APPLIED PROCESS DESIGN

FOR CHEMICAL AND PETROCHEMICAL PLANTS

Volume 3, Third Edition

Contains process design and equipment details for heat transfer, refrigeration systems, compression equipment, and mechanical drivers

Ernest E. Ludwig

APPLIED PROCESS DESIGN

FOR CHEMICAL AND PETROCHEMICAL PLANTS

Volume 3, Third Edition

Volume 1:

1. Process Planning, Scheduling, Flowsheet Design
2. Fluid Flow
3. Pumping of Liquids
4. Mechanical Separations
5. Mixing of Liquids
6. Ejectors
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8. Distillation
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10. Heat Transfer
11. Refrigeration Systems
12. Compression Equipment (Including Fans)
13. Reciprocating Compression Surge Drums
14. Mechanical Drivers

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Volume 3, Third Edition

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To my wife, Sue, for her patient encouragement and help

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Foreword to the Second Edition

The techniques of process design continue to improve as the science of chemical engineering develops new and better interpretations of fundamentals. Accordingly, this second edition presents additional, reliable design methods based on proven techniques and supported by pertinent data. Since the first edition, much progress has been made in standardizing and improving the design techniques for the hardware components that are used in designing process equipment. This standardization has been incorporated in this latest edition, as much as practically possible.

The "heart" of proper process design is interpreting the process requirements into properly *arranged* and *sized* mechanical hardware expressed as (1) off-the-shelf mechanical equipment (with appropriate electric drives and instrumentation for control); (2) custom-designed vessels, controls, etc.; or (3) some combination of (1) and (2). The unique process conditions must be attainable in, by, and through the equipment. Therefore, it is essential that the process designer carefully visualize physically and mathematically just how the process will behave in the equipment and through the control schemes proposed.

Although most of the chapters have been expanded to include new material, some obsolete information has been removed.

Chapter 10, "Heat Transfer," has been updated and now includes several important design techniques for difficult condensing situations and for the application of thermosiphon reboilers.

Chapter 11, "Refrigeration Systems," has been improved with additional data and new systems designs for light hydrocarbon refrigeration.

Chapter 12, "Compression Equipment," has been generally updated.

Chapter 13, "Compression Surge Drums," presents several new techniques, as well as additional detailed examples.

Chapter 14, "Mechanical Drivers," has been updated to inlcude the latest code and standards of the National Electrical Manufacturer's Association and information on the new energy efficient motors.

Also, the new appendix provides an array of basic reference and conversion data.

Although computers are now an increasingly valuable tool for the process design engineer, it is beyond the scope of these three volumes to incorporate the programming and mathematical techniques required to convert the basic process design methods presented into computer programs. Many useful computer programs now exist for process design, as well as optimization, and the process designer is encouraged to develop his/her own or to become familiar with available commercial programs through several of the recognized firms specializing in design and simulation computer software.

The many aspects of process design are essential to the proper performance of the work of chemical engineers and other engineers engaged in the process engineering design details for chemical and petrochemical plants. Process design has developed by necessity into a unique section of the scope of work for the broad spectrum of chemical engineering.

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Preface to the Third Edition

This volume of *Applied Process Design* is intended to be a chemical engineering process design manual of methods and proven fundamentals with supplemental mechanical and related data and charts (some in the expanded appendix). It will assist the engineer in examining and analyzing a problem and finding a design method and mechanical specifications to secure the proper mechanical hardware to accomplish a particular process objective. An expanded chapter on safety requirements for chemical plants and equipment design and application stresses the applicable codes, design methods, and the sources of important new data.

This manual is not intended to be a handbook filled with equations and various data with no explanation of application. Rather, it is a guide for the engineer in applying chemical processes to the properly detailed hardware (equipment), because without properly sized and internally detailed hardware, the process very likely will not accomplish its unique objective. This book does not develop or derive theoretical equations; instead, it provides direct application of sound theory to applied equations useful in the immediate design effort. Most of the recommended equations have been used in actual plant equipment design and are considered to be some of the most reasonable available (excluding proprietary data and design methods), which can be handled by both the inexperienced as well as the experienced engineer. A conscious effort has been made to offer guidelines of judgment, decisions, and selections, and some of this will also be found in the illustrative problems. My experience has shown that this approach at presentation of design information serves well for troubleshooting plant operation problems and equipment/systems performance analysis. This book also can serve as a classroom text for senior and graduate level chemical plant design courses at the university level.

The text material assumes that the reader is an undergraduate engineer with one or two years of engineering fundamentals or a graduate engineer with a sound knowledge of the fundamentals of the profession. This book will provide the reader with design techniques to actually design as well as mechanically detail and specify. It is the author's philosophy that the process engineer has not adequately performed his or her function unless the results of a process calculation for equipment are specified in terms of something that can be economically built or selected from the special designs of manufacturers and can by visual or mental techniques be *mechanically* interpreted to actually perform the process function for which it was designed. Considerable emphasis in this book is placed on the mechanical Codes and some of the requirements that can be so important in the specifications as well as the actual specific design details. Many of the mechanical and metallurgical specifics that are important to good design practice are not usually found in standard mechanical engineering texts.

The chapters are developed by *design function* and not in accordance with previously suggested standards for unit operations. In fact, some of the chapters use the same principles, but require different interpretations that take into account the *process* and the *function* the equipment performs in the process.

Because of the magnitude of the task of preparing the material for this new edition in proper detail, it has been necessary to omit several important topics that were covered in the previous edition. Topics such as corrosion and metallurgy, cost estimating, and economics are now left to the more specialized works of several fine authors. The topic of static electricity, however, is treated in the chapter on process safety, and the topic of mechanical drivers, which includes electric motors, is covered in a separate chapter because many specific items of process equipment require some type of electrical or mechanical driver. Even though some topics cannot be covered here, the author hopes that the designer will find design techniques adaptable to 75 percent to 85+ percent of required applications and problems.

The techniques of applied chemical plant process design continue to improve as the science of chemical engineering develops new and better interpretations of the fundamentals for chemistry, physics, metallurgical, mechanical, and polymer/plastic sciences. Accordingly, this third edition presents additional reliable design methods based on proven techniques developed by individuals and groups considered competent in their subjects and who are supported by pertinent data. Since the first and second editions, much progress has been made in standardizing (which implies a certain amount of improvement) the hardware components that are used in designing process equipment. Much of the important and basic standardization has been incorporated in this latest edition. Every chapter has been expanded and updated with new material.

All of the chapters have been carefully reviewed and older (not necessarily obsolete) material removed and replaced by newer design techniques. It is important to appreciate that not all of the material has been replaced because much of the so-called "older" material is still the best there is today, and still yields good designs. Additional charts and tables have been included to aid in the design methods or explaining the design techniques.

The author is indebted to the many industrial firms that have so generously made available certain valuable design data and information. Thus, credit is acknowledged at the appropriate locations in the text, except for the few cases where a specific request was made to omit this credit.

The author was encouraged to undertake this work by Dr. James Villbrandt and the late Dr. W. A. Cunningham and Dr. John J. McKetta. The latter two as well as the late Dr. K. A. Kobe offered many suggestions to help establish the usefulness of the material to the broadest group of engineers and as a teaching text.

In addition, the author is deeply appreciative of the courtesy of the Dow Chemical Co. for the use of certain non-credited materials and their release for publication. In this regard, particular thanks is given to the late N. D. Griswold and Mr. J. E. Ross. The valuable contribution of associates in checking material and making suggestions is gratefully acknowledged to H. F. Hasenbeck, L. T. McBeth, E. R. Ketchum, J. D. Hajek, W. J. Evers, and D. A. Gibson. The courtesy of the Rexall Chemical Co. to encourage completion of the work is also gratefully appreciated.

Ernest E. Ludwig. P.E.

Chapter

10

Heat Transfer

Heat transfer is perhaps the most important, as well as the most applied process, in chemical and petrochemical plants. Economics of plant operation often are controlled by the effectiveness of the use and recovery of heat or cold (refrigeration). The service functions of steam, power, refrigeration supply, and the like are dictated by how these services or utilities are used within the process to produce an efficient conversion and recovery of heat.

Although many good references (5, 22, 36, 37, 40, 61, 70, 74, 82) are available, and the technical literature is well represented by important details of good heat transfer design principles and good approaches to equipment design, an unknown factor that enters into every design still remains. This factor is the scale or fouling from the fluids being processed and is wholly dependent on the fluids, their temperature and velocity, and to a certain extent the nature of the heat transfer tube surface and its chemical composition. Due to the unknown nature of the assumptions, these fouling factors can markedly affect the design of heat transfer equipment. Keep this in mind as this chapter develops. Conventional practice is presented here; however, Kern[71] has proposed new thermal concepts that may offer new approaches.

Before presenting design details, we will review a summary of the usual equipment found in process plants.

The design of the heat transfer process and the associated design of the appropriate hardware is now almost always being performed by computer programs specifically developed for particular types of heat transfer. This text does not attempt to develop computer programs, although a few examples are illustrated for specific applications. The important reason behind this approach is that unless the design engineer working with the process has a "feel" for the expected results from a computer program or can assess whether the results calculated are proper, adequate, or "in the right ball park," a plant design may result in improperly selected equipment sizing. *Unless the user-designer has some knowledge of what a specific computer program can accomplish, on what specific heat transfer equations and concepts the program is based, or which of these concepts have been incorporated into the program, the user-designer can be "flying blind" regarding the results,* not knowing whether they are proper for the particular conditions required. Therefore, one of the intended values of this text is to provide the designer with a basis for manually checking the expected equations, coefficients, etc., which will enable the designer to accept the computer results. In addition, the text provides a basis for completely designing the process heat transfer equipment (except specialized items such as fired heaters, steam boiler/generators, cryogenic equipment, and some other process requirements) and sizing (for mechanical dimensions/details, but not for pressure strength) the mechanical hardware that will accomplish this function.

Types of Heat Transfer Equipment Terminology

The process engineer needs to understand the terminology of the heat transfer equipment manufacturers in order to properly design, specify, evaluate bids, and check drawings for this equipment.

The standards of the Tubular Exchanger Manufacturers Association (TEMA)[107] is the only assembly of unfired mechanical standards including selected design details and *Recommended Good Practice* and is used by all reputable exchanger manufacturers in the U.S. and many manufacturers in foreign countries who bid on supplying U.S. plant equipment. These standards are developed, assembled, and updated by a technical committee of association members. The standards are updated and reissued every 10 years. These standards do not designate or recommend thermal design methods or practices for specific process applications but do outline basic heat transfer fundamentals and list suggested fouling factors for a wide variety of fluid or process services.

The three classes of mechanical standards in TEMA are Classes R, C, and B representing varying degrees of mechanical details for the designated process plant applications' severity. The code designations [TEMA−1988 Ed] for mechanical design and fabrication are:

RCB—Includes all classes of construction/design and are identical; shell diameter (inside) not exceeding 60 in., and maximum design pressure of 3,000 psi.

R—Designates severe *requirements* of petroleum and other related processing applications.

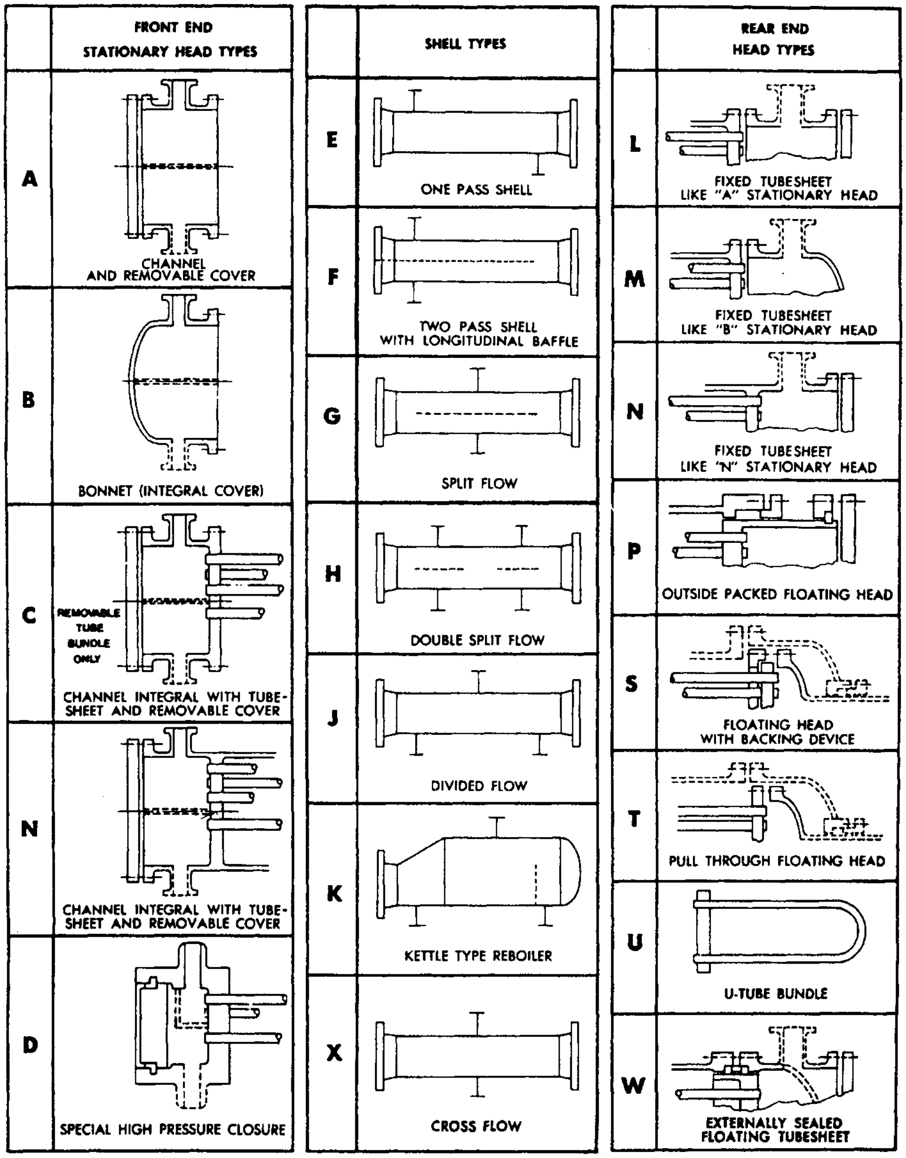


Figure 10-1A. Nomenclature for Heat Exchanger Components. Figures 10-1A–G used by permission: *Standards of Tubular Exchanger Manufacturers Association*, 7th Ed., Fig. N-1.2, © 1988. Tubular Exchanger Manufacturers Association, Inc.

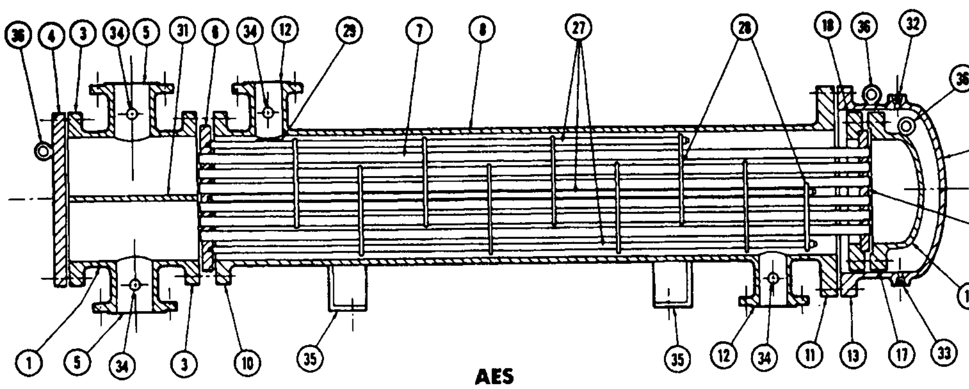


Figure 10-1B. Floating head. (© 1988 by Tubular Exchanger Manufacturers Association, Inc.)

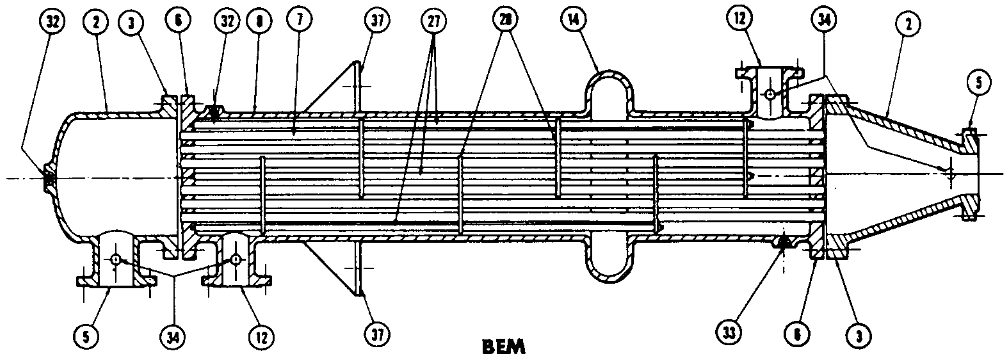


Figure 10-1C. Fixed tubesheet. (© 1988 by Tubular Exchanger Manufacturers Association, Inc.)

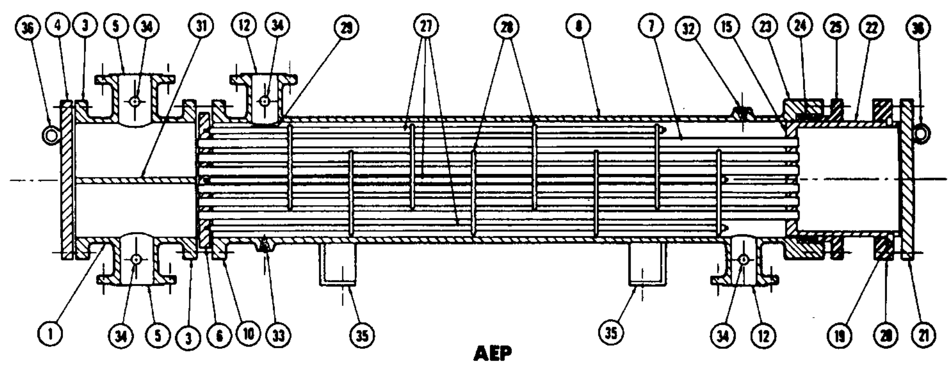


Figure 10-1D. Floating head—outside packed. (© 1988 by Tubular Exchanger Manufacturers Association, Inc.)

C—Indicates generally moderate requirements of *commercial* and general process applications.

B—Specifies design and fabrication for chemical process service.

RGP—*Recommended Good Practice,* includes topics outside the scope of the basic standards.

Note: The petroleum, petrochemical, chemical, and other industrial plants must specify or select the design/fabrication code designation for their individual application as the standards do not dictate the code designation to use. Many chemical plants select the most severe designation of Class R rather than Class B primarily because they prefer a more rugged or husky piece of equipment.

In accordance with the TEMA Standards, the individual vessels must comply with the American Society of Mechanical Engineers (ASME) Boiler and Pressure Vessel Code, Section VIII, Div. 1, plus process or petroleum plant location state and area codes. The ASME Code Stamp is required by the TEMA Standards.

Figures 10-1A–G and Table 10-1 from the Standards of Tubular Exchanger Manufacturers Association[107] give the nomenclature of the basic types of units. Note the nomenclature type designation code letters immediately below each illustration. These codes are assembled from Table 10-1 and Figures 10-1A–G.

Many exchangers can be designed without all parts; specifically the performance design may not require (a) a floating head and its associated parts, or (b) an impingement baffle but may require a longitudinal shell side baffle (see Figures 10-1F and 10-1G). It is important to recognize that the components in Figures 10-1B–K are associated with the basic terminology regardless of type of unit. An application and selection guide is shown in Table 10-2 and Figures 10-2 and 10-3.

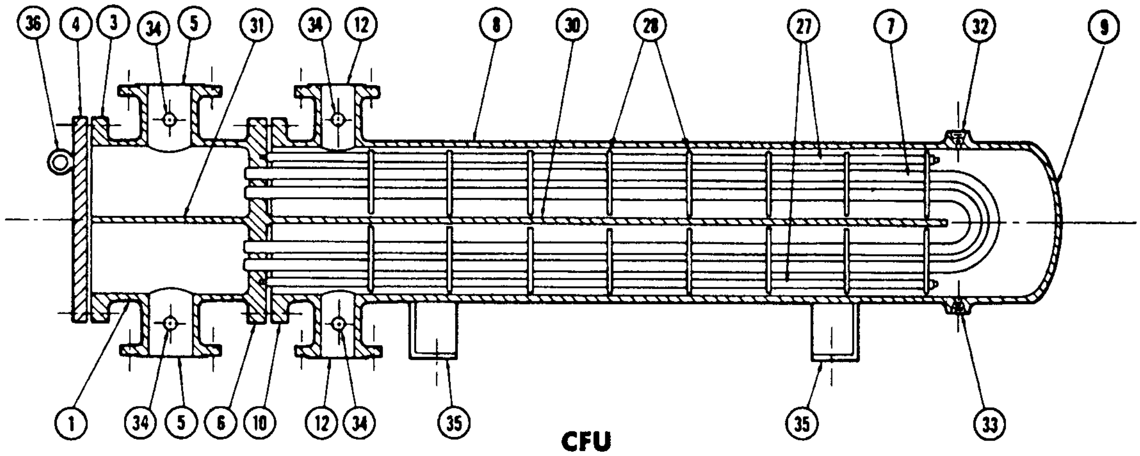


Figure 10-1E. Removable U-bundle. (© 1988 by Tubular Exchanger Manufacturers Association, Inc.)

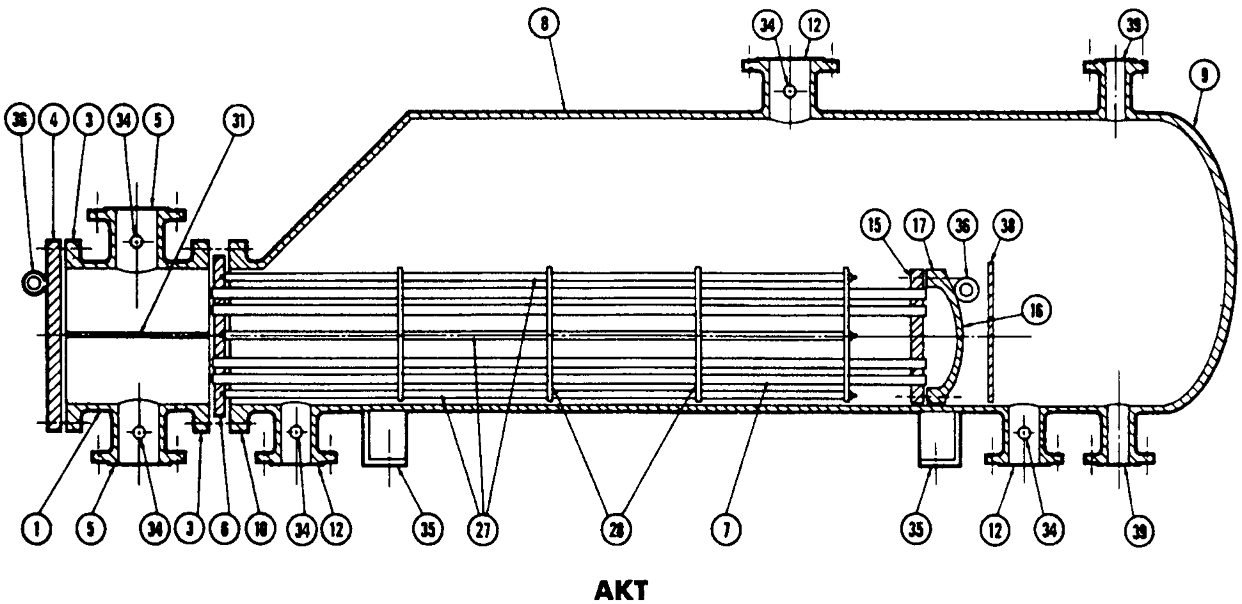


Figure 10-1F. Kettle reboiler. (© 1988 by Tubular Exchanger Manufacturers Association, Inc.)

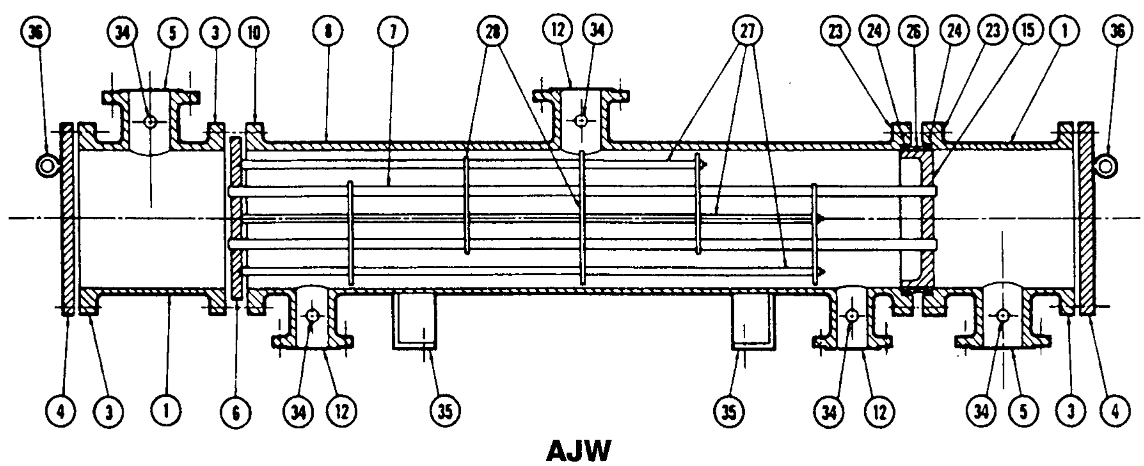


Figure 10-1G. Divided flow—packed tubesheet. (© 1988 by Tubular Exchanger Manufacturers Association, Inc.)

Table 10-1
Standard TEMA Heat Exchanger Terminology/Nomenclature*

1. Stationary Head—Channel	21. Floating Head Cover—External
2. Stationary Head—Bonnet	22. Floating Tubesheet Skirt
3. Stationary Head Flange—Channel or Bonnet	23. Packing Box
4. Channel Cover	24. Packing
5. Stationary Head Nozzle	25. Packing Gland
6. Stationary Tubesheet	26. Lantern Ring
7. Tubes	27. Tierods and Spacers
8. Shell	28. Transverse Baffles or Support Plates
9. Shell Cover	29. Impingement Plate
10. Shell Flange—Stationary Head End	30. Longitudinal Baffle
11. Shell Flange—Rear Head End	31. Pass Partition
12. Shell Nozzle	32. Vent Connection
13. Shell Cover Flange	33. Drain Connection
14. Expansion Joint	34. Instrument Connection
15. Floating Tubesheet	35. Support Saddle
16. Floating Head Cover	36. Lifting Lug
17. Floating Head Cover Flange	37. Support Bracket
18. Floating Head Backing Device	38. Weir
19. Split Shear Ring	39. Liquid Level Connection
20. Slip-on Backing Flange	

*Key to Figures 10-1B–G. See Figure 10-1A for Nomenclature Code.

Used by permission: *Standards of Tubular Exchanger Manufacturers Association,* 7th Ed., Table N-2, © 1988. Tubular Exchanger Manufacturers Association, Inc. All rights reserved.

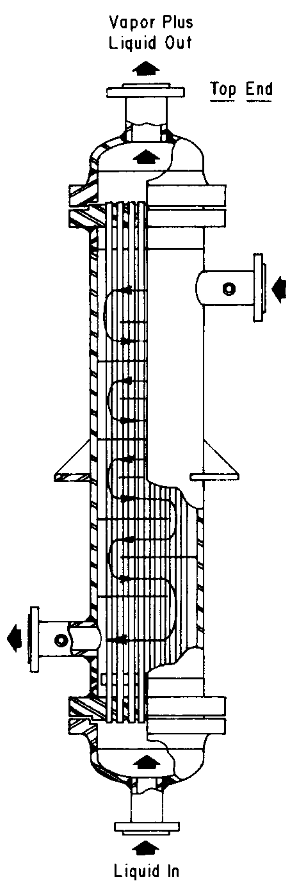


Figure 10-1H. Fixed tubesheet, single-tube pass vertical heater or reboiler. (Used by permission: Engineers & Fabricators, Inc., Houston.)

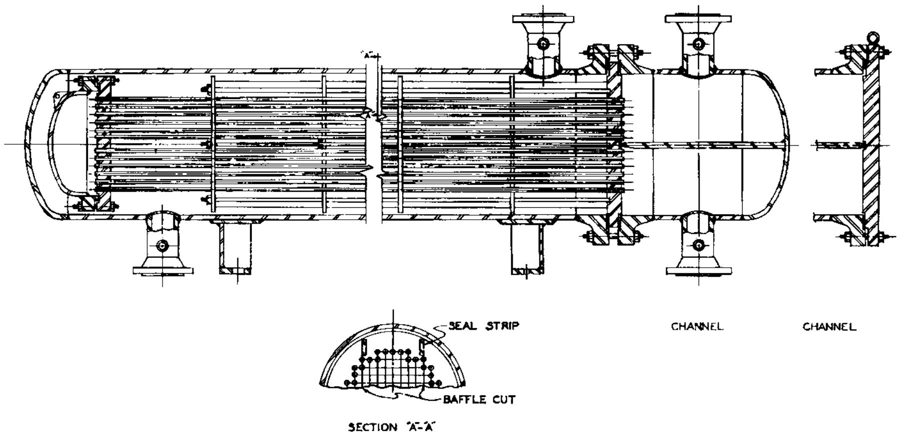


Figure 10-1I. Floating head, removable type. (Used by permission: Yuba Heat Transfer Division of Connell Limited Partnership.)

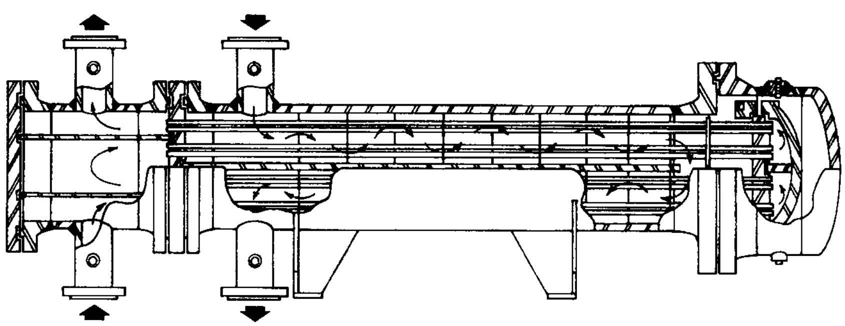


Figure 10-1J. Split-ring removable floating head, four-pass tube-side and two-pass shell-side. (Used by permission: Engineers & Fabricators, Inc., Houston.)

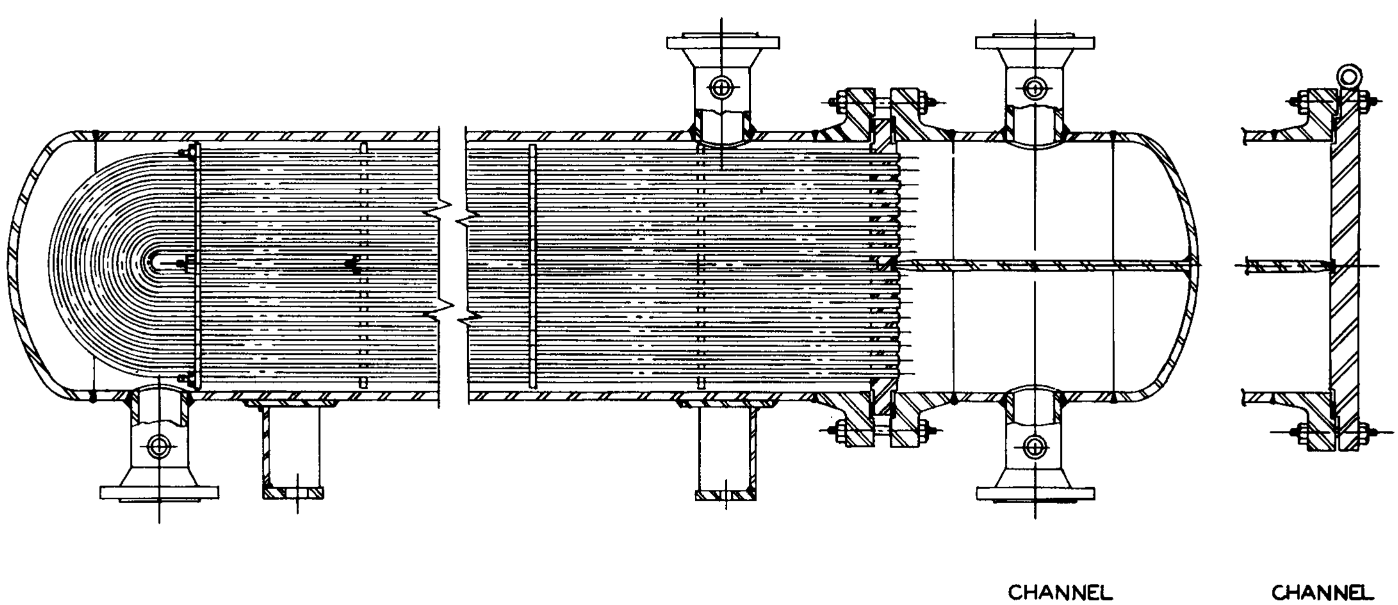


Figure 10-1K. U-tube exchanger. (Used by permission: Yuba Heat Transfer Division of Connell Limited Partnership.)

Table 10-2
Selection Guide Heat Exchanger Types

Type Designation	Figure No.	Significant Feature	Applications Best Suited	Limitations	Approximate Relative Cost in Carbon Steel Construction
Fixed TubeSheet	10–1C 10–1H	Both tubesheets fixed to shell.	Condensers; liquid-liquid; gas-gas; gas-liquid; cooling and heating, horizontal or vertical, reboiling.	Temperature difference at extremes of about 200°F due to differential expansion.	1.0
Floating Head or Tubesheet (removable and nonremovable bundles)	10–1B 10–1D 10–1G 10–1I 10–1J	One tubesheet "floats" in shell or with shell, tube bundle may or may not be removable from shell, but back cover can be removed to expose tube ends.	High temperature differentials, above about 200°F extremes; dirty fluids requiring cleaning of inside as well as outside of shell, horizontal or vertical.	Internal gaskets offer danger of leaking. Corrosiveness of fluids on shell-side floating parts. Usually confined to horizontal units.	1.28
U-Tube; U-Bundle	10–1E 10–1K	Only one tubesheet required. Tubes bent in U-shape. Bundle is removable.	High temperature differentials, which might require provision for expansion in fixed tube units. Clean service or easily cleaned conditions on both tube side and shell side. Horizontal or vertical.	Bends must be carefully made, or mechanical damage and danger of rupture can result. Tube side velocities can cause erosion of inside of bends. Fluid should be free of suspended particles.	0.9–1.1
Kettle	10–1F	Tube bundle removable as U-type or floating head. Shell enlarged to allow boiling and vapor disengaging.	Boiling fluid on shell side, as refrigerant, or process fluid being vaporized. Chilling or cooling of tube-side fluid in refrigerant evaporation on shell side.	For horizontal installation. Physically large for other applications.	1.2–1.4
Double Pipe	10–4A 10–4B 10–4C 10–4D	Each tube has own shell forming annular space for shell-side fluid. Usually use externally finned tube.	Relatively small transfer area service, or in banks for larger applications. Especially suited for high pressures in tube (greater than 400 psig).	Services suitable for finned tube. Piping-up a large number often requires cost and space.	0.8–1.4
Pipe Coil	10–5A 10–5B	Pipe coil for submersion in coil-box of water or sprayed with water is simplest type of exchanger.	Condensing, or relatively low heat loads on sensible transfer.	Transfer coefficient is low, requires relatively large space if heat load is high.	0.5–0.7
Open Tube Sections (water cooled)	10–5A 10–5B	Tubes require no shell, only end headers, usually long, water sprays over surface, sheds scales on outside tubes by expansion and contraction. Can also be used in water box.	Condensing, relatively low heat loads on sensible transfer.	Transfer coefficient is low, takes up less space than pipe coil.	0.8–1.1
Open Tube Sections (air cooled); Plain or Finned Tubes	10–6	No shell required, only end headers similar to water units.	Condensing, high-level heat transfer.	Transfer coefficient is low, if natural convection circulation, but is improved with forced air flow across tubes.	0.8–1.8
Plate and Frame	10–7A 10–7B 10–7C	Composed of metal-formed thin plates separated by gaskets. Compact, easy to clean.	Viscous fluids, corrosive fluids slurries, high heat transfer.	Not well suited for boiling or condensing; limit 350–500°F by gaskets. Used for liquid-liquid only; not gas-gas.	0.8–1.5
Small-tube Teflon	10–8	Chemical resistance of tubes; no tube fouling.	Clean fluids, condensing, cross-exchange.	Low heat transfer coefficient.	2.0–4.0
Spiral	10–9A 10–9B 10–9C 10–9D	Compact, concentric plates; no bypassing, high turbulence.	Cross-flow, condensing, heating.	Process corrosion, suspended materials.	0.8–1.5

Details of Exchange Equipment Assembly and Arrangement

The process design of heat exchange equipment depends to a certain extent upon the basic type of unit considered for the process and how it will be arranged together with certain details of assembly as they pertain to that particular unit. It is important to recognize that certain basic types of exchangers, as given in Table 10-2, are less expensive than others and also that inherently these problems are related to the fabrication of construction materials to resist the fluids, cleaning, future reassignment to other services, etc. The following presentation alerts the designer to the various features that should be considered. Also see Rubin.[281]

1. Construction Codes

The American Society of Mechanical Engineers (ASME) *Unfired Pressure Vessel Code*[119] is accepted by almost all states as a requirement by law and by most industrial insurance underwriters as a basic guide or requirement for fabrication of pressure vessel equipment, which includes some components of heat exchangers.

This code does not cover the rolling-in of tubes into tubesheets.

For steam generation or any equipment having a direct fire as the means of heating, the ASME *Boiler Code*[6] applies, and many states and insurance companies require compliance with this.

These classes are explained in the TEMA Standards and in Rubin.[99, 100, 133]

2. Thermal Rating Standards

The TEMA Code[107] does not recommend thermal design or rating of heat exchangers. This is left to the rating or design engineer, because many unique details are associated with individual applications. TEMA does offer some common practice rating charts and tables, along with some tabulations of selected petroleum and chemical physical property data in the third (1952) and sixth (1978) editions.

3. Exchanger Shell Types

The type of shell of an exchanger should often be established before thermal rating of the unit takes place. The shell is always a function of its relationship to the tubesheet and the internal baffles. Figures 10-1, 10-2, and 10-3 summarize the usual types of shells; however, remember that other arrangements may satisfy a particular situation.

The heads attached to the shells may be welded or bolted as shown in Figure 10-3. Many other arrangements may be found in references 37, 38, and 61.

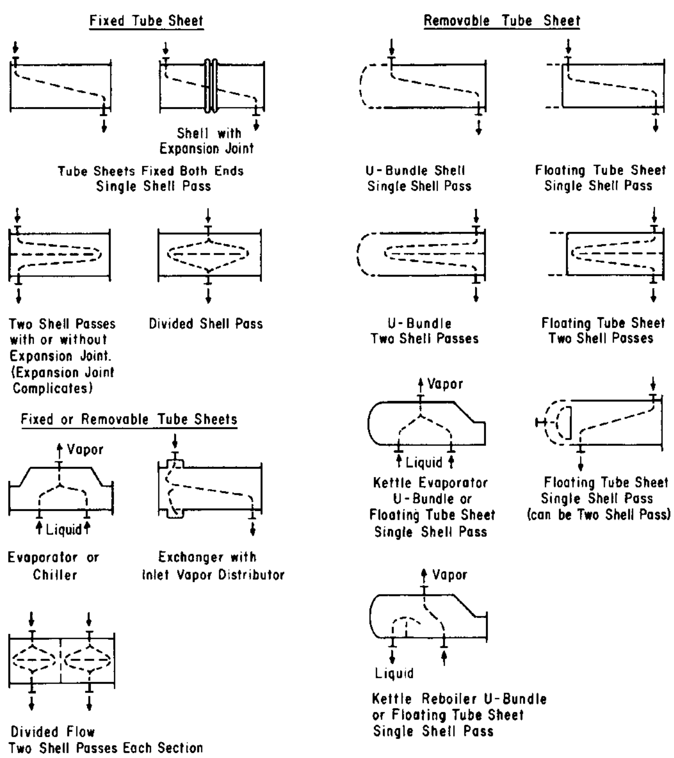


Figure 10-2. Typical shell types.

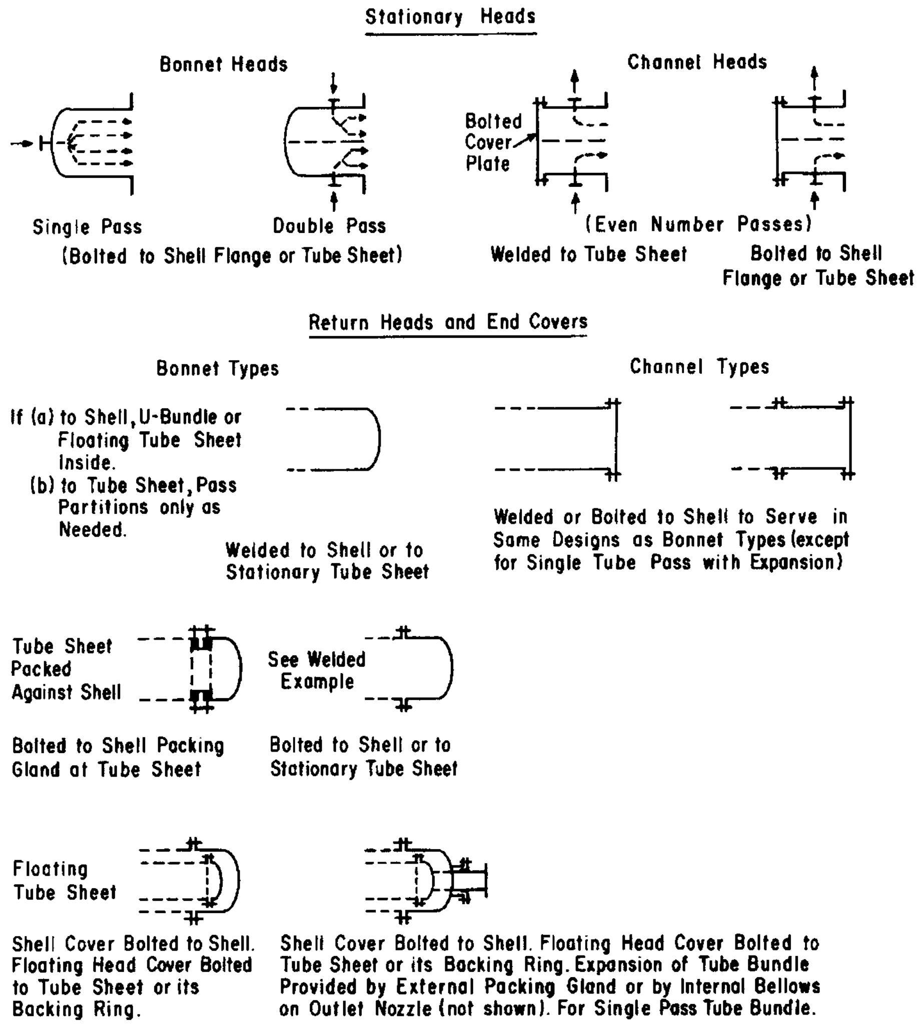


Figure 10-3. Typical heads and closures.

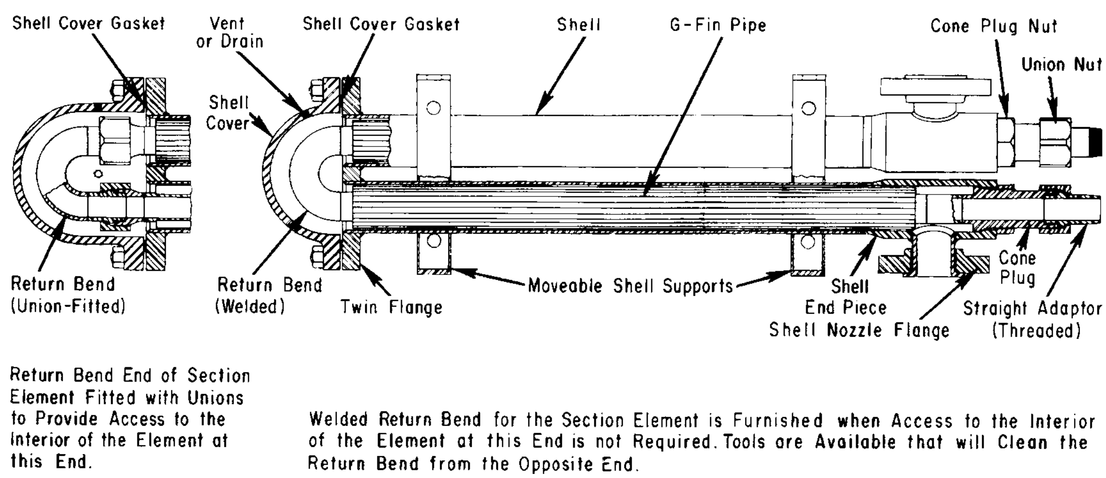


Figure 10-4A(1). Double-pipe longitudinal Twin G-Finned exchanger. (Used by permission: Griscom-Russell Co./Ecolaire Corp., Easton, PA, Bul. 7600.)

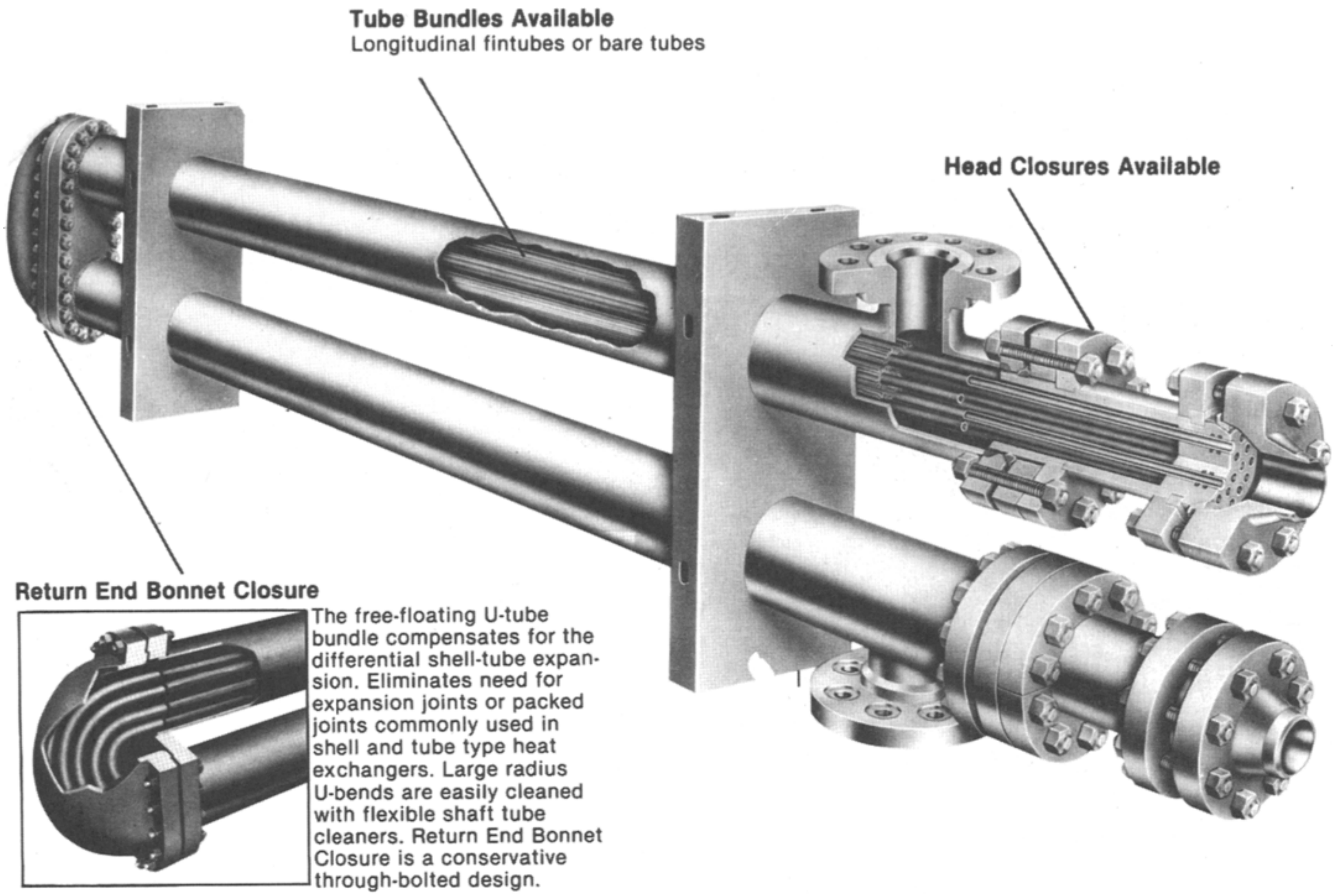


Figure 10-4A(2). Multitube hairpin fintube heat exchangers. The individual shell modules can be arranged into several configurations to suit the process parallel and/or series flow arrangements. The shell size range is 3–16 in. (Used by permission: Brown Fintube Co., A Koch® Engineering Co., Bul. B–30–1.)

4. Tubes

The two basic types of tubes are (a) plain or bare and (b) finned—external or internal, see Figures 10-4A–E, 10-10, and 10-11. The plain tube is used in the usual heat exchange application. However, the advantages of the more common externally finned tube are becoming better identified. These tubes are performing exceptionally well in applications in which their best features can be used.

Plain tubes (either as solid wall or duplex) are available in carbon steel, carbon alloy steels, stainless steels, copper, brass and alloys, cupro-nickel, nickel, monel, tantalum, carbon, glass, and other special materials. Usually there is no great problem in selecting an available tube material. However, when its assembly into the tubesheet along with the resulting fabrication problems are considered, the selection of the tube alone is only part of a coordinated design. Plain-tube mechanical data and dimensions are given in Tables 10-3 and 10-4.

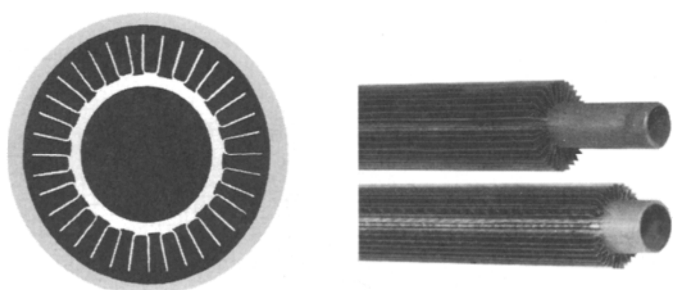


Figure 10-4A(3). Longitudinal fins resistance welded to tubes. The welding of the fins integral to the parent tube ensures continuous high heat transfer efficiency and the absence of any stress concentrations within the tube wall. (Used by permission: Brown Fintube Co., A Koch® Engineering Co., Bul. 80–1.)

The duplex tube (Figure 10-11) is a tube within a tube, snugly fitted by drawing the outer tube onto the inner or by other mechanical procedures.

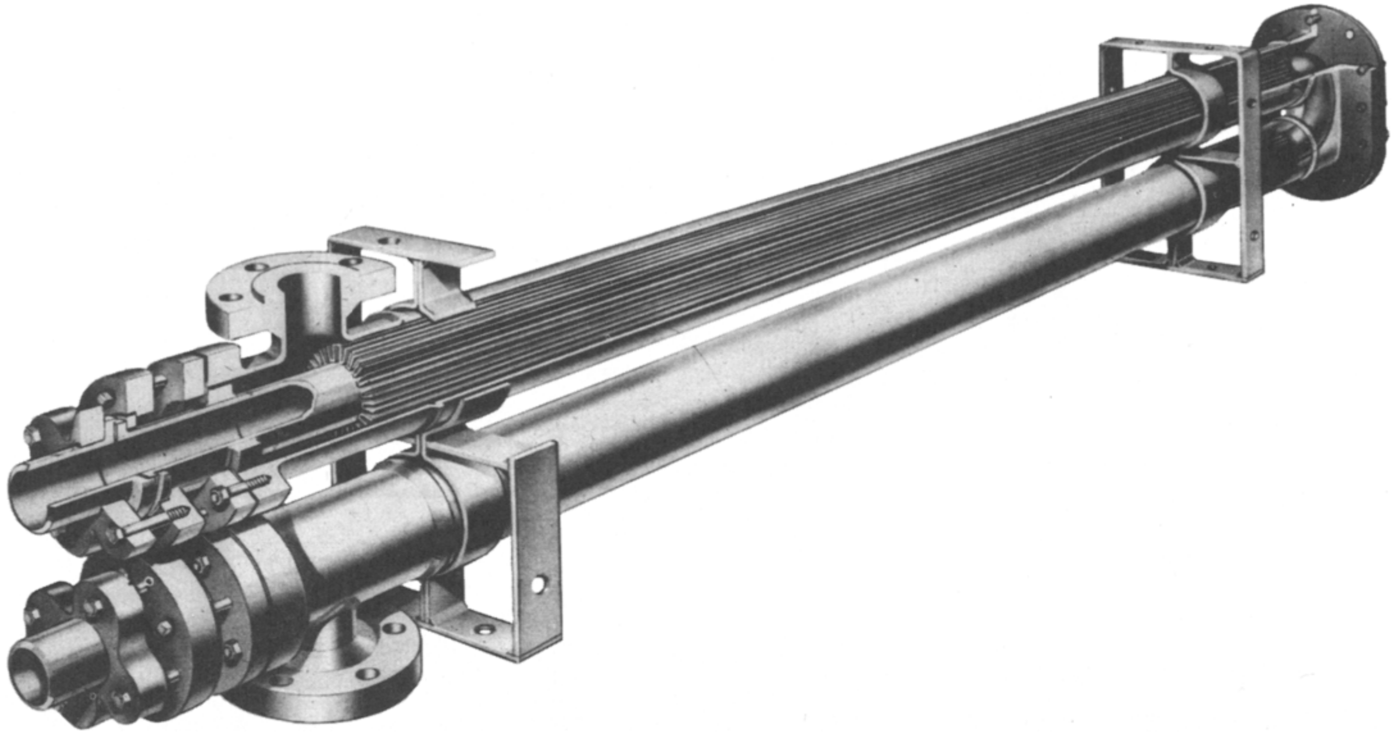
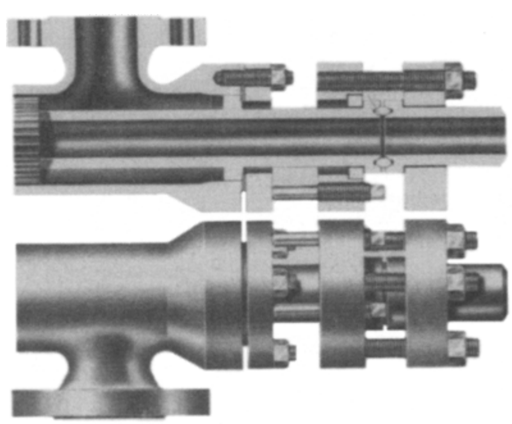
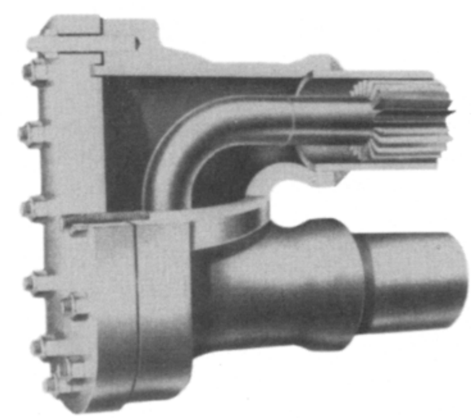


Figure 10-4B. Cutaway view of finned double-pipe exchanger. (Used by permission: ALCO Products Co., Div. of NITRAM Energy, Inc.)



FIXED-END CLOSURE



RETURN END CLOSURE

Figure 10-4C. High-pressure fixed-end closure and return-end closure. (Used by permission: ALCO Products Co., Div. of NITRAM Energy, Inc.)

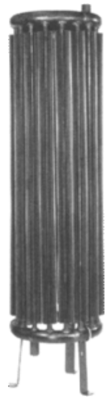


Figure 10-4D. Vertical longitudinal finned-tube tank heater, which is used in multiple assemblies when required. (Used by permission: Brown Fintube Co., A Koch® Engineering Co., Bul. 4–5.)

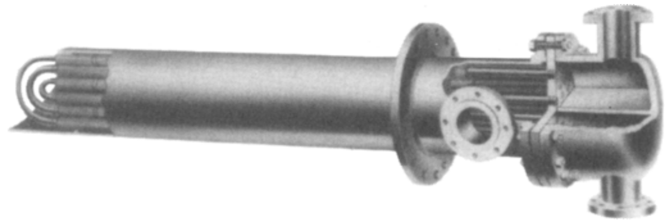


Figure 10-4E. Longitudinal finned-tube tank suction direct line heater. (Used by permission: Brown Fintube Co., A Koch® Engineering Co., Bul. 4–5.)

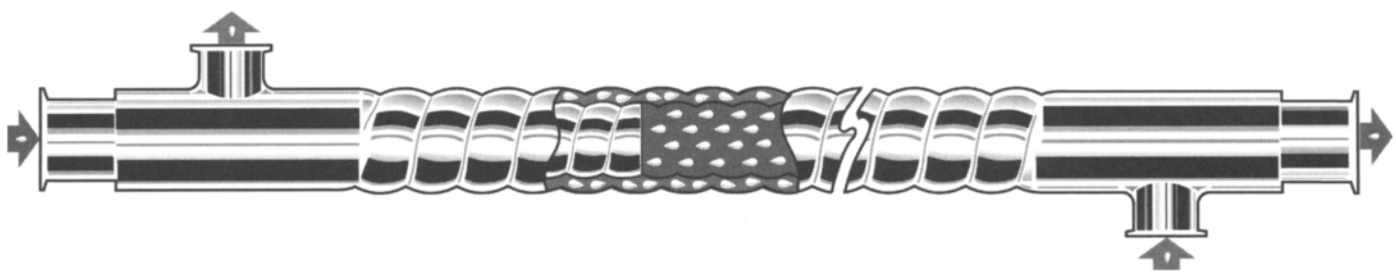


Figure 10-4F(1). Single concentric corrugated tube in single corrugated shell. (Used by permission: APV Heat Transfer Technologies.)

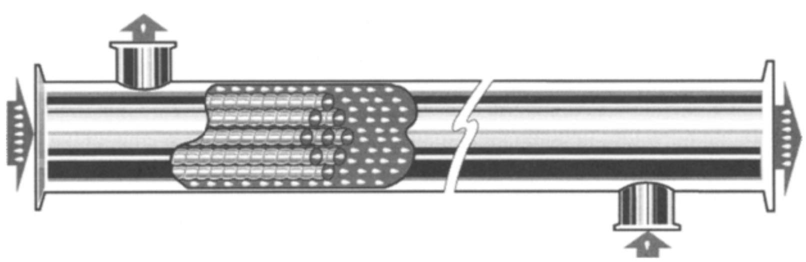


Figure 10-4F(2). Multicorrugated tubes in single shell. (Used by permission: APV Heat Transfer Technologies.)

This tube is useful when the shell-side fluid is not compatible with the material needed for the tube-side fluid, or vice versa. The thicknesses of the two different wall materials do not have to be the same. As a general rule, 18 ga is about as thin as either tube should be, although thinner gages are available. In establishing the gage thickness for each component of the tube, the corrosion rate of the material should be about equal for the inside and outside, and the wall thickness should still withstand the pressure and temperature conditions after a reasonable service life.

More than 100 material combinations exist for these tubes. A few materials suitable for the inside or outside of the tube include copper, steel, cupro-nickel, aluminum, lead, monel, nickel, stainless steel, alloy steels, various brasses, etc. From these combinations most process conditions can be satisfied. Combinations such as steel outside and admiralty or cupro-nickel inside are used in ammonia condensers cooled with water in the tubes. Tubes of steel outside and cupro-nickel inside are used in many process condensers using sea water. These tubes can be bent for U-bundles without loss of effective heat transfer. However, care must be used, such as by bending sand-filled or on a mandrel. The usual minimum radius of the bend for copper-alloy–steel type duplex tube is three times the O.D. of the tube. Sharper bends can be made by localized heating; however, the tube should be specified at the time of purchase for these conditions.

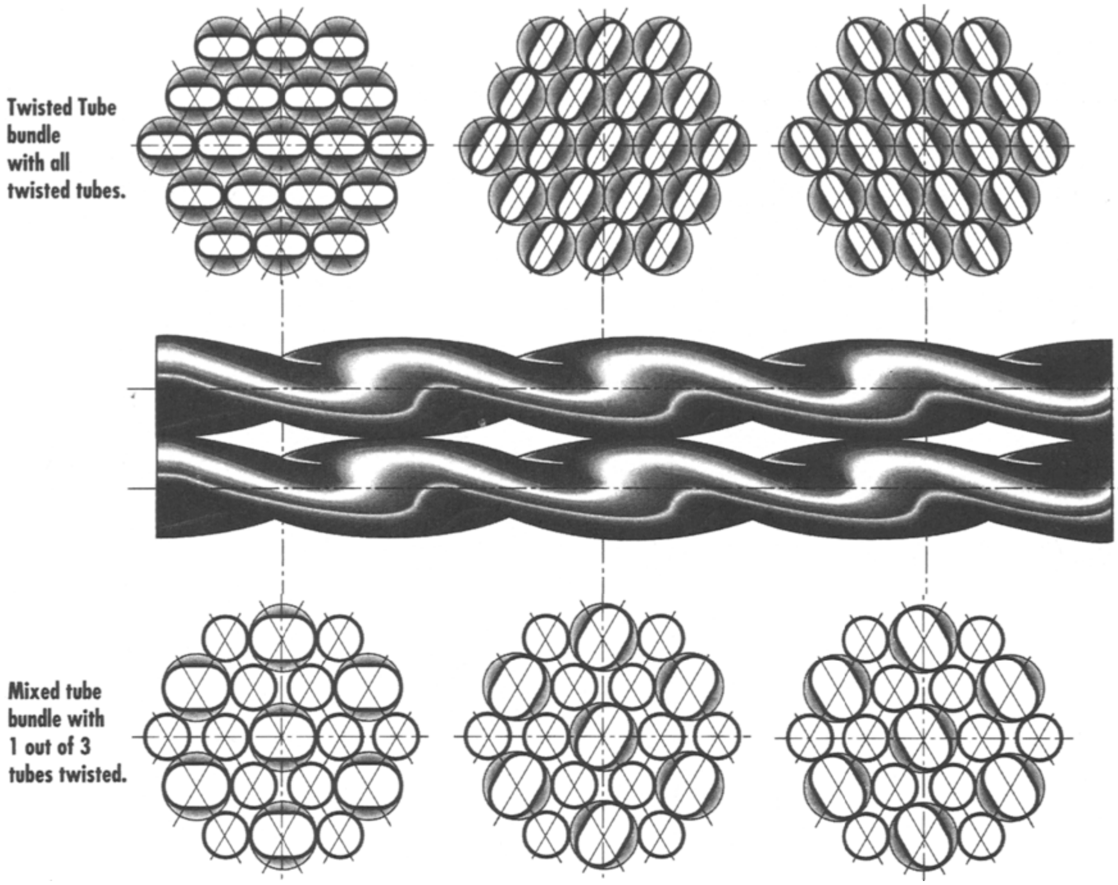


Figure 10-4G. Twisted tubes with heat exchanger bundle arrangements. (Used by permission: Brown Fintube Co., A Koch® Engineering Co., Bul. B-100-2.)

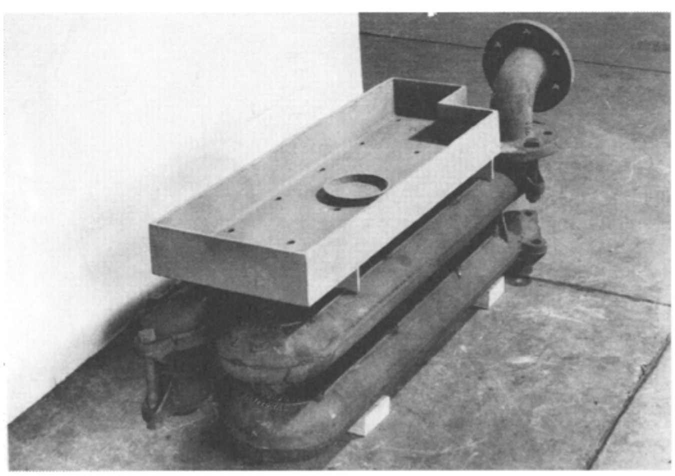


Figure 10-5A. Cast iron sections; open coil cooler-coil and distribution pan.

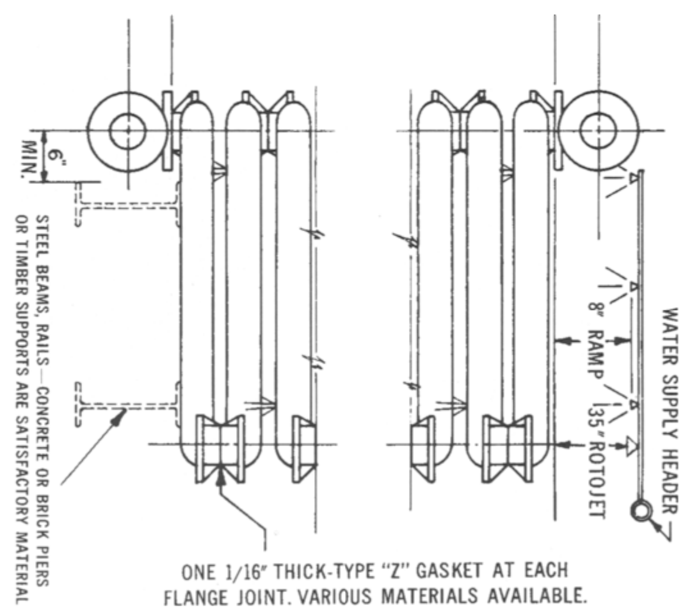


Figure 10-5B. Elevation assembly—cast iron cooler sections.

Figure 10-6. Open tube sections. (Used by permission: Griscom-Russell Co./Ecolaire Corp., Easton, PA.)

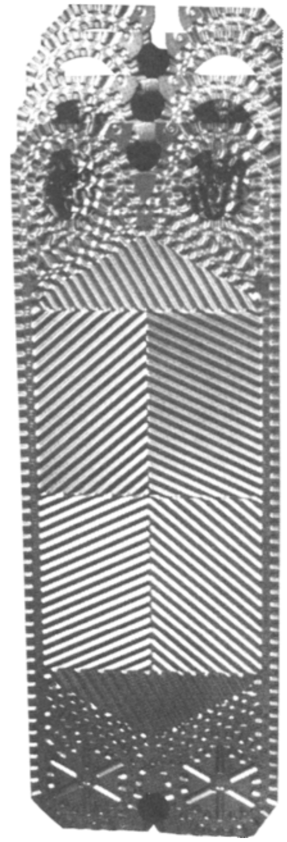


Figure 10-7A. Typical one side of Plate for Plate and Frame Exchanger. (Used by permission: Graham Manufacturing Company, Inc., Bul. PHE 96–1.)

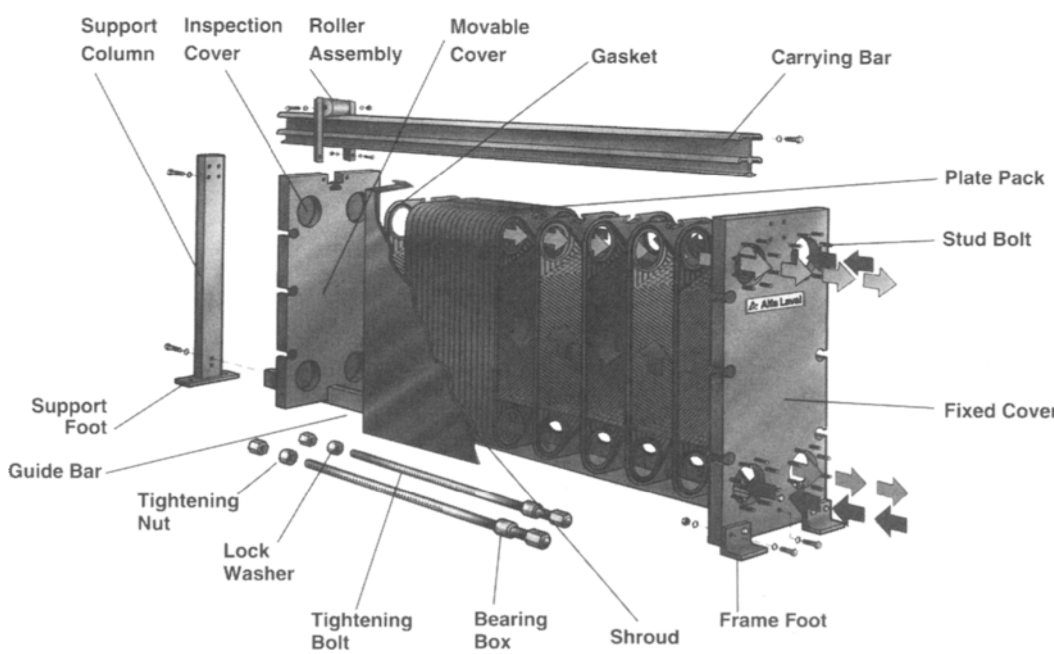


Figure 10-7. "Plate and Frame" heat exchanger basic components. (Used by permission: Alfa Laval Thermal, Inc., Bul. G101)

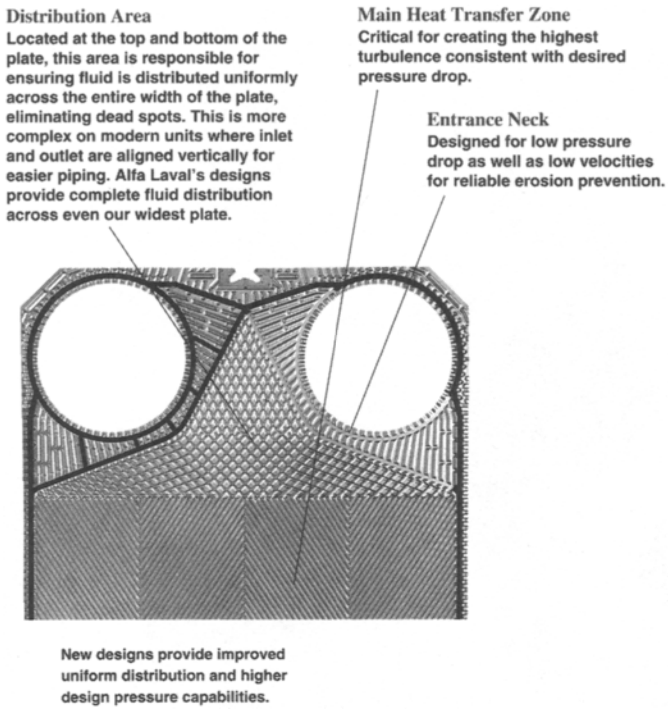


Figure 10-7B. Typical flow patterns of fluid flow across one side of plate. The opposing fluid is on the reverse side flowing in the opposite direction. (Used by permission: Alfa Laval Thermal Inc, Bul. G–101.)

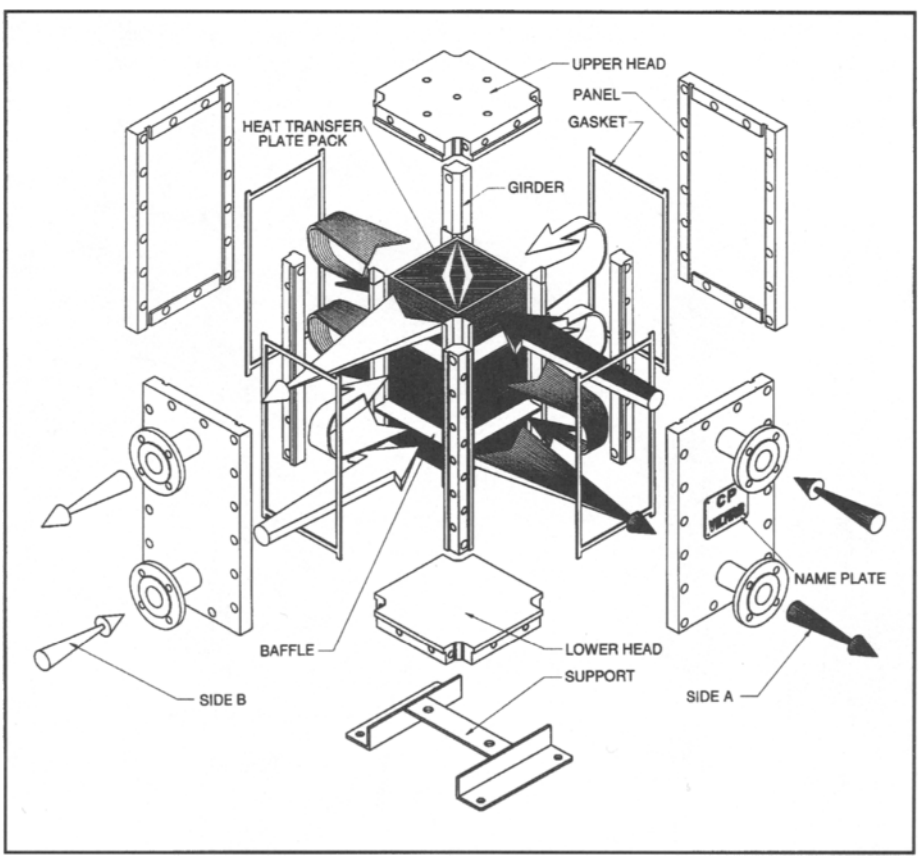


Figure 10-7C. The patented COMPABLOC® welded plate heat exchanger is technologically advanced, compact, and efficient. The fully welded design (but totally accessible on both sides) combines the best in performance, safety maintenance, and capital/maintenance costs. (Used by permission: Vicarb Inc., Canada, publication VNT–3110 © 1997.)

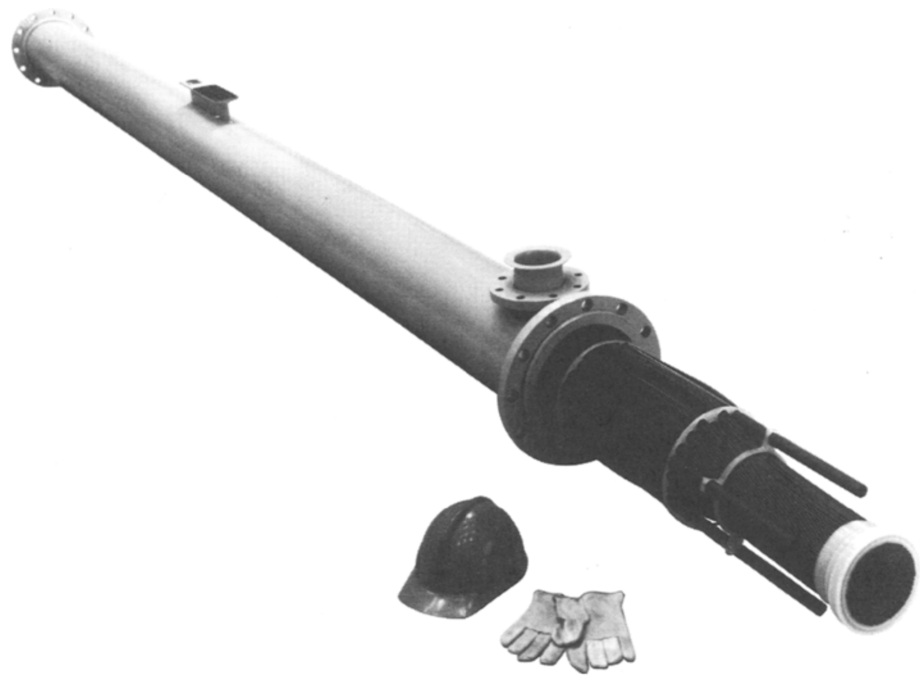


Figure 10-8. Single-pass shell and tube Teflon® tube heat exchanger, countercurrent flow. Tube bundles are flexible tube Teflon® joined in integral honeycomb tubesheets. Shell-side baffles are provided for cross-flow. Standard shell construction is carbon steel shell plain or Teflon (LT)® lined. Heads are lined with Teflon®. Tube diameters range from 0.125–0.375 in. O.D.; the temperature range is 80–400°F; pressures range from 40–150 psig. (Used by permission: AMETEK, Inc., Chemical Products Div., Product Bulletin "Heat Exchangers of Teflon®.")

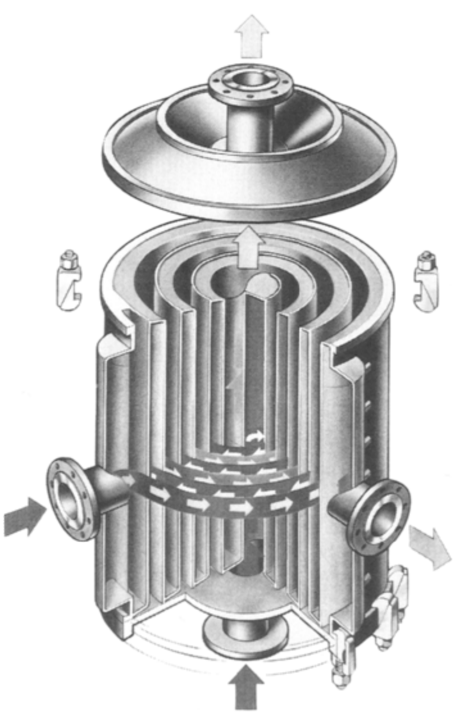
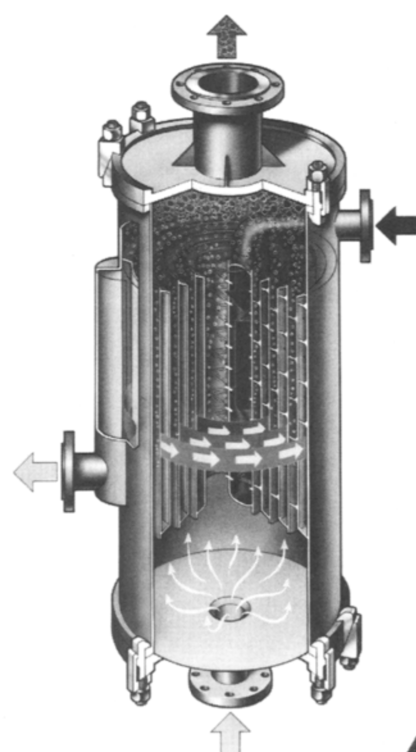


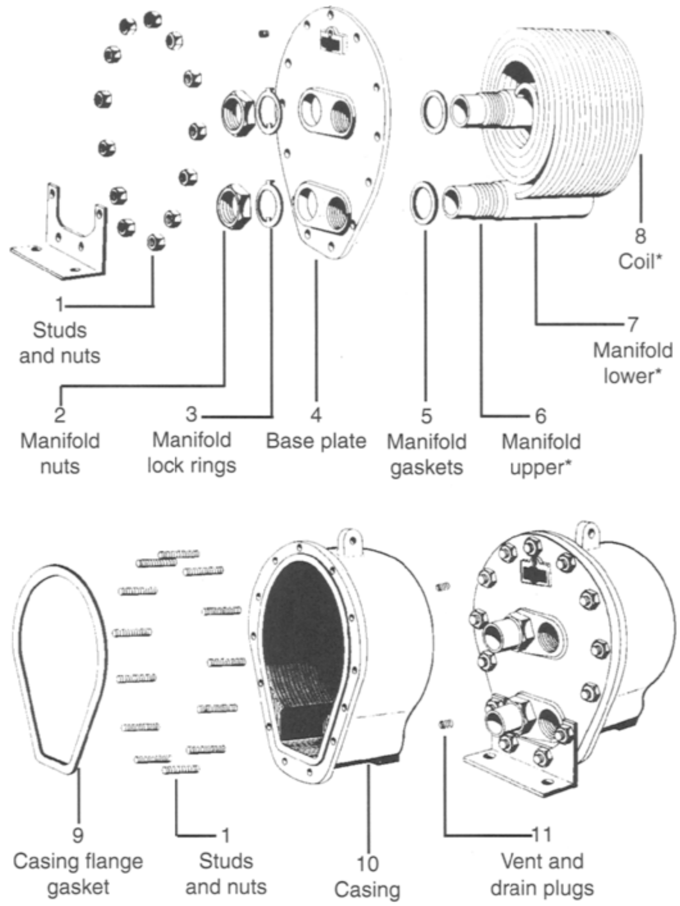
Figure 10-9A. Spiral flow heat exchanger, cross-flow arrangement for liquids, gases, or liquid/gaseous (condensable) fluids. (Used by permission: Alfa Laval Thermal Inc., Bul. 1205 © 1993.)



Alfa Laval

Figure 10-9B. Spiral flow heat exchanger; vaporizer. (Used by permission: Alfa Laval Thermal Inc., Bul. 1205 © 1993.)

Figure 10-9C. Coil Assembly for bare tube Heliflow® exchanger. Tube sizes range from $^1/_4$ –$^3/_4$ in. O.D. Tube-side manifold connections are shown for inlet and outlet fluid. (Used by permission: Graham Manufacturing Company, Inc., Bul. HHE–30 © 1992.)



*Although they are numbered separately for clarity in explaining the Heliflow® heat exchanger, Items 6, 7, and 8 are not separate items. Coil and manifolds are a one-piece factory assembly.

Figure 10-9D. Assembly of components of Heliflow® spiral heat exchanger. (Used by permission: Graham Manufacturing Company, Bul. "Operating and Maintenance Instructions for Heliflow®.")

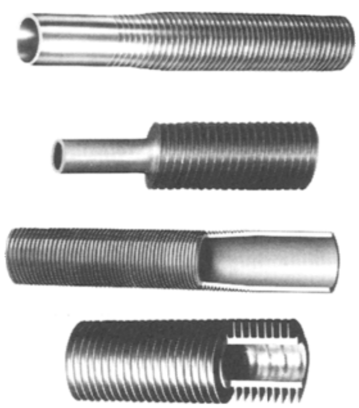


Figure 10-10A. Circular-type finned tubing. (Used by permission: Wolverine Tube, Inc.)

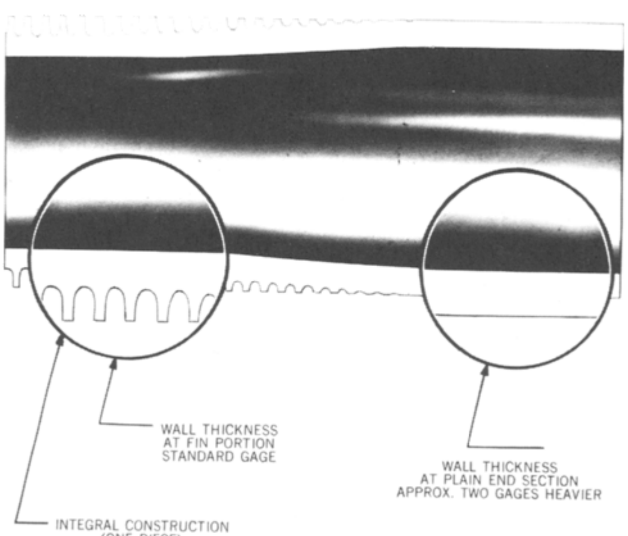


Figure 10-10B. Low-finned integral tube details. (Used by permission: Wolverine Tube, Inc.)

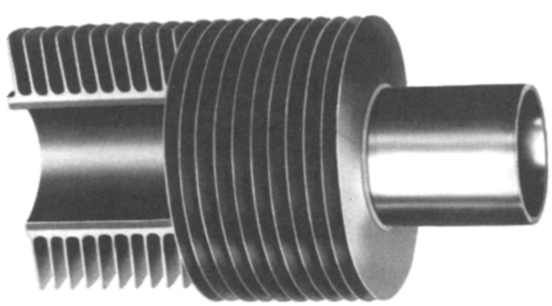


Figure 10-10C. Bimetal high-finned tube. (Used by permission: Wolverine Tube, Inc.)

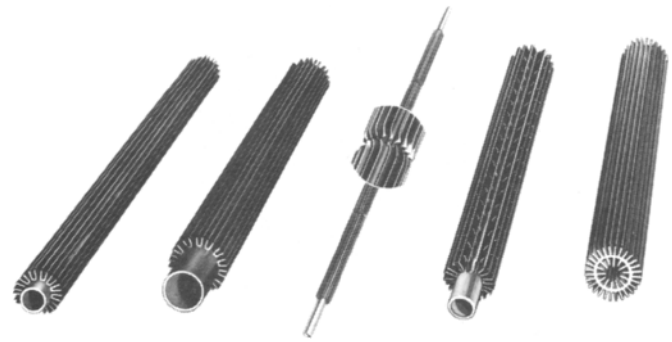


Figure 10-10D. Longitudinal fin tubes. (Used by permission: Brown Fintube Co., A Koch® Engineering Co.)

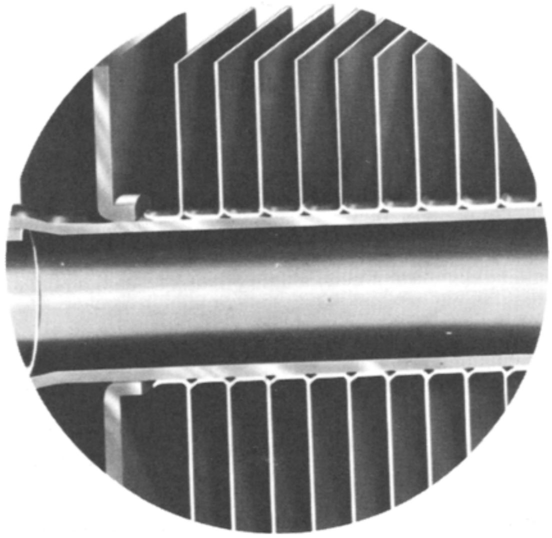


Figure 10-10E. A cutaway section of plate-type fins showing the continuous surface contact of the mechanically bonded tube and fins. (Used by permission: The Trane® Co., La Crosse, Wis.)

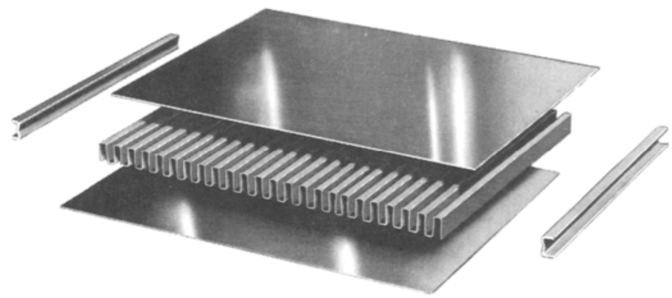


Figure 10-10F. Flat plate extended surface used in low-temperature gas separation plants; exploded view of brazed surfaces. (Used by permission: The Trane® Co., La Crosse, Wis.)

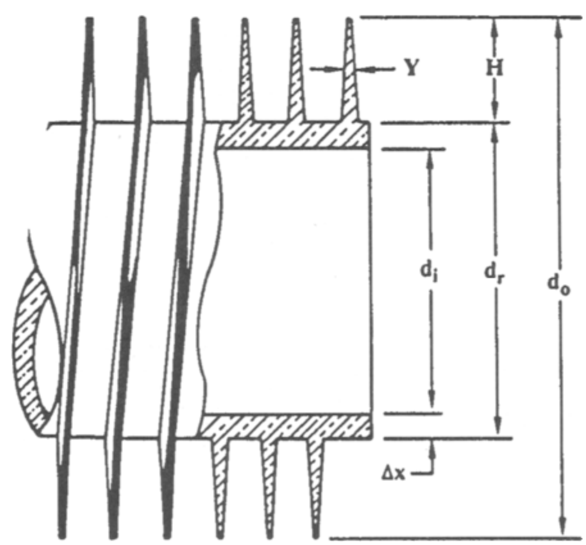


Figure 10-10H. Geometrical dimensions for High-Finned Wolverine Trufin® tubes. The fins are integral with the basic tube wall. (Used by permission: Wolverine Tube, Inc., *Engineering Data Book*, II, © 1984.)

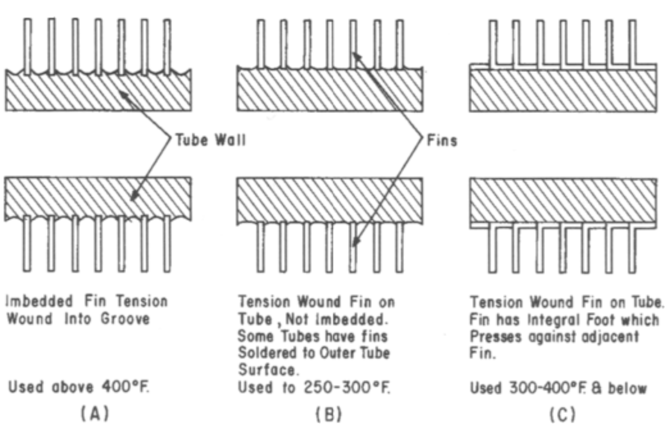


Figure 10-10G. Tension wound fins.

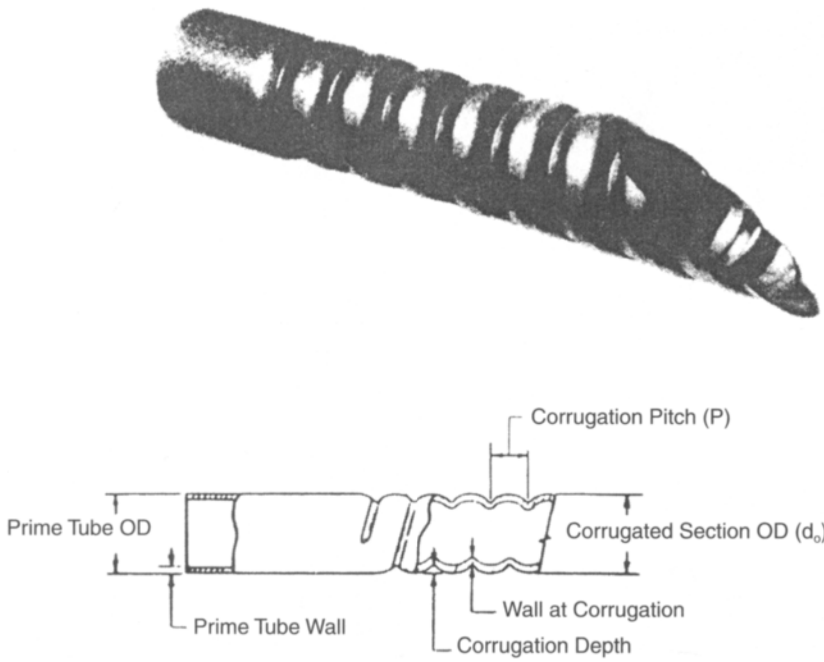


Figure 10-10I. Koro-Chil® corrugated tube, used primarily for D-X water-type chillers, water-cooled outside, refrigerant expanding/boiling inside. (Used by permission: Wolverine Tube, Inc.)

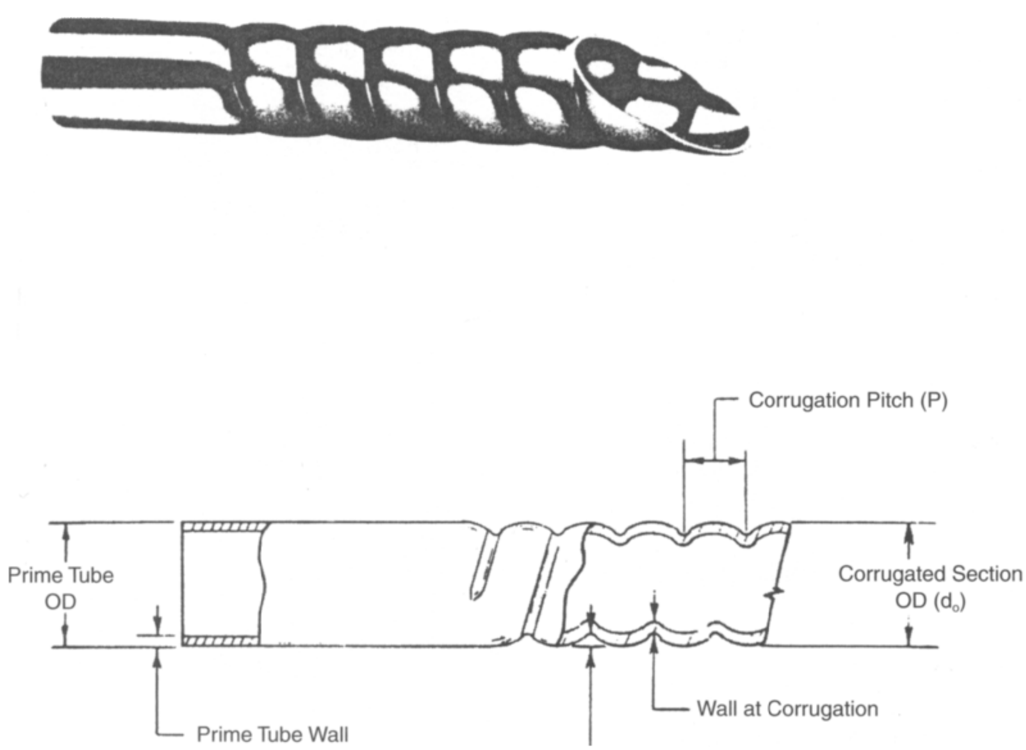
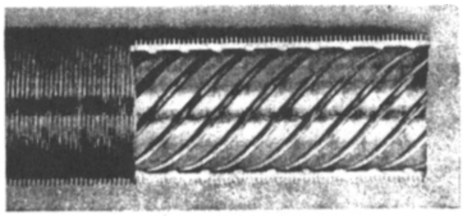


Figure 10-10J. Korodense® corrugated tube. Used primarily in steam condensing service and other power plant applications. Efficiency is reported at up to 50% greater than plain tubes. (Used by permission: Wolverine Tube, Inc.)



DIMENSIONAL NOMENCLATURE USED FOR TYPE S/T TURBO-CHIL

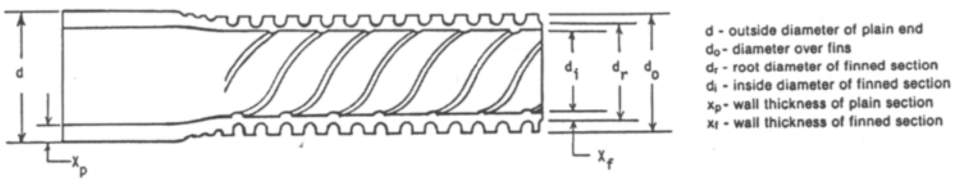


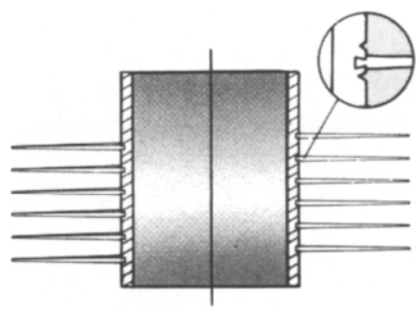
Figure 10-10K. Type S/T Turbo-Chil® finned tube with internal surface enhancement by integral ridging. (Used by permission: Wolverine Tube, Inc.)

Applied Fins manufactured on McElroy machines:

Base Tube Diameter:	From ⅝" min. to 2" max. (15.88mm—50.8mm)
Fin height:	From ¼" min. to ¾" max. (6.35mm—19.05mm)
Fin pitch:	From 5 fins/inch min. to 11.5 fins/inch max. (196—453 fins/metre)
Fin thickness:	From 0.012" min. to 0.028" max. (0.30mm—0.71mm)

'G' FIN (or Embedded fin)

The strip is tension wound into a machined groove and securely locked in place by back-filling with base tube material. This ensures that maximum heat transfer is maintained at high tube metal temperatures.

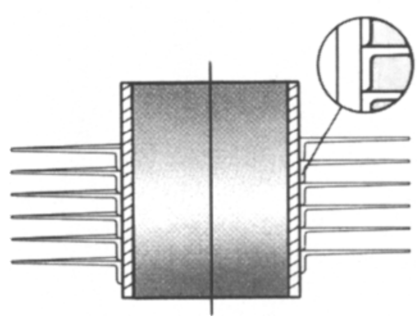


Maximum temperature: 450°C.

Fin material: Aluminium, Copper or steel.
Tube material: Carbon steel, Cr Mo steel, stainless steel, copper, copper alloys, incolloy, etc.

'L' FIN

Controlled deformation of the strip under tension gives optimum contact pressure of the foot on the base tube to maximise heat transfer performance.



The helical fin foot gives considerable corrosion protection to the base tube.

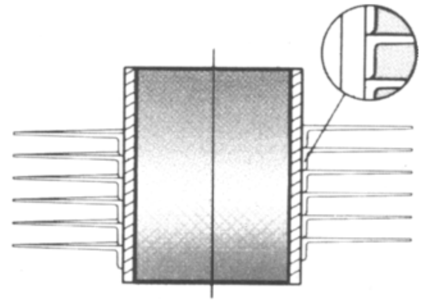
Maximum temperature: 150°C.

Fin material: Aluminium or copper.
Tube material: Any metallic material.

Figure 10-10L. Various fin manufacturing techniques used by Profins, Ltd., "Finned and Plain Tubes" bulletin. (Used by permission: Profins, Ltd., Burdon Drive, North West Industrial Estate, Peterlee, Co. Durham SR82HX, England.)

'KL' FIN

Manufactured exactly as the 'L' fin except that the base tube is knurled before application of the L-footed fin, then the fin foot is knurled into the corresponding knurling in the outer wall of the tube thereby giving much better thermal contact. This type of fin is much more resistant to thermal cycling than 'L' fin.

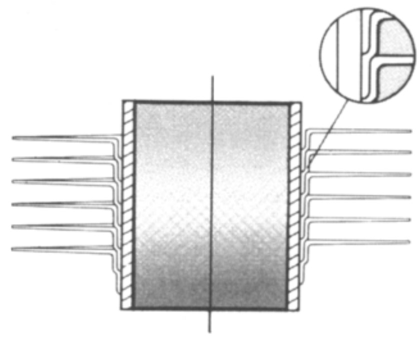


Maximum temperature: 260°C.

Fin material: Aluminium or copper.
Tube material: Any metallic material.

'LL' FIN

Manufactured by the same process as 'L' fin, the overlapped fin foot gives complete corrosion protection to the base tube.



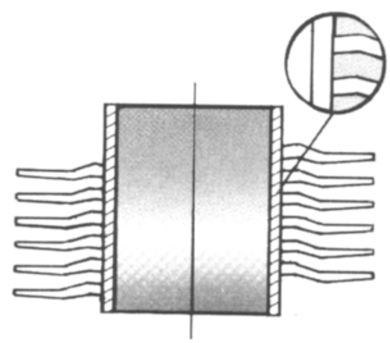
This is often used as an alternative to the more expensive extruded fin tube in hostile environments.

Maximum temperature: 180°C

Fin material: Aluminium or copper.
Tube material: Any metallic material.

Fins manufactured on Razmussen Machine

Semi-crimped fin is a non taper fin wrapped under tension around the outside of the base tube.



Fin is tack welded to the base tube at each end of the finned section or wherever the finning is interrupted.

Maximum temperature: 250°C.

Base tube diameter:	5/8"—4½" (15.88mm—114mm)
Fin height:	¼"—1" (6.4mm—25.4mm)
Fin pitch:	3 fins/inch—10 fins/inch (118 fins/metre—394 fins/metre)

Generally tube and fin is in carbon steel or stainless steel.

Figure 10-10L. Continued.

Table 10-3
Characteristics of Tubing

Tube O.D. Inches	B.W.G. Gage	Thickness In.	Internal Area In^2	Ft^2 External Surface Per Ft Length	Ft^2 Internal Surface Per Ft Length	Weight Per Ft Length Steel Lb*	Tube I.D. In.	Moment of Inertia In.^{4}	Section Modulus In.^{3}	Radius of Gyration In.	Constant C**	O.D. / I.D.	Transverse Metal Area In.^{2}
1/4	22	0.028	0.0296	0.0654	0.0508	0.066	0.194	0.00012	0.00098	0.0791	46	1.289	0.0195
	24	0.022	0.0333	0.0654	0.0539	0.054	0.206	0.00010	0.00083	0.0810	52	1.214	0.0158
	26	0.018	0.0360	0.0654	0.0560	0.045	0.214	0.00009	0.00071	0.0823	56	1.168	0.0131
	27	0.016	0.0373	0.0654	0.0571	0.040	0.218	0.00008	0.00065	0.0829	58	1.147	0.0118
3/8	18	0.049	0.0603	0.0982	0.0725	0.171	0.277	0.00068	0.0036	0.1166	94	1.354	0.0502
	20	0.035	0.0731	0.0982	0.0798	0.127	0.305	0.00055	0.0029	0.1208	114	1.230	0.0374
	22	0.028	0.0799	0.0982	0.0835	0.104	0.319	0.00046	0.0025	0.1231	125	1.176	0.0305
	24	0.022	0.0860	0.0982	0.0867	0.083	0.331	0.00038	0.0020	0.1250	134	1.133	0.0244
1/2	16	0.065	0.1075	0.1309	0.0969	0.302	0.370	0.0021	0.0086	0.1555	168	1.351	0.0888
	18	0.049	0.1269	0.1309	0.1052	0.236	0.402	0.0018	0.0071	0.1604	198	1.244	0.0694
	20	0.035	0.1452	0.1309	0.1126	0.174	0.430	0.0014	0.0056	0.1649	227	1.163	0.0511
	22	0.028	0.1548	0.1309	0.1162	0.141	0.444	0.0012	0.0046	0.1672	241	1.126	0.0415
5/8	12	0.109	0.1301	0.1636	0.1066	0.601	0.407	0.0061	0.0197	0.1865	203	1.536	0.177
	13	0.095	0.1486	0.1636	0.1139	0.538	0.435	0.0057	0.0183	0.1904	232	1.437	0.158
	14	0.083	0.1655	0.1636	0.1202	0.481	0.459	0.0053	0.0170	0.1939	258	1.362	0.141
	15	0.072	0.1817	0.1636	0.1259	0.426	0.481	0.0049	0.0156	0.1972	283	1.299	0.125
	16	0.065	0.1924	0.1636	0.1296	0.389	0.495	0.0045	0.0145	0.1993	300	1.263	0.114
	17	0.058	0.2035	0.1636	0.1333	0.352	0.509	0.0042	0.0134	0.2015	317	1.228	0.103
	18	0.049	0.2181	0.1636	0.1380	0.302	0.527	0.0037	0.0119	0.2044	340	1.186	0.089
	19	0.042	0.2299	0.1636	0.1416	0.262	0.541	0.0033	0.0105	0.2067	359	1.155	0.077
	20	0.035	0.2419	0.1636	0.1453	0.221	0.555	0.0028	0.0091	0.2090	377	1.126	0.065
3/4	10	0.134	0.1825	0.1963	0.1262	0.833	0.482	0.0129	0.0344	0.2229	285	1.556	0.259
	11	0.120	0.2043	0.1963	0.1335	0.808	0.510	0.0122	0.0326	0.2267	319	1.471	0.238
	12	0.109	0.2223	0.1963	0.1393	0.747	0.532	0.0116	0.0309	0.2299	347	1.410	0.219
	13	0.095	0.2463	0.1963	0.1466	0.665	0.560	0.0107	0.0285	0.2340	384	1.339	0.195
	14.	0.083	0.2679	0.1963	0.1529	0.592	0.584	0.0098	0.0262	0.2376	418	1.284	0.174
	15	0.072	0.2884	0.1963	0.1587	0.522	0.606	0.0089	0.0238	0.2411	450	1.238	0.153
	16	0.065	0.3019	0.1963	0.1623	0.476	0.620	0.0083	0.0221	0.2433	471	1.210	0.140
	17	0.058	0.3157	0.1963	0.1660	0.429	0.634	0.0076	0.0203	0.2455	492	1.183	0.126
	18	0.049	0.3339	0.1963	0.1707	0.367	0.652	0.0067	0.0178	0.2484	521	1.150	0.108
	20	0.035	0.3632	0.1963	0.1780	0.268	0.680	0.0050	0.0134	0.2531	567	1.103	0.079
7/8	10	0.134	0.2894	0.2291	0.1589	1.062	0.607	0.0221	0.0505	0.2662	451	1.442	0.312
	11	0.120	0.3167	0.2291	0.1662	0.969	0.635	0.0208	0.0475	0.2703	494	1.378	0.285
	12	0.109	0.3390	0.2291	0.1720	0.893	0.657	0.0196	0.0449	0.2736	529	1.332	0.262
	13	0.095	0.3685	0.2291	0.1793	0.792	0.685	0.0180	0.0411	0.2778	575	1.277	0.233
	14	0.083	0.3948	0.2291	0.1856	0.703	0.709	0.0164	0.0374	0.2815	616	1.234	0.207
	15	0.072	0.4197	0.2291	0.1914	0.618	0.731	0.0148	0.0337	0.2850	655	1.197	0.182
	16	0.065	0.4359	0.2291	0.1950	0.563	0.745	0.0137	0.0312	0.2873	680	1.174	0.165
	17	0.058	0.4525	0.2291	0.1987	0.507	0.759	0.0125	0.0285	0.2896	706	1.153	0.149
	18	0.049	0.4742	0.2291	0.2034	0.433	0.777	0.0109	0.0249	0.2925	740	1.126	0.127
	20	0.035	0.5090	0.2291	0.2107	0.314	0.805	0.0082	0.0187	0.2972	794	1.087	0.092
1	8	0.165	0.3526	0.2618	0.1754	1.473	0.670	0.0392	0.0784	0.3009	550	1.493	0.433
	10	0.134	0.4208	0.2618	0.1916	1.241	0.732	0.0350	0.0700	0.3098	656	1.366	0.365
	11	0.120	0.4536	0.2618	0.1990	1.129	0.760	0.0327	0.0654	0.3140	708	1.316	0.332
	12	0.109	0.4803	0.2618	0.2047	1.038	0.782	0.0307	0.0615	0.3174	749	1.279	0.305
	13	0.095	0.5153	0.2618	0.2121	0.919	0.810	0.0280	0.0559	0.3217	804	1.235	0.270
	14	0.083	0.5463	0.2618	0.2183	0.814	0.834	0.0253	0.0507	0.3255	852	1.199	0.239
	15	0.072	0.5755	0.2618	0.2241	0.714	0.856	0.0227	0.0455	0.3291	898	1.168	0.210
	16	0.065	0.5945	0.2618	0.2278	0.650	0.870	0.0210	0.0419	0.3314	927	1.149	0.191
	18	0.049	0.6390	0.2618	0.2361	0.498	0.902	0.0166	0.0332	0.3367	997	1.109	0.146
	20	0.035	0.6793	0.2618	0.2435	0.361	0.930	0.0124	0.0247	0.3414	1060	1.075	0.106
1 1/4	7	0.180	0.6221	0.3272	0.2330	2.059	0.890	0.0890	0.1425	0.3836	970	1.404	0.605
	8	0.165	0.6648	0.3272	0.2409	1.914	0.920	0.0847	0.1355	0.3880	1037	1.359	0.562
	10	0.134	0.7574	0.3272	0.2571	1.599	0.982	0.0742	0.1187	0.3974	1182	1.273	0.470
	11	0.120	0.8012	0.3272	0.2644	1.450	1.010	0.0688	0.1100	0.4018	1250	1.238	0.426
	12	0.109	0.8365	0.3272	0.2702	1.330	1.032	0.0642	0.1027	0.4052	1305	1.211	0.391
	13	0.095	0.8825	0.3272	0.2775	1.173	1.060	0.0579	0.0926	0.4097	1377	1.179	0.345
	14	0.083	0.9229	0.3272	0.2838	1.036	1.084	0.0521	0.0833	0.4136	1440	1.153	0.304
	16	0.065	0.9852	0.3272	0.2932	0.824	1.120	0.0426	0.0682	0.4196	1537	1.116	0.242
	18	0.049	1.0423	0.3272	0.3016	0.629	1.152	0.0334	0.0534	0.4250	1626	1.085	0.185
	20	0.035	1.0936	0.3272	0.3089	0.455	1.180	0.0247	0.0395	0.4297	1706	1.059	0.134
1 1/2	10	0.134	1.1921	0.3927	0.3225	1.957	1.232	0.1354	0.1806	0.4853	1860	1.218	0.575
	12	0.109	1.2908	0.3927	0.3356	1.621	1.282	0.1159	0.1545	0.4933	2014	1.170	0.476
	14	0.083	1.3977	0.3927	0.3492	1.257	1.334	0.0931	0.1241	0.5018	2180	1.124	0.369
	16	0.065	1.4741	0.3927	0.3587	0.997	1.370	0.0756	0.1008	0.5079	2300	1.095	0.293
2	11	0.120	2.4328	0.5236	0.4608	2.412	1.760	0.3144	0.3144	0.6660	3795	1.136	0.709
	12	0.109	2.4941	0.5236	0.4665	2.204	1.782	0.2904	0.2904	0.6697	3891	1.122	0.648
	13	0.095	2.5730	0.5236	0.4739	1.935	1.810	0.2586	0.2586	0.6744	4014	1.105	0.569
	14	0.083	2.6417	0.5236	0.4801	1.701	1.834	0.2300	0.2300	0.6784	4121	1.091	0.500

*Weights are based on low carbon steel with a density of 0.2836 lb/in.^{3} For other metals multiply by the following factors:

Aluminum	0.35	A.I.S.I. 300 Series S/Steels	1.02	Nickel-Chrome-Iron	1.07	Nickel-Copper	1.12
Titanium	0.58	Aluminum Bronze	1.04	Admiralty	1.09	Copper and Cupro-Nickels	1.14
A.I.S.I. 400 Series S/Steels	0.99	Aluminum Brass	1.06	Nickel	1.13		

$$\text{**Liquid Velocity} = \frac{\text{lb per tube hour}}{\text{C} \times \text{sp. gr. of liquid}} \text{ in. ft per sec (sp. gr. of water at 60°F = 1.0)}$$

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Table 10-4
Thermal Conductivity of Metals

Temp. °F	70	100	200	300	400	500	600	700	800	900	1,000	1,100	1,200	1,300	1,400	1,500
	Btu/hr ft °F															
Material																
Carbon Steel	30.0	29.9	29.2	28.4	27.6	26.6	25.6	24.6	23.5	22.5	21.4	20.2	19.0	17.6	16.2	15.6
C-1/2 Moly Steel	24.8	25.0	25.2	25.1	24.8	24.3	23.7	23.0	22.2	21.4	20.4	19.5	18.4	16.7	15.3	15.0
1 Cr-1/2 Mo & 1-1/4 Cr-1/2 Mo	21.3	21.5	21.9	22.0	21.9	21.7	21.3	20.8	20.2	19.7	19.1	18.5	17.7	16.5	15.0	14.8
2-1/4 Cr-1 Mo	20.9	21.0	21.3	21.5	21.5	21.4	21.1	20.7	20.2	19.7	19.1	18.5	18.0	17.2	15.6	15.3
5 Cr-1/2 Mo	16.9	17.3	18.1	18.7	19.1	19.2	19.2	19.0	18.7	18.4	18.0	17.6	17.1	16.6	16.0	15.8
7 Cr-1/2 Mo	14.1	14.4	15.3	16.0	16.5	16.9	17.1	17.2	17.3	17.2	17.1	16.8	16.6	16.2	15.6	15.5
9 Cr-1 Mo	12.8	13.1	14.0	14.7	15.2	15.6	15.9	16.0	16.1	16.1	16.1	16.0	15.8	15.6	15.2	15.0
3-1/2 Nickel	22.9	23.2	23.8	24.1	23.9	23.4	22.9	22.3	21.6	20.9	20.1	19.2	18.2	16.9	15.5	15.3
13 Cr	15.2	15.3	15.5	15.6	15.8	15.8	15.9	15.9	15.9	15.9	15.8	15.6	15.3	15.1	15.0	15.0
15 Cr	14.2	14.2	14.4	14.5	14.6	14.7	14.7	14.8	14.8	14.8	14.8	14.8	14.8	14.8	14.8	14.8
17 Cr	12.6	12.7	12.8	13.0	13.1	13.2	13.3	13.4	13.5	13.6	13.7	13.8	13.9	14.1	14.3	14.5
TP 304 Stn. Stl.	8.6	8.7	9.3	9.8	10.4	10.9	11.3	11.8	12.2	12.7	13.2	13.6	14.0	14.5	14.9	15.3
TP 316 & 317 Stn. Stl.	7.7	7.9	8.4	9.0	9.5	10.0	10.5	11.0	11.5	12.0	12.4	12.9	13.3	13.8	14.2	14.6
TP 321 & 347 Stn. Stl.	8.1	8.4	8.8	9.4	9.9	10.4	10.9	11.4	11.9	12.3	12.8	13.3	13.7	14.1	14.6	15.0
TP 310 Stn. Stl.	7.3	7.5	8.0	8.6	9.1	9.6	10.1	10.6	11.1	11.6	12.1	12.6	13.1	13.6	14.1	14.5
Nickel 200			38.8	37.2	35.4	34.1	32.5	31.8	32.5	33.1	33.8					
Ni-Cu Alloy 400	12.6	12.9	13.9	15.0	16.1	17.0	17.9	18.9	19.8	20.9	22.0					
Ni-Cr-Fe Alloy 600	8.6	8.7	9.1	9.6	10.1	10.6	11.1	11.6	12.1	12.6	13.2	13.8	14.3	14.9	15.5	16.0
Ni-Fe-Cr Alloy 800	6.7	6.8	7.4	8.0	8.6	9.1	9.6	10.1	10.6	11.1	11.6	12.1	12.7	13.2	13.8	14.5
Ni-Fe-Cr-Mo-Cu Alloy 825			7.1	7.6	8.1	8.6	9.1	9.6	10.0	10.4	10.9	11.4	11.8	12.4	12.9	13.6
Ni-Mo Alloy B		6.1	6.4	6.7	7.0	7.4	7.7	8.2	8.7	9.3	10.0	10.7				
Ni-Mo-Cr Alloy C-276		5.9	6.4	7.0	7.5	8.1	8.7	9.2	9.8	10.4	11.0	11.5	12.1		13.2	
Aluminum Alloy 3003	102.3	102.8	104.2	105.2	106.1											
Aluminum Alloy 6061	96.1	96.9	99.0	100.6	101.9											
Titanium	12.7	12.5	12.0	11.7	11.5	11.3	11.2	11.1	11.2	11.3	11.4	11.6				
Admiralty			70.0	75.0	79.0	84.0	89.0									
Naval Brass			71.0	74.0	77.0	80.0	83.0									
Copper			225.0	225.0	224.0	224.0	223.0									
90-10 Cu-Ni			30.0	31.0	34.0	37.0	42.0	47.0	49.0	51.0	53.0					
70-30 Cu-Ni			18.0	19.0	21.0	23.0	25.0	27.0	30.0	33.0	37.0					
Muntz			71.0													
Zirconium			12.0													
Cr-Mo Alloy XM-27			11.3													
Cr-Ni-Fe-Mo-Cu-Cb (Alloy 20Cb)			7.6													
Ni-Cr-Mo-Cb (Alloy 625)	5.67	5.83	6.25		7.25		8.17		9.08		10.08		11.0		12.0	

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American Brass Co.
Airco, Inc.
Cabot-Stellite
Carpenter Technology
International Nickel Co.

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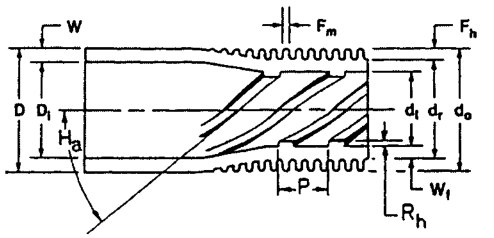
Errata Note: k = BTU/(hr)(ft)(°Ft).

DEFINITION OF "MICRORIB" I.D. - 272710

27 NUMBER OF RIBS
27 HELIX ANGLE OF RIBS
10 RIB HEIGHT (THOUSANDTHS OF AN INCH)



D — Outside Diameter of Plain End
D_i — Inside Diameter of Plain End
d_r — Root Diameter
d_o — Diameter Over Fins
d_i — Inside Diameter of Fin Section
W — Wall Thickness of Plain End
W_f — Wall Thickness Under Fin
F_h — Height of Fin
F_m — Mean Fin Thickness
P — Mean Rib Pitch
R_h — Height of Rib
H_a — Rib Helix Angle

Figure 10-10M. Finned tube with internal ribs enhances heat transfer inside as well as outside the tubes. (Used by permission: High Performance Tube, Inc., "Finned Tube Data Book.")

Finned tubes may have the fin externally or internally. The most common and perhaps adaptable is the external fin. Several types of these use the fin (a) as an integral part of the main tube wall, (b) attached to the outside of the tube by welding or brazing, (c) attached to the outside of the tube by mechanical means. Figure 10-10 illustrates several different types. The fins do not have to be of the same material as the base tube, Figure 10-11.

The usual applications for finned tubes are in heat transfer involving gases on the outside of the tube. Other applications also exist, such as condensers, and in fouling service where the finned tube has been shown to be beneficial. The total gross external surface in a finned exchanger is many times that of the same number of plain or bare tubes.

Tube-side water velocities should be kept within reasonable limits, even though calculations would indicate that improved tube-side film coefficients can be obtained if the water velocity is increased. Table 10-24 suggests guidelines that recognize the possible effects of erosion and corrosion on the system.

Bending of Tubing

The recommended minimum *radius* of bend for various tubes is given in Table 10-5. These measurements are for 180° U-bends and represent minimum values.

TEMA, Par. RCB 2.31 recommends the minimum wall thinning of tubes for U-Bends by the minimum wall thickness in the bent portion before bending, t_1.

$$t_o = t_1\left[1 + \frac{d_o}{4R}\right] \tag{10–1}$$

where t_o = original tube wall thickness, in.
t_1 = minimum tube wall thickness calculated by code rules for straight tube subjected to the same pressure and metal temperature.
d_o = O.D. or tube in in.
R = mean radius of bend, in.

See TEMA for more details.

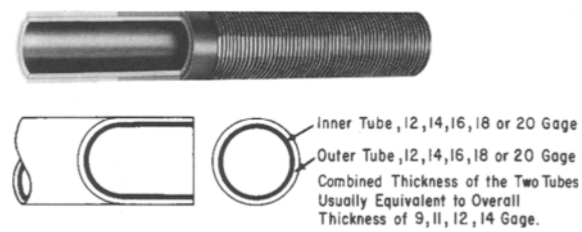


Figure 10-11. Duplex tube. Note inside liner is resistant to tube-side fluid and outer finned tube is resistant to shell-side fluid. (Used by permission: Wolverine Tube, Inc.)

Table 10-5
Manufacturers' Suggested Minimum Radius of Bend for Tubes

Tube O.D., In.	Bend Radius, In.	Center-to-Center Distance
Duplex, all sizes	3 × tube O.D.	6 × tube O.D.
*Plain: $^5/_8$ in.	$^{13}/_{16}$ in.	1 $^5/_8$ in.
$^3/_4$ in.	1 in.	2 in.
1 in.	1$^3/_{16}$ in.	2 $^3/_8$ in.

*For bends this sharp, the tube wall on the outer circumference of the tube may thin down 1$^1/_2$–2 gage thicknesses, depending on the condition and specific tube material. More generous radii will reduce this thinning. TEMA[107] presents a formula for calculating the minimum wall thickness.

5. Baffles

Baffles are a very important part of the performance of a heat exchanger. Velocity conditions in the tubes as well as those in the shell are adjusted by design to provide the necessary arrangements for maintenance of proper heat transfer fluid velocities and film conditions. Consider the two classes of baffles described in the following sections.

A. Tube Side Baffles

These baffles are built into the head and return ends of an exchanger to direct the fluid through the tubes at the proper relative position in the bundle for good heat transfer as well as for fixing velocity in the tubes, see Figures 10-1D and 10-3.

Baffles in the head and return ends of exchangers are either welded or cast in place. The arrangement may take any of several reasonable designs, depending upon the number of tube-side passes required in the performance of the unit. The number of tubes per pass is usually arranged about equal. However, depending upon the physical changes in the fluid volume as it passes through the unit, the number of tubes may be significantly different in some of the passes. Practical construction limits the number of tube-side passes to 8–10, although a larger number of passes may be used on special designs. It is often better to arrange a second shell unit with fewer passes each. The pass arrangements depend upon the location of entrance and exit nozzle connections in the head and the position of the fluid paths in the shell side. Every effort is usually made to visualize the physical flow and the accompanying temperature changes in orienting the passes. Figures 10-12 and 10-13 illustrate a few configurations.

Single-pass Tube Side. For these conditions, no baffle is in either the head or the return end of the unit. The tube-side fluid enters one end of the exchanger and leaves from the opposite end. In general, these baffles are not as convenient from a connecting pipe arrangement viewpoint as units with an even number of passes in which the tube-side fluid enters and leaves at the same end of the exchanger. See Figures 10-1C and 10-1G and Table 10-1.

Two-pass Tube Side. For these conditions one head end baffle is usually in the center, and no baffle is in the return end, as the fluid will return through the second pass of itself. See Figures 10-1A and 10-1B.

Three-pass Tube Side; five-pass Tube Side. These are rare designs because they require baffles in both heads, and the outlet connection is at the end opposite the inlet. This provides the same poor piping arrangement as for a single-pass unit.

Four-pass Tube Side; Even Number of Passes Tube Side. These conditions are often necessary to provide fluid velocities high enough for good heat transfer or to prevent the deposition of suspended particles in the tubes and end chambers. The higher the number of passes, the more expensive the unit.

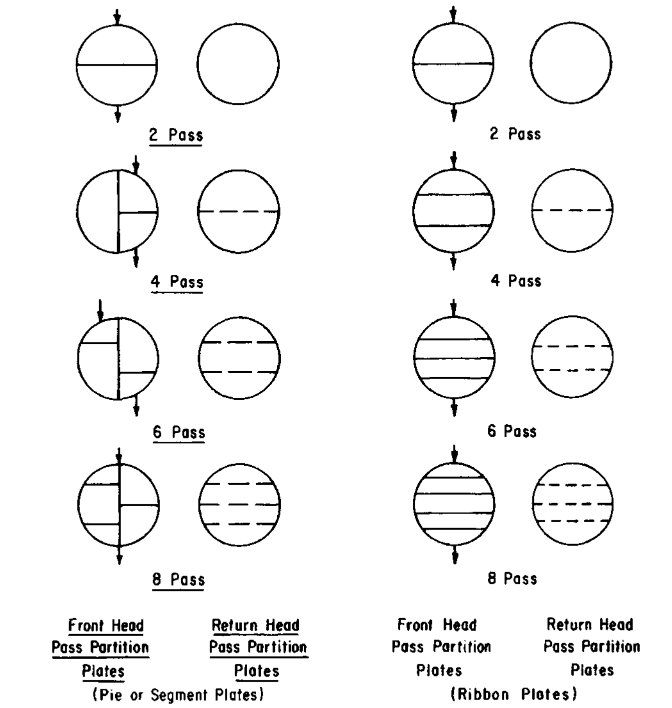


Figure 10-12. Tube-side pass arrangements.

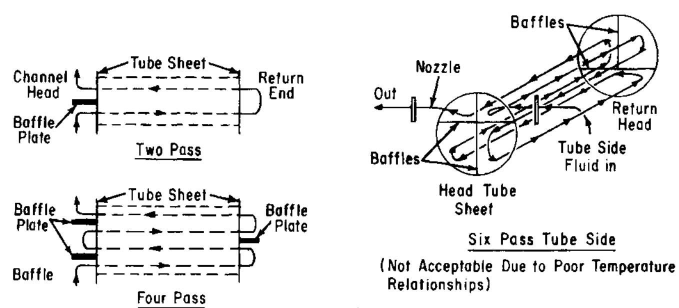


Figure 10-13. Tube-side baffles.

The more passes in a head, the more difficult the problem of fluid by-passing through the gasketed partitions becomes, unless expensive construction is used. Seating of all partitions due to warping of the metals, even though machined, is a real problem. At high pressure above about 500 psig, multiple-pass units are only sparingly used. See Figure 10-1J.

B. Shell-Side Baffles and Tube Supports

Only a few popular and practical shell baffle arrangements exist, although special circumstances can and do require many unique baffling arrangements. The performance of the shell side of the exchanger depends upon the designer's understanding the effectiveness of fluid contact with the tubes as a direct result of the baffle pattern used.

The baffle cut determines the fluid velocity between the baffle and the shell wall, and the baffle spacing determines the parallel and cross-flow velocities that affect heat transfer and pressure drop. Often the shell side of an exchanger is subject to low-pressure drop limitations, and the baffle patterns must be arranged to meet these specified conditions and at the same time provide maximum effectiveness for heat transfer. The plate material used for these supports and baffles should not be too thin and is usually $^{3}/_{16}$-in. minimum thickness to $^{1}/_{2}$-in. for large units. TEMA has recommendations. Figure 10-14 summarizes the usual arrangements for baffles.

a. Tube Supports. Tube supports for horizontal exchangers are usually segmental baffle plates cut off in a vertical plane to a maximum position of one tube past the centerline of the exchanger and at a minimum position of the centerline. The cut-out portion allows for fluid passage. Sometimes horizontally cut plates are used when baffles are used in a shell, and extra tube supports may not be needed. It takes at least two tube supports to properly support all the tubes in an exchanger when placed at maximum spacing. A tube will sag and often vibrate to destruction if not properly supported. However, because only half of the tubes can be sup-

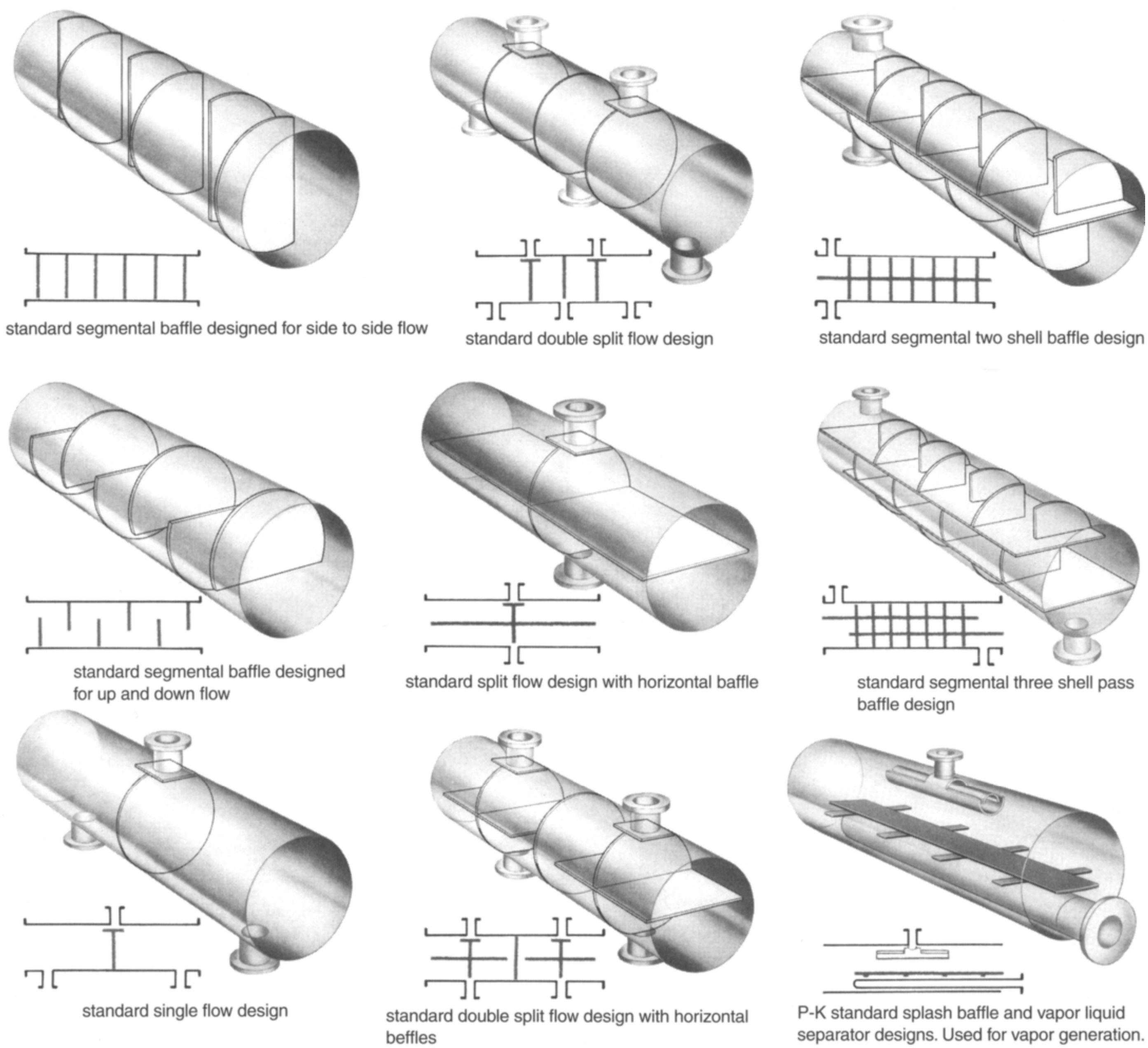


Figure 10-14. Shell baffle arrangements. (Used by permission: Patterson-Kelley Div., a Harsco Company, "Manual No. 700A.")

Table 10-6
Maximum Unsupported Straight Tube Spans
(All Dimensions in In.)

Tube O.D.	Tube Materials and Temperature Limits (°F): Carbon Steel & High Alloy Steel (750) Low Alloy Steel (850) Nickel-Cooper (600) Nickel (850) Nickel-Chromium-Iron(1000)	Aluminum & Aluminum Alloys, Copper & Copper Alloys, Titanium Alloys at Code Maximum Allowable Temperature
$^1/_4$	26	22
$^3/_8$	35	30
$^1/_2$	44	38
$^5/_8$	52	45
$^3/_4$	60	52
$^7/_8$	69	60
1	74	64
$1^1/_4$	88	76
$1^1/_2$	100	87
2	125	110

Notes:

(1) Above the metal temperature limits shown, maximum spans shall be reduced in direct proportion to the fourth root of the ratio of elastic modulus at temperature to elastic modulus at tabulated limit temperature.

(2) In the case of circumferentially finned tubes, the tube O.D. shall be the diameter at the root of the fins and the corresponding tabulated or interpolated span shall be reduced in direct proportion to the fourth root of the ratio of the weight per unit length of the tube, if stripped of fins to that of the actual finned tube.

(3) The maximum unsupported tube spans in Table 10-6 do not consider potential flow-induced vibration problems. Refer to Section 6 for vibration criteria.

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ported by one support, the support plate must be alternated in orientation in the shell. The approximate maximum unsupported tube length and maximum suggested tube support spacing are given in Table 10-6.

Although detailed calculations might indicate that for varying materials with different strengths the spacing could be different, it is usually satisfactory to follow the guides in Table 10-6 for any material commonly used in heat exchangers. Practice allows reasonable deviation without risking trouble in the unit.

The tube support acts as a baffle at its point of installation and should be so considered, particularly in pressure-drop calculations. Tube supports are often ignored in heat transfer coefficient design. They should also be provided with openings in the lower portion at the shell to allow liquid drainage to the outlet. Holes for tubes are drilled $^1/_{64}$-in. larger than tube O.D. when unsupported length is greater than 36 in. and are drilled $^1/_{32}$-in. larger when the unsupported tube length is 36 in. or less, per TEMA standards, and are free of burrs. If there is much clearance, the natural flow vibration will cause the edge of the support to cut the tube. Pulsating conditions require special attention, and holes are usually drilled tight to tube O.D.

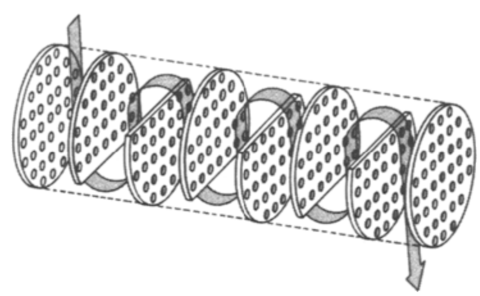

Figure 10-15. Horizontal cut segmental baffles. (Used by permission: B.G.A. Skrotzki, B.G.A. Power, © June 1954. McGraw-Hill, Inc. All rights reserved.)

b. Segmental Baffles. This type of baffle is probably the most popular. It is shown in Figures 10-15 and 10-16 for horizontal and vertical cuts, respectively. A segmental baffle is a circle of near shell diameter from which a horizontal or vertical portion has been cut. The cut-out portion, which represents the free-flow area for shell-side fluid, is usually from 20 to near 50% of the open shell area. The net flow area in this space must recognize the loss of flow area covered by tubes in the area. Tube holes are drilled as for tube supports.

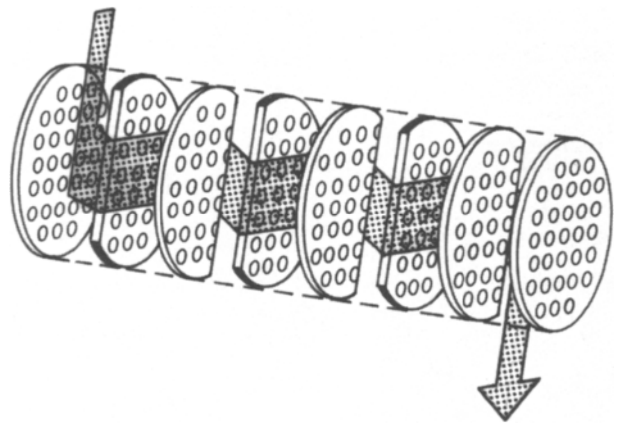

Figure 10-16. Vertical cut segmental baffles. (Used by permission: B.G.A. Skrotzki, B.G.A. Power, © June 1954, by McGraw-Hill, Inc. All rights reserved.)

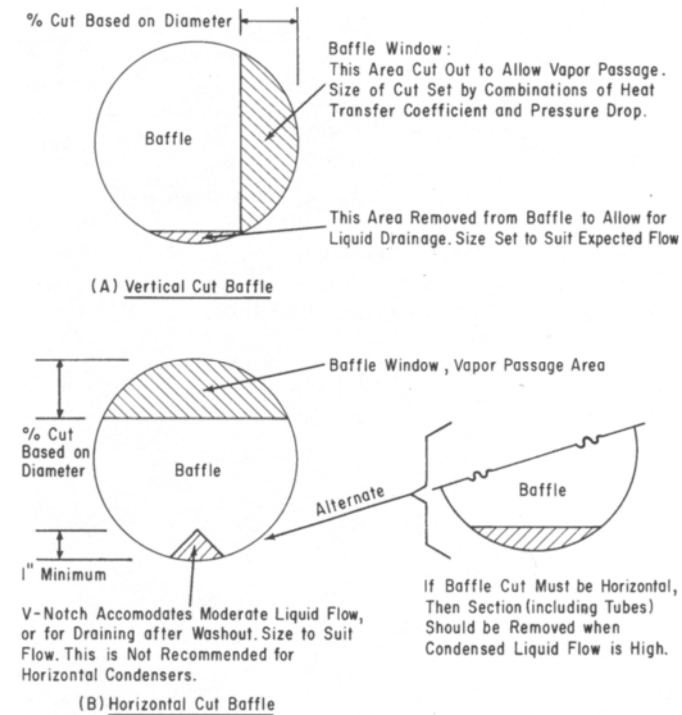

Figure 10-17. Baffle details.

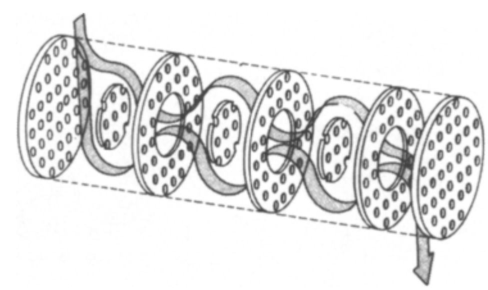

Figure 10-18. Disc and doughnut baffles. (Used by permission: B.G.A. Skrotzki, B.G.A. Power, © June 1954, by McGraw-Hill, Inc. All rights reserved.)

The baffle edge is usually vertical for service in horizontal condensers, reboilers, vaporizers, and heat exchangers carrying suspended matter or with heavy fouling fluids. With this arrangement, noncondensable vapors and inert gases can escape or flow along the top of the unit. Thus, they prevent vapor binding or vapor lock causing a blanking to heat transfer of the upper portion of the shell. Also as important as vapor passage is liquid released from the lower portion of the shell as it is produced. Although provision should be made in the portion of the baffle that rests on the lower portion of the shell for openings to allow liquid passage, it is a good practice to use the vertical baffle cut to allow excess liquid to flow around the edge of the baffle without building up and blanking the tubes in the lower portion of the exchanger, Figure 10-17.

The horizontal cut baffles are good for all gas-phase or all liquid-phase service in the shell. However, if dissolved gases in the liquid can be released in the exchanger, this baffling should not be used, or notches should be cut at the top for gas passage. Notches will not serve for any significant gas flow, just for traces of released gas. Liquids should be clean; otherwise sediment will collect at the base of every other baffle segment and blank off part of the lower tubes to heat transfer.

c. Disc and Doughnut Baffles. The flow pattern through these baffles is uniform through the length of the exchanger. This is not the case for segmental baffles. The disc and the doughnut are cut from the same circular plate and are placed alternately along the length of the tube bundle as shown in Figure 10-18.

Although these baffles can be as effective as the segmental ones for single-phase heat transfer, they are not used as often. The fluid must be clean; otherwise sediment will deposit behind the doughnut and blank off the heat transfer area. Also, if inert or dissolved gases can be released, they cannot be vented effectively through the top of the doughnut. If condensables exist, the liquid cannot be drained without large ports or areas at the base of the doughnut.

d. Orifice Baffles. This baffle is seldom used except in special designs, as it is composed of a full circular plate with holes drilled for all tubes about $^1/_{16}$-in. to $^1/_8$-in. larger than the outside diameter of the tube (see Figure 10-19). The clean fluid (and it must be very clean) passes through the annulus between the outside of the tube and the drilled hole in the baffle. Considerable turbulence is at the orifice but very little cross-flow exists between baffles. Usually condensables can be drained through these baffles unless the flow is high, and noncondensables can be vented across the top. For any performance, the pressure drop is usually high, and it is mainly for this and the cleanliness of fluid requirements that these baffles find few industrial applications.

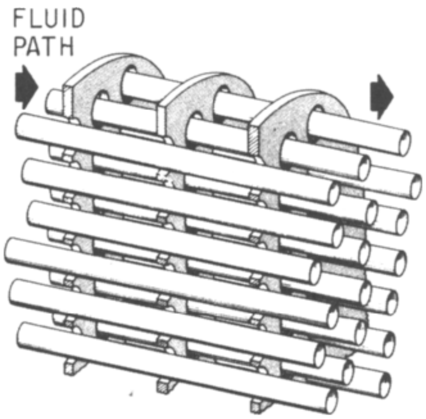

Figure 10-19. Baffles with annular orifices. (Used by permission: B.G.A. Skrotzki, B.G.A. Power, © June 1954, by McGraw-Hill, Inc. All rights reserved.)

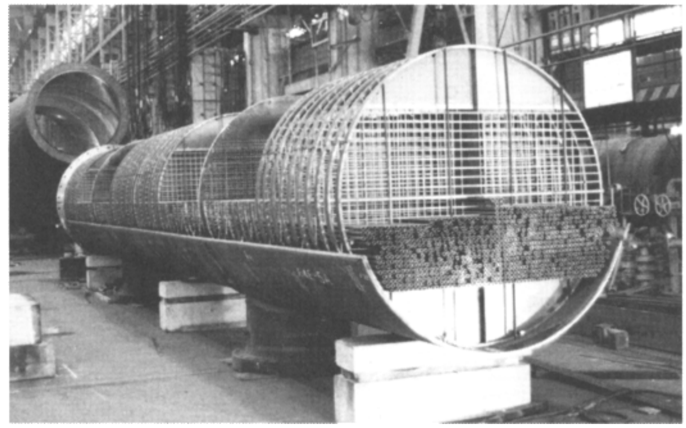

Figure 10-20B. RODbaffle® Intercooler in fabrication, 67 in. × 40 ft, 2,232–$^3/_4$-in. O.D. copper-nickel tubes, 1.00 in. pitch. TEMA AHL. (Used by permission: © Phillips Petroleum Company, Licensing Div., Bul. 1114–94-A–01.)

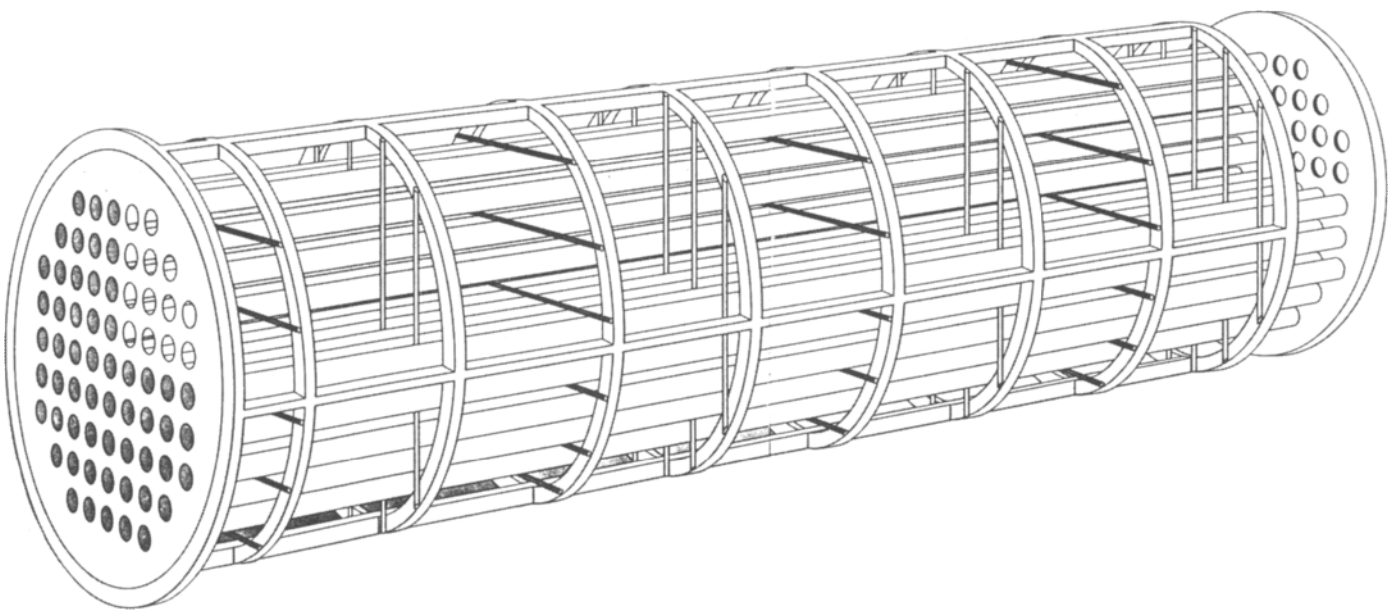

Figure 10-20A. RODbaffle® exchanger cross-section showing assembly, using TEMA E, F, H, J, K, and X shells. (Used by permission: © Phillips Petroleum Company, Licensing Div., Bul. 1114–94-A–01.)

e. RODbaffles®. These baffles are rods set throughout the shell side of the tube bundle (see Figures 10-20A–D). The primary objective in using this style of baffle is to reduce tube failure from the vibrational damage that can be caused by the various metal baffles versus metal tube designs. The RODbaffles® are designed to overcome the tube vibration mechanisms of (a) vortex shedding, (b) turbulence, and (c) fluid elastic vibration. For proper application and design, the engineer should contact Phillips Petroleum Company Licensing Division for names of qualified design/manufacturing fabricators. This unique design has many varied applications, but they can be handled only by licensed organizations.

f. Impingement Baffles. These baffles are located at inlet flow areas to the shell side of tube bundles to prevent suspended solid particles or high-velocity liquid droplets in gas streams from cutting, pitting, and otherwise eroding portions of the tubes. Several arrangements exist for effectively placing these baffles as shown in Figures 10-21A–C.

Besides preventing a destruction of the tubes, impingement plates serve to spread out and distribute the incoming fluid into the tube bundle. If they are used in proper relation to the bundle cross-flow baffles, the fluid can be effectively spread across the bundle near the inlet end. If this is not accomplished, part of the tube area will be stagnant, and its heat transfer will be less than the other parts of the

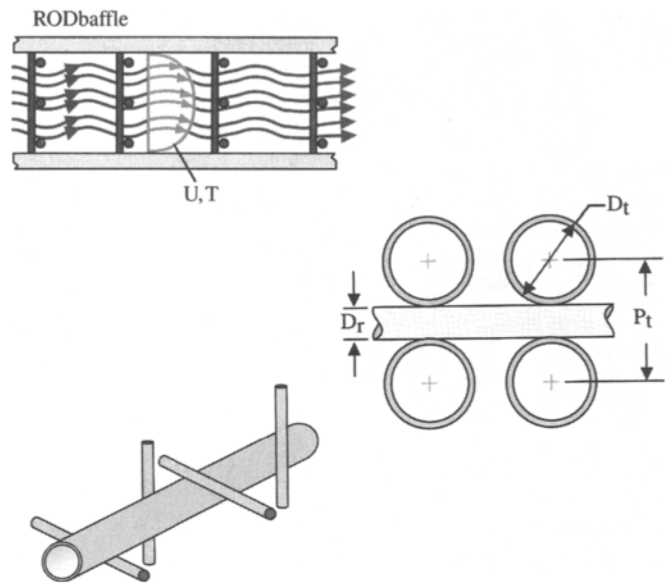

Figure 10-20C. RODbaffle® tube-baffle details. (Used by permission: © Phillips Petroleum Company, Licensing Div., Bul. 1114–94-A–01.)

exchanger. Some indications are that these stagnant partially effective areas may be 10–20% of the total exchanger surface in a 16-ft long bundle.[55] It is apparent that this portion of the design requires a close visualization of what will occur as the fluid enters the unit. Braun[17] suggests flow patterns as shown in Figures 10-21A and 10-21B.

Some exchanger designs require that inlet nozzles be placed close to the tubesheet to obtain the best use of the surface in that immediate area. Fabrication problems limit this dimension. Therefore, internal baffling must be used to force the incoming fluid across the potentially stagnant areas.

g. Longitudinal Baffles. Longitudinal baffles are used on the shell side of a unit to divide the shell-side flow into two or more parts, giving higher velocities for better heat transfer, or to provide a divided area of the bundle for the subcooling of liquid or the cooling of noncondensable vapors as they leave the shell. The baffle must be effectively sealed at the shell to prevent bypassing. Depending upon the shell diameter, the usual sealing methods are (a) welding, (b) sliding slot, and (c) special packing. Figure 10-22 illustrates some of these techniques.

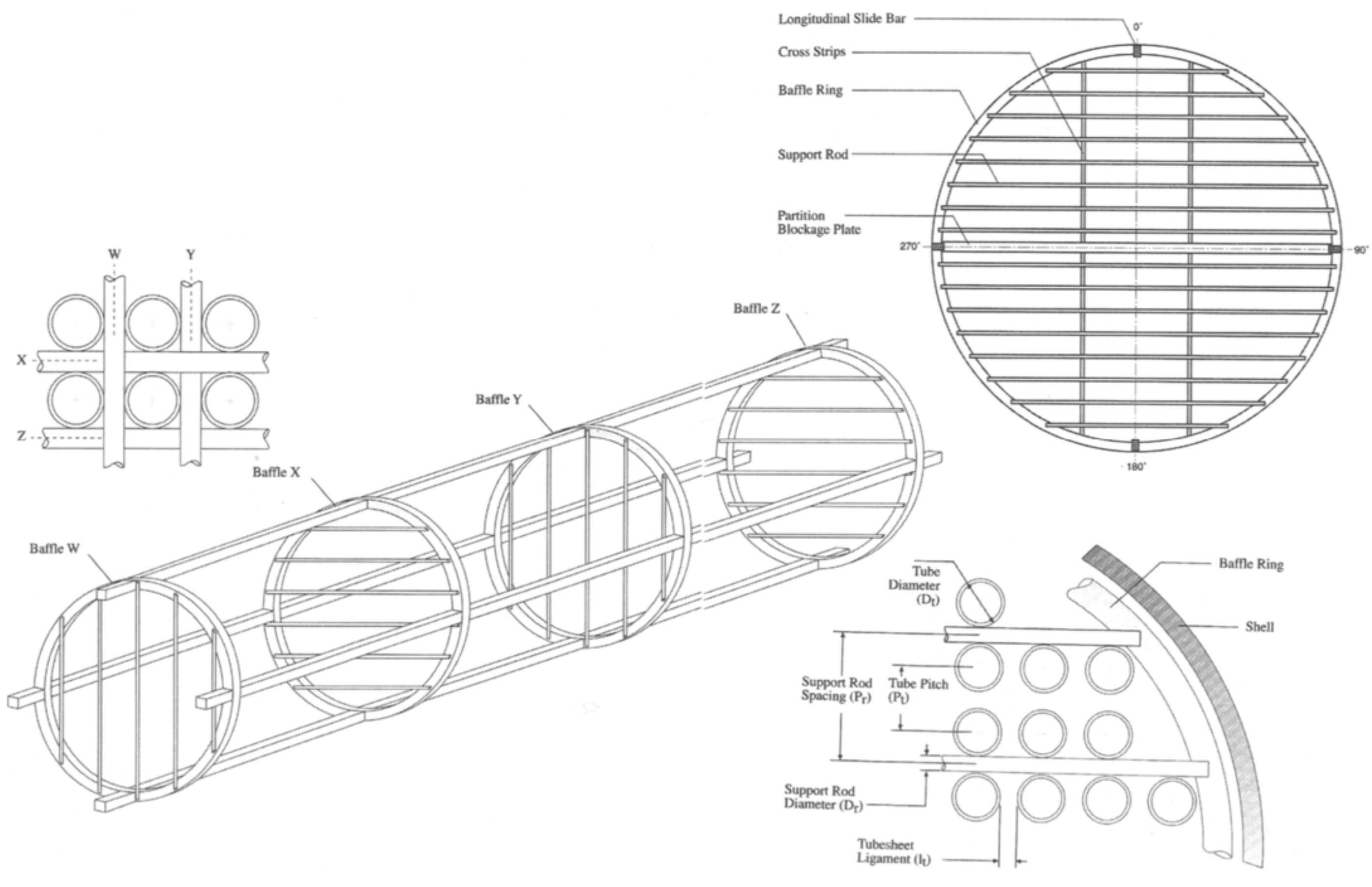

Figure 10-20D. RODbaffle® layout details. Key elements are support rods, circumferential baffle rings, cross-support strips, and longitudinal tie bars. Four different RODbaffle® configurations are used to form a set: baffles W, X, Y, and Z. (Used by permission: © Phillips Petroleum Company, Licensing Div., Bul. 1114–94-A–01.)

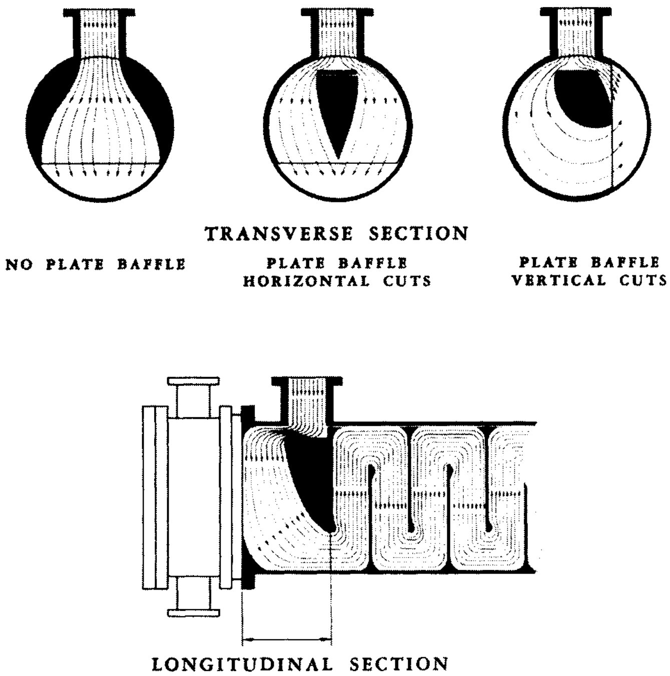

Figure 10-21A. Impingement baffles and fluid-flow patterns. (Used by permission: Brown & Root, Inc.)

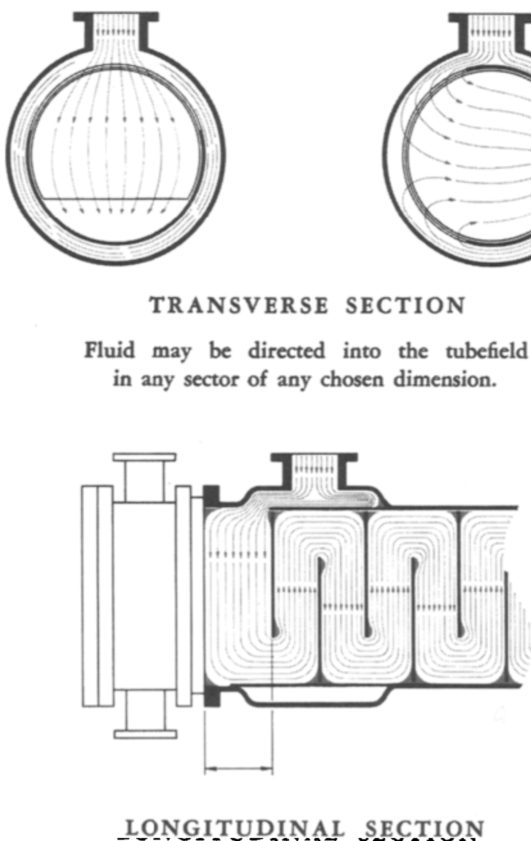

Figure 10-21B. Impingement fluid-flow pattern with annular inlet distributor. (Used by permission: Brown & Root, Inc.)

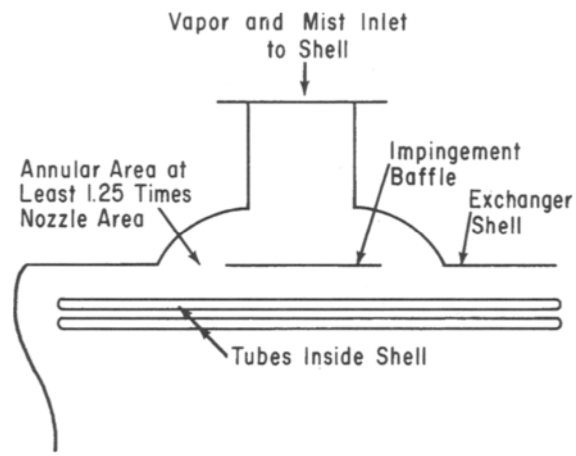

Figure 10-21C. Impingement baffle located in inlet nozzle neck.

Longitudinal baffles must also be compatible with the shell-side fluid, so they normally will be of the same material as tubes or baffles. This baffle never extends the full inside length of the shell, because fluid must flow by its far end for the return pass in reaching the exchanger outlet.

6. Tie Rods

Tie rods with concentric tube spacers are used to space the baffles and tube supports along the tube bundle. The baffles or supports must be held fixed in position because any chattering or vibration with respect to the tubes may wear and eventually destroy the tube at the baffle location. The number of tie rods used depends upon the size and construction of the exchanger bundle. The material of the rods and spacers must be the same or equivalent to that of the baffles or bundle tubes. Provision must be made in the tubesheet layout for these rods, which is usually accomplished by omitting a tube (or more) at selected locations on the outer periphery of the tube bundle. The rod is usually threaded into the back of only one of the tubesheets, being free at the other end, terminating with the last baffle or support by means of lock washers or similar fool-proof fastening. See the upper portion of Figure 10-22 for tie rod spacers.

Table 10-7 shows suggested tie rod count and diameter for various sizes of heat exchangers, as recommended by TEMA[107]. Other combinations of tie rod number and diameter with equivalent metal area are permissible; however, no fewer than four tie rods, and no diameter less than $^3/_8$-in., should be used. Any baffle segment requires a minimum of three points of support.

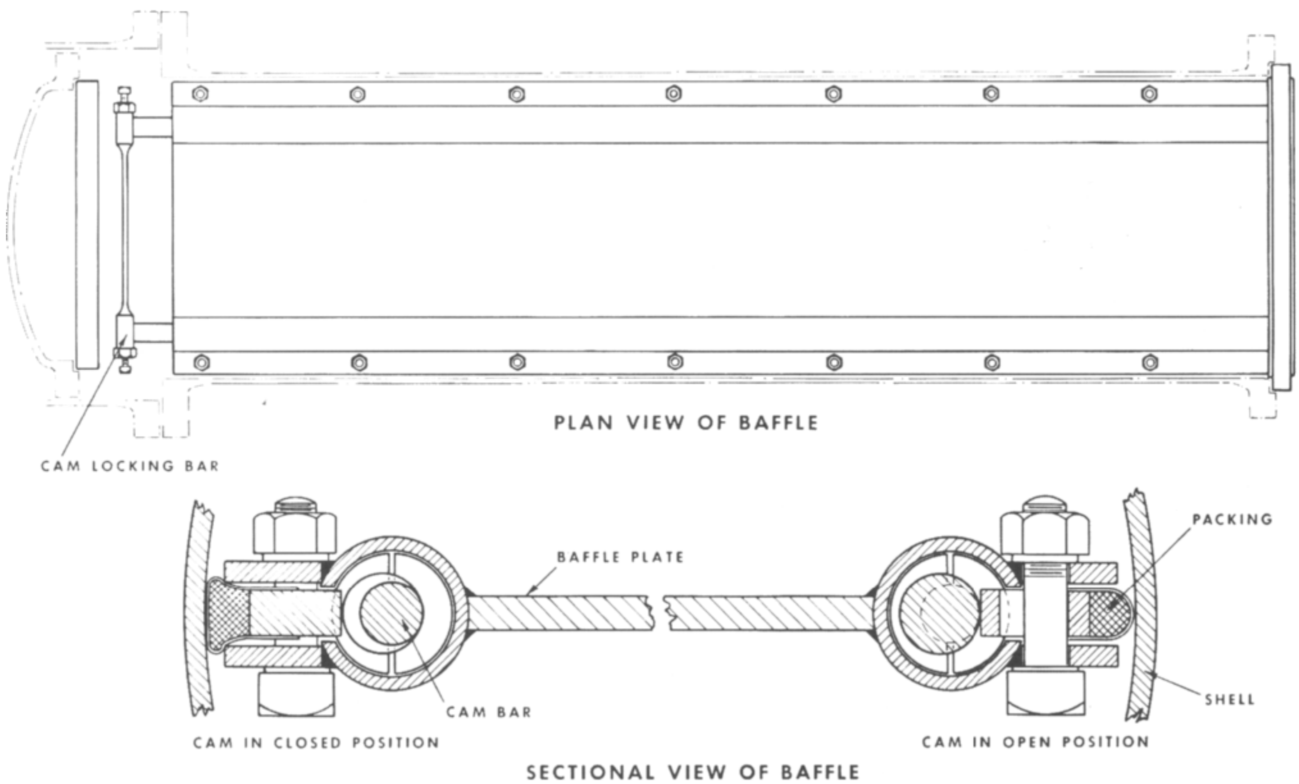

Figure 10-22A. Construction details of two-pass expanding shell-side baffle. (Used by permission: Struthers-Wells Corp., Bul. A-22.)

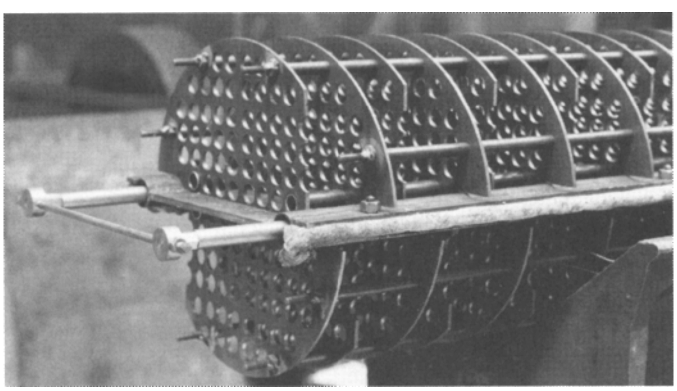

Figure 10-22B. Assembled two-pass shell baffle for installation in shell of exchanger. (Used by permission: Struthers-Wells Corp. Bul. A-22.).

Table 10-7
Tie Rod Standards
(All Dimensions in In.)

Nominal Shell Diameter	Tie Rod Diameter	Minimum Number of Tie Rods
6–15	$^3/_8$	4
16–27	$^3/_8$	6
28–33	$^1/_2$	6
34–48	$^1/_2$	8
49–60	$^1/_2$	10

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7. Tubesheets

Tubesheets form the end barriers to separate the shell-side and tube-side fluids. Most exchangers use single plates for tubesheets. However, for hazardous or corrosive materials such as chlorine, hydrogen chloride, sulfur dioxide, etc., where the intermixing due to leakage from shell- to tube-side or vice versa would present a serious problem, the double tubesheet is used as shown in Figure 10-23. This is considerably more expensive for fabrication, not only due to the plate costs, but also to the extra grooving of these sheets and rolling of the tubes into them. Because they must be aligned true, the machining must be carefully handled; otherwise assembly of the unit will be troublesome.

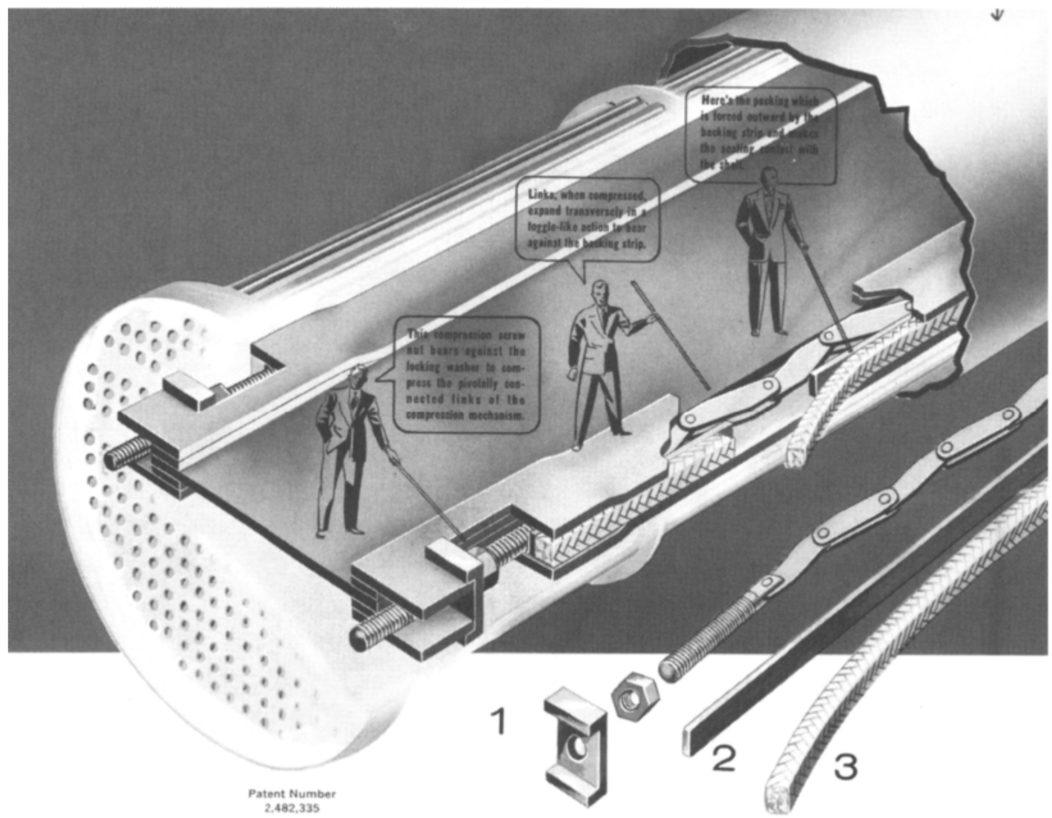

Figure 10-22C. Longitudinal shell-pass baffle. (Used by permission: Henry Vogt Machine Co., Patent No. 2,482,335.)

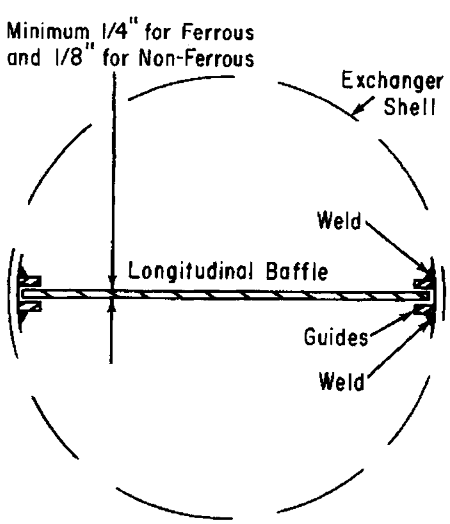

Figure 10-22D. Longitudinal baffle, sliding slot detail.

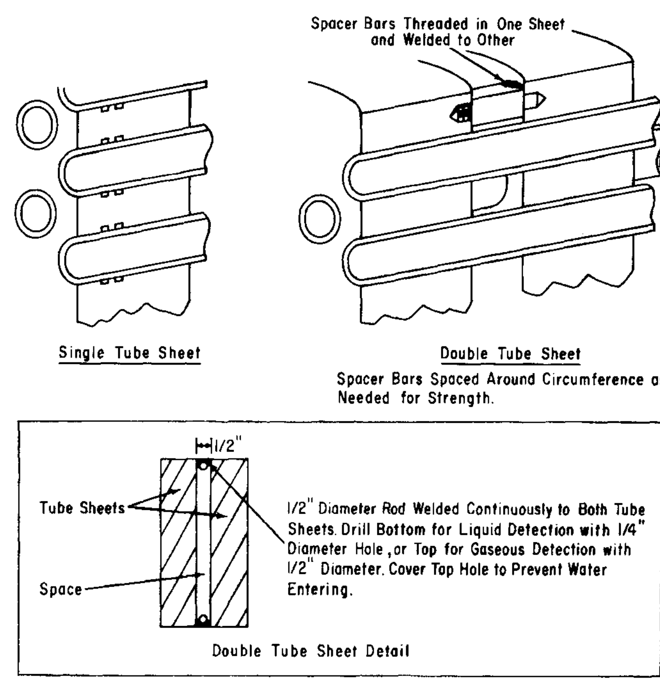

Figure 10-23. Tube-to-double tubesheet assembly detail.

8. Tube Joints in Tubesheets

The quality of the connection between the tube and tubesheet is extremely important. A poor joint here means leakage of shell-side fluid into the tube side or vice versa. This joint can be one of several designs, depending upon the service and type of exchanger. In general, it is good to standardize on some type of grooved joint as compared to the less expensive plain joint. In Figures 10-23 and 10-24, these joints are indicated, as well as special types for the duplex-type tube. The plain joint is used in low-pressure services where the differential pressure across the tubesheet is 5–50 psi, and the differential expansion of tubes with respect to shell is very low, as gaged by a rule of thumb. The maximum temperature differential anywhere in the unit between fluids is not more than 200°F for steel or copper alloy construction.

The serrated and grooved joints are used for high-pressure differentials but usually not in services exceeding 200°F as a rule. Actually these joints will withstand more than twice the push or pull on the tube as a plain joint. The serrations or grooves provide points of strength and effect a better seal against fluid leakage.

The welded joint is used only for high system pressures above 1,000 psig, or high temperatures greater than 300°F, where the properties of the fluid make it impossible to hold a seal with grooved or serrated joints due to temperature stresses or where extra precautions must be taken against cross-contamination of the fluids. If a weld is used, it must be considered as the only sealing and strength part of the connection, because tubes cannot be safely rolled into the tubesheet after welding for fear of cracking a weld. The rolls made prior to welding are usually separated by the heat of the welding operation. This means that the weld cannot be a seal weld, but must truly be a strength weld and so designed. Tubes to be welded into the tubesheet should be spaced farther apart to allow for the weld, without the welds of adjacent tubes touching. The details will depend upon the materials of construction.

Tubes may be inserted into a tubesheet, and packing may be added between them and the tubesheet. A threaded ferrule is inserted to tighten the packing. This type of joint is used only for special expansion problems.

If conditions are such as to require a duplex tube, it is quite likely that a plain end detail for the tube will not be satisfactory. Grooved or serrated joints are recommended for this type of tube, and the ends should be flared or beaded. Table 10-8 gives recommended flare or bell radii for copper-based alloys. Also see Table 10-8A. In service where galvanic corrosion or other corrosive action may take place on the outside material used in the tube, a ferrule of inside tube

Table 10-8
Recommended Diameter of Flared Inlet Holes in Tubesheets for Copper and Copper Alloys

O.D. of Tube, In.	Flare Diameter, In.	Radius of Flare, In.	Tangent Point to Tubesheet, In.
$^1/_2$	0.60	0.38	0.21
$^5/_8$	0.75	0.47	0.26
$^3/_4$	0.90	0.56	0.31
1	1.20	0.75	0.42

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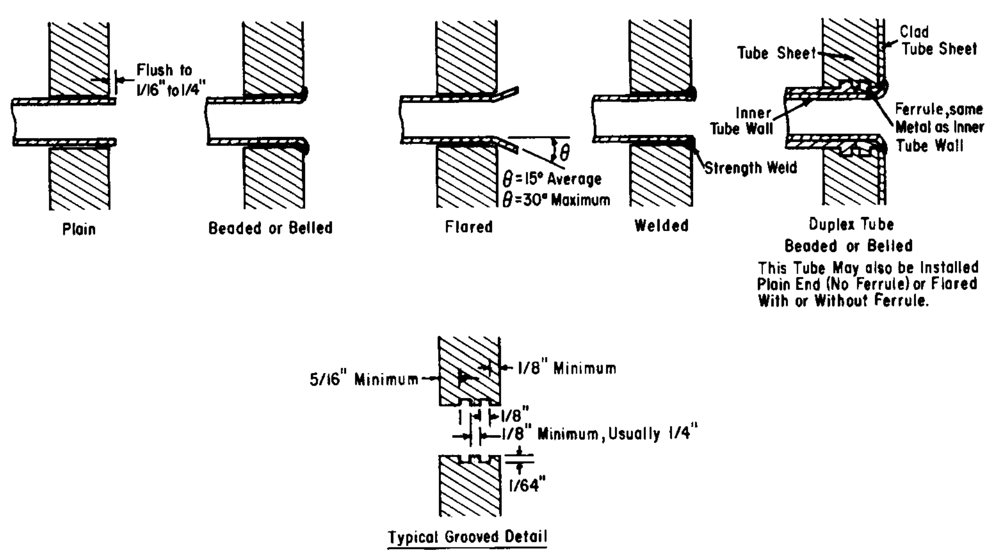

Figure 10-24. Tube to tubesheet joint details.

Table 10-8A
TEMA Standard Tube Hole Diameters and Tolerences
(All Dimensions in In.)

	Nominal Tube Hole Diameter and Under Tolerance				Over Tolerance: 96% of tube holes must meet value in column (c). Remainder may not exceed value in column (d).	
	Standard Fit (a)		Special Close Fit (b)			
Nominal Tube O.D.	Nominal Diameter	Under Tolerance	Nominal Diameter	Under Tolerance	(c)	(d)
$^1/_4$	0.259	0.004	0.257	0.002	0.002	0.007
$^3/_8$	0.384	0.004	0.382	0.002	0.002	0.007
$^1/_2$	0.510	0.004	0.508	0.002	0.002	0.008
$^5/_8$	0.635	0.004	0.633	0.002	0.002	0.010
$^3/_4$	0.760	0.004	0.758	0.002	0.002	0.010
$^7/_8$	0.885	0.004	0.883	0.002	0.002	0.010
1	1.012	0.004	1.010	0.002	0.002	0.010
$1^1/_4$	1.264	0.006	1.261	0.003	0.003	0.010
$1^1/_2$	1.518	0.007	1.514	0.003	0.003	0.010
2	2.022	0.007	2.018	0.003	0.003	0.010

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material should be used on the outside in the tubesheet only to avoid this contact, as shown in Figure 10-24. As an added sealing feature, the end of the duplex tube may be beaded over to seal against surface tension effects.

As a caution, the rolling of tubes into their tubesheets is a very special job that requires experience and "feel," even though today there are electronically controlled rolling and expanding tools. The tubes must be just right, not over nor under expanded, to give a good joint and seal.

Example 10-1. Determine Outside Heat Transfer Area of Heat Exchanger Bundle

To determine the outside heat transfer area of a heat exchanger bundle consisting of 100 tubes, $^3/_4$ in. O.D. tubing, 18 BWG (gauge thickness) × 16 ft long. For fixed tubesheets (2), thickness is 1.0 in. each.

From Table 10-3, read:

External surface area/foot length for these tubes = 0.1963 ft^2.

Note: $^1/_8$ = projection of tubes past exterior face of two tube sheets

Total external tube surface for this bundle:

$$\text{Interior face-to-face of the two tube sheets} = 16\text{ ft} - 2^1/_4\text{ in.} = 15\text{ ft, } 9.75\text{ in.}$$

$$\text{Net tube surface per tube} = (15.8125\text{ ft length net})(0.1963\text{ ft}^2/\text{ft}) = 3.1039\text{ ft}^2/\text{tube.}$$

For 100 tubes, total heat exchanger NET outside tube surface area:

$$=(100)(3.1039) = 310.39\text{ ft}^2$$

Tubesheet Layouts

The layout of the heat exchanger tubesheet determines the number of tubes of a selected size and pitch that will fit into a given diameter of shell. The number of tubes that will fit the shell varies depending upon the number of tube-side passes and even upon whether there is a shell-side pass baffle that divides the shell itself into two or more parts.

The usual tube sizes for most exchangers are $^3/_4$-in. O.D. and 1-in. O.D. The $^5/_8$-in. and $^1/_2$-in. O.D. tubes are used in package exchangers with refrigeration and other systems. However, they present problems in both internal and external cleaning as well as fabrication. Tubes of 1 $^1/_4$ in. and 1 $^1/_2$ in. O.D. and sometimes larger are used in boilers, evaporators, reboilers, and special designs. Tubes of 3 in., 3 $^1/_2$ in. and 4 in. are used in direct fired furnaces and a few special process exchanger designs.

Tube Counts in Shells

Although there are several relations for numerically calculating the number of tubes in a shell, the counts presented in Table 10-9 have been carefully prepared.

Errors in tube count can cause recalculation in expected exchanger performances.[125]

Number of tubes/shell:

(1) Triangular pitch

$$N_t = \frac{[(D_s - K_1)^2\pi/4 + K_2] - p\,(D_s - K_1)[K_3(n) + K_4]}{1.223(p)^2} \qquad (10\text{-}2)$$

(2) Square pitch

$$N_t = \frac{[(D_s - K_1)^2\pi/4 + K_2] - p\,(D_s - K_1)[K_3(n) + K_4]}{(p)^2} \qquad (10\text{-}3)$$

where

N_t = Number tubes in shell
D_s = Inside diameter of shell, in.
p = Tube pitch, in.
n = Number of tube passes
K_1, K_2, K_3, K_4 = Constants depending on the tube size and layout. Use the following table.

Table of K Values

Tube Size In.	Arrangement	Pitch In.	K_1	K_2	K_3	K_4
$^3/_4$	Triangular	$^{15}/_{16}$	1.080	−0.900	0.690	−0.800
$^3/_4$	Triangular	1	1.080	−0.900	0.690	−0.800
$^3/_4$	Square	1	−1.040	−0.100	0.430	−0.250
1	Triangular	1 $^1/_4$	1.080	−0.900	0.690	−0.800
1	Square	1 $^1/_4$	−1.040	−0.100	0.430	−0.250

The tube layouts given in Figures 10-25A–K are samples of the convenient form for modifying standard layouts to fit special needs, such as the removal of certain tubes.

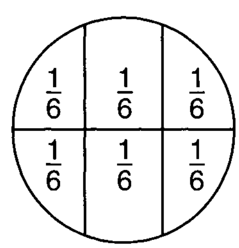
Figures 10-25A–E are for U-bundle tubes and require a wide blank space across the center of the tubesheet to recognize the U-bend requirement at the far end of the tube bundle, see Figures 10-1E, 10-1A item U, and 10-1K. Figures 10-25F–K are for fixed tubesheet layouts.

Before fabrication, an exact layout of the tubes, clearances, etc., must be made; however, for most design purposes, the tube counts for fixed tubesheets and floating heads as given in Tables 10-9 and 10-10 are quite accurate. A comparison with tube counts in other references (19, 70) indicates an agreement of ± 3% in the small diameters up to about 23 $^1/_4$ -in. I.D. shell and graduating up to about ± 10% for the larger shells. The counts as presented have checked manufacturers' shop layouts ± one tube for 8-in. to 17 $^1/_4$ -in. I.D. shell; ± 5 tubes for 21 $^1/_4$ -in. to 27-in. I.D.; and ± 10–20 tubes for the larger shells. No allowances for impingement baffles are made in these layouts, although channel and head baffle lanes have been considered. A standard manufacturing tolerance of $-^3/_8$ in. has been maintained between the specified inside diameter of the shell to the nearest point on any tube (tube clearance).

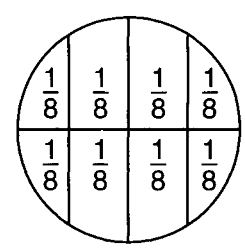
Tube spacing arrangements are shown in Figure 10-26 for the usual designs. Countless special configurations exist for special purposes, such as wide pitch dimensions to give larger ligaments to provide access for cleaning tools to clean scale and fouling films from the outside of tubes by mechanical means. Often chemical cleaning is satisfactory, and wide lanes are not justified. Wide spaces also give low pressure drops but require special care to avoid low transfer coefficients, or at least these conditions should be recognized. Special directional tube lanes, as in steam surface condensers for power plants, allow the fluid to penetrate the large bundle and, thereby, give good access to the surface.

Method of Figuring Tube Counts—Use with Table 10-9

A. Fixed Tubes: All pitches and tube sizes
Pass: One: Straight through
Two: Half-circle per pass
Four: 15$^1/_4$ in. shell I.D. and smaller, used pie shape baffle layout
17$^1/_4$ in. shell I.D. and larger, used ribbon baffle layout
Six: Ribbon baffle layout for all shell I.D.
Eight: Ribbon baffle layout for all shell I.D.



Six Pass



Eight Pass

B. U-Tubes: Radius of Bend = 2 $^1/_2$ times tube O.D. for all pitches and tube sizes
Pass: Two: Pie shape layout
Four: Pie shape layout
Six: Vertical baffle layout
Eight: Vertical baffle layout

C. Allowances: For tie rods:
8–13 $^1/_4$ in. shell I.D., removed 4 tubes
15 $^1/_4$–29 in. shell I.D., removed 6 tubes
31–37 in. shell I.D., removed 8 tubes

Table 10-9
Heat Exchanger Tubesheet Layout Tube Count Table

Note the right column for tubesheet and number of passes per configuration.

37	35	33	31	29	27	25	23 1/4	21 1/4	19 1/4	17 1/4	15 1/4	13 1/4	12	10	8	I.D. of Shell (in.)		
1,269	1,143	1,019	881	763	663	553	481	391	307	247	193	135	105	69	33	3/4 in. on 15/16 in.Δ	Fixed Tubes	One-Pass
1,127	1,007	889	765	667	577	493	423	343	277	217	157	117	91	57	33	3/4 in. on 1 in.Δ		
965	865	765	665	587	495	419	355	287	235	183	139	101	85	53	33	3/4 in. on 1 in.□		
699	633	551	481	427	361	307	247	205	163	133	103	73	57	33	15	1 in. on 1 1/4 in.Δ		
595	545	477	413	359	303	255	215	179	139	111	83	65	45	33	17	1 in. on 1 1/4 in. □		
1,242	1,088	964	846	734	626	528	452	370	300	228	166	124	94	58	32	3/4 in. on 15/16 in.Δ	Fixed Tubes	Two-Pass
1,088	972	858	746	646	556	468	398	326	264	208	154	110	90	56	28	3/4 in. on 1 in.Δ		
946	840	746	644	560	486	408	346	280	222	172	126	94	78	48	26	3/4 in. on 1 in.□		
688	608	530	462	410	346	292	244	204	162	126	92	62	52	32	16	1 in. on 1 1/4 in.Δ		
584	522	460	402	348	298	248	218	172	136	106	76	56	40	26	12	1 in. on 1 1/4 in.□		
1,126	1,008	882	768	648	558	460	398	304	234	180	134	94	64	34	8	3/4 in. on 15/16 in.Δ	U Tubes[2]	
1,000	882	772	674	566	484	406	336	270	212	158	108	72	60	26	8	3/4 in. on 1 in.Δ		
884	778	688	586	506	436	362	304	242	188	142	100	72	52	30	12	3/4 in. on 1 in.□		
610	532	466	396	340	284	234	192	154	120	84	58	42	26	8	XX	1 in. on 1 1/4 in.Δ		
526	464	406	356	304	256	214	180	134	100	76	58	38	22	12	XX	1 in. on 1 1/4 in. □		
1,072	1,024	904	788	680	576	484	412	332	266	196	154	108	84	48	XX	3/4 in. on 15/16 in.Δ	Fixed Tubes	Four-Pass
1,024	912	802	692	596	508	424	360	292	232	180	134	96	72	44	XX	3/4 in. on 1 in.Δ		
880	778	688	590	510	440	366	308	242	192	142	126	88	72	48	XX	3/4 in. on 1 in.□		
638	560	486	422	368	308	258	212	176	138	104	78	60	44	24	XX	1 in. on 1 1/4 in.Δ		
534	476	414	360	310	260	214	188	142	110	84	74	48	40	24	XX	1 in. on 1 1/4 in.□		
1,092	976	852	740	622	534	438	378	286	218	166	122	84	56	28	XX	3/4 in. on 15/16 in.Δ	U Tubes[2]	
968	852	744	648	542	462	386	318	254	198	146	98	64	52	20	XX	3/4 in. on 1 in.Δ		
852	748	660	560	482	414	342	286	226	174	130	90	64	44	24	XX	3/4 in. on 1 in.□		
584	508	444	376	322	266	218	178	142	110	74	50	36	20	XX	XX	1 in. on 1 1/4 in.Δ		
500	440	384	336	286	238	198	166	122	90	66	50	32	16	XX	XX	1 in. on 1 1/4 in.□		
1,106	964	844	732	632	532	440	372	294	230	174	116	80	XX	XX	XX	3/4 in. on 15/16 in.Δ	Fixed Tubes	Six-Pass
964	852	744	640	548	464	388	322	258	202	156	104	66	XX	XX	XX	3/4 in. on 1 in.Δ		
818	224	634	536	460	394	324	266	212	158	116	78	54	XX	XX	XX	3/4 in. on 1 in.□		
586	514	442	382	338	274	226	182	150	112	82	56	34	XX	XX	XX	1 in. on 1 1/4 in.Δ		
484	430	368	318	268	226	184	154	116	88	66	44	XX	XX	XX	XX	1 in. on 1 1/4 in.□		
1,058	944	826	716	596	510	416	358	272	206	156	110	74	XX	XX	XX	3/4 in. on 15/16 in.Δ	U Tubes[2]	
940	826	720	626	518	440	366	300	238	184	134	88	56	XX	XX	XX	3/4 in. on 1 in.Δ		
820	718	632	534	458	392	322	268	210	160	118	80	56	XX	XX	XX	3/4 in. on 1 in.□		
562	488	426	356	304	252	206	168	130	100	68	42	30	XX	XX	XX	1 in. on 1 1/4 in.Δ		
478	420	362	316	268	224	182	152	110	80	60	42	XX	XX	XX	XX	1 in. on 1 1/4 in.□		
1,040	902	790	682	576	484	398	332	258	198	140	94	XX	XX	XX	XX	3/4 in. on 15/16 in.Δ	Fixed Tubes	Eight-Pass
902	798	694	588	496	422	344	286	224	170	124	82	XX	XX	XX	XX	3/4 in. on 1 in.Δ		
760	662	576	490	414	352	286	228	174	132	94	XX	XX	XX	XX	XX	3/4 in. on 1 in.□		
542	466	400	342	298	240	190	154	120	90	66	XX	XX	XX	XX	XX	1 in. on 1 1/4 in.Δ		
438	388	334	280	230	192	150	128	94	74	XX	XX	XX	XX	XX	XX	1 in. on 1 1/4 in.□		
1,032	916	796	688	578	490	398	342	254	190	142	102	68	XX	XX	XX	3/4 in. on 15/16 in.Δ	U Tubes[2]	
908	796	692	600	498	422	350	286	226	170	122	82	52	XX	XX	XX	3/4 in. on 1 in.Δ		
792	692	608	512	438	374	306	254	194	146	106	70	48	XX	XX	XX	3/4 in. on 1 in.□		
540	464	404	340	290	238	190	154	118	90	58	38	24	XX	XX	XX	1 in. on 1 1/4 in.Δ		
456	396	344	300	254	206	170	142	98	70	50	34	XX	XX	XX	XX	1 in. on 1 1/4 in.□		
37	**35**	**33**	**31**	**29**	**27**	**25**	**23 1/4**	**21 1/4**	**19 1/4**	**17 1/4**	**15 1/4**	**13 1/4**	**12**	**10**	**8**	**I.D. of Shell (in.)**		

[1]Allowance made for tie rods.

[2]R.O.B. = 2 1/2 × tube diameter. Actual number of "U" tubes is one-half the figure shown in the table.

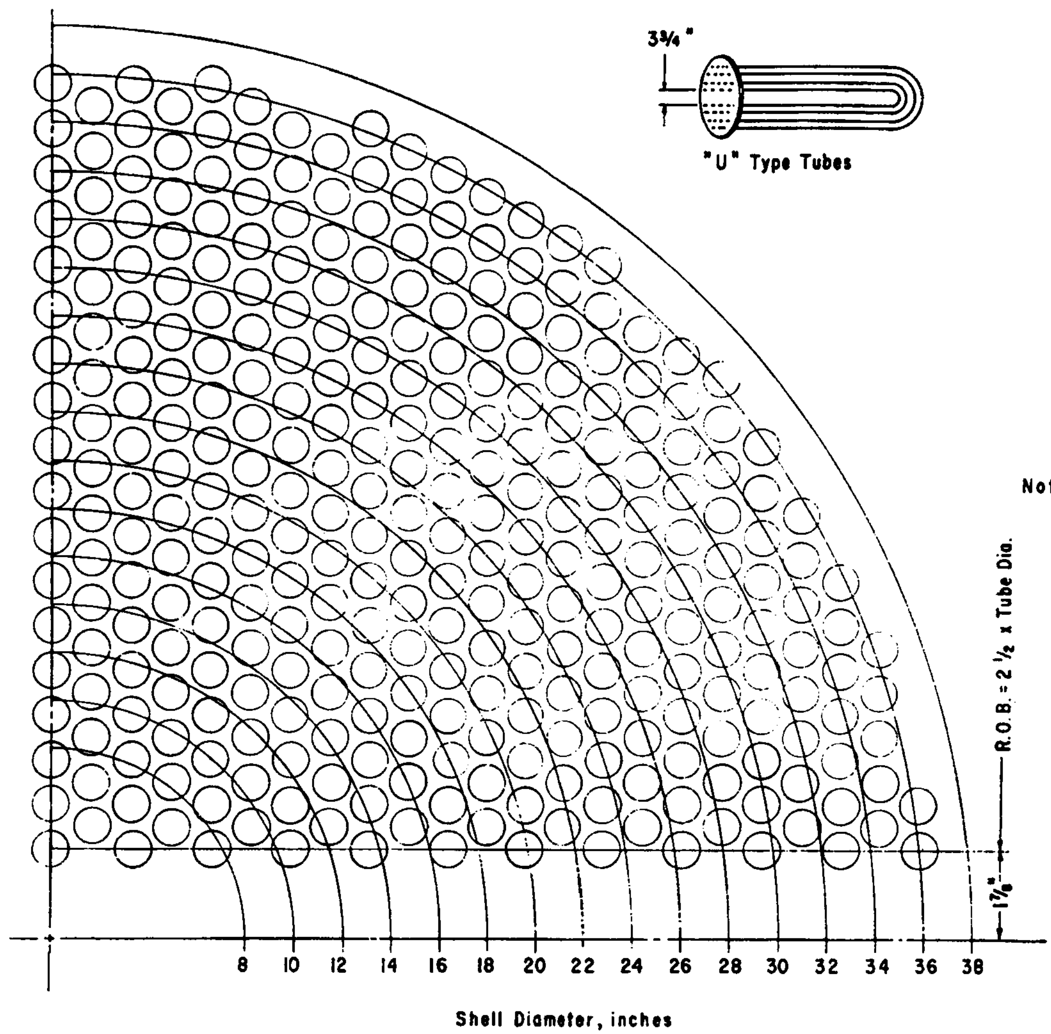
Applications of Tube Pitch Arrangements, Figure 10-26

Triangular Pitch, Apex Vertical or Facing Oncoming Flow. Most popular, generally suitable for nonfouling or fouling services handled by chemical treatment, medium- to high-pressure drop, gives better coefficients than in-line square pitch.

In-Line Triangular Pitch, Apex Horizontal or at Right Angle to Oncoming Flow. Not as popular as the staggered triangular pitch; coefficients not as high, but better than in-line square pitch; pressure drop about medium to high; generally suitable for fouling conditions same as preceding.

In-Line Square Pitch. Popular for conditions requiring low-pressure drop and/or cleaning lanes for mechanical cleaning of outside of tubes; coefficient lower than triangular pitch.

Diamond Square Pitch. Popular arrangement for reasonable low-pressure drop (not as low as in-line square), mechanical cleaning requirements, and better coefficient than in-line square pitch.



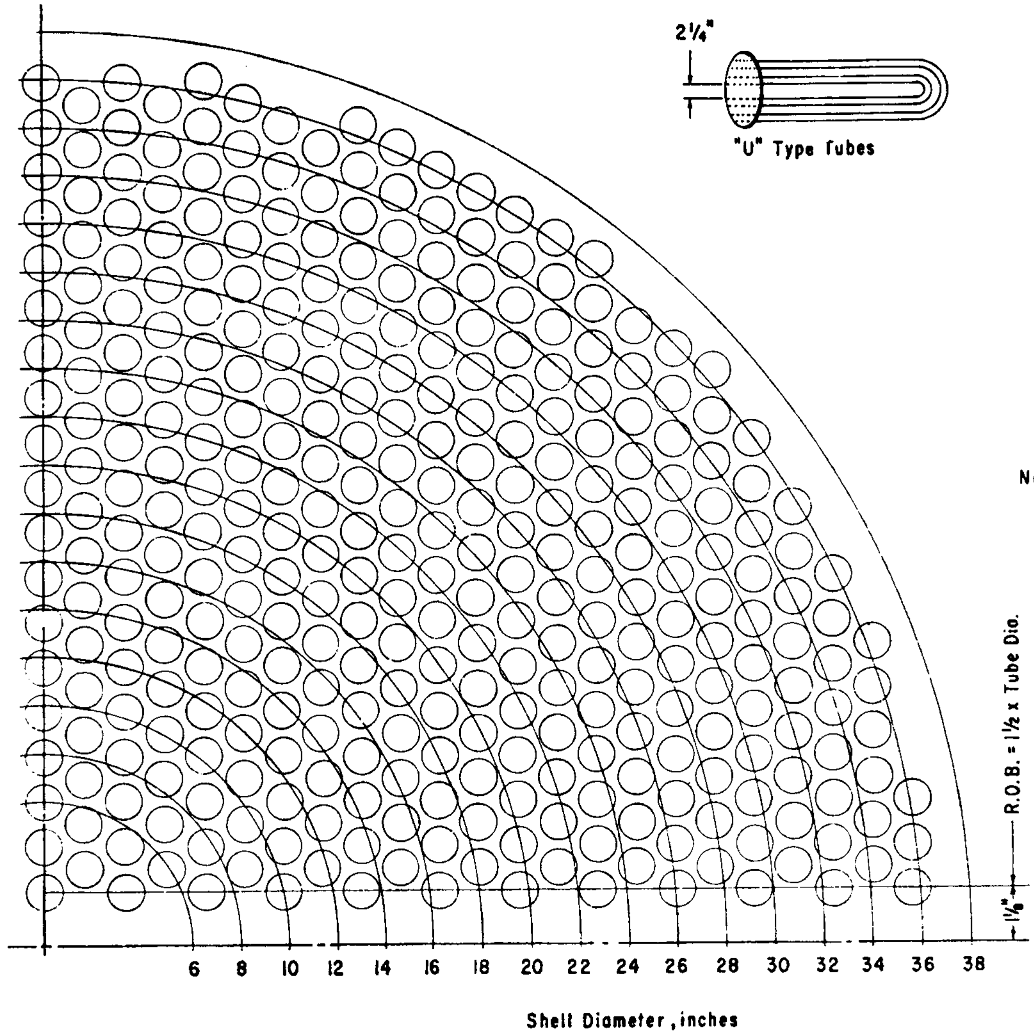
I.D. of Shell	No. of Holes	I.D. of Shell	No. of Holes
8	12	25	466
10	38	26	512
12	68	27	564
13¼	98	28	610
14	110	29	654
15¼	140	30	720
16	152	31	776
17¼	186	32	826
18	210	33	890
19¼	240	34	932
20	276	35	1,016
21¼	310	36	1,082
22	346	37	1,134
23¼	404	38	1,212
24	416		

Note: Total Number of Holes in the Tube Sheet is Based on ⅜" Minimum Clearance Between Shell and Tubes. Actual Number of "U" Tubes is one-half the Above Figures.
For 4 Tube Pass Layout "Count-out" Holes on Vertical Center Line.
Omit Tubes as Required for Tie Rods and Impingement Plates.
When Calculating Surface Area for Bundle, Reduce Tube Lengths to Allow for Shorter Tubes Near Center.

Tube Sheet Layout
U-TUBE EXCHANGER

Tube Passes: Two or Four
Tube Size & Pitch: 3/4" on 5/16"Δ

Figure 10-25A. Tubesheet layout for U-tube exchanger. Tube passes: two or four. Tube sizes and pitch: $^3/_4$ in. on $^{15}/_{16}$ in.Δ. Radius of bend: 2 $^1/_2$ × tube diameter.



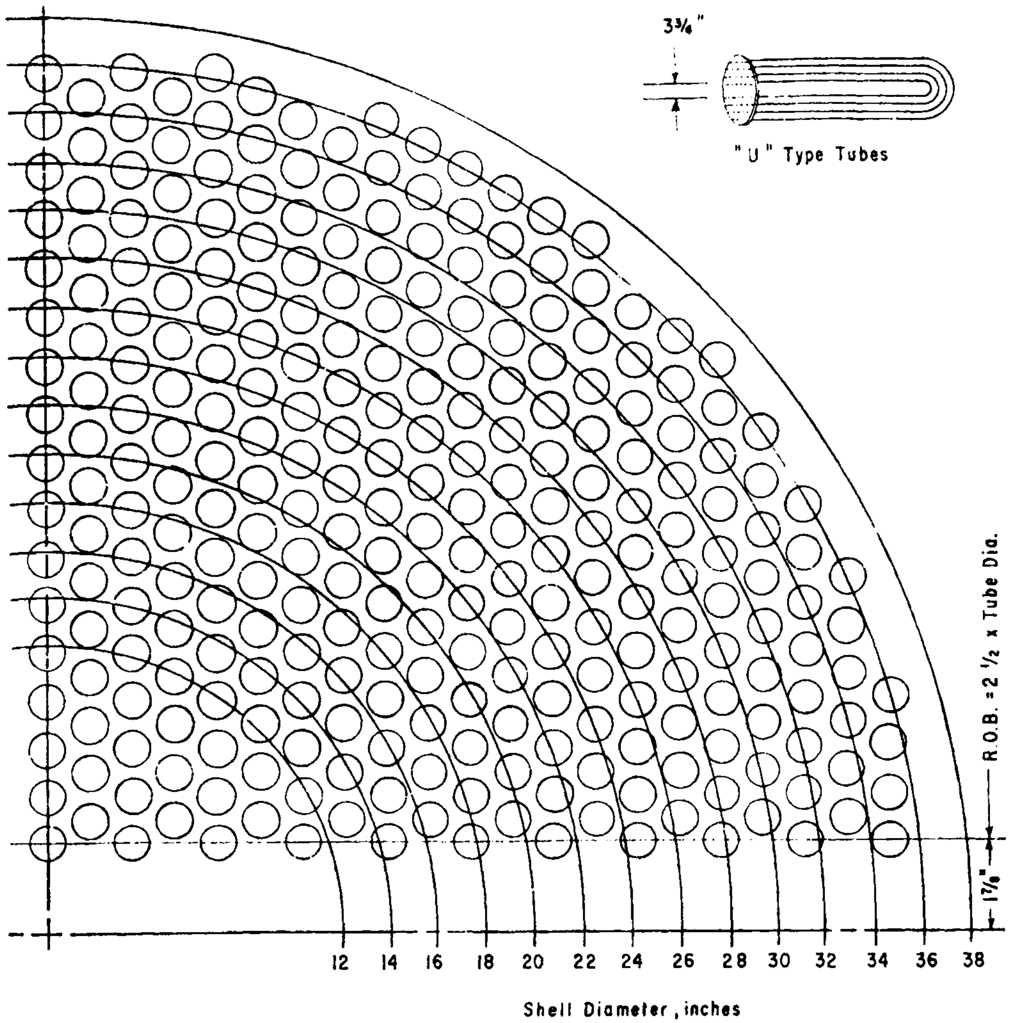
I.D. of Shell	No. of Holes	I.D. of Shell	No. of Holes
8	26	25	516
10	52	26	552
12	90	27	606
13¼	108	28	662
14	132	29	708
15¼	166	30	782
16	186	31	818
17¼	220	32	900
18	240	33	948
19¼	282	34	1,014
20	314	35	1,086
21¼	348	36	1,148
22	392	37	1,212
23¼	434	38	1,294
24	470		

Note: Total Number of Holes in the Tube Sheet is Based on 3/8" Minimum Clearance Between Shell and Tubes. Actual Number of "U" Tubes is one-half the Above Figures.
For 4 Tube Pass Layout "Count-out" Holes on Vertical Center Line.
Omit Tubes as Required for Tie Rods and Impingement Plates.
When Calculating Surface Area for Bundle, Reduce Tube Lengths to Allow for Shorter Tubes Near Center.

Tube Sheet Layout
U-TUBE EXCHANGER

Tube Passes: Two or Four
Tube Size & Pitch: 3/4" on 15/16"Δ

Figure 10-25B. Tubesheet layout for U-tube exchanger. Tube passes: two or four. Tube size and pitch: $^3/_4$ in. on $^{15}/_{16}$ in.Δ. Radius of bend: 1 $^1/_2$ × tube diameter.



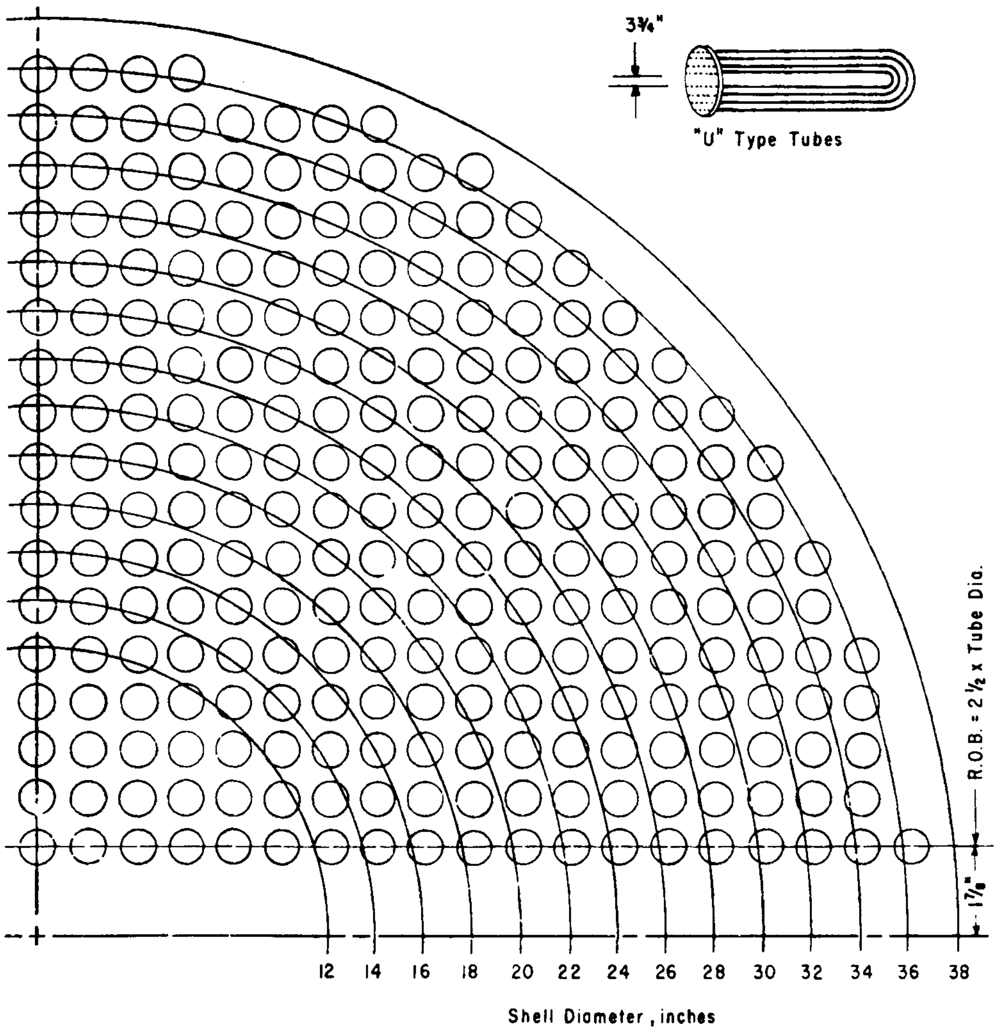
I.D. of Shell	No. of Holes	I.D. of Shell	No. of Holes
12	64	26	458
14	98	28	540
16	140	30	634
18	186	32	728
20	240	34	834
22	306	36	952
24	380	38	1,078

Note: Total Number of Holes in the Tube Sheet is Based on ³/₈" Minimum Clearance Between Shell and Tubes. Actual Number of "U" Tubes is one-half the Above Figures.
For 4 Tube Pass Layout "Count-out" Holes on Vertical Center Line.
Omit Tubes as Required for Tie Rods and Impingement Plates.
When Calculating Surface Area for Bundle, Reduce Tube Lengths to Allow for Shorter Tubes Near Center.

Tube Sheet Layout
U-TUBE EXCHANGER

Tube Passes: Two or Four
Tube Size & Pitch: 3/4" on 1"Δ

Figure 10-25C. Tubesheet layout for U-tube exchanger. Tube passes: two or four. Tube size and pitch: $^3/_4$ in. O.D. on 1 in.Δ. Radius of bend: 2 $^1/_2$ × tube diameter.



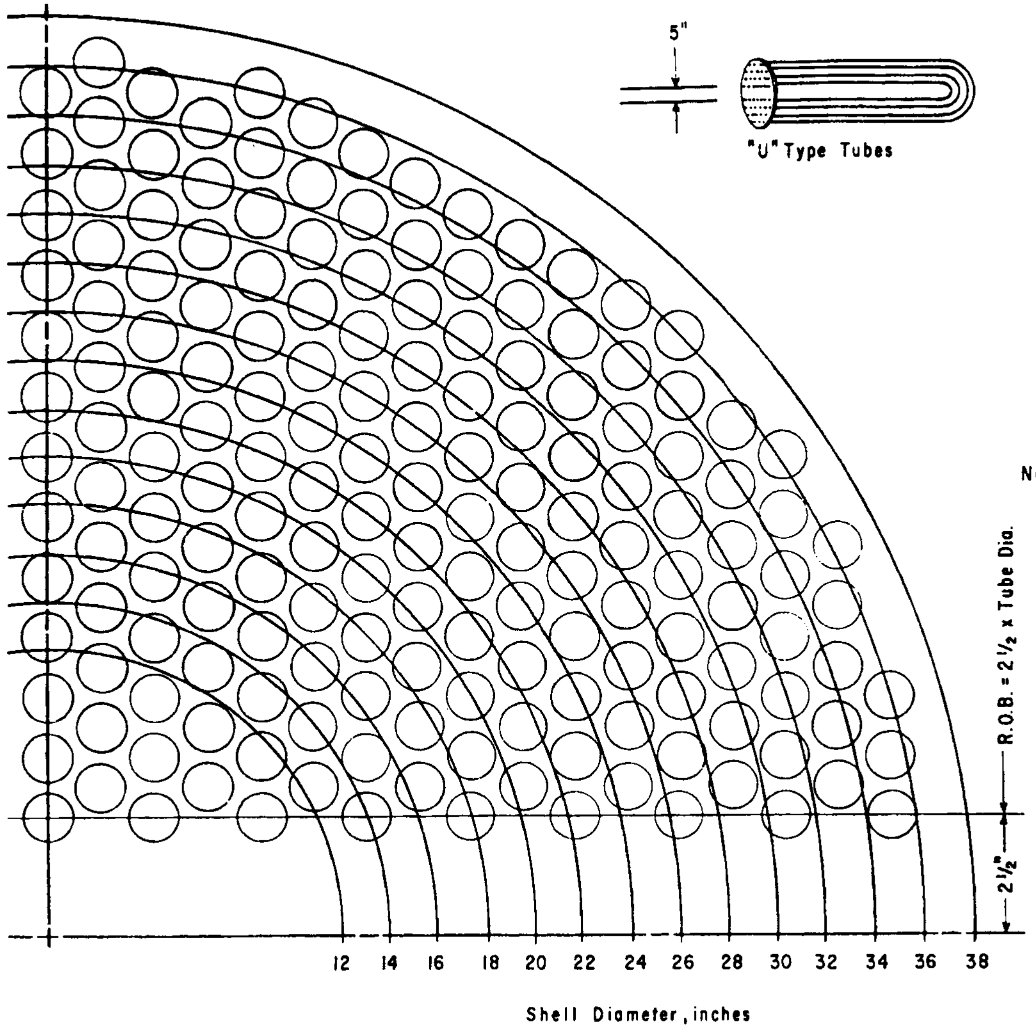
I.D. of Shell	No. of Holes	I.D. of Shell	No. of Holes
12	56	26	410
14	86	28	476
16	128	30	562
18	166	32	652
20	220	34	746
22	278	36	840
24	336	38	912

Note: Total Number of Holes in the Tube Sheet is Based on ³/₈" Minimum Clearance Between Shell and Tubes. Actual Number of "U" Tubes is one-half the Above Figures.
For 4 Tube Pass Layout "Count-out" Holes on Vertical Center Line.
Omit Tubes as Required for Tie Rods and Impingement Plates.
When Calculating Surface Area for Bundle, Reduce Tube Lengths to Allow for Shorter Tubes Near Center.

Tube Sheet Layout
U-TUBE EXCHANGER

Tube Passes: Two or Four
Tube Size & Pitch: 3/4" on 1"Δ

Figure 10-25D. Tubesheet layout for U-tube exchanger. Tube passes: two or four. Size and pitch: $^3/_4$ in. on 1 in.Δ. Radius of bend: 2 $^1/_2$ × tube diameter.



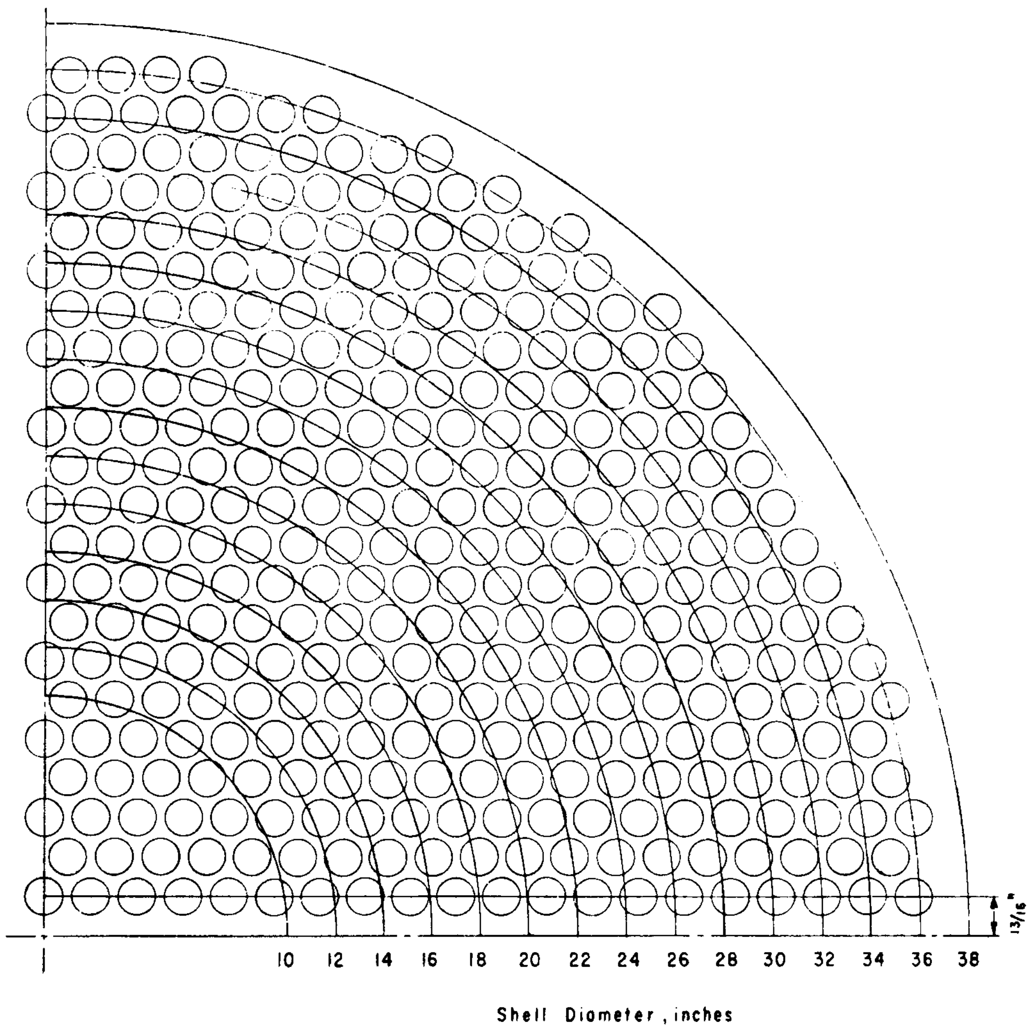
I.D. of Shell	No. of Holes	I.D. of Shell	No. of Holes
12	30	26	268
14	46	28	314
16	76	30	372
18	98	32	438
20	140	34	494
22	174	36	568
24	218	38	654

Note: Total Number of Holes in the Tube Sheet is Based on 3/8" Minimum Clearance Between Shell and Tubes. Actual Number of "U" Tubes is one-half the Above Figures.
For 4 Tube Pass Layout "Count-out" Holes on Vertical Center Line.
Omit Tubes as Required for Tie Rods and Impingement Plates.
When Calculating Surface Area for Bundle, Reduce Tube Lengths to Allow for Shorter Tubes Near Center.

Tube Sheet Layout
U - TUBE EXCHANGER

Tube Passes: Two or Four
Tube Size & Pitch: 1" on 1 1/4" Δ

Figure 10-25E. Tubesheet layout for U-tube exchanger. Tube passes: two or four. Tube size and pitch: 1 in. on 1 $^1/_4$ in.Δ. Radius of bend: 1 $^1/_2$ × tube diameter.

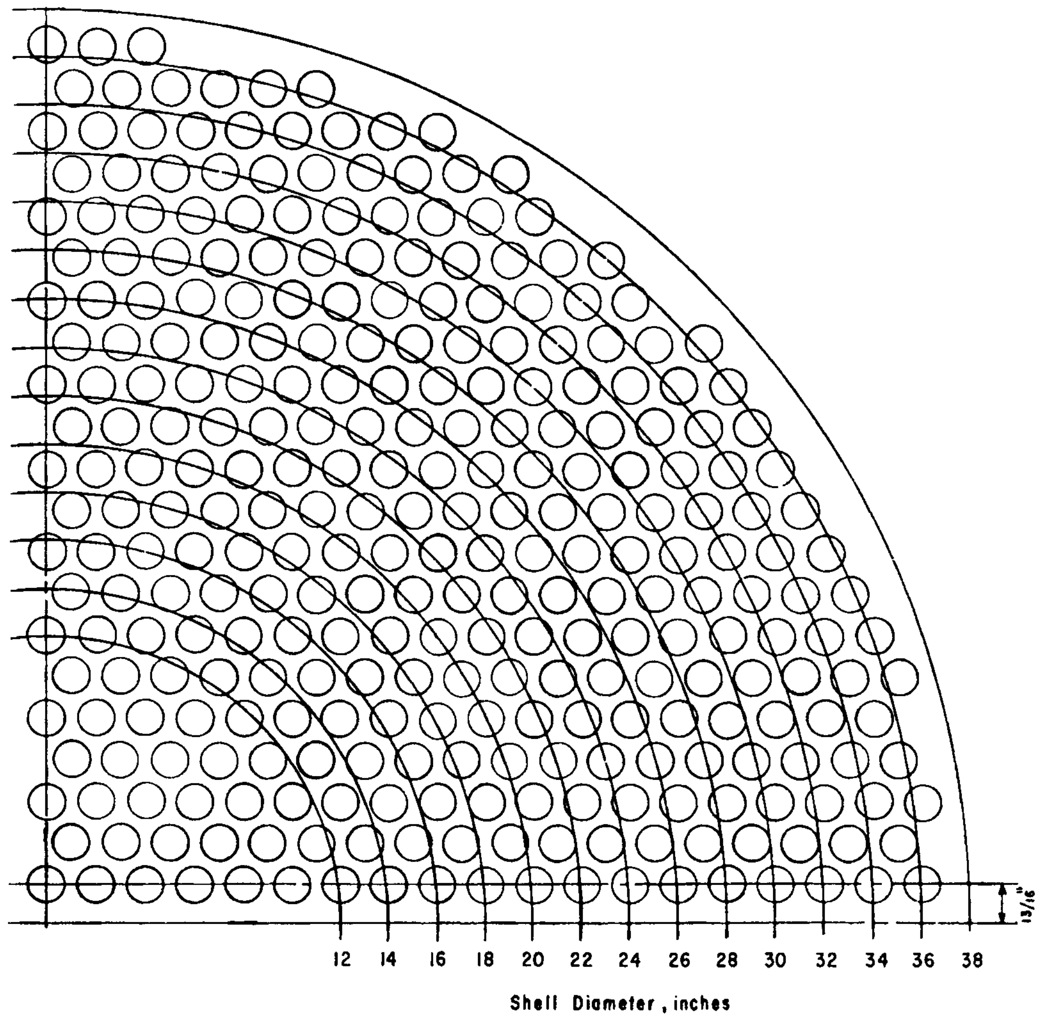


I. D. of Shell	No. of Holes	I. D. of Shell	No. of Holes
8	36	25	534
10	62	26	596
12	98	27	632
13¼	128	28	696
14	148	29	740
15¼	172	30	810
16	204	31	854
17¼	234	32	934
18	266	33	972
19¼	306	34	1,056
20	332	35	1,096
21¼	376	36	1,198
22	420	37	1,250
23¼	458	38	1,326
24	498		

Note : Total Number of Holes in Tube Sheet is Based on 3/8" Minimum Clearance Between Shell and Tubes.
Make Allowances for Tie Rods and Impingement Plates.

Tube Sheet Layout
FIXED TUBE SHEET EXCHANGER
(Also Non-Removable Floating Head)
Tube Passes : Two
Tube Size & Pitch : 3/4" on 15/16" Δ

Figure 10-25F. Fixed tubesheet layout (also nonremovable floating head). Tube passes: two. Tube size and pitch: $^3/_4$ in. on $^{15}/_{16}$ in.Δ.

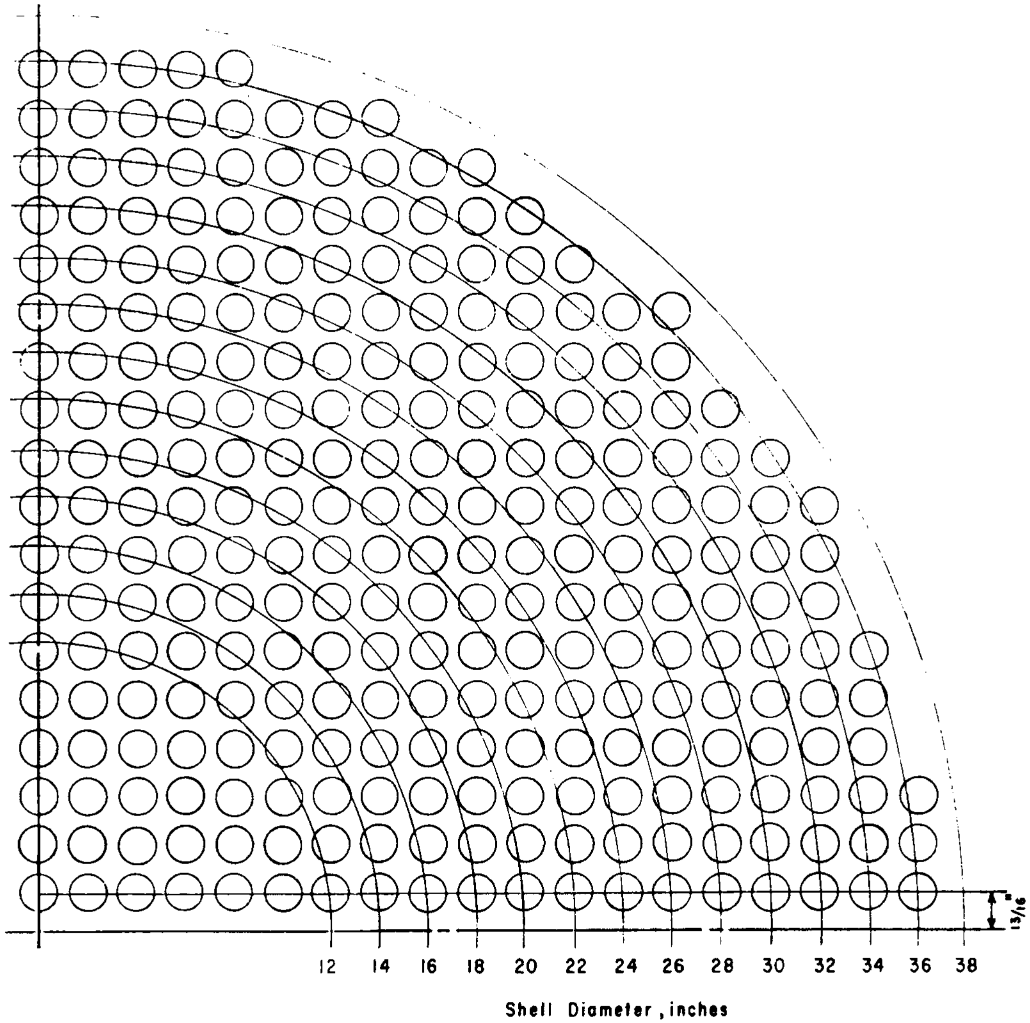


I.D. of Shell	No. of Holes	I.D. of Shell	No. of Holes
12	94	26	558
14	132	28	652
16	176	30	752
18	230	32	862
20	294	34	976
22	364	36	1092
24	438	38	1190

Note : Total Number of Holes in Tube Sheet is Based on 3/8" Minimum Clearance Between Shell and Tubes.
Make Allowances for Tie Rods and Impingement Plates.

Tube Sheet Layout
FIXED TUBE SHEET EXCHANGER
(Also Non-Removable Floating Head)
Tube Passes : Two
Tube Size & Pitch: 3/4" on 1"Δ

Figure 10-25G. Fixed tubesheet layout (also nonremovable floating head). Tube passes: two. Tube size and pitch: $^3/_4$ in. on 1 in.Δ.

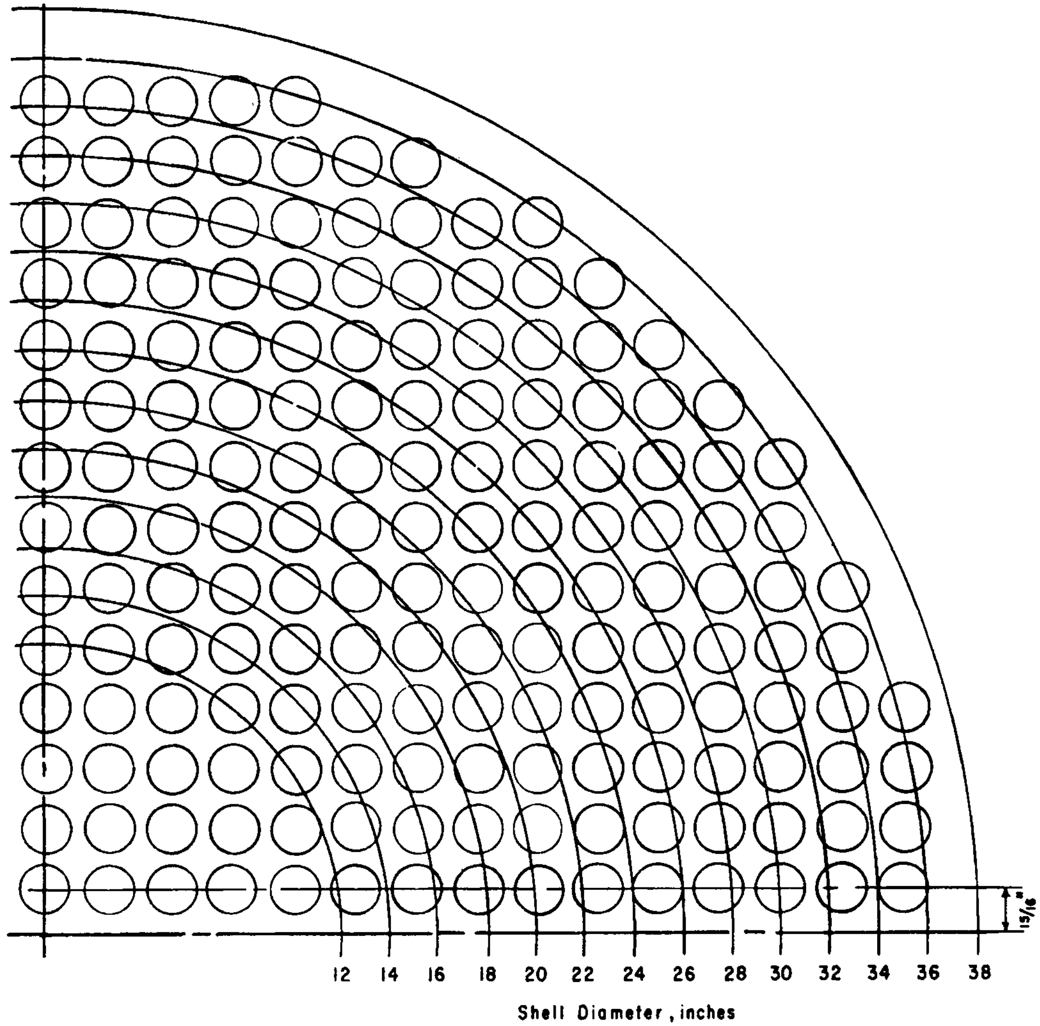


I.D. of Shell	No. of Holes	I.D. of Shell	No. of Holes
12	82	26	460
14	112	28	530
16	162	30	632
18	204	32	718
20	258	34	816
22	324	36	922
24	386	38	1,032

Note : Total Number of Holes in Tube Sheet is Based on 3/8" Minimum Clearance Between Shell and Tubes.
Make Allowances for Tie Rods and Impingement Plates.

Tube Sheet Layout
FIXED TUBE SHEET EXCHANGER
(Also Non-Removable Floating Head)
Tube Passes : Two
Tube Size & Pitch: 3/4" on 1" □

Figure 10-25H. Fixed tubesheet layout (also nonremovable floating head). Tube passes: two. Tube size and pitch: $^3/_4$ in. on 1 in.□.

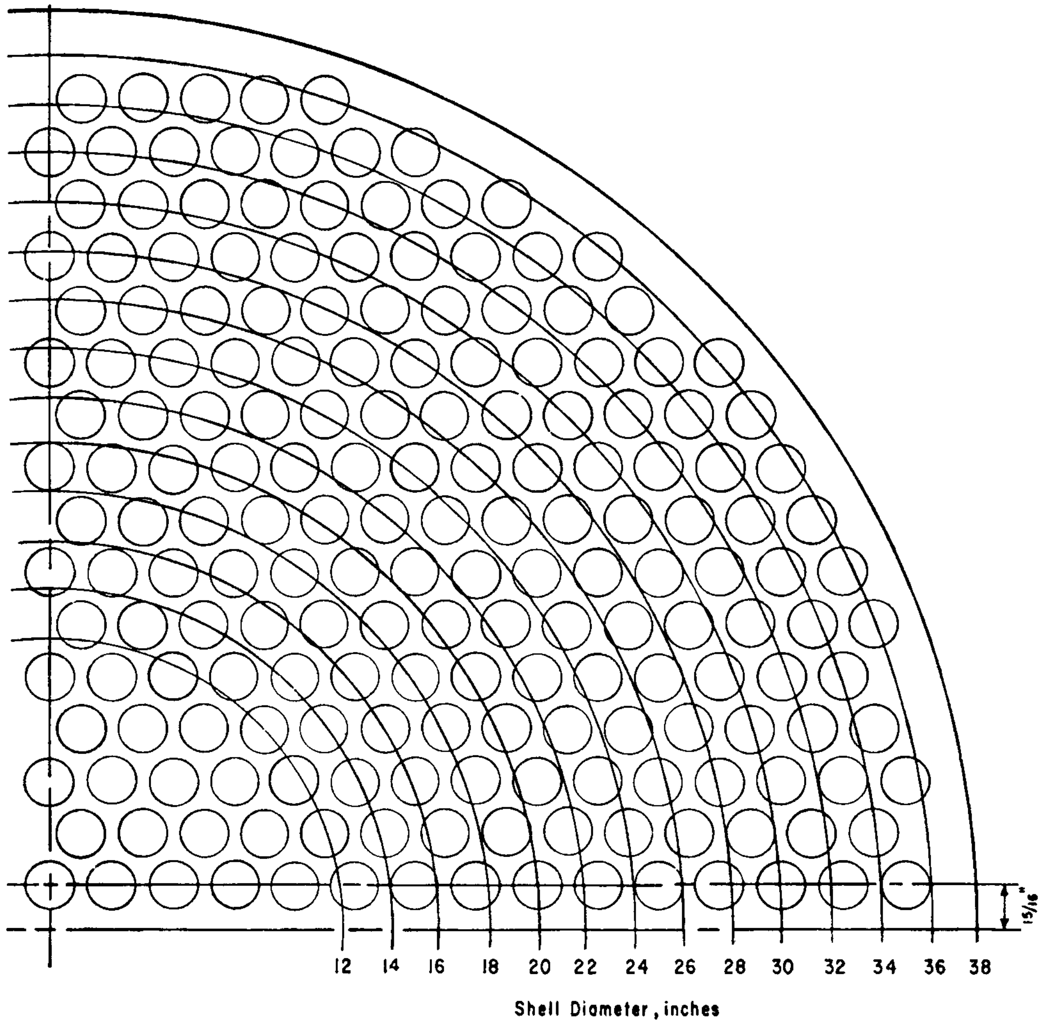


I.D. of Shell	No. of Holes	I.D. of Shell	No. of Holes
12	52	26	280
14	70	28	336
16	90	30	394
18	124	32	452
20	158	34	514
22	196	36	570
24	234	38	640

Note : Total Number of Holes in Tube Sheet is Based on 3/8" Minimum Clearance Between Shell and Tubes.
Make Allowances for Tie Rods and Impingement Plates.

Tube Sheet Layout
FIXED TUBE SHEET EXCHANGER
(Also Non-Removable Floating Head)
Tube Passes : Two
Tube Size & Pitch: 1" on 1 1/4" □

Figure 10-25I. Fixed tubesheet layout (also nonremovable floating head). Tube passes: two. Tube size and pitch: 1 in. on 1 $^1/_4$ in.□.

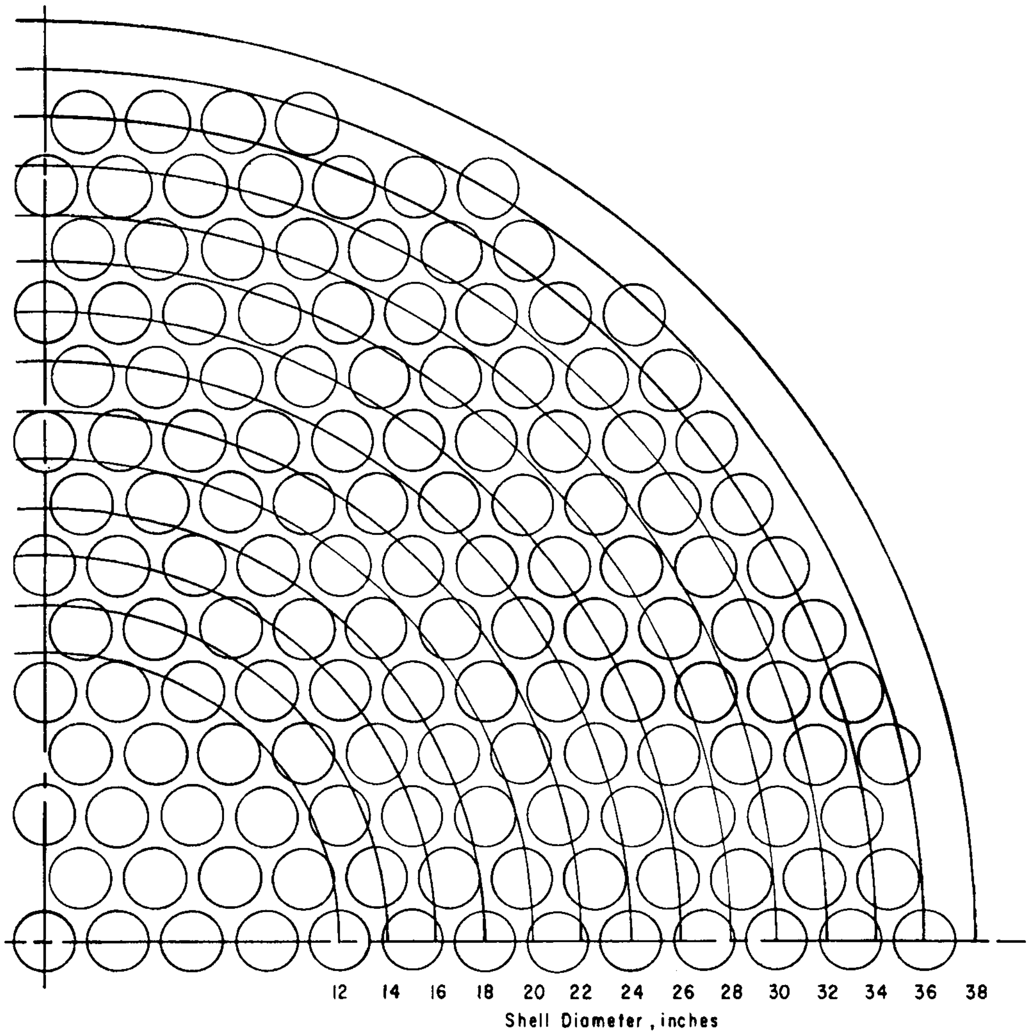


I.D. of Shell	No. of Holes	I.D. of Shell	No. of Holes
12	56	26	328
14	82	28	384
16	110	30	454
18	144	32	518
20	180	34	584
22	222	36	660
24	274	38	744

Note : Total Number of Holes in Tube Sheet is Based on 3/8" Minimum Clearance Between Shell and Tubes.
Make Allowances for Tie Rods and Impingement Plates.

Tube Sheet Layout
FIXED TUBE SHEET EXCHANGER
(Also Non-Removable Floating Head)
Tube Passes : Two
Tube Size & Pitch: 1" on 1 1/4" Δ

Figure 10-25J. Fixed tubesheet layout (also nonremovable floating head). Tube passes: one. Tube size and pitch: 1 in. on 1 $^1/_4$ in.Δ.



I.D. of Shell	No. of Holes	I.D. of Shell	No. of Holes
12	37	26	241
14	61	28	271
16	85	30	313
18	109	32	363
20	123	34	421
22	163	36	463
24	199	38	517

Note : Total Number of Holes in Tube Sheet is Based on 3/8" Minimum Clearance Between Shell and Tubes.
Make Allowances for Tie Rods and Impingement Plates.

Tube Sheet Layout
FIXED TUBE SHEET EXCHANGER
(Also Non-Removable Floating Head)
Tube Passes : One
Tube Size & Pitch : 1 1/4" on 1 1/2"Δ

Figure 10-25K. Fixed tubesheet layout (also nonremovable floating head). Tube passes: one. Tube size and pitch: 1 $^1/_4$ in.Δ.

Table 10-10A
Full Circle Tube Layouts Floating Head Exchanger $^3/_4$-in. O.D. Tubes on $^{15}/_{16}$-in. Triangular Pitch

Size (In.)	Number of Passes 1	2	4	6	8	Net Free Distance 2 Passes	Rows Across
8	42	40	32	32	24	3.75	13
10	73	68	56	54	52	4.63	17
12	109	106	88	86	80	4.00	21
14	130	124	110	108	104	4.50	25
16	187	176	162	152	144	5.00	29
18	241	232	214	216	204	5.88	33
20	308	302	282	274	264	6.50	37
22	384	372	352	348	336	7.13	41
24	472	458	432	420	406	7.75	45
26	555	538	510	510	502	8.63	48
28	649	636	610	606	580	8.13	53
30	764	744	716	708	700	8.75	57
32	868	850	822	812	796	9.38	63
34	994	970	930	928	912	9.75	65
36	1131	1108	1066	1058	1028	10.50	71
38	1268	1246	1204	1190	1172	11.25	75
40	1414	1390	1360	1338	1316	12.06	79
42	1558	1544	1502	1482	1464	11.50	83

Table 10-10B
Full Circle Tube Layouts Floating Head Exchanger $^3/_4$-in. O.D. Tubes on 1-in. Square Pitch

Size (In.)	Number of Passes 1	2	4	6	8	Net Free Distance 2 Passes	Rows Across
8	32	32	26	24	24	3.50	6
10	56	52	48	48	48	4.13	8
12	82	82	78	72	72	4.50	11
14	104	96	92	88	88	6.00	11
16	140	136	128	120	120	6.50	14
18	185	180	172	168	164	6.88	16
20	241	236	224	212	212	7.38	17
22	300	280	280	268	268	7.75	20
24	360	350	336	332	332	8.25	21
26	424	412	402	392	392	8.75	23
28	402	488	480	472	472	9.25	25
30	580	566	566	548	548	9.75	27
32	665	648	644	628	628	10.00	29
34	756	758	730	728	728	10.19	31
36	853	848	832	816	816	11.69	33
38	973	950	938	932	932	12.19	35
40	1085	1064	1052	1036	1036	12.69	37
42	1201	1176	1162	1148	1148	13.19	39

Table 10-10C
Full Circle Tube Layouts Floating Head Exchanger $^3/_4$-in. O.D. Tubes on 1-in. Triangular Pitch

Size (In.)	Number of Passes 1	2	4	6	8	Net Free Distance 2 Passes	Rows Across
8	37	30	26	26	24	4.13	11
10	61	56	52	46	44	4.88	17
12	92	90	86	78	72	4.38	21
14	121	110	102	98	92	5.50	21
16	163	152	146	140	132	6.25	25
18	212	202	194	188	184	5.88	29
20	269	260	250	240	236	6.63	35
22	337	330	314	300	296	7.33	39
24	421	404	380	378	364	8.00	43
26	499	476	460	450	440	8.88	47
28	579	562	542	538	520	9.63	51
30	668	648	636	624	612	10.33	53
32	766	744	732	714	712	11.00	57
34	870	850	834	828	808	10.50	61
36	986	978	942	932	920	11.38	67
38	1108	1100	1060	1060	1036	12.13	71
40	1236	1228	1200	1190	1164	12.75	75
42	1367	1350	1322	1306	1288	13.25	77

Table 10-10D
Full Circle Tube Layouts Floating Head Exchanger, 1-in. O.D. Tubes on 1 $^1/_4$-in. Square Pitch

Size (In.)	Number of Passes 1	2	4	6	8	Net Free Distance 2 Passes	Rows Across
8	21	16	16	....	....	4.13	5
10	37	32	32	32	....	4.25	6
12	48	52	48	48	48	4.25	8
14	61	60	60	52	52	5.50	10
16	89	84	80	76	76	5.50	11
18	113	112	112	108	108	5.38	13
20	148	148	140	136	136	5.38	14
22	184	178	172	168	164	7.38	16
24	221	220	212	208	208	7.38	17
26	266	266	258	252	252	7.38	19
28	316	308	304	292	292	7.38	20
30	368	360	352	344	340	9.38	21
32	421	410	402	392	392	9.38	23
34	481	472	464	452	452	9.69	24
36	545	540	532	524	524	9.69	26
38	608	608	588	588	588	9.69	28
40	680	680	656	664	660	9.69	29
42	750	738	728	728	728	11.69	31

Table 10-10E follows.

Table 10-10E
Full Circle Tube Layouts Floating Head Exchanger, 1-in. O.D. Tubes on 1 $^1/_4$-in. Triangular Pitch

Size (In.)	Number of Passes 1	2	4	6	8	Net Free Distance 2 Passes	Rows Across
8	22	20	18	16	12	4.13	9
10	38	36	32	32	28	4.50	13
12	60	52	48	46	44	4.63	15
14	73	68	60	58	56	4.25	19
16	97	98	86	82	80	4.25	21
18	130	126	118	114	112	6.50	25
20	170	164	152	150	144	6.75	27
22	212	202	196	188	184	7.00	31
24	258	250	242	232	228	7.25	33
26	304	302	286	278	272	7.50	37
28	361	348	338	336	324	7.75	39
30	421	408	400	394	388	8.00	43
32	482	472	456	446	440	8.25	47
34	555	538	524	520	500	11.06	49
36	625	618	592	588	572	10.44	51
38	700	688	672	660	640	10.69	55
40	786	776	752	742	736	11.00	59
42	872	850	834	824	816	11.32	61

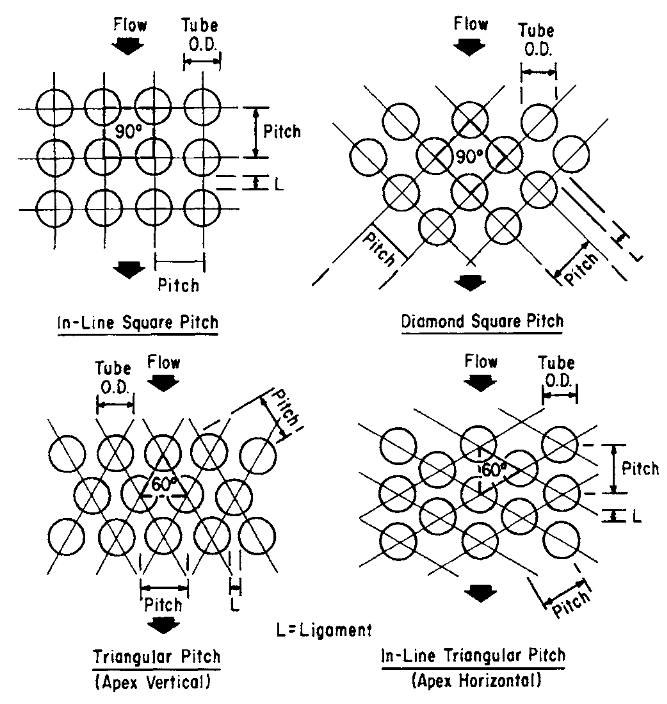


Figure 10-26. Tube spacing layouts for tubesheets.

Exchanger Surface Area

The actual surface area available for heat transfer is determined from the fabricator's shop drawings. From these details, the following are fixed:

A. Number of tubes.

The actual number of tubes to be installed in the unit. Manufacturing tolerances may require elimination of some tubes that preliminary design layouts and tables indicated might be installed in the unit. Figures 10-25A–K and Table 10-9 have considered known fabrication tolerances. Sometimes extra tie rods for baffles must be added, or in some cases, eliminated. The outer tube circle limit for each exchanger is determined by the type of shell to be used. That is, (1) if commercial pipe, greater out-of-round tolerances might be required or (2) if formed on shop rolls, the out-of-round tolerance will be known, but not necessarily the same for each diameter shell.

B. Exact distance between faces of tubesheets.

Tubes are usually ordered in even lengths, such as 8, 10, 12, 16, 24, or 32 ft, and the tubesheets are from $^1/_4$ in. to $^1/_2$ in. shorter between outer faces.

C. Net effective tube length.

This is the net length of tube exposed inside the shell and available for contact by the shell-side fluid. This length accounts for the thickness of each tubesheet (and for the double tubesheets when used). For design purposes, it is usually estimated from experience, allowing about

1. 1 $^1/_2$ in. per tubesheet for low-pressure units.
2. 2–3 in. per tubesheet for high-pressure exchangers, 200 psi–400 psi.

D. Exact baffle spacing.

In some instances the baffle spacing must be rearranged to allow for a nozzle or coupling connection. It is important that changes in baffle location be reviewed, as performance or pressure drop can be seriously affected. This is of extreme importance in vacuum units. Baffle orientation is sometimes misinterpreted by the fabricator, and this can cause serious problems where liquid drainage is concerned, or the revised vapor flow path can allow for bypassing the tube surface.

E. Impingement baffle location.

When scale drawings are made, the effectiveness of impingement baffles can be evaluated easily. Sometimes it is necessary to relocate or make slight size changes in order to properly protect the tubes and direct the vapor flow.

Effective Tube Surface

The effective tube surface is usually evaluated on the outside tube surface. Use net tube length.

Net effective outside area for plain or bare tubes is:

A_n = (ft^2 external surface per ft length from Table 10-3) (L_e, net effective tube length) (N_t, number of tubes) (10-4)

Net effective outside area for finned tube is:

A_{nf} = (ft^2 external finned surface per ft length from Table 10-39 or other specific tube data) (L_e, net effective tube length) (N_t, number of tubes) (10-5)

Effective Tube Length for U-Tube Heat Exchangers

One challenge in the design of U-tube heat exchangers is to determine the effective length of the tubes. For example, when U-tube bundles are fabricated from 12-ft tubes, the maximum length tube in the bundle is 12 ft which is in the outside tube row. The inside tube is the shortest and is less than 12 ft long.

The effective tube length, L_e, of the bundle for surface area calculations is the mean of the tube lengths between the outside tubes and the inside tubes. See Figure 10-27A.

In calculating the optimum U-tube heat exchanger design, most designers estimate the effective tube length for each of the various heat exchangers. After a specific heat exchanger design is selected, the effective tube length is determined accurately by the fabricator. If the estimated length differs significantly from the actual length, additional design calculations may be necessary.

To more easily determine the effective tube lengths for U-tubes, the correction chart shown in Figure 10-27B[124] is convenient. The chart is based on many actual U-tube bundle layouts. Values read from the chart are not more than 1% lower than those obtained by calculations, except where the curve is extrapolated to lower tube counts. Such extrapolations result in errors of 3%, 4%, and higher, giving larger values than those calculated. This does not apply to higher tube count extrapolations. The chart is limited to $^3/_4$-in. and 1-in. O.D. tubes. The accuracy of extrapolation to other diameters has not been determined. The chart is applicable to low-finned tubes, as well as to plain tubes. However, it is restricted to either two- or four-tube passes. For arrangements other than those used in the chart preparation, the chart may be used at the designer's discretion. As an example of those "beyond limits," the following is a comparison of calculated areas with actual areas for various U-tube bundles installed in a chemical plant:

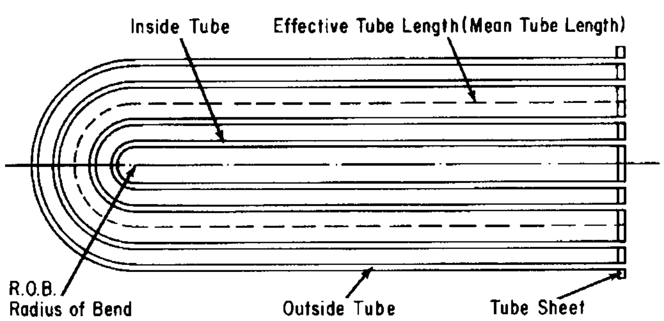


Figure 10-27A. U-tube bundle.

No. of U-Tubes	Tube Size	Inside Tube Pitch	Calc'd* Tube R.O.B.	Area Ft²	Act. Area	% Error
66	$^3/_4$ in. O.D. × 32 ft	1 in.□	2 in.	400.0	405.7	+1.43
88	$^3/_4$ in. O.D. × 32 ft	1 in.□	2 in.	534	533	−0.19
54	1 in. O.D. × 32 ft	1 $^1/_4$ in.□	3 in.	439	448	+2.05

Note: R.O.B. = Radius of bend.

*The calculated areas were based on Figure 10-27B using only the respective tube size and number of tubes.

Example 10-2. Use of U-Tube Area Chart[124]

Case 1

Given:

Number of U-tubes: 168

Tubes: 0.75 in. O.D. × 16 ft (nominal) plain tubes

Required:

Total effective exposed area.

From Figure 10-27B:

Effective tube length = 14.5 ft.

$$\text{Total effective exposed area} = (14.5)(168)\,\pi\left(\frac{0.75}{12}\right) = 478.3\ \text{ft}^2$$

Case 2

Given:

Tubes: 1 in. × 12 ft (nominal) low-finned tubes (19 fins/in.).

$$\text{Outside area of tubes} = 0.678\ \frac{\text{ft}^2}{\text{linft}}$$

Exposed area of bundle = 2,142 ft^2

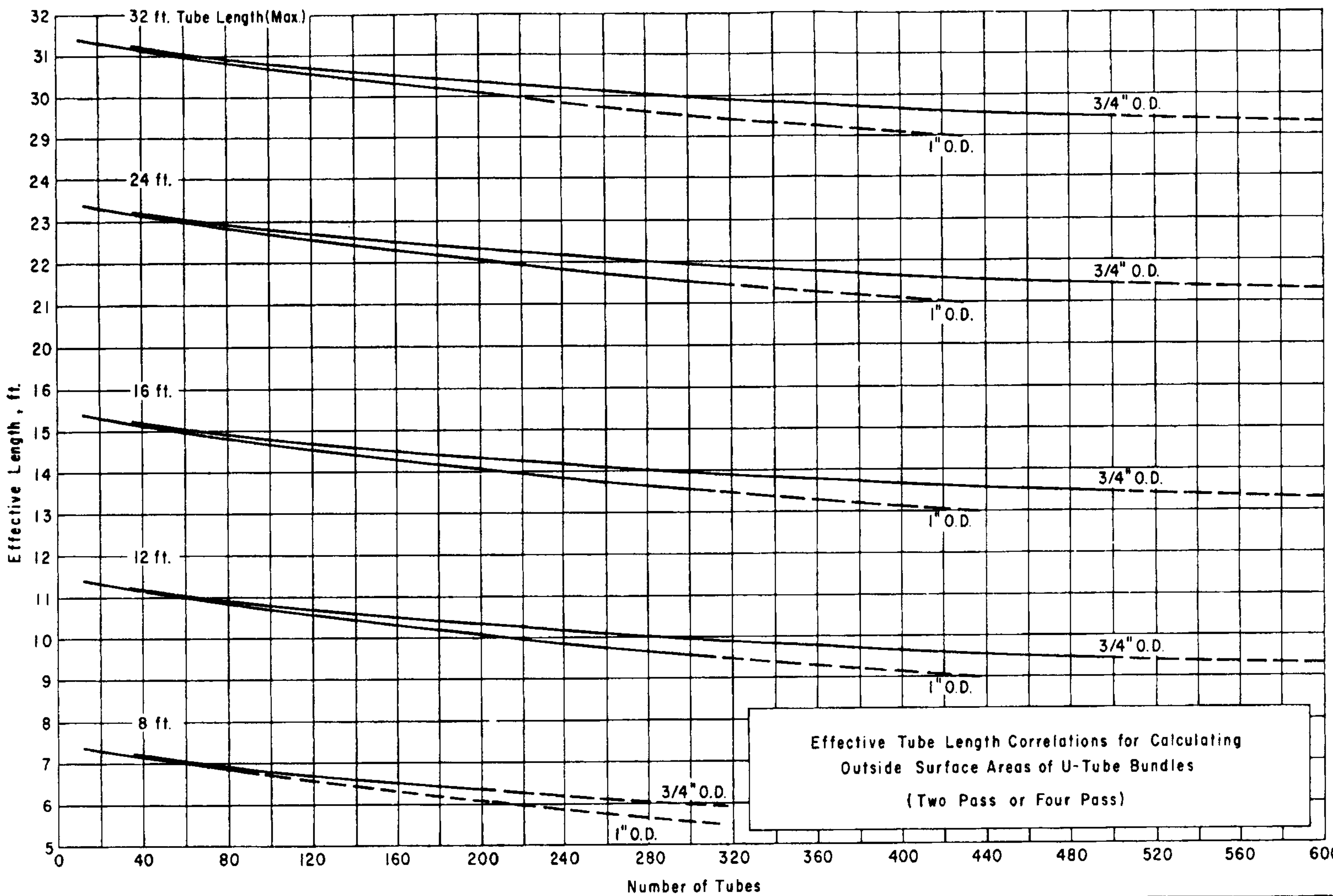
Required:

Number of tubes required.

A brief trial and error procedure is necessary.

Assume effective tube length = 11 ft.

$$\text{Thus, the number of tubes required} = \frac{2{,}142}{(11)(0.678)} = 288$$



Basis For Chart Preparation:

1. Tube O.D.	Minimum Radius Bend	Pitch
3/4"	1.5 Tube Dia.	1" Δ
1"	2.5 Tube Dia.	1.25" Δ and □

2. Tubes on Exchanger Center Line.
3. 3/8" Clearance Between U-Bundle and Inside of Shell.
4. Allowance of 1.5" for Thickness of Tube Sheet.

Accuracy:
Low Tube Counts (10 ≅ 4-6%   All Other Counts (0.1 - 1.0%

Use:
Total Effective Bundle Outside Tube Surface Area := (Effective Length, ft.) (No. Tubes)(Outside Surface Area per ft. of Length).

Notes:
1. Number of Tube Holes in Tube Sheet is Twice Number of Tubes.
Chart Gives Results 0.2-2% High for 1" Tubes on 1.5 Minimum Radius Bend.
Chart Gives Acceptable Results for 3/4" Tubes on 1" □ Pitch.
2. See Discussion Regarding Chart Accuracy for Tube Arrangements Other Than Those Listed Here.

Figure 10-27B. Effective tube length correlations for calculating outside surface areas of U-tube bundles (two or four passes). (Used by permission: D. L. Whitley and E. E. Ludwig, *Chemical Engineering.* V. 67, No. 6 © 1960. McGraw-Hill, Inc. All rights reserved.)

From Figure 10-27B:

Effective tube length for 288 tubes = 9.6 ft.

$$\text{Calculate new number of tubes} = \frac{2{,}142}{(9.6)(0.678)} = 330$$

Effective tube length = 9.42 ft.

$$\text{Calculate new number of tubes} = \frac{2{,}142}{(9.42)(0.678)} = 336$$

Effective tube length for 336 tubes = 9.4 ft.
Thus, required number of tubes = 336

The equations of Figure 10-27B correlations are as follows:

L_e = Effective tube length, ft.
N_t = Number of U-tubes
L = Nominal tube length, ft.

For $^3/_4$-in. U-tubes:

$$L_e = (L - 0.5) - (7.4007 \times 10^{-3})(N_t) + (8.5791 \times 10^{-6})(N_t)^2 - (3.7873 \times 10^{-9})(N_t)^3 \qquad (10\text{-}6)$$

For 1-in. U-tubes:

$$L_e = (L - 0.5) - (9.2722 \times 10^{-3})(N_t) + (1.1895^{-5})(N_t)^2 - (8.4977 \times 10^{-9})(N_t)^3 \qquad (10\text{-}7)$$

Nozzle Connections to Shell and Heads

Inlet and outlet liquid nozzles are sized by conventional pressure drop evaluations or by the more common velocity guides. For low-pressure vacuum services, velocities should not be used to establish any critical connection size. (Figure 10-63 is a useful guide for the usual case.)

Safety valves are often required on the shell side of exchangers and sometimes on the tube side. These valves may require sizing based upon process reaction, overpressure, etc., or on external fire. For details, see Chapter 7, Vol. I on safety-relieving devices.

Drains are necessary on the shell and on the bottom of most heads. Sometimes several drains are necessary on the shell side to facilitate drainage between baffles when flushing is a part of the operation.

Vents are usually placed on the shell and on the tube-side heads to allow venting of inert gasses or other material. A 1 in.-6,000 lb. half or full-coupling is recommended for both vent and drain, unless other sizes are indicated.

Couplings are handy to have on the process inlet and outlet nozzles on both the tube and shell sides. These may be used for flushing, sampling, or thermometer wells, thermocouple bulbs, or pressure gages.

Types of Heat Exchange Operations

The process engineer identifies heat exchange equipment in a process by the operation or function it serves at a particular location in the flow cycle. For example, the bottom vaporizer on a product finishing distillation column is usually termed "Finishing Column Reboiler E-16," or "Reboiler E-16;" the overhead vapor condenser on this column is termed "Condenser E-17;" etc. The usual operations involved in developing a process flowsheet are described in Table 10-11, or Chapter 1, Volume 1.

Thermal Design

Engineering thermal design of heat transfer equipment is concerned with heat flow mechanisms of the following three types—simply or in combination: (1) conduction, (2) convection, and (3) radiation. Shell and tube exchangers are concerned primarily with convection and conduction; whereas heaters and furnaces involve convection and radiation.

Radiation is not generally considered in conventional heat transfer equipment *except* for direct gas/oil-fired heaters and cracking units. These later types are not a part of this chapter, because they are specialty items of their own as far as design considerations are concerned.

Conduction is heat transfer through a solid nonporous barrier when a temperature difference exists across the barrier. The thermal transfer capability of the specific barrier or wall material, known as *thermal conductivity,* determines the temperature gradient that will exist through the material.

$$Q = \frac{k_a}{L_c}A(t_2 - t_1) = \frac{k_a}{L_c}A\Delta t \qquad (10\text{-}8)$$

Referring to Figure 10-28, conduction occurs through the tube wall and is represented by a temperature drop $t_4 - t_5$ and through the scale of fouling by the drops $t_3 - t_4$ and $t_5 - t_6$.

Convection is heat transfer between portions of a fluid existing under a thermal gradient. The rate of convection heat transfer is often slow for natural or free convection to rapid for forced convection when artificial means are used to mix or agitate the fluid. The basic equation for designing heat exchangers is

$$Q = UA(t_2 - t_1) = UA\Delta t \qquad (10\text{-}9)$$

where

$(t_2 - t_1)$ represents the temperature difference across a *single fluid* film. Referring to Figure 10-28, convection occurs through the fluid $t_1 - t_3$ and also $t_6 - t_8$.

where

A = net external surface area of tubes exposed to fluid heat transfer (not just the length of the individual tubes), ft^2.
Q = heat load, Btu/hr
U = overall heat-transfer coefficient, Btu/(hr-ft^2 - °F)
ΔT = mean temperature difference, °F, corrected

Table 10-11
Heat Exchange Operations

Equipment Designation	Process Operation
Condenser	(a) Condenses all vapors (pure or mixed) entering. (b) Condenses all condensable vapor, cools the gases—termed a cooler-condenser.
Partial Condenser	Condenses only part of the total entering vapors; condensed liquid removed as reflux or as "fractionation mixture;" vapor passes out unit to a second condenser, or on for other processing.
Cooler	Cools process stream, usually by water, but can be by air as in air cooler or by other process fluid.
Chiller	Cools process stream by refrigerant at temperature lower than prevailing water, can be chilled by water cooling the process fluid or by refrigerant such as ammonia, propylene, and freon. (Also see "Evaporator.")
Evaporator	(a) Evaporates process fluid by some heating medium such as steam. (b) Evaporates refrigerant such as ammonia, propylene, etc., while cooling (or chilling or condensing) process fluid. Usually refrigerant on shell side of exchanger. (c) Evaporates part of process mixture while concentrating remainder as liquid. (See "Vaporizer.")
Vaporizer	Vaporizes or evaporates all or part of liquid fed to unit by means of heating medium, such as steam, Dowtherm, etc.
Reboiler	Boils liquid by heating medium in a recirculation cycle. Feed may flow by
(a) Forced Circulation	(a) Pumped through tubes (usually), vaporizing main portion on leaving, termed "Forced Circulation Reboiler."
(b) Natural Circulation or Thermosiphon	(b) Natural static and thermal heads through tubes, vaporizing part of fluid near outlet, termed "Natural Circulation" or "Thermosiphon Reboiler."
Heater	Heats fluid (adds sensible heat) but does not vaporize except for effect of temperature on vapor pressure. Heating medium is usually steam, Dowtherm, or similar fluid that condenses at pressure and temperature desired, imparting its latent heat to fluid (gas or liquid).
Steam Generator	Produces steam from condensate or boiler feed water by combustion of waste oil, tars, or "off-gas" in direct-fired equipment.
Waste Heat Boiler	Produces steam from condensate or boiler feed water by removal of sensible heat from high temperature level process or waste gas streams. (Sometimes liquid streams serve this function.)
Exchanger (a) Cross Exchanger	(a) Exchanges sensible heat between two process streams, either liquids or gases, cooling one while heating the other. Sometimes termed *cross-exchanger.*
(b) Heat-Exchanger	(b) May exchange heat for type of streams noted in (a), or any combination of specifically identified types mentioned previously, such as Cooler, Heater, etc. Usually limited to sensible heat exchange.

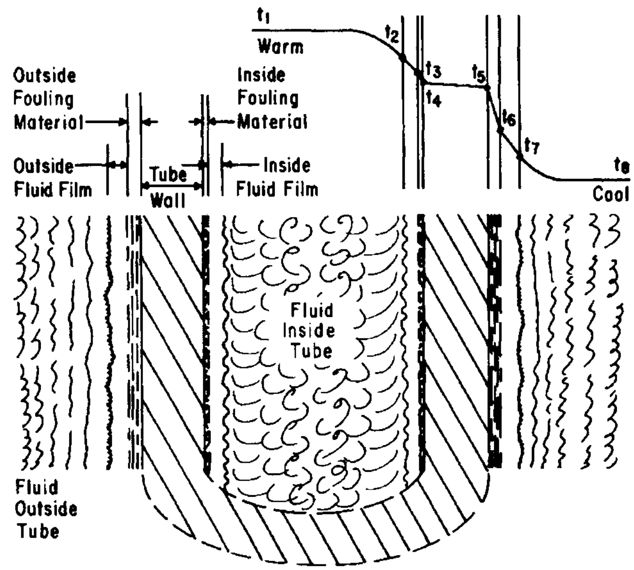


Figure 10-28. Tube wall conditions affecting overall heat transfer and associated temperature profile.

An important step in accurately establishing the required net surface area of an exchanger is to determine the true ΔT.

For example, the *simplest* temperature difference involves *constant temperature* on *each side of the tube,* such as steam condensing on one side at about 410°F and an organic hydrocarbon compound boiling at constant temperature of about 250°F. Use this *simple* temperature difference for Equation 10-9:

$$\Delta T = 410 - 250 = 160°F$$

This applies regardless of the fluid flow pattern in the unit[129]. Such a unit could be like the one shown in Figure 10-1C; also see Figure 10-29B.

For counter-current flow of the fluids through the unit with sensible heat transfer only, this is the most efficient temperature driving force with the largest temperature cross in the unit. The temperature of the outlet of the hot stream can be cooler than the outlet temperature of the cold stream, see Figure 10-29:

Hot: 200°F→100°F
Cold: 80°F→150°F

Note that the Log Mean Temperature Difference (LMTD) is somewhat less than the arithmetic mean, represented by the following:

$$[(T_1 - t_2) + (T_2 - t_1)] \div 2,$$

or, (hot - cold terminal temperature difference) ÷ 2 (10-10)

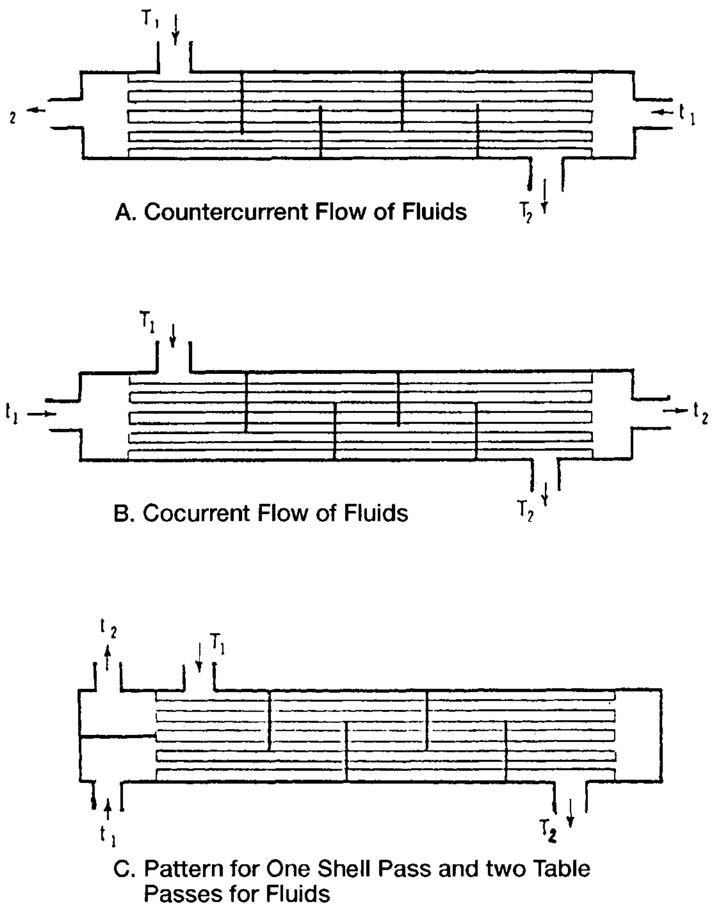


Figure 10-29. Three flow patterns for examining ΔT and LMTD. Note: T_1 = shell-side fluid inlet, and t_1 = tube-side fluid inlet.

The ΔT for this flow is given by reference 129, Equation 10-11, using end conditions of exchanger. Thus:

$$\Delta T = \frac{(T_2 - t_1) - (T_1 - t_2)}{\ln\left(\dfrac{T_2 - t_1}{T_1 - t_2}\right)} = \frac{GTD - LTD}{\ln \dfrac{GTD}{LTD}} \tag{10-11}$$

$$= LMTD$$

where

GTD = Greater Terminal Temperature Difference, °F
LTD = Lesser Terminal Temperature Difference, °F
LMTD = Logarithmic Mean Temperature Difference, °F = ΔT
T_1 = Inlet temperature of hot fluid, °F
T_2 = Outlet temperature of hot fluid, °F
t_1 = Inlet temperature of cold fluid, °F
t_2 = Outlet temperature of cold fluid, °F

Note that the logarithmic mean temperature difference should be used when the following conditions generally apply[107] for conditions of true counter-current or co-current flow:

- Constant overall heat transfer coefficient.
- Complete mixing within any shell cross pass or tube pass.
- The number of cross baffles is large (more than 4).
- Constant flow rate and specific.
- Enthalpy is a linear function of temperature.
- Equal surfaces in each shell pass or tube pass.
- Negligible heat loss to surroundings or internally between passes.

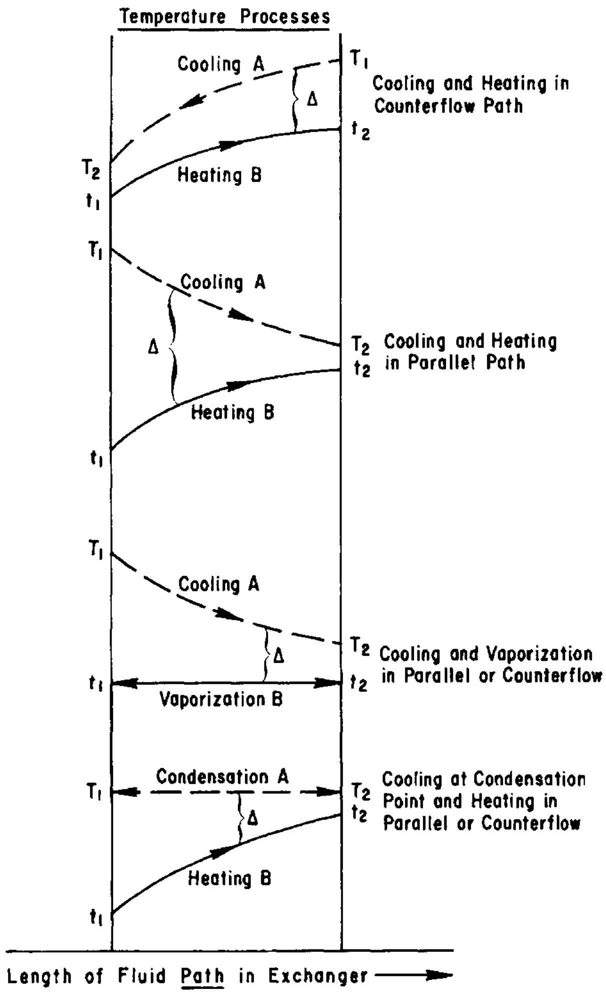
For co-current flow (see Figure 10-29B), the temperature differences will be $(T_1 - t_1)$, and the opposite end of the unit will be $(T_2 - t_2)$. This pattern is not used often, because it is not efficient and will not give as good a transfer and counter-current[129] flow. Because the temperature cannot cross internally, this limits the cooling and heating of the respective fluids. For certain temperature controls related to the fluids, this flow pattern proves beneficial.

For one shell and multipass on the tube side, it is obvious that the fluids are not in true counter-current flow (nor co-current). Most exchangers have the shell side flowing through the unit as in Figure 10-29C (although some designs have no more than two shell-side passes as in Figures 10-1J and 10-22, and the tube side fluid may make two or more passes as in Figure 10-1J); however, more than two passes complicates the mechanical construction.

Temperature Difference: Two Fluid Transfer

The temperature difference, Δt, °F, required to satisfy the basic heat transfer relation Q = UA Δt is the logarithmic mean to the differences in temperatures at the opposite ends of the paths of flow of the two fluids. The temperature flow paths can be represented as shown in Figures 10-30 and 10-31.

In true *counterflow* operation (sensible heat transfer), the one fluid, A, being cooled is flowing at all times in a near 180° direction to the fluid being heated, B, Figure 10-30. Note that because A is being cooled, it comes into the exchanger at a temperature, T_1, which is hotter than the inlet, t_1, of the fluid being heated. In this case the fluid B can leave at a temperature, t_2 which is greater than the outlet temperature T_2 of fluid A. The vertical distance between the two curves at any point along the travel length of the fluid is the temperature difference (T' − t') (or Δ, Figure 10-30) at that point.



Δ Represents the Difference in Temperature Between the Flowing Fluids at any Point in the Exchanger

Figure 10-30. Temperature paths in heat exchangers.

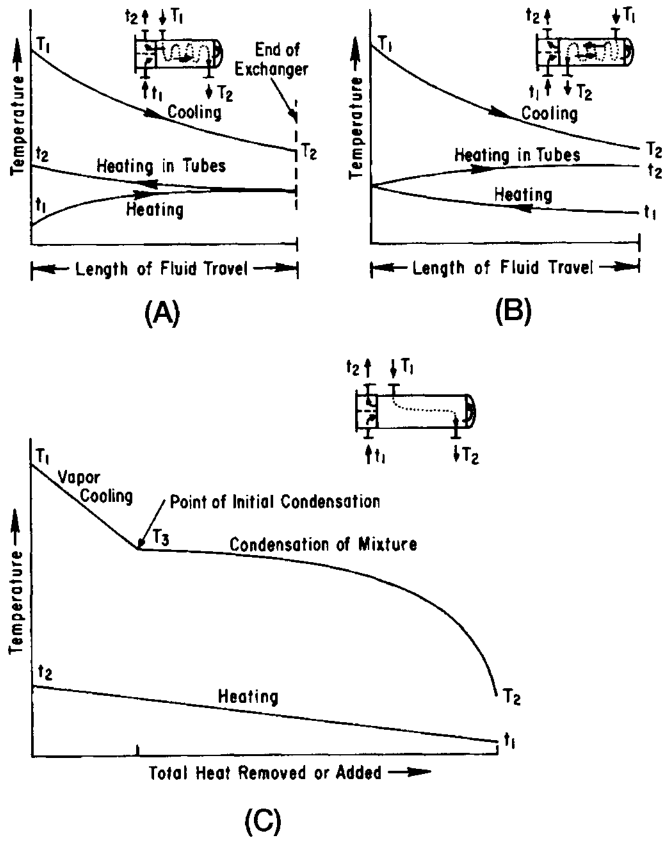


Figure 10-31. Fluid flows through two passes in tubes: part of flow is parallel to shell-side fluid, and part is counterflow.

In *parallel* operation (sensible heat transfer), fluids A and B (Figure 10-30) flow in the same direction along the length of travel. They enter at the same general position in the exchanger, and their temperatures rise and fall respectively as they approach the outlet of the unit and as their temperatures approach each other as a limit. In this case the outlet temperature, t_2, of fluid B, Figure 10-30, cannot exceed the outlet temperature, T_2, of fluid A, as was the case for counterflow. In general, parallel flow is not as efficient in the use of available surface area as counterflow.

In *condensation* one fluid remains at constant temperature throughout the length of the exchanger while the fluid B that is absorbing the latent heat of condensation is rising in temperature to an outlet of t_2. Note that as fluid A condenses, it does not flow the length of the travel path. Fluid A drops to the bottom of the exchanger and flows out the outlet at temperature T_2, which is the same as T_1, the temperature of condensation, providing no subcooling occurs to lower the temperature of the liquid to less than T_2. In this case, t_2 approaches but never reaches T_2. When viewed from the condensation operation, the unit is termed a *condenser*; however, if the main process operation is the *heating* of a fluid with the latent heat of another stream, such as steam, then the unit is termed a *heater*. If boiling follows sensible heating, the unit is a *reboiler*.

Temperature *crosses* in an exchanger can prevent the unit from operating. Figure 10-32 indicates two situations, one involving desuperheating and condensing a vapor, and the second requiring the heating and vaporizing of a fluid. In the first instance note that it is not simply the desire to remove a fluid t_2 at a temperature greater than T_2, but more fundamentally involves the shape of the temperature profile curves. To be certain of performance, the heating, cooling, condensing, or vaporizing curves for the fluids should be established. Although a unit may calculate to give performance based on end limits of temperature, if a cross exists inside between these limits, the expected heat exchange will not be accomplished.

For the average fluid temperature and/or true caloric temperature see "Temperature for Fluid Properties Evaluation—Caloric Temperature" later in this chapter.

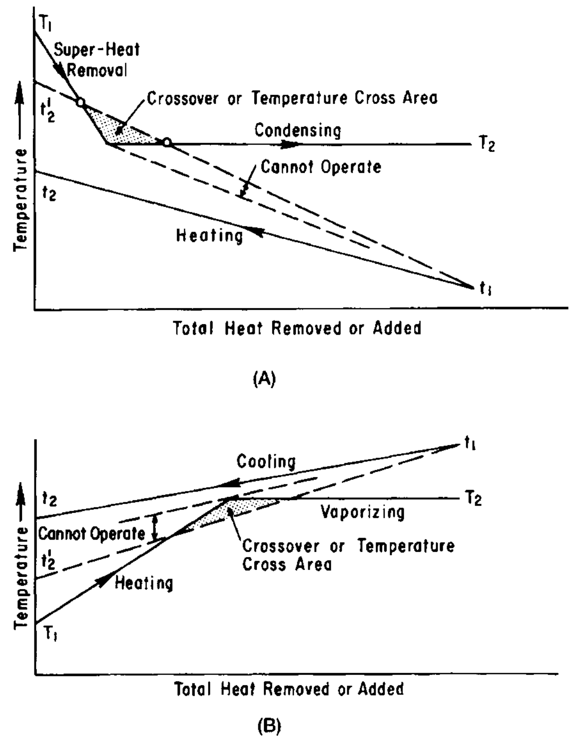


Figure 10-32. Typical temperature situations that contain cross-over points, preventing exchanger operation. (Adapted and used by permission: Brown and Root, Inc.)

Counter-current or co-current flow of the two (usual) fluids in a heat transfer operation is the most efficient of the several alternate design combinations. The most efficient transfer occurs in a straight-through, single-pass operation, such as shown in Figures 10-2 and 10-29, and design-wise Figure 10-1H (but not as a reboiler). Usually for these cases the logarithmic mean temperature difference may be applied as Murty[132] discusses a calculation method for establishing the maximum possible cross in a parallel counterflow exchanger, Figure 10-13. In the following example, this technique is outlined.

Example 10-3. One Shell Pass, 2 Tube Passes Parallel-Counterflow Exchanger Cross, After Murty[132]

Find the minimum temperature that a hot fluid at 410°F can be cooled if the cold fluid is heated from an inlet temperature of 167°F to 257°F. Also find the theoretical temperature cross and theoretical minimum hot fluid shell-side outlet temperature, T_2.

Using equations from reference 132:

$$T_{2\,min} - t_1 = \frac{(T_1 - t_1)}{1 - \dfrac{(\Delta t)}{(2(T_1 - t_1))}} - (T_1 - t_1) \tag{10-12}$$

$$\Delta t = t_2 - t_1$$

$$\text{Substituting: } T_{2\,min} - 167 = \frac{[(410 - 167)]}{1 - \dfrac{((257 - 167))}{(2(410 - 167))}} - (410 - 167)$$

$$T_{2\,min} - 167 = 55°F$$

$$(\text{actual})\ T_{2\,min} = 222°F$$

Maximum temperature cross:

$$\frac{[t_2 - T_2]}{[T_1 - t_1]} = \frac{(2 - 2^{1/2})(2^{1/2} - 1)}{2^1/_2} = 0.1715$$

The theoretical maximum possible temperature cross in *this* style exchanger = $(t_2 - T_{2\,min}) = 0.1715$.

Theoretical $(t_2 - T_2)_{max} = 0.1715\ (T_1 - t_1)$

Then, theoretical $T_{2\,min} = t_2 - (t_2 - T_2)_{max}$

Then, for the example: the theoretical maximum possible temperature cross:

$$(t_2 - T_2)_{max} = 0.1715\,(410 - 167) = 41.6°F$$

$$\text{Theoretical } T_{2min} = 257 - 41.6 = 215.4°F$$

or, when, $T_1 - t_1 > \Delta T$, then the following approximation applies:

$$T_{2\,min} - t_1 \cong (T_1 - t_1)\left[1 + \frac{\Delta t}{2(T_1 - t_1)}\right] - (T_1 - t_1) \cong \Delta t/2$$

Use the preceding equation when $(T_1 - t_1) \geq 50$.

T = Shell-side fluid, °F
t = Tube-side fluid, °F
T_{2Min} = Minimum hot fluid exit temperature achievable, °F
1 = Inlet (hot)
2 = Outlet (cool)

In *vaporization*, one fluid, B, vaporizes at constant temperature while the second fluid, A, is cooled from T_1 to T_2. When a refrigerant such as propylene is being vaporized to condense ethylene vapors, the unit actually operates at a fixed temperature difference for the entire length of the exchanger. In this latter situation, t_1 equals t_2 and T_1 equals T_2. In an evaporator, one fluid is vaporized as the heating fluid is cooled to T_2.

Mean Temperature Difference or Log Mean Temperature Difference

For these cases, the logarithmic mean temperature difference may be applied as:

$$\Delta t = LMTD = MTD = \frac{\Delta t_2 - \Delta t_1}{\ln\dfrac{\Delta t_2}{\Delta t_1}} = \frac{\Delta t_2 - \Delta t_1}{2.3\log_{10}\dfrac{\Delta t_2}{\Delta t_1}} \tag{10-13}$$

ΔT = Log Mean Temperature difference = LMTD

here: Δt_1 = Temperature difference at one end of exchanger (smaller value), see Figure 10-29.

Δt_2 = Temperature difference at other end of exchanger (larger value)

ln = Natural logarithm to base e

MTD = Mean Temperature Difference, °F, see Figure 10-33 = Log mean temperature difference

LTD = Δt_1 = Least terminal temperature difference

GTD = Δt_2 = Greater terminal temperature difference

Figure 10-33 is a useful means of solving the LMTD calculation.

Correction factors are given in Figures 10-34A–F to modify the true *counter-current* LMTD for the *multipass* exchanger actual flow paths and accompanying temperature deviations.

Note that where Figures 10-34A–J represent corrections to the LMTD for the physical configuration of the exchanger, Figures 10-35A–C represent the temperature efficiency of the unit and are not the same as the LMTD correction.

Often, a reasonable and convenient way to understand the heat transfer process in a heat exchanger unit is to break down the types of heat transfer that must occur: such as, vapor subcooling to dew point, condensation, and liquid subcooling. Each of these demands heat transfer of a different type, using different ΔT values, film coefficients, and fouling factors. This is illustrated in Figure 10-36. It is possible to properly determine a weighted overall temperature

$$MTD = \frac{GTD - LTD}{\log_e \frac{GTD}{LTD}}$$

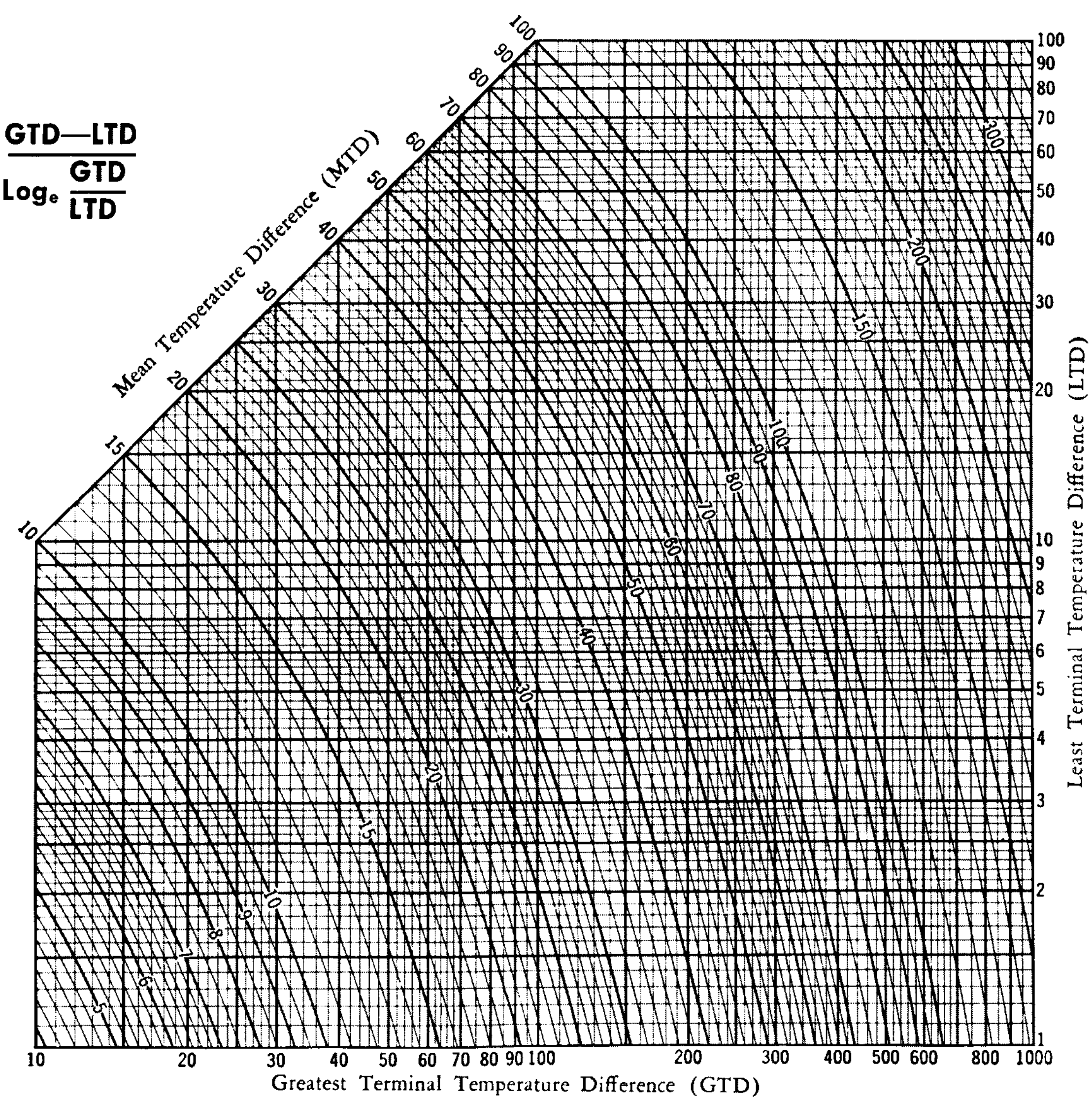


Figure 10-33. Mean temperature difference chart. (Used by permission: The Griscom-Russell/Ecolaire Corporation.)

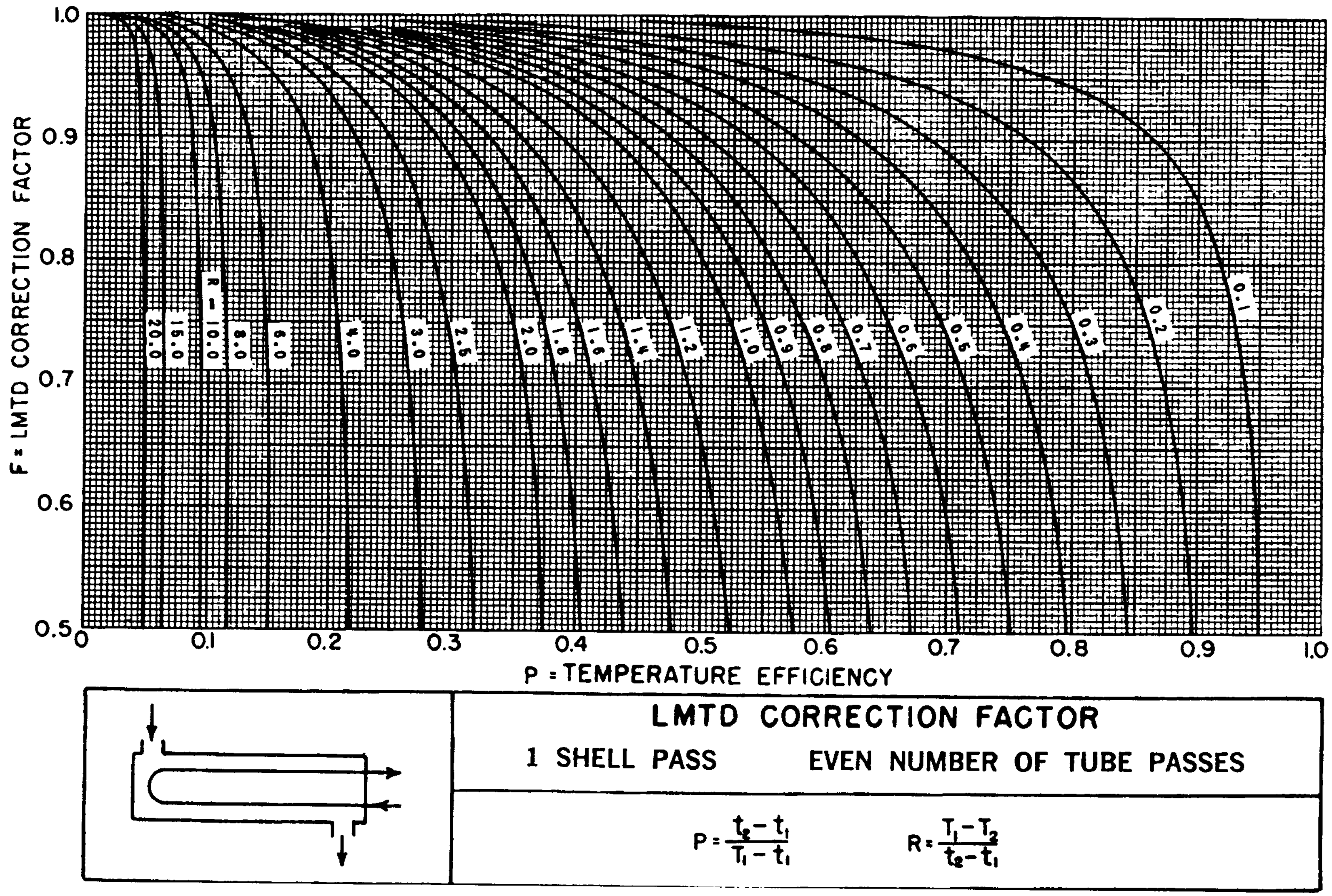


Figure 10-34A. MTD correction factor, 1 shell pass, even number of tube passes. (Figures 10-34A–10-34J used by permission: *Standards of Tubular Exchanger Manufacturers Association*, 7th Ed., Figure T-32, ©1988. Tubular Exchanger Manufacturers Association, Inc.)

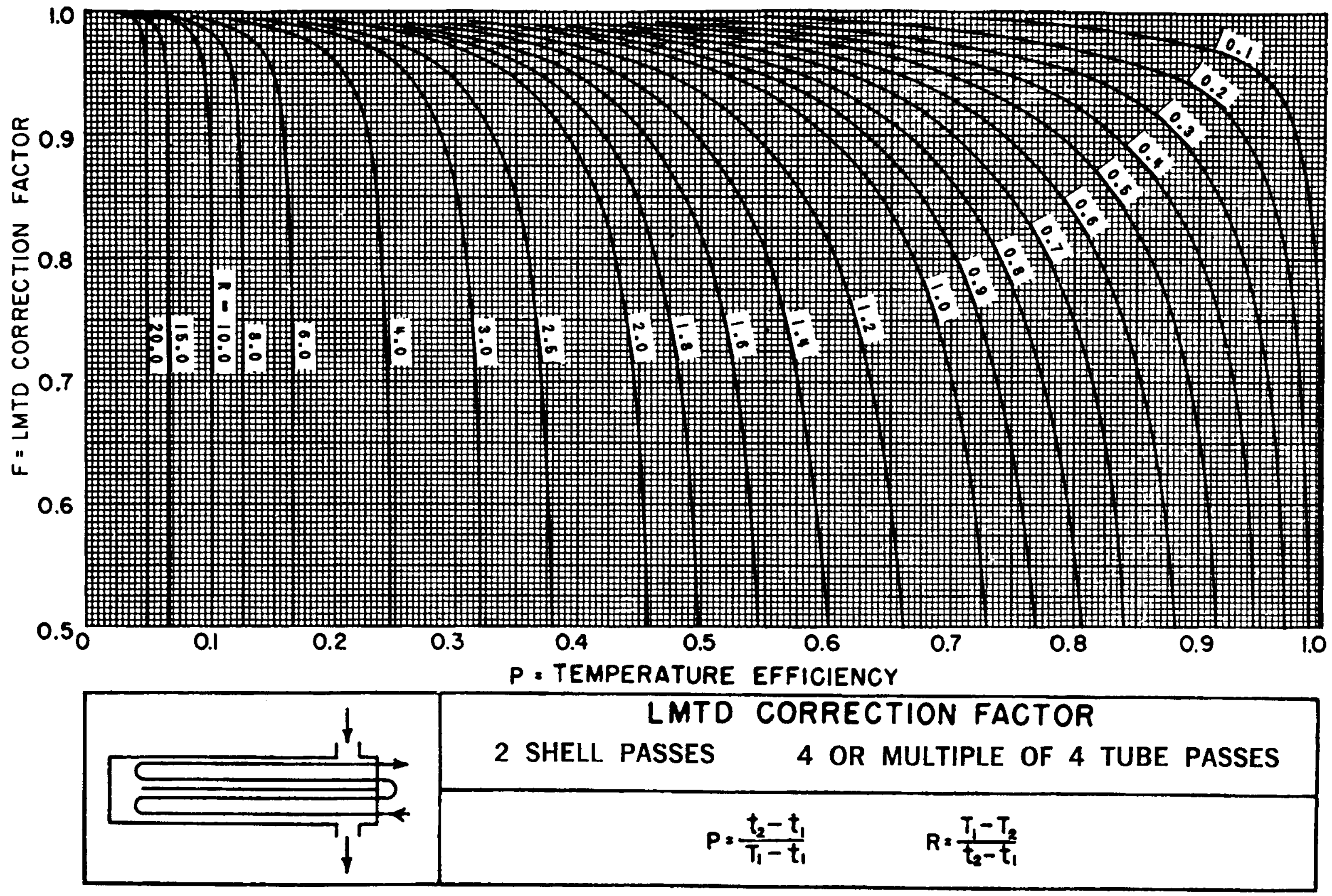
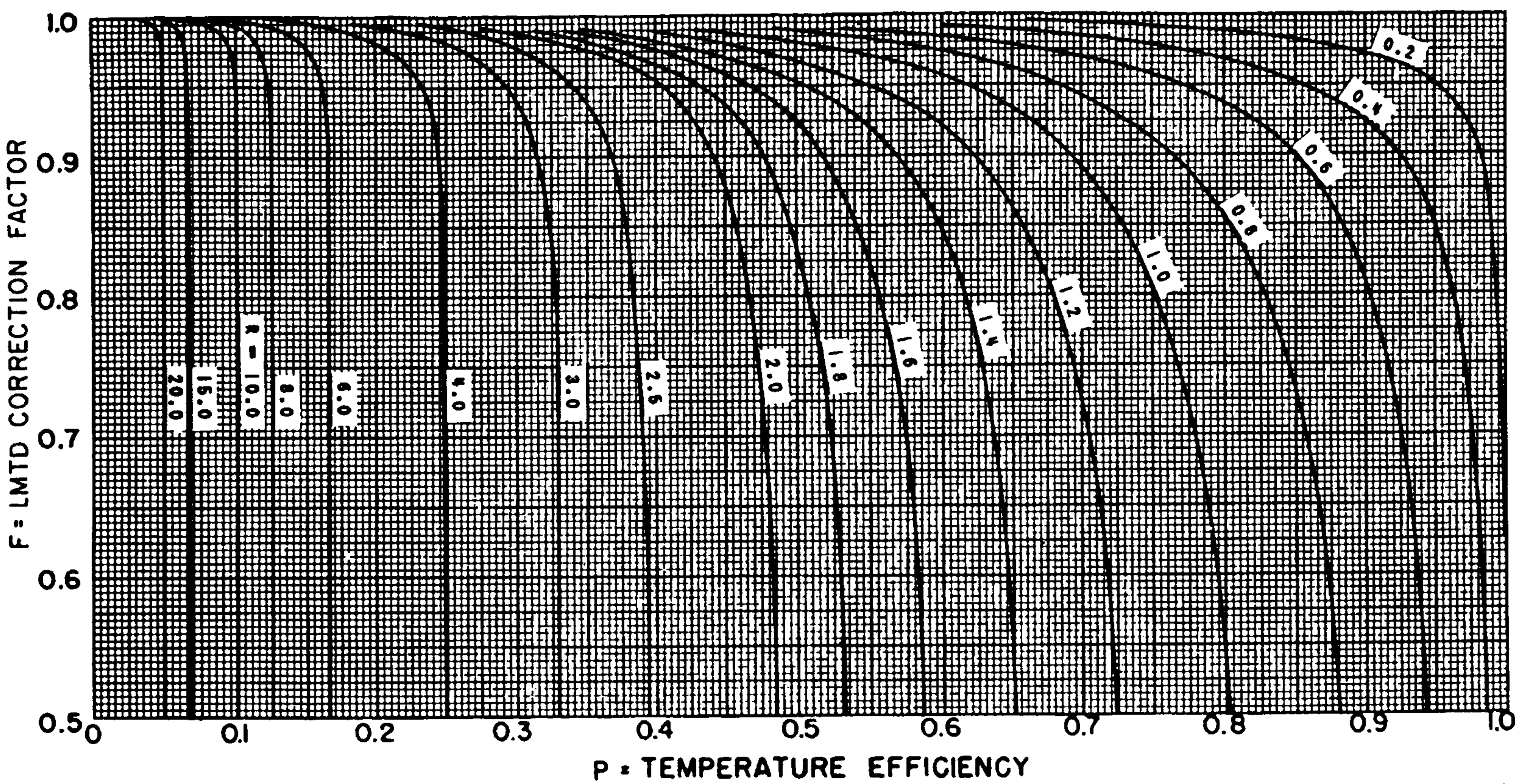
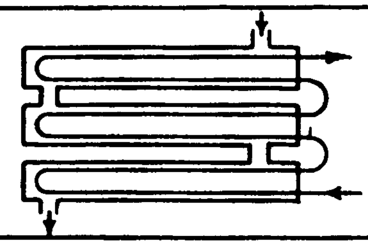


Figure 10-34B. MTD correction factor, 2 shell passes, 4 or a multiple of 4 tube passes.





LMTD CORRECTION FACTOR

3 SHELL PASSES 6 OR MORE EVEN NUMBER OF TUBE PASSES

$$P = \frac{t_2 - t_1}{T_1 - t_1} \qquad R = \frac{T_1 - T_2}{t_2 - t_1}$$

Figure 10-34C. MTD correction factor, 3 shell passes, 6 or more even number of tube passes.

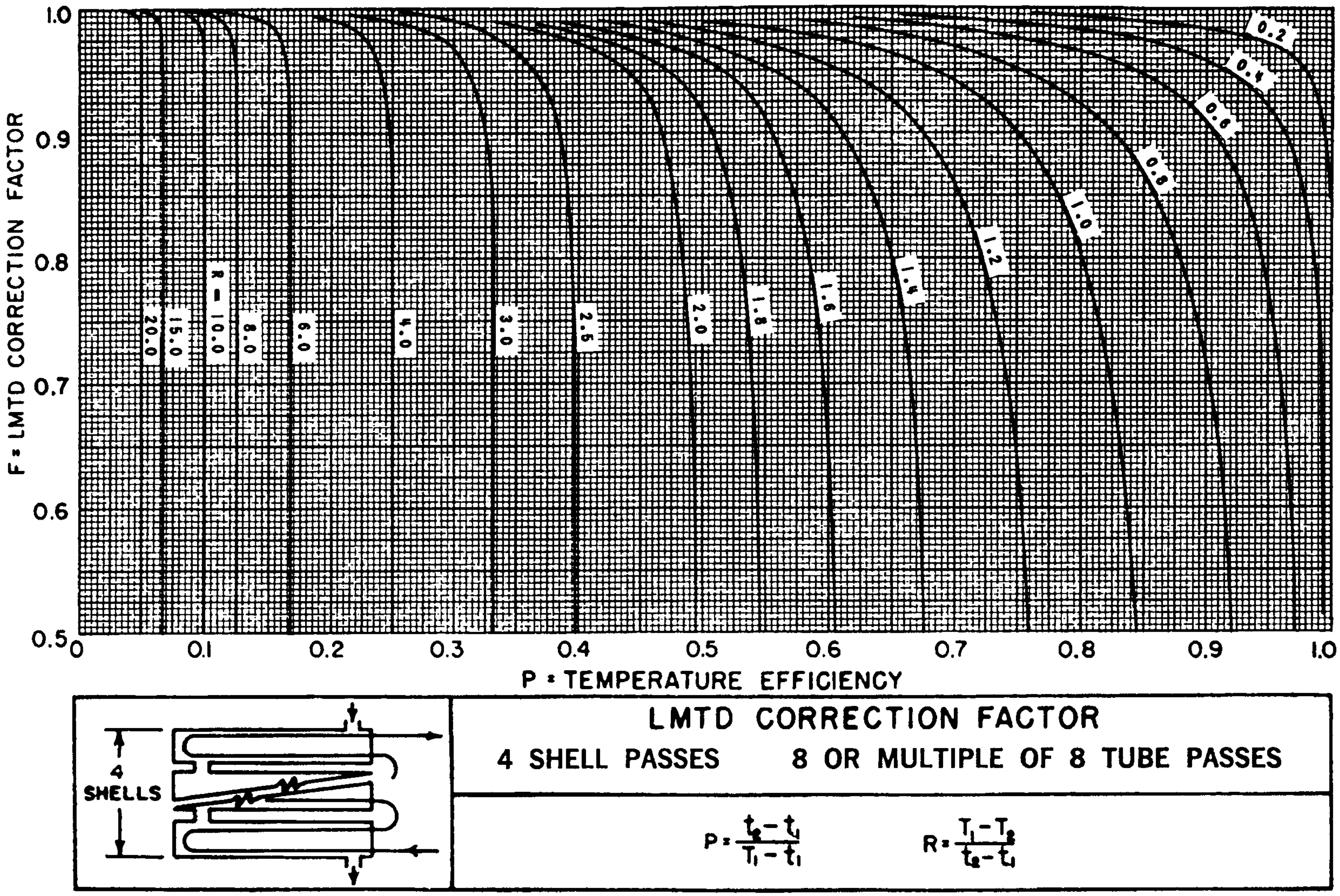


Figure 10-34D. MTD correction factor, 4 shell passes, 8 or a multiple of 8 tube passes.

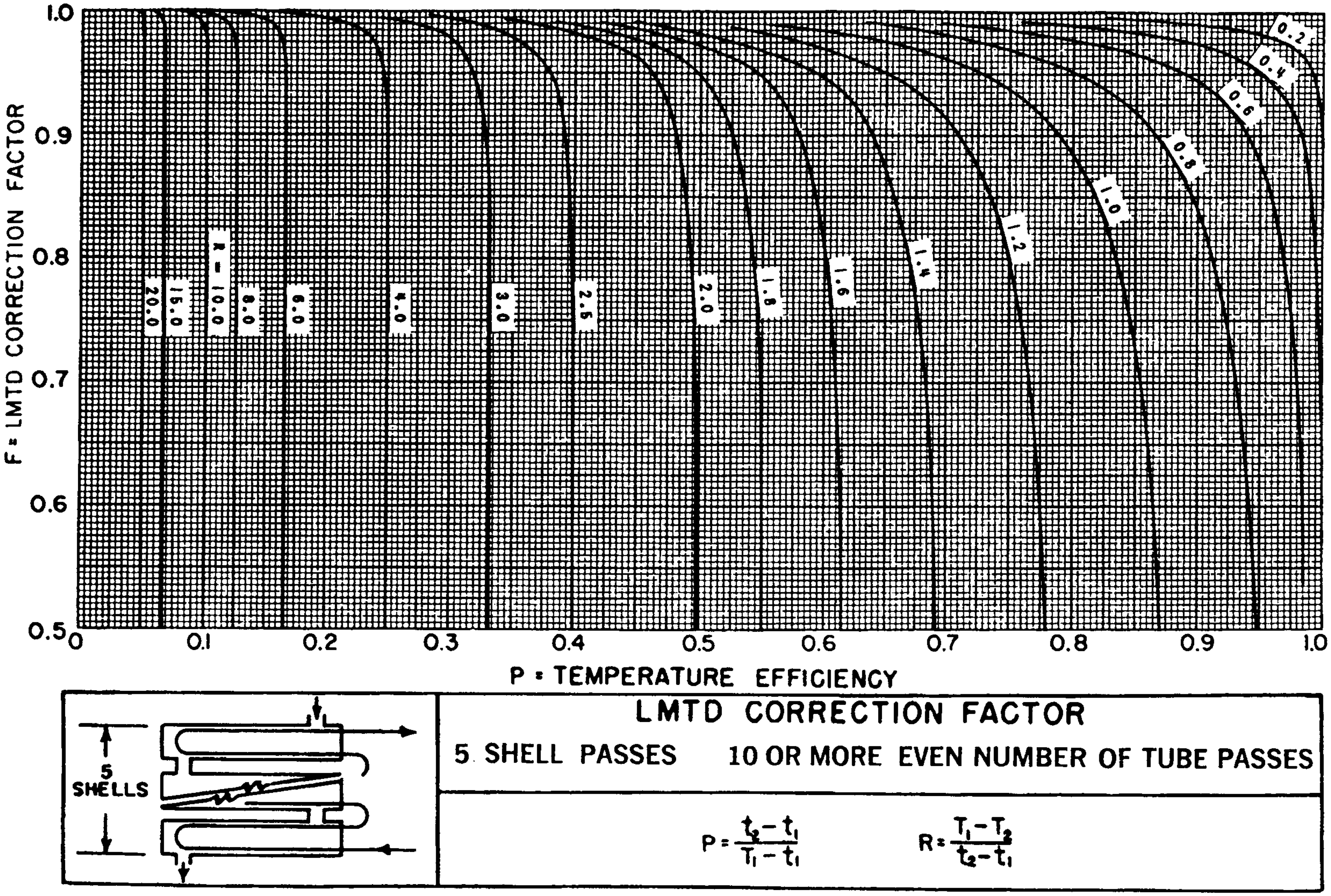


Figure 10-34E. MTD correction factor, 5 shell passes, 10 or more even number of tube passes.

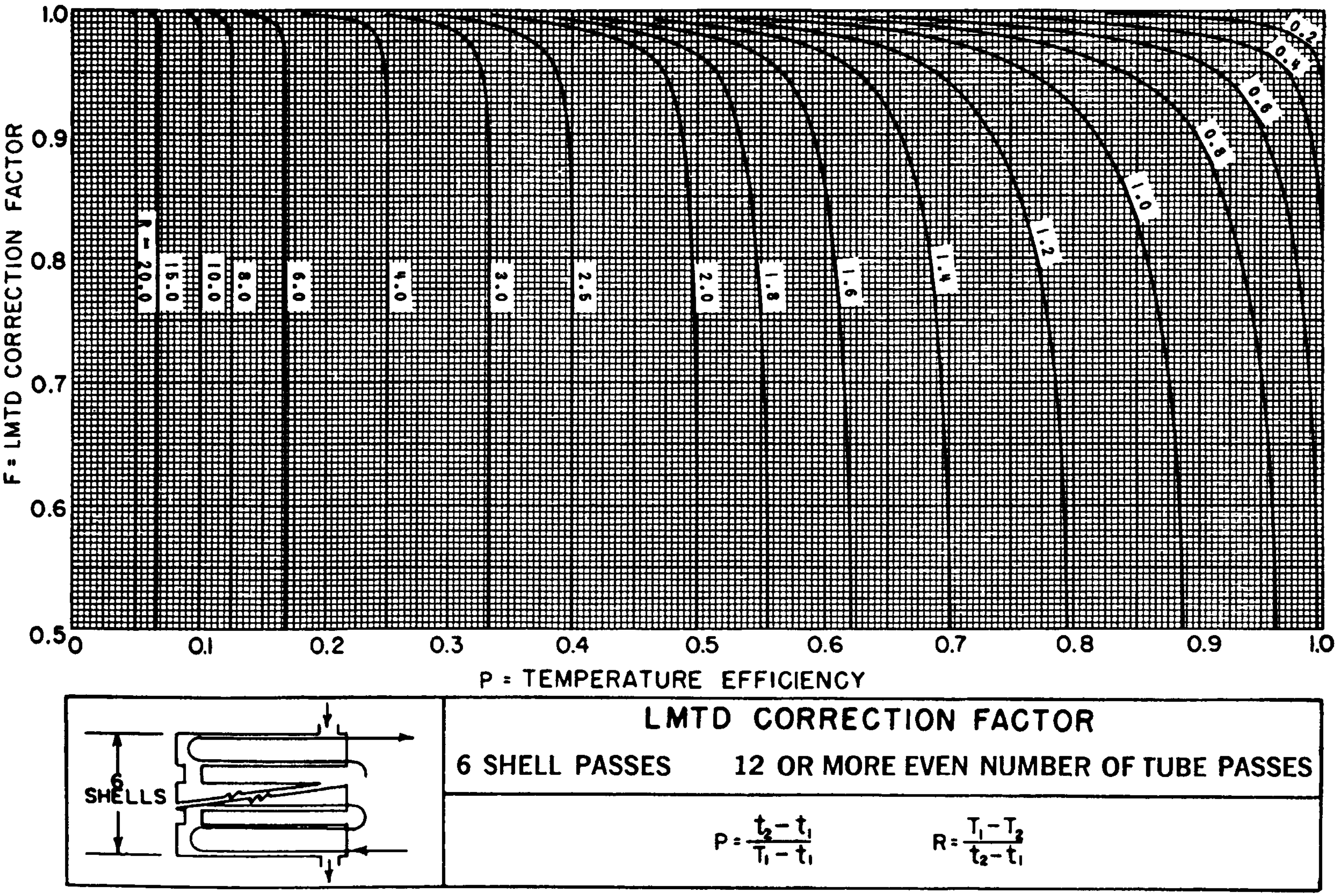
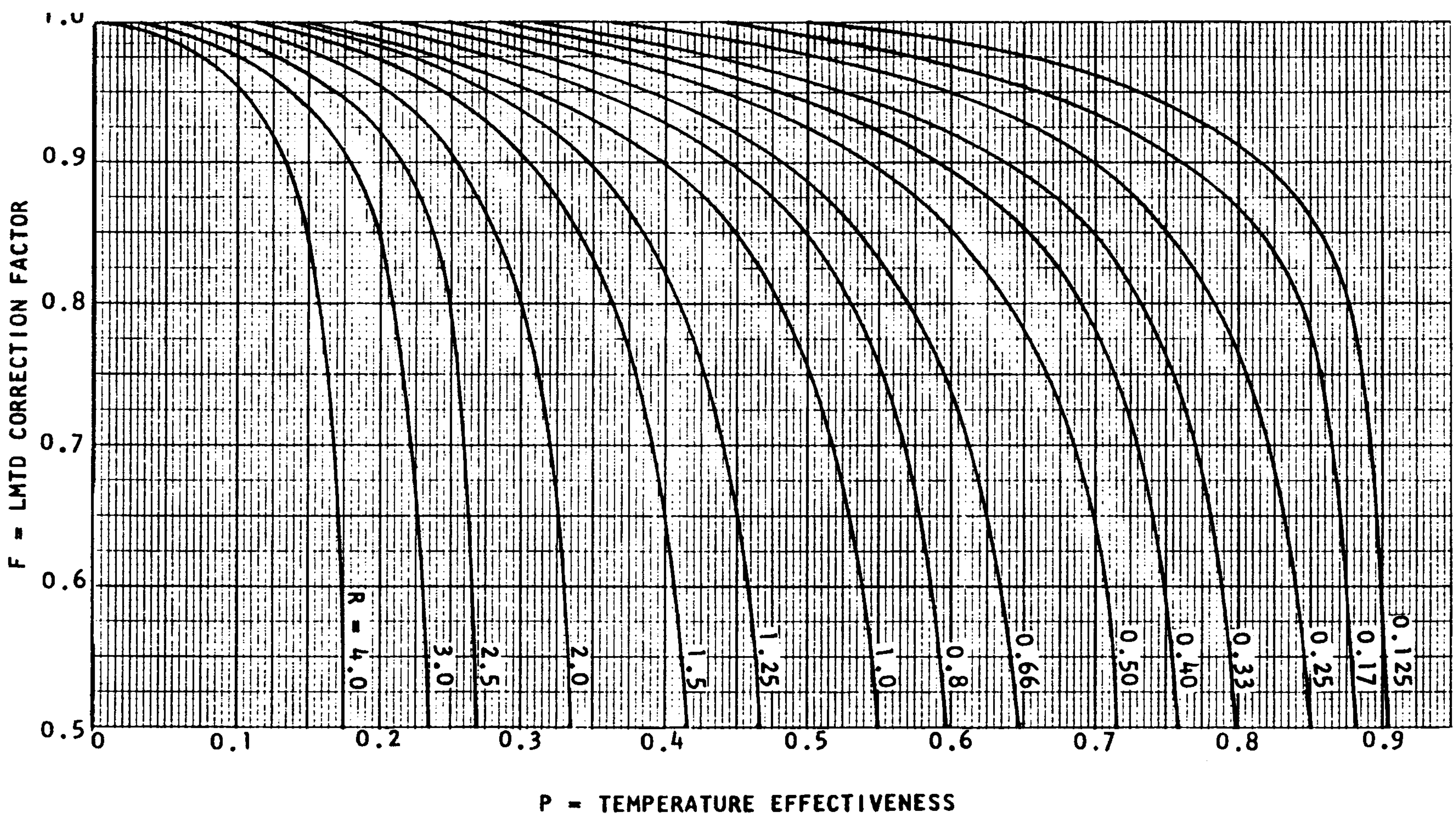
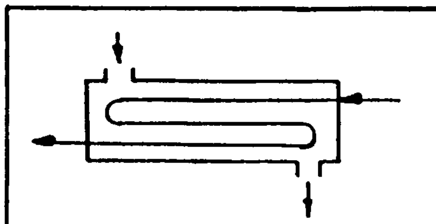


Figure 10-34F. MTD correction factor, 6 shell passes, 12 or more even number of tube passes.





LMTD CORRECTION FACTOR

1 SHELL PASS 3 TUBE PASSES (2 COUNTER AND 1 COCURRENT)

$$P = \frac{t_2 - t_1}{T_1 - t_1} \qquad R = \frac{T_1 - T_2}{t_2 - t_1}$$

Figure 10-34G. MTD correction factor, 1 shell pass, 3 tube passes (2 counter and 1 co-current).

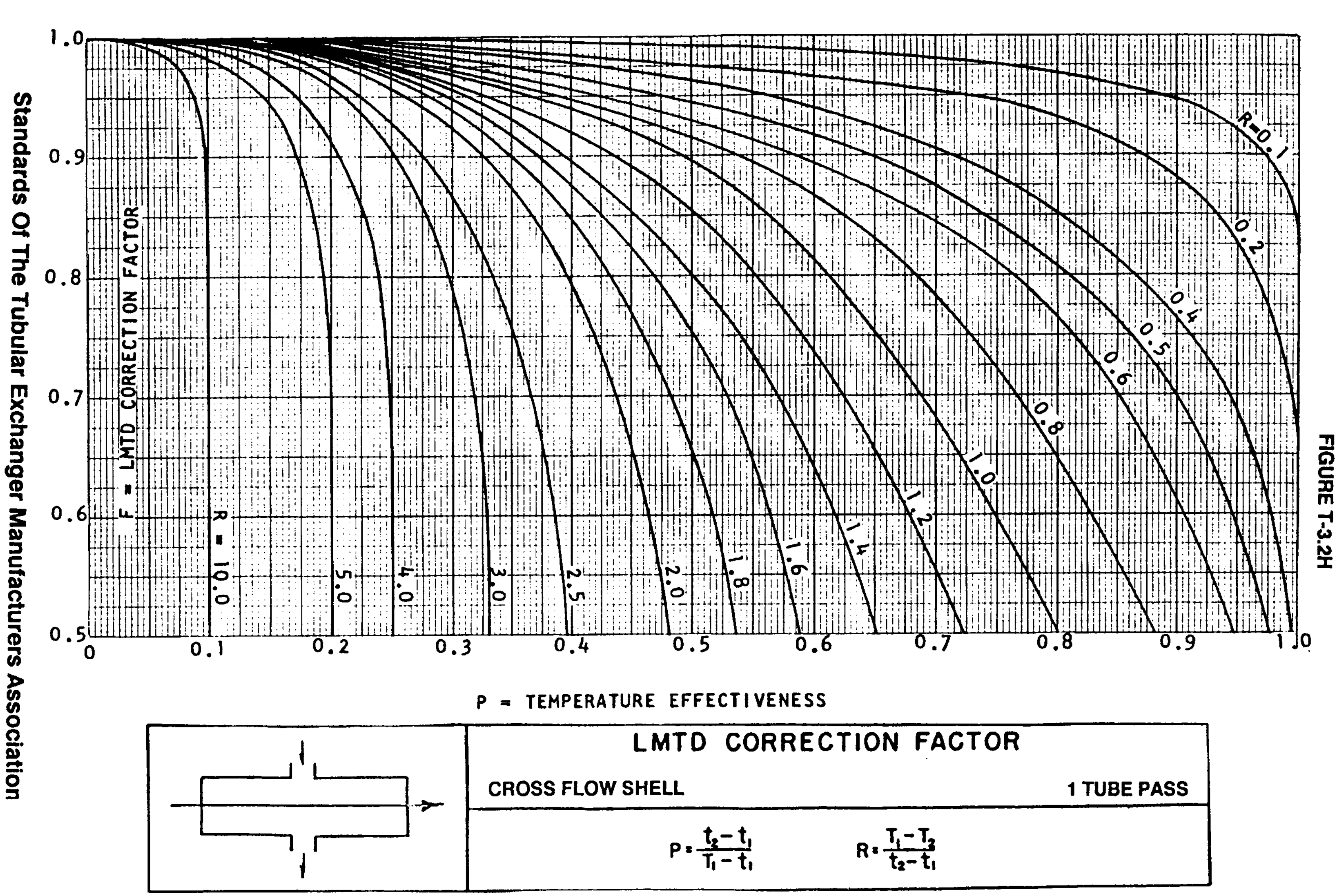


Figure 10-34H. MTD correction factor, crossflow shell, 1 tube pass.

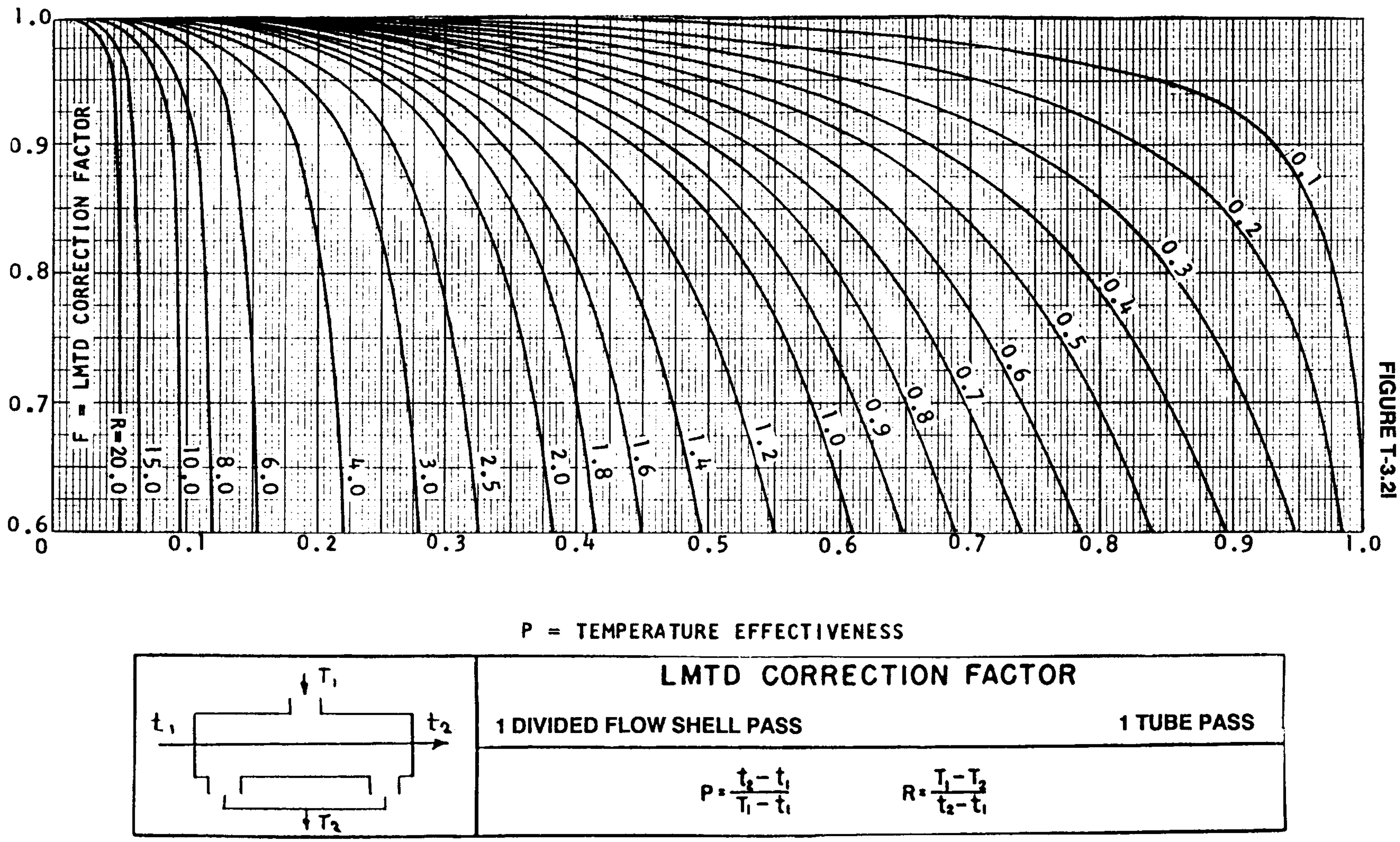


Figure 10-34I. MTD correction factor, 1 divided flow shell pass, even number of tube passes.

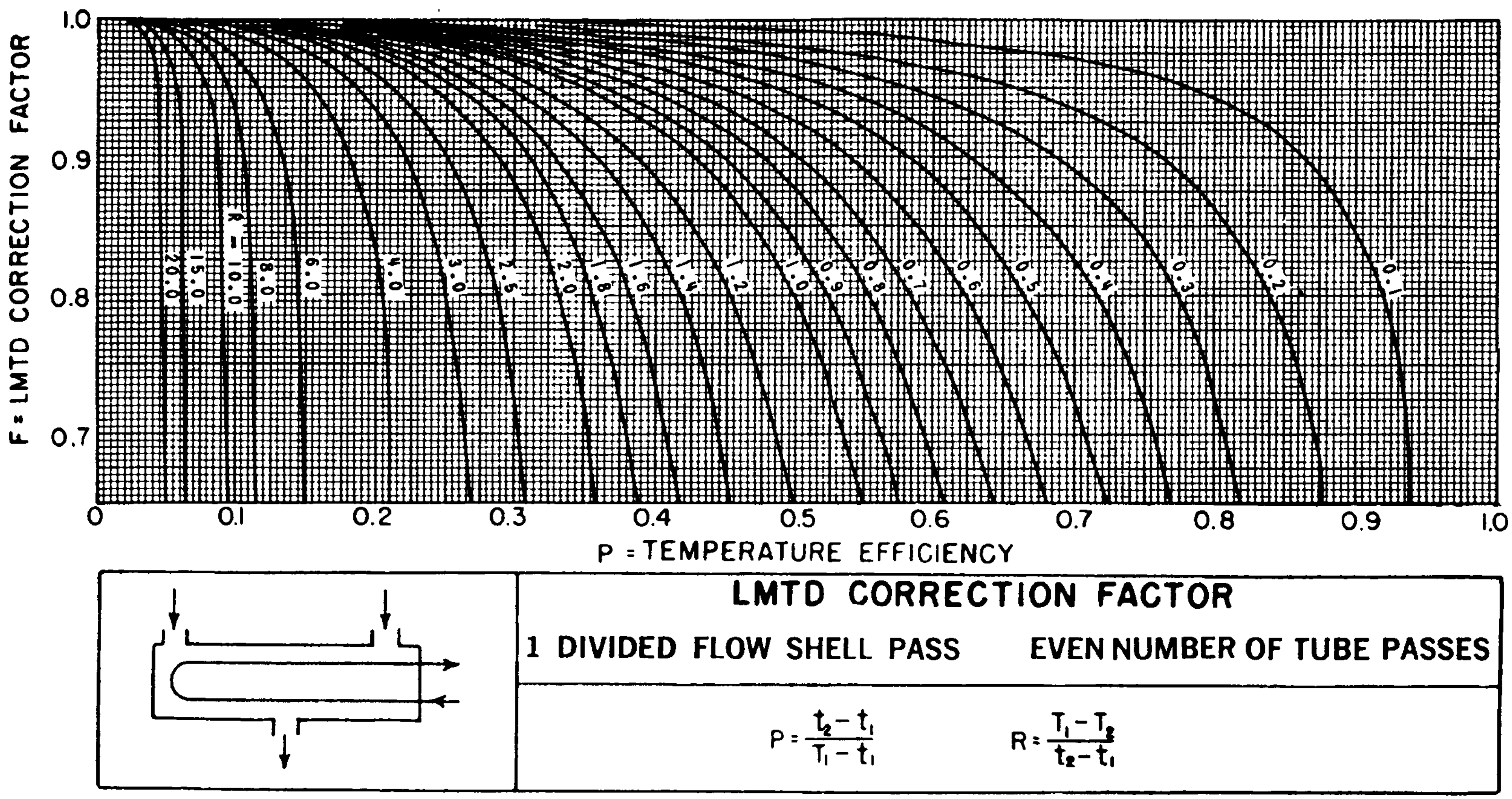


Figure 10-34J. MTD correction factor, split flow shell, 2 tube passes.

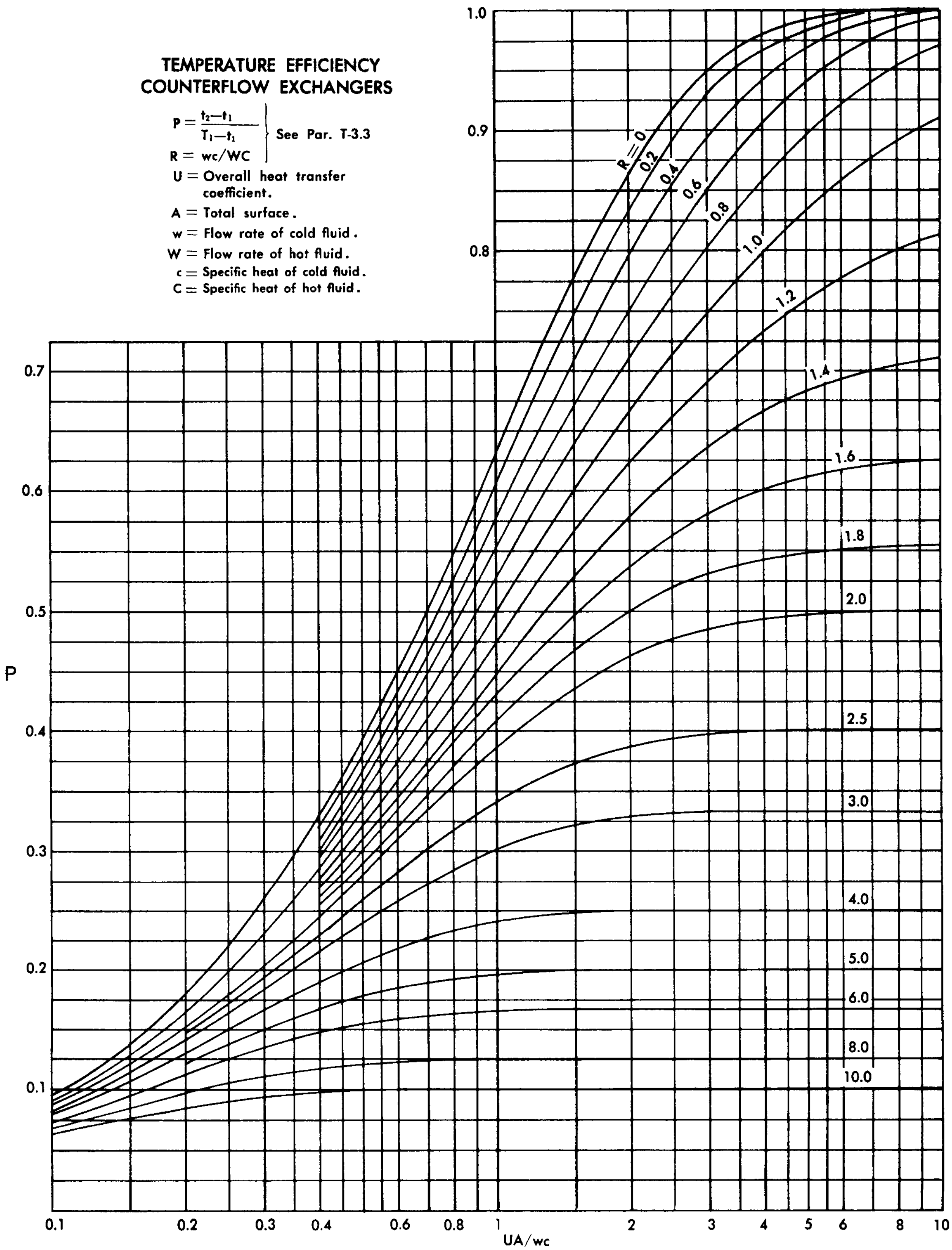


Figure 10-35A. Temperature efficiency for counterflow exchangers. (Used by permission: *Standards of Tubular Exchanger Manufacturers Association*, 7th Ed., Figure T-3.3, ©1988. Tubular Exchanger Manufacturers Association, Inc. All rights reserved.)

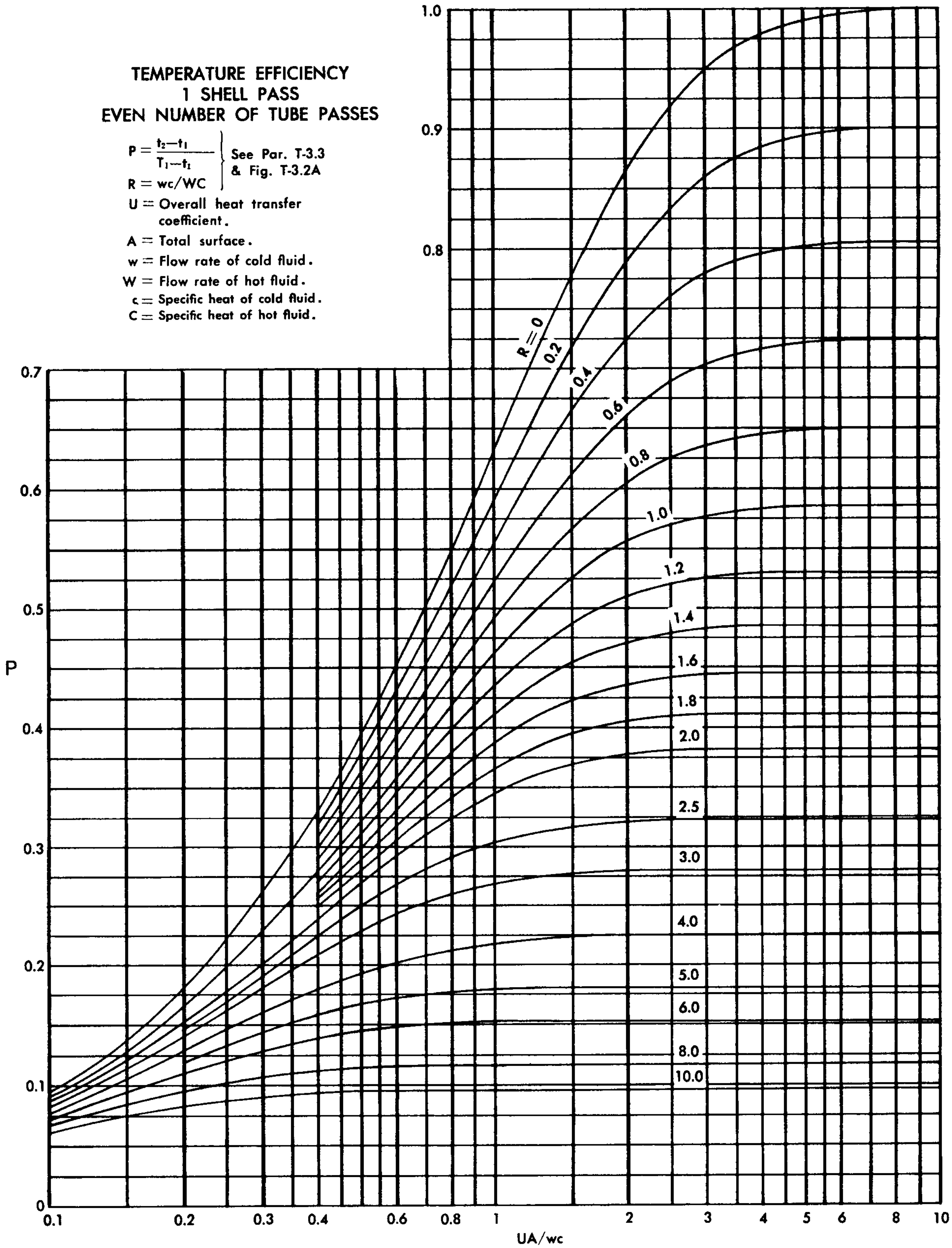


Figure 10-35B. Temperature efficiency, 1 shell pass, even number of tube passes. (Used by permission: *Standards of Tubular Exchanger Manufacturers Association*, 7th Ed., Figure T–3.3A, ©1988. Tubular Exchanger Manufacturers Association, Inc. All rights reserved.)

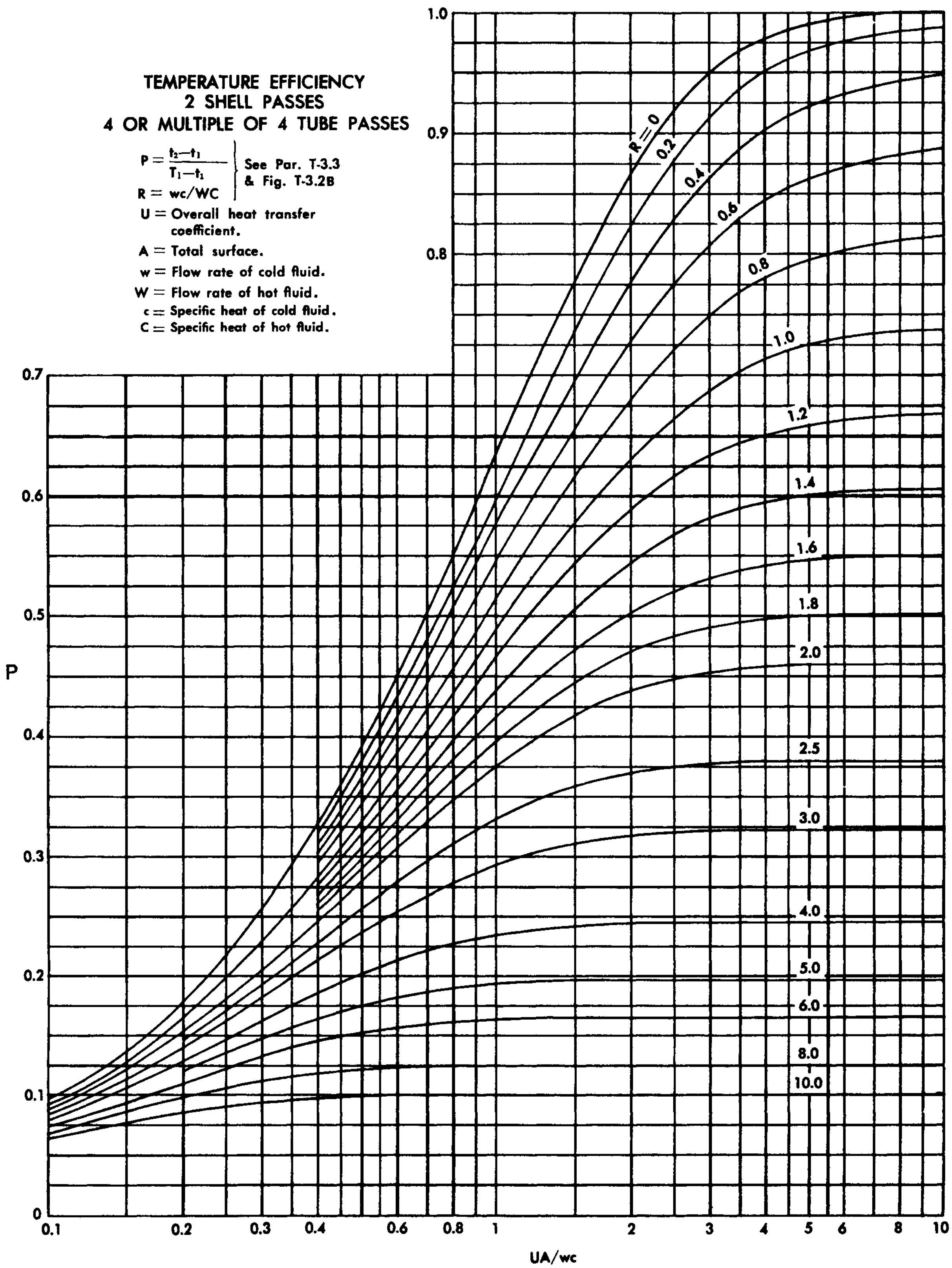


Figure 10-35C. Temperature efficiency, 2 shell passes, 4 or a multiple of 4 tube passes. (Used by permission: *Standards of Tubular Exchanger Manufacturers Association*, 7th Ed., Figure T-3.3B, ©1988. Tubular Exchanger Manufacturers Association, Inc. All rights reserved.)

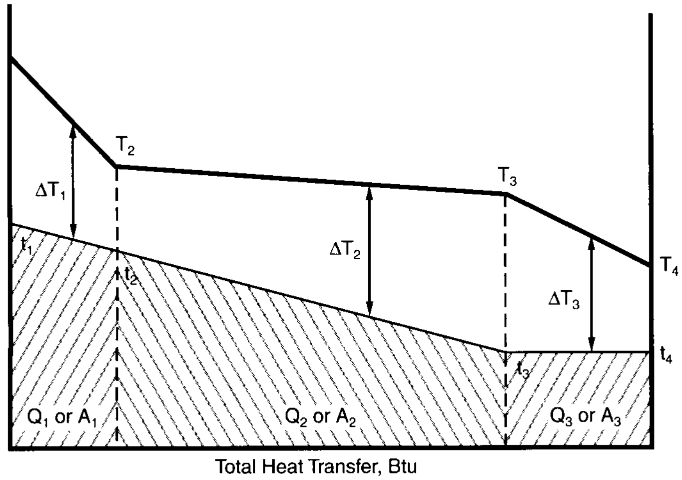


Figure 10-36. Breakdown of heat transfer zones in an exchanger.

difference and to calculate the total heat transfer area directly:

$$\Delta \text{ T (log-mean weighted)} = \frac{Q_{total}}{\frac{Q_1}{\Delta T_1} + \frac{Q_2}{\Delta T_2} + \frac{Q_3}{\Delta T_3}} \qquad (10\text{-}14)$$

where

Q = Heat transferred in specific section of the exchanger, Btu/hr

ΔT = Corresponding LMTD for the respective heat transfer area, °F

Subscripts 1, 2, and 3 = segments of heat exchanger corresponding to the Q and ΔT values.

These MTD (or LMTD) correction factors are read from the appropriate chart, which describes the exchanger mechanical and temperature terminal operational conditions. The P and R ratios must be calculated as represented in the diagrams; otherwise the factor read will have no meaning. A true counterflow or parallel flow exchanger does not require any correction to the LMTD.

Correction for Multipass Flow through Heat Exchangers

In most multipass exchangers, a combination of counter-current and co-current flow exists as the fluid flows through alternate passes (see Figure 10-29). The mean temperature is less than the logarithmic mean calculated for counter-current flow and greater than that based on co-current flow.

The corrected mean temperature difference is calculated:

$$(\text{LMTD})_{cor} = (\text{LMTD})_{calc}(\text{F}) \qquad (10\text{-}15)$$

F = Correction factor to LMTD for counter-current flow for various mechanical pass configurations, see Figures 10-34A–J.

Subscripts: cor = Corrected for determining exchanger area requirements.

calc = Calculated from Equation 10-13.

Use $(\text{LMTD})_{cor}$ in determining exchanger area requirements.

$$A_o = \frac{Q}{U(\text{LMTD})_{cor}} = Q/(U\Delta T_m) \qquad (10\text{-}16)$$

A_o = Required effective outside heat transfer surface area based on net exposed tube area. *Note*: Later in text A_o = A.

ΔT_m = Corrected mean temperature difference.

U = Overall heat transfer (fouled) coefficient, Equation 10-37

To determine the true overall temperature difference, the correction factors, F, shown in Figure 10-34 are used to correct for the deviations involved in the construction of multi-passes on the shell and tube sides of the exchanger. Note that R of the charts represents the heat capacity rate ratio[107], and P is the temperature efficiency of the exchanger.

$$\text{MTD}_{cor} = \Delta T_c = (\text{F})(\text{LMTD}) \qquad (10\text{-}17)$$

where

LMTD = defined by Equation 10-13

F = Correction factor as defined by the charts of Figure 10-34

Note: F = 1.0 for pure counter-current flow. As co-current flow increases in design arrangement (not flow rate), the F is reduced, and the exchanger efficiency falls, to a usual practical lower limit of 0.75–0.80[131]

Ratnam and Patwardhan[134] present graphs to aid in analyzing multipass exchangers, based on equations developed. Turton, et al. [135] also presents performance and design charts based on TEMA charts (Figure 10-34J) and combining these with Temperature Efficiency Charts[107] from TEMA (Figures 10-35A–C).

MTD (c) = ΔT_c = corrected LMTD for specific exchanger design/style

Example 10-4. Performance Examination for Exit Temperature of Fluids

In this example, we use the method of Turton[135], which incorporates combined working charts not included in this text.

A 1–2 (one shell, two tube passes) shell and tube exchanger is described as follows:

For shell side: M = 65,000 lb/hr

Cp_s = 0.58 Btu/lb (°F)

$T_1 = 210°F$

For tube side: $m = 155{,}000$ lb/hr

$Cp = 0.55$ Btu/lb (°F)

$t_1 = 135°F$; $t_2 = 168°F$

Now, estimate the exchanger performance if the hot fluid (shell-side) is to be increased 35% greater than the original design: A = 187 ft². Note that the method does not specifically incorporate fouling, but it should be acknowledged.

Determine the outlet temperature when U is established.

1. On shell side: M = 1.35 (65,000) = 87,750 lb/hr; the Cp_s and T_1 are kept the same.
2. On tube side: m, Cp, t_1, U, and A remain the same.
3. Now calculate $R = \dfrac{T_1 - T_2}{t_2 - t_1} = m\,Cp\,/MCPs$

   $= (155{,}000)(0.55) / (87{,}750)(0.58)$
   $= 1.675$
4. UA/(mc) = (170)(187) / (155,000)(0.55) = 0.37, see Figure 10-35.
5. Using Figure 10-35, at UA/mc = 0.37 and R = 1.675, read P = 0.25.
6. Now using Figure 10-34A, at P = 0.25 and R = 1.675, read F = 0.954.
7. Using P and R to find exit temperatures:
   $t_2 = P(T_1 - t_1) + t_1 = 0.25\,(210 - 133) + 133 = 152°F$
   $T_2 = T_1 - R(t_2 - t_1) = 210 - 1.675\,(152 - 133) = 178°F$

This same concept incorporating the TEMA charts can be used to (1) determine the ft² heat transfer area required for an exchanger and (2) determine flow rate and outlet temperature of the fluids (shell or tube side)[135].

where

A = Heat exchanger surface area, ft².
Cp = Specific heat capacity (tube or cold side), Btu/(lb)(°F)
Cp_s = Specific heat capacity (shell or hot side), Btu/(lb)(°F)
F = MTD correction factor, Figure 10-34.
F = Heat exchanger efficiency, dimensionless.
m = Mass flow rate (tube or cold side), lb/hr.
M = Mass flow rate (shell or hot side), lb/hr.
P = $(t_2 - t_1) / (T_1 - t_1)$, Figure 10-34.
q = Rate of heat transfer, Btu/hr.
R = Ratio of the heat capacities of tube-side to shell-side fluid, dimensionless

$= \dfrac{(T_1 - T_2)}{(t_2 - t_1)}$, Figure 10-34.

t = Temperature of tube-side fluid, °F.
T = Temperature of shell-side fluid, °F.
ΔT_{lm} = Log mean temperature difference for countercurrent flow, °F.
U = Overall heat transfer coefficient in exchanger, Btu/(ft²)(hr)(°F).

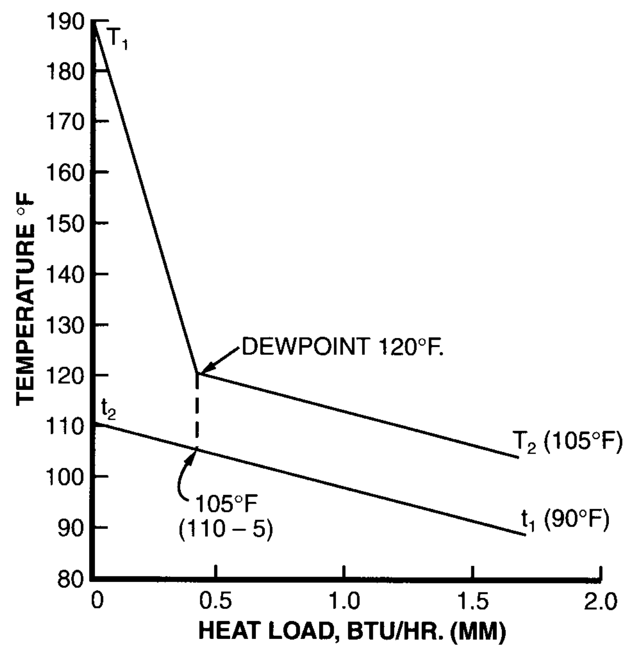


Figure 10-37. Finding a counterflow weighted MTD. (Reprinted with permission: Gulley, Dale E., *Heat Exchanger Design Handbook*, © 1968 by Gulf Publishing Company, Houston, Texas. All rights reserved.)

Subscripts:

1 = Inlet
2 = Outlet

Weighted Mean Temperature Difference (MTD) applies to the more complicated shell and tube heat exchangers. Gulley[59] discusses several important cases in which the conventional LMTD for fluid temperature change and the corresponding MTD correction factors (Figure 10-34A–J) do not adequately represent the design requirements, see Figure 10-37. From the sample listing that follows, recognize that the heat release for each section of an exchanger is necessary to properly analyze the condition. It can be misleading if end point conditions previously cited are used to describe some of the special cases. It is necessary to break the heat transfer calculations into zones and calculate the weighted MTD. Typical services requiring the use of weighted MTS's are[59, 70]

1. Overhead condensers with steam and hydrocarbon condensing.
2. Exchangers with change of phase.
3. Amine overhead condensers.
4. Pure component condensers with subcooling.
5. Condensers with large desuperheating zones such as for refrigerants, chemicals, and steam.
6. Pure component vaporizing with superheating.
7. Vertical reboilers in vacuum service.
8. Desuperheating-condensing-subcooling.
9. Condensing in presence of noncondensable gases.

Example 10-5. Calculation of Weighted MTD[59]

A gas is to be cooled from 190°F to 105°F with partial condensation taking place. The dew point is 120°F. The cooling water enters at 90°F and leaves at 110°F. Figure 10-37 illustrates the basic exchanger functions.

1. The heat load is

$$\begin{aligned} Q\text{ (Desuperheat)} &= 420{,}000 \text{ Btu/hr} \\ Q\text{ (Condensation)} &= \underline{1{,}260{,}000 \text{ Btu/hr}} \\ & 1{,}680{,}000 \text{ Btu/hr} \end{aligned}$$

2. The cooling water temperature rise in each zone is

$$\begin{aligned} \text{Desuperheating: } \Delta T, °F &= (110 - 90)\ (420{,}000)/1{,}680{,}000 \\ &= 5°F \\ \text{Condensation: } \Delta T, °F &= (110 - 90)\ (1{,}260{,}000)/1{,}680{,}000 \\ &= 15°F \end{aligned}$$

3. For desuperheating, the LMTD is

$$\begin{array}{ll} 190 \rightarrow & 120 \\ \underline{110} \leftarrow & \underline{90} \\ 80 & 15 \end{array}$$

Reading Figure 10-33:
LMTD = 38.8°F

4. For condensation, the LMTD is

$$\begin{array}{ll} 120 \rightarrow & 105 \\ \underline{105} \leftarrow & \underline{90} \\ 15 & 15 \end{array}$$

LMTD = 15°F

5. The weighted LMTD for calculations is

$$\text{Wt. LMTD} = \frac{Q_{total}}{\dfrac{Q_{des}}{(\text{LMTD})_{des}} + \dfrac{Q_{cond}}{(\text{LMTD})_{cond}}}$$

$$= \frac{1{,}680{,}000}{\dfrac{420{,}000}{38.8} + \dfrac{1{,}260{,}000}{15}} = 17.7°F$$

For shells in series it is necessary to develop a weighted MTD correction factor, and a graphical technique is presented by Gulley.[59]

Heat Balance

In heat exchanger design, the exchange of the heat between fluids is considered to be complete (i.e., 100%) except in those cases when heat losses to the atmosphere or other outside medium are either known or planned.

Plant[130] presented a technique for comparative heat exchange performance evaluation that is based on his efficiency method and can include almost any style and application of exchanger.

In condensers where heat loss is desired, insulation often is omitted from piping carrying hot fluids to take advantage of the heat loss to the atmosphere. In any heat exchange equipment the heat released or lost by one fluid must be accounted for in an equivalent gain by a second fluid, provided that heat losses are negligible or otherwise considered.

Heat Load or Duty

The heat load on an exchanger is usually determined by the process service conditions. For example, the load on a condenser for vapors from a distillation column is determined by the quantity and latent heat of vaporization at the condensing conditions, or for gas coolers, by the flow of gas and the temperature range required for the cooling.

For sensible heat changes:

$$q = W c_p (t_2 - t_1) \tag{10-18}$$

For latent heat changes:

$$q = W l_v \tag{10-19}$$

For cooling (or heating) and latent heat change (condense or boil):

$$Q = q' = Wc_p (t_2 - t_1) + Wl_v \tag{10-20}$$

An item of heat exchange equipment can be used for any of these heat changes, or any combination of them, provided the loads are established to correspond with the physical and thermal changes actually occurring or expected to occur in the unit. Thus, the heat load must be known for the design of an exchanger, although it may be determined on existing equipment from operating data. This latter is termed *performance evaluation.*

Example 10-6. Heat Duty of a Condenser with Liquid Subcooling

The overhead condenser on a distillation column is to subcool the condensed vapors from the condensation temperature of 46.4°F down to 35°F. The specific heat of the liquid is 0.3 Btu/lb (°F), and the latent heat of vaporization at 46.4°F is 265 Btu/lb. The vapor rate to the condenser is 740.3 lb/hr. What is the total heat load on the condenser?

Latent duty: $q_1 = 740.3\,(265) = 196{,}180$ Btu/hr

Sensible duty: $q_2 = 740.3\,(0.3)\,(46.4 - 35) = 2532$ Btu/hr

Total heat duty: $Q = q_1 + q_2 = 198{,}712$ Btu/hr

Transfer Area

The heat transfer area, A ft^2, in an exchanger is usually established as the outside surface of all the plain or bare tubes or the total finned surface on the outside of all the finned tubes in the tube bundle. As will be illustrated later, factors that inherently are a part of the inside of the tube (such as the inside scale, transfer film coefficient, etc.) are often corrected for convenience to equivalent outside conditions to be consistent. When not stated, transfer area in conventional shell and tube heat exchangers is considered as *outside tube area.*

Example 10-7. Calculation of LMTD and Correction

An oil cooler is to operate with an inlet of 138°F and an outlet of 103°F, and the cooling water enters at 88°F and is to be allowed to rise to 98°F. What is the corrected MTD for this unit, if it is considered as (a) a concentric pipe counterflow unit, (b) a single-pass shell–two-pass tube unit, and (c) a parallel flow unit?

1. Diagram the temperature pattern

(a) Counterflow

$$T_1 = 138°F \rightarrow 103°F = T_2$$
$$t_2 = 98°F \leftarrow 88°F = t_1$$
$$\Delta t_2 = 40°F \qquad \Delta t_1 = 15°F$$

$$\text{LMTD} = \frac{40 - 15}{\ln \frac{40}{15}} = 25.5°F$$

(Calculate or read from Figure 10-33.)

(b) Shell and tube 1–2 (1 pass shell–2 pass tubes)

$$T_1 = 138°F \rightarrow 103°F = T_2$$
$$t_2 = 98°F \leftarrow 88°F = t_1$$
$$\Delta t_2 = 40°F \qquad \Delta t_1 = 15°F$$

$$\text{LMTD} = \frac{40 - 15}{\ln \frac{40}{15}} = 25.5°F$$

(c) Parallel flow

$$T_1 = 138°F \rightarrow 103°F = T_2$$
$$t_1 = 88°F \rightarrow 98°F = t_2$$
$$\Delta t_2 = 50°F \qquad \Delta t_1 = 5°F$$

$$\text{LMTD} = \frac{50 - 5}{\ln \frac{50}{5}} = 19.5°F$$

2. Corrections to LMTD (See Figure 10-34.)

$$P = \frac{t_2 - t_1}{T_1 - t_1} \qquad R = \frac{T_1 - T_2}{t_2 - t_1}$$

(a) Counterflow

No correction factor, F = 1.0, corrected LMTD = 25.5°F.

(b) Shell and tube, 1–2 (This is part parallel, part counterflow.)

$$P = \frac{98 - 88}{138 - 88} = \frac{10}{50} = 0.2$$

$$R = \frac{138 - 103}{98 - 88} = \frac{35}{10} = 3.5$$

F read from Figure 10-34A = 0.905

Corrected LMTD = (0.905) (25.5) = 23.1°F

(c) Parallel flow

Correction factor does not apply,

LMTD = 19.5 ∘ F

3. For an exchanger design, the unit requiring the smallest area will be the counterflow having the largest corrected LMTD = 25.5°F for this example.

Correction factors should seldom be used when they fall below a value that lies on a curved portion of the P–R curves, Figure 10-34. That is, values on the straight portions of the curves have little or no accuracy in most cases. For the "single-shell pass—two or more than two passes" unit chart, an F of less than about 0.8 would indicate consideration of a two-shell pass unit. As a general guide, F factors less than 0.75 are not used. To raise the F factor, the unit flow system, temperature levels, or both must be changed.

Temperature for Fluid Properties Evaluation—Caloric Temperature

For most exchanger conditions, the arithmetic mean temperatures of the shell side and tube side, respectively, are satisfactory to evaluate the properties of the fluids, which in turn can be used to determine the overall coefficient, U. This means that the LMTD as determined is correct.

When you can determine that the overall coefficient U or fluid properties vary markedly from the inlet to the exit conditions of the unit, the arithmetic mean is no longer satisfactory for fluid property evaluation. *For this case, the proper temperature of each stream is termed the caloric temperature for each fluid.* The F fraction is the smallest of the values calculated and applies to both streams. Although the caloric temperature applies to counter-current and parallel flow only, it can be used with reasonable accuracy for multipass flow.[107]

The caloric value of hot fluid (from Kern,[70] by permission):

$$t_h = t_{h2} + F_c(t_{h1} - t_{h2}) \qquad (10\text{-}21)$$

The caloric value of the cold fluid:

$$t_c = t_{c1} + F_c(t_{c2} - t_{c1}) \qquad (10\text{-}22)$$

$$F_c = \frac{t_c - t_1}{t_2 - t_1} \text{ or } = \frac{t_h - t_{h1}}{t_{h2} - t_{h1}} \qquad (10\text{-}23)$$

where

t_h = caloric value or hot fluid, °F
t_{h1} = inlet hot fluid temperature, °F
t_{h2} = outlet hot fluid temperature, °F
t_c = caloric value of cold fluid, °F
t_{c1} = inlet cold fluid temperature, °F
t_{c2} = outlet cold fluid temperature, °F
F_c = correction factor, F, (see Reference 70 for details), Figure 10-38. The insert allows for more rapid calculation for petroleum fractions.

For heat exchangers in true counter-current (fluids flowing in opposite directions inside or outside a tube) or true co-current (fluids flowing inside and outside of a tube, parallel to each other in direction), with essentially constant heat capacities of the respective fluids and constant heat transfer coefficients, the log mean temperature difference may be appropriately applied, see Figure 10-33.[107]

For a variation in heat transfer coefficient from one end of the exchanger to the other where the average fluid temperature is considered approximately linear, the physical properties of the fluids can be approximated by evaluating them using Figure 10-38. To use this figure, the temperature change of each fluid multiplied by the F factor from the chart is added to its respective *cold* terminal temperature to obtain the average temperature. The Δt_c and Δt_h from the figure represent the cold and hot terminal temperature differences, and C of the chart represents the fractional change in heat transfer coefficient as a parameter of the chart. For hydrocarbon fluids, the C may be simplified using the insert in Figure 10-38.[107] Other less frequently used exchanger arrangements are discussed by Gulley.[129] Note that the F factor for Figure 10-38 is *not* the same F factor as given in Figures 10-34A–J.

The C ratio (disregard sign if negative) is evaluated from the estimated overall coefficients based on the temperatures at the cold and hot ends, respectively. For Figure 10-38, the hot terminal difference is $t_h = t_{h1} - t_{c2}$; the cold terminal temperature difference is $t_c = t_{h2} - t_{c1}$.

Tube Wall Temperature

Refer to Figure 10-28. The temperature of the outside of the tube wall is based on hot fluid being on the outside of the tubes:

$$t_w = t_h - \frac{h_{io}}{h_{io} + h_o}(t_h - t_c) \qquad (10\text{-}24)$$

or

$$t_w = t_c + \frac{h_o}{h_{io} + h_o}(t_h - t_c) \qquad (10\text{-}25)$$

where

h_{io} = inside film coefficient referred to outside of tube, Btu/hr (ft^2) (°F)
h_o = outside film coefficient referred to outside of tube, Btu/hr (ft^2) (°F)
t_w = temperature of outside wall of tube, °F

The outside tube wall temperature for hot fluid on the inside of the tubes is

$$t_w = t_c + \frac{h_{io}}{h_{io} + h_o}(t_h - t_c) \qquad (10\text{-}26)$$

or

$$t_w = t_h - \frac{h_o}{h_{io} + h_o}(t_h - t_c) \qquad (10\text{-}27)$$

An alternate and possibly less accurate calculation (but not requiring the calculation of caloric temperature) using reasonably assumed or calculated film coefficients and bulk rather than caloric temperature is

For hot fluid on shell side:

$$t_w = t_c + \left[\frac{h_o}{h_i + h_o}\right](t_h - t_c) \qquad (10\text{-}28)$$

t_w = the outside surface temperature of wall

For hot fluid in tubes:

$$t_w = t_c + \left[\frac{h_i}{h_i + h_o}\right](t_h - t_c) \qquad (10\text{-}29)$$

where

t_w = tube wall temperature, neglecting fouling and metal wall drop, °F
t_c = cold fluid bulk temperature, °F
t_h = hot fluid bulk temperature, °F

Often this may be assumed based upon the temperature of the fluids flowing on each side of the tube wall. For a more accurate estimate, and one that requires a trial-and-error solution, neglecting the drop-through tube metal wall (usually small):

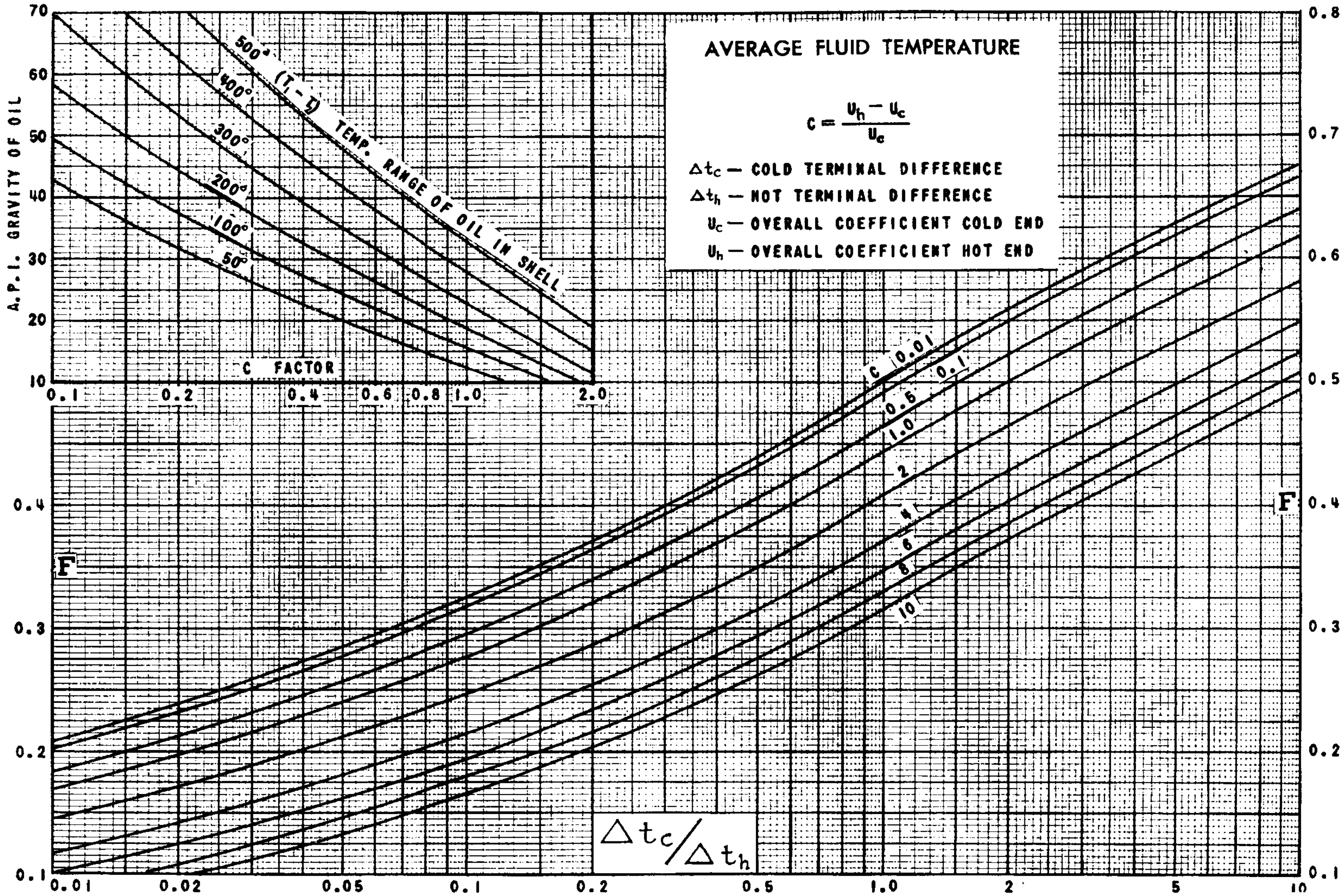


Figure 10-38. Caloric or true average fluid temperature. (Used by permission: *Standards of Tubular Exchanger Manufacturers Association*, ©1959 and 1968. Tubular Exchanger Manufacturers Association, Inc. All rights reserved.)

$$t_w = \frac{h_i t_i + h_o t_o}{h_i + h_o} \quad (10\text{-}30)$$

where

t_i = bulk temperature of fluid inside tube
t_o = bulk temperature of fluid outside tube
h_i = film coefficient for fluid inside tube
h_o = film coefficient of fluid outside tube

To determine a reasonably good value for t_w, either t_w must be estimated and used to calculate h_i and h_o, or h_i and h_o must be assumed and a t_w calculated. In either case, if the calculated values do not check reasonably close to the assumed values, the new calculated results should be used to recalculate better values. Film temperatures are generally taken as the arithmetic average of the tube wall, t_w, and the bulk temperature, t_i or t_o. This approach neglects the effect of tube wall fouling (i.e., it is for clean tube conditions). Corrections can be made to account for the fouling if considered necessary. Usually this fouling is accounted for in the overall U.

The temperature for calculating film properties is as follows.

For streamline flow:

$$t_f = t_{av} + 1/4\,(t_w - t_{av}) \quad (10\text{-}31)$$

For turbulent flow:

$$t_f = t_{av} + 1/2\,(t_w - t_{av}) \quad (10\text{-}32)$$

Ganapathy[161] presents a shortcut technique for estimating heat exchanger tube wall temperature, which so often is needed in establishing the fluid film temperature at the tube wall:

$$t_w = (h_i t_i + h_o t_o)/(h_i + h_o) \quad (10\text{-}33)$$

where

h_i = inside tube coefficient, Btu/(hr) (ft^2) (°F)
h_o = outside tube film coefficient, Btu/(hr) (ft^2) (°F)
t_i = temperature of inside fluid entering tubes, °F
t_o = temperature of outside fluid on tubes °F

For example, a hot flue gas flows outside a tube and shell exchanger at 900°F (t_o) while a hot liquid is flowing into the tubes at 325°F (t_i). The film coefficients have been estimated to be h_i = 225°F and h_o = 16 Btu/(hr) (ft^2) (°F). Estimate the tube wall temperature using h_i as h_{io} corrected to the outside surface for the inside coefficient:

$$t_w = [(225)(325) + (16)(900)]/(225 + 16) = 363°\text{F}$$

This calculation neglects the temperature drop across the metal tube wall and considers the entire tube to be at the temperature of the outside surface of the wall, t_w. Kern[70] for the same equation suggests using the caloric temperature for t_i and t_o.

Edmister and Marchello[165] present a tube wall temperature equation:

$$t_2 = \frac{D_i h_i t_i + D_o h_o t_4}{D_i h_i + D_o h_o},\ °\text{F} \quad (10\text{-}34)$$

where

t_1 = fluid No. 1 mean fluid temperature, °F
t_4 = fluid No. 2 mean fluid temperature, °F
t_2 = fluid No. 1 fluid film tube wall temperature, °F
t_3 = fluid No. 3 fluid film tube wall temperature, °F
D_o = tube outside diameter, in.
D_i = tube inside diameter, in.
h_i = inside tube film (surface) coefficient, Btu/(hr) (ft^2) (°F)
h_o = outside tube film (surface) coefficient, Btu/(hr) (ft^2) (°F)

This relationship can be used for estimating surface temperature and for back-checking estimating assumptions. For many situations involving liquids and their tube walls, the temperature difference ($t_2 - t_3$) across the wall is small and equals $t_2 - t_3$ for practical purposes.

Fouling of Tube Surface

Most process applications involve fluids that form some type of adhering film or scale onto the surfaces of the inside and outside of the tube wall separating the two systems (Figure 10-28). These deposits may vary in nature (brittle, gummy), texture, thickness, thermal conductivity, ease of removal, etc. Although no deposits are on a clean tube or bundle, the design practice is to attempt to compensate for the reduction in heat transfer through these deposits by considering them as resistances to the heat flow. These resistances or fouling factors have not been accurately determined for very many fluids and metal combinations, yet general practice is to "throw in a fouling factor." This can be disastrous to an otherwise good technical evaluation of the expected performance of a unit. Actually, considerable attention must be given to such values as the temperature range, which affects the deposit, the metal surface (steel, copper, nickel, etc.) as it affects the adherence of the deposit, and the fluid velocity as it flows over the deposit or else moves the material at such a velocity as to reduce the scaling or fouling.

The percentage effect of the fouling factor on the effective overall heat transfer coefficient is considerably more on units with the normally high value of a clean unfouled coefficient than for one of low value. For example, a unit with a clean overall coefficient of 400 when corrected for 0.003 total fouling ends up with an effective coefficient of 180, but a unit with a clean coefficient of 60, when corrected for a 0.003 fouling allowance, shows an effective coefficient of 50.5 (see Figure 10-39).

Fouling factors as suggested by TEMA[107] are shown in Table 10-12. These values are predominantly for petroleum

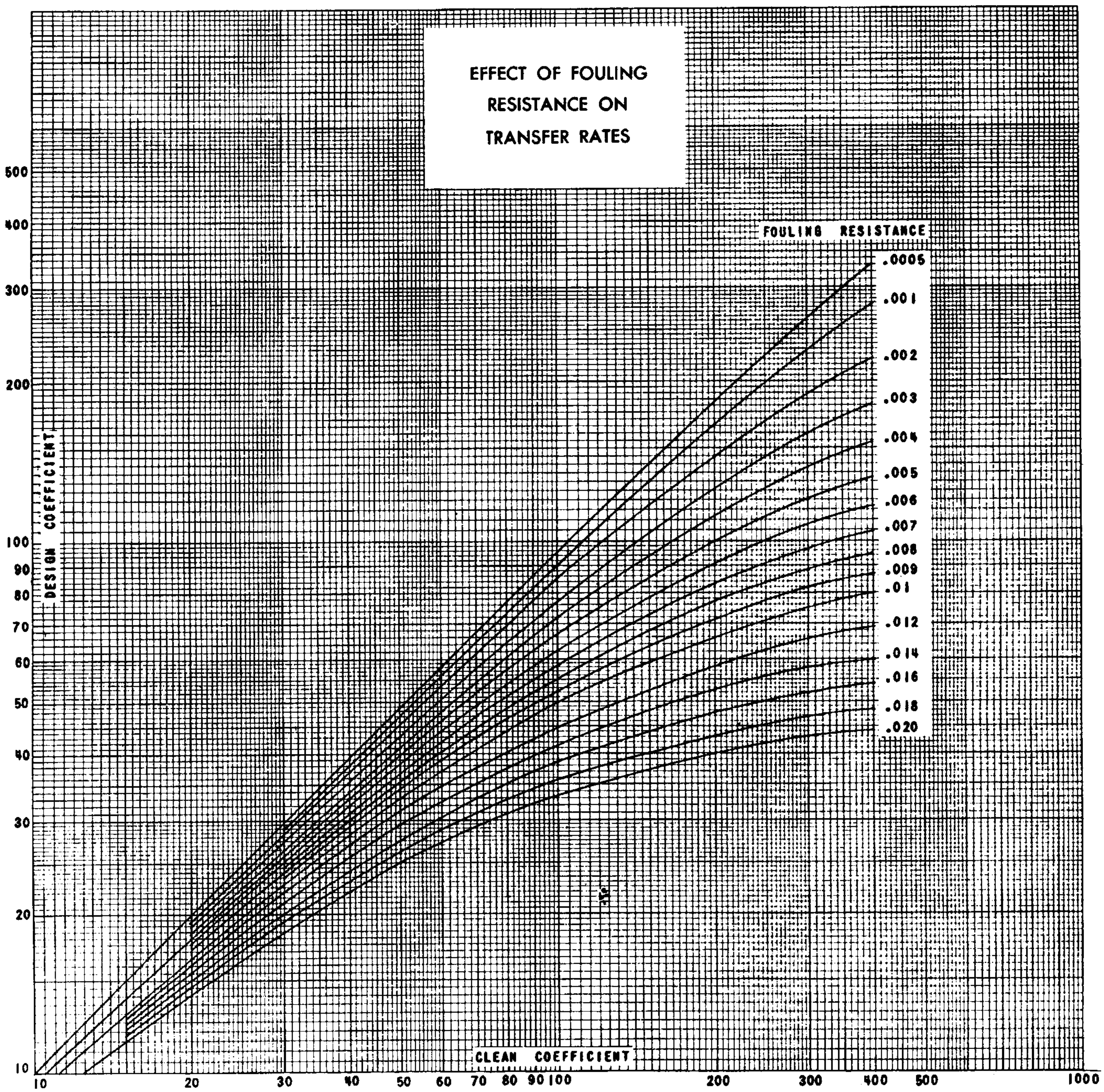


Figure 10-39. Chart for determining U-dirty from values of U-clean and the sum of tube-side and shell-side fouling resistances. Note: Factors refer to outside surface. Fouling resistance is sum of ($r_i + r_o$), as hr-ft²-°F/Btu. (Used by permission: *Standards of Tubular Exchanger Manufacturers Association* ©1959 and 1968. Tubular Exchanger Manufacturers Association, Inc. All rights reserved.)

operations, although portions of the table are applicable to general use and to petrochemical processes. In general, some experience in petrochemical mixtures indicates the values for certain waters, organic materials, and a few others to be too low but not as high as suggested by Fair.[44] High values for some materials do not seem justified in the light of normal process economics. Table 10-13 presents a selected group of materials with suggested fouling factors. Other references dealing with various types and conditions of heat exchanger fouling include 145, 146, 147, 148, 149, 150, 151, 152, 153, 154, 155, 156, 157, and 158.

Nelson[89] presents data in Figure 10-40A on some fluids showing the effects of velocity and temperature. Also see Figure 10-40B.

The fouling factors are applied as part of the overall heat transfer coefficient to both the inside and outside of the

Table 10-12
Guide to Fouling Resistances

RGP-T2.4 Design Fouling Resistances (hr-ft^2-°F/Btu)

The purchaser should attempt to select an optimal fouling resistance that will result in a minimum sum of fixed, shutdown and cleaning costs. The following tabulated values of fouling resistances allow for oversizing the heat exchanger so that it will meet performance requirements with reasonable intervals between shutdowns and cleaning. These values do not recognize the time related behavior of fouling with regard to specific design and operational characteristics of particular heat exchangers.

Fouling Resistances For Industrial Fluids

Oils:	
Fuel oil #2	0.002
Fuel oil #6	0.005
Transformer oil	0.001
Engine lube oil	0.001
Quench oil	0.004
Gases and Vapors:	
Manufactured gas	0.010
Engine exhaust gas	0.010
Steam (nonoil-bearing)	0.0005
Exhaust steam (oil-bearing)	0.0015–0.002
Refrigerant vapors (oil-bearing)	0.002
Compressed air	0.001
Ammonia vapor	0.001
CO_2 vapor	0.001
Chlorine vapor	0.002
Coal flue gas	0.010
Natural gas flue gas	0.005
Liquids:	
Molten heat transfer salts	0.0005
Refrigerant liquids	0.001
Hydraulic fluid	0.001
Industrial organic heat transfer media	0.002
Ammonia liquid	0.001
Ammonia liquid (oil-bearing)	0.003
Calcium chloride solutions	0.003
Sodium chloride solutions	0.003
CO_2 liquid	0.001
Chlorine liquid	0.002
Methanol solutions	0.002
Ethanol solutions	0.002
Ethylene glycol solutions	0.002

Fouling Resistances for Chemical Processing Streams

Gases and Vapors:	
Acid gases	0.002–0.003
Solvent vapors	0.001
Stable overhead products	0.001
Liquids:	
MEA and DEA solutions	0.002
DEG and TEG solutions	0.002
Stable side draw and bottom product	0.001–0.002
Caustic solutions	0.002
Vegetable oils	0.003

Fouling Resistances for Natural Gas-Gasoline Processing Streams

Gases and Vapors:	
Natural gas	0.001–0.002
Overhead products	0.001–0.002
Liquids:	
Lean oil	0.002
Rich oil	0.001–0.002
Natural gasoline and liquefied petroleum gases	0.001–0.002

Fouling Resistances for Oil Refinery Streams

Crude and Vacuum Unit Gases and Vapors:	
Atmospheric tower overhead vapors	0.001
Light Naphtha	0.001
Vacuum overhead vapors	0.002

Crude and Vacuum Liquids:

Crude oil

	0 to 250°F velocity ft/sec			250 to 350°F velocity ft/sec		
	<2	2–4	>4	<2	2–4	>4
DRY	0.003	0.002	0.002	0.003	0.002	0.002
SALT*	0.003	0.002	0.002	0.005	0.004	0.004
	350 to 450°F velocity ft/sec			**450°F and more velocity ft/sec**		
	<2	2–4	>4	<2	2–4	>4
DRY	0.004	0.003	0.003	0.005	0.004	0.004
SALT*	0.006	0.005	0.005	0.007	0.006	0.006

*Assumes desalting @ approx. 250°F

Gasoline	0.002
Naphtha and light distillates	0.002–0.003
Kerosene	0.002–0.003
Light gas oil	0.002–0.003
Heavy gas oil	0.003–0.005
Heavy fuel oils	0.005–0.007
Asphalt and Residuum:	
Vacuum tower bottoms	0.010
Atmosphere tower bottoms	0.007
Cracking and Coking Unit Streams:	
Overhead vapors	0.002
Light cycle oil	0.002–0.003
Heavy cycle oil	0.003–0.004

Fouling Resistances for Oil Refinery Streams (Continued)

Cracking and Coking Unit Streams (Continued)	
Light coker gas oil	0.003–0.004
Heavy coker gas oil	0.004–0.005
Bottoms slurry oil (4.5 ft/sec min.)	0.003
Light liquid products	0.002
Catalytic Reforming, Hydrocracking, and Hydrodesulfurization Streams:	
Reformer charge	0.0015
Reformer effluent	0.0015
Hydrocracker charge and effluent*	0.002
Recycle gas	0.001
Hydrodesulfurization charge and effluent*	0.002
Overhead vapors	0.001
Liquid product greater than 50°A.P.I.	0.001
Liquid product 30–50°A.P.I.	0.002

*Depending on charge, characteristics and storage history, charge resistance may be many times this value.

Light Ends Processing Streams:	
Overhead vapors and gases	0.001
Liquid products	0.001
Absorption oils	0.002–0.003
Alkylation trace acid streams	0.002
Reboiler streams	0.002–0.003
Lube Oil Processing Streams:	
Feed stock	0.002
Solvent feed mix	0.002
Solvent	0.001
Extract*	0.003
Raffinate	0.001
Asphalt	0.005
Wax slurries*	0.003
Refined lube oil	0.001

*Precautions must be taken to prevent wax deposition on cold tube walls.

Visbreaker:	
Overhead vapor	0.003
Visbreaker bottoms	0.010
Naphtha Hydrotreater:	
Feed	0.003
Effluent	0.002
Naphtha	0.002
Overhead vapors	0.0015
Catalytic Hydro Desulfurizer:	
Charge	0.004–0.005
Effluent	0.002
H.T. sep. overhead	0.002
Stripper charge	0.003
Liquid products	0.002
HF Alky Unit:	
Alkylate, deprop. bottoms, main fract. overhead, main fract. feed	0.003
All other process streams	0.002

Fouling Resistances for Water

Temperature of Heating Medium	**Up to 240°F**		**240 to 400°F**	
Temperature of Water	**125°**		**More Than 125°**	
	Water Velocity Ft/Sec		**Water Velocity Ft/Sec**	
	3 and Less	**More Than 3**	**3 and Less**	**More Than 3**
Sea water	0.0005	0.0005	0.001	0.001
Brackish water	0.002	0.001	0.003	0.002
Cooling tower and artificial spray pond:				
Treated makeup	0.001	0.001	0.002	0.002
Untreated	0.003	0.003	0.005	0.004
City or well water	0.001	0.001	0.002	0.002
River water:				
Minimum	0.002	0.001	0.003	0.002
Average	0.003	0.002	0.004	0.003
Muddy or silty	0.003	0.002	0.004	0.003
Hard (more than 15 grains/gal)	0.003	0.003	0.005	0.005
Engine jacket	0.001	0.001	0.001	0.001
Distilled or closed cycle				
Condensate	0.0005	0.0005	0.0005	0.0005
Treated boiler feedwater	0.001	0.0005	0.001	0.001
Boiler blowdown	0.002	0.002	0.002	0.002

If the heating medium temperature is more than 400°F and the cooling medium is known to scale, these ratings should be modified accordingly.

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heat transfer surface using the factor that applies to the appropriate material or fluid. As a rule, the fouling factors are applied without correcting for the inside diameter to outside diameter, because these differences are not known to any great degree of accuracy.

For fouling resistances of significant magnitude, a correction is usually made to convert all values to the outside surface of the tube; see Equation 10-37. Sometimes only one factor is selected to represent both sides of the transfer fouling films or scales.

Table 10-13
Suggested Fouling Factors in Petrochemical Processes
r = (hr) (ft^2) (°F)/Btu

Fluid	Velocity, Ft/Sec	Temperature Range <100°F	>100°F
Waters:			
Sea (limited to 125°F max.)	<4	0.002	0.003
	>7	0.0015	0.002
River (settled)	<2	0.002	0.002–0.003
	>4	0.0005–0.0015	0.001–0.0025
River (treated and settled)	<2	0.0015	0.002
	>4	0.001	0.0015
4 mils baked phenolic coating[65]		0.0005	
15 mils vinyl-aluminum coating		0.001	
Condensate (100°–300°F)	<2	0.001	0.002–0.0004
	>4	0.0005	0.001
Steam (saturated) oil free			0.0005–0.0015
with traces oil			0.001–0.002
Light hydrocarbon liquids (methane, ethane, propane, ethylene, propylene, butane-clean) } Light hydrocarbon vapors: (clean) }		0.001	
Chlorinated hydrocarbons (carbon tetrachloride, chloroform, ethylene dichloride, etc.)			
Liquid		0.001	0.002
Condensing		0.001	0.0015
Boiling		0.002	0.002
Refrigerants (vapor condensing and liquid cooling)			
Ammonia		0.001	
Propylene		0.001	
Chloro-fluoro-refrigerants		0.001	
Caustic liquid, salt-free			
20% (steel tube)	3–8	0.0005	
50% (nickel tube)	6–9	0.001	
73% (nickel tube)	6–9	0.001	
Gases (industrially clean)			
Air (atmos.)		0.0005–0.001	
Air (compressed)		0.001	
Flue gases		0.001–0.003	
Nitrogen		0.0005	
Hydrogen		0.0005	
Hydrogen (saturated with water)		0.002	
Polymerizable vapors with inhibitor		0.003–0.03	
High temperature cracking or coking, polymer buildup		0.02–0.06	
Salt brines			
(125°F max.)	<2	0.003	0.004
	>4	0.002	0.003
Carbon dioxide[93] (sublimed at low temp.)		0.2–0.3	

In the tables the representative or typical fouling resistances are referenced to the surface of the exchanger on which the fouling occurs—that is, the inside or outside tubes. Unless specific plant/equipment data represents the fouling in question, the estimates in the listed tables are a

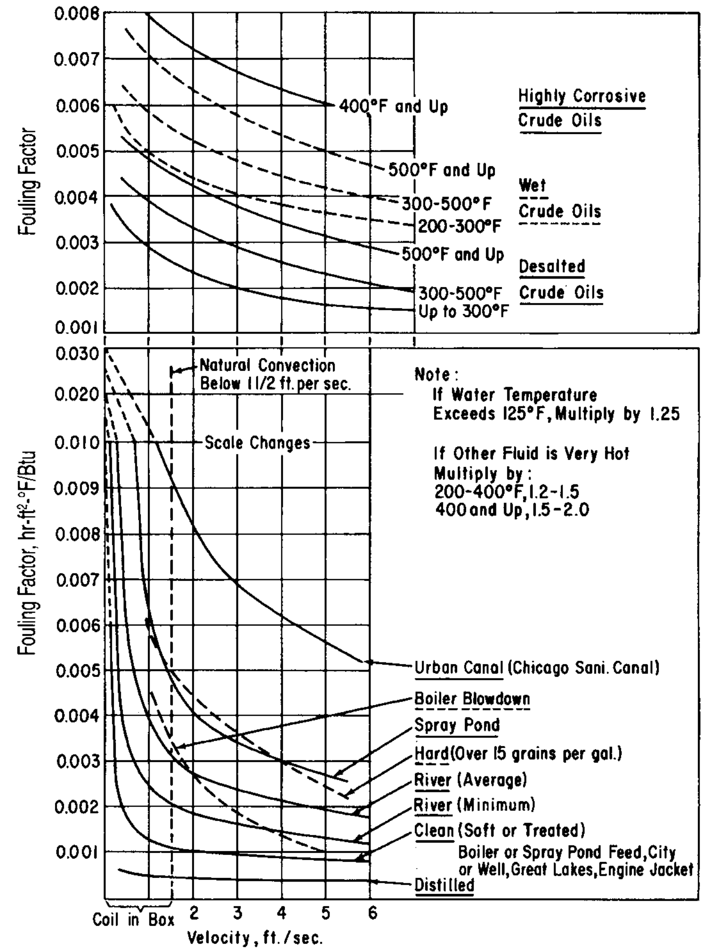

Figure 10-40A. Fouling factors as a function of temperature and velocity. (Used by permission: W. L. Nelson, No. 94 in series, *Oil and Gas Journal.* ©PennWell Publishing Company.)

reasonable starting point. It is not wise to keep adjusting the estimated (or other) fouling to achieve a specific overall heat transfer coefficient, U, which is the next topic to be discussed.

Fouling generally can be kept to a minimum provided that proper and consistent cleaning of the surface takes place. Kern[269] discusses fouling limits. Inside tubes may be rodded, brushed, or chemically cleaned, and most outside tube surfaces in a shell can be cleaned only by chemical or by hydraulic/corncob external cleaning or rodding/brushing between tube lanes provided that the shell is removable as in Figures 10-1A, 10-1B, 10-1D–F, 10-1I, and 10-1K.

Unless a manufacturer/fabricator is guaranteeing the performance of an exchanger in a specific process service, they cannot and most likely will not accept responsibility for the fouling effects on the heat transfer surface. Therefore, the owner must expect to specify a value to use in the thermal design of the equipment. This value must be determined

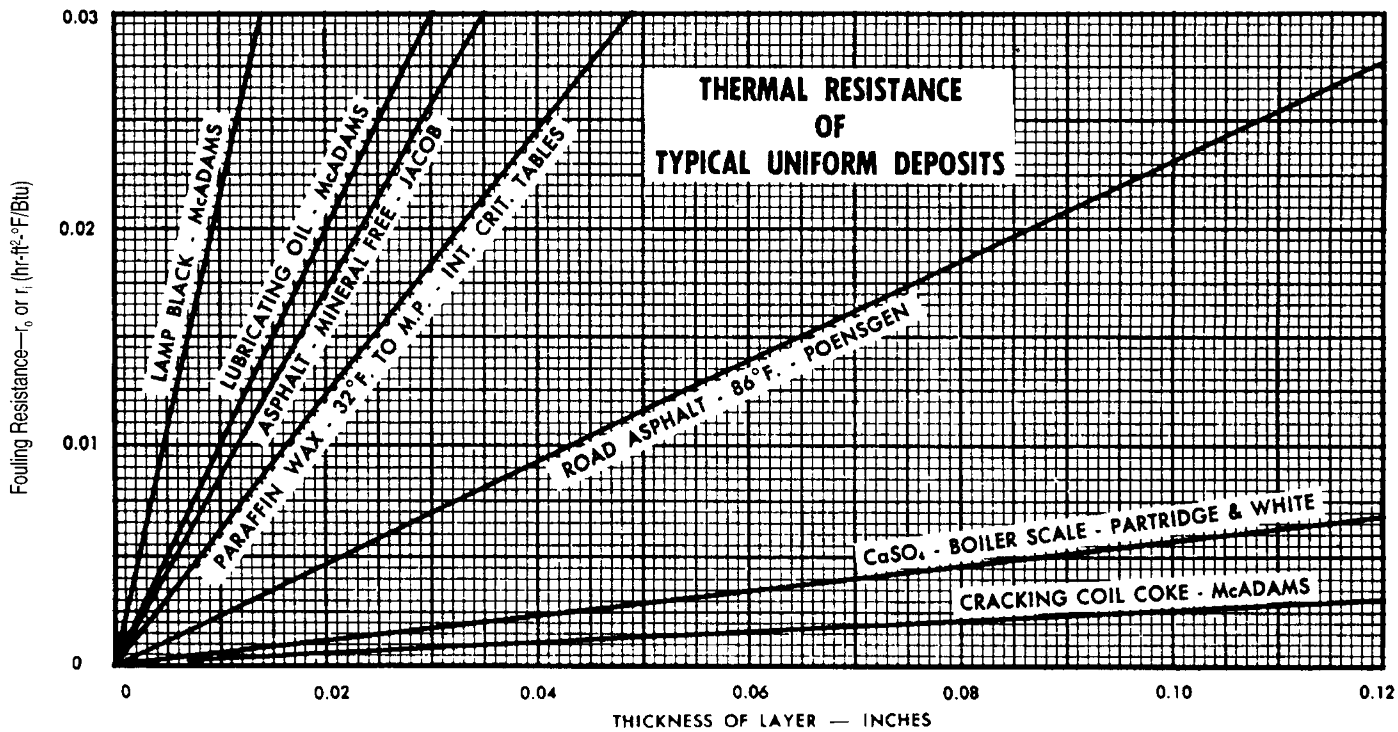

Figure 10-40B. Fouling resistance for various conditions of surface fouling on heat exchanger surfaces. Thermal resistance of typical uniform deposits. Note that the abscissa reads for either the inside, r_i, or outside, r_o, fouling resistance of the bulidup of the resistance layer or film on/in the tube surface. (Used by permission: *Standards of Tubular Exchanger Manufacturers Association,* 6th Ed, p. 138, © 1978. Tubular Exchanger Manufacturers Association, Inc. All rights reserved.)

with considerable examination of the fouling range, both inside and outside of the tubes, and by determining the effects these have on the surface area requirements. Just a large unit may not be the proper answer. Fouling of the tube surfaces is usually expressed[107] as follows:

$$r_o = \text{fouling resistance on outside of tube, } \frac{(\text{hr})(°\text{F})(\text{ft}^2 \text{ outside surface})}{(\text{Btu})}$$

$$r_I = \text{fouling resistance on inside of tube, } \frac{(\text{hr})(°\text{F})(\text{ft}^2 \text{ inside surface})}{(\text{Btu})}$$

Fouling of the tube surfaces (inside and/or outside) can be an important consideration in the economical and thermal design of a heat exchanger. Most fouling can be categorized by the following characteristics.[107] Note that biological fouling is not included.

- Linear
- Falling-rate
- Asymptotic

Essentially all three of these types are time-dependent regarding the buildup or increase in the thickness and/or density of the fouling material. These authors[137] suggest the need for specific time-dependent data to better define fouling, and they propose calculation techniques but no actual physical data. If the time required to reach a certain level of fouling is measured or observed operationally, then cleaning maintenance schedules can be better coordinated by considering production downtime, rather than having the need to improve the heat transfer become a surprise or "crater" situation.

Epstein[145, 146] lists six types of fouling:

- Precipitation or scaling fouling: precipitation on hot surfaces or due to inverse solubility.
- Particulate fouling: suspended particles settle on heat transfer surface.
- Chemical reaction fouling: deposits formed by chemical reaction in the fluid system.
- Corrosion fouling: corrosion products produced by a reaction between fluid and heat transfer surface, and tube surface becomes fouled.
- Solidification fouling: liquid and/or its components in liquid solution solidify on tube surface.
- Biological fouling: biological organisms attach to heat transfer surface and build a surface to prevent good fluid contact with the tube surface.

Many cooling waters have inverse solubility characteristics due to dissolved salt compounds (organic and inorganic); others carry suspended solids that deposit on the tube at low-flowing velocities (<2 ft/sec). Biological fouling usually does not occur and is not a serious problem in most plant waters that are treated with biocides. Inverse solubility of dissolved salts in water occurs when the water contacts warm surfaces and the salts deposit on the tube surfaces[140]. Do not design an exchanger by selecting a fouling resistance that has not fully developed, but rather select a value that has stabilized over a period of time, see Figure 10-41. It is also quite important to appreciate that fluid velocity often affects the fouling material thickness and hence its ultimate value for exchanger design; note Tables 10-12 and 10-13 and Figure 10-42 as examples for some waters.[140] Although TEMA[107] presents suggested fouling resistances, r, these are average values to consider and do not identify the actual effects of hot surfaces, fluid velocity, or composition of the deposited film, solid suspension, or other scale. Thus, the designer must establish from usually meager data (if any) the fouling resistances to use in an actual design, and often only through experience. Some field plant operating performance can aid in establishing the ultimate magnitude of the fouling. Knudsen[140] reports useful data in Chenoweth[142] and Koenigs.[141]

Zanker[143] has presented a graphical technique for determining the fouling resistance (factor) for process or water fluid systems based on selected or plant data measurements, as shown in Figures 10-43A, 10-43B, and 10-43C. The design determination procedure presented by Zanker[143] is quoted here and used by permission from Hydrocarbon Processing ©1978 by Gulf Publishing Company, all rights reserved. It is based on: $R_t = R^* \left(1 - e^{-BT}\right)$. The asymptotic value, R^*, is the expected fouling resistance after operating at time θ_2 and is the value proposed for the use in design.

"In order to use these nomographs two sets of data have to be known: t_1, R_1, and t_2, R_2.

Prior to using these nomographs, the auxiliary values have to be computed:

$$\theta = t_1/t_2 \text{ and } \alpha - R_1/R_2$$

The following steps are used with the nomograph Part 1, Figure 10-43A.

Find the intersecting point of the curves of known values of θ and α on the grid in the center of nomograph.

Interpolate if necessary; mark this point A.

2) Connect Point A, with a ruler, to the known value of θ on the "primary scale." Extend this line up to the intersecting point with the Y scale. (Read the Y value as an intermediate result).
3) Connect the Y value, with a ruler, to the known value of R_1 on the appropriate scale.

Read the final result, R^*, at the intersection point of the line with the oblique R^* scale.

The following steps are used with nomograph Part 2 (Figure 10-43B).

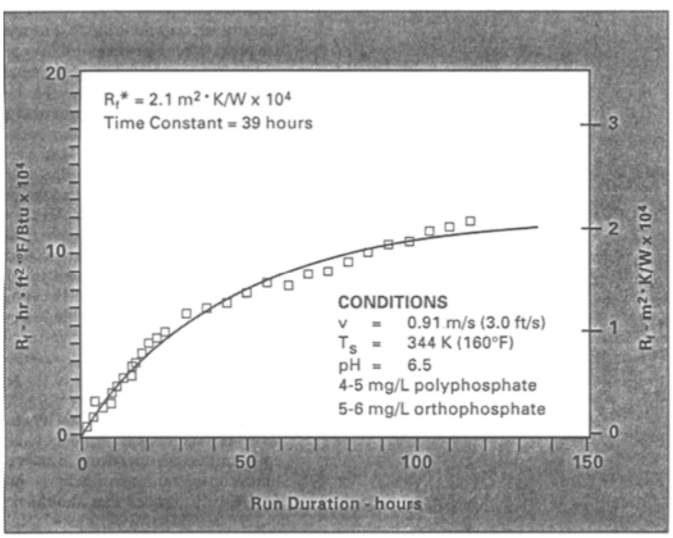

Figure 10-41. For many cooling waters, the fouling resistance increases rapidly, then decreases, and finally approaches an asymptotic value. (Used by permission: Knudsen, J. G., *Chemical Engineering Progress*. V. 87, No. 4, ©1991. American Institute of Chemical Engineers. All rights reserved.)

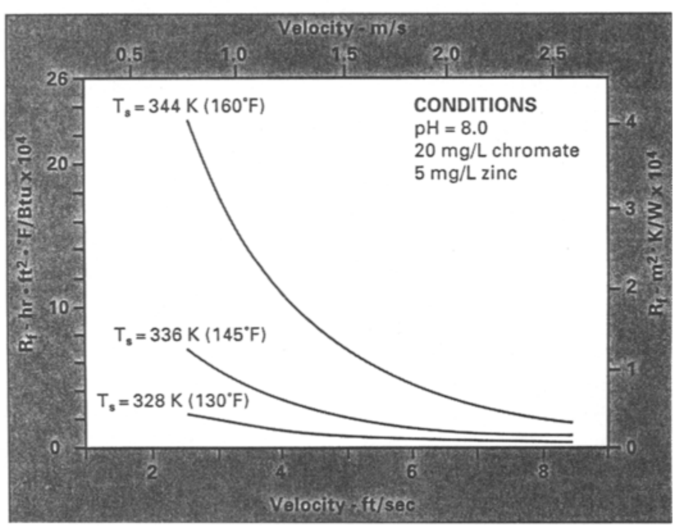

Figure 10-42. It is important to understand the relationships among velocity, surface temperature, and fouling resistance for a given exchanger. (Used by permission: Knudsen, J. G., *Chemical Engineering Progress*. V. 87, No. 4, ©1991. American Institute of Chemical Engineers. All rights reserved.)

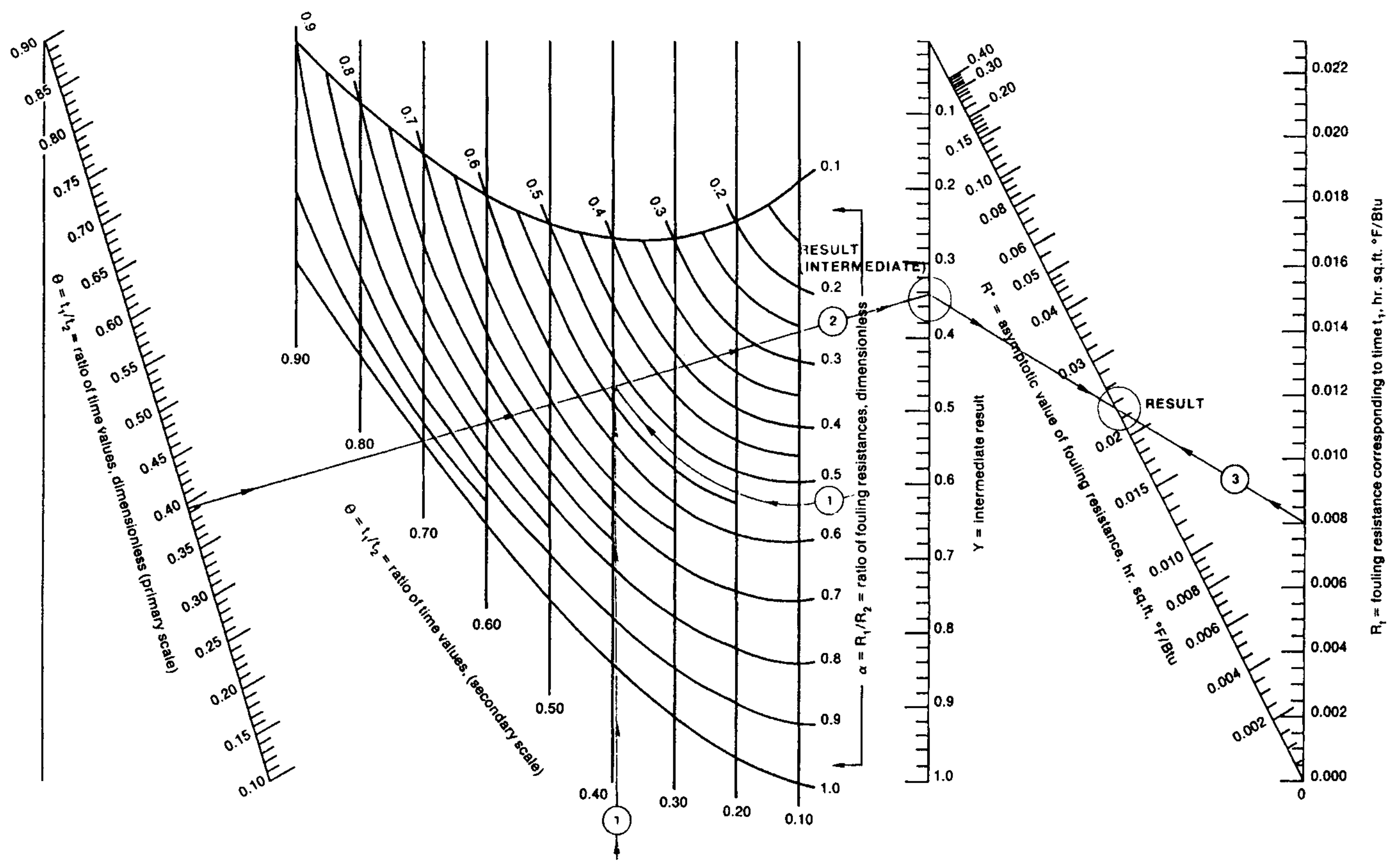

Figure 10-43A. Predict fouling by nomograph, Part 1. Calculation of R* value for fouling factor; use in conjunction with Figures 10-43B and 10-43C. (Used by permission: Zanker, A., *Hydrocarbon Processing.* March 1978, p. 146. ©Gulf Publishing Company, Houston, Texas. All rights reserved.)

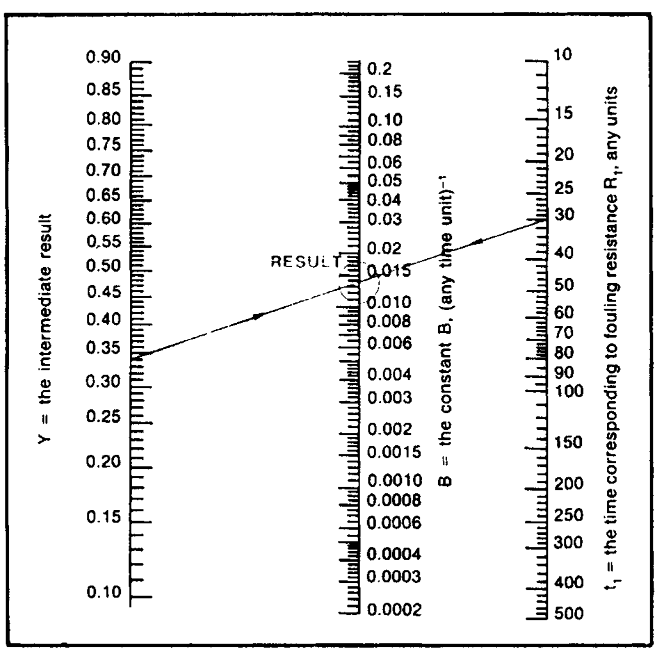

Figure 10-43B. Fouling Nomograph, Part 2. Calculation of B Value for fouling factor; use with Figures 10-43A and 10-43C. (Used by permission: Zanker, A., *Hydrocarbon Processing.* March 1978, p. 147. ©Gulf Publishing Company, Houston, Texas., All rights reserved.)

1) Connect, with a ruler, the known values of Y (found by means of Nomograph, Part 1) and t_1.
2) Read the final result, B, at the intersection point with the central B scale.

After using both nomographs, the constants R* and B are known, and equation can be solved with R_1 as the unknown.

It has to be emphasized that the units of t and B are opposite (reciprocals). If t is in days, then B is in days^{-1}.

The Nomograph Part 3 (Figure 10-43C) may be used in a number of ways. For example, what will the fouling resistance, R_t, be after an arbitrarily chosen time, t, or it can calculate the thickness of a fouling deposit after an arbitrarily chosen time t, providing the thermal conductivity of the deposited material is known. It can calculate thermal conductivity of a deposit, providing thickness is known, or estimated.

And, finally, it can answer: after how much time will the fouling resistance achieve a desired percentage of the asymptotic fouling value, R*?

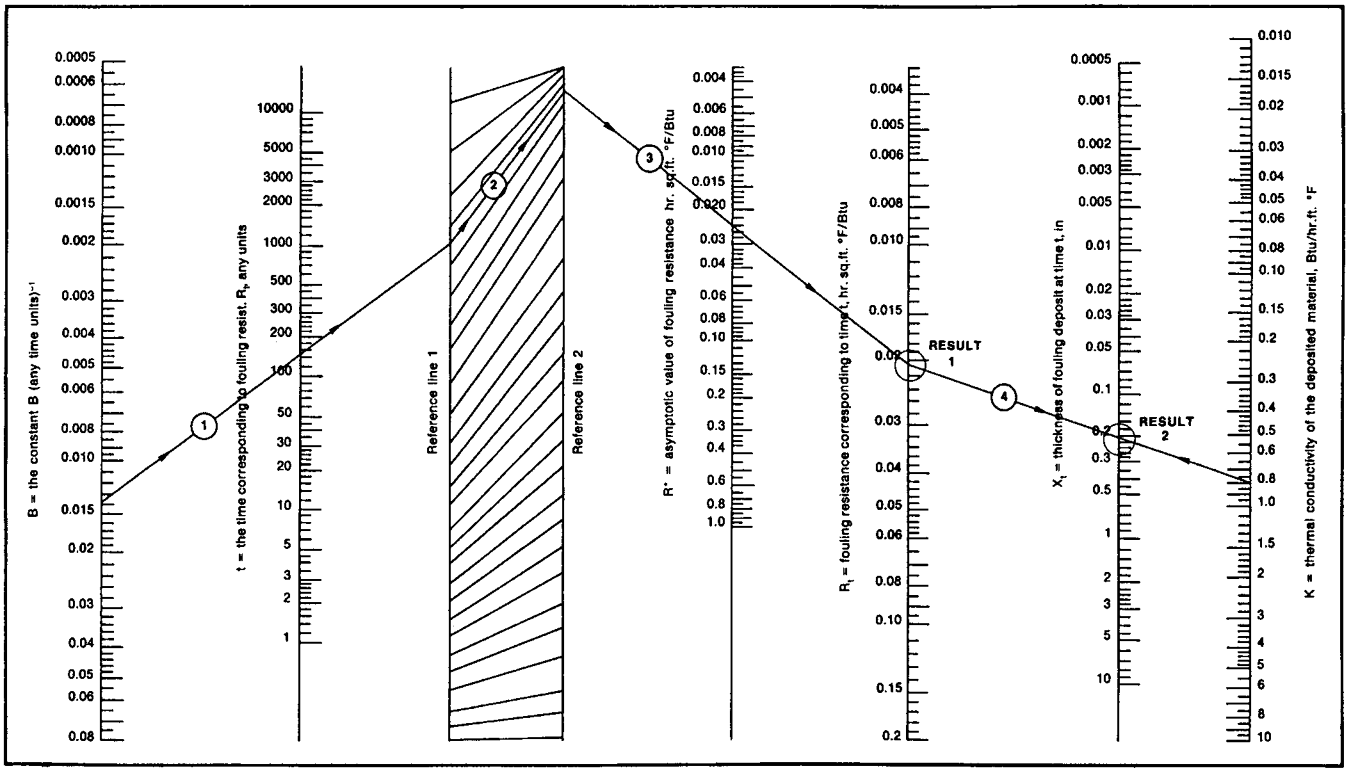

Figure 10-43C. Fouling Nomograph, Part 3. Final and practical calculation for fouling factor; use in conjunction with Figures 10-43A and 10-43B. (Used by permission: Zanker, A., *Hydrocarbon Processing*. March 1978, p. 148. ©Gulf Publishing Company, Houston, Texas. All rights reserved.)

The simplest calculation is to find the unknown fouling resistance after a time t. For this purpose, the nomograph is used as follows:

1) Connect, with a ruler, the known values of B and t on the appropriate scales. Extend this line up to the intersection point with Reference Line 1. Mark this Point A.
2) Transfer Point A from Reference Line 1 to Reference Line 2, using the oblique tie-lines as a guide. Mark A^1 the transferred point.
3) Connect Point A^1, with a ruler, to the known value of R* on the appropriate scale. Extend this line up to the intersection with the R_t scale.
   Read the *final result,* R_t, at this intersecting point.
4) If the thickness of deposit X_t is the desired value, and the conductivity, K is known, connect with a ruler the value of R_t to the known value of K on the appropriate scale and read *the thickness* X_t at the intersection point of this line with the X_t scale. Conversely, if the thickness X_t is known and the value of conductivity is desired, it can be easily found.

The Nomograph Part 3 may also be used to predict when the fouling resistance will reach an appropriate percentage of the asymptotic R* value, or any desired R value.

The procedure for this purpose is as follows:

1) Find the desired or calculated R value on the R_t scale and connect it with the known value of R* on the R* scale; extend this line up to the intersection with Reference Line 2. Mark this Point A^1.
2) Transfer Point A^1 from Reference Line 2 to Reference Line 1 using the oblique tie-lines as a guide. Mark the transferred Point A.
3) Connect Point A to the known value of B on the appropriate scale.

Read the final result, t, at the intersection point of this line with the t scale.

where

R_t = Fouling resistance at time T, (hr) (ft^2) (°F)/Btu
R* = Asymptotic value of fouling resistance, (hr) (ft^2) (°F)/Btu
t = Time, corresponding to the fouling resistance, R_t, (any units)
B = Constant, describing the rate of fouling, (any time units)$^{-1}$ (the constant B, is measured in the *reciprocals* of the same time units, as t)
e = Base of natural logarithms, (2.71821). As a simplification it is assumed that:
$R_t = X_t/K$

X_t = Thickness or deposit formed at time t (ft)
K = Thermal conductivity of deposited material, (Btu/(hr) (ft) (°F)."

Ganapathy[144] presents a working chart, Figure 10-44, to plot actual operating U_a values to allow projection back to infinity and to establish the "base" fouling factor after the operating elapsed time. Note that the flow rate inside or outside the tubes is plotted against the overall coefficient, U. For a heat exchanger in which gas flows inside the tubes.[144]

$$1/U = A W^{-0.8} + B \tag{10-35}$$

and for gas flowing over or outside plain or finned tubes:

$$1/U = A W^{-0.6} + B \tag{10-36}$$

where

U = overall heat transfer coefficient, Btu/(hr) (ft^2) (°F)
W = flow rate of the fluid controlling the heat transfer, lb/hr
B = constant, equal to fouling factor at infinity flow rate
F or ff = fouling factor

As the value of B or the fouling factor increases with time, the engineer can determine when the condition will approach that time when cleaning of the exchanger is required. Gas flows are used because usually the gas film controls in a gas-liquid exchanger.

Overall Heat Transfer Coefficients for Plain or Bare Tubes

The overall heat transfer coefficient as used in the relationship Q = UA Δt is the sum of the individual coefficient of heat transfer for the (a) fluid film inside the tube, (b) scale or fouling film inside the tube, (c) tube wall, (d) scale or fouling film outside the tube, and (e) fluid film outside the tube. These must each be individually established when making a new design, or they may be grouped together as U when obtaining data on an existing unit.

Najjar, Bell, and Maddox[162] studied the influence of physical property data on calculated heat transfer film coefficients and concluded that accurate fluid property data is extremely important when calculating heat transfer coefficients using the relationships offered by Dittus-Boelter, Sieder-Tate, and Petukhov. Therefore, the designer must strive to arrive at good consistent physical/thermal property data for these calculations.

Figure 10-28 illustrates the factors affecting the overall heat transfer expressed in equation form:

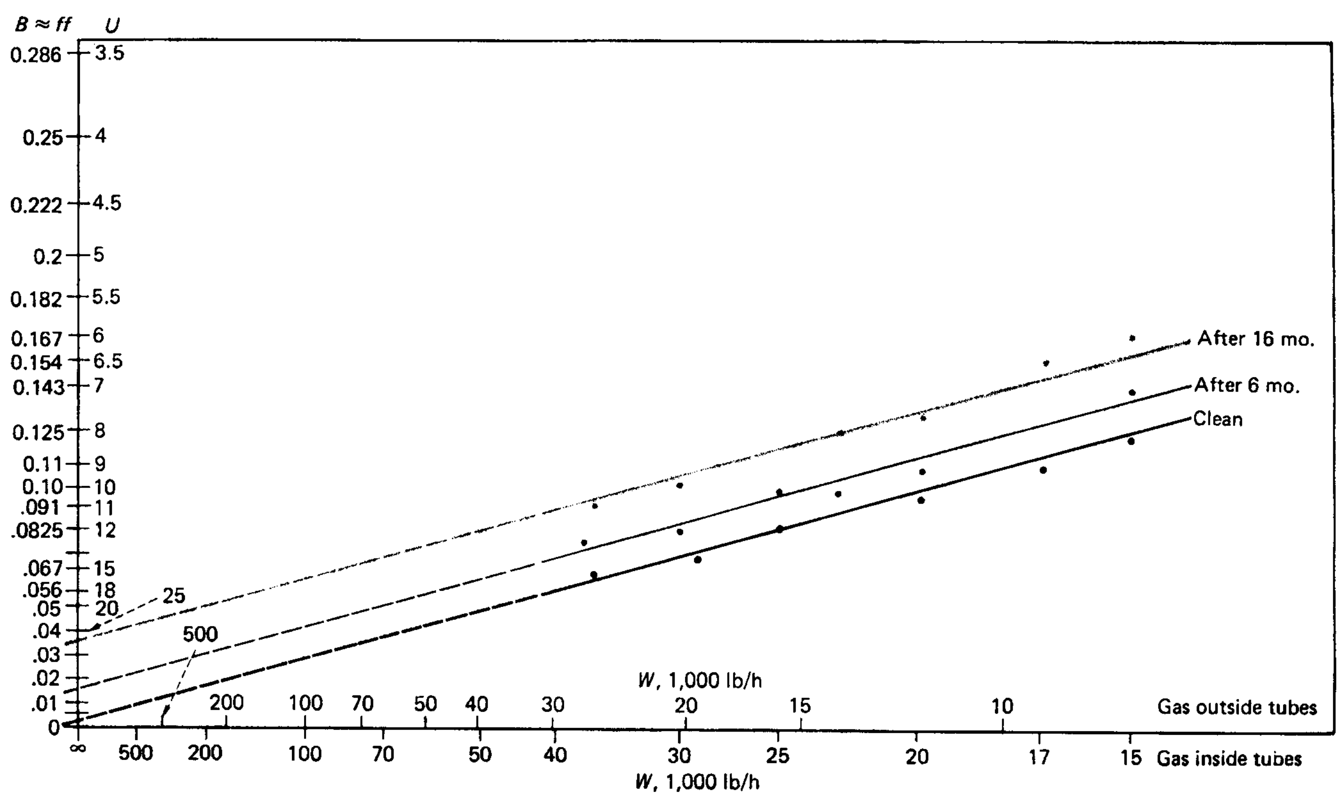

Figure 10-44. Keep track of fouling by monitoring the overall heat transfer coefficient as a function of flow rate. (Used by permission: Ganapathy, V., *Chemical Engineering,* Aug. 6, 1984, p. 94. ©McGraw-Hill, Inc. All rights reserved.)

$$U_o = \frac{1}{\frac{1}{h_o} + r_o + r_w\left(\frac{A_o}{A_{avg}}\right) + r_i\left(\frac{A_o}{A_i}\right) + \frac{1}{h_i\left(\frac{A_i}{A_o}\right)}} \quad (10\text{-}37)$$

where

U_o = overall heat transfer coefficient corrected for fouling conditions, Btu/hr (ft²) (°F), and referenced to outside tube surface area
h_o = film coefficient of fluid on outside of tube, Btu/hr (ft²) (°F)
h_i = film coefficient of fluid on inside of tube, Btu/hr (ft²) (°F)
r_o = fouling resistance (factor) associated with fluid on outside of tube, hr (ft²) (°F)/Btu
r_i = fouling resistance (factor) associated with fluid on inside of tube, hr (ft²) (°F)/Btu
*r_w = resistance of tube wall = L_w/k_w, hr (ft²) (°F.)/Btu
t = L_w = thickness of tube wall, in. or ft, as consistent
**k_w = thermal conductivity of material of tube wall, (Btu-ft)/[(hr) (ft²) (°F)]
A_o = outside area of unit length of tube, ft²/ft, Table 10-3
A_i = inside area of unit length of tube, ft²/ft, Table 10-3
A_{avg} = average of inside and outside tube area for unit length, ft²/ft
Note: Btu/(hr) (ft²) (°F/ft) = 12 Btu/(hr) (ft²) (°F/in.)
Δt = corrected mean temperature difference, °F
A_o = A = total required *net* effective outside heat transfer surface referenced to tube length measured between inside dimensions between tubesheets
r_w = resistance of tube wall referred to outside surface of tube wall, including extended surface, if present[107]
* r_w = also for bare tubes: $\frac{d}{24k}\left[\ln\frac{(d)}{(d-2t)}\right]$ reference,[107] (hr) (°F) (ft² outside surface)/Btu
d = O.D. bare tube (or, root diameter of fin tube), in.
t = tube wall thickness, in.
N = number of fins/in.
** K = k = thermal conductivity, Btu ft/(hr) (ft²) (°F) (Note the difference in units.) For conversion, see Table 10-13A.
w = fin height, in.
ln = natural logarithm
** = must be consistent units, also *

For integral circumferentially finned tubes:[107]

$$r_w = \frac{t}{12k}\,\frac{[d + 2Nw(d + w)]}{(d - t)} \quad (10\text{-}38)$$

In actual exchanger operational practice, the U values at the hot and cold terminals of the heat exchanger are not the same and can be significantly different if evaluated only at the spot conditions. In order to obtain an overall coefficient U that represents the transfer of heat throughout the exchanger, the U should be evaluated at the caloric temperature for physical properties and individual film coefficients of the fluids. Often, bulk average temperatures are used, but these may not be sufficiently accurate. Film coefficients *should be more accurately evaluated at or close to* the tube wall temperatures.

The ratio multiplier, A_o/A_i, is usually omitted for general process design from the r_i factor for inside fouling. For thin-walled (18–12 ga) and highly conductive metal tubes, such as high copper alloys, the resistance of the tube wall can usually be omitted with little, if any, significant effect on U. Each of these omissions should be looked at in the light of the problem and not as a blind rule. It is important to appreciate that the tube wall resistance of such useful tube wall materials as Teflon® and other plastics, Karbate® and other impervious graphites, glass, plastic-lined steel, and even some exotic metals, etc., cannot be omitted as they are usually sufficiently thick as to have a significant impact. Refer to Table 10-13A for thermal conductivity, k, values for many common metal tubes and allow a calculation of r_w, tube wall resistance. Note that the conversion for thermal conductivity is

$$\frac{\text{Btu}}{(\text{hr})(\text{ft}^2)(°\text{F/in.})} \times 0.08333 = \frac{\text{Btu}}{(\text{hr})(\text{ft}^2)(°\text{F/ft})} \quad (10\text{-}39)$$

and Btu/ (hr) (ft) (°F) = Btu-ft/ (hr) (ft²) (°F)

Table 10-14 tabulates a few unusual and useful thermal conductivity data.

Note that individual heat transfer coefficients are not additive, but their reciprocals, or resistances, are

$$\frac{1}{U} = \frac{1}{h_o} + r_o + r_w\left(\frac{A_o}{A_{avg}}\right) + r_i + \frac{1}{h_i\left(\frac{a_i}{A_o}\right)} \quad (10\text{-}40)$$

$1/h_o$ = resistance of outside fluid film
$1/h_i$ = resistance of inside fluid film

Sometimes one of the fluid-side scale resistances can be neglected or assumed to be so small as to be of little value, in which case only the significant resistances and/or film coefficients need to be used in arriving at the overall coefficient, U. Note that A_o, A_i, and A_{avg} can be substituted by d_o, d_{it}, and d_{avg} respectively. Theoretically, d_{avg} and A_{avg} should be the logarithmic average, but for most practical cases, the use of the arithmetic average is completely satisfactory.

Recognize that only the heat that flows through the sum of all the resistances can flow through any one resistance considered individually, even though by itself, a resistance may be capable of conducting or transferring more heat.

Film coefficients defined on an *inside* tube surface area basis *when converted* to the larger outside surface area become

$$h_{io} = h_i\left(\frac{A_i}{A_o}\right) = h_i\left(\frac{d_{it}}{d_o}\right) \quad (10\text{-}41)$$

Table 10-13A
Metal Resistance of Tubes and Pipes

OD Tube	Gauge	Factor	Carbon Steel	Admiralty	Red Brass, .45% Ars. Copper	4–6% Chrome 1/2 % Moly Steel 80–20 CU-NI	70–30 CU-NI	Monel	Copper 99.9+ % CU	Nickel	Aluminum	Stainless AISI Type 302 & 304	Yorkalbro, Alum. Brass
Thermal Conductivity, K (Value of r_w)			25	63	89	21	17	14.6	238	34.4	85	10	57.7
5/8 in.	18	.00443	.000177	.000070	.000050	.000211	.000261	.000303	.000019	.000129	.000052	.000443	.000077
	16	.00605	.000242	.000096	.000068	.000288	.000356	.000414	.000025	.000176	.000071	.000605	.000105
	14	.00798	.000319	.000127	.000090	.000380	.000469	.000547	.000034	.000232	.000094	.000798	.000138
	18	.00437	.000175	.000069	.000049	.000208	.000257	.000299	.000018	.000127	.000051	.000437	.000076
	16	.00593	.000237	.000094	.000067	.000282	.000349	.000406	.000025	.000172	.000070	.000593	.000103
3/4 in.	14	.00778	.000311	.000123	.000087	.000370	.000458	.000533	.000033	.000226	.000092	.000778	.000135
	13	.00907	.000363	.000144	.000102	.000432	.000534	.000621	.000038	.000264	.000107	.000907	.000157
	12	.01063	.000425	.000169	.000119	.000506	.000625	.000728	.000045	.000309	.000125	.001063	.000184
	18	.00429	.000172	.000068	.000048	.000204	.000252	.000294	.000018	.000125	.000050	.000429	.000074
	16	.00579	.000232	.000092	.000065	.000276	.000341	.000397	.000024	.000168	.000068	.000579	.000100
1 in.	14	.00754	.000302	.000120	.000085	.000359	.000444	.000516	.000032	.000219	.000089	.000754	.000131
	13	.00875	.000350	.000139	.000098	.000417	.000515	.000599	.000037	.000254	.000103	.000875	.000152
	12	.01019	.000408	.000162	.000114	.000485	.000599	.000698	.000043	.000296	.000120	.001019	.000177
	10	.01289	.000516	.000205	.000145	.000614	.000758	.000883	.000054	.000375	.000152	.001289	.000223
	8	.01647	.000659	.000261	.000185	.000784	.000967	.001128	.000069	.000479	.000194	.001647	.000285
	18	.00425	.000170	.000067	.000048	.000202	.000250	.000291	.000018	.000124	.000050	.000425	.000074
	16	.00571	.000228	.000091	.000064	.000272	.000336	.000391	.000024	.000166	.000067	.000571	.000099
1 1/4 in.	14	.00741	.000296	.000118	.000083	.000353	.000436	.000508	.000031	.000215	.000087	.000741	.000128
	13	.00857	.000343	.000136	.000096	.000408	.000504	.000587	.000036	.000249	.000101	.000857	.000149
	12	.00995	.000398	.000158	.000112	.000474	.000585	.000682	.00042	.000289	.000117	.000995	.000172
	10	.01251	.000500	.000199	.000141	.000596	.000736	.000857	.000053	.000364	.000147	.001251	.000217
	8	.01584	.000634	.000251	.000178	.000754	.000932	.001085	.000067	.000460	.000186	.001584	.000275
	18	.00422	.000169	.000067	.000047	.000201	.000248	.000289	.000018	.000123	.000050	.000422	.000073
	16	.00566	.000226	.000090	.000064	.000270	.000333	.000388	.000024	.000165	.000067	.000566	.000098
1 1/2 in.	14	.00732	.000293	.000116	.000082	.000349	.000431	.000501	.000031	.000213	.000086	.000732	.000127
	13	.00845	.000338	.00134	.000095	.000402	.000497	.000579	.000036	.000246	.000099	.000845	.000146
	12	.00979	.000392	.000155	.000110	.000466	.000576	.000671	.000041	.000285	.000115	.000979	.000170
	10	.01226	.000490	.000195	.000138	.000584	.000721	.000840	.000052	.000356	.000144	.001226	.000212
	8	.01545	.000618	.000245	.000174	.000736	.000909	.001058	.000065	.000449	.000182	.001545	.000268
	18	.00419	.000168	.000067	.000047	.000200	.00246	.000287	.000018	.000122	.000049	.000419	.000073
	16	.00560	.000224	.000089	.000063	.000267	.000329	.000384	.000024	.000163	.000066	.000560	.000097
2 in.	14	.00722	.000289	.000115	.000081	.000344	.000425	.000495	.000030	.000210	.000085	.000722	.000125
	13	.00831	.000332	.000132	.000093	.000396	.000489	.000569	.000035	.000242	.000098	.000832	.000144
	12	.00961	.000384	.000153	.000108	.000458	.000565	.000658	.000040	.000279	.000113	.000961	.000167
	10	.01197	.000479	.000190	.000134	.000570	.000704	.000820	.000050	.000348	.000141	.001197	.000207
	8	.01499	.000600	.000238	.000168	.000714	.000882	.001027	.000063	.000436	.000176	.001499	.000260
Pipe													
3/4 in.	St'd	.01055	.000422	.000167	.000119	.000502	.000621	.000723	.000044	.000307	.000124	.001055	.000183
IPS	X Hvy	.01504	.000602	.000239	.000169	.000716	.000885	.001030	.000063	.000437	.000177	.001504	.000261
	Sch. 160	.0229	.000916	.000363	.000257	.001090	.001347	.00157	.000096	.000666	.000269	.00229	.000397
	XX Hvy	.0363	.00145	.000576	.000408	.00173	.00214	.00249	.000153	.001055	.000427	.00363	.000629
Pipe													
1 1/2 in.	St'd	.01308	.000523	.000208	.000147	.000623	.000769	.000896	.000055	.000380	.000154	.001308	.000227
IPS	X Hvy	.01863	.000745	.000296	.000209	.000887	.001096	.001276	.000078	.000542	.000219	.001863	.000323
	Sch. 160	.0275	.00110	.000437	.000309	.00131	.00162	.00188	.000116	.000799	.000324	.00275	.000477
	XX Hvy	.0422	.00169	.000670	.000474	.00201	.00248	.00289	.000177	.001227	.000496	.00422	.000731

r_w for 1 in. O.D. 16 BWG steel tube with 18 BWG admiralty liner = .00031

For other materials, divide *factor* number by the thermal conductivity of various materials:

60–40 Brass 55	Chrome Vanadium Steel	Tin 35	Everdur #50 19
Zinc 64	SAE 6120 .... 23.2	Lead 20	Wrought Iron 40
			Alcunic 56.5

K in Btu/(hr)(ft²)(°F/ft) r_w in $\dfrac{1}{\text{Btu/(hr)(ft}^2\text{)(°F)}}$

Used by permission: Griscom-Russell/Ecolaire Corporation.

Table 10-13B
Preliminary Design Resistances

Basis: Pressures Used in Commercial Fractionations		
Heating Side, r_p'	**Clean**	**Service**
Condensing steam	0.0005	0.0010
Cooling hot water	0.0025	0.0045
Cooling hot oil	0.0080	0.0100
Combustion gases	*	*
Boiling Side, r_b'	**Clean**	**Service**
C_2–C_4 hydrocarbons	0.0030	0.0040
Gasoline and naphthas	0.0050	0.0060
Aromatics	0.0030	0.0040
C_2–C_7 alcohols	0.0030	0.0040
Chlorinated hydrocarbons	0.0040	0.0070
Water (atm. pressure)	0.0015	0.0025

*For direct-fired reboilers, estimate area on basis of heat flux:
Radiant zone q/A = 10,000 Btu/(hr)(ft²)(°F)
Convection zone q/A = 3,500 Btu/(hr)(ft²)(°F)
Used by permission: Fair, J. R., *Petroleum Refiner.* Feb. 1960, reference 45. ©Gulf Publishing Company, Houston, Texas. All rights reserved.

Table 10-14
Thermal Conductivity—Special Materials

Material	k, Btu/(hr) (ft²) (°F)/(ft)
Carbon	3
Graphite	90
Karbate®, carbon base	3
Karbate®, graphite base	80
Teflon®	0.11
Glass (chemical resistant)	6.9

*To convert to Btu/(hr) (ft²) (°F)/(in.), multiply by 12.
To convert to gram calories/(sec) (cm²) (°C)/(cm), multiply by 0.004134.

This value is then used to represent the film coefficient equivalent to the converted inside coefficient, as h_{io}.

Figure 10-45 is convenient for solving for a clean U using known or estimated film coefficients only.

Example 10-8. Calculation of Overall Heat Transfer Coefficient from Individual Components

An exchanger has been examined, and the following individual coefficients and resistances determined. What is the overall coefficient of heat transfer referenced to outside coefficients? (Methods for determining these film coefficients are given later).

Film coefficient outside tube, $h_o = 175$
Film coefficient inside tube, $h_i = 600$
Fouling factor outside tube, $r_o = 0.001$
Fouling factor inside tube, $r_i = 0.0025$
Tube wall 1 in.–16 ga, Admiralty, 0.065-in. thick, $k_w = 768$ Btu/(ft²) (hr)/(°F)/(in.)
Inside area/ft = $A_i = 0.2278$ ft²/ft
Outside area/ft = $A_o = 0.2618$ ft²/ft

$$U_o = \frac{1}{\frac{1}{175} + 0.001 + \frac{0.065}{768} + 0.0025 + \frac{1}{600\left(\frac{0.2278}{0.2618}\right)}}$$

$$U_o = \frac{1}{0.00571 + .001 + 0.0000846 + 0.0025 + 0.00192}$$

$$U_o = 1/0.01121 = 89.1 \text{ Btu/hr (ft}^2\text{) (°F)}$$

Note the relative effects of the tube wall resistance when compared to the fouling factors in this case.

Approximate Values for Overall Coefficients

Various fluid heat transfer operations can be characterized in a general way by values of the overall coefficient, U. The values given in Perry[94] cannot be all-inclusive for every situation. However, they are suitable for use in estimating exchanger performance and in checking (approximately) the calculated values and similar nonexact comparisons. Table 10-15 lists a variety of applications and the corresponding overall U values and fouling factors. In general, these units have performed without difficulty, although the questions that cannot be answered are whether they may have been too large or how much too large they may have been.

Tables 10-16, 10-17, 10-18, and 10-18A give general estimating overall coefficients, and Table 10-19 gives the range of a few common film coefficients. Table 10-20 illustrates the effect of tube-wall resistance for some special construction materials. Table 10-20A lists estimating coefficients for glass-lined vessels. Also see Reference 215. See Table 10-24 for suggested water rates inside tubes.

For steam jacketed, agitated closed reactor kettles, the overall U usually will range from 40–60 Btu/hr (ft²) (°F). Of course, the significant variables are the degree or type of internal wall turbulence and the viscosity and thermal characteristics of the internal fluid. For water or other liquid cooling in the reactor jacket, the U value usually ranges from 20–30.

For duPont's Teflon® tube ($^1/_4$-in. diameter) heat exchangers (Figure 10-8) for condensing, heating, and cooling service, the U values range from 15–35. Little or no fouling occurs on the Teflon® surface.

Figure 10-39 presents the effect of total fouling on the overall coefficient. For example, if a clean nonfouled coefficient is corrected to the fouled condition by one overall fouling factor, the effect of changing the expected amount of fouling to another value can be readily determined.

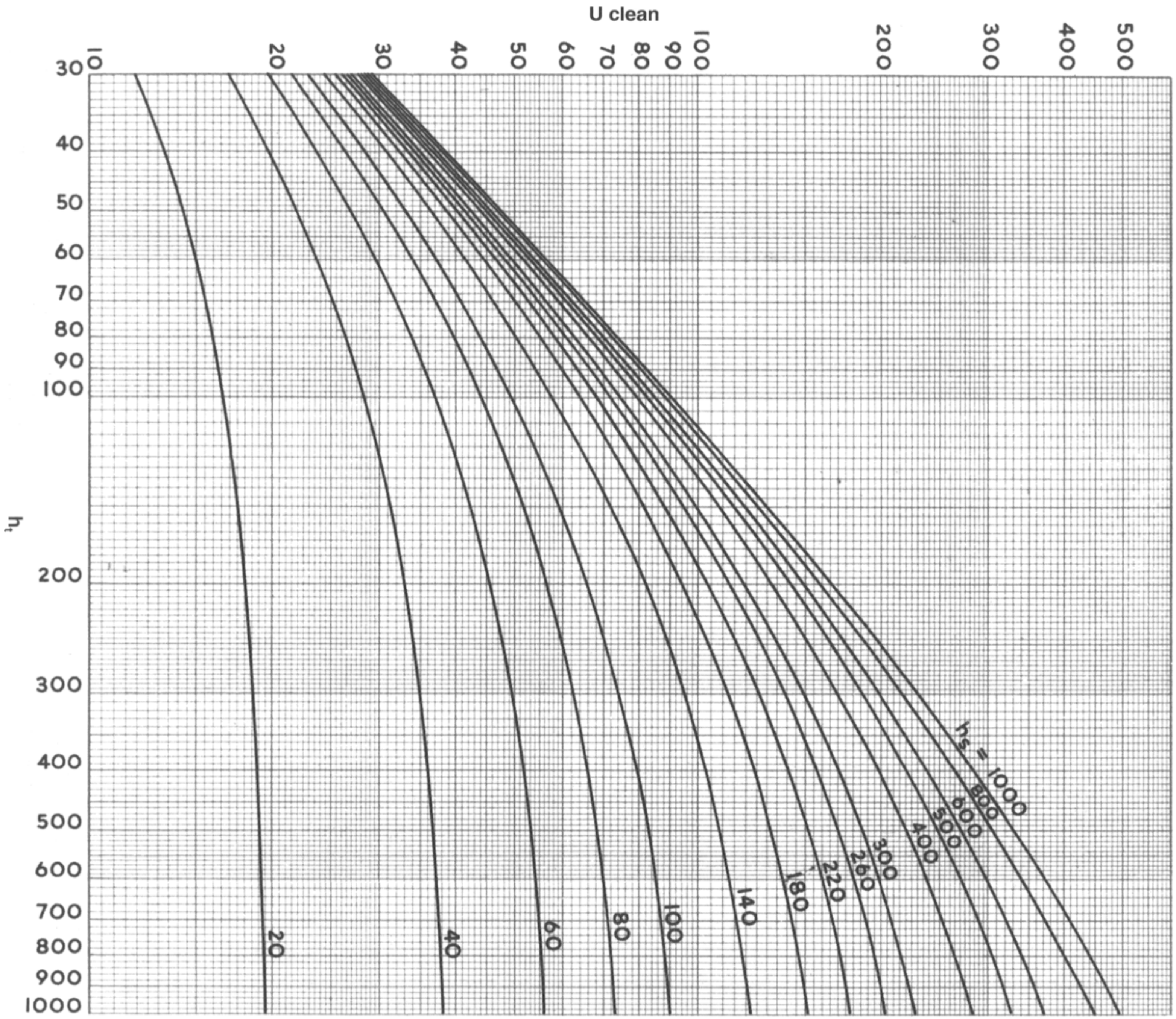

Figure 10-45. Chart for determining U-clean from tube-side and shell-side fluid film coefficients; no fouling included. Note: s = shell side, t = tube-side. (Used by permission: *Elements of Heat Transfer,* ©1957. Brown and Root, Inc.)

Table 10-15
Overall Coefficients in Typical Petrochemical Applications

U, Overall Coefficient, Btu/(hr) (ft^2)(°F)

In Tubes	Outside Tubes	Type Equipment	Velocities, Ft/Sec Tube	Velocities, Ft/Sec Shell	Overall Coefficient	Temp. Range, °F	Estimated Fouling Tube	Estimated Fouling Shell	Estimated Fouling Overall
A. Heating-Cooling									
Butadiene mix. (super-heating)	Steam	H	25–35	...	12	400–100	...	...	0.04
Solvent	Solvent	H	...	1.0–1.8	35–40	110–30	...	...	0.0065
Solvent	Propylene (vaporization)	K	1–2	...	30–40	40–0	...	...	0.006
C_4 unsaturates	Propylene (vaporization)	K	20–40	...	13–18	100–35	...	...	0.005
Solvent	Chilled water	H	...	...	35–75	115–40	0.003	0.001	...
Oil	Oil	H	...	...	60–85	150–100	0.0015	0.0015	...
Ethylene-vapor	Condensate and vapor	K	...	...	90–125	600–200	0.002	0.001	...
Ethylene vapor	Chilled water	H	...	...	50–80	270–100	0.001	0.001	...
Condensate	Propylene (refrigerant)	K–U	...	...	60–135	60–30	0.001	0.001	...
Chilled water	Transformer oil	H	...	...	40–75	75–50	0.001	0.001	...
Calcium brine-25%	Chlorinated C_1	H	1–2	0.5–1.0	40–60	−20–+10	0.002	0.005	...
Ethylene liquid	Ethylene vapor	K–U	...	...	10–20	−170–(−100)	...	...	0.002
Propane vapor	Propane liquid	H	...	...	6–15	−25–100	...	...	0.002
Lights and chlor. HC	Steam	U	...	...	12–30	−30–260	0.001	0.001	...
Unsat. light HC, CO, CO_2, H_2	Steam	H	...	...	10–2	400–100	...	...	0.3
Ethanolamine	Steam	H	...	...	15–25	400–40	0.001	0.001	...
Steam	Air mixture	U	...	...	10–20	−30–220	0.0005	0.0015	...
Steam	Styrene and tars	U (in tank)	...	...	50–60	190–230	0.001	0.002	...
Chilled water	Freon-12	H	4–7	...	100–130	90–25	0.001	0.001	...
Water*	Lean copper solvent	H	4–5	...	100–120	180–90	...	...	0.004
Water	Treated water	H	3–5	1–2	100–125	90–110	...	...	0.005
Water	C_2-chlor. HC, lights	H	2–3	...	6–10	360–100	0.002	0.001	...
Water	Hydrogen chloride	H	...	...	7–15	230–90	0.002	0.001	...
Water	Heavy C_2-chlor.	H	...	...	45–30	300–90	0.001	0.001	...
Water	Perchlorethylene	H	...	...	55–35	150–90	0.001	0.001	...
Water	Air and water vapor	H	...	...	20–35	370–90	0.0015	0.0015	...
Water	Engine jacket water	H	...	...	230–160	175–90	0.0015	0.001	...
Water	Absorption oil	H	...	...	80–115	130–90	0.0015	0.001	...
Water	Air-chlorine	U	4–7	...	8–18	250–90	...	...	0.005
Water	Treated water	H	5–7	...	170–225	200–90	0.001	0.001	...
B. Condensing									
C_4 unsat.	Propylene refrig.	K	v	...	58–68	60–35	...	...	0.005
HC unsat. lights	Propylene refrig.	K	v	...	50–60	45–3	...	...	0.0055
Butadiene	Propylene refrig.	K	v	...	65–80	20–35	...	...	0.004
Hydrogen chloride	Propylene refrig.	H	...	...	110–60	0–15	0.012	0.001	...
Lights and chloro-ethanes	Propylene refrig.	KU	...	...	15–25	130–(−20)	0.002	0.001	...
Ethylene	Propylene refrig.	KU	...	...	60–90	120–(−10)	0.001	0.001	...
Unsat. Chloro HC	Water	H	7–8	...	90–120	145–90	0.002	0.001	...
Unsat. Chloro HC	Water	H	3–8	...	180–140	110–90	0.001	0.001	...
Unsat. Chloro HC	Water	H	6	...	15–25	130–(−20)	0.002	0.001	...
Chloro-HC	Water	KU	...	...	20–30	110–(−10)	0.001	0.001	...
Solvent and noncond.	Water	H	...	...	25–15	260–90	0.0015	0.004	...
Water	Propylene vapor	H	2–3	...	130–150	200–90	...	...	0.003
Water	Propylene	H	...	...	60–100	130–90	0.0015	0.001	...
Water	Steam	H	...	...	225–110	300–90	0.002	0.0001	...
Water	Steam	H	...	...	190–235	230–130	0.0015	0.001	...
Treated water	Steam (exhaust)	H	...	...	20–30	220–130	0.0001	0.0001	...
Oil	Steam	H	...	...	70–110	375–130	0.003	0.001	...
Water	Propylene cooling and cond.	H	...	...	{ 25–50 110–150	30–45 (C) 15–20 (Co)	0.0015	0.001	...

In Tubes	Outside Tubes	Type Equipment	Velocities, Ft/Sec Tube	Shell	Overall Coefficient	Temp. Range, °F	Estimated Fouling Tube	Shell	Overall
Chilled water	Air-chlorine (part. cond.)	U	...	...	8–15 20–30	8–15 (C) 10–15 (Co)	0.0015	0.005	...
Water	Light HC, cool and cond.	H	...	...	35–90	270–90	0.0015	0.003	...
Water	Ammonia	H	...	...	140–165	120–90	0.001	0.001	...
Water	Ammonia	U	...	...	280–300	110–90	0.001	0.001	...
Air-Water vapor	Freon	KU	...	...	10–50 10–20	60–10	...	...	0.01
C. Reboiling									
Solvent, Copper-NH_3	Steam	H	7–8	...	130–150	180–160	...	...	0.005
C_4 Unsat.	Steam	H	...	...	95–115	95–150	...	...	0.0065
Chloro. HC	Steam	VT	...	...	35–25	300–350	0.001	0.001	...
Chloro. unsat. HC	Steam	VT	...	...	100–140	230–130	0.001	0.001	...
Chloro. ethane	Steam	VT	...	...	90–135	300–350	0.001	0.001	...
Chloro. ethane	Steam	U	...	...	50–70	30–190	0.002	0.001	...
Solvent (heavy)	Steam	H	...	...	70–115	375–300	0.004	0.0005	...
Mono–di-ethanolamines	Steam	VT	...	...	210–155	450–350	0.002	0.001	...
Organics, acid, water	Steam	VT	...	...	60–100	450–300	0.003	0.0005	...
Amines and water	Steam	VT	...	...	120–140	360–250	0.002	0.0015	...
Steam	Naphtha frac.	Annulus, long. F.N.	...	...	15–20	270–220	0.0035	0.0005	...
Propylene	C_2, C_2	KU	...	...	120–140	150–40	0.001	0.001	...
Propylene-butadiene	Butadiene, unsat.	H	...	25–35	15–18	400–100	...	...	0.02

*Unless specified, all water is untreated, brackish, bay or sea.

Notes: H = horizontal, fixed or floating tubesheet; T = thermosiphon; V = vertical; (C) = cooling range Δt
U = U-tube horizontal bundle; v = variable; R = reboiler; (Co) = condensing range $\Delta\tau$
K = kettle type; HC = hydrocarbon; Data/results based on actual and specific industrial equipment.

Table 10-16
General Evaporator Overall Coefficient, U

	U, Btu/hr(ft²)(°F)
Long-tube vertical evaporator	
Natural circulation	200–600
Forced circulation	400–2,000
Short-tube evaporators	
Horizontal	200–400
Calandria (vertical, thermosiphon)	150–500
Coil evaporators	200–400
Agitated-film evaporators, Newtonian liquid	
1 centipoise	400
100 centipoise	300
10,000 centipoise	120

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Figure 10-45 can be used to solve the overall coefficient, U, equation for the clean coefficient, composed of the tube-side and shell-side film coefficients only. Correction for tube-side and shell-side scaling and tube-wall resistance can be added by looking up the clean U on Figure 10-39 and reading the dirty or fouled U value or by using Equation 10-42 developed by Hedrick,[159] which is reported to produce smooth curves for all values of L/d from 2 to 50 and across the Reynolds number range of 2,000 to 10,000.

$$h_{io} = (16.1/d_o)\,[B_l\,k\,(c\mu/k)^{1/3}\,(\mu/\mu_w)^{0.14}] \qquad (10\text{-}42)$$

where

$$B = (-3.08 + 3.075X + 0.32567X^2 - 0.02185X^3)\,(10\,d_i/L)\,[1 - (X/10)^{0.256}] \qquad (10\text{-}43)$$

$X = N_{Re} / 1{,}000$
R_{Re} = Reynolds number
d_i = inside tube diameter, in.
d_o = outside tube diameter, in.
k = thermal conductivity, Btu/hr-ft-°F
c = specific heat of fluid, Btu/lb-°F
μ = viscosity of fluid, centipoise
μ_w = viscosity at the wall, centipoise
L = tube length, ft
h_{io} = inside film coefficient based on the outside tube diameter, Btu/(hr) (ft²)(°F)

Table 10-17
Approximate Overall Heat Transfer Coefficient, U*

Use as a guide to the order of magnitude and not as limits to any value. Coefficients of actual equipment may be smaller or larger than the values listed.

Hot Fluid	Cold Fluid	U, Btu/hr (ft^2)(°F)
Condensing		
Steam (pressure)	Water	350–750
Steam (vacuum)	Water	300–600
Saturated organic solvents near atmospheric	Water	100–200
Saturated organic solvents, vacuum with some noncondensable	Water, brine	50–120
Organic solvents, atmospheric and high noncondensable	Water, brine	20–80
Aromatic vapors, atmospheric with noncondensables	Water	5–30
Organic solvents, vacuum and high noncondensables	Water, brine	10–50
Low boiling atmospheric	Water	80–200
High boiling hydrocarbon, vacuum	Water	10–30
Heaters		
Steam	Water	250–750
Steam	Light oils	50–150
Steam	Heavy oils	10–80
Steam	Organic solvents	100–200
Steam	Gases	5–50
Dowtherm	Gases	4–40
Dowtherm	Heavy oils	8–60
Flue gas	Aromatic HC and steam	5–15
Evaporators		
Steam	Water	350–750
Steam	Organic solvents	100–200
Steam	Light oils	80–180
Steam	Heavy oils (vacuum)	25–75
Water	Refrigerants	75–150
Organic solvents	Refrigerants	30–100
Heat Exchangers (No Change of Phase)		
Water	Water	150–300
Organic solvents	Water	50–150
Gases	Water	3–50
Light oils	Water	60–160
Heavy oils	Water	10–50
Organic solvents	Light oil	20–70
Water	Brine	100–200
Organic solvents	Brine	30–90
Gases	Brine	3–50
Organic solvents	Organic solvents	20–60
Heavy oils	Heavy oils	8–50

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Table 10-18
Approximate Overall Heat Transfer Coefficient, U

Condensation		
Process Side (Hot)	**Condensing Fluid (Cold)**	
Hydrocarbons (light)	Water	100–160
Hydrocarbons w/ inerts (traces)	Water	30–75
Organic vapors	Water	70–160
Water vapor	Water	150–340
Water vapor	Hydrocarbons	60–150
Exhaust steam	Water	280–450
Hydrocarbons (light)	Refrigerant	45–110
Organics (light)	Cooling brine	50–120
Gasoline	Water	65–130
Ammonia	Water	135–260
Hydrocarbons (heavy)	Water	40–75
Dowtherm vapor	Liquid organic	75–115
Vaporization		
Process Side (Vaporized)	**Heating Fluid**	
Hydrocarbons, light	Steam	90–210
Hydrocarbons, C4–C8	Steam	75–150
Hydrocarbons, C3–C4 (vac)	Steam	45–175
Chlorinated HC	Steam	90–210
HCl solution (18–22%)	Steam	120–240
Chlorine	Steam	130–220
Coils in Tank		
Coil Fluid	**Tank Fluid**	
Steam	Aqueous sol'n. (agitation)	140–210
Steam	Aqueous sol'n. (no agitation)	60–100
Steam	Oil-heavy (no agitation)	10–25
Steam	Oil-heavy (agitation)	25–55
Steam	Organics (agitation)	90–140
Hot water	Water (no agitation)	35–65
Hot water	Water (agitation)	90–150
Hot water	Oil-heavy (no agitation)	6–25
Heat transfer oil	Organics (agitation)	25–50
Salt brine	Water (agitation)	50–110
Water (cooling)	Glycerine (agitation)	50–75

Film Coefficients with Fluid Inside Tubes, Forced Convection

Useful dimensionless groups for heat transfer calculation are[70]

Symbol	Name	Function
Gz	Graetz number	wc/kL
Gr	Grashof number	$D^3\rho^2 g\beta\Delta t/\mu^2$
Nu	Nusselt number	$h\,D/k$
Pe	Peclet number	$DG\,c/k$
Pr	Prandtl number	$c\,\mu/k$
Re	Reynolds number	DG/μ, or, $D\,u\,\rho/\mu$
Sc	Schmidt number	$\mu/\rho k_d$
St	Stanton number	h/cG

where

C or c = specific heat of fluid, Btu/(lb)(°F)
D = inside diameter of tube, ft
G = mass velocity, lb/hr (ft^2)
g = acceleration of gravity, ft/(hr^2), or ft/sec/sec
Depends on the system of units
H = heat transfer film coefficient, Btu/(hr) (ft^2)(°F)
j_H = factor for heat transfer, dimensionless
k = thermal conductivity, Btu/(hr) (ft^2)(°F/ft)
L = length, ft
Δt = temperature difference for heat transfer, °F
v = velocity, ft/hr
μ = viscosity, absolute, lb/(hr)(ft)
ρ = density, lb/ft^3
k_d = diffusivity (volumetric), ft^2/hr
β = thermal coefficient of expansion, 1/°F

Gas and liquid heat transfer inside tubes has been studied by Sieder and Tate[101] and is represented by Figure 10-46. Also see references 270 and 271.

Table 10-19
Approximate Film Coefficients, h_i or h_o

	Film Coefficient, Btu/hr. (ft^2)(°F)
No Change of Phase	
Water	300–2000
Gases	3–50
Organic solvents	60–500
Oils	10–120
Condensing	
Steam	1000–3000
Organic solvents	150–500
Light oils	200–400
Heavy oils (vacuum)	20–50
Ammonia	500–1000
Evaporation	
Water	800–2000
Organic solvents	100–300
Ammonia	200–400
Light oils	150–300
Heavy oils	10–50

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Table 10-18A
Miscellaneous Heating Overall Heat Transfer Coefficient,[1] U

	Equipment	Material Treated	Agitation	Heating Surface	Overall Coeff. Btu/hr (Ft^2)(°F)	Controlling Resistance
Steam	Paper drying rolls	Paper		Cast iron	20–58	Paper film*
Steam	Yankee drying rolls	Paper		Cast iron	100	Metal wall*
Steam	Jacketed kettle	Paraffin wax	None	Copper	27.4	Product[δ]
Steam	Jacketed kettle	Paraffin wax	Scraper	Cast iron	107	Product[δ]
Steam	Jacketed kettle	Boiling water	None	Steel	187	Product[δ]
DOWTHERM A (vapor phase)	Jacketed kettle	Varnish	None	Steel	20–50	Product
DOWTHERM A (vapor phase)	Jacketed kettle	Asphalt	None	Steel	8–20	Product
DOWTHERM A (vapor phase)	Heat exchanger	Fatty acids	Forced circn.	Steel	45–50	Product
DOWTHERM A (vapor phase)	Heat exchanger	Rosin	Forced circn.	Steel	40	Product
DOWTHERM A (vapor phase)	Heat exchanger	Asphalt	Forced circn.	Steel	25–50	Product
DOWTHERM A (vapor phase)	Reboiler	Fatty acids	None		80	Product
DOWTHERM A (liquid phase)	Jacketed kettle	Varnish	Forced circn.	Steel	15–40	Product
DOWTHERM A (vapor phase)	Heat exchanger	Edible oil	Forced circn.	Steel	124–150	Product
DOWTHERM A (vapor phase)	Heat exchanger	Cocoa butter	Forced circn.	Steel	70–75	Product
Mercury vapor	Heat exchanger	DOWTHERM A MEDIUM	Forced circn.	Steel	220–350	Product

Approximate Overall Design Coefficient[1]

	Hot Fluid	Cold Fluid	Overall U
Coolers	Light organic	Water	75–150
Heaters	Steam	Light organic	100–200
Exchangers	Light organic	Light organic	40–75
	Heavy organic	Light organic	30–60
	Light organic	Heavy organic	10–40

[1] Values include dirt factors of 0.003 and allowable pressure drops at 5–10 psi on the controlling steam. DOWTHERM fluid would be included as light organic. (Kern. *Process Heat Transfer,* 1st Ed., p. 840 ©1950.)

*Montgomery, A. E. "Heat Transfer Calens. In Paper Machine." *Paper Trade Journal,* Oct. 3, 1946, p. 29.

[δ]Perry. *Chemical Engineers Handbook,* 3rd Ed., p. 482, ©1950.

Table 10-20
Effect of Tube Wall Material and Film Conditions on Overall Coefficient

Tube	Overall Coefficient, U, Btu/hr. (ft²)(°F) Percentages in ( ) refer to graphite tube			
	Heating Water with Steam	Condensing Organic Vapor with Water	Cooling Organic Liquid with Water	Cooling Viscous Organic Liquid with Water
Stainless steel, 304–16 BWG	184(92.5)	79(96.5)	43(100)	18.9(100)
Impervious graphite, $^{3}/_{16}$ in. tk. wall	199(100)	82(100)	43(100)	18.9(100)
Glass, 0.0625 in. wall	89(44.7)	56(68.3)	36(82.5)	17.3(91.6)
*Stainless steel 304, reactor, $^{21}/_{32}$ in. wall	83(41.7)	54(65.2)	35(80.9)	17.0(89.9)
*Glassed-steel reactor, pipe, $^{11}/_{16}$ in. steel wall	71(35.7)	48(58.5)	32(73.8)	16.3(86.2)
Film coefficients only, $h_i + h_o$	300	100	50	20

*Thickness based on 1,000-gal reactors for service at same pressure.

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Table 10-20A
Estimating Overall Coefficient, U, for Special Applications

Material of Construction (Barrier Material)	Overall Heat Transfer Coefficient (Service U)* Btu/(hr) (ft²)(°F)**			
	Heating Water with Steam	Heating Water with Heat Transfer Oil	Cooling Organic Liquid with Water	Cooling Viscous Organic Liquid with Water
Stainless steel reactor, 0.656 in. wall[§]	90.2	62.2	35.1	16.7
Glasteel reactor, 0.05 in. glass, 0.688 in. steel[§]	77.0	55.6	32.6	16.5
Combined film conductance, $\frac{h_i h_o}{h_i + h_o}$	300	137	50	20

*Fouling factors typical to process fluids and materials of construction are included.
**Multiply by 4.882 for conversion to kcal/(hr)(m²)(°C).
§Thickness based on 1,000-gal reactors for service at same pressures.

The equations representing portions of the graph are as follows:

A. For viscous streamline flow of organic liquids, water solutions (not water) and gases with DG/μ <2,100 in horizontal or vertical tubes: deviation 6 12%.[70]

$$\frac{h_i D}{k_a} = 1.86\left[\left(\frac{DG}{\mu}\right)\left(\frac{c\mu}{k_a}\right)\left(\frac{D}{L}\right)\right]^{1/3}\left(\frac{\mu}{\mu_w}\right)^{0.14} \quad (10\text{-}44)$$

$$= 1.86\left(\frac{4_{wc}}{\pi k_a L}\right)^{1/3}\left(\frac{\mu}{\mu_w}\right)^{0.14} \quad (10\text{-}45)$$

$$\frac{h_i}{cG} = 1.86\left(\frac{\mu}{DG}\right)^{2/3}\left(\frac{k_a}{\mu c}\right)^{2/3}\left(\frac{D}{L}\right)^{1/3}\left(\frac{\mu}{\mu_w}\right)^{0.14} \quad (10\text{-}46)$$

B. For turbulent flow of viscous fluids as organic liquids, water solutions (not water) and gases with DG/μ>10,000 in horizontal or vertical tubes: deviation 115% to 210%.[70]

$$\frac{h_i D}{k_a} = 0.023\left(\frac{DG}{\mu}\right)^{0.8}\left(\frac{c\mu}{k_a}\right)^{1/3}\left(\frac{\mu}{\mu_w}\right)^{0.14} \quad (10\text{-}47)$$

C. For transition region between streamline and turbulent flow, use Figure 10-46.

where

c = specific heat of fluid at constant pressure, Btu/(lb) (°F)
D = inside diameter of tube, ft
G = fluid mass velocity, lb/(hr) (ft² tube cross-section flow area)

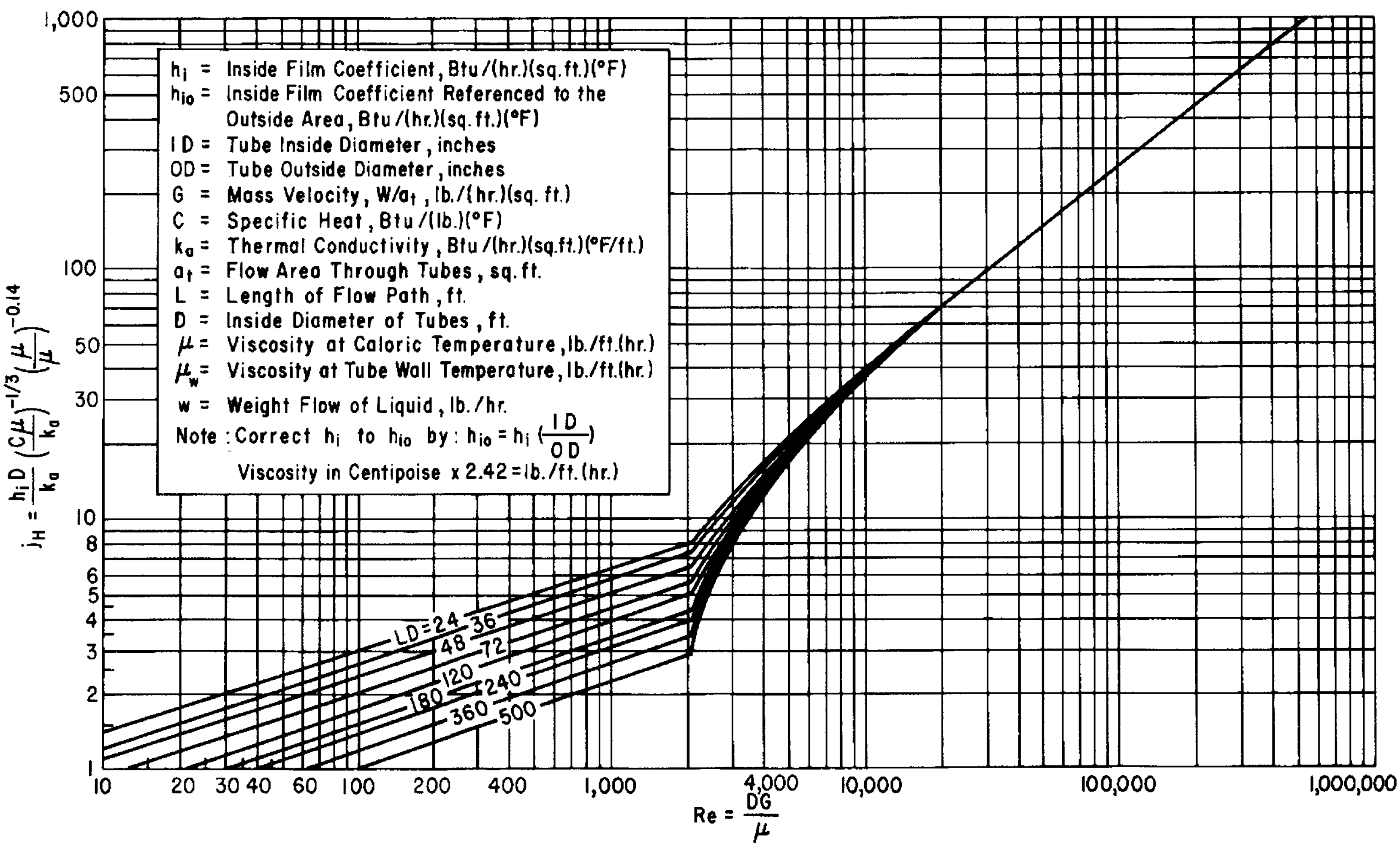


Figure 10-46. Tube-side heat transfer, heating and cooling. (Adapted and used by permission: Kern, D. Q. *Process Heat Transfer*, 1st Ed., ©1950. McGraw-Hill Book Co. All rights reserved. Originally adapted by Kern from Sieder and Tate.)

h_i = film heat transfer coefficient inside tube, Btu/(hr) (ft²)(°F)
k_a = thermal conductivity of fluid at average bulk temperature of fluid, Btu/hr (ft²)/(°F/ft)
L = total heated or cooled length of heat transfer path, ft
w = weight rate of fluid flow per tube, lb/hr
μ = viscosity of fluid, lb/(hr) (ft)
μ_w = viscosity of fluid at wall temperature, lb/(hr) (ft)

Figures 10-47, 10-48, and 10-49 are useful in solving the equivalent of Equation 10-47 for turbulent as well as streamline flow of gases and vapors inside tubes. To use the charts

A. For turbulent flow:

1. Determine ϕ_p using fluid properties (Figure 10-47).
2. Determine tube-side film coefficient, h_i, based on inside tube surface (Figure 10-48).
3. Correct h_i for the effect of tube size by multiplying by the accompanying factor shown in Figure 10-48.

B. For streamline flow:

1. Determine ϕ_p (Figure 10-47).
2. Determine h_i (Figure 10-49).
3. Correct h_i by multiplying by the tube size and heated length factors accompanying Figure 10-49.

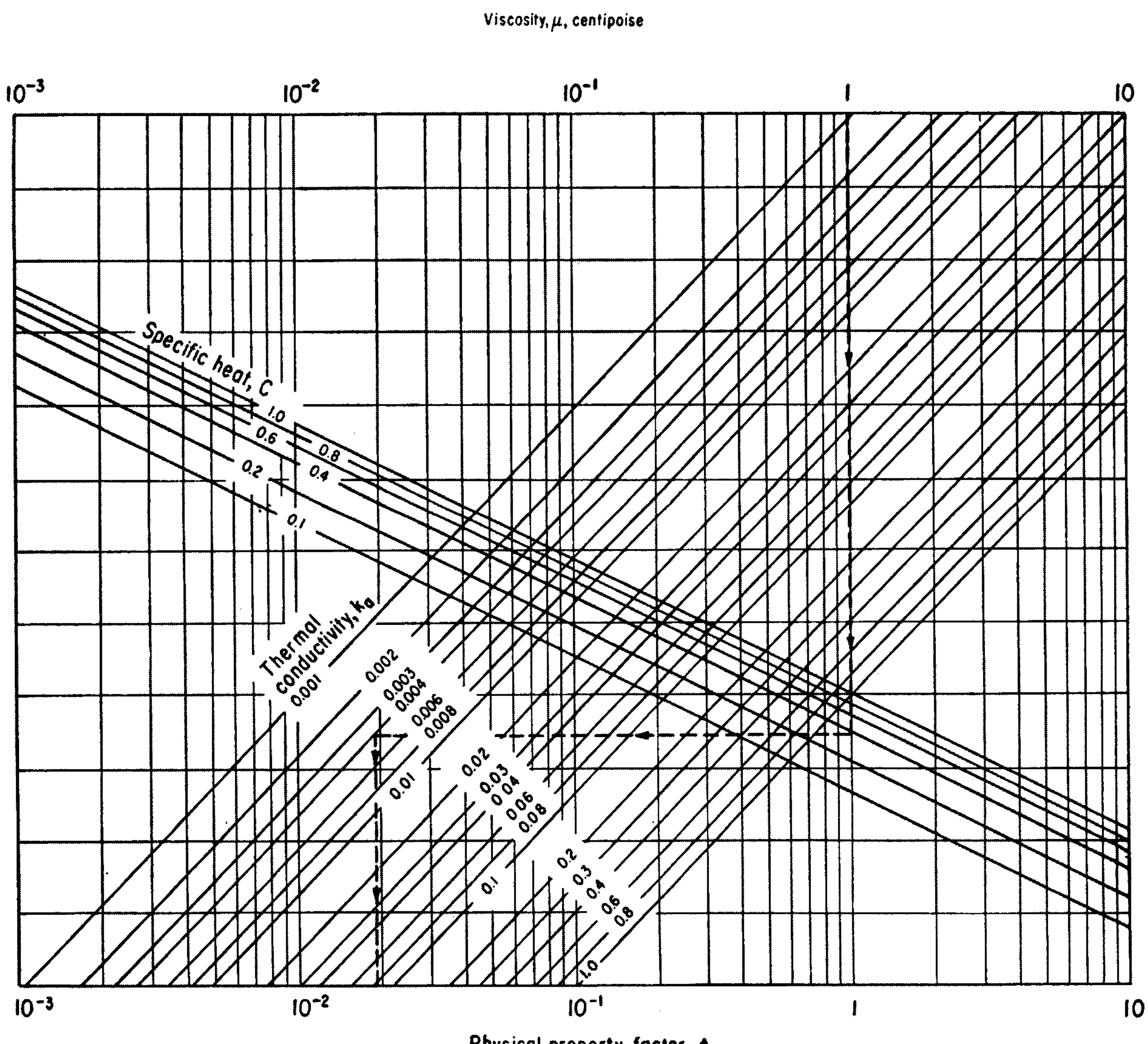
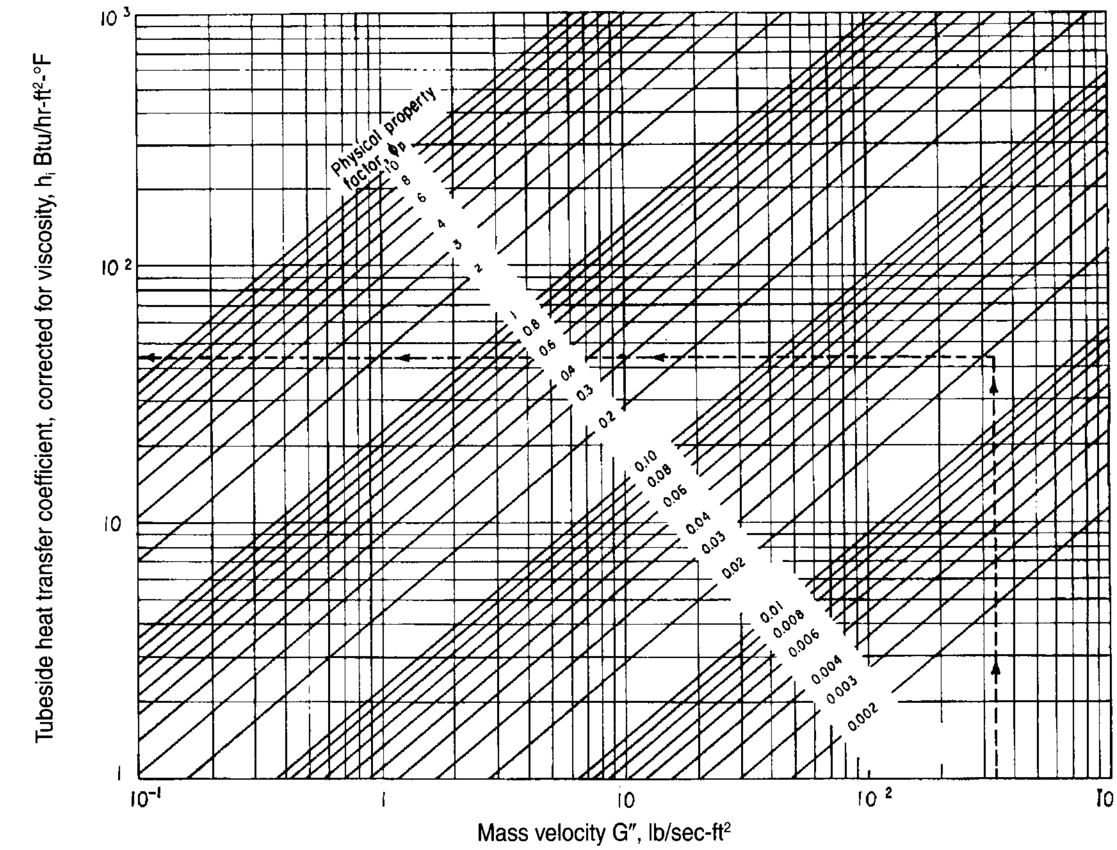


Figure 10-47. Flow inside tubes for gas and vapors. Physical property factor depends on viscosity, specific heat, and thermal conductivity. (Used by permission: Ning Hsing Chen, *Chemical Engineering*, V. 66, No. 1, ©1959. McGraw-Hill, Inc. All rights reserved.)



Tube Size Correction Factors for Turbulent Flow

$^3/_4$" × 14 BWG	1.011	1" × 14 BWG	0.942
16	1.000	$1^1/_4$" × 10	0.912
1" × 10	0.967	12	0.903
12	0.955	14	0.894

Figure 10-48. Flow inside tubes for gases and vapors. Heat transfer coefficient for vapors and gases in turbulent flow. (Used by permission: Ning Hsing Chen, *Chemical Engineering*, V. 66, No. 1, ©1959. McGraw-Hill, Inc. All rights reserved.)

D. For water, the inside film coefficient is represented by Figure 10-50A. Furman[49] presents charts that reduce the expected deviation of the film coefficient from the ±20% of Figure 10-50A, 10-50B, 10-50C, and 10-50D.

E. For heating and cooling turbulent gases and other low viscosity fluids at $DG/\mu > 8{,}000$; the Dittus-Boelter relation is used. See Figures 10-46, 10-51, and 10-52.

$$\frac{h_i D}{k_a} = 0.0243\left(\frac{DG}{\mu}\right)^{0.8}\left(\frac{c\mu}{k_a}\right)^{0.4}\left(\frac{\mu}{\mu_w}\right)^{0.14} \quad (10\text{-}48)$$

For cooling, $DG/\mu > 8{,}000$, the following is sometimes used in place of the preceding relation, Figure 10-51:

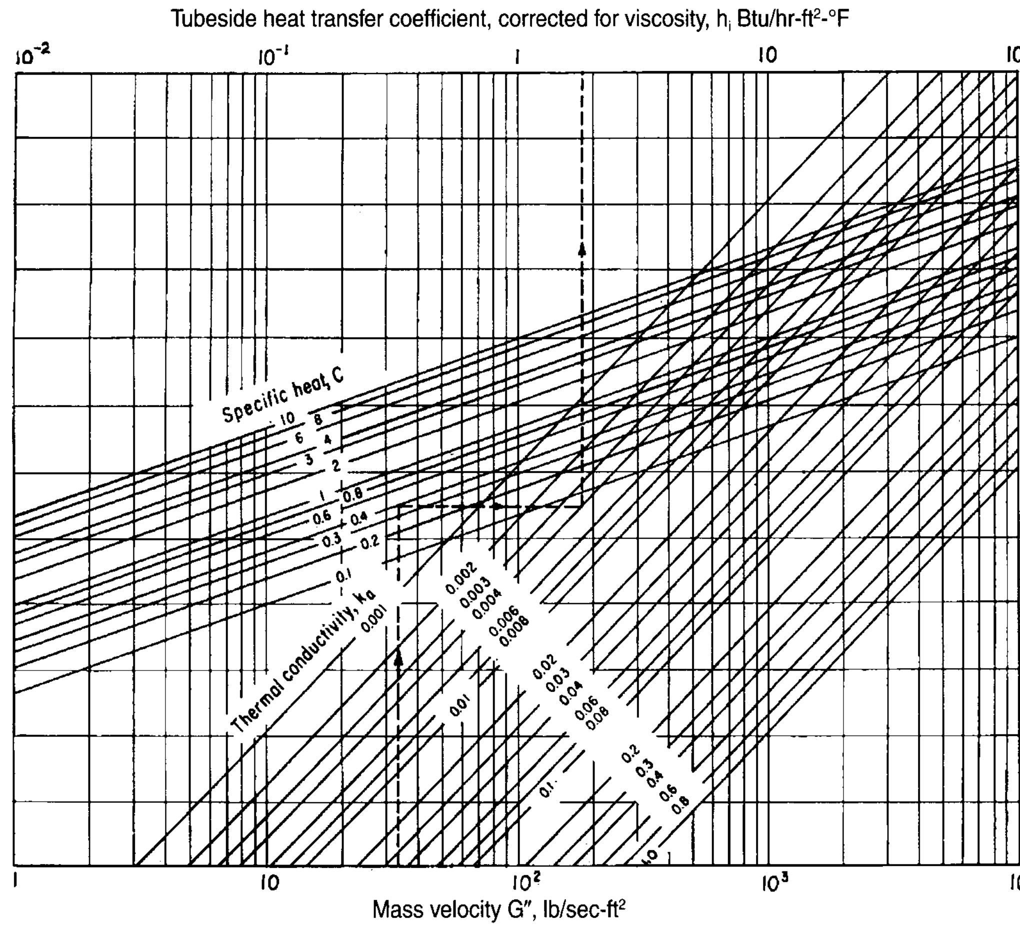
$$\frac{h_i D}{k_a} = 0.023\left(\frac{DG}{\mu}\right)^{0.8}\left(\frac{c\mu}{k_a}\right)^{0.4}\left(\frac{\mu}{\mu_w}\right)^{0.14} \quad (10\text{-}49)$$

When the Prandtl number $(c\mu/k_a)$ can be used at 0.74, as is the case for so many gases such as air, carbon monoxide, hydrogen, nitrogen, oxygen, a close group of ammonia (0.78), and hydrogen sulfide (0.77), this relation reduces to the following:

$$\frac{h_i D}{k_a} = 0.026\left(\frac{DGC}{k_a}\right)^{0.8} \quad (10\text{-}50)$$

Note that the values of the initial coefficients on the right-side of the preceding equations vary significantly among several respected references; therefore, the engineer should not be surprised to note these variations in the literature.

Pierce[164] proposes and illustrates good agreement between the test data and the correlation for a smooth continuous curve for the Colburn factor over the entire range of Reynolds numbers for the laminar, transition, and turbulent flow regimes inside smooth tubes:



Heated Length Correction Factors, Streamline Flow			
8 ft	1.26	16 ft	1.00
10	1.17	18	0.96
12	1.10	20	0.96
14	1.05		

Tube Size Correction Factors for Streamline Flow			
$^3/_4$ in. × 14 BWG	1.060	1 in. × 14 BWG	0.744
16	1.000	1 $^1/_4$ in. × 10 BWG	0.631
1 in. × 10	0.846	12	0.600
12	0.793	14	0.571

Figure 10-49. Flow inside tubes for gases and vapors. Heat transfer coefficient for streamline flow. (Used by permission: Ning Hsing Chen, *Chemical Engineering,* V. 66, No. 1, ©1959. McGraw-Hill, Inc. All rights reserved.)

$$J_4 = \left[\left(\frac{1}{N_{Re}^{9.36}}\right)\frac{1}{\left[\dfrac{N_{Re}^{1.6}}{7.831(10^{-14})} + \left(\dfrac{1.969(10^6)}{N_{Re}}\right)^8\right]^{3/2}}\right]^{1/12} \quad (4)$$

(10-51)

Colburn Factor, J:

$$J = J_4\ (\mu_b/\mu_w)^{0.14} \qquad (10\text{-}52)$$

Then, convective heat transfer coefficient:

$$h = J\ (Cp\ \rho\ v/N_{Pr}^{2/3}) \qquad (10\text{-}53)$$

where

Cp = specific heat, J/kg K = J/kg-Kelvin
D = diameter, m, meter
J = Colburn factor
J_4 = Colburn factor given by equation proposed by Pierce
L = length of tube, m
N_{Pr} = Prandtl number
N_{Re} = Reynolds number
v = velocity, m/sec
μ = dynamic viscosity, sPa (pascal-sec)
ρ = density, kg/m^3
b = evaluate at bulk temperature
w = evaluate at wall temperature
kg = kilogram

Buthod[22] presents Figure 10-52 for gases flowing inside tubes. Note that the coefficient refers to the outside tube surface area. It is useful for gases other than those shown because the scale can be multiplied by 10 to obtain the proper order of magnitude for specific heat.

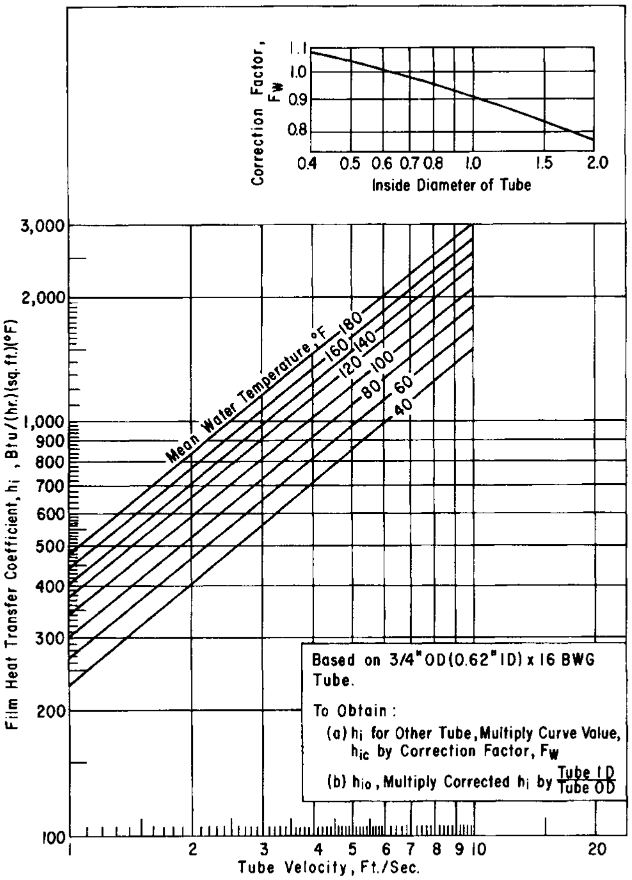


Figure 10-50A. Tube-side film heat transfer coefficient for water. (Used by permission: Kern, D. Q., *Process Heat Transfer*, 1st Ed., ©1950. McGraw-Hill, Inc. All rights reserved. Original adapted from Eagle and Ferguson, *Proc. Royal Society* A 127, 450, ©1930.)

Simplify the relation for heating and cooling gases, using

$c\mu/k_a = 0.78$ and $\mu = 0.435$ (Reference 81)

$$h = 0.0144\frac{cG^{0.8}}{D^{0.2}} \quad (10\text{-}54)$$

Note that below $G = 1{,}200P^{2/3}$, results may be too conservative.
Gases in turbulent flow in circular helical coils:[81]
Multiply h_i for straight tubes by $[1 + 3.5d_{it}/D_H]$

where

d_{it} = inside tube diameter, in.
D_H = diameter of helix of coil, in.
P = absolute pressure, atm. (this equation only)

Ganapathy[263] developed nomograms for solving for film coefficients for superheated steam, gases, liquids, and vapor refrigerants flowing inside exchanger tubes. See Figures 10-53A, 10-53B, 10-53C, and 10-53D.

Also see Rubin, reference 280.

Liquids in turbulent flow in circular helical coils[80, 81] should be handled the same as for gases or use $1.2 \times h_1$ for straight tubes.

Film Coefficients with Fluids Outside Tubes
Forced Convection

Film coefficients for turbulent flow that exist on the outside or shell side of the conventional baffled shell and tube exchanger are correlated for hydrocarbons, organic compounds, water, aqueous solutions, and gases[5, 70] by

$$\frac{h_oD_e}{k_a} = 0.36\left(\frac{D_eG_s}{\mu}\right)^{0.55}\left(\frac{c\mu}{k_a}\right)^{1/3}\left(\frac{\mu}{\mu_w}\right)^{0.14} \quad (10\text{-}55)$$

and as represented in Figure 10-54, deviation: 0 to +20 percent. The G_s is correlated for both cross- and parallel-flow through the bundle by using the hydraulic radius along the tubes only.[70] Figure 10-55 is helpful in visualizing shell-side fluid flow.

where

h_o = film coefficient outside of tubes in bundle, Btu/hr (ft^2) (°F)
k_a = thermal conductivity, Btu/hr (ft^2) (°F/ft)
G_s = mass rate, lb/hr (ft^2)
D_e = equivalent tube diameter, ft
d_e = equivalent tube diameter, in.
a_s = flow area across the tube bundle, ft^2
B = baffle spacing, in.
c = specific heat of fluid, Btu/lb (°F)
μ = viscosity at the caloric temperature, lb/ft (hr)
μ_w = viscosity at the tube wall temperature, lb/ft (hr)

Kern's[70] correlation checks well for the data of Short,[102] Bowman,[7] and Tinker[116] for a wide variety of baffle cuts and spacing for segmental baffles with and without leakage as summarized by Donohue.[36] Short's data for disc and doughnut baffles is better calculated by[36]

$$\frac{hD}{k_a} = 0.23(d_e)^{0.6}\left(\frac{D_oG_w}{\mu}\right)^{0.6}\left(\frac{c\mu}{k_a}\right)^{0.33} \quad (10\text{-}56)$$

where

d_e = equivalent tube diameter for the shell side = 4 (flow area/wetted perimeter), in.
D_o = outside diameter of tube, ft
G_w = weighted mass velocity = $w/S_e = w/(G_cG_b)^{0.5}$ in lb/(hr) (ft^2)
S_e = weighted flow area = [(cross flow area) (baffle window area)]$^{0.5}$, ft^2
G_c = cross-flow mass velocity, lb/hr (ft^2)
G_b = mass velocity through baffle window opening, based on the area of the opening less the area of tubes passing through it, lb/hr (ft^2)
μ = viscosity, lb/hr (ft)

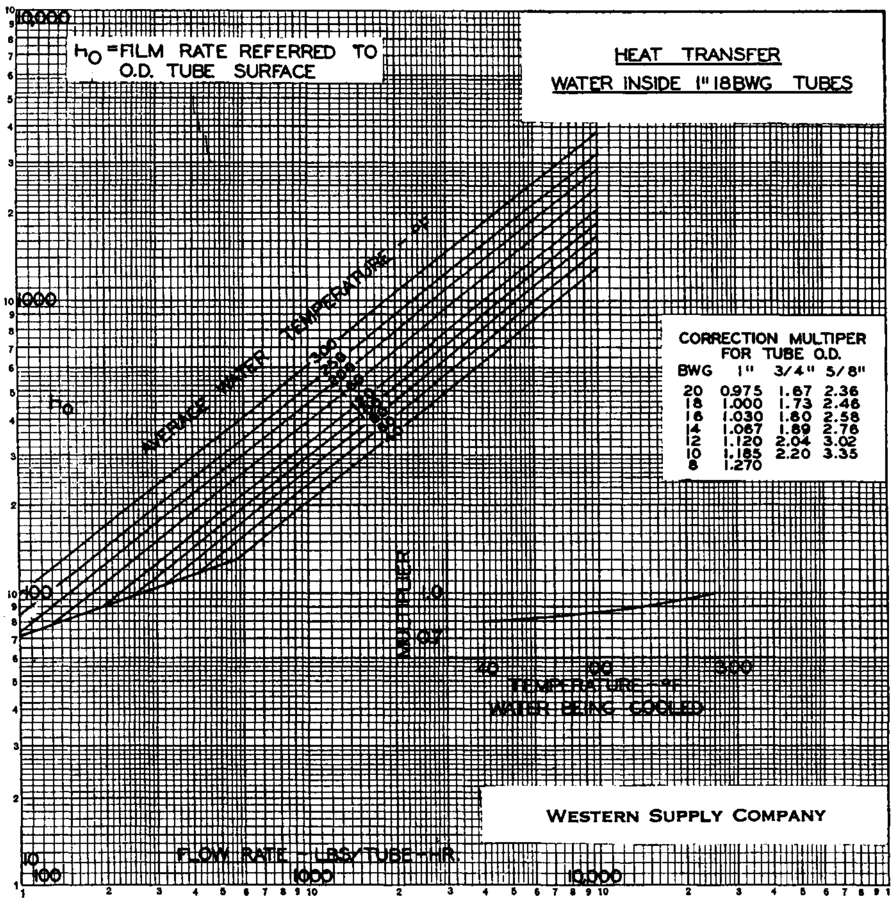


Figure 10-50B. Heat transfer film coefficient for water flowing inside 1 in. × 18 BWG tubes referred to outside tube surface area for plain tubes. Note the corrections for tubes of wall gauges other than 18 BWG. (Used by permission: J. B. Co., Inc., Western Supply Div., Tulsa, Okla.)

Shell-Side Equivalent Tube Diameter[70]

See Figure 10-56 and Table 10-21.

Best results are obtained when baffle pitch or spacing between baffles is between one-fifth to one shell diameter.

For square pitch tubes, the shell-side equivalent diameter is

$$d_e = \frac{4(p^2 - \pi d_o^2/4)}{\pi d_o}, \text{ in.} \tag{10-57}$$

For 60° triangular equilateral pitch tubes:

$$d_e = \frac{4[(0.5p)(0.86p) - 0.5\,\pi d_o^2/4]}{\dfrac{\pi d_o}{2}}, \text{ in.} \tag{10-58}$$

where

d_e = equivalent diameter, in., shell side for cross flow
p = tube pitch, in.
d_o = outside diameter of tube, in.

Cross-flow area for Figure 10-54 is based upon the maximum flow area at the nearest tube row to the centerline of the shell.[70] The length of the flow area is the baffle spacing.

$$a_s = \frac{D_s(c'B)}{p(144)}, \text{ ft}^2 \tag{10-59}$$

$$G_s = \frac{W}{a_s}, \text{ lb/(hr) (ft}^2) \tag{10-60}$$

where

D_s = shell inside diameter, in.
c′ = clearance between tubes measured along the tube pitch, in.
B = baffle spacing, in.
W = weight flow of fluid, lb/hr
p = tube pitch, in.

Baffling on the shell side of an exchanger is usually most beneficial in convection transfer and must be considered from both the heat transfer and pressure drop viewpoints. Close baffle spacing increases heat transfer and pressure drop for a given throughput. The average segmental baffle will have an open "window" for fluid passage of 25% of the shell diameter, or 75% of the shell diameter will have a baffle covering it from flow.

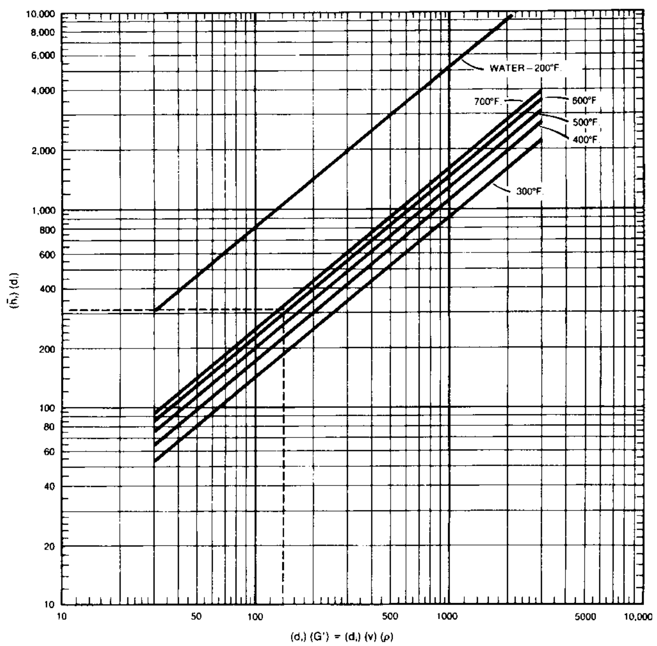


Figure 10-50C. Tube-side (inside tubes) liquid film heat transfer coefficient for Dowtherm®. A fluid inside pipes/tubes, turbulent flow only. Note: $\bar{h}$ = average film coefficient, Btu/hr-ft^2-°F; d_i = inside tube diameter, in.; G′ = mass velocity, lb/sec/ft^2; v = fluid velocity, ft/sec; k = thermal conductivity, Btu/hr (ft^2)(°F/ft); μ = viscosity, lb/(hr)(ft); C_p = specific heat, Btu/(lb)(°F). (Used by permission: *Engineering Manual for Dowtherm Heat Transfer Fluids*, ©1991. The Dow Chemical Co.)

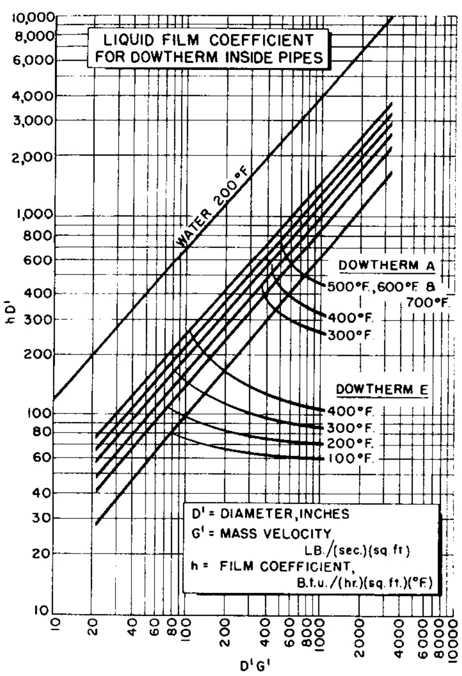


Figure 10-50D. Tube-side (inside pipes or tubes) liquid film heat transfer coefficient for Dowtherm® A and E at various temperatures. (Used by permission: *Engineering Manual for Heat Transfer Fluids*, ©1991. The Dow Chemical Co.)

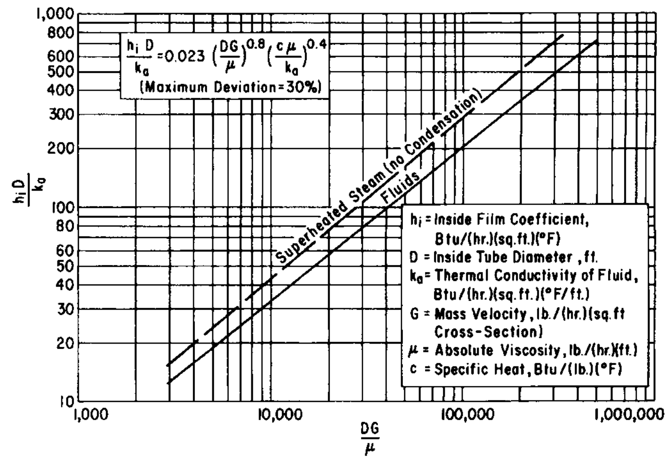


Figure 10-51. Convection inside film coefficient for gases and low viscosity fluids inside tubes—heating and cooling. (Used by permission: McAdams, W. H. *Heat Transmission*, 2nd Ed., ©1942. McGraw-Hill, Inc. All rights reserved.)

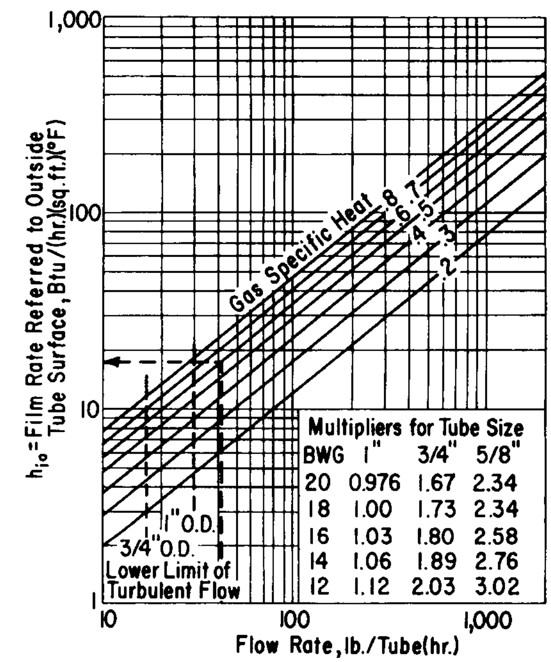


Figure 10-52. Heat transfer to gases inside tubes. (Used by permission: Buthod, A. P. *Oil & Gas Journal*, V. 58, No. 3, ©1960. PennWell Publishing Company. All rights reserved.)

The smallest baffle window is 15–20% of the diameter of the shell, and the largest is close to 51%. Some design relations in other references use this as a percentage of the shell cross-section area, and the corresponding relations must be used. In exchanger design, this cutout is varied to help obtain good operating performance; however, the spacing between baffles (baffle pitch) is much more significant in its effect on the film coefficient for a given baffle

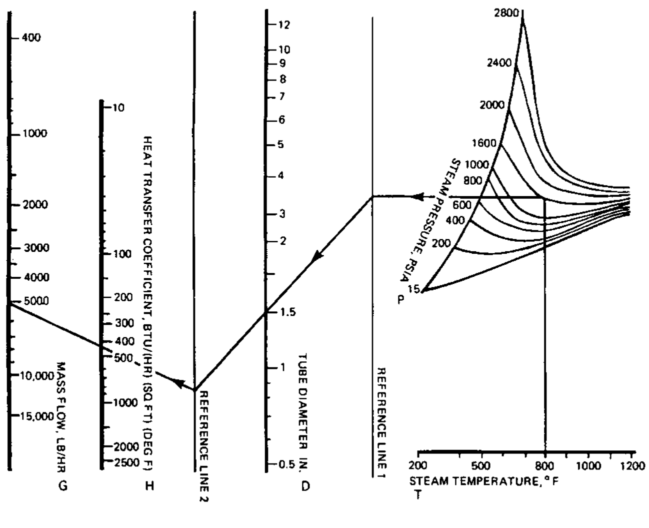
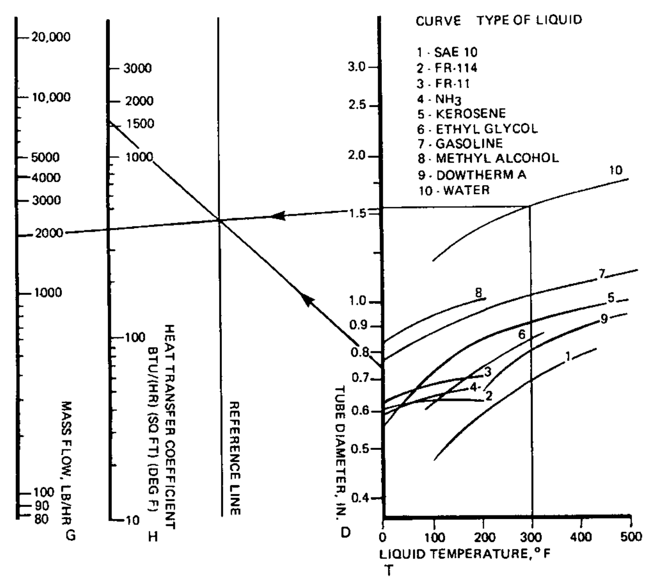


Figure 10-53A. Determine the inside heat transfer coefficient for superheated steam. (Used by permission: Ganapathy, V. *Hydrocarbon Processing*, Sept. 1977. ©Gulf Publishing Company, Houston, Texas. All rights reserved.)



©Figure 10-53C. Determine the inside heat transfer coefficient of common liquids. (Used by permission: Ganapathy, V. *Hydrocarbon Processing*, Sept. 1977. ©Gulf Publishing Company, Houston, Texas. All rights reserved.)

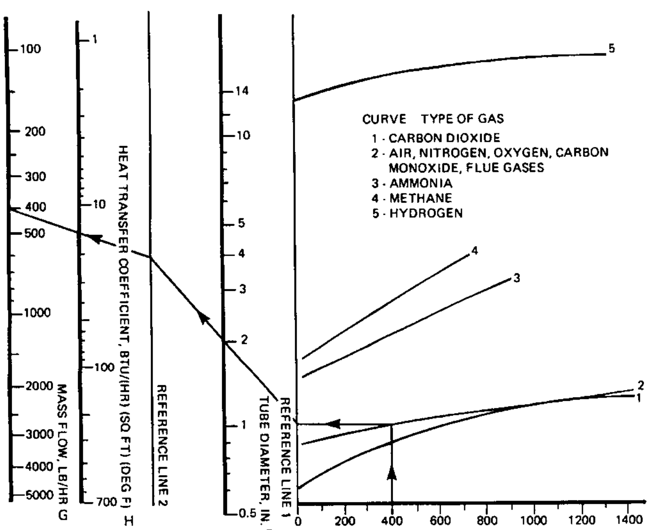


Figure 10-53B. Determine the inside heat transfer coefficient of common gases (Used by permission: Ganapathy, V. *Hydrocarbon Processing*, Sept. 1977. ©Gulf Publishing Company, Houston, Texas. All rights reserved.)

cut. If twice the number of baffles is used for a fixed fluid flow, the velocity across the tube bundle is doubled, and the increase in film coefficient is about 44%. However, the pressure drop will approach four times its value before doubling the number of baffles (see Figure 10-57).

Figure 10-58 illustrates a low pressure drop baffle arrangement. Each situation must be examined, as no generalities will solve all detailed designs. Baffles should be held to a minimum spacing of $^1/_5$ the shell diameter or 2 in., whichever is larger. Baffles spaced equally to shell diameter are found to give good average performance, and this guide is often used in estimating the initial spacing for baffles. Where possible the baffle spacing and percent baffle cut should provide equal flow area. This is of particular importance in pressure drop calculations. Figure 10-59 is useful for this equalization.

Shell-side film coefficients can be conveniently obtained from the charts of Chen,[25] Figures 10-60, 10-61, and 10-62. These are based on Donohue's[38] equation

$$\frac{h_o D_o}{K_a} = 0.22\left(\frac{D_o G_w}{\mu}\right)^{0.6}\left(\frac{c_p \mu}{k_a}\right)^{0.333}\left(\frac{\mu}{\mu_w}\right)^{0.14} \qquad (10\text{-}61)$$

Equivalent tube diameter for shell-side heat transfer calculations is used by permission from Kern and Kraus.[206]

The volumetric equivalent diameter, d_e in., is again calculated on the basis of 4× the hydraulic radius; see Figure 10-56.

$$d_e = \frac{4 \times \text{free area}}{\text{wetted perimeter}}, \text{ in.} \qquad (10\text{-}62)$$

(a) Equivalent Diameter, D_e, for Annulus

$$D_e = \frac{D_2^2 - D_1^2}{D_1} = 4r_h = \frac{4\,(\text{flow area})}{(\text{wetted perimeter})} \qquad (10\text{-}62A)$$

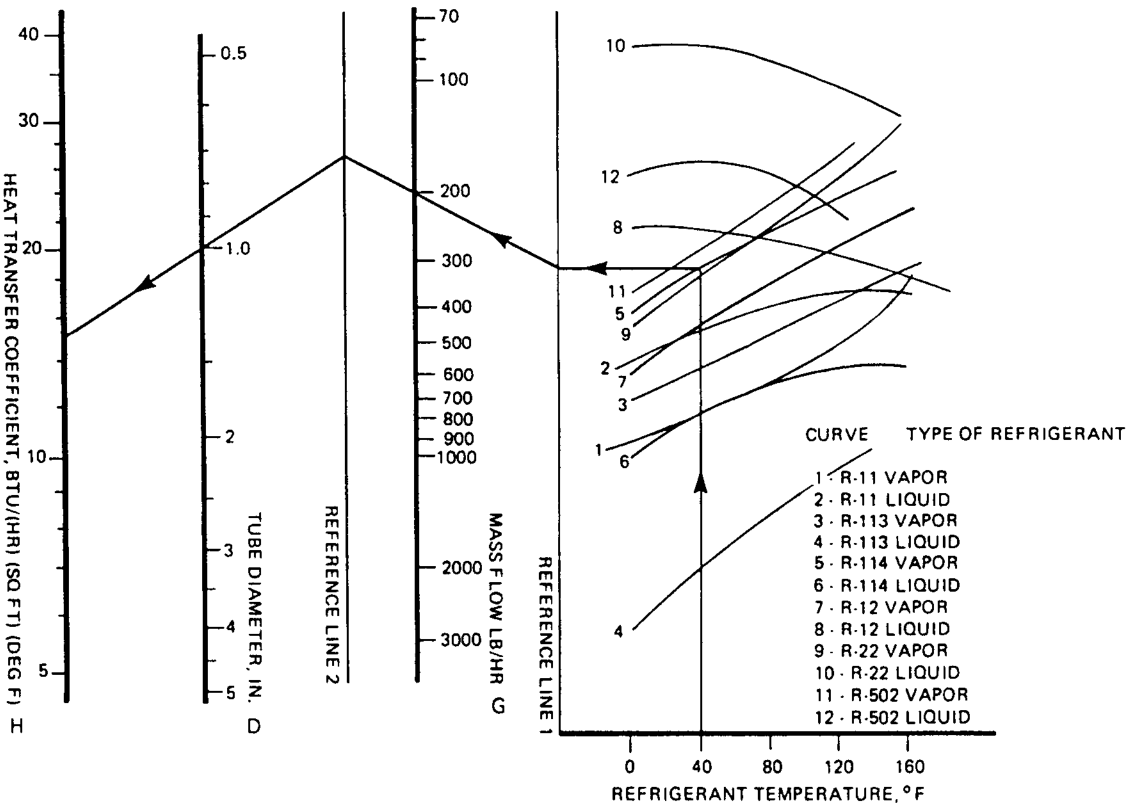


Figure 10-53D. Determine the inside heat transfer coefficient of several common vapor/liquid refrigerants. (Used by permission: Ganapathy, V. *Hydrocarbon Processing,* Sept. 1977. ©Gulf Publishing Company, Houston, Texas. All rights reserved.)

where

D_1 = outside diameter of inner tube, ft
D_2 = inside diameter of outer pipe, ft
r_h = hydraulic radius, ft = (radius of a pipe equivalent to the annulus cross-section)

(b) Square Pitch and Rotated Square Pitch

$$d_e = \frac{4p^2 - \pi(d_e')^2/4}{\pi d_e'} \tag{10-63}$$

Where p is the tube pitch, in.

(c) Triangular Pitch

$$d_e = \frac{4[0.5p(0.86p) - 0.5\pi(d_e')^2/4]}{0.5\pi d_e'} \tag{10-64}$$

For plain tubing, the nominal O.D. replaces d_e'. The volumetric equivalent diameter does not distinguish between square pitch and square pitch rotated by 45°.

where

d_e' = equivalent diameter of plain tube (used to correlate heat transfer and pressure drop) corresponding to the metal volume of a finned tube, in. It is the volumetric equivalent diameter of the root tube plus the addition to the root-tube O.D. if the volume of the fin metal were added to it to form a new root-tube O.D.[206] See Figure 10-10H.
d_e = equivalent diameter, in.

For use in the equivalent diameter equations, the following volumetric d_e', in., values are taken from reference 206.

Plain Tube Section Tube O.D. in.	19fins/in. × $^1/_{16}$ in. High Equivalent Diameter d_e', in.	16fins/in. × $^1/_{16}$ in. High Equivalent Diameter d_e', in.
0.625	0.535	0.540
0.750	0.660	0.665
0.875	0.785	0.790
1.000	0.910	0.917

Used by permission: Based on data from Kern and Kraus, pp. 512–513, ©1972. McGraw-Hill, Inc.

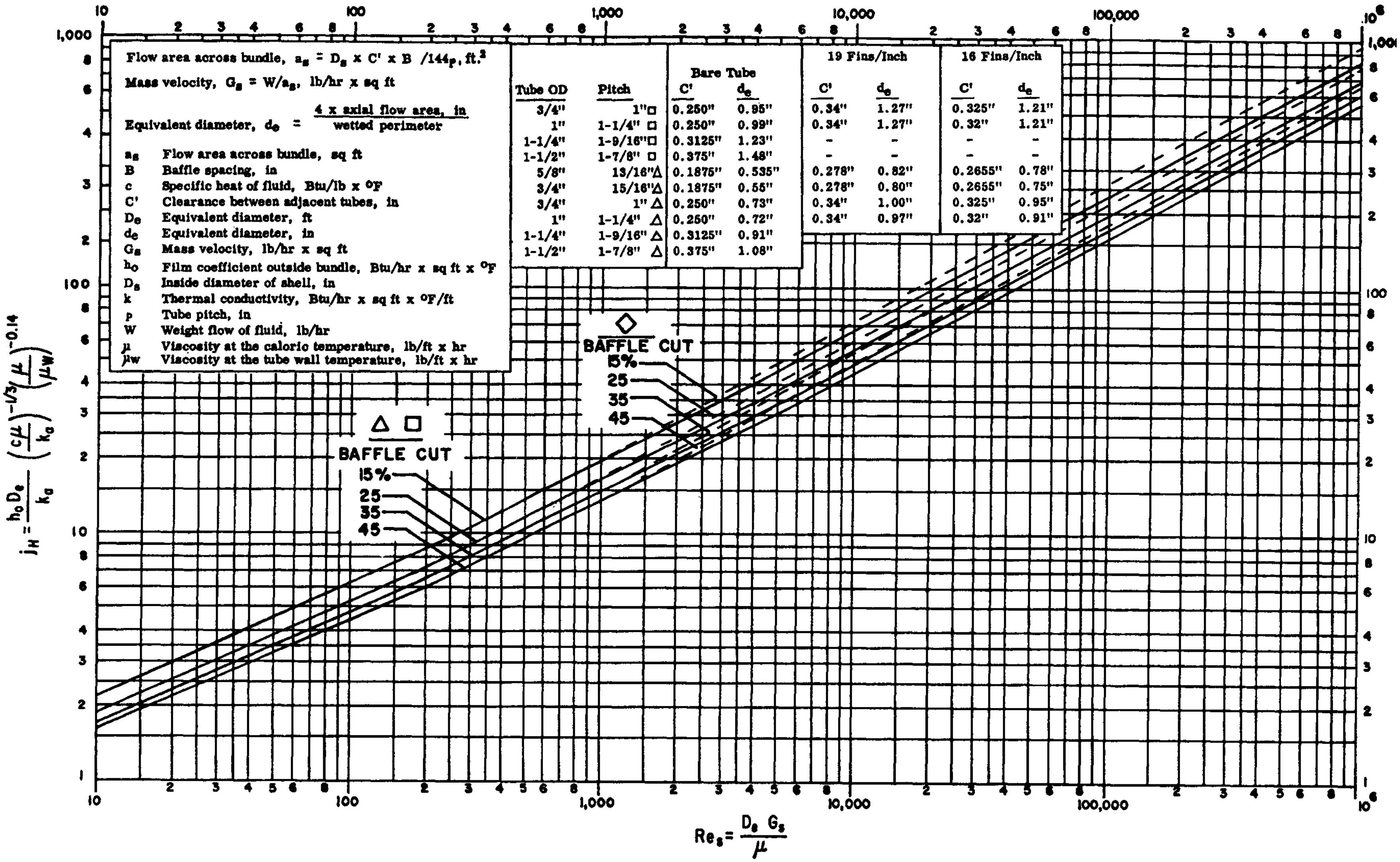


Figure 10-54. Shell-side heat transfer curve for segmental baffles. (Used by permission: *Engineering Data Book Section II*, ©1959. Wolverine Tube, Inc.)

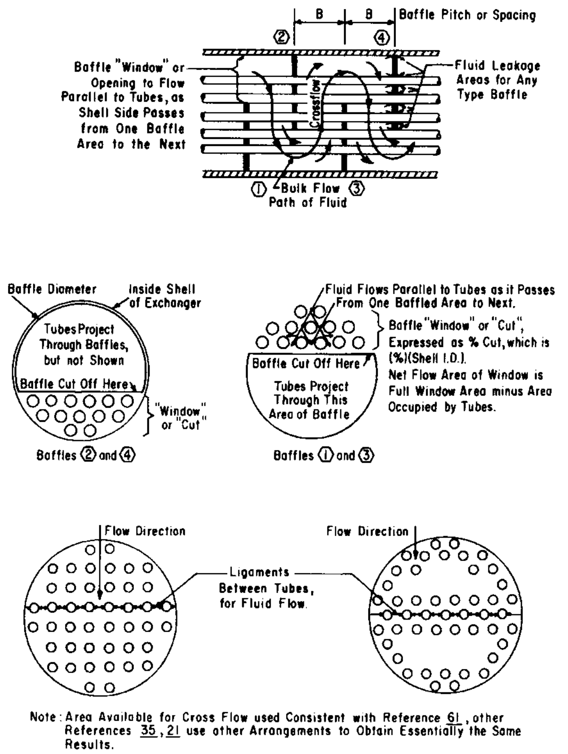


Figure 10-55. Shell-side baffles and cross-flow area.

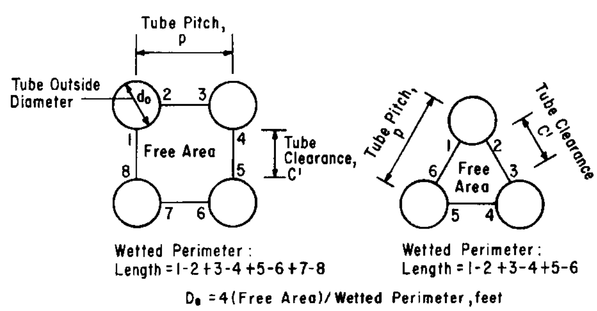


Figure 10-56. Equivalent diameter for tubes on shell side of exchanger taken along the tube axis. (a) Square pitch, (b) triangular pitch on 60° equilateral angles. (Used by permission: Kern, D. Q. *Process Heat Transfer,* 1st Ed., ©1959. McGraw-Hill, Inc. All rights reserved.)

The charts are used as follows:

1. Determine geometric mean mass velocity, G_e', using Figure 10-60.
   (a) Cross-flow area for this method[38] equals the horizontal shell diameter minus the space occupied by the tubes along this diameter, multiplied by the baffle spacing. Determine G_c', lb/sec (ft^2) by dividing the shell-side flow rate by the cross-flow area.

Table 10-21
Shell-Side Equivalent Tube Diameters for Various Tube Arrangements

Tube O.D. In.	Pitch	Equivalent Diameter, d_e, In.
$^1/_2$	$^5/_8$ triangular	0.36
$^1/_2$	$^3/_4$ triangular	0.74
$^3/_4$	$^{15}/_{16}$ triangular	0.55
$^3/_4$	1 triangular	0.73
1	1 $^1/_4$ triangular	0.72
1 $^1/_4$	1 $^9/_{16}$ triangular	0.91
$^1/_2$	$^5/_8$ square	0.48
$^1/_2$	$^3/_4$ square	0.88
$^3/_4$	$^{15}/_{16}$ square	0.72
$^3/_4$	1 square	0.95
1	1 $^1/_4$ square	0.99
1 $^1/_4$	1 $^9/_{16}$ square	1.23

Used by permission: *Engineering Data Book Section,* ©1960 and 1984. Wolverine Tube Inc.; and Kern, D.Q. *Process Heat Transfer,* ©1950. McGraw-Hill Inc. All rights reserved.

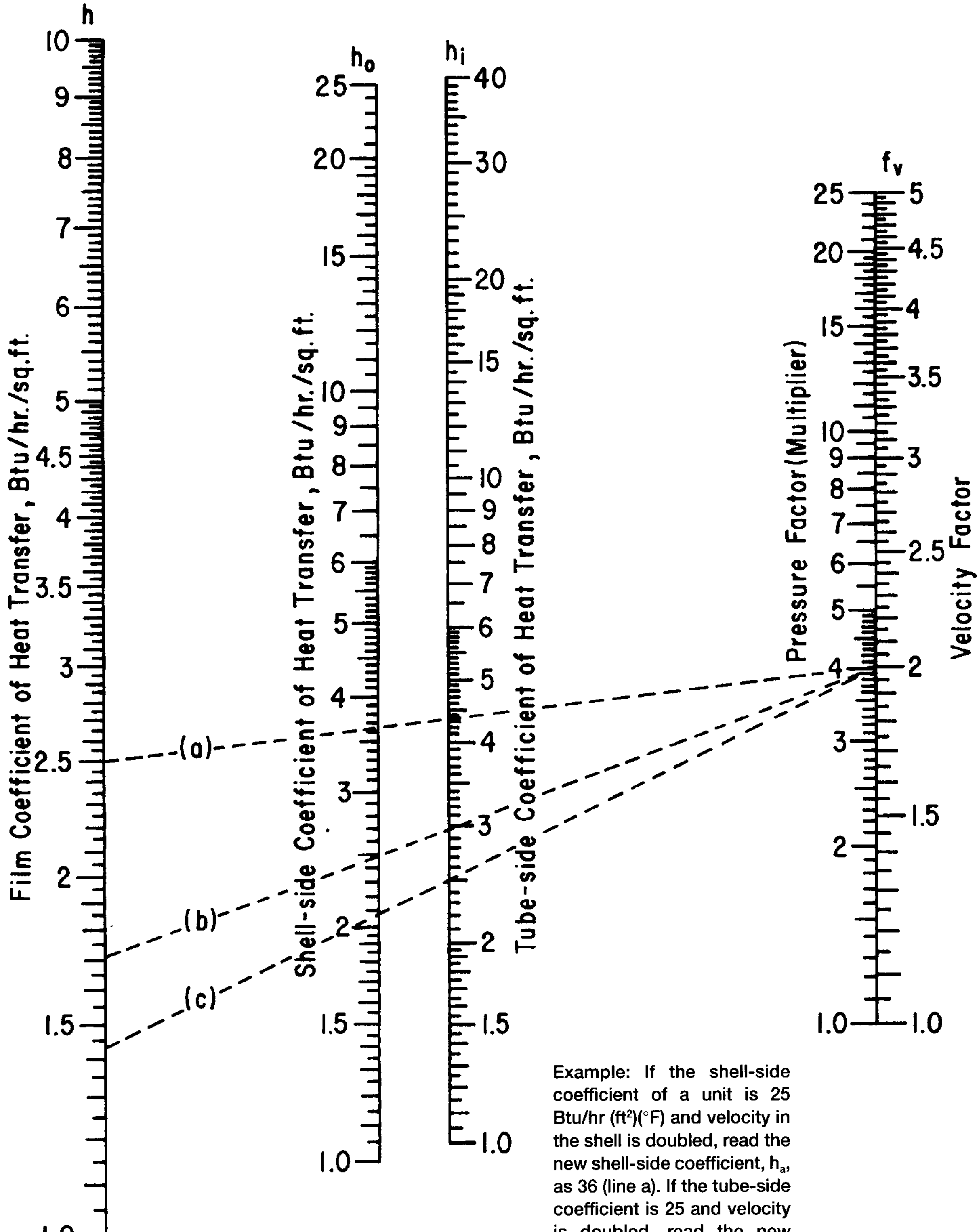
   (b) The baffle window cut-out area minus the area occupied by the tubes passing through this area is the net baffle opening flow area. Determine G_b', as lb/sec (ft^2) by dividing the flow rate of the shell side by this new baffle opening flow area.
   (c) Read G_e', lb/sec (ft^2), from Figure 10-60 at the intersection of G_c' and G_b'.
2. Determine the physical property factor, ϕ_p', using Figure 10-61.
3. Read the outside film coefficient, h_o, using Figure 10-62.
   Note: This has the viscosity correction $(\mu/\mu_w)^{0.14}$, included. A correction multiplier must be used to correct the results of Figure 10-62 for tubes different than $^5/_8$-in. O.D.

The charts of Rubin[98] are somewhat similar and also useful for solving the equation by graph rather than by calculator.

Shell-Side Velocities

Figure 10-63 suggests reasonable maximum velocities for gases and vapors through heat exchangers. If entrained liquid or solids are present, the velocities should be reduced. Pressure drop must be checked to determine the acceptability of any selected velocity. Table 10-22 presents suggested maximum velocities for fluids flowing through exchanger nozzles. The effect of entrance and exit losses on pressure losses should be checked, as they become important in low pressure systems.

Figure 10-64 is convenient in selecting pipe or nozzle sizes.



Example: If the shell-side coefficient of a unit is 25 Btu/hr (ft^2)(°F) and velocity in the shell is doubled, read the new shell-side coefficient, h_a, as 36 (line a). If the tube-side coefficient is 25 and velocity is doubled, read the new tube coefficient, h_i, as 43.1 (line a). In other cases, pressure drop would increase by a factor of 4. Note: This may be used in reverse for reduced flow.

Figure 10-57. Effect of velocity on heat transfer rates and pressure drop: shell-side and tube-side. (Used by permission: Shroff, P. D. *Chemical Processing,* No.4, ©1960. Putnam Publishing Co., Itasca, Ill. All rights reserved.)

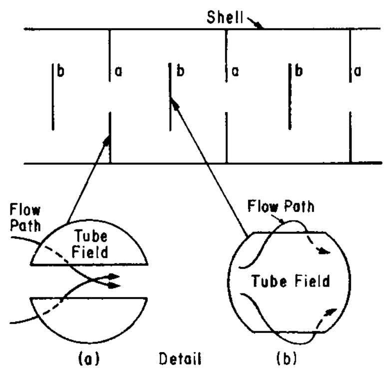


Figure 10-58. Baffling for low pressure drop shell-side designs.

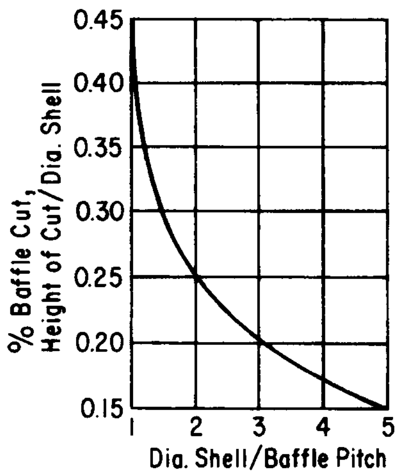


Figure 10-59. Determination of equal flow areas in bundle cross-flow and baffle window shell-side performance. (Used by permission: *Engineering Data Book Section II,* ©1959. Wolverine Tube, Inc.)

Table 10-22
Maximum Recommended Velocities through Nozzle Connections, Piping, Etc. Associated with Shell and/or Tube Sides of Heat Exchanger

Viscosity in Centipoise	Maximum Velocity, Ft/Sec	Remarks
Liquids:		
More than 1500	2	Very heavy oils
1000–500	2.5	Heavy oils
500–100	2.5	Medium oils
100–35	5	Light oils
35–1	6	Light oils
Less than 1	8	. . .
Vapors and Gases:		
Use 1.2 to 1.4 of the value shown on Figure 10-63 for velocity through exchangers.		

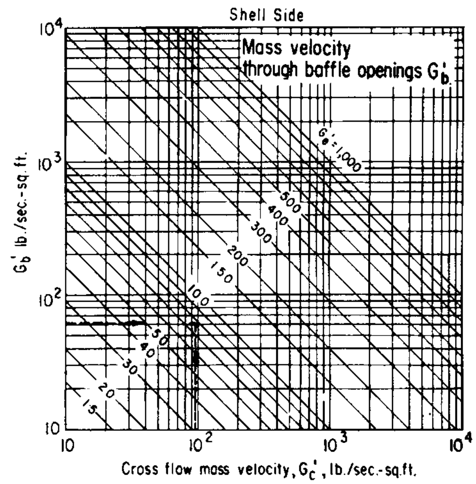


Figure 10-60. Shell-side mass velocity through baffle opening G_b'. (Used with permission: Ning Hsing Chen, *Chemical Engineering,* V. 65, ©1958. McGraw-Hill, Inc. All rights reserved.)

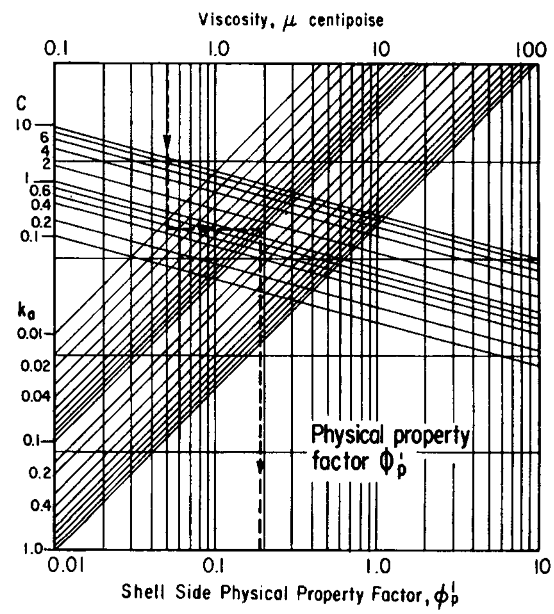


Figure 10-61. Shell-side physical property factor for ϕ_p'. (Used with permission: Ning Hsing Chen, *Chemical Engineering,* V 65, Oct. 1958. ©McGraw-Hill, Inc. All rights reserved.)

Design Procedure for Forced Convection Heat Transfer in Exchanger Design

1. Establish physical properties of fluids at the caloric or arithmetic mean temperature, depending upon the temperature range and order of magnitude of the properties.
2. Establish the heat duty of the exchanger.

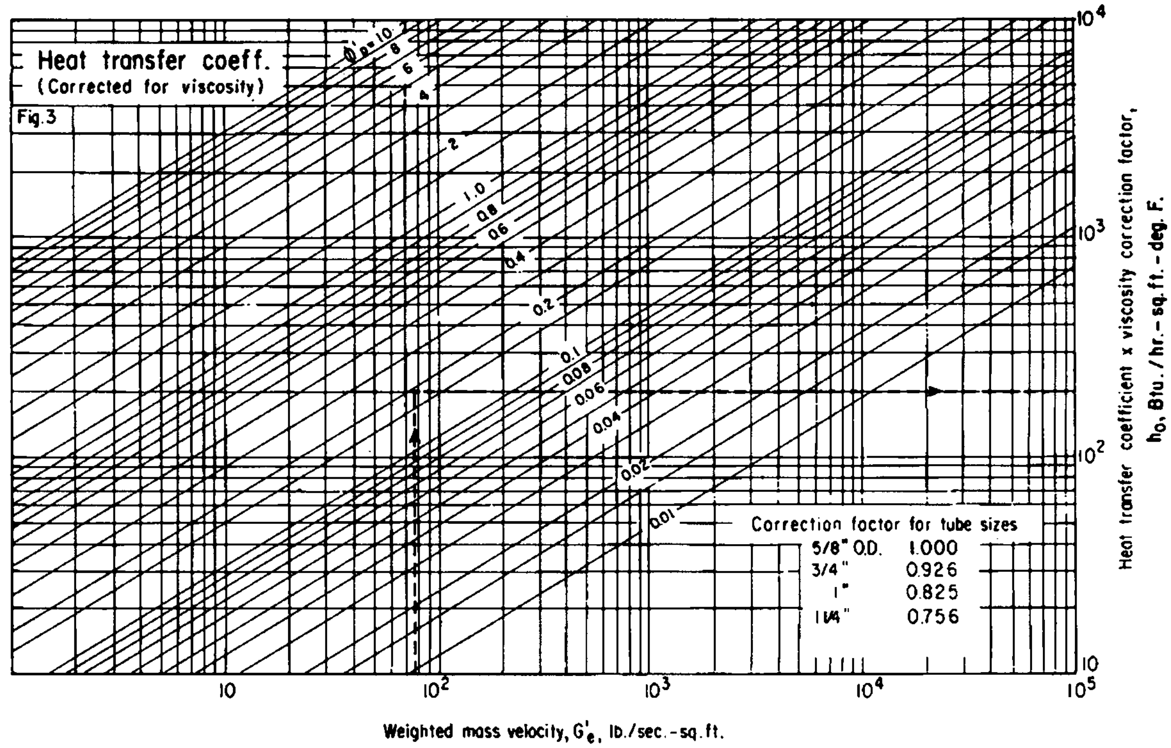


Figure 10-62. Shell-side film coefficient. (Used with permission: Ning Hsing Chen, *Chemical Engineering,* V. 65, Oct. 1958. ©McGraw-Hill, Inc. All rights reserved.)

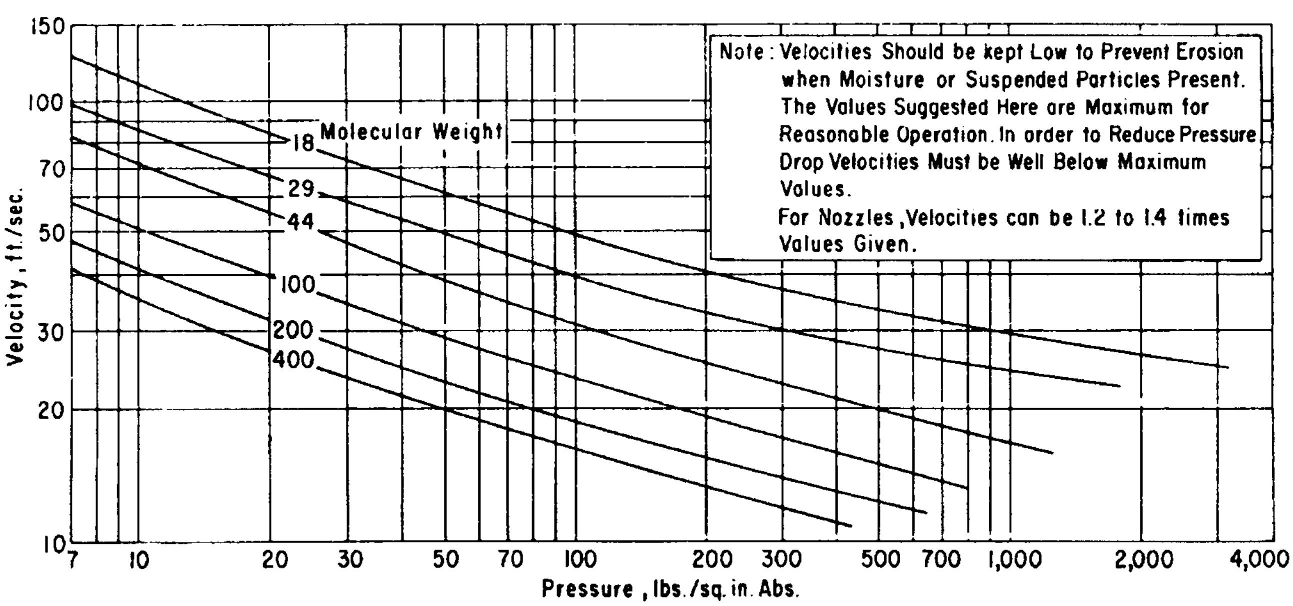


Figure 10-63. Maximum velocity for gases and vapors through heat exchangers on shell side.

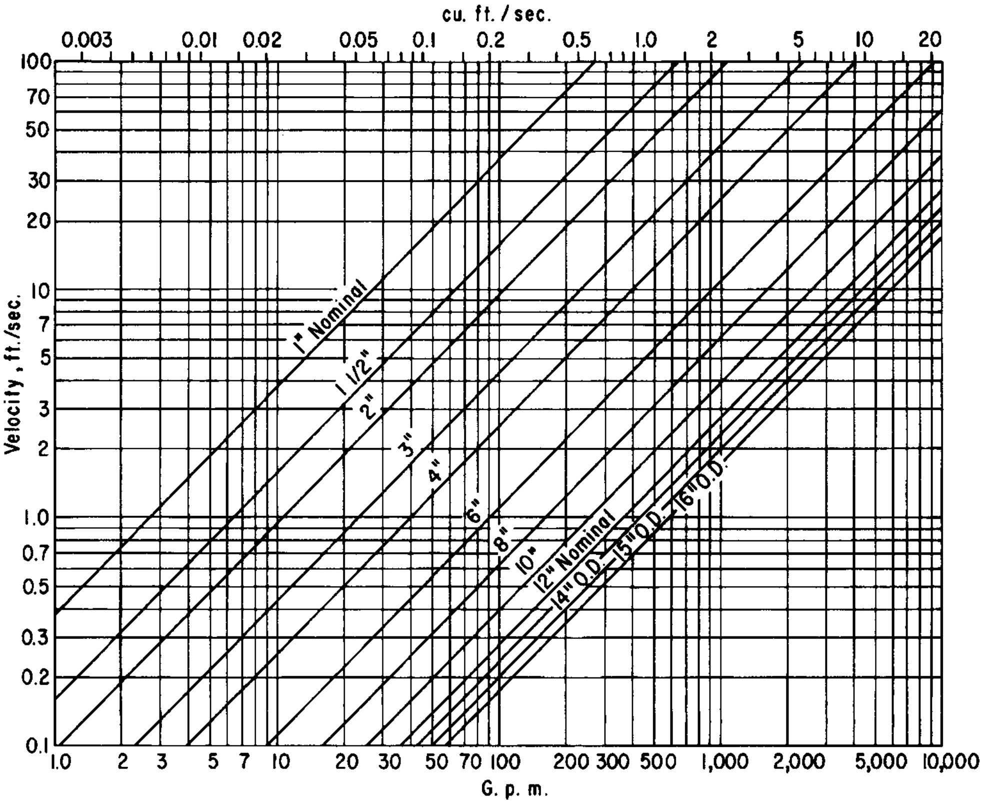


Figure 10-64. Nozzle sizes for fluid flow. (Used by permission: ITT Technologies, ITT Standard. All rights reserved.)

3. Estimate or assume a specific unit and define its size and characteristics, based upon reasonable values of overall U and LMTD.
4. Determine the LMTD, with correction if needed from Figures 10-33 and 10-34.
5. Calculate the tube-side flow rate based upon the assumed number of tubes per pass and the heat balance.
6. Determine the tube-side film coefficient for water, using Figure 10-50A or 10-50B. For other liquids and gases, use Figure 10-46. Correct h_i to the outside tube surface by

$$h_{io} - h_i\left(\frac{\text{I.D.}}{\text{O.D.}}\right) \tag{10-65}$$

7. Determine the shell-side film coefficient for an assumed baffle spacing.

(a) Establish G_s from Equation 10-60.
(b) Calculate the Reynold's number, R_e, expressed as

$$R_e = \frac{D_e G_s}{\mu} \tag{10-66}$$

(c) Read j_H from Figure 10-54. Note that 25% is a good average value for many designs using segmental baffles.
(d) Calculate h_o from

$$j_H = \frac{h_o D_e}{k}\left(\frac{c\mu}{k}\right)^{-1/3}\left(\frac{\mu}{\mu_w}\right)^{-0.14} \tag{10-67}$$

Let $\mu/\mu_w = 1.0$

(e) If h_o appears too low, assume closer baffle spacing, up to $^1/_5$ of the shell diameter and recalculate G_s and h_o. If this second trial is obviously too low, then a larger shell size may be indicated; therefore, return to step 3, re-evaluating the assumed U to be certain that it is attainable.

8. If the h_o appears to have possibilities of satisfying the design, continue to a conclusion by assuming the tube-side and shell-side fouling (Tables 10-12 and 10-13; Figures 10-39, 10-40A, 10-41, 10-42, and 10-43).
9. Calculate the overall coefficient using Equation 10-37. Neglect the tube-wall resistance, unless special situations indicate that it should be included.

10. Calculate the area required using Equation 10-9.
11. Calculate the net available area in the assumed unit, using only the effective tube length.
12. Compare values calculated in steps 10 and 11. If the calculated unit is too small, re-assume a new larger unit for step 3 or try closer baffle spacing in step 7 but do not get baffles closer than $^1/_5$ the shell I.D.
13. Calculate the percent of excess area. A reasonable figure is 10–20%.
14. Calculate the shell-side pressure drop. (Refer to the later section on "Pressure Drop Relations" and Figure 10-140. If ΔP is too high, reassume unit (step 3).
15. Calculate the tube-side pressure drop. (Use Figure 10-139 for the end return losses. For water in tubes, use Figure 10-138 for tube losses. For other liquids and gases in tubes, use Figure 10-137.
    Total pressure drop = (end return + tube) losses, psi.

If the tube-side pressure drop exceeds a critical allowable value for the process system, then recheck by either lowering the flow rate and changing the temperature levels or reassume a unit with fewer passes on tube side or more tubes per pass. The unit must then be rechecked for the effect of changes on heat transfer performance.

Example 10-9. Convection Heat Transfer Exchanger Design

See Figure 10-65.

The liquid bottoms from a distillation column must be cooled from 176°F to 105°F. The cooling water is untreated at 90°F.

Operating data: Bottom flow, 6,350 lb/hr
Average Cp, 0.333 Btu/lb (°F)
Average k_a, 0.055 Btu/hr (ft^2) (°F/ft)
Average μ, 0.404 centipoise
Average sp.gr, 0.78
Physical properties are based on values at 140°F average temperature.
Caloric fluid temperature for property evaluation can be calculated from Equation 10-21.

The caloric value of hot liquid on the shell side is

$$t_h = t_{h2} + F(t_{h1} - t_{h2})$$

$$\text{Rough estimate } U_c \text{ at cold end} = \frac{150{,}000}{(38.2)(15^\circ)} = 262$$

$$U_h \text{ at hot end} = \frac{150{,}000}{38.2(81^\circ)} = 48.5$$

$$C = \frac{U_h - U_c}{U_c} = \frac{48.5 - 262}{262} = -0.815$$

(Note: disregard sign)

$\Delta t_c = 15^\circ$

$\Delta t_h = 81^\circ$

$$\frac{\Delta t_c}{\Delta t_h} = \frac{15}{81} - 0.815$$

Reading Figure 10-38,

$F = 0.32$

Then: $t_h = 105 + 0.32(176 - 105) = 127.7°F$

Note that the arithmetic average [$^1/_2$ (176 − 105) + 105 = 140°F] would be quite satisfactory for this design, because the properties do not vary significantly with temperature.

1. Heat duty = (6350) (176 − 105) (0.33)
   = 150,000 Btu/hr
2. Estimated unit

Assume: U = 100
LMTD = 39.2

$$A = \frac{150{,}000}{(100)(39.2)} = 38.2 \text{ ft}^2$$

Tubes: 1-in. O.D. × 14 BWG × 8 ft long

$$\text{No. required} = \frac{38.2}{(0.2618 \text{ ft}^2/\text{ft})(8 - 6 \text{ in.}/12)}$$
$$= 20 \text{ tubes}$$

Trial:

10-inch I.D. shell with 24 1-in. tubes on 1 $^1/_4$-in. triangular pitch, 4 tube passes.

3. Log mean temperature difference (Figure 10-33),

176° cooling → 105°
95° ← warming 90°
81° 15°

4. Water rate,

$$w = \frac{150{,}000}{(1)(95 - 90)} = 30{,}000 \text{ lb/hr}$$

$$\text{gpm} = \frac{30{,}000}{(8.33)(60)} = 60$$

At $^{24}/_4$ = 6 tubes/pass,

$$\text{Cross-sectional area/tube} = \frac{0.546}{144} = 0.00379 \text{ ft}^2/\text{tube}$$

Flow area/pass = (0.00379) (6 tubes) = 0.0227 ft^2 flow area

DWG. NO. A

Item No.

By

EXCHANGER RATING

Job No.

Date:- 4-1-59

Charge No.

Apparatus T-58 Bottoms Cooler Plant

Min. Req. Eff. O. S. Area Exposed in Shell, Sq. Ft. Outside 89 Inside: –

Number of Units: Operating One Spares None

DESIGN DATA PER Unit

UNIT DATA		SHELL SIDE	TUBE SIDE
Fluid		Col. T-58 Bottoms	Untreated water
Fluid Flow	Lbs./Hr.	6350	50,000 (100 gpm.)
Temperature In	°F.	176	90
Temperature Out	°F.	105	93
Operating Pressure	PSIG	15	50
Density	Lbs./CF	48.7	62.3
Specific Heat	Btu/Lb./°F.	0.333	
Latent Heat	Btu/Lb.	—	
Therm. Cond.	Btu/Hr./Sq. Ft./°F./Ft.	0.055	
Viscosity	Centipoise	0.404	
Molecular Weight			
No. of Passes		1	4
Pressure Drop	PSI	Calc: 0.0835 Used: 1.25	Calc: 5.24 Used: 6
Fouling Factor		0.002	0.001

Heat Transferred - BTU/Hr: 150,000

LMTD 39.6 °F. Overall U: Calc. 47.2 Used 42.5

CONSTRUCTION

	Shell	Tube
Max. Oper. Pressure	30 PSI	50 PSI
Max. Oper. Temperature	200 °F.	150 °F.

Type of Unit Fixed Tube Sheet Tube Pitch 1¼" Triangular Joint Roll & Flare

Tubes - Material: Admiralty No. (Approx.) 44 O.D. 1" BWG 14 Length 8'-0"

Shell - Material: Steel Diameter (Approx.): 12" I.D.

Channel Material: Steel Supports Material: Admiralty

Tube Sheet Material: Admiralty Baffle Material: Admiralty

Corrosion Allowance - Shell Side 1/16" Tube Side 1/32"

Connections - Shell In: 2" Out 2" Flange 150# Raised Face

Channel In: 3" Out 3" Flange 150# Raised Face

Others Vent and Drain Size 3/4" Flange 3000# Coupling

Bolts: Alloy Steel Gaskets: Asbestos

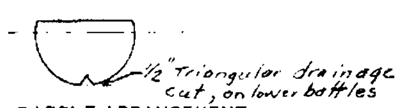
Code TEMA - C Stamp Yes X-Ray – SR –

Insulation Personnel Protection Class PP Cathodic Protection No

Baffles: 25% Cut Segmental Horizontal Cut

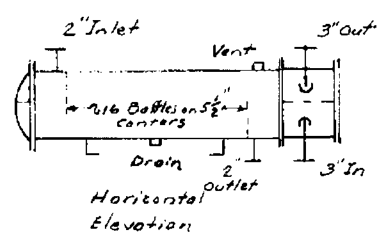
16 Total on 5½" Centers

First and last baffle to be close to nozzles. Nozzles to be as close as practical to tube sheets



BAFFLE ARRANGEMENT

Tube sheet layout based on quadrants



NOZZLE ARRANGEMENT

Remarks:-

Checked Date Approved Date

Rev. By Date B/M No.

Figure 10-65. Exchanger rating example.

$$\text{Water velocity} = \frac{60}{(60)(7.48)(0.0227)} = 5.88 \text{ ft/sec}$$

5. Film coefficient, tube side,
From Figure 10-50A and 10-50B, at 5.88 fps and 93°F, read:

h_i = 1,340 Btu/hr (ft^2) (°F)
Correction for tube I.D. of 0.834 in., F_w = 0.94
Correction to outside of tube:
h_{io} = (1,340) (0.94) (0.834)/1.0 = 1,050 Btu/hr (ft^2) (°F)

6. Film coefficient, shell side,
From Figure 10-54, read:

$$\text{Cross-flow area} = a_s = \frac{\text{I.D.}(c'B)}{p(144)}$$

Assume baffle spacing of 10 in. = B
I.D. = shell I.D. = 10 in.
c' = 1.25 − 1.0 = 0.25 in.
W = 6,350 lb/hr
p = 1.25

$$a_s = \frac{(10)(0.25)(10)}{(1.25)(144)} = 0.139 \text{ ft}^2$$

$$G_s = \frac{W}{a_s} = \frac{6{,}350}{0.139} = 45{,}700 \text{ lb/hr (ft}^2)$$

$$\text{Reynold's number: Re} = \frac{D_e G_s}{\mu}$$

D_e = (0.72/12) = 0.06 ft (Table 10-21)
μ = (0.404) (2.42) = 0.978 lb/ft (hr)

$$R_e = \frac{(0.06)(45{,}700)}{0.978} = 2{,}800$$

Reading Figure 10-54,

j_H = 28 (for 25% cut segmental baffles)

From Figure 10-140,

f = 0.0027 (for 25% cut segmental baffles)
Note: "f" from figure 10-140. is divided by 1.2 for plain tubes (not finned).

$$j_H = \frac{h_o D_e}{k_a}\left(\frac{c\mu}{k_a}\right)^{-1/3}\left(\frac{\mu}{\mu_w}\right)^{-0.14} = 28$$

$$h_o = 28\left(\frac{0.055}{0.06}\right)\left(\frac{(0.333)(0.404)(2.42)}{0.055}\right)^{1/3}(1)$$

Note that (μ/μ_w) is taken as 1.0 for fluids of low viscosity where the change in temperature does not introduce a significant increase in viscosity.

h_o = 46.4 Btu/hr (ft^2) (°F)

Try to obtain a better coefficient by closer baffling, check extreme of 2-in. baffle spacing.

7. Shell-side film coefficient based on 2-in. baffle spacing,

$$G_s = \frac{(6{,}350)(144)(1.25)}{(10)(0.25)(2)} = 229{,}000 \text{ lb/hr (ft}^2)$$

$$R_e = \frac{(0.06)(229{,}000)}{0.978} = 14{,}080$$

From Figure 10-54,

j_H = 66

$$h_o = \frac{(66)(0.055)(1.81)}{0.06} = 109.5 \text{ Btu/hr (ft}^2)(°\text{F})$$

8. Assume fouling,

Shell side = 0.002
Tube side = 0.001

9. Overall coefficient,

$$U = \frac{1}{\frac{1}{1{,}050} + 0.001 + 0.002 + \frac{1}{109.5}}$$
$$= 76.2 \text{ Btu/hr (ft}^2)(°\text{F})$$

10. Area required,

$$A = \frac{150{,}000}{(76.5)(39.2)} = 50 \text{ ft}^2$$

11. Area available in assumed unit,

A = (0.2618) (24) (8 − 6 in./12) = 47 ft^2

Therefore, the assumed unit is too small.

12. Second trial,
Assume:
12-in. I.D. shell, 44 tubes, 1-in. O.D. × 14 BWG × 8 ft long on 1 $^1/_4$-in. triangular pitch, 4 tube passes. For the revised water rate, allow only 3°F rise.

$$\text{lb hr} = \frac{150{,}000}{(1)(3°)} = 50{,}000$$

$$\text{gpm} = \frac{50{,}000}{(8.33)(60)} = 100$$

Number of tubes per pass = 11
Water flow area = (11)(0.00379) = 0.0417 ft^2

$$\text{Water velocity} = \frac{100}{(60)(7.48)(0.0417)} = 5.34 \text{ ft/sec}$$

Film coefficient, tube side,
From Figure 10-50A and 10-50B, read:
$h_i = 1{,}220$
$h_{io} = (1{,}220)(0.94)(0.834) = 956$ Btu/hr (ft^2)(°F)
Film coefficient, shell side,
Select baffle spacing of 5.5 in., equal to 16 baffles.

$$G_s = \frac{(6{,}350)(144)(1.25)}{(12)(0.25)(5.5)} = 69{,}300 \text{ lb/hr (ft}^2)$$

$$R_e = \frac{(0.06)(69{,}300)}{0.978} = 4{,}250$$

Reading Figure 10-54, $j_H = 35$
Reading Figure 10-140,
f = 0.0025 (for plain tubes)

$$h_o = \frac{(35)(0.055)(1.81)}{0.06} = 58.1$$

Use same fouling factors as for first trial.
Overall coefficient,

$$U = \frac{1}{\dfrac{1}{58.1} + 0.002 + 0.001 + \dfrac{1}{956}}$$
$$= 47.2 \text{ Btu/hr (ft}^2)(°\text{F})$$

LMTD,

$$\begin{array}{ccc} & \text{Shell} & \\ 176 & \longrightarrow & 105 \\ & \text{Tubes} & \\ \underline{93} & \longleftarrow & \underline{90} \\ 83 & & 15 \end{array}$$

LMTD = 39.8°F

Correction to LMTD read Figure 10-34,

$$P = \frac{93 - 90}{176 - 90} = 0.0349$$

$$R = \frac{176 - 105}{93 - 90} = 23.6$$

F = 0.99
Corrected LMTD = (0.99)(39.8) = 39.6°F
Area required,

$$A = \frac{150{,}000}{(47.2)(39.6)} = 80.2 \text{ ft}^2$$

Area available in assumed unit, second trial,
A = (0.2618)(44)(8 − 3 in./12) = 89.2 ft^2
(For low-pressure design, 3 in. is sufficient allowance for two tubesheets.)
Percent excess area,

$$\% = \frac{89.2 - 80.2}{80.2}(100) = 10.2\%$$

This is satisfactory.
Pressure drop, shell side (see "Pressure Drop" section),

$$\Delta P_s = \frac{f(G)^2(D_s)(N_c + 1)}{5.22\,(10)^{10}(D_e)(s)(\phi_s)} \qquad (10\text{-}67A)$$

where

$f = 0.0025$
$G = 69{,}300$
$D_s = 12/12 = 1$ ft
$N_c + 1 = 16 + 1 = 17$
$D_e = 0.72/12 = 0.06$
$s = 0.78$
$\phi_s = 1.0$

$$\Delta P_s = \frac{(0.0025)(69{,}300)^2(1)(17)}{5.22\,(10)^{10}(0.06)(0.78)} = 0.0835 \text{ psi}$$

Use ΔP_s = 1.0 to 1.5 ×0.0835 psi for any critical pressure drop considerations. This should be safe.
Pressure drop, tube side, from Figure 10-139.
End return loss, ΔP_r = (0.75) (4) = 3.00 psi,
from Figure 10-138.
Lb water per tube/pass = 50,000/11 = 4,550
ΔP = (7/100) (8 ft) (4 passes) = 2.24 psi
Total pressure drop = 2.24 + 3.0 = 5.24 psi
Use ΔP = 6 psi
Nozzle sizes:
Inlet water rate = 100 gpm
Velocity in 3-in. connection = 4.34 ft/sec
Head loss = 0.0447 (6 in./12) = 0.022 ft water
Outlet nozzle to be same.
Inlet shell side (bottom flow):

$$\text{Liquid rate} = \frac{6{,}350}{(60)(8.33)(0.78)} = 16.2 \text{ gpm}$$

Kinematic viscosity = 0.404/0.78 = 0.52 centistokes
From Cameron Miscellaneous Liquids Table (Fluid Flow Chapter, Vol. I),
Velocity in 2-in. connection = 1.53 ft/sec
Note that a 1 $^1/_2$-in. connection is satisfactory; however many plants prefer minimums of 2-in. connections on process vessels for main stream flows.
Pressure loss is negligible = 0.006 (6 in./12) = 0.003 ft fluid
Outlet shell side:
Use same size as inlet, 2 in.

Spiral Coils in Vessels

Spiral coils can be useful in transferring heating and cooling from the helical or nonhelical coil to and from a volume of liquid in a process vessel or storage tank. These coils in a stagnant or noncirculating tank are almost useless; therefore, the best arrangement is to use the coil in an agitated/mixing tank. See Chapter 5 of Volume 1, 3rd Edition of this series.

Tube-Side Coefficient

Kern[70] reports that tube-side coefficients can be approximately 20% greater in a spiral coil than in a straight pipe or tube using the same velocities. The Sieder-Tate correlation is shown in Equations 10-44 and 10-45 and for streamline flow is $DG/\mu < 2{,}100$. For transition and turbulent flow, see Equation 10-46 and Figure 10-46 or Figure 10-50A and 10-50B for straight pipes and tubes. McAdams[81] suggests multiplying the h value obtained by $(1 + 3.5\ (D/D_H))$, when D is the inside diameter of the tube and D_H is the diameter of the helix, in ft.[70]

Outside Tube Coefficients

This design is not well adapted to free-convection heat transfer outside a tube or coil; therefore, for this discussion only agitation is considered using a submerged helical coil, Oldshue[241] and Kern[70].

$$\frac{h_c D_j}{k} = 0.87\left(\frac{L^2 N\rho}{\mu}\right)^{2/3}\left(\frac{C_p\mu}{k}\right)^{1/3}\left(\frac{\mu}{\mu_w}\right)^{0.14} \tag{10-68}$$

Using the nomenclature of Equation 10-44, in addition:

h_c = heat transfer coefficient for outside of coil, Btu/(hr) (ft²)((F)
D_j = diameter of inside of vessel, ft
L = tube length, ft
N = agitator speed, rev/hr
ρ = density, lb/ft³
μ = viscosity, lb/ft-hr
k = thermal conductivity of liquid, Btu/(hr) (ft²) (°F/ft)
C_p = specific heat, Btu/(lb) (°F)

A related but somewhat more recent work by Oldshue[241] presents heat transfer to and from helical coils in a baffled tank, using standard baffling of T/12 located either inside the coil diameter or outside:

$$h_o(\text{coil})\frac{d_t}{k} = 0.17\left(\frac{d^2N\rho}{\mu}\right)^{0.67}\left(\frac{C_p\mu}{k}\right)^{0.37}\left(\frac{D}{T}\right)^{0.1}\left(\frac{d}{T}\right)^{0.5}\left(\frac{\mu}{\mu_s}\right)^{m} \tag{10-69}$$

where
(use conventional units for symbols)

m = $0.1\ (\mu\ 8.621 \times 10^{-5})^{-0.21}$
C_p = heat capacity, Btu/(lb) (°F)
D = impeller diameter, ft
d_o = tube diameter, ft
d_t = tube O.D., ft
h_o = outside (process fluid side) heat transfer coefficient
k = thermal conductivity of liquid, Btu/(hr) (ft²) ((F/ft)
m = experimental exponent, usually 0.14.
N = impeller speed, rev/hr
T = tank diameter, ft
μ = viscosity, bulk fluid, lb/(ft) (hr)
μ_s = viscosity of fluid at film temperature at heat transfer surface, lb/(ft) (hr)
ρ = liquid density, lb/ft³
U_o = overall heat transfer coefficient based on outside tube area

Condensation Outside Tube Bundles

Film-type condensation is considered to be the usual condition for most pure vapors, although drop-type condensation gives transfer coefficients many times larger when it does occur. For practical purposes, film-type is considered in design.

Figure 10-66 indicates the usual condensing process, which is not limited to a vertical tube (or bundle) as shown, but represents the condensing/cooling mechanism for any tube. The temperature numbers correspond to those of Figure 10-28.

Vertical Tube Bundle[70]

See Figure 10-67A and 10-67B.

Figure 10-67A has been initially represented by McAdams[82] from several investigators. This figure represents the mean coefficient for the entire vertical tube for two values of the Prandtl number, Pr_f, which = $c\mu/k$.

where

c = specific heat of fluid, Btu/(lb) (°F)
μ = fluid viscosity, lb/(ft) (hr)
k = thermal conductivity, Btu/(hr) (ft²) (°F/ft)

Note that the break at Point A on Figure 10-67B at Re_c = 2,100 indicates where the film is believed to become turbulent.[172] McAdams[82] discusses the two regions on the figure, streamlined at the top and turbulent on the way down, with a transition region in between:

$$Re_c = \frac{4\,\Gamma}{\mu_l} \tag{10-70}$$

where

$G' = \Gamma = w/\rho_t$, condensate loading for each vertical tube, lb/(hr) (ft)
w = flow rate, rate of condensation per tube, W/N_t, lb/(hr)(tube), from lowest point of tube(s)

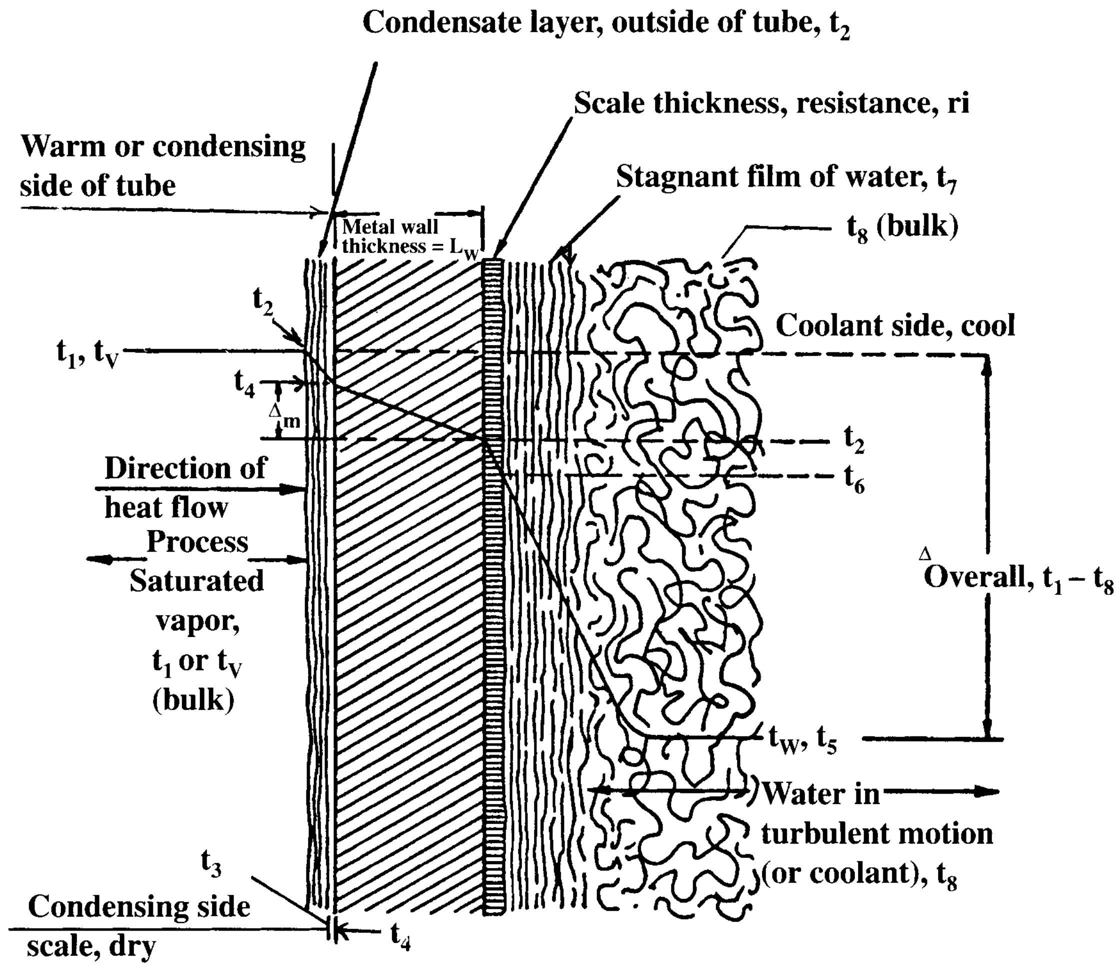


Figure 10-66. Condensing vapors on cooling metal (or other) wall (also see Figure 10-28). Note that t_4 and t_5 are wall temperatures and may be essentially equal to t_w = wall. This illustration is not for vertical tube, but represents the condensing/cooling mechanism.

ρ_t = πd_o for vertical tube (perimeter), ft
d_o = tube outside diameter, ft
Γ' = mass rate of flow of condensate from lowest point on condensing surface divided by the breadth (unit perimeter), lb/(hr) (ft). For a vertical tube: $\Gamma' = w/\pi D$.
G'' = condensate loading for horizontal tubes, lb/(hr) (ft)
G' = condensate loading for vertical tubes, lb/(hr) (ft).

McAdams[82] and Kern[70] both suggest the same relationship for condensation on the outside of vertical tubes:

$$h_c = \frac{(\mu_f^2)^{1/3}}{(k_f^3\rho_f^2 g)} = 1.47\frac{(4G')^{-1/3}}{(\mu_f)} \qquad (10\text{-}71)$$

g = acceleration of gravity, 4.17×10^8, ft/(hr) (hr)

Bell[172] suggests the relation:

$$h_c = 0.943\frac{[k_l^3\rho_l(\rho_l - \rho_v)\lambda g]^{1/4}}{[\mu_l L(T_{sat} - T_w)]} \qquad (10\text{-}72)$$

where
k_l = liquid thermal conductivity, Btu/(hr) (ft) (°F)
ρ_l = liquid density, lb/ft^3
ρ_v = vapor density, lb/ft^3
λ = latent heat of vaporization, Btu/lb
g = acceleration of gravity, ft/(sec) (sec)
L = tube length, ft
T_{sat} = saturation temperature, °F
T_w = surface temperature, °F
μ_l = liquid viscosity, lb/(ft) (hr)

$$\text{For } \frac{4G'_o}{\mu_t} < 2{,}000 \qquad (10\text{-}73)$$

$$h_{cm} = 0.945\left[\frac{k_f^3\rho_f^2 g}{\mu_f G_o'}\right]^{1/3} = 0.945\left[\frac{k_f^3\rho^2 tg\pi N_t D_o}{\mu_f W}\right]^{1/2} \qquad (10\text{-}73A)$$

$$G_o' = \frac{W}{\pi N_t D_o}\text{, lb./hr. (linear foot)} \qquad (10\text{-}73B)$$

For 4 $G_o'/\mu_f > 2{,}000$ (reference 82),the following equation is usually applicable to long tubes and high flow rates; the average film coefficient:

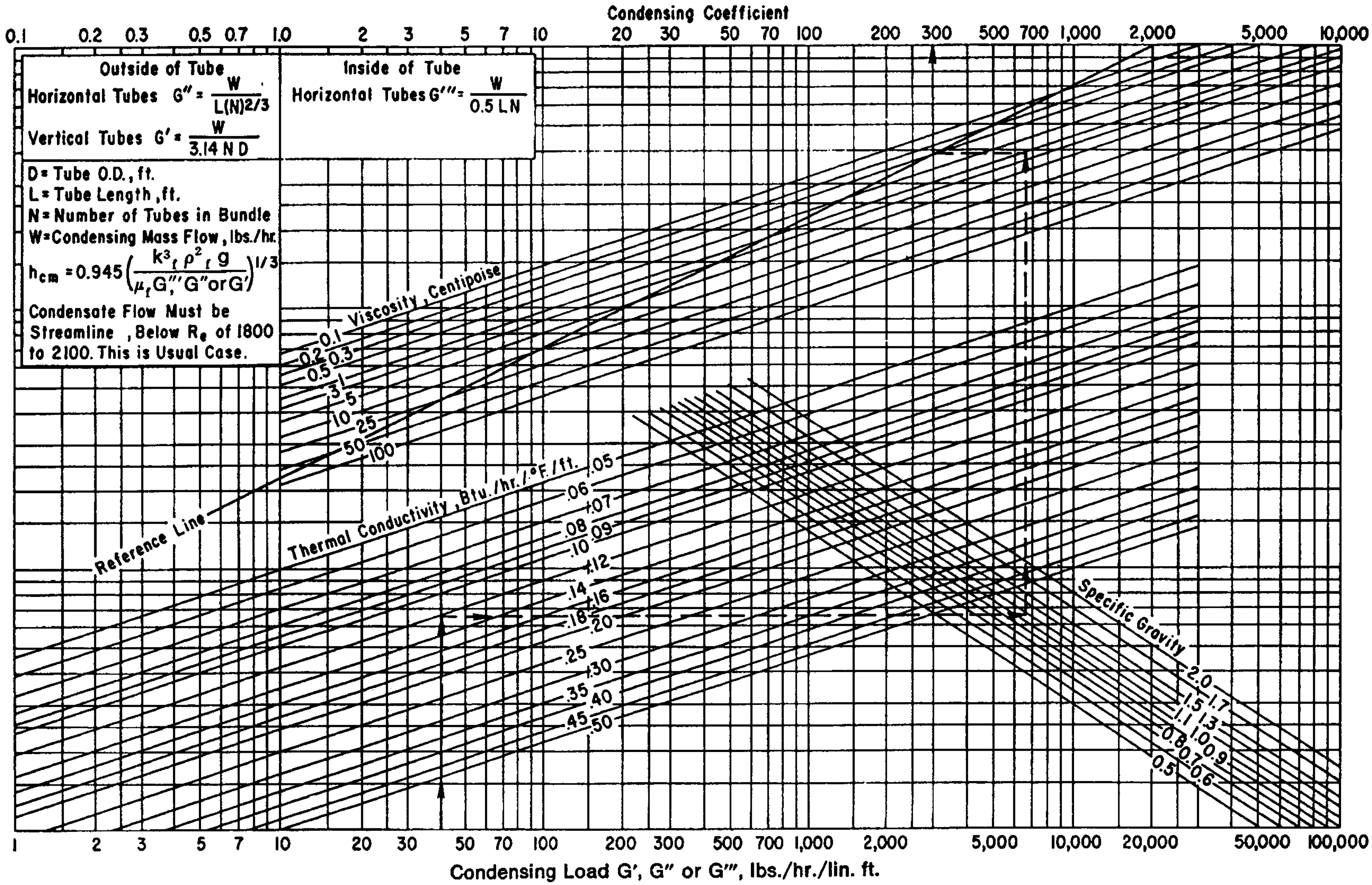


Figure 10-67A. Condensing film coefficients outside horizontal or vertical tubes. (Used by permission: Kern, D.Q. *Process Heat Transfer,* 1st Ed., ©1950. McGraw-Hill, Inc. All rights reserved.)

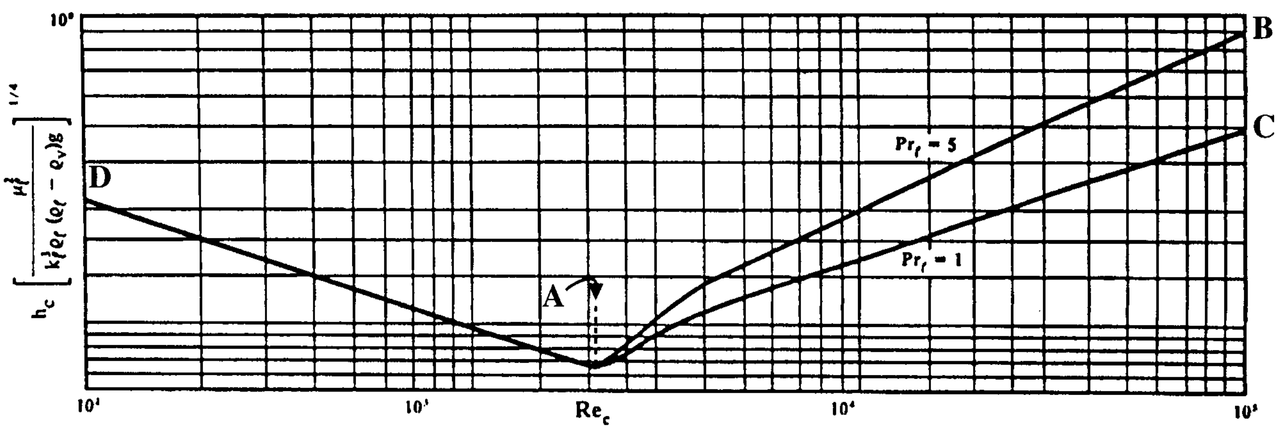


Figure 10-67B. Correlation of McAdams[82] representing the condensing film coefficient on the outside of vertical tubes, integrated for the entire tube length. This represents the streamline transition and turbulent flow conditions for Prandtl numbers 1 and 5. Do not extrapolate Prandtl numbers, Pr_f, beyond 5. (Used by permission: *Engineering Data Book II* ©1984, Wolverine Tube, Inc.)

$$h_{cm} = 0.0077\left(\frac{k_f^3\rho_f^2 g}{\mu_f^2}\right)^{1/3}\left(\frac{4W}{\mu_f \pi D_o}\right)^{0.4} \tag{10-74}$$

For steam at atmospheric pressure and Δt from 10°F–150°F:[82]

$$h_{cm} = \frac{4{,}000}{L^{1/4}\Delta t^{1/3}} \tag{10-75}$$

where

L = tube length, ft

$\Delta t = t_{sv} - t_w$ = (temperature of saturation of dew point—temperature of tube wall surface), °F.

Horizontal Tube Bundle[70]

See Figures 10-67A and 10-67B.

For single pure vapors Kern[70] recommends the following, due to the splashing of condensed liquid (outside) from horizontal tubes as it drips/splashes to and off of the lower tubes in the bundle:

$$G'' = \frac{W}{LN_t^{2/3}}, \text{ lb/(hr)(linear ft); see Figure 10-67A} \tag{10-76}$$

Then the heat transfer for condensation is[70, 82] on a horizontal bundle:

$$h_{cm}\frac{(\mu_f{}^2)^{1/3}}{(k_f^3\rho_f^2 g)} = \frac{1.5\,(4G'')^{-1/3}}{(\mu_f)} \tag{10-77}$$

The preceding equation automatically allows for the effect of the number of vertical rows of horizontal tubes as proposed by Kern[70] and cited later in this discussion.[82] The flow should be streamlined (laminar) flow, with a Reynolds Number of 1,800– 2,100 for the condensation,[82] see Figures 10-67A and 10-67B.

$$N_{Re,f} = 2\,\Gamma'/\mu_f \tag{10-78}$$

The critical Reynolds Number of 2,100 corresponds to $4\Gamma'/\mu_f$ of 4,200 for horizontal tube.[82]

Γ' = w/L, per horizontal tube, lb/ (ft) (hr)

The thickness of the film[94A] for

Reynolds Number $< 2{,}100 = (3\mu\Gamma'/\rho^2 g)^{1/3}$

For steam at atmospheric pressure,[82] the average film coefficient is

$$h_{cm} = \frac{5{,}800}{(N_v D_o')^{1/4}(\Delta t_m)^{1/3}} \tag{10-79}$$

where

N_v = number of rows of tubes in a vertical tier

D_o' = tube O.D., in.

$\Delta t_m = (t_v - t_w)/2$, °F

t_v = temperature of vapor, °F

t_w = temperature of tube wall, °F

h_{cm} = average value of condensing film coefficient, Btu/hr (ft^2) (°F), for vertical rows of horizontal tubes

k_f = thermal conductivity at film temperature, Btu/(hr) (ft^2) (°F/ft)

ρ_f = density lb/ft^3 at film temperature, t_f

g = acceleration of gravity, ft/(hr) (hr) = 4.17×10^8

μ_f = viscosity at film, lb/(ft) (hr) = centipoise $\times$ 2.42 = lb/ (ft) (hr)

W = flow rate, lb/hr, condensate

D_o = outside diameter of tubes, ft

N_t = total number of tubes in bundle used for condensation

L = tube length, ft, straight

The charts of Chen[26] are also useful for solving the equations for condensing coefficients; however, the correction for the effect of multiple fluid stream is not included. Therefore, the results should be conservative.

Devore[34] has presented useful charts for solving a multitube condenser design as shown in Figures 10-68, 10-69, and 10-70. Figure 10-71 is useful for condensing steam. The charts all follow Nusselt's basic presentation; however, a correction for turbulence of the film and other deviations is included.

Rohsenow and Hartnett[166] present Nusselt's relation for the heat transfer average for horizontal tubes in a bundle condensing vertically from tube to tube, top to bottom tube:

$$h = 0.728\frac{[g\rho_l(\rho_l - \rho_v)k^3h'_{fg}]^{1/4}}{[nD_o\mu\Delta T]},$$

average for horizontal tubes in vertical bank (10-80)

where

g = acceleration of gravity, 32.17 ft/hr^2 = [ft/sec^2 (3,600)^2]

D_o = tube outside diameter, ft

c = specific heat of liquid at constant pressure

h = heat transfer coefficient, Btu/(hr) (ft^2)(°F)

h_{fg}' = h_{fg} + ($^3/_8$) c(T_s − T_w)

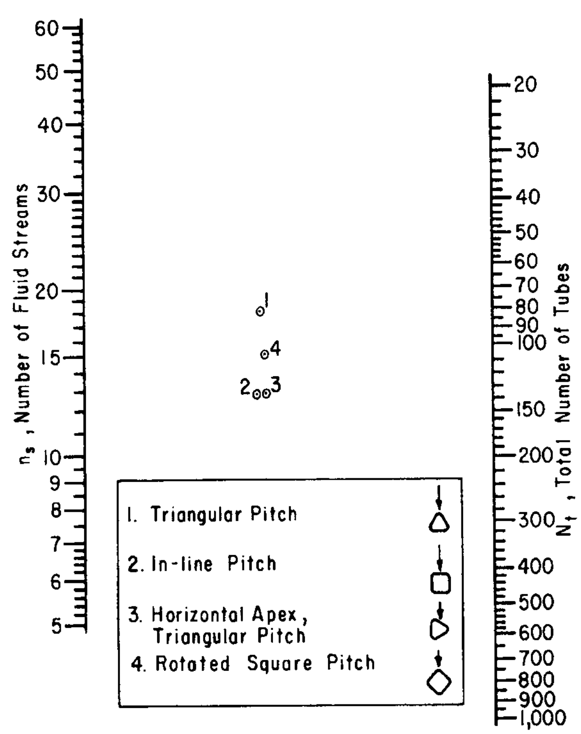


Figure 10-68. Number of condensate streams in a horizontal bundle. (Used by permission: Devore, A. *Petroleum Refiner,* V. 38, No. 6, ©1959. Gulf Publishing Company, Houston, Texas. All rights reserved.)

FIGURE 6 (below)—In order to use this nomogram for turbulence correction factor for horizontal multitube banks, first evaluate N_s by using Figure 3 to find n_s and then Equation (35) to solve for N_a.

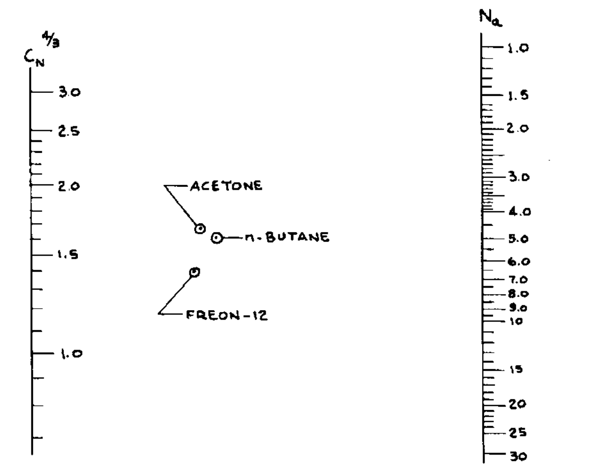
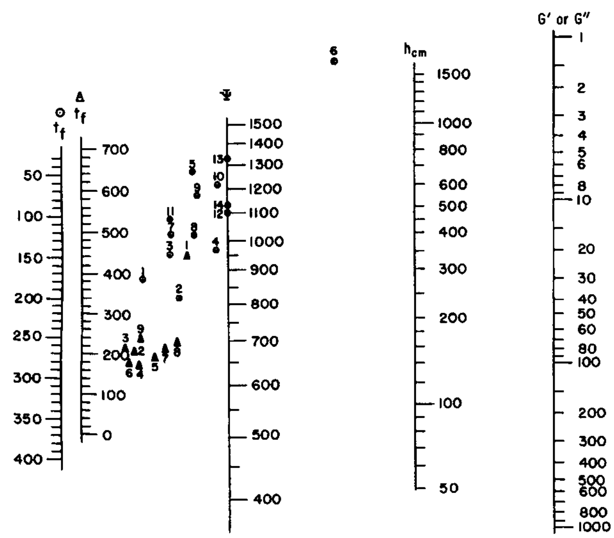


Figure 10-69. Turbulence correction factor for horizontal multitube banks. Evaluate N_a by solving for n_s from Figure 10-68. (Used by permission: Devore, A. *Petroleum Refiner,* V. 38, No. 6, ©1959. Gulf Publishing Company, Houston, Texas. All rights reserved.)



Note: These Condensate Film Coefficients for Vertical Surfaces are Restricted to $G'/\mu'_f \leqq 1090$. To Obtain Theoretical h_{cm} for Horizontal Tubes, use the above Nomogram and Multiply Result by 0.8.

Substance	Range, °F	Point
n-Propanol	32-122	1
n-Propanol	122-380	2
Ethanol	32-158	3
Ethanol	158-377	4
Methanol	32-172	5
Methanol	172-365	6
CCl_4	32-118	7
CCl_4	118-390	8
$CHCl_3$	32-154	9
$CHCl_3$	154-320	10
Benzene	32-114	11
Benzene	114-383	12
Acetone	32-300	13

Substance	Range, °F	Point
Diethyl Ether	32-257	14
Eutectic Mixture Of Diphenyl And Diphenyl Oxide	500-700	△1
Dichlorodifluoromethane	32-140	△2
Chlorodifluoromethane	0-120	△3
sym-Dichlorotetrafluoroethane	32-175	△4
n-Pentane	50-150	△5
n-Pentane	150-250	△6
n-Hexane	50-218	△7
n-Hexane	218-350	△3
n-Octane	50-250	△8
n-Octane	250-400	△9

Figure 10-70. Condensate film coefficients—vertical or horizontal. (Used by permission: Devore, A. *Petroleum Refiner,* V. 38, No. 6, ©1959. Gulf Publishing Company, Houston, Texas. All rights reserved.)

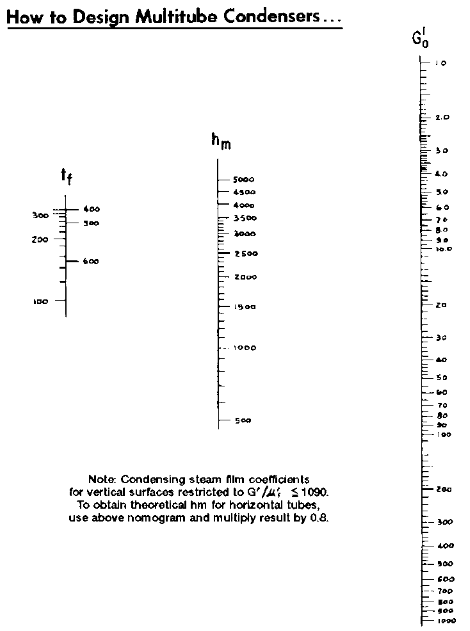


Figure 10-71. Condensing steam film coefficients for vertical surfaces or horizontal tubes. G_o'/μ_f' restricted to $\leq$ 1,090. For theoretical h_m for horizontal tubes, use and multiply results by 0.8. G_o' = condensate mass flow per unit tube outside circumference, vertical tubes, lb/(hr)(ft). (Used by permission: Devore, A. *Petroleum Refiner,* V. 38, No. 6, ©1959. Gulf Publishing Company, Houston, Texas. All rights reserved.)

h_{fg} = λ = latent heat, Btu/lb
λ = latent heat of vaporization, Btu/lb
k = thermal conductivity of the liquid at film temperature, Btu/(hr) (ft) (°F)
n = number of horizontal tubes in a vertical bank
ΔT = $T_s - T_w$, °F
T_s = temperature at saturation pressure, °F
T_w = temperature at wall, °F
μ = viscosity of liquid, lb/(ft) (hr)
ρ_l = density of liquid, lb/ft³
ρ_v = density of vapor, lb/ft³

Reference 166 points out that the preceding equation provides results lower than actual experience.

As reported by references 166 and 168, Chen's[167] proposed relationship provides better results; Chen assumes subcooling is removed from the condenser:

$$h = 0.728 \frac{[1 + 0.2(c\Delta T)(n - 1)]}{[h_{fg}]} \frac{g\rho_l(\rho_l - \rho_v)k^3h'_{fg}]^{1/4}}{[nD\mu\Delta T]} \quad (10\text{-}81)$$

where symbols are the same as for reference 166. Agreement with test data is good when $(c\Delta T/h_{fg}) < 2$.

Bell and Mueller[172] present the following equation, which is similar to several of the others for condensing outside single horizontal tubes:

$$h_c = 0.728 \frac{[k^3\rho_l(\rho_l - \rho_v)\lambda g]^{1/4}}{[\mu_l D_o(T_{sat} - T_w)]} \quad (10\text{-}82)$$

or

$$h_c = 0.951 \frac{[k_l^3\rho_l(\rho_l - \rho_v)gL]^{1/3}}{[\mu_l W]} \quad (10\text{-}83)$$

where

h_c = average condensing coefficient on outside of tube, Btu/(hr) (ft²) (°F)
T_{sat} = saturation temperature of the vapor, °F
T_w = wall temperature of tube, °F
L = length of tube for heat transfer, ft
W = vapor weight (mass) flow rate, lb/hr
D_o = outside tube diameter, ft

Subscripts:

l = liquid
c = condensing
v = vapor

The preceding equations are reported to predict actual heat transfer coefficients only about 15% lower than experimental values—the difference can be attributed to the rippling of the film and early turbulence and drainage instabilities on the bottom side of the tube.[172]

General design practice is to assume that the average coefficient calculated for a single tube is the same as for an entire bundle, based on test data.[172]

In horizontal condensers (outside tubes), for N tubes in a vertical row, with the condensate flowing uniformly from one tube to the one below without extensive splashing, the mean condensing coefficient, h_m, for the entire row of N tubes (per Knudsen in reference 94A) is related to a film coefficient for the top, h_l, single tube by:

$$h_{m(new)} = h_l N^{-1/4}, \text{ (a severe penalty)} \quad (10\text{-}84)$$

h_l is calculated by the previous listed equations.

Kern[70] recommends:

$$h_{m(row)} = h_l (N)^{-1/6} \quad (10\text{-}85)$$

Short and Brown[174] in reference 172 found no net penalty against the single tube coefficient in a single row 20 tubes high. Bell[172] concurs that this is borne out in industrial experience, and "current design practice is to assume that the average coefficient for the entire tube bank is the same as for a single tube."

Stepwise Use of Devore Charts

1. Based upon condensing heat load, log Δt and an assumed overall coefficient, U, estimate the required surface area.

2. Determine the number and size of tubes required to calculate this area. Also set the number of tube passes and tube pitch.
3. Determine the inside film coefficient by methods previously outlined for convection.
4. Estimate film temperature of fluid on the outside of tubes and determine Δt across condensing film.
5. For vertical tubes, determine condensate loading G_o' (Equation 10-73B). For these charts, G_o' (viscosity in centipoise, at film temperature) is limited to 1,090.
6. For horizontal tubes, use Figure 10-68 to determine the equivalent number of condensate streams, n_s, based on the total number in a circular bundle, N_t.
   (a) Then calculate condensate loading (horizontal tubes):

$$G'' = W/Ln_s \tag{10-86}$$

   Note that this varies from the form used in the Kern relation.
   (b) Determine the average number of tubes, N_a:

$$N_a = N_t/n_s \tag{10-87}$$

   (c) Determine turbulence correction factor $C_N^{4/3}$ from Figure 10-69. For compounds other than those shown, select the nearest type for reference and evaluate C_N. To obtain more conservative results, reduce the value of $C_N^{4/3}$ but never to less than 1.0.
7. Film coefficients: h_{cm}
   (a) For vertical tubes, use Figure 10-70 or 10-71 and corresponding scales for compounds at t_f and G' (as defined for use with *these* charts).
   (b) For horizontal tubes, use Figure 10-70 or 10-71 and corresponding scales for compounds at t_f and G'' (as defined for use with *these* charts).
8. Evaluate the overall clean coefficient, U_c.
9. Check the assumed temperature drop across the condensate film, Δt.

$$\Delta t = (U_c/h_{cm})(t_h - t_c) \tag{10-88}$$

   If these values are not in good agreement, reassume and recalculate.
10. Calculate the overall fouling coefficient, adding the appropriate fouled factors to clean, U_c.
11. Determine the required surface area:

$$A = Q/U\,\Delta t \tag{10-89}$$

If this surface area is slightly less than that assumed for the unit, say 10–20%, the unit should be acceptable. If the required area is larger, the new number of same length can be determined by the ratio of required area/assumed area × the number of tubes in the assumed unit when the effect of any new shell diameter should be reviewed; otherwise the unit should perform satisfactorily.

Subcooling

The literature is limited on design data/correlations for subcooling condensed liquids in or on vertical or horizontal condenser units. Certain analytical logic can be used to examine what is taking place and the corresponding heat transfer functions can be used to establish the relations to break the desired unit into its components (in terms of heat transfer) and combine them to develop a single condensing/subcooling unit. Also see reference 70.

Subcooling Condensate Outside Vertical Tubes

The area concerned with the subcooling only can be evaluated using a film coefficient calculated from Figure 10-54 for liquids outside tubes. This assumes that the liquid being cooled is held in the area around the tube by a level control or pipe seal, allowing drainage at the rate it builds up and covering a portion of the tubes.

Care should be used in determining the temperatures that prevail at tube inlet and outlet, as well as the shell side in and out for the subcooling portion. This becomes particularly tedious for multipass units.

Subcooling Inside Vertical Tubes

Colburn et. al.[173] conducted some fundamental studies using organic liquids; they developed the *subcooling coefficient* when $R_e > 2100$:

$$h_s = 7.5\left[\frac{(k^2\rho^2C_p)}{(\mu_f)}\,\frac{(4\Gamma)}{(\mu_f)}\right]^{1/3} \tag{10-90}$$

which is used to calculate the area required, and then this area is added to an earlier calculated condensing area.

where
h_s = subcooling film coefficient, pcu/(hr) (ft^2) (°F)
k = thermal conductivity, Btu/(hr) (ft) (°F)
ρ = liquid density, lb/ft^3
C_p = heat capacity of condensate, pcu/(lb) (°C)
Γ = condensate rate per unit periphery, lb/(hr) (ft)
μ_f = viscosity of condensate at average film temperature, lb/(hr) (ft)

Note: 1.0 Btu/(ft^2) (hr)(°F) = 1.0 pcu/(hr) (ft^2)(°C)

Subcooling Condensate Outside Horizontal Tubes

Subcooling in horizontal condensers is accomplished by a liquid seal on the liquid outlet or by a baffle, which dams the liquid and allows it to overflow through the outlet. Here also, the subcooling area is calculated separately from the condensing area. The two are then added to obtain the total. Usually the liquid held to be subcooled covers only about 15–30% of the total surface, although some units may run as high as 50%. If the quantity of the liquid is very large, handling it in a separate liquid cooler where higher coefficients can be obtained is possibly a better solution.

The cooling of the condensate by free convection is[70]

$$h_c = 116\left[\left(\frac{k_f^3 \rho_f c_f \beta}{\mu_f'}\right)\left(\frac{\Delta t}{d_o}\right)\right]^{0.25} \qquad (10\text{-}91)$$

where

μ_f' = viscosity in centipoise
Δ_t = temperature difference between tube surface and fluid, °F
d_o = O.D. tube, in.
β = coefficient of thermal expansion, %/°F
ρ = density, lb/ft^3
k_f = thermal conductivity of film, Btu/hr (ft^2) (°F/ft)
c_f = specific heat, Btu/lb (°F) at film conditions
t_f = $(t_w + t_a)/2$, °F, average, film temperature
t_a = bulk fluid temperature, °F
g = acceleration of gravity, ft/(hr)^2
G_o' = condensate mass flow per unit tube outside circumference, vertical tubes, lb./(hr) (ft)

The usual range of film coefficient values is 40–50 for organic solvents and light petroleum fractions such as hexanes; 25 for heavier materials such as aniline, straw oil, etc.; and 0.5–3 for low temperature (10–40°F) subcooling of heavier organics and inorganics such as chlorine.

Film Temperature Estimation for Condensing

Kern[70] recommends the temperature to use in estimating or determining fluid properties:

$$t_f = 1/2\ (t_b - t_w) \qquad (10\text{-}92)$$

where

t_b = bulk temperature of fluid, °F.
t_w = wall temperature of tube surface, °F

$$\Delta t_f = t_f - t_w$$

McAdams[82] recommends:

$$t_f = t_{sv} - 3/4\ \Delta t \qquad (10\text{-}93)$$
$$\Delta t = t_{sv} - t_w$$

where

t_{sv} = saturation or dew point temperature, °F.

In most instances the effect of the difference on physical properties will be small.

Condenser Design Procedure

The usual total condenser will follow the following design steps:

1. Establish condensing temperature of vapors, either by the conditions of other parts of the process (distillation column, vacuum jet, etc.) or by the temperature approach to cooling water, remembering that a close approach will require relatively large surface area. Select the cooling water temperature to ensure performance in the summer months and consider the conditions during the winter (see step 8q).
2. Establish film temperature for condensation from Equation 10-26 or 10-28.
3. Establish physical properties of fluids, shell side at a different temperature than tube side.
4. Calculate the heat load of condensation from latent heat. (This may be a weighted value for a mixture.)
5. Set an allowable temperature rise for the cooling water.
6. Calculate water rate:

$$W = Q/c_p\ \Delta t,\ \text{lb/hr} \qquad (10\text{-}94)$$

Q = Btu/hr
Δt = temperature rise of water, °F
c_p = Btu/lb °F

$$\text{gpm} = \frac{W}{(8.33)(60)} \qquad (10\text{-}95)$$

(for water, otherwise correct 8.33 lb/gal for sp. gr. of coolant)

$$\text{ft}^3/\text{sec} = \frac{\text{gpm}}{(7.48)(60)} = \text{cfs} \qquad (10\text{-}96)$$

7. Estimate the number of tubes per pass to maintain minimum water velocity.
   Set minimum velocity in tubes at 3.5–6 ft/sec = v.
   Water flow area = cfs/v, ft^2 = a
   Select tube size and calculate flow area available:

$$\text{Flow area/tube} = \frac{\text{tube cross-section, in}^2}{144} = \text{ft}^2/\text{tube} \qquad (10\text{-}97)$$

$$\text{Estimated no. tubes/pass} = \frac{a}{\text{ft}^2/\text{tube (cross-sect.)}} = n' \qquad (10\text{-}98)$$

8. Assume a unit:
   (a) Estimate overall coefficient, U, from Tables 10-15, 10-17, and 10-18 or by your own experiences.
   (b) Roughly calculate a log mean temperature difference ΔT.
   (c) Estimate area = $Q/U\ \Delta t$, ft^2
   (d) The total tube footage required = A/(ft^2 surface/ft tube length), ft = F_L

(e) Assuming a tube length, l:

$$\text{No. passes} = \frac{F_l}{(n')(l)} = P_a \qquad (10\text{-}99)$$

If this value is not reasonable, reassume the tube length, and/or the size of tubes. Try to keep the number of passes fewer than 8 except in special cases, as construction is expensive.

(f) From Table 10-9 pick an exchanger shell diameter that closely contains the required number of tubes at the required number of passes.

(g) From the actual tube count selected, establish the actual number of tubes/pass. They may be a few tubes more or less than initially figured.

Calculate the flow area/pass = (number of tubes/pass) (cross-section flow area/tube), ft²/pass.

(h) Calculate velocity in tubes = cfs/(ft²/pass), ft/sec.

(i) For the film coefficient, tube side, read h_i from Figure 10-50A or 10-50B at the mean water temperature and calculated velocity of (h).

Correct to the outside of tube:

$$h_{io} = (h_i)(\text{tube dia. film correction, } F_w)\left(\frac{\text{tube I.D.}}{\text{tube O.D.}}\right),$$

$$\text{Btu/hr (ft}^2)(^\circ\text{F}) \qquad (10\text{-}100)$$

(j) For the film coefficient, shell side, calculate G_o'' from Equation 10-73B or 10-76, lb/hr (lin. ft)

Do not use full tube length as effective, reduce "l" by the estimated tubesheet thickness at each end; usually $1\,^1/_2$ in. per tubesheet for low pressure (to 150 psi) and 3 in. for higher pressure (to about 600 psi) is satisfactory.

From Figure 10-67A, read h_o, Btu/hr (ft²) (°F).

(k) Select fouling factors from tube side and shell side, from Table 10-12 or 10-13 or your own experience.

(l) Calculate the overall coefficient:

$$U = \frac{1}{\dfrac{1}{h_o} + r_o + r_{io} + \dfrac{1}{h_{io}}}, \text{ Btu/(hr)(ft}^2)(^\circ\text{F}) \qquad (10\text{-}101)$$

Usually the tube wall resistance can be neglected, but if you doubt its effect, add to the resistances.

(m) Calculate log mean temperature difference by using Figure 10-33 or Equation 10-13.

(n) Area required:

$$A = Q/U\,(Dt), \text{ ft}^2 \text{ net}$$

(o) Area available in assumed unit:

$$A = (\text{ft}^2 \text{ surface/ft tube})(\text{number of tubes total})(\text{net tube length}) \qquad (10\text{-}102)$$

(p) Compare, and if the available area is equal to or greater than the required area, the selected unit will perform satisfactorily. If the required area is greater than the available area, select a new unit with more tubes, longer tubes, larger tubes, or some combination. Repeat from step 8, unless the minimum water velocity can safely be changed, then repeat from step 7.

(q) Check the effect of winter operating conditions on the importance of (1) maintaining constant yearly outlet condensate temperature; (2) subcooled condensate as a result of excess surface area due to lower inlet cooling water; and (3) maintaining a minimum water velocity in tubes.

Example 10-10. Total Condenser

Ammonia vapors from a stripping operation are to be condensed. Select the condenser pressure, which sets the top of stripper pressure, and design a condenser. Water at 90°F is to be used.

Flow: 1,440 lb/hr ammonia, at dewpoint.

1. The condenser operating pressure should be so selected as to give a reasonable temperature difference between the condensing temperature and the water temperature. The quantity of water required should not be penalized by requiring a small temperature rise in the water.
   By referring to a Mollier diagram for ammonia, the condensing temperature at 220 psig is 106.6°F. This is about the lowest operating pressure possible to keep a ΔT of greater than 10°F between water and ammonia.
2. Heat load:

$$Q = (1440)(470.5 \text{ Btu/lb latent heat}) = 680{,}000 \text{ Btu/hr}$$

3. Minimum water tube velocity: set at 5 ft/sec.
4. Water required for 10°F temperature rise:

$$Q = Wc_p\,\Delta T$$

$$W = \frac{680{,}000}{(1)(10)} = 68{,}000 \text{ lb/hr}$$

$$\text{gpm} = \frac{68{,}000}{(8.33)(60)} = 136$$

$$\text{ft}^3/\text{sec} = \frac{136}{(7.48)(60)} = 0.303$$

5. Water flow area:

Total tube cross-section flow area = 0.303/5 ft/sec = 0.0606 ft²

6. Number of tubes, using 1-in., 12 BWG tube:

Tube flow area = 0.479 in.²/144 = 0.00333 ft²/tube

$$\text{No. tubes} = \frac{0.0606}{0.00333}$$

=18.2 per pass to keep the 5 ft/sec water velocity

7. Assumed condenser unit:

Assume U = 200
LMTD = 10°

$$A = \frac{680{,}000}{(200)(10)} = 340 \text{ ft}^2$$

Tube length required:

Outside area/tube = 0.2618 ft²/ft
Total tube footage = 340/0.2618 = 1,300 ft

Try 15.25-in. I.D. shell 4 pass, with 78 tubes total, 16-ft long on 1 $^1/_4$-in. triangular pitch.
Trial area = (78) (0.2618) (16-ft long) = 326 ft²

This does not allow for tube area lost inside the two tubesheets of a fixed T. S. unit.

8. Flow area:

Number of tube side passes = 78/18.2 = 4.28, use 4 tube passes
Number of tubes/pass = 78/4 = 19.5, use 19
Flow area/pass = (19) (0.00333) = 0.0633 ft²

$$\text{Velocity in tubes} = \frac{0.303 \text{ cfs}}{0.0633 \text{ ft}^2/\text{pass}} = 4.8 \text{ fps/pass}$$

9. For the film coefficient, tube side, use Figure 10-50A or 10-50B:

Mean water temperature = 95°F
Vel = 4.8
read h_i = 1,150 Btu/hr (ft²) (°F)
correction for 1-in. tube Fw = 0.96 at 0.782-in. I.D.
Correction to outside of tube:
h_{io} = (1,150) (0.96) (0.782/1.0) = 864 Btu/hr (ft²) (°F)

10. Film coefficient, shell side:

$$\text{Condensate loading: } G_o'' = \frac{W}{LN_t^{2/3}}$$

$$G_o'' = \frac{1440}{(16 \text{ ft} - 6 \text{ in.}/12)(78)^{2/3}} = 5.1 \text{ lb/hr (lin ft)}$$

(allowing 3-in. thickness/tubesheet, thus reducing effective tube length)

Thermal conductivity, k_a = 0.29 Btu/(hr) (ft²) (°F/ft)
Specific gravity = 0.59
Viscosity = 0.085 centipoise
From Figure 10-67A or 10-67B, read:
h_o = 2500 Btu/hr(ft²) (°F)
Although this is satisfactory, because it is so large, use
h_o = 1500

In many cases, a "safety" factor of greater than 10–15% is not justified.
Select the tube-side fouling of water = 0.002.
Select the shell-side fouling of ammonia = 0.001

11. Overall coefficient:

$$\frac{1}{U} = \frac{1}{1{,}500} + 0.001 + 0.002 + \frac{1}{864}$$

$$\frac{1}{U} = 0.00067 + .003 + 0.00116 = 0.00483$$

U = 207 Btu/hr (ft²) (°F) (neglecting tube wall resistance)

12. Log mean temperature difference:

$$\begin{array}{ccc} & \text{Condensing} & \\ 106.6°\text{F.} & \longrightarrow & 106.6°\text{F.} \\ \underline{100°\text{F.}} & \xleftarrow{\text{Warming}} & \underline{90°\text{F.}} \\ 6.6 & & 16.6 \end{array}$$

$$\text{LMTD} = \frac{16.6 - 6.6}{\ln \dfrac{16.6}{6.6}} = 10.8°\text{F (Figure 10-33)}$$

13. Area required:

$$A = \frac{Q}{U\Delta t} = \frac{680{,}000}{(207)(10.8)} = 304 \text{ ft}^2$$

14. Area available in selected unit:

A = (0.2618) (78) (15.5) = 316 ft² net

$$U = \frac{680{,}000}{(316)(10.8)} = 199 \text{ Btu/hr (ft}^2\text{)(°F)}$$

15. Safely factor or percent excess surface:

$$\% = \frac{(316 - 304)(100)}{304} = 3.9\%$$

16. Pressure drop, tube side:

End return loss (from Figure 10-139) at 4.8 ft/sec = (0.63) (4 passes) = 2.52 psi
For the tube pressure drop, use Figure 10-138:

$$\text{At water rate} = \frac{68{,}000 \text{ lb/hr}}{19 \text{ tubes/pass}} = 3{,}580 \text{ lb/hr (tube)(pass)}$$

chart reads $\Delta p_t = 6.2$ psi/100 ft

$$\text{Total exchanger } \Delta p_t = \left(\frac{6.2}{100}\right)(4 \text{ passes})(16 \text{ ft tubes})$$
$$= 3.87 \text{ psi}$$

For usual purposes the effect of water temperature is not great, and Figure 10-138 can be used. The preceding value checks Stoever[109] with $\Delta p_t = 3.85$ psi.

Total tube side $\Delta p_t = 2.52 + 3.87 = 6.39 \text{ psi} \approx 6.4$ psi
Allow: 8 psi

Total shell side Δp_s (refer to the pressure drop section of this chapter) can be neglected for pressure units unless an unusual condition or design exists. To check, follow procedure for unbaffled shell pressure drop.

17. For the specification sheet, see Figure 10-72.

18. *Winter operation*—In order not to allow the water velocity in the tubes to fall below 3 ft/sec in the winter, you may have to compromise with the selected unit as based on 90°F water. If the average four-month winter temperature drops to 70°F, the quantity of water required will be reduced as will the velocity through the tubes. The low velocity is the point of concern. Check to determine the prevailing conditions.

LMTD:

$$\begin{array}{ccc} 106.6 & \longrightarrow & 106.6 \\ \underline{80} & \longleftarrow & \underline{70} \\ 26.6 & & 36.6 \end{array}$$

LMTD = 31.5°F

This assumes the same quantity of water in order to keep the velocity the same.
Revised tube side film coefficient:

At 75°F, $h_i = 1{,}025$

Corrected:

$$h_{io} = (1{,}025)(0.96)(.782/1.0) = 768 \text{ Btu/hr (ft}^2\text{)(°F)}$$

Assume the shell-side film coefficient unchanged, because the previous selection was on conservative side.

$$U = \frac{1}{\dfrac{1}{1{,}500} + 0.001 + 0.002 + \dfrac{1}{660}} = 201 \text{ Btu/hr (ft}^2\text{)(°F)}$$

Revised area required:

$$A = \frac{680{,}000}{(201)(31.5)} = 107 \text{ ft}^2$$

This is a significant reduction in area and is primarily due to the increased Δt during winter operation. Actually, the unit will subcool the condensate with the excess surface during the winter. Note that this result is based on maintaining the same water quantity through the tubes. If a lower velocity is acceptable for the water conditions, then a higher temperature rise can be taken, which reduces the liquid subcooling. Very few waters are acceptable for cooling without excessive scaling when the velocity falls below 1 ft/sec.

19. Although the previously discussed unit will perform as required, it may be larger than necessary. The water velocity of 4.8 ft/sec is not as high as would be preferred.
20. Redesign, using $^3/_4$-in. bimetal tubes.
Try for a minimum tube velocity of 6.5 ft/sec:

gpm = (136)(10/5) = 272 gpm
ft^3/sec. = 0.606
Water flow area = 0.606/6.5 = 0.0934 ft^2

For $^3/_4$-in. tubes with an I.D. equivalent to about a 12 BWG tube:

Tube flow area = 0.223/144 = 0.00155 ft^2/tube
Number of tubes = 0.0934/0.00155 = 30.1 tubes/pass

Assumed unit for trial conditions:

$$A = \frac{680{,}000}{(200)(15°)} = 226 \text{ ft}^2 \text{ (using 15° } \Delta t \text{ for estimating)}$$

Outside area/tube = 0.1963 ft^2
Number of feet tube = 226/0.1963 = 1,550 ft
Using 16-ft tubes:
Number required = 1,150/16 = 72
For a 4-pass, fixed tubesheet unit, $^3/_4$-in. tubes on $^{15}/_{16}$-in. triangular pitch, shell I.D. = 12-in., number of tubes = 84:
No. tubes/pass = 84/4 = 21
Flow area/pass = (21) (0.00155) = 0.0326 ft^2
ft^3/sec for 6.5 ft/sec = (6.5) (0.0326) = 0.212
gpm = 0.212 (7.48) (60) = 95.2
lb/hr = (95.2) (8.33) (60) = 47,700
For the film coefficient, tube side, use a mean water temperature of 85°F instead of 95°F. This will lean a little more to the winter operation and be safe for summer.
$h_i = 1{,}400$
correction for $^3/_4$-in. tube = 1.15

$$h_{io} = (1{,}400)(1.15)\left(\frac{0.532}{0.75}\right) = 1{,}140 \text{ Btu/hr (ft}^2\text{)(°F)}$$

Film coefficient, shell side:

$$G_o'' = \frac{1{,}440}{15.5(84)^{2/3}} = 4.84 \text{ lb/hr (lin ft)}$$

DWG. NO. A

Item No. E-1

EXCHANGER RATING

By [illegible]

Job No.

Date:- 5-18-58

Charge No.

Apparatus Overhead Condenser Col. T-12 Plant Ammonia Clean-Up

Min. Req. Eff. O. S. Area Exposed in Shell, Sq. Ft. ______ Outside 304 Inside: –

Number of Units: Operating 1 Spares none

DESIGN DATA PER Unit

UNIT DATA		SHELL SIDE	TUBE SIDE
Fluid		Ammonia	Brackish Water
Fluid Flow	Lbs./Hr.	1440	68,000 (136 gpm)
Temperature In	°F.	106.6	90
Temperature Out	°F.	106.6	100
Operating Pressure	PSIG	220	60
Density	Lbs./CF	37.0	62.4
Specific Heat	Btu/Lb./°F.		1.0
Latent Heat	Btu/Lb.	470.5	
Therm. Cond.	Btu/Hr./Sq. Ft./°F-/Ft.	0.29	
Viscosity	Centipoise	0.085	
Molecular Weight		17	
No. of Passes		1	4
Pressure Drop	PSI	Calc: Neglect Used: None	Calc: 6.4 Used: 8
Fouling Factor		0.001	0.002

Heat Transferred - BTU/Hr: 680,000

LMTD 10.8 °F. Overall U: Calc. 207 Used 199

CONSTRUCTION

	SHELL SIDE	TUBE SIDE
Max. Oper. Pressure	250 PSI	100 PSI
Max. Oper. Temperature	450 °F.	450 °F.

Type of Unit Fixed Tube Sheet Tube Pitch 1¼" △ Joint Roll and flare

Tubes - Material: Duplex * No. (Approx.) 78 O.D. 1" BWG * Length 16'-0"

Shell - Material: Steel Diameter (Approx.): 15¼" I.D.

Channel Material: Steel Supports Material: Steel

Tube Sheet Material: Steel with ¼" Monel Clad Baffle Material: Steel

Corrosion Allowance - Shell Side 1/16" Tube Side 1/16"

Connections - Shell In: 3" Out 3" Flange 300 psi Raised Face

Channel In: 4" Out 4" Flange 150 psi Raised Face

Others Vents and Drains Size 1" Flange 3000 psi Coupling

Bolts: Alloy Steel Gaskets: Asbestos impregnated

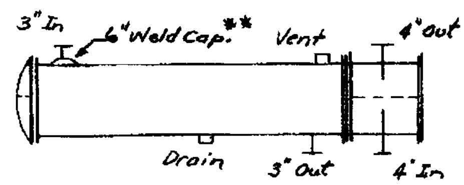
Code TEMA-Cl. C Stamp Yes X-Ray No SR No

Insulation None Class – Cathodic Protection No

Tube Sheet Layout standard for 4 pass tube side, quadrant design.

No baffles required; Provide tube supports on maximum spacing.

BAFFLE ARRANGEMENT



** With internal impingement plate having ½" ϕ holes with free area at least 0.5 Cross-Sect.

NOZZLE ARRANGEMENT

Remarks:- * Bi-metal tube 18 BWG Steel outside and 16 BWG Cupro-Nickel inside

Checked ______ Date ______ Approved ______ Date ______

Rev. ______ By ______ Date ______ B/M No. ______

Figure 10-72. Exchanger rating for overhead condenser.

Because the value of h_o read from Figure 10-67A is still about 2,500, use $h_o = 1500$.
Overall coefficient:

$$U = \frac{1}{\frac{1}{1500} + 0.001 + 0.002 + \frac{1}{1140}}$$

$$= 220 \text{ Btu/hr (ft}^2\text{)(°F)}$$

Water flow of 95.2 gpm gives:

$$\text{Water temperature rise} = \frac{680{,}000 \text{ Btu/hr}}{(1)(95.2)(8.33)(60)} = 14.3°\text{F}$$

Log mean temperature difference:

Summer: based upon water 90°F → 104.3°F
LMTD = 7.2°F as calculated previously
Winter: based upon water 70°F → 84.3°F
LMTD = 29.0°F as calculated previously

Area required:

Summer: $$A = \frac{680{,}000}{(220)(7.2)} = 430 \text{ ft}^2$$

Winter: $$A = \frac{680{,}000}{(220)(29.0)} = 106 \text{ ft}^2$$

Area available in selected unit:

$$A = (0.1963)(84)(15.5) = 256 \text{ ft}^2$$

To make this unit acceptable for summer operation (the calculated required surface is greater than that available), assume that the water rate can be increased, thereby decreasing water ΔT and increasing LMTD.

$$\begin{array}{ccc} 106.6 & \longrightarrow & 106.6 \\ \underline{95} & \longleftarrow & \underline{90} \\ 11.6 & & 16.6 \end{array}$$

LMTD = 14°

Summer area required (not making any correction for change in water film coefficient or condensing coefficient):

$$A = 286\left(\frac{7.2}{14}\right) = 147$$

$$\text{Water rate} = \frac{680{,}000}{(5°)(1)} = 136{,}000 \text{ lb/hr}$$

$$\text{gpm} = \frac{136{,}000}{(8.33)(60)} = 272$$

$$\text{ft}^3\text{/sec} = \frac{272}{(7.48)(60)} = 0.606$$

$$\text{ft/sec} = \frac{0.606}{(0.00155)(21)} = 18.6$$

This velocity is too high for satisfactory operation. Therefore, the only way to get more flow area is more tubes and requires the same size shell as the previously designed unit. I recommend using the unit with 1-in. tubes.

21. Nozzles should be sized with or checked against the sizes of the incoming or outgoing lines. Often the exchanger nozzle must be larger than the pipe in order to keep velocities low to prevent erosion or high pressure drop.

Water Connections

Flow rate = 136 gpm
Design for (136) (1.5) = 204 gpm
Maximum allowable velocity 6 ft/sec
Referring to Cameron hydraulic tables:
Select 4-in. nozzle, vel = 5.3 ft/sec
Select head loss = (0.046 ft) (6 in./12) = 0.023 ft liquid

Condensed Ammonia Liquid Out

Referring to Cameron Miscellaneous Liquid Table in Fluid Flow Chapter, Vol. I:

$$\text{gpm} = \frac{1{,}440 \text{ lb/h.}}{(60)(8.33)(0.59)} = 4.87$$

Design rate (4.87) (1.5) = 7.8 gpm

Select 3-in. nozzle, head loss less than 0.00035 ft (negligible). Use large nozzle to ensure free drainage of unit and no vapor binding in outlet line. Actually a 1-in. connection would safely carry the liquid flow with a head of about 0.08 ft of liquid. A condenser must be free draining and capable of handling surges.

Ammonia Vapor Inlet

Design rate = (1440) (1.5) = 2160 lb/hr

Referring to Figure 10-63, at 220 psia and 17 mol wt, the maximum suggested vapor velocity through a nozzle is

(40) (1.2) = 48 ft/sec max.

For a 3-in. nozzle, Schedule 40,

Cross-section area to flow = 0.0513 ft^2
Sp. Vol NH_3 vapor = 1.282 ft^3/lb
Total flowing ft^3/hr = (2,160) (1.282) = 2,770

$$ft^3/sec = \frac{2770}{3600} = 0.8 \text{ cfs}$$

Velocity in 3-in. pipe:

$$\frac{0.80}{0.0513} = 15.6 \text{ ft/sec}$$

This is satisfactory, although a 2-in. nozzle would have a velocity of 34.3 ft/sec. Because this condenser has entering vapors at the dew point, entrainment of some particles is always a real possibility; therefore, a low inlet velocity is preferred. Also, overhead vapor lines should have low pressure drop for vapor at its dew point and a 3-in. line might be indicated when this line is checked.

RODbaffled® (Shell-Side) Exchangers

(See Figures 10-20A, 10-20-B, 10-20-C, and 10-20-D.)

The design techniques of this system of baffling are not adequately published in literature, because they are proprietary information of Phillips Petroleum Co. and are licensed to design firms for specifically designing the units to ensure proper application details. Figures 10-73 and 10-74 illustrate some of the improved features claimed for heat transfer and pressure drop.

The specific features of this new type of shell-side construction provide improved resistance to destructive tube vibration compared to the usual plate baffle designs. These units have been applied in practically all of the usual process heat exchanger services including externally finned tubes. These units have been fabricated in large shell diameters. Technical references include Gentry;[169] Gentry, Young, and Small;[170] and Gentry and Small.[171]

Condensation Inside Tubes

Horizontal Tube Bundles[70, 82, 94]

Kern's[70, 94A] modification of the Nusselt development is considered useful.

$$h_c = 0.761 \frac{[Lk_l^3\rho_l(\rho_l - \rho_v)g]^{1/3}}{[W_T\mu_l]} = 0.612 \frac{[k_l^3\rho_l(\rho_l - \rho_v)g]^{1/4}}{[\mu_l D_i \Delta t]} \quad (10\text{-}103)$$

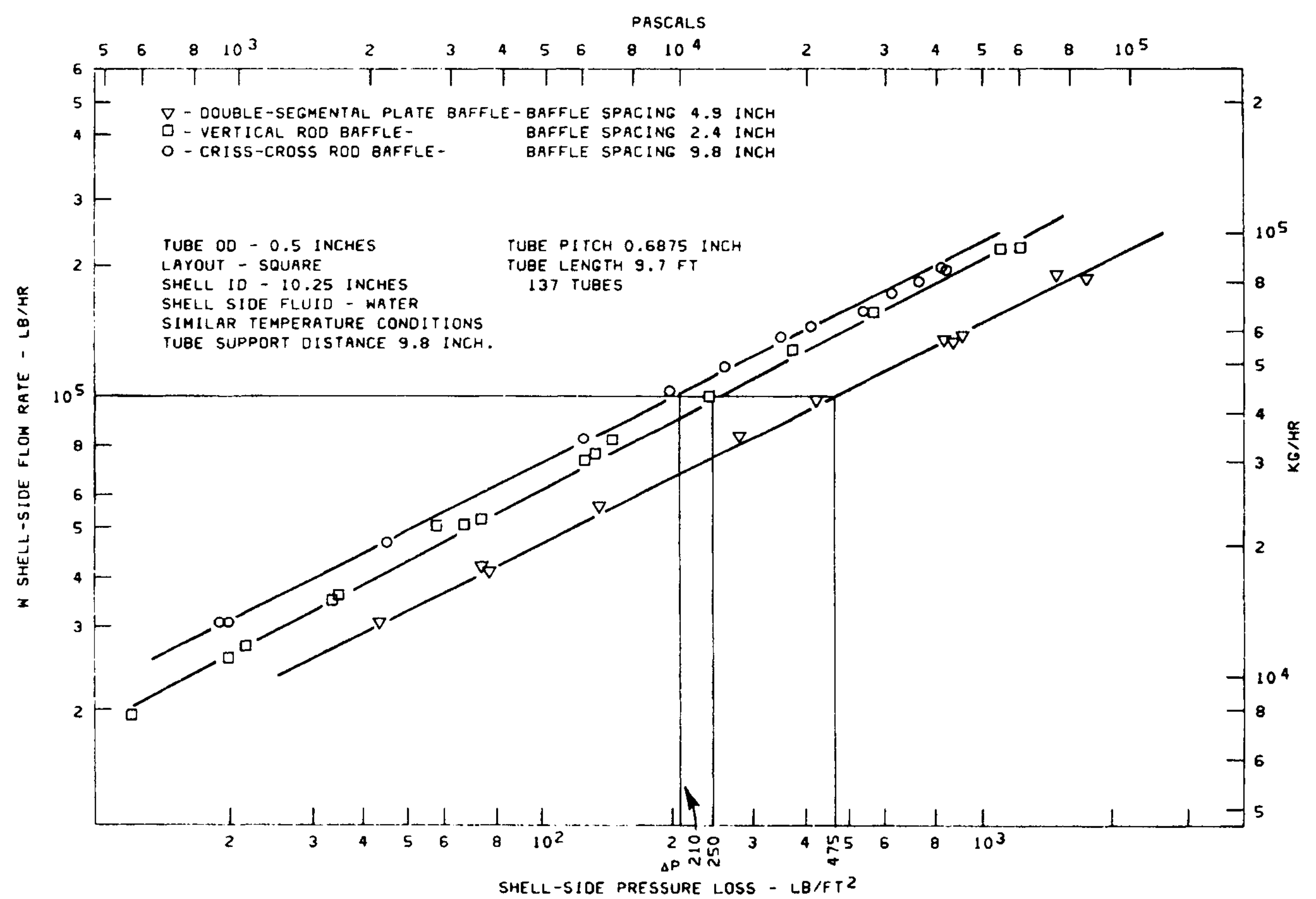


Figure 10-73. Shell-side pressure loss for 3 shell-side baffle configurations—RODbaffles®. (Used by permission: Small, W. M., and Young, R. K. *Heat Transfer Engineering*, V. 2, ©1979. Taylor and Francis, Inc., Philadelphia, PA. All rights reserved.)

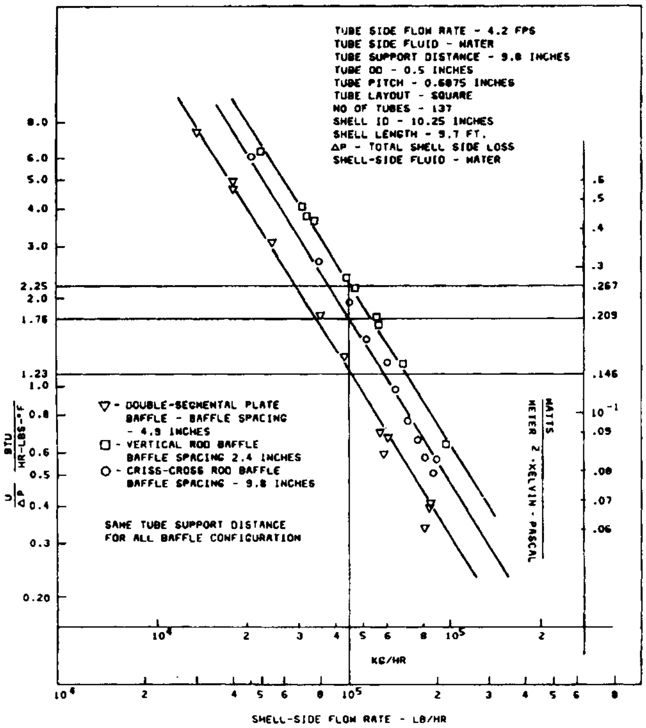


Figure 10-74. Ratio of heat transfer to pressure loss for 3 shell-side configurations—RODbaffles®. (Used by permission: Small, W. M., and Young, R. K. *Heat Transfer Engineering,* V. 2, ©1979. Taylor and Francis, Inc., Philadelphia, PA. All rights reserved.)

where

k_l = liquid thermal conductivity, Btu/(hr) (ft^2) (unit temperature gradient, °F/ft)
ρ_l = liquid density, lb/ft^3
ρ_v = vapor density, lb/ft^3
μ_l = liquid viscosity, lb/(hr) (ft), [= centipoise × 2.42 = lb/(hr) (ft)]
D_i = inside diameter, ft
g = acceleration of gravity, 4.18×10^8 ft/(hr)^2
Δt = temperature difference = $(t_{sv} - t_s)$ °F
t_s = surface temperature, °F
t_{sv} = saturated vapor temperature, °F
h_c = condensing film coefficient, mean, Btu/(hr) (ft^2) (°F)
W_T = total vapor condensed in one tube, lb/hr
L = tube length, ft (effective for heat transfer)

Because the condensate builds up along the bottom portion of horizontal tubes, the layer builds up thicker and offers more resistance to heat transfer. Kern[70] proposes good agreement with practical experience using the following

$$G'' = W/(0.5\,LN_t), \text{ special } G'' \text{ loading for a single horizontal tube, lb/(hr) (ft)} \quad (10\text{-}104)$$

$$h_c = \frac{[(\mu_f^2)]^{1/3}}{[(k_f^3\rho_f^2 g)]} = 1.51\,\frac{[(4G'')]^{-1/3}}{[\mu_f]} \quad (10\text{-}105)$$

where

N_t = number of effective tubes for condensation
L = tube length, ft
W = condensate flow rate, lb/hr

Other symbols as listed previously.

Subscript:

f = liquid film

These relations are good for single-pass tube side units; however, for multipass units, the number of available vapor tubes must be determined at the end of the first and each succeeding pass, as the lower liquid carrying tubes must not be considered as available tubes. Thus, G'' should be evaluated for each pass, and the individuals evaluated separately, or an average determined as the average of the pass average values of h_{cm}.

Condensing Inside Horizontal Tubes

The correlation of Akers, et. al.,[1] has given good results in some industrial designs. The authors report that some vertical and inclined tube data is also correlated on the same basis. The sharp break in the data occurs around a Reynolds number of 5×10^4 as shown in Figure 10-75. The mass flow rate used to correlate is the arithmetic average of inlet and outlet liquid condensate and vapor flows:

$$G_e = \bar{G}_L + \bar{G}_g\,(\rho_L/\rho_g)^{1/2} \quad (10\text{-}106)$$

where

G_e = equivalent mass flow inside tubes, lb/hr (ft^2 of flow cross section)
$\bar{G}_L$ and $\bar{G}_g$ = arithmetic averages of condensate and vapor flow respectively, lb/hr (ft^2 of flow cross section)

The relation applies to systems that potentially are condensable as contrasted to those systems containing noncondensable gases such as air, nitrogen, etc. All of the vapor does not have to be condensed in the unit for the correlation to apply.

Subcooling Condensate in Vertical Tubes

The total unit size is the sum of the area requirements for condensation plus subcooling of the liquid to the desired outlet temperature. For the subcooling portion:

1. McAdams[82] recommends:

$$\frac{h_{cm}}{\sqrt{k_f^3\rho_f^2 g/\mu_f^2}} = 0.01\left(\frac{c\mu}{k}\right)^{1/3}\left(\frac{4W}{\mu_f \pi D_i}\right)^{1/3} \quad (10\text{-}107)$$

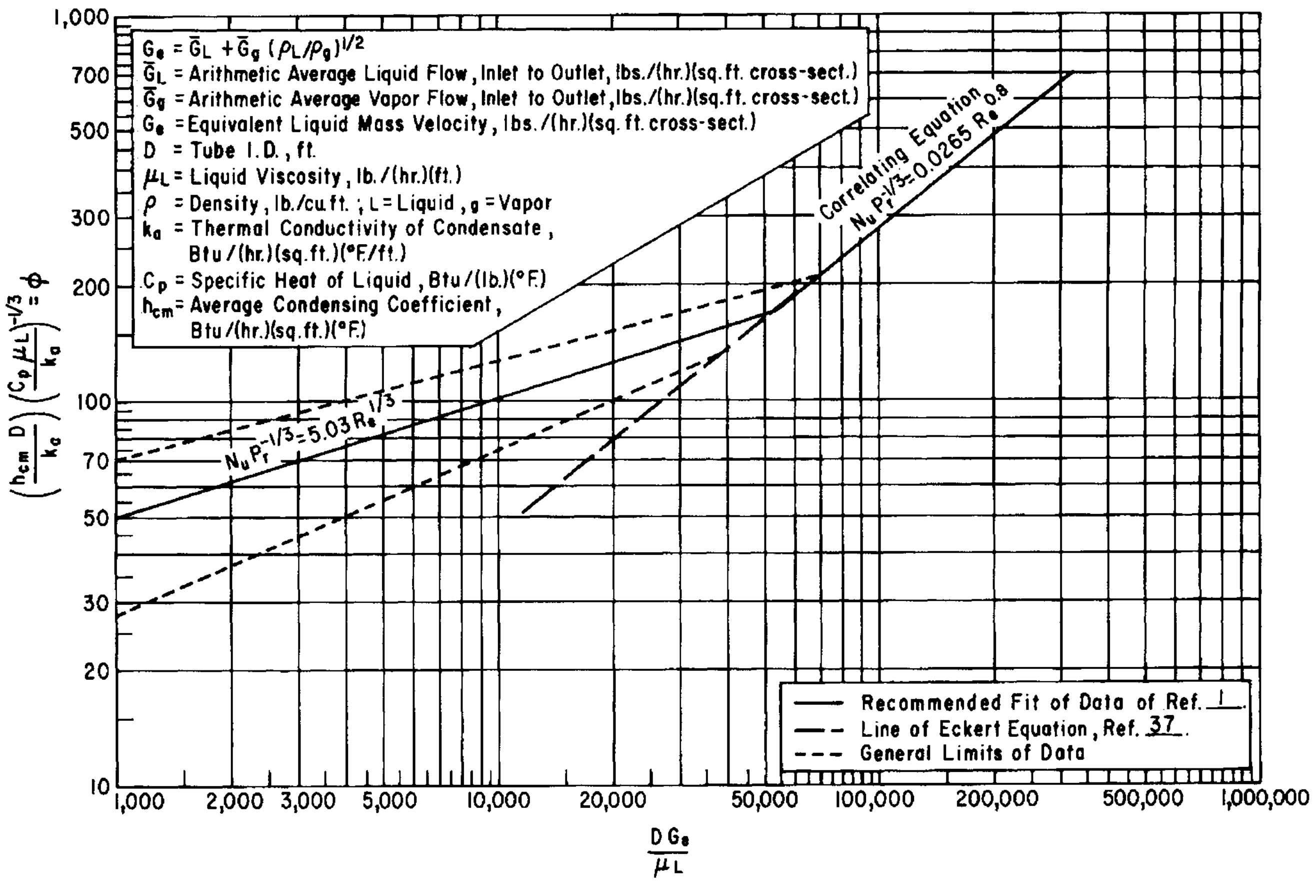


Figure 10-75. Condensing inside horizontal tubes. (Used by permission: Akers, W. W., Deans, H .A., and Crosser, O. K. *Chemical Engineering Progress,* V. 55, No. 29, ©1959. American Institute of Chemical Engineers. All rights reserved.)

2. The mean temperature of condensate film before subcooling:[82]

$$t_m = t_{sv} - 3\,(t_{sv} - t_w)/8 \qquad (10\text{-}108)$$

where

t_{sv} = temperature of saturated vapor
t_w = temperature of surface

Vertical Tube Bundles, Single Pass Downward

Figure 10-76 is the semi-empirical curve of Colburn[30] as recommended by Kern.[70]

Upward flow should be avoided as the film coefficient falls considerably below the value for the downward flow; however, see later section for details.

The condensation inside vertical tubes is similar as to mechanisms for condensation outside each tube. As the condensate flows down the tube (inside), the liquid film becomes thicker and normally changes from streamline to turbulent flow at some region between the top and bottom of the tube. Accordingly, the local film coefficient decreases until the fluid film becomes turbulent, and then the film coefficient increases. Colburn[30, 70] has indicated that at a point on Figure 10-76 where $4G'/\mu_f = 2{,}100$ the transition occurs, and the mean coefficient for condensation inside a vertical tube when $4G'/\mu > 2{,}100$ satisfies the entire tube. Nusselt's recommendation is $4G'/\mu_f = 1{,}400$.[70]

The distance from the top of the tube to the transition region is expressed[70]:

$$x_c = \frac{2{,}668\lambda\mu_f^{5/3}}{\rho^{2/3} k g^{1/3}(T_v - t_w)} \qquad (10\text{-}109)$$

where

units are previously listed

T_v = temperature of vapor, °F
t_w = temperature of tube wall, °F
x_c = distance from top (effective) of tube, ft

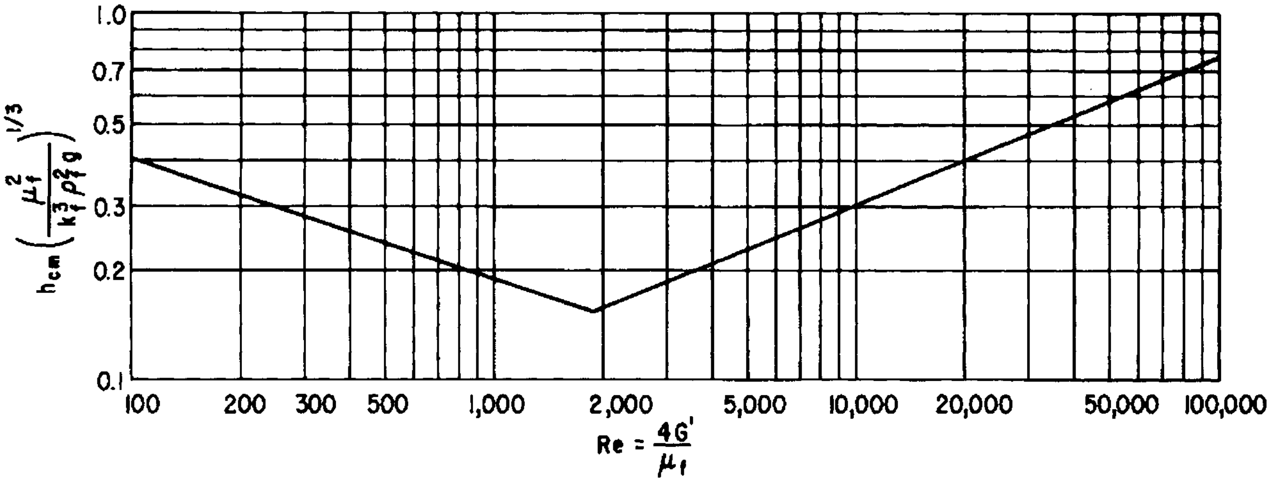


Figure 10-76. Condensation down-flow in vertical tubes. Note: h_{cm} = average value of condensing coefficient between two points; G′ = condensate loading, lb/(hr)(ft) = w′ / P, lb/(hr)(lin. ft); w′ = W/N_t, lb/(hr)(tube); W = condensate, lb/hr; p = perimeter, ft per tube. (Used by permission: Colburn, A. P. *Transactions of American Institute of Chemical Engineers,* V. 30., ©1934. American Institute of Chemical Engineers. All rights reserved.)

Condensing Single Pass Up-Flow in Vertical Tubes

This mechanical configuration is not the usual situation for most vapor condensers; however, it is convenient for special arrangements and in particular to mount directly above a boiling vessel for refluxing vapors. It can also be used in special designs to take very hot vapors and generate steam; however, for all cases a very real limitation must be recognized.

Clements and Colver[29] developed the modified Nusselt equation to correlate hydrocarbon and hydrocarbon mixtures in turbulent film condensation:

$$\frac{h_x x}{k_l} = Nu_x = 1.88 \times 10^{-8}\left[\frac{x^3 g \rho_l \lambda}{\mu_l k_l \Delta T}\right]^{0.75} \quad (10\text{-}110)$$

with an average deviation of date of 35.7%,

where

x = distance film has fallen
g = gravitational constant
ρ_l = liquid density
λ = latent heat of vaporization
μ = liquid viscosity
k = liquid thermal conductivity
ΔT = temperature difference = $(T_{bubble\ point} - T_{surface})$
Nu_x = local Nusselt number, $h_x x/k_l$
h_x = local heat transfer coefficient

Note: The inner wall temperature data agreed with the bubble point temperature.

Figures 10-77 and 10-78 illustrate the test performance data, which is valuable in understanding the mechanism.

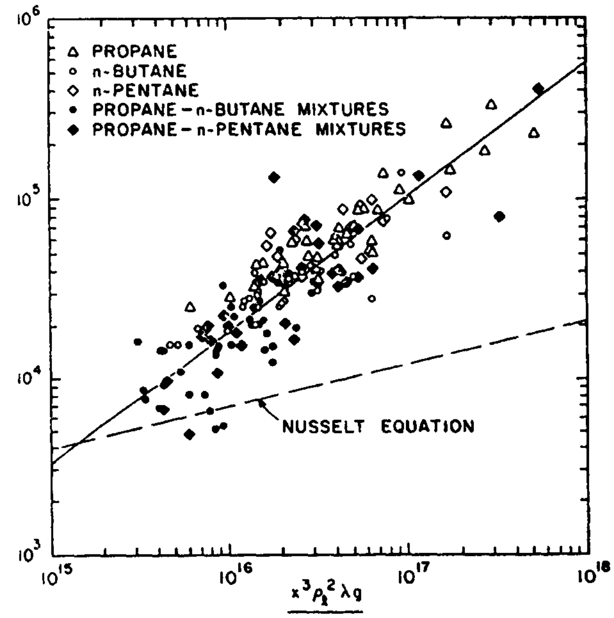


Figure 10-77. Turbulent film condensation of light hydrocarbons and their mixtures—up-flow. (used by permission: Clements, L. D., and Colver, C. P. *AIChE Heat Transfer Symposium* V. 131, No. 69, ©1973. American Institute of Chemical Engineers. All rights reserved.)

Flooding in an up-flow in a vertical condenser is an important design consideration, because this flooding poses a limit on flows for any selected design. To select the number of tubes required to obtain the area for up-flow without flooding, the diameter of the tubesheet to hold these tubes becomes quite large. The selected number of tubes, to

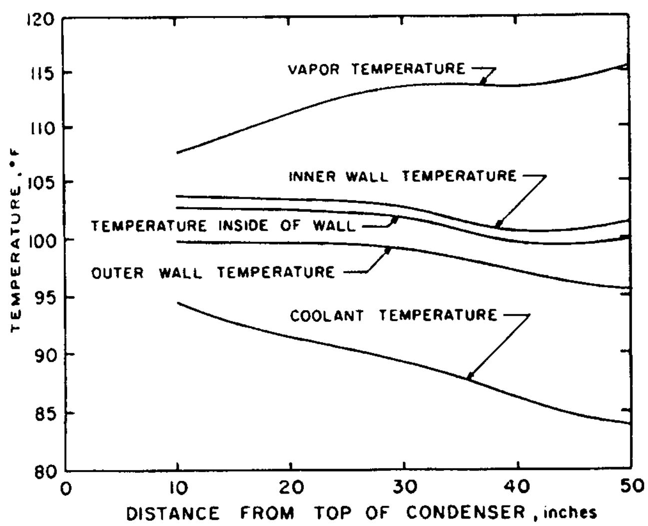


Figure 10-78. Typical condenser temperature profiles for 43% propane–57% *n*-butane mixture at 176 psi abs.—up-flow. (Used by permission: Clements, L. D., and Colver, C. P. *AIChE Heat Transfer Symposium*, V. 131, No. 69, ©1973. American Institute of Chemical Engineers. All rights reserved.)

obtain the actual heat transfer required, may dictate very short tubes, such as 2- or 3-ft long. This is an unrealistic design. Therefore, tube size may change the balance for the design, or it may be impractical, and a down-flow unit may be more economical.

For a given size vertical condenser in up-flow, the lightest liquid and gas rates occur at the entrance to the tubes; therefore flooding begins at this location. Some advantages exist for particular applications, including (a) mounting vertically over refluxing equipment as it can save a separator and instruments, (b) often lower fouling rates for the tube side due to the liquid washing effect, and (c) fractional condensation of multicomponent mixture allowing lighter components to flow out vertically. According to English et. al.[42] the correlation for the flooding condition is:

$$G = 1550\, D^{0.3}\, \rho_l^{0.46}\, \sigma^{0.09}\, \rho_G^{0.5} / \mu_l^{0.14}\, (\cos \Theta)^{0.32}\, (L/G)^{0.07} \quad (10\text{-}111)$$

where

D = tube inside diameter, in.
G = superficial vapor mass flow rate, lb/hr ft^2 (total vapor entering base of vertical condenser tube)
L = superficial liquid mass flow rate, lb/hr ft^2 (liquid leaving the base of the condenser tube, plus entrained liquid)
μ_l = liquid viscosity, centipoise
ρ_G = gas density, lb/ft^3
ρ_l = liquid density, lb/ft^3
σ = surface tension, dynes/cm
Θ = tube-taper angle (measured from horizontal), degrees

According to this investigation, the allowable gas rate at flooding can be increased by having the outlet tube ends extend through the bottom tubesheet and be cut off at an angle to the horizontal, rather than just a "square" cut-off. The angle measured from the horizontal for a vertical tube is as follows:

Angle	% Increase* in Maximum Allowable Gas Rate
30°	5
60°	25
75°	54

*Increase compared to square end tubes $\Theta = 0°$

The studies of Diehl and Koppany[35] further examined vertical up-flow limitations.

The critical diameter above which the flooding velocity is independent of diameter is given by:

$$d_{ic} = \frac{\sigma}{80}, \text{ in.} \quad (10\text{-}112)$$

where

d_i = inside tube diameter
subscript c = critical condition
σ = surface tension of liquid, dynes/cm

For example, consider Dowtherm at 20 in. Hg vacuum:

σ = 20.8 dynes/cm

$$d_{ic} = \frac{20.8}{80} = 0.26 \text{ in.}$$

Now, at 20 psig, σ = 13.05 dynes/cm

$$d_{ic} = \frac{13.05}{80} = 0.16 \text{ in.}$$

Therefore, for flooding in vertical tubes for a range of these conditions, the tube I.D. must be greater than 0.26 in.; generally, the recommendation is to use 0.5–1.0-in. I.D. tubes, approximately, to move far enough away from the critical condition.

Flooding correlation (no tapered inlet tube considered):

$$V_f = F_1 F_2 \left(\frac{\sigma}{\rho_g}\right)^{0.5}, \text{ for } F_1 F_2 \left(\frac{\sigma}{\rho_g}\right)^{0.5} > 10 \quad (10\text{-}113)$$

For, $F_1 F_2 \left(\frac{\sigma}{\rho_g}\right)^{0.5} < 10$

$$V_f = 0.71 \left[F_1 F_2 \left(\frac{\sigma}{\rho_g}\right)^{0.5}\right]^{1.15} \quad (10\text{-}114)$$

where

$$F_1 = \left[d_i / \left(\frac{\sigma}{80}\right)\right]^{0.4}, \text{ for } d_i / \left(\frac{\sigma}{80}\right) < 1.0$$

$$F_1 = 1.0, \text{ for } d_i / \left(\frac{\sigma}{80}\right) \geq 1.0$$

$$F_2 = (L/G)^{-0.25}$$

By assuming no entrainment in each condenser tube, the liquid rate out the tube must equal the vapor rate entering the tubes (assuming no noncondensables), so L/G = 1, and at steady state, $F_2 = 1$.

Using the Dowtherm figures cited previously at 20-in. Hg. vacuum,

$\sigma = 20.8$ dynes/cm

$\rho_g = 0.0877\ \mathrm{lb/ft^3}$

$$\left(\frac{\sigma}{\rho_g}\right)^{0.5} = 15.4$$

$$V_f = F_1F_2\left(\frac{\sigma}{\rho_g}\right)^{0.5}$$

$$V_f = (1)(1)(15.4) = 15.4\ \mathrm{ft/sec}$$

This is the velocity of the vapors in the tube, which will result in flooding at this low pressure.

For the condition of 20 psig pressure:

$\sigma = 13.05$ dynes/cm

$\rho_g = 0.5587\ \mathrm{lb/ft^3}$

$$\left(\frac{\sigma}{\rho_g}\right)^{0.5} = \left(\frac{13.05}{0.5587}\right)^{0.5} = 4.83$$

$$F_1 = F_2 = 1$$

Because:

$$\left(\frac{\sigma}{\rho_g}\right)^{0.5} < 10$$

$$V_f = 0.71\left[F_1F_2\left(\frac{\sigma}{\rho_g}\right)^{0.5}\right]^{1.15}$$

$$V_f = 4.35\ \mathrm{ft/sec}$$

By comparing with solving for the same conditions using a 1-in. tube in the English[42] correlation at 20-in. Hg vacuum, $V_f = 14.02$ ft/sec, compared to 15.4 ft/sec from the preceding calculation.

English's[42] flooding correlation incorporates an entrainment load of E/G from 0.01–0.05 lb liquid per lb of vapor.

The effect of the tapered inlet tube (as earlier discussed) as now determined by English[42] is only significant at the 60° and 75° tapers, both producing about the same increase in vapor capacity. Diehl's correlation is shown in Figure 10-79.

where

- E = superficial liquid entrainment rate, lb/hr/ft²
- F_1, F_2 = correlation factors defined by equations
- V_f = superficial flooding velocity of the vapor, ft/sec
- V_{f7} = superficial flooding velocity of the vapor when inlet tube taper is 70°, ft/sec

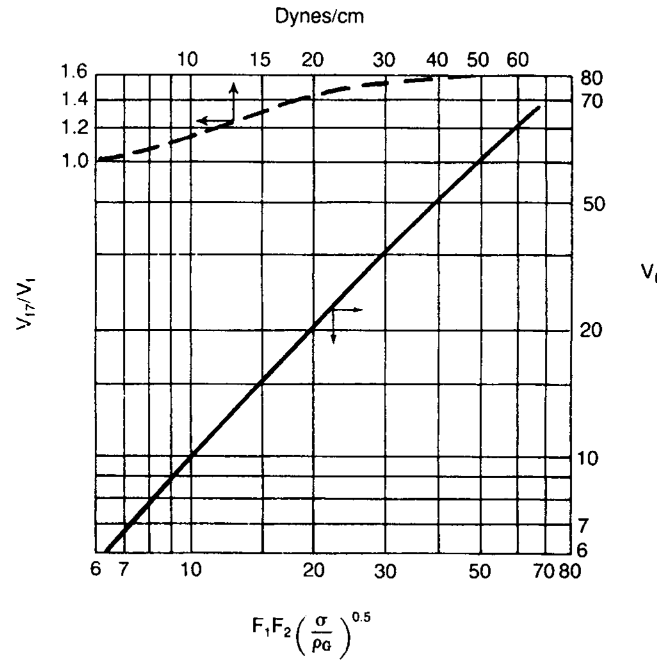


Figure 10-79. Effect of entrance tube taper on flooding velocity. (Used by permission: Diehl, J. E., and C.R. Kop Company, *Chemical Engineering Progress Symposium, Heat Transfer*, V. 65, No. 92, ©1968. American Institute of Chemical Engineers. All rights reserved.)

Example 10-11. Desuperheating and Condensing Propylene in Shell

See Figure 10-80.

A refrigeration system requires that 52,400 lb/hr of propylene refrigerant vapor from the compressors be desuperheated and then condensed.

Propylene inlet: 265 psia and 165°F
Propylene dew point: 265 psia and 112°F
Cooling water in: 90°F

Assume that this load can be handled best in two units operating in parallel. In this scenario, if one condenser develops trouble, the entire refrigeration system, and consequently the plant process, is not shut down.

Heat Duty

Heat content of propylene vapor at 165°F = 512 Btu/lb
Heat content of propylene vapor at 112°F = 485 Btu/lb
Sensible heat duty = 52,400 (512 − 485) = 1,415,000 Btu/hr
Latent heat of vaporization at 112°F = 126 Btu/lb
Latent heat duty = 52,400 (126) = 6,600,000 Btu/hr

SPEC. DWG. NO.
A-

Job No.____________________

B/M No.____________________

EXCHANGER RATING & SPECIFICATIONS

Page 1 of 1 Pages	
Unit Price	
No. Units	2
Item No.	

Service Propylene Condensing
Min. Req. Eff. Outside Area Exposed In Shell, Sq. Ft. 1960
Number of Units: Operating 2 in parallel Spares None

DESIGN DATA PER Unit

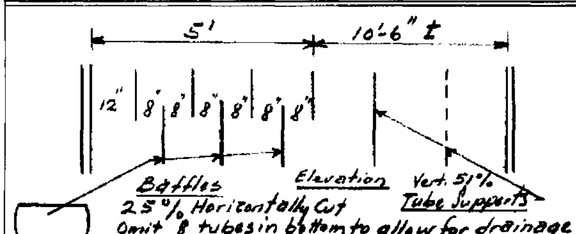
UNIT DATA		SHELL SIDE	TUBE SIDE
Fluid		Propylene	Sea water
Fluid Flow	Lbs./Hr.		
Temperature In	°F.	165	90
Temperature Out	°F.	112	97
Operating Pressure	PSIG	250	60
Density	Lbs./CF	Vap. 2.2 Liq. 29.4	
Specific Heat	Btu/Lb./°F.	Vap. 0.55 Liq. not used	
Latent Heat	Btu/Lb.	126.4	
Therm. Cond.	Btu/Hr./Sq. Ft./°F./Ft.	Vap. 0.0128 Liq. 0.0725	
Viscosity	Centipoise	Vap. 0.0109 Liq. 0.087	
Molecular Weight		42	
No. of Passes		1	
Pressure Drop	PSI	Calc: 1.02 Used: 2	Calc: 7.75 Used: 7.75
Fouling Factor		Vap. 0.001 Liq. 0.0005	0.002

Heat Transferred - BTU/Hr. Cooling 707,500 Condensing 3,300,000
LMTD 36.2° and 19° °F. Overall U: Calc. 39.6 and 142 Used 31 and 131

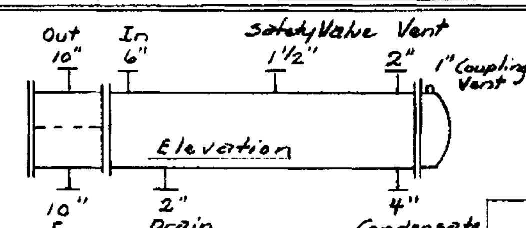
CONSTRUCTION

	Shell	Tube
Max. Oper. Pressure	300 PSIG	100 PSIG
Max. Oper. Temperature	450 °F.	450 °F.

Type of Unit Fixed Tube Sheet Shell Dia: 29" I.D.
Tubes: No. (Approx.) 646-8 = 638 O.D. 3/4 BWG. 12 Length 16'-0" Pitch 1" △ Joint Roll & Bead
Material: Tubes Cupro-Nickel, 90-10 Hi Fe Tube Sheet Steel, Monel 1/4" Minimum Clad
Channel Steel Baffle Steel Corr. Allow. 1/16"
Shell Steel Baffle Steel Corr. Allow. 1/16"
Connections - Shell In: 6" Out 4" Flange 300# R.F.
Channel In: 10" Out 10" Flange 150# R.F.
Others: Safety, (Vent, drain) Size 2 Out x 1 1/2" In Flange 300# R.F.
Bolts: Alloy steel Gaskets: Impregnated asbestos
Code ASME Stamp Yes TEMA Class C X-Ray No S.R. No
Insulation None Class — Cathodic Protection Yes in channels



BAFFLE ARRANGEMENT



NOZZLE ARRANGEMENT

Remarks: Cut lower baffles to insure drainage, Remove 8 lower tubes.
Note: Top entrance of 6" Inlet vapor prevents use of vertical cut baffles which would allow drainage with out omission of 8 tubes.

BY	Chk'd.	App.	Rev.	Rev.	Rev.
Date					

P.O. To:____________________

Figure 10-80. Exchanger ratings and specifications for propylene condenser.

Water Required

Assume a 7°F rise in sea water temperature:

lb water/hr = (6,600,000 + 1,415,000)/(1) (7°) = 1,144,000

Water temperature at the dew point:

$$t = 90 + \frac{6{,}600{,}000}{(1{,}144{,}000)(1)} = 90 + 5.78 = 95.8°F$$

For log mean temperature differences, see Figure 10-73.

$$\text{LMTD desuperheating} = \frac{68 - 16.2}{2.3 \log \frac{68}{16.2}} = 36.2°F$$

$$\text{LMTD condensing} = 19 \circ F$$

Temperature difference correction for desuperheating section only; see Figure 10-30 for 1 shell pass, 2 or more tube passes.

$$P = \frac{t_2 - t_1}{T_1 - t_1} = \frac{97 - 95.8}{165 - 95.8} = 0.0173$$

$$R = \frac{T_1 - T_2}{t_2 - t_1} = \frac{165 - 112}{97 - 95.8} = 44.2$$

Note that in reading the chart, values are off the scales, but by approximate interpolation, a value of F = 1.0 is not unreasonable. In any case the error in using this value will be small.

No correction is necessary for the condensing section.

Assume overall U values to establish initial order-of-magnitude of area required:

Condensing U = 130
Desuperheating U = 35

$$\text{Area estimated} = A = \frac{6{,}600{,}000}{130(19)} + \frac{1{,}415{,}000}{35(36.2)}$$
$$= 2{,}670 + 1{,}118 = 3{,}788 \text{ ft}^2$$

Try two parallel units of approximately 1,894 ft^2 each.

Select:

$^3/_4$-in. O.D. tubes × 12 BWG cupro-nickel × 16 ft, 0-in. long

$$\text{No. tubes} = \frac{1{,}894}{(0.196)(15 \text{ ft long tubes})} = 645$$

Use two passes in tubes; tubes on 1-in. triangular pitch. From the tube count in Table 10-9, a 29-in. I.D. shell will hold 646 tubes, including an allowance for tie rods. The number of tubes per pass = 646/2 = 323.

Tube-side coefficient:

$$\text{water flow/unit} = \frac{8{,}015{,}000 \text{ Btu/hr total}}{(2)(7°F)} = 573{,}000 \text{ lb/hr}$$

gpm = 573,000/(8.33) (60) = 1,148
ft^3/sec. = 573,000/(62.4) (3,600) = 2.55

Flow area for water = (323) (0.233 in.^{2}/tube)/144 = 0.5 ft^2
Velocity of water through tubes = 2.55 cfs/0.5 = 5.1 ft/sec
From Figure 10-50A or 10-50B, for water,
h_i = 1,200 Btu/hr (ft^2) (°F)
Referenced to outside surface, tube I.D. = 0.532 in.

$$h_{io} = 1{,}200\left(\frac{0.532}{0.75}\right) = 850 \text{ Btu/hr (ft}^2\text{)(°F)}$$

Shell-side coefficient: condensing.
Assume $^2/_3$ of tube length is used for condensing ≅ 10 ft.
Referring to Figure 10-67,

Tube loading = G_o'' = $W/LN_t^{2/3}$ = (52,400/2)/(10) (646)$^{2/3}$
G_o'' = 26,200/(10)(74) = 35.4 lb/lin. ft
Propylene properties at 112°F (liquid):
Sp. gr. = 0.473
μ_f = 0.087 centipoise
k_f = 0.0725 Btu/hr (ft^2) (°F/ft)
Read, h_o = 320, use 300 Btu/hr (ft^2) (°F)

Overall U for condensing:
Assume: water side fouling = 0.002
propylene side fouling (oil) = 0.0005
neglect tube wall resistance

$$\frac{1}{U} = \frac{1}{300} + 0.0005 + 0.002 + \frac{1}{850}$$

$$\frac{1}{U} = 0.00333 + 0.0005 + 0.002 + 0.00117 = 0.0070$$

$$U = 142 \text{ Btu/hr (ft}^2\text{) (} \circ \text{F)}$$

$$\text{Condensing area} = A = \frac{(6{,}600{,}000/2)}{(142)(19)} = 1{,}220 \text{ ft}^2\text{/unit}$$

Shell-side coefficient: vapor desuperheating or cooling.
Tube length allowed for this = approximately 15 ft − 10 ft = 5 ft
Refer to Figure 10-81.
Assume a baffle cut of 25% and spacing as shown.
Note that allowance must be made for the entrance nozzle, which often means that baffles cannot be spaced too close to the tubesheet.
Tube bundle cross flow area:
a_s = (D_s) (c') (B)/144 (p)
c' = 0.25 in. between tubes
a_s = (29 in.) (0.25) (8 in.)/144 (1-in. pitch)
a_s = 0.403 ft^2
G_s = W/a_s = (52,400/2)/0.403 = 65,000 lb/hr (ft^2)
Vapor properties at 140°F:
c_p = 0.55 Btu/lb (°F)
k_a = 0.0128 Btu/hr (ft^2) (°F/ft)
μ' = 0.0109 centipoise
μ = 0.0109 (2.42) = 0.0264 lb/hr (ft)
Vapor density = 2.2 lb/ft^3
$(c_p\mu/k_a)^{1/3}$ = [(0.55) (0.0264)/0.0128]$^{1/3}$ = 1.042
The Reynolds number for Figure 10-54:
D_e = 0.73 in. ≅ (0.73/12) ft

$$R_e = D_eG_s/\mu = \frac{(0.73 \text{ in./12})(65{,}000)}{0.0264}$$
$$= 149{,}800$$

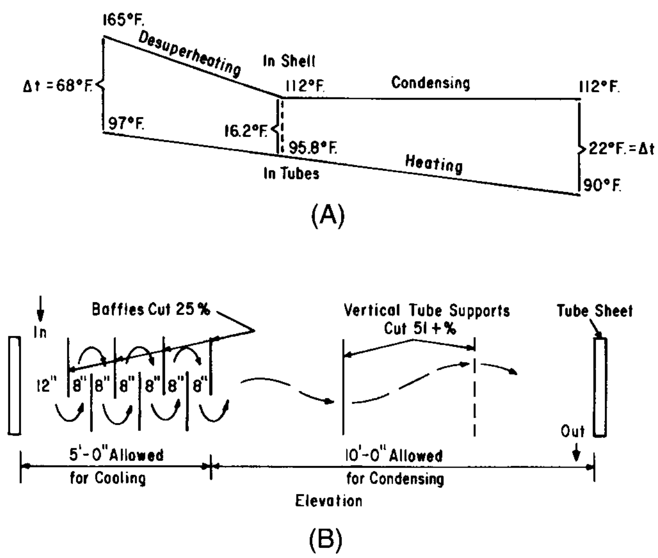


Figure 10-81. Illustration for Example 10-11. (A) Temperature profile for fluid desuperheating and condensing. (B) Baffle and tube support layout.

Reading Figure 10-54,

$j_H = 240$

$$h_o = \frac{j_H k}{D_e}\left(\frac{c_p\mu}{k_a}\right)^{1/3}\left(\frac{\mu}{\mu_w}\right)^{0.14}$$

μ/μ_w = approximately 0.5 as lowest ratio

$$h_o = \frac{240(0.0128)(1.042)(0.9)}{(0.73/12)}$$

h_o = 47.3 Btu/hr (ft^2) (°F)

For overall U cooling, assume:
water side fouling = 0.002
propylene side fouling = 0.001
neglect tube wall resistance

$$\frac{1}{U} = \frac{1}{47.3} + 0.001 + 0.002 + \frac{1}{850}$$

$$= 0.0212 + 0.001 + 0.002 + 0.00117 = 0.02537$$

U = 39.6 Btu/hr (ft^2) (°F)

Area required for gas cooling:

$$A_{gc} = \frac{(1{,}415{,}000/2)}{(39.6)(36.2)} = 494 \text{ ft}^2 \text{ per unit}$$

Total area per unit:
A = 1,220 + 494 = 1,714 ft^2

Total area available per unit:
A = (0.196) (15.5 ft) (646) = 1,960 ft^2

Note that this assumes 3 in. as the thickness for each tubesheet: 16 ft − 6 in = 15.5 ft

$$\text{Overall "factor of safety"} = \frac{1{,}960 - 1{,}714}{1{,}714}(100) = 14.3\%$$

This is not excessive.

Area available for gas cooling = (5 ft) (0.196) (646) = 632 ft^2
Area calculated required for gas cooling = 494 ft

$$\text{Percent extra area} = \frac{632 - 494}{494}(100) = 28\%$$

Area available for condensing = (10.5 ft) (0.196) (646) = 1330 ft^2
Area calculated required for condensing = 1220 ft^2

$$\text{Percent extra area} = \frac{1{,}330 - 1{,}220}{1{,}220}(100) = 9\%$$

The baffling for the gas cooling area could be adjusted to make more area available for condensing and, thereby, balance the unit a little better. In operation these areas will become balanced, and some condensing will undoubtedly take place in the gas cooling area. In either case the unit size is within the range to allow reasonable plant operating flexibility without increasing the capital cost of the unit significantly.

For the tube side pressure drop, refer to Figure 10-138:

At 5.1 ft/sec
Read 15 psi/100 ft of tube
Tube length = (15/100) (16 + 16) = 4.7 psi
Allow 20% for fouling:
Δp = 4.7 (1.2) = 5.65 psi

From Figure 10-139,
For two-pass exchanger:

1 entrance
1 return
<u>1</u> exit
3

At 5.1 ft/sec tube velocity, Δp_r = 0.7 psi/pass
Then: 3 (0.7) = 2.1 psi (This is conservative, as some designers use (1) (0.7) = 0.7 psi per pass to cover a unit of this type.)
Total tube side Δp_t = 5.65 + 2.1 = 7.75 psi
This should be the maximum expected value.

Shell-side pressure drop due to gas cooling:
Reading Figure 10-140 at R_e = 149,800,
f_a = 0.0017 from chart/1.2 = 0.00142

$$\Delta p_s = \frac{f_s G^2 D_s (N_c + 1)}{2 g \rho D_e (\mu/\mu_w)^{0.14}}$$

$$= \frac{(0.00142)(65{,}000)^2(29/12)(7 + 1)}{2(4.17 \times 10^8)(2.2)(.73/12)(0.9)}$$

Δp_s = 1.16 psi

The pressure drop due to condensing is usually negligible in a unit of this type. As a maximum, it may be taken as one half of the gas flow drop calculated for one baffle. This would be 1.16/8 = 0.145 psi for the condensing portion. Note that this does not recognize tube supports at 50% cut area, but for pressure units, this pressure drop will be nil.

Example 10-12. Steam Heated Feed Preheater—Steam in Shell

Design a preheater for heating the feed to a distillation column. The 54,180 lb/hr of feed consists primarily of ethyl benzene and styrene and is to be heated from 50°F to 207°F. Steam is available at 10 psig.

The average physical properties of the feed have been calculated over the temperature range, at 128°F:

Molecular weight: 104
Specific heat, c_p: 0.428 Btu/lb (°F)
Viscosity: 0.4765 centipoise
Thermal conductivity: 0.0891 Btu/lb (ft²) (°F/ft)
Density: 53.4 lb/ ft³
Heat duty Q = (54,180) (0.428) (207 − 50) = 3,640,000 Btu/hr

Heat Transfer Surface Required

Try an 18-in. diameter shell unit with 82 × 1-in. O.D. × 12 BWG steel tubes × 12 ft long × 6 pass tubes.

Tube I.D. = 0.782
Tubes/pass = (82/6) = 13.67

$$\text{cross-section/pass} = 13.67\left[\frac{(0.782)^2(0.7854)}{(144)}\right] = 0.0455 \text{ ft}^2$$

$$\text{ft/sec tube velocity} = \frac{(54{,}180)}{(53.4)(3{,}600)(0.0455)} = 6.20 \text{ ft/sec}$$

$$D = \frac{(0.782)}{(12)} = 0.06512 \text{ ft, tube I.D.}$$

$$G = \frac{(54{,}180)}{(0.0455)} = 1{,}190{,}000 \text{ lb/hr (ft}^2)$$

$$R_e = \frac{(0.06512)(1{,}190{,}000)}{(0.4765)(2.42)} = 67{,}200$$

j_H = 182, from Figure 10-54.

$$h_i = (182)\left(\frac{k}{D}\right)\left(\frac{c_p\mu}{k_a}\right)^{1/3}\left(\frac{\mu}{\mu_w}\right)^{0.14}$$

$$h_i = \frac{(182)(0.0891)}{(0.06512)}\left[\frac{(0.4281)(0.4765)(2.42)}{0.0891}\right]^{0.333}$$

$$= (249)(5.55)^{0.333}$$

h_i = 440 Btu/hr (ft²) (°F)
$1/h_i$ = 1/440 = 0.002270
$1/h_{io}$ = (0.002270) (1.00/0.782) = 0.00290
Assume inside fouling, r_i = 0.001
r_{io} = (0.0010) (1.00/0.782) = 0.00128
Tube resistance, 1/k = 0.00024
Assume outside (steam) fouling: r_o = 0.00050
Steam side film, h_o, assumed:
$1/h_o$ = 1/1,500 = <u>0.00067</u>
Σr = 0.00559

Note for oil-free steam, it is usually quite safe to assume the steam film coefficient = 1,500. Most calculated values will be considerably greater than this.

U_o = 1/0.00559 = 179 Btu/hr (ft²) (°F)

Log mean temperature difference:

239.3°F	239.3°F
207.0°F	50.0°F
Δ = 32.3°F	Δ = 189.3°F

$$\Delta_t = \frac{189.3 - 32.3}{\ln\frac{189.3}{32.3}} = \frac{157.0}{\ln 5.8607} = \frac{157}{1.767} = 88.85$$

Δt correction = 1.0, because this is total condensing on one side.

$$\text{area required} = \frac{(3{,}640{,}000)}{(179)(88.85)(1.0)} = 229 \text{ ft}^2$$

Area available = (82) (11.75) (0.2618) = 252 ft²

$$\text{Safety factor based on fouling condition: } = \frac{(252)(100)}{(229)} = 1.10$$

≅ 10%

Tube-Side Pressure Drop

R_e = 67,200

From Figure 10-137,

f = 0.000165
G = 1,190,000 lb/hr ft²
D = 0.06512 ft
L = 12.00 ft
n = 6.00 passes
ρ_L = 53.4 lb/ft³

$$\Delta p_t = \frac{(0.000165)(1{,}190{,}000)^2(12.00)(6.0)}{(2)(417{,}000{,}000)(53.4)(0.06512)}$$

= 5.80 psi, uncorrected for tube passes

For pass corrections use fluid flow expansion and contraction as an illustration of one approach to these pressure drop calculations.

Assume channel diameter = 17.25 in. (based on approximate layout)
Sectional area of channel = (17.25)²(0.7854) = 233.5 in.²
≅ 17.25 in. diameter
Sectional area of tubes = (82) (0.782)²(0.7854) = 39.5 in.²
≅ 7.1 in. diameter

$$\frac{d}{D} = \frac{7.1}{17.25} = 0.410$$

Because the number of tubes per pass is equal per pass, assume that the corresponding area of the channel is equal for all passes. Use the data from Table 2-2 and Figure 2-21, Chapter 2, Fluid Flow, Volume 1, 3rd Ed. Reading data from Standards of the Hydraulic Institute:

k contraction = (0.375) (6 pass + 1 exit nozzle) = 2.63
k expansion = (0.700) (6 pass + 1 inlet nozzle) = 4.90
$\Sigma K = 7.53$
$k = Kv^2/2g = 7.53\ (6.20)^2/2\ (32.2) = 4.5$ ft liquid
= 4.5/2.3 ft/psi = 1.95 psi
Δp_t = total = 5.80 + 1.95 = 7.75 psi
Use 8.5 psi for design purposes.
Note: Here, k is resistance coefficient, also K.

Shell-Side Pressure Drop: Negligible

The unit proposed has been checked as satisfactory for the service. Other designs could be assumed and balanced for reasonable velocities, pressure drops, and area.

Example 10-13. Gas Cooling and Partial Condensing in Tubes

Design a partial condenser to cool a mixture of hydrogen chloride-water vapor from 178°F to 90°F using 60 gal per min of chilled water at 70°F. The unit is to have the acid mixture in the tubes, because this will allow for a cheaper construction than if this material were in the shell. The tube-side material is to be impervious graphite, and the shell and shell-side baffles are to be steel. The acid vapor is essentially at its dew point.

The specification sheet summarizing the design is given in Figure 10-82.

Head Load

1. Cool: 1,496.8 lb/hr HCI from 178°F to 90°F
   156.6 lb/hr H_2O vapor from 178°F to 90°F
   1,653.4 lb/hr to condenser

2. Condense: 149.6 lb/hr H_2O vapor and 102 lb/hr HCI = 251.6 lb/hr

Condensing heat = 902.1 Btu/lb condensed

Q cooling	= (1,496.3) (0.192) (178 − 90) =	25,300 Btu/hr
	= (156.6) (0.450) (178 − 90 =	6,200
Q condensing	= (902.1) (251.6) =	227,000
Total heat duty		258,500 Btu/hr

Water Temperature Rise

60 gpm ≅ 30,000 lb/hr

$$\Delta T = \frac{Q}{Wc_p} = \frac{(258{,}500)}{(30{,}000)(1)} = 8.62°F$$

Exit water temperature = 70 + 8.62 = 78.62°F

ΔT Determination (Water Available at 70°F)

°F Vapor Side	Q, Btu/hr	°F Water	ΔT (vapor-water)
178	0	78.62	99.4
165	87,900	75.69	89.3
145	172,700	72.86	72.1
125	220,400	71.27	53.7
104	251,500	70.23	33.8
90	258,500	70.00	20.0

Integrated ΔT = 76.48°F, see Figure 10-83

Log mean temperature difference is not used because the distribution of the exchanger area varies through the unit, due to changing heat load.

Δt correction: use Figure 10-34

$$P = \frac{178-90}{178-70} = \frac{88}{108} = 0.815$$

$$R = \frac{78.62-70}{178-90} = \frac{8.62}{88} = 0.0979$$

Δt correction = 0.935
Δt corrected = (76.48) (0.935) = 71.5°F (integrated value)

Tube-Side Coefficient

Tubes are $1\,{}^1/_4$-in. O.D. × ${}^7/_8$-in. I.D.

Condensing Coefficient

Use method of Akers et. al.[1] (Method preferably used for pure pressure rather than mixed vapors.)

$$G_e = \bar{G}_L + G_g\,(\rho_L/\rho_v)^{1/2} \qquad (10\text{-}114A)$$
$$\bar{G}_L = (0 + 251.6)/2 = 125.8 \text{ lb/hr}$$
$$\bar{G}_L = 125.8/0.1128 \text{ ft}^2 = 1{,}115 \text{ lb/hr (ft}^2\text{ cross-sect.)}$$

$$\text{Flow Cross-section area/pass} = \left(\frac{55}{2}\right)\frac{(3.14)}{(4)}\left(\frac{0.875}{144}\right)^2$$
$$= 0.1128 \text{ ft}^2\text{/pass}$$

$\bar{G}_g$ = average mass velocity of vapor, in to out

$$= \left[\frac{1{,}653.4 + 1{,}401.8}{(2)}\right]\frac{1}{0.1128} = \frac{1527.6}{0.1128}$$

$\bar{G}_g$ = 13,520 lb/hr (ft² tube cross-sect.)
$\bar{G}_g = (\rho_L/\rho_g)^{1/2} = 13{,}520\ (72.4/0.0831)^{1/2} = 399.800$
G_e = 1,115 + 399,800 = 400,915 lb/hr (ft² cross-sect.)
Avg. mol. wt. = 33.14 + 36.36/2 = 34.75

Note: 114°F is integrated average temperature for the following physical properties:

DWG. NO. A

Item No.

By | **EXCHANGER RATING** | Job No.

Date:- | | Charge No.

Apparatus Hcl Partial Condenser | Plant

Minimum Surface Area per Shell, Sq. Ft.: 157 | Outside | Inside:

Number of Units: Operating One | Spares None

DESIGN DATA PER Shell		
UNIT DATA	SHELL SIDE	TUBE SIDE
Fluid	Chilled Water	Hcl + Water Vapor
Fluid Flow Lbs./Hr.	30,000	1653.4
Temperature In °F.	70	178
Temperature Out °F.	78.6	90
Operating Pressure PSIG	25	0.8
Density Lbs./CF		0.0759 (Avg.)
Specific Heat Btu/Lb./°F.	1.0	0.2166 (Avg.)
Latent Heat Btu/Lb.		902.1 (1)
Therm. Cond. Btu/Hr./Sq. Ft./°F-/Ft.	0.348	0.0098 (Avg.)
Viscosity Centipoise	0.931	0.0148 (Avg.)
Molecular Weight	18	34.65 (Avg.)
No. of Passes	One	Two
Pressure Drop PSI	Calc: 1.23 Used: 2.25	Calc: 0.24 Used: 0.30
Fouling Factor	0.0015	0.0010

Heat Transferred - BTU/Hr: 258,500

LMTD (2) 71.50 °F. Overall U: Calc. 43.2 Used 23.0

CONSTRUCTION		
Max. Oper. Pressure	50 PSI	50 PSI
Max. Oper. Temperature	340 °F.	340 °F.

Type of Unit Impervious Graphite | Tube Pitch 1 5/8" Δ | Joint

Tubes - Material: Impervious Graphite No. 55 O.D. 1 1/4" 7/8" I.D. Length 9 ft.

Shell - Material: Steel | Diameter (Approx.):

Channel Material: Impervious Graphite | Supports Material:

Tube Sheet Material: Impervious Graphite | Baffle Material:

Corrosion Allowance - Shell Side | Tube Side

Connections - Shell In: 4" Steel | Out 4" Steel | Flange 150 R.F.

Channel In: 4" Graphite | Out 4" Graphite | Flange

Others Nozzle Cplgs. | Size 3/4" Shell Side | Flange 3000 # Cplg. Plugged

Bolts: | Gaskets: Neoprene

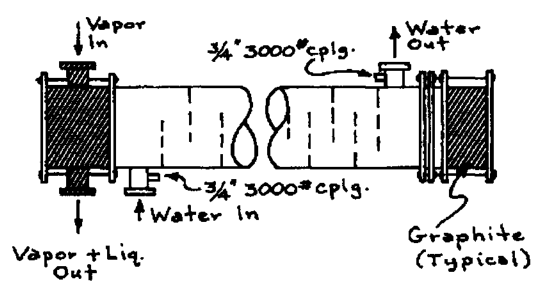
Code None | Stamp | X-Ray | SR

Insulation None | Class | Cathodic Protection None

Channel:
One horiz. pass baffle nozzle end

Shell:
22 Baffles (25% Horiz. Cut) on
4" Center to Center Spacing.
Provide 1/2" V-notch on the
bottom of lower baffles
for drainage.

BAFFLE ARRANGEMENT



NOZZLE ARRANGEMENT

Remarks:- (1) Total phase heat change @ 90°F. Including heat sol'n.
102 lb./hr. Hcl absorbed in 149.6 lb./hr. H_2O condensed.
(2) Integrated average from Q vs. ΔT plot = 76.48°F.
ΔT corrected = 71.50°F.

Checked | Date | Approved | Date

Rev. | By | Date | B/M No.

Figure 10-82. Exchanger rating specifications for hydrogen chloride partial condenser.

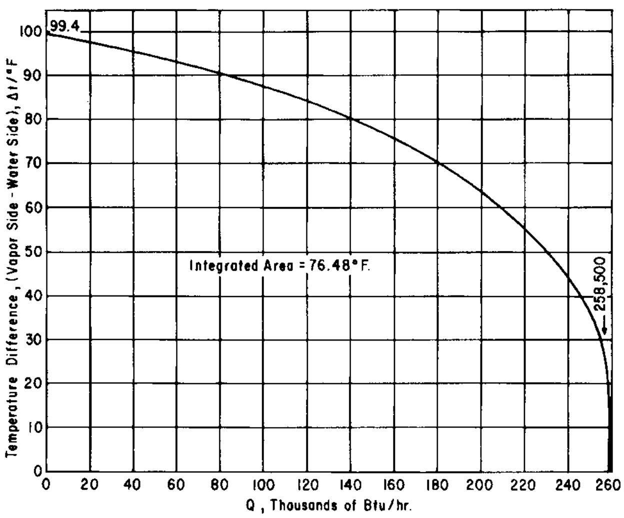


Figure 10-83. Heat duty variation with temperature difference as vapor flows through unit.

ρ_L at 114°F = 72.4 lb/ft^3
c_{pL} at 114°F = 0.615 Btu/lb (°F)
k_{aL} at 114°F = 0.280 Btu/hr (ft^2) (°F/ft)
μ_L at 114°F = 1.30 centipoise (35 wt. % avg.) = 3.15 lb/(hr) (ft)

$$R_e = \frac{DG_e}{\mu_L} = \frac{(0.0729)(400{,}915)}{3.15} = 9.260$$

Reading Figure 10-75,

$$\phi = \frac{h_{cm}D}{k_a}\left(\frac{c_p\mu}{k_a}\right)^{-1/3} = 100$$

Allowing for the spread of data and tending to be conservative, use 85% of value read from chart:

use, $\phi = 85$

$$h_i = \frac{(85)(0.280)}{(0.0729)}\left(\frac{0.615 \times 3.15}{0.280}\right)^{1/3} = (326.5)(6.92)^{1/3}$$

$$h_i = (326.5)\ (1.904) = 622 \text{ Btu/hr (ft}^2\text{) (°F)}$$

Referencing to outside of tube:

$$h_{io} = \frac{(622)(0.875)}{(1.250)} = 436 \text{ Btu/hr (ft}^2\text{)(°F)}$$

Sensible Gas Cooling Coefficient

μ_v = 0.0358 lb/hr (ft)
D = 0.0729 ft
c_p = 0.217 Btu/lb (°F) (avg.)
k_a = 0.00979 Btu/hr (ft^2) (°F/ft)
R_e = 27,650
j_H = 90.0 (see Figure 10-54)

$$h_i = j_H \frac{k}{D}\left(\frac{c_p\mu}{k}\right)^{1/3}$$

$$h_i = \frac{(90)(0.00979)}{(0.0729)}\left(\frac{0.217 \times 0.0358}{0.00979}\right)^{1/3} = (12.10)(0.794)^{1/3}$$

$$h_i = (12.10)\ (0.9262) = 11.20$$

$$h_{io} = 11.20\left(\frac{0.875}{1.25}\right) = 7.84$$

Shell-Side (Water) Coefficient

Use 4-in. baffle spacing and 25% cut baffles in 6-in. O.D. shell.

Use flow rate = 30,000 lb/hr.

Shell-side equivalent diameter:

$$d_e = \frac{4\left[(p/2)(.86)(p) - \frac{\pi}{2}\,d_o^2/4\right]}{\frac{\pi d_o}{2}}$$

$$= \frac{4\left[\left(\frac{1.625}{2}(.86) \times 1.625\right) - \frac{\pi}{2}\,\frac{(1.25)^2}{4}\right]}{\frac{\pi}{2}(1.25)}$$

d_e = 1.06 in. ≅ 0.0886 ft
D_e' = 0.0886 ft

Shell-side bundle cross flow area:

$$a_s = \frac{(D_s)(c')(B)}{p(144)} = \frac{(15.25)(0.375)(4)}{1.625(144)} = 0.0977 \text{ ft}^2$$

G_s = (30,000)/(0.0977) = 307,000 lb/hr (ft^2)
μ(water at 74.3°F) = 0.931 centipoise = 2.25 lb/hr (ft)

$$Re_s = \frac{(0.0886)(307{,}000)}{(2.25)} = 12{,}070$$

j_H = 6.10, Figure 10-54
c_p = 1.0
k_a = 0.348 Btu/hr (ft^2) (°F/ft)

$$h_o = j_H \frac{k}{D_e}\left(\frac{c_p\mu}{k}\right)^{1/3} = \frac{(61.0)(0.348)}{(0.0886)} = \left(\frac{1.0 \times 2.25}{0.348}\right)^{1/3}$$

$$h_o = (239)\ (6.46)^{1/3} = (239)\ (1.862) = 445 \text{ Btu/hr (ft}^2\text{) (°F)}$$

Fouling Factors and Tube Resistance

Assume inside fouling factor = 0.00010

Assume water side (shell) fouling factor = 0.0015

Tube resistance for $^3/_8$-in. impervious graphite wall:

$$r_t = L_w/k = \frac{0.375}{1{,}020 \text{ Btu/hr (ft}^2\text{)(°F/in.)}}$$

r_t = 0.000368

Condensing Surface Area

$$U_o = \frac{1}{\frac{1}{h_{io}} + r_i(d_o/d_i) + r_t + r_o + \frac{1}{h_o}}$$

$$= \frac{1}{\frac{1}{436} + 0.001(1.25/0.875) + 0.000368 + 0.0015 + \frac{1}{445}}$$

$$U_o = \frac{1}{0.00783} = 127.6 \text{ Btu/hr (ft}^2)(^\circ\text{F})$$

Area: $A = Q/U\,\Delta t$

$$A = \frac{(227{,}000)}{(127.6)(71.50)} = 24.9 \text{ ft}^2$$

Sensible Gas Cooling Area

$$U_o = \frac{1}{\frac{1}{7.84} + 0.001428 + 0.000368 + 0.0015 + \frac{1}{445}}$$

$$U_o = \frac{1}{0.1275 + 0.001428 + 0.000368 + 0.0015 + 0.002245}$$

$$U_o = \frac{1}{0.133041} = 7.51$$

$$\text{Area: } A = \frac{(25{,}300 + 6{,}200)}{(7.51)(71.50)} = 58.7 \text{ ft}^2$$

Total Area

$A = 24.9 + 58.7 = 83.6 \text{ ft}^2$
"Safety Factor," or excess area:
% S.F. with unit containing 157 ft^2 = 157/83 .6 = 1.878 = 87.8%

This is normally too large a value to be considered an economical design. However, in this case, future flow rates indicate that the 157 ft^2 will be close to the needed area. Rather than handle and provide piping and special valves for two units, it is cheaper and easier from an operation viewpoint to install one large unit at this time.

Overall U Calculation

$$U = \frac{258{,}500}{(83.6)(71.50)} = 43.2$$

Overall U Used (Based on the Over-Sized Unit)

$$U = \frac{(258{,}500)}{(157)(71.50)} = 23.0$$

Density Avg. Vapor Flow @ Avg. Temperature of 134°F 0.65 psig

Avg. MW vapor @ inlet and exit avg. conditions = 34.65

$$\rho_v = \frac{(34.65)(15.35)(520)}{(379)(14.70)(594)} = 0.0835 \text{ lb/ft}^3$$

Shell-Side ΔP

R_e = 12,070, previous calculation, from Figure 10-140
f = 0.00205

$$\Delta p_s = \frac{f_s G_s^2 D_s'(N_c + 1)}{2 g \rho D_e' \phi_s}$$

where
Number of baffles = 22
$g = 4.17 \times 10^8$
G_s = 307,000 lb/hr (ft^2)
D_e' = 0.0886 ft
$D_s' = 15.^{25}/_{12} = 1.27$ ft
ρ_L = 62.27 lb/ft^3
ϕ_s = essentially 1.0

$$\Delta p_s = \frac{(0.00205)(307{,}000)^2(1.27)(22 + 1)}{(834{,}000{,}000)(62.27)(0.0886)}$$

Δp_s = 1.23 psi; use 2.25 psi for system allowances

Note that this pressure loss does not account for nozzle entrance or exit losses. These losses may be neglected provided velocities are low and no unusual conditions are imposed upon these connections. For low pressure systems, these losses cannot be ignored.

Tube-Side Pressure Drop

$R_e = D\,G_t/\mu_v$
μgas @ 114°F: 90.25 mol % HCI in feed, 99.7 wt. % exit
Avt. wt. % HCI = 95.0 wt. %
μ' = (0.950) (0.0150) + (1 + 0.950) (0.0105)
= 0.01478 centipoise
$\mu \cong$ 0.0358 lb/ft hr

$$R_e = \frac{(0.0729)(13{,}560)}{(0.0358)} = 27{,}650$$

f = 0.000204 (Figure 10-137)

$$\Delta p_t = \frac{(0.000204)(13{,}560)^2(6)(2)}{(2)(417{,}000{,}000)(0.0831)(0.0729)} = 0.089 \text{ psi}$$

ρ_v = at 114°F and 1 atm = 0.0831 lb/ft^3

For bundle entrance and exit losses, refer to copyrighted graph of Donohue.[38]

Δp_r = (0.051) (2) = 0.102 psi
Δp_{tt} calculated total = 0.089 + 0.102 = 0.191 psi
Δp_{tt} used = (0.191) (1.2) = 0.23 psi

Condensing Vapors in Presence of Noncondensable Gases

A stream containing a noncondensable and vapors to be condensed must be considered so that the continually changing gas vapor physical properties (and some thermal properties), gas film heat transfer coefficient, and mass gas flow rate are adequately represented. This operation is usually a constant pressure process. The vapor condenses at its dew-point on the tubes, thereby providing a wet surface; a noncondensable gas film surrounds this surface; and the vapor of the stream diffusing through this film condenses into the liquid film of the condensate on the tube, see Figure 10-84. The sensible heat and latent heat of the vapor are transferred through the gas film and the liquid film to the tube surface (except when considerable subcooled condensate film exists, in which cases there may be condensation or fogging in the gas film). The rigorous method of design of Colburn[30] and Colburn and Hougen[31] involving trial-and-error calculations is considered the most accurate of the various alternate procedures published to date. Kern[70] presents a very useful analysis of special design problems with examples.

The effect of a noncondensable gas in the system with a condensable vapor is to significantly reduce the condensing side film coefficient. Henderson and Marcello[62] present data to illustrate the effect. Figures 10-85, 10-86, and 10-86A present the effect of ΔT with a steam-air system and toluene-nitrogen. The following is a reasonable short-cut approach that can be acceptable for many applications but certainly is not as accurate as the Colburn-Hougen[30, 31] method:

$$H = \frac{1}{1 + Cy} \tag{10-115A}$$

where

- H = heat transfer coefficient ratio, h_M/h_{Nu}
- h_M = effective heat transfer film coefficient, Btu/hr-ft^2-°F
- h_{Nu} = condensing film coefficient by Nusselt equation Btu/hr-ft^2-°F
- y = mol (volume) percent noncondensable gas in bulk stream.
- C = see following table[62]

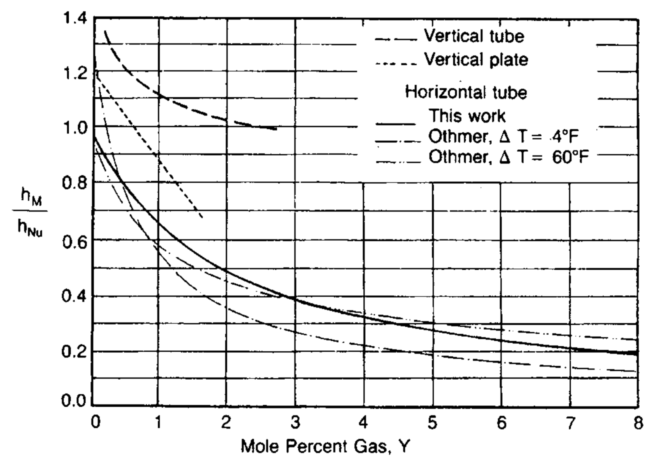


Figure 10-86. Heat transfer ratio correlations for steam and air system. (Used by permission: Henderson, C. L., and Marchello, J. M. *ASME Transactions Journal of Heat Transfer,* V. 91, No. 8, p. 44, ©1969. American Society of Mechanical Engineers. All rights reserved.)

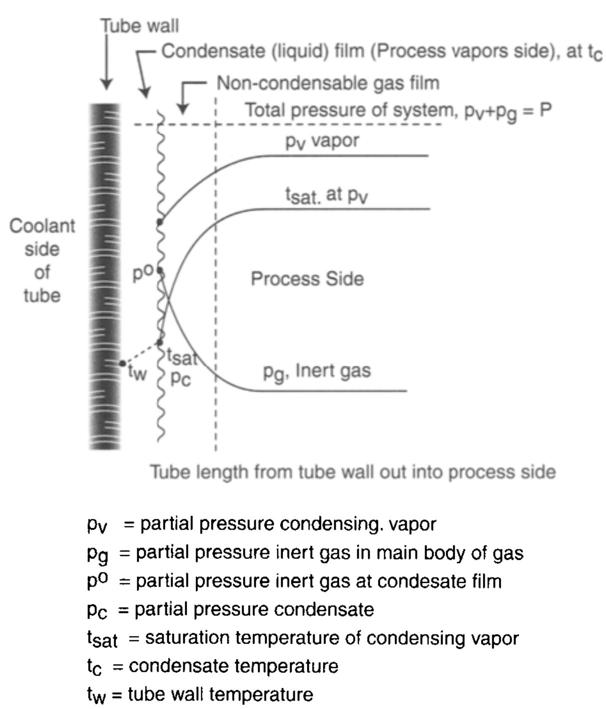


Figure 10-84. Condensable vapors in presence of a noncondensable gas.

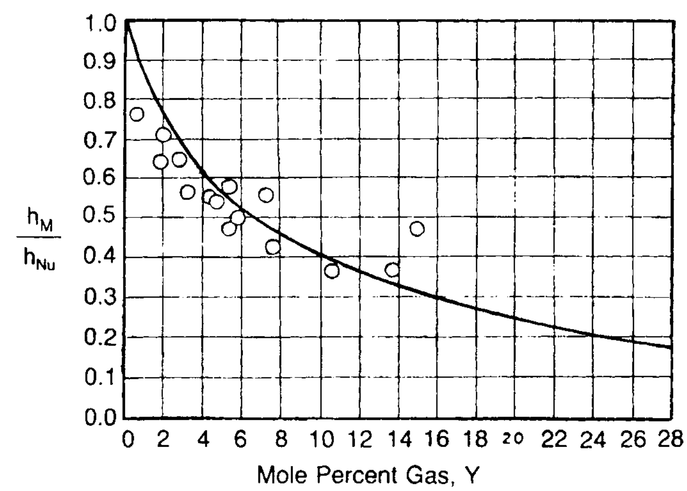


Figure 10-85. Heat transfer ratio for toluene and nitrogen. (Used by permission: Henderson, C. L., and Marchello, J.M. *ASME Transactions Journal of Heat Transfer,* V. 91, No. 8, p. 44, ©1969. American Society of Mechanical Engineers. All rights reserved.)

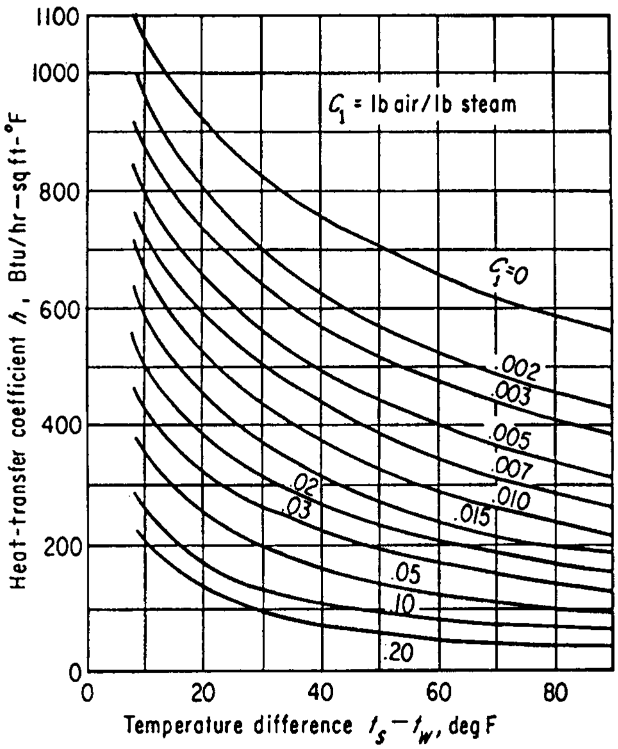


Figure 10-86A. Influence of air content on the heat transfer coefficient of steam containing air. (Used by permission: Edmister, W. C., and Marchello, J. M. *Petro/Chem. Engineer*, June 1966, p. 48. © Petroleum Engineer International.)

System	C	% Range Non-condensable	% Standard Deviation
Steam-air	0.51	0.64–25.1	9.2
Toluene-nitrogen	0.149	0.71–59.1	8.7
Benzene-nitrogen	0.076	7.1–20.3	14.3

Figures 10-85, 10-86, and 10-86A and Equation 10-115A represent the effective reduction of the pure component (condensable) when inert gases are present, resulting in the *reduced effective* heat transfer for condensing the mixture.

Although it is not stated in the study, from a practical industrial standpoint, the effects of air, nitrogen, and other common inert gases can be expected to be about the same for other organic systems.

A computer program developed by Volta[121] handles the problem of condensing in the presence of a noncondensable gas for down-flow of either a saturated or superheated gas-vapor mixture *inside* vertical tubes. The program is based on a modification of Colburn-Hougen and Bras and is certainly more accurate and easier to use than the lengthy manual calculations. Although the program was written for vertical tubes, it can be used to approximate the results in a horizontal unit, and if the correction factor between vertical and horizontal tube condensation is applied, the comparison may be improved. The method uses diffusion coefficients. An example using vertical tubes is included.

The survey of Marto[181] includes several excellent references to this topic. The proposal of Rose[182] is reported to give good agreement with selected experimental data.

Example 10-14. Chlorine-Air Condenser, Noncondensables, Vertical Condenser

A chlorine-air mixture is to be cooled and the water vapor condensed using chilled water. The design conditions are as follows:

Flow: 92.3 lb gas mixture, per tube. Estimated tube bank: 4.48 lb water, per tube
Gas in: 110°F (saturated)
Required gas out: 58°F
Water in: 48°F
System pressure: 1 atm
Number of tubes, assumed: 416; 0.75 in. O.D. × 20 BWG

The computer print-out of good results is presented in Table 10-23. A brief interpretation of the result follows:

Water condensed, total 815 lb/hr
Partial pressure water vapor in: 0.087
Partial pressure water vapor out: 0.011
Cooling water out (counter flow): 58.14°F
Inside film coefficient, Btu/hr (ft^2) °F: 13.12 (avg.)
Internal tube surface (calculated): 786 ft^2
Internal tube surface (recommended): 867 ft^2

The design method of Colburn and Hougen[31, 70] has withstood many examinations and is considered the best for any problem of this type. However, it is somewhat long and tedious and several approximation methods have been proposed.[9, 10, 11, 12, 23, 79, 123]

The graphical methods of Bras[9, 175, 176] provide helpful short-cuts to avoid the tedious trial-and-error solutions required of the rigorous methods. Reference 9 is the most recent and perhaps the easier to use. The results agree in general within about 10%.

The graphical method of Hulden[68] is also helpful as it is not as tedious as the arithmetic methods, and based on his comparison with the Colburn-Hougen method, the proposed results are within 1%.

All of these have some limitations and have not been thoroughly compared against the Colburn method, which is considered to be within 10% of any correct solution. Cairns[23, 176] has compared his proposal with 6 different systems and 4 other approximation methods. In general, the agreement with the Colburn-Hougen method is excellent.

The selection of the number of temperature increments is important as it affects the accuracy of the final heat transfer area. In the majority of cases, the selection of a limited

Table 10-23
Computer Printout for Example 10-14

Cooler Condenser Design
Program No. 710402
Modified Colburn-Hougen, Bras Method
Design Calc. for Sec. Coolers—Water Sat'd at 110°F
Pass No. 2

Lb-Water Water Rate Tube 4.48 lb min tube	Lb-Gas, Gas Rate Tube 92.30 lb hr tube	HLI, Water & Tube 480.00 Overall Coeff. I.D.	Acc-F, Program Accuracy .10 Controlled to 0.1 F
TL-in. Cooling Water In. °F 48.00	Gas in Top	PI-Atm, Press. of Sys 1.000	

Temp. Gas In. °F (TV)	Partial Press. Water at Interface (PV)	Temp. of Interface (TO)	Temp. Water Out. °F (TL)	Heat Duty Tube Cum. Btu hr tube (Q)	Film Coeff. (HF)	Cumulative Water Condensed, lb hr tube (WC)	Reynolds No. Gas Re No.	Cumulative Area Tube, inside, ft^2 (AREA)	Length of Tube, ft (L)
110.00	.087	77.71	58.14	.59	14.8	.00	61665.2	0.00	0.00
109.00	.078	76.98	57.14	269.3	14.65	.24	61571.5	.03	.18
108.00	.071	74.62	56.28	499.0	14.51	.44	61508.6	.06	.35
107.00	.064	72.58	55.55	694.3	14.40	.61	61472.6	.09	.52
106.00	.059	70.59	54.93	863.2	14.30	.76	61457.3	.12	.67
105.00	.054	68.79	54.38	1011.2	14.21	.88	61458.5	.15	.83
104.00	.050	67.16	53.89	1142.1	14.13	.99	61473.0	.17	.97
103.00	.046	65.58	53.46	1259.0	14.06	1.09	61498.5	.20	1.12
102.00	.043	64.25	53.07	1363.9	14.00	1.18	61533.3	.22	1.26
101.00	.040	63.03	52.71	1458.8	13.94	1.25	61576.1	.25	1.40
100.00	.037	61.91	52.39	1545.0	13.89	1.32	61625.8	.27	1.54
99.00	.035	60.88	52.10	1623.6	13.84	1.38	61681.6	.30	1.67
98.00	.033	59.94	51.83	1695.5	13.80	1.44	61742.6	.32	1.81
97.00	.031	59.07	51.59	1761.6	13.76	1.49	61808.4	.35	1.95
96.00	.029	58.36	51.36	1822.4	13.72	1.53	61878.4	.37	2.08
95.00	.027	57.62	51.15	1878.5	13.68	1.57	61952.4	.40	2.22
94.00	.026	56.93	50.96	1930.5	13.65	1.61	62029.7	.42	2.36
93.00	.024	56.30	50.78	1978.8	13.62	1.64	62110.3	.44	2.50
92.00	.023	55.71	50.61	2023.7	13.59	1.67	62193.7	.47	2.64
91.00	.022	55.17	50.46	2065.5	13.56	1.70	62279.8	.49	2.78
90.00	.021	54.66	50.31	2104.6	13.53	1.73	62368.3	.52	2.92
89.00	.020	54.20	50.18	2141.1	13.51	1.75	62459.0	.55	3.07
88.00	.019	53.77	50.05	2175.3	13.48	1.77	62551.9	.57	3.21
87.00	.018	53.37	49.93	2207.5	13.46	1.79	62646.7	.60	3.36
86.00	.018	53.00	49.82	2237.7	13.44	1.81	62743.3	.63	3.51
85.00	.017	52.66	49.71	2266.2	13.42	1.82	62841.6	.65	3.67
84.00	.017	52.35	49.61	2293.0	13.40	1.84	62941.5	.68	3.82
83.00	.016	51.96	49.52	2318.6	13.38	1.85	63043.0	.71	3.98
82.00	.015	51.71	49.43	2342.7	13.36	1.86	63145.6	.74	4.15
81.00	.015	51.46	49.34	2365.7	13.35	1.87	63249.7	.77	4.32
80.00	.015	51.23	49.26	2387.5	13.33	1.88	63355.0	.80	4.49
79.00	.014	51.01	49.18	2408.2	13.31	1.89	63461.5	.83	4.66
78.00	.014	50.81	49.11	2428.0	13.30	1.90	63569.2	.86	4.84
77.00	.014	50.62	49.04	2447.0	13.28	1.90	63677.9	.90	5.03
76.00	.013	50.44	48.97	2465.1	13.27	1.91	63787.6	.93	5.22
75.00	.013	50.28	48.91	2482.6	13.25	1.92	63898.3	.96	5.42
74.00	.013	50.12	48.84	2499.4	13.24	1.92	64009.9	1.00	5.62
73.00	.013	49.98	48.78	2515.6	13.23	1.93	64122.4	1.04	5.83
72.00	.013	49.84	48.72	2531.3	13.21	1.93	64235.6	1.08	6.05
71.00	.012	49.71	48.67	2546.5	13.20	1.93	64349.7	1.12	6.28
70.00	.012	49.59	48.61	2561.3	13.19	1.94	64464.5	1.16	6.51
69.00	.012	49.47	48.56	2575.8	13.17	1.94	64580.1	1.20	6.76
68.00	.012	49.36	48.51	2589.9	13.16	1.94	64696.3	1.25	7.02
67.00	.012	49.25	48.45	2603.7	13.15	1.94	64813.2	1.30	7.29
66.00	.012	49.15	48.40	2617.3	13.14	1.95	64930.7	1.35	7.57
65.00	.012	49.05	48.35	2630.6	13.12	1.95	65048.8	1.40	7.87
64.00	.012	49.05	48.31	2648.5	13.11	1.95	65167.5	1.45	8.19
63.00	.012	48.84	48.26	2656.4	13.10	1.95	65287.1	1.52	8.52
62.00	.012	48.84	48.21	2669.3	13.09	1.95	65406.8	1.58	8.88
61.00	.012	48.66	48.16	2682.0	13.08	1.95	65527.4	1.65	9.26
60.00	.012	48.66	48.12	2694.7	13.06	1.96	65648.1	1.72	9.68
59.00	.012	48.49	48.07	2707.1	13.05	1.96	65769.8	1.80	10.12
58.00	.011	48.49	48.02	2719.8	13.04	1.96	65891.4	1.89	10.62

Exit Re No of Condensate: 22

number of increments, 5–7, will produce results on the high side. In one case studied the use of 6 points compared to 17 points resulted in an area 36% too high. An important factor in this analysis is the shape of the heat transfer curve. Increments should be chosen smaller in the areas where the rate of change of heat load with temperature is the greatest (see Figures 10-87 and 10-88).

The work of Dmytryszyn[39] indicates that the best agreement between actual and calculated surface areas using the Colburn-Hougen method, when tested with vapors outside a single vertical tube, requires a graphical solution to calculate the heat transfer surface required to cool the incoming gas mixture to its dew point (area described 1, 2, 3, 4), Figure 10-89. The area (described as 1, 2, 3, 5, 1) calculated for gas desuperheating is too large when determined by the usual equations; likewise, calculations based on an overall condensing coefficient give results that are too low.[39]

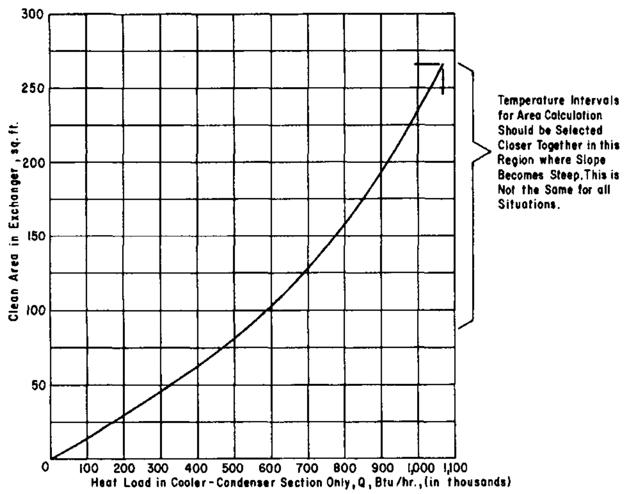


Figure 10-87. Plot of exchanger surface area without fouling for gas-cooling-condensing section only, Example 10-15.

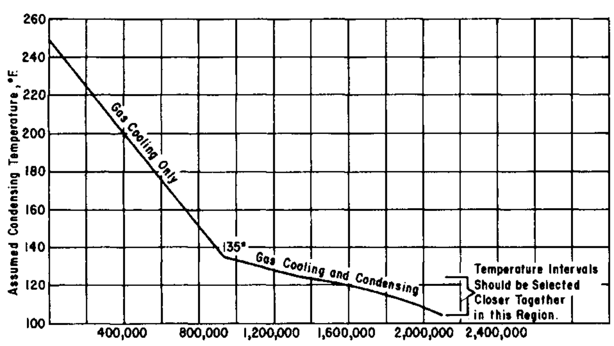


Figure 10-88. Heat load curve for condensing presence of noncondensables for Example 10-15.

The general method as outlined by Kern[70] has been supplemented in the following discussion. The test of Revilock[96] indicates the general applicability of the method.

1. The method is applicable only to gas-vapor mixtures with the vapor at saturation. However, systems involving superheated mixtures and subcooling can be handled as separate problems and added to the cooler-condenser area requirements to form a complete unit.
2. Assume temperature increments of condensation from the inlet temperature to the outlet. The increments should be smaller near the inlet as most of this heat load will be transferred at the higher temperature level. The number of increments is a function of the desired accuracy. However, as a rule, the minimum should be 4, with 6 or more being preferred.
3. Calculate the gas cooling and condensing heat loads for each increment separately and plot a curve representing the total heat load versus temperature.
4. Assume an exchanger unit, establishing shell size, number of tubes, and number of passes. Because the estimation of overall U values for this type of unit is much more variable than for some of the other units, a rough value may be taken between 30 and 60 Btu/hr (ft^2) (°F) as a start (see Table 10-15). The actual weighted Δt will be somewhat larger than an LMTD value; however, this is difficult to approximate without a trial or two or unless a condensate film temperature, t_f, can be estimated for the inlet and outlet conditions of several intervals. An average difference value of these t_f values and the inlet gas temperature to the interval will give a reasonable estimating value for the temperature difference in determining the estimating area, A.

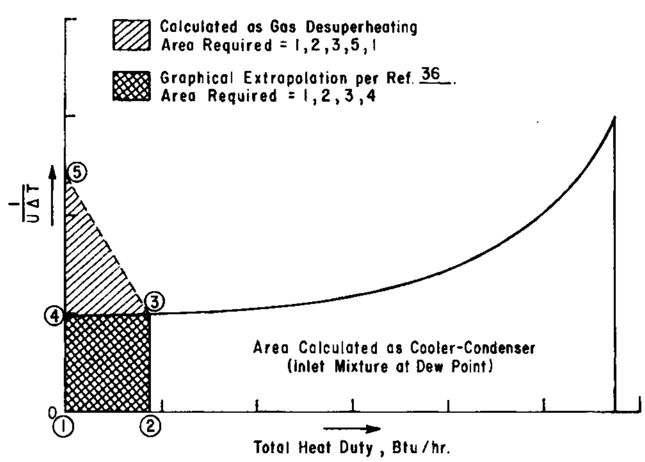


Figure 10-89. Graphical evaluation of gas desuperheating area for a noncondensable-condensable mixture.

Calculate LMTD:

$t_1 \rightarrow t_2$ (cool and condense)
$t_4 \leftarrow t_3$ (water temperature rise)
Assume gas cooling and condensing coefficients, then

$$\text{Trial area} = \frac{\text{q gas cooling}}{\text{U gas cool (LMTD)}} + \frac{\text{q cond.}}{\text{U cond. (LMTD)}} \quad (10\text{-}115)$$

5. Determine the gpm tube-side flow rate and temperature rise for the overall unit to be certain that they are reasonable and consistent with heat load.
6. Calculate the tube-side film coefficient, h_i, and reference it to the outside of the tube, h_{io}. Calculate the condensing film coefficient:

$$\text{Then } \frac{1}{h_{io}'} = \frac{1}{h_{cond.}} + \frac{1}{h_{io}}$$

This value will remain constant throughout the design.

7. Calculate the shell-side dry-gas film coefficient, h_g or h_o, for outside tube conditions. Assume a baffle spacing or about equal to one shell diameter. Use the shell-side method described in Equation 10-48 and Figure 10-54. This is necessary for inlet conditions and then must be checked and recalculated if sufficient change occurs in the mass flow rate, G, to yield a change in h_g.
8. Calculate mass transfer coefficient, K_g using h_o:

$$K_g = \frac{h_o(c\mu/k_a)^{2/3}}{cp_{gf} M_a(\mu/\rho k_d)^{2/3}} = \frac{\text{(Expression)}}{p_{gf}} \quad (10\text{-}116)$$

Gilliland[285] correlation for one gas diffusing through another:

$$k_d = 0.0166 \frac{T_k^{3/2}}{p_t'(V_A^{1/3} - V_B^{1/3})^2}\left(\frac{1}{M_A} + \frac{1}{M_B}\right)^{1/2} \quad (10\text{-}117)$$

where

V_A = molecular volume for component A, diffusing gas
V_B = molecular volume for component B, diffused gas (See chapter on "Packed Towers," Volume 2, 3rd Ed. for further discussion.) Compute from atomic volumes:
k_d = diffusivity, ft^2/hr
p_t' = total pressure, atm
T_k = absolute temperature, ° Kelvin
M_A and M_B = molecular weights of the gases

relating K_g to the change in logarithmic difference in inerts in the main gas body and at the condensate film. This value of K_g may have to be recalculated each time a new h_o is determined, these values being re-evaluated with physical properties at the interval temperatures.

9. For each selected temperature interval, calculate a balance on fundamental relation:

$$h_o\,(t_g - t_c) + K_g M\,\lambda\,(p_v' - p_c) = h_{io}'\,(t_c - t_w') = U\,(t_g - t_w') \quad (10\text{-}118)$$

h_{io}' = inside film coefficient corrected to outside, plus outside condensing film coefficient clean basis, Btu/hr (ft^2) (°F)
h_o = dry gas coefficient, on shell side, Btu/hr (ft^2) (°F)
K_g = diffusion coefficient, lb-mol/(hr) (ft^2) (atm)
M_v = average molecular weight of vapor, dimensionless
p_c = partial pressure of vapor at the condensate film, °F
p_v' = partial pressure of vapor in gas body, atm
t_c = temperature of condensate film, °F
t_g = temperature of dry gas (inerts), °F
t_w' = temperature of water, °F
λ = latent heat of vaporization, Btu/lb

This procedure involves the following:

a. Establish

t_g = inlet temperature of interval, °F.
p_v = vapor pressure of condensate at t_g, psia or atm
P = inlet gas-vapor mixture absolute pressure to interval, allowing for estimated pressure drop where necessary, psia or atm
$p_g = P - p_v$, psia or atm

$$t_w' = t_{wo} - \frac{q_i}{W_t(c_p)_t}, \text{°F} \quad (10\text{-}119)$$

t_w' = temperature of inlet water to interval, °F
t_{wo} = temperature of outside tube wall, °F
W_t = tube side flow rate, lb/hr
$(c_p)_t$ = tube side specific heat, Btu/lb (°F)
q_i = heat load of previous interval, Btu/hr

b. Assume:

t_c = temperature of condensate film, °F
p_c = vapor pressure of condensate at t_c, psia or atm

c. Calculate: $p_g' = P - p_c$, psia or atm.

$$p_{gf} = \frac{p_g' - p_g}{2.3 \log \frac{p_g'}{p_g}}, \text{ psia or atm.} \quad (10\text{-}120)$$

d. Substitute in balance equation and try for as close a balance as reasonable, depending upon the magnitude of the heat load and significance of changes in t_c. Usually ±5% is acceptable. If check is not obtained, reassume t_c and continue as per (b), (c), and (d).

e. Calculate $U\Delta t$ = average of the value of the two sides of equation of (d).

$$U\Delta t = \frac{[h_o(t_g - t_c) + K_g M_v \lambda(p_v - p_c)] + h_{io}(t_c - t_w)}{2} \quad (10\text{-}121)$$

f. Calculate U:

$$U = \frac{U\Delta t}{t_g - t_w} = \frac{U\Delta t}{\Delta t} \quad (10\text{-}122)$$

g. Summarize results of the intervals

Interval	$t_g(°F)$	$t_c(°F)$	$\frac{1}{U\Delta t}$	$\left(\frac{1}{U\Delta t}\right)_{avg}$	q_i	$\Delta A = \frac{\Delta q}{(U\Delta t)_{avg}}$	Δt	Δt_{avg}	$\frac{q_i}{\Delta t_{avg}}$
↓									
•	•	•	•	•	•	•	•	•	•
•	•	•	•	•	•	•	•	•	•
				*	Total	Total		*	Total

*If interval is large, use log mean average.

h. Calculate weighed Δt:

$$\Delta t = \frac{Q(\text{total})}{\sum\left(\frac{q}{\Delta t_{avg}}\right)} \tag{10-123}$$

$$U(\text{clean}) = \frac{Q(\text{total})}{(A, \text{total})(\Delta t)_{\text{weighed}}} \tag{10-124}$$

Apply fouling:

$$U(\text{dirty}) = \frac{1}{\frac{1}{U_c} + f} \tag{10-125}$$

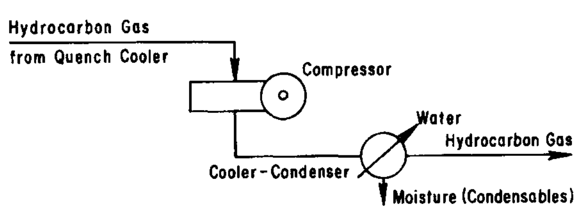
i. The total area for cooler-condenser is

$$A = \frac{Q(\text{total})}{U(\text{dirty})(\Delta t)_{\text{weighed}}} \tag{10-126}$$

j. If the area of (i) does not acceptably match the assumed area, a new unit must be assumed and the calculations repeated for a new balance. If the difference is not great, approximations and ratios can serve to adjust the results to an acceptable figure.

Example 10-15. Condensing in Presence of Noncondensables, Colburn-Hougen Method[31, 70]

A hydrocarbon vapor compressor is discharging a mixture of 970 lb mol/hr dry hydrocarbon gas plus water vapor equivalent to saturation at its inlet of 104°F and 14.2 psia. The gas enters a condenser unit at 250°F and 34 psia and is to be cooled to 104°F. The molecular weight of the dry gas is 14.0.



1. Water vapor entering compressor at 104°F and 14.2 psia and entering cooler-condenser at 250°F.

Vapor pressure at 104°F from steam tables = 1.069 psia

$$\text{lb mol water} = \frac{970}{\left(\frac{14.2 - 1.069}{14.2}\right)}\left(\frac{1.069}{14.2}\right) = 78.8 \text{ lb mol/hr}$$

$$= 1{,}420 \text{ lb/hr}$$

2. Water vapor leaving cooler condenser at 104°F and an assumed 31 psia, allowing 3 psi for pressure drop through unit.

lb mol water vapor

$$= \frac{970}{\left(\frac{31 - 1.069}{31}\right)}\left(\frac{1.069}{31}\right) = 34.6 \text{ lb mol/hr}$$

$$= 622 \text{ lb/hr}$$

3. Water condensed and removed from unit = 1420 − 622 = 798 lb/hr
4. Dew point

Partial pressure entering water

$$= \frac{78.8}{(78.8 + 970)}(34.0) = 2.55 \text{ psia}$$

Dew point from steam tables = 135°F
Partial pressure inerts = 34.0 − 2.55 = 31.45 psia

$$\text{Partial pressure leaving water} = \frac{34.6}{(34.6 + 970)}(31.0) = 1.07 \text{ psia}$$

Assume gas cooling condition down to 135°F and condensation with gas cooling beyond this point. Select temperature intervals for exchanger analysis:

Gas cooling: 250°F to 135°F
Cooling-condensing: 135°F to 130°F
130°F to 125°F
125°F to 115°F
115°F to 104°F

Gas cooling heat load:

c_p gas = 7.85 Btu/mol (°F) average for gas mixture
Enthalpy vapor at 250°F = 1164.0 Btu/lb (steam tables)*
Enthalpy saturated vapor at 135°F = 1119.9 Btu/lb (steam tables)
Gas cooling = (970) (7.85) (250−135) = 876,000 Btu/hr
Water vapor cooling
= (1420) (1164.0−119.9)* = 62,700 Btu/hr
Total = 938,700 Btu/hr
*Correcting for superheat, enthalpy at 2.55 psia at 250°F = 1172.6 Btu/lb
Water vapor cooling would then
= 1420 (1172.6 − 1119.9) = 75,000 Btu

Cooler-condenser heat load:

Interval 135°–130°
Vapor pressure water at 130°F = 2.223 psia (steam tables) = p_v
Pressure of inerts = [34.0 − 1.5]−2.223 = 30.28 psia = p_g
lb mol water vapor in gas phase at 130°F

$$= \frac{970}{\frac{[34.0 - 1.5) - 2.223]}{(34.0 - 1.5)}} \left(\frac{2.223}{34.0 - 1.5}\right)$$

(Note that an allowance of 1.5 psi has been made as pressure drops to this point; this reduces the pressure at the end of this interval to 32.5 psia.)

= 71.2 lb mol/hr
Mol water condensed in this section = 78.8 − 71.2 = 7.6 mol/hr
Heat load of inerts = (970) (7.85) (135 − 130) = 38,100 Btu/hr
Enthalpy water vapor at 130°F = 1117.9 Btu/lb
Heat load of water vapor = [(71.2) (18)] (1119.9 − 1117.9)
= 1540 Btu/hr
Enthalpy of liquid at 130°F = 97.9 Btu/lb
Heat of condensation = [(7.6) (18)] (1119.9 − 97.9)
= 140,000 Btu/hr
Total heat load this section = 179,640 Btu/hr

Interval 130°F–125°F
Vapor pressure water at 125° = 1.942 psia
Enthalpy water vapor at 125° = 1115.8 Btu/lb
Enthalpy water liquid at 125° = 92.91 Btu/lb
Pressure of inerts = 32.5 − 1.942 = 30.56 psia
Mol water vapor in gas phase at 125°F

$$= \frac{970}{\frac{30.56}{32.5}} \left(\frac{1.942}{32.5}\right) = 61.8 \text{ mol/hr}$$

Mol water condensed in this section = 71.2 − 61.8 = 9.4 mol/hr
Heat load of inerts = (970) (7.85) (130 − 125) = 38,100 Btu/hr
Heat load of water vapor = (61.8) (18) (1117.9 − 1115.8)
= 2,340 Btu/hr
Heat of condensation = (9.4)(18)(1117.9 − 92.91)
= 173,500 Btu/hr
Total = 213,940 Btu/hr

Interval 125°F–115°F
Vapor pressure water at 115°F = 1.4709 psia
Enthalpy water vapor at 115°F = 1111.6 Btu/lb
Enthalpy water liquid at 115°F = 82.93 Btu/lb
Pressure of inerts = 32.0 − 1.47 = 30.53 psia

(Note the pressure at this interval is assumed to be 32.0 psia to allow for additional pressure drop.)

Mol water vapor in gas phase at 115°F

$$= \frac{970}{\frac{30.53}{32.0}} \left(\frac{1.47}{32.0}\right) = 46.8 \text{ mol/hr}$$

Mol water condensed in this section = 61.8 − 46.8
= 15.0 mol/hr
Heat load of inerts = (970) (7.85) (125 − 115) = 76,000 Btu/hr
Heat load of water vapor = (46.8) (18) (1115.8 − 1111.6)
= 3540 Btu/hr
Heat of condensation = (15) (18) (1115.8 − 82.93)
= 279,000 Btu/hr
Total = 358,540 Btu/hr

Interval 115°–104°F
Vapor pressure water at 104°F = 1.0695 psia
Enthalpy water vapor at 104°F = 1106.9 Btu/lb
Enthalpy water liquid at 104°F = 71.96 Btu/lb
Pressure of inerts = 31.0 − 1.069 = 29.93 psia

(Note the pressure at the end of this interval is taken at 31 psia.)

Mol water vapor in gas phase at 104°F = 34.6 mol/hr
(from calculation, part 2)
Mol water condensed in this section = 46.8 − 34.6
= 12.2 mol/hr
Heat load of inerts = (970) (7.85) (115 − 104) = 83,800 Btu/hr
Heat load of water vapor = (34.6)(18)(111.6 − 1106.9)
= 2930 Btu/hr
Heat of condensation = 12.2 (18) (111.6 − 71.96)
= 228,000 Btu/hr
Total = 314,730 Btu/lb

Grand total heat load
= 938,700 + 179,640 + 213,940 + 358,540 + 314,730
= 2,005,550 Btu/hr

See Figure 10-88 for heat load curve.

5. Assume a unit,

If U = 100 , and estimate Δt = 30,

$$A = \frac{2,000,000}{(100)(30)} = 670 \text{ ft}^2$$

Try: U-tube bundle (for large temperature differential)
4 tube passes
1-in. tubes, duplex 18 BWG cupro-nickel inside, 18 BWG steel outside on 1 $^1/_4$-in. triangular spacing, 32 ft overall length

Shell I.D. = 25 inches (Table 10-9)
Number of holes in tubesheet = 218
(including allowance for tie rods)
Approximate surface area:
Effective length of 24 ft U-bent tube = 29.5 ft (Figure 10-25).
Effective area = (22.6) (0.2618 ft^2/ft) (109) = 645 ft^2
Number of tubes = 218/2 = 109 ≅ 645 ft^2

Triangular spacing was selected in preference to square primarily due to the large amount of simple gas cooling in this unit.
The I.D. of this plain surface duplex tube is determined by deducting the gage thickness from the O.D. = 0.804 in. I.D. of cupro-nickel.
Use sea water at 7 ft/sec in tubes.
Number of tubes for liquid entrance/pass = 218/4 = 54.5. Use 54.
Note that for U-bent bundles the reverse end of the bent tube acts like a head on a normal fixed tubesheet unit. Therefore, the number of tubes and the number of tube holes are not to be confused in determining velocities.

lb/hr water per pass at 7 ft/sec
= (vel.) (C) (sp. gr.) (number of tubes) (Table 10-3)
C = 793 for tube I.D. = 0.805-in.
lb/hr = (7) (793) (1) (54) = 299,000
gpm = 299,000/500 = 598

6. Temperature rise in cooling water:

$$\Delta T_w = \frac{Q}{c_p(W)} = \frac{2,005,550}{(1)(299,000)} = 6.7°F$$

This rise can be increased and gpm decreased if necessary; however, in brackish and sea water, it is preferable to keep velocity as high as possible. Therefore, design this unit as shown.

7. Tube-side film coefficient (clean):

h_i = 1500(0.95) = 1420 Btu/hr (ft^2) (°F), Figure 10-50A or 10-50B

$$h_{io} = (1,420)\left(\frac{0.804}{1.0}\right) = 1,140 \text{ Btu/hr (ft}^2)(°F)$$

8. Shell-side fluid properties. From previous process calculations, the following properties were determined for the dry gas stream:

k_a = 0.0545 Btu/hr (ft^2) (°F/ft)
μ = 0.0409 lb/ft (hr)
c_p = 0.566 Btu/lb (°F)

For water vapor
k_a = 0.0118 Btu/hr (ft^2) (°F/ft)
μ = 0.0283 lb/ft (hr)
c_p = 0.45 Btu/lb (°F)

Average properties for mixture at average shell temperature:

$$k_a = 0.0545\left(\frac{970}{970 + 78.8}\right) + 0.0118\left(\frac{78.8}{970 + 78.8}\right)$$
$$= 0.0513 \text{ Btu/hr (ft}^2)(°F/ft)$$

$$\mu = 0.0409\frac{(970)(14)}{(970)(14) + 1,420} + 0.0283\frac{1,420}{(970)(14) + 1,420}$$
$$= 0.0397 \text{ lbs/hr (ft)}$$

$$c_p = 0.566\left(\frac{(970)(14)}{(970)(14) + 1,420}\right) + 0.45\left(\frac{1,420}{(970)(14) + 1,420}\right)$$
$$= 0.554 \text{ Btu/lb (°F)}$$

9. Shell side coefficients
a. Gas cooling interval 250°F–135°F.
From equation 10-48 for use with Figure 10-54, assume 18-in. baffle spacing:

$$a_s = \frac{D_s(C'B)}{p(144)} = \frac{(25)(0.25)(18)}{(1.25)(144)} = 0.624 \text{ ft}^2$$

$$G_s = \frac{W}{a_s} = \frac{(970)(14) + 1,420}{0.624} = 24,000 \text{ lb/hr (ft}^2)$$

$$D_e = 0.72/12 = 0.06 \text{ (Table 10-21)}$$

$$R_e = \frac{DG}{\mu} = \frac{(0.06)(24,000)}{0.0397} = 36,200$$

From Figure 10-54 for 25% baffle cut:

j_H = 110

$$h_o = \frac{j_H k}{D_e}\left(\frac{C\mu}{k_a}\right)^{1/3}\left(\frac{\mu}{\mu_w}\right)^{0.14}$$

h_o = 70.8 Btu/hr (ft^2) (°F)

Because no condensation (theoretically) occurs in this section, no heat is diffused through condensate film.

$$h_o(t_g - t_c) = h_{io}'(t_c - t_w') = U(t_g - t_w')$$
$$70.8(250 - t_c) = 1140(t_c - (90 + 6.7))$$
$$t_c = 105.6 \text{ °F}$$

$$U\Delta t = \frac{70.8(250 - 105.6) + 1,140(105.6 - 96.7)}{2}$$
$$= \frac{10,230 + 10,150}{2} = 10,190$$

$$U = \frac{10,190}{250 - 96.7} = 66.4 \text{ Btu/hr (ft}^2)(°F)$$

b. Cooler-condenser zone, diffusivity:

$$k_d = 0.0166 \frac{T_k^{3/2}}{p't(V_A^{1/3} + V_b^{1/3})^2}\left(\frac{1}{M_A} + \frac{1}{M_B}\right)^{1/2}$$

V_A for water vapor = 14.8
V_B for hydrocarbon mixture:

	V_A	Mol Fraction	Weighted V_B
C_2H_2	37	0.148	5.45
H_2	14.3	0.531	7.6
CH_4	29.6	0.098	2.9
CO	22.2	0.223	4.95
			$20.90 = V_B$

Use average temperature of (135 + 104)/2 for $k_d \cong 322°K$

$$k_d = 0.0116 \frac{(322)^{3/2}}{\frac{32.5}{14.7}[(14.8)^{1/3} + (20.9)^{1/3}]^2}\left(\frac{1}{18} + \frac{1}{14}\right)^{1/2}$$

$$k_d = 0.572 \text{ ft}^2/\text{hr}$$

Average mol weight of mixture:

$$M = \frac{(14)(970) + 1{,}420}{970 + 78.8} = 14.33 \text{ lb/mol}$$

Average lb/ft³ of mixture in cooler-condenser zone:

$$\rho = \frac{14.33}{359}\left(\frac{460 + 32}{460 + 120}\right)\left(\frac{32.5}{14.7}\right) = 0.0766 \text{ lb/ft}^3$$

$$\left(\frac{\mu}{\rho k_d}\right)^{2/3} = \left(\frac{0.0397}{(0.0766)(0.572)}\right)^{2/3} = 0.936$$

$$\left(\frac{c_p\mu}{k}\right)^{2/3} = \left(\frac{(0.554)(0.0397)}{0.0513}\right)^{2/3} = 0.569$$

c. Relation between diffusion and heat transfer:

$$K_G = \frac{h_o(c\mu/k)^{2/3}}{cp_{gf}M_m\left(\frac{\mu}{\rho k_d}\right)^{2/3}} \qquad (10\text{-}102)$$

h_o = same as in gas cooling zone = 70.8

$$K_G = \frac{(70.8)(0.569)}{(0.554)(p_{gf})(14.33)(0.936)}$$

$$= \frac{5.41}{p_{gf}} \text{ mol/hr (ft}^2\text{)(atm)}$$

10. Cooler-condenser interval: 135°F–130°F.
*Note: The value of h_{io} is not correct for the condensing coefficient as described in step 6 of this outline. In this case, the error is small, but that is not necessarily true for other situations.

t_g = 135°F
p_v = 2.537 psia at 135°F
p_g = 32.5 − 2.537 = 29.96

$$t_w = 96.7 - \frac{938{,}700}{(299{,}000)(1)} = 93.6°\text{F}$$

Note that this assumes that water is warmed from the gas cooling before it enters this section. This is not rigorously correct.

Δt = 135 − 93.6 = 41.4°F
Latent heat at 135°F = 1,017.0 Btu/lb = λ
Assume temperature of condensate, t_c = 100°F
then: p_c = 0.949 psia (steam tables)
p_g' = 32.5−0.949 = 31.55 psia

$$p_{gf}' = \frac{p_g' - p_g}{2.3 \log \frac{p_g'}{p_g}} = \frac{31.55 - 29.96}{2.3 \log \frac{31.55}{29.96}} = 31.7 \text{ psia}$$

$$h_o (t_g - t_c) + K_G M_v \lambda (p_v - p_c) =$$

$$h_{io}' (t_c - t_w') \quad 70.8(135 - 100)$$

$$+ \left[\frac{5.41}{\left(\frac{31.7}{14.7}\right)}\right](18)(1{,}017.0)\left(\frac{2.537 - 0.949}{14.7}\right)$$

$$= 1{,}140(100.0 - 93.6)$$

$$2{,}480 + 4{,}960 = 7{,}300$$

$$7{,}440 = 7{,}300$$

(See the preceding note regarding h_{io}').
Close enough check.

$$U\Delta t = \frac{7{,}440 + 7{,}300}{2} = 7{,}370$$

$$U = \frac{U\Delta t}{t_g - t_w} = \frac{73.70}{135 - 93.6} = 177 \text{ Btu/hr (ft}^2\text{)(°F)}$$

Interval 130°F–125°F
t_g = 130°F
p_v = 2.223
p_g = 32.5 − 2.223 = 30.277 psia

$$t_w = 93.6 - \frac{179{,}640}{(299{,}000)(1)} = 93.0°\text{F}$$

Flow gas-vapor rate = (970)(14) + (61.8)(18) = 14,713

$$G_s = \frac{W}{a_s} = \frac{14{,}713}{0.624} = 23{,}600 \text{ lb/hr (ft}^2\text{)}$$

Because this rate is so close to the previous G_s of 24,000, it is reasonable to assume that the h_o value will be close also. Use h_o = 70.0.

Latent heat, λ, at 130°F = 1,020.0 Btu/lb
Assume condensate temperature, t_c = 98.5°F
P_c at 98.5°F – 0.907 psia
P_g = 32.5 – 0.907 = 31.593

$$P_{gf} = \frac{31.593 - 30.27}{2.3 \log \frac{31.593}{30.27}} = 31.6$$

$$h_o = (t_g - t_c) + K_G M_V \lambda\ (p_v - p_c) = h_{io}'\ (t_c - t_w')$$

(See the previous note regarding h_{io}'.)

$$70(130 - 98.5) + \left[\frac{5.41}{\frac{31.6}{14.7}}\right](18)(1{,}020)\frac{(2.223 - 0.907)}{14.7} = 1{,}140(98.5 - 93)$$

2,210 + 4,140 = 6,270
6,350 = 6,270

Close enough check (slightly > 1%).

$$U\Delta t = \frac{6{,}350 + 6{,}270}{2} = 6{,}310$$

$$U = \frac{6{,}310}{130 - 93} = 170 \text{ Btu/hr (ft}^2\text{)(°F)}$$

Interval 125°F–115°F
t_g = 125
p_v = 1.942 psia at 125°F
p_s = 32.0 – 1.942 = 30.058 psia

$$t_w = 93.0 \text{ (from previous interval)} - \frac{213{,}940}{(299{,}000)(1)} = 92.3\text{°F}$$

New gas rate = (970)(14) + (46.8)(18) - 14,442

$$G_s = \frac{14{,}442}{0.624} = 23{,}100 \text{ lb/hr (ft}^2\text{)}$$

This is still not enough change in G_s from 24,000 to justify a calculation of a new h_o. Use of the water vapor flow leaving the assumed interval means the actual G_s through that interval will be equal to or greater than the value for conditions at 115°F. This value is on the safe side for h_o.

Latent heat at 125°F, λ = 1022.9 Btu/lb
Assume condensate temperature of t_c = 97.0°F

p_c = 0.867 psia at 97°F
p_g' = 32.0 – 0.867 = 31.133 psia

$$p_{gf} = \frac{31.133 - 30.058}{2.3 \log \frac{31.133}{30.058}} = 30.2 \text{ psia}$$

$$70(125 - 97) + \left[\frac{5.41}{\left(\frac{30.2}{14.7}\right)}\right](18)(1{,}022.9)\left(\frac{1.942 - 0.867}{14.7}\right)$$

$$= 1{,}140(97 - 92.3)$$

1,960 + 3,540 = 5,350
5,500 = 5,350

Close enough check.

$$U\Delta t = \frac{5{,}500 + 5{,}350}{2} = 5425$$

$$U = \frac{5{,}425}{125 - 92.3} = 166 \text{ Btu/hr (ft}^2\text{)(°F)}$$

Interval 115°F–104°F
Point at 104°F
t_g = 104°F
p_v = 1.069 psi at 104°F
p_g = 31.0 – 1.069 = 29.931 psia

$$t'_w = 91.1 - \frac{314{,}730}{299{,}00(1)} = 90.05\text{°F, close enough check to 90°F}$$

New gas rate = 970(14) + 34.6(18) = 14,223 lb/hr

Note that this is the same as the previous interval; the difference is discussed there. Using Kern's outline, this effect is as discussed and not as demonstrated here. In this case, the difference is negligible.

Use h_o = 65
Latent heat at 104°F = 1034.9 Btu/lb
Assume condensate temperature, t_c, of 91.7°F
p_c = 0.736 psia at 92°F
p'_g = 3.10 – 0.736 = 30.264 psia

$$p_{gf} = \frac{30.264 - 29.931}{2.3 \log \frac{30.264}{29.931}} = 30.0$$

$$65(104-92) + \left[\frac{5.41}{\frac{30.0}{14.7}}\right](18)(1{,}034.9)\frac{1.069 - 0.736}{14.7}$$

$$= 1{,}140(91.7 - 90)$$

800 + 1120 = 1940
1920 = 1940

checks:

$$U\Delta t = \frac{1{,}920 + 1{,}940}{2} = 1930$$

$$U = \frac{1{,}930}{104 - 90} = 138 \text{ Btu/hr (ft}^2)(^\circ\text{F})$$

$t_g = 115°F$
$p_v = 1.471$ psi at 115°F
$p_g = 31.0 - 1.471 = 29.529$

$$t_w' = 92.3 - \frac{358{,}540}{299{,}000(1)} 91.1^\circ\text{F}$$

New gas rate = 970(14) + 34.6(18) = 14,223 lb/hr

$$G = \frac{14{,}223}{0.624} = 22{,}800 \text{ lb/hr (ft}^2)$$

This is only 5% of the original G_s and still within normal expected accuracy of this method. For improved accuracy, the physical properties, R_e, h_o, and diffusivity K_G, could be recalculated for the conditions of this interval. This particular problem does not seem to warrant the extra work, although many problems may require a recalculation at every interval or two.

Use $h_o = 65$
Latent heat at 115°F = 1028.7 Btu/hr
Assume condensate temperature, t_c, of 94.5°F
$p_o = 0.802$ psia at 94.5°F
$p_g' = 31.0 - 0.802 = 30.198$

$$p_{gf} = \frac{30.198 - 29.529}{2.3 \log \frac{30.198}{29.529}} = 30.03$$

$$65(115 - 94.5) + \left[\frac{5.41}{\frac{30.03}{14.7}}\right](18)(1{,}028.7)\frac{1.471 - 0.802}{14.7}$$

$$= 1{,}140(94.5 - 91.1)$$

$$1330 + 2220 = 3880$$

$$3550 = 3880$$

Close enough check.

$$U\Delta t = \frac{3{,}550 + 3{,}880}{2} = 3{,}715$$

$$U = \frac{3{,}715}{115 - 91.1} = 155 \text{ Btu/hr (ft}^2)(^\circ\text{F})$$

11. Areas required:

a. Gas cooling zone

LMTD: 250 → 135
96.7 ← 93.6
153.3   41.4
LMTD = 85°F

Correction factor:

$$R = \frac{250 - 135}{96.7 - 93.6} = 37.1$$

$$P = \frac{96.7 - 93.6}{250 - 93.6} = 0.0198$$

F = 0.975 (approx., Figure 10-34)
Corrected LMTD = 85 (0.975) = 82.8°F
Clean U = 66.4 (Part 9)
Fouling assumed = 0.004
(overall, 0.0015 tube side, 0.0025 shell side)

$$U(\text{dirty}) = \frac{1}{\frac{1}{66.4} + 0.004} = 52.3 \text{ Btu/hr (ft}^2)(^\circ\text{F})$$

$$\text{Area required this zone} = \frac{938{,}700}{(52.3)(82.8)} = 216.0 \text{ ft}^2$$

b. Cooler-condenser zone, see plot of Figures 10-87 and 10-88.

$$\text{Weighted } \Delta t = \frac{Q}{\left(\frac{\sum q}{\Delta t_{avg.}}\right)} = \frac{1{,}066{,}850}{39{,}990} = 26.7$$

$$U(\text{clean}) = \frac{Q}{A\Delta t} = \frac{1{,}066{,}850}{(267.9)(26.7)} = 149$$

Same fouling applies to this portion of the unit = 0.004

$$U(\text{dirty}) = \frac{1}{\frac{1}{149} + 0.004} = 93.4 \text{ Btu/hr (ft}^2)(^\circ\text{F})$$

Area required for cooler-condenser zone:

$$A = \frac{1{,}066{,}850}{(93.4)(26.7)} = 427 \text{ ft}^2$$

Total exchanger area:
Gas cooling zone = 216
Cooler-condenser zone = 427
Total = 643 ft²

Available area in assumed unit = 645 ft²

$$\text{Percent excess area} = \frac{(645 - 643)(100)}{629} = 0.3\%$$

This excess area is smaller than usual practice; 10% being a preferred figure. The accuracy of the relations used are not claimed to be better than 20%

Interval	t_g (°F)	t_o (°F)	$1/U\Delta t$	$(1/U\Delta t)$ avg.*	Δq (Btu/hr)	$\Delta A = \frac{\Delta q}{(U\Delta t)\text{ avg.}}$ (ft²)	Δt	$\Delta t_{avg.}$ (°F) *	$\frac{\Delta q}{\Delta t_{avg.}}$
135 to	135	100.0	0.0001359	...	...	...	41.4	...	...
130 to	130	98.5	0.0001585	0.0001472	179,640	26.5	37.0	39.2	4,580
125 to	125	97.0	0.0001842	0.0001713	213,940	36.6	32.7	34.8	6,140
115 to	115	94.5	0.0002695	0.0002268	358,540	81.2	23.9	28.4	12,620
104	104	91.7	0.000518	0.0003937	314,730	123.6	14.0	18.9	16,650
					1,066,850	267.9			39,990

*If the interval becomes large, use the log mean average.

(in some situations, and these are indeterminate without prior experience). For the uncertainties of this type of problem, a preferred and considerably safer unit would be:

Shell I.D. = 25 in. (same)
Tube passes = 4 (same)
Tubes: 1-in. duplex (same)
Tube length: 32 ft (compared to assumed 24 ft)
Number of tubes: 109 (same)
Effective area: 870 ft² (645 ft²)

Multizone Heat Exchange

To select the proper heat transfer relations to represent the functions, you need to analyze the heat transfer functions that will take place in the unit-tube and/or on the shell side. Some units may have several functions, such as the example in Rubin's[179] recommendations on this subject; that is, steam desuperheating and hydrocarbon condensing; steam and hydrocarbon condensing, and condensate subcooling. Rubin[180] presents an excellent interpretation of multizone operation for several different sets of conditions. See Figures 10-91A and 10-91B.

The presence of even a small amount of noncondensable gas in the condensing mixture can significantly reduce the condensing heat transfer rates and needs to be recognized. See Figure 10-85.

Fluids in Annulus of Tube-in-Pipe or Double Pipe Exchanger, Forced Convection

This unit consists of two pipes or tubes, the smaller centered inside the larger as shown in Figure 10-92. One fluid flows in the annulus between the tubes; the other flows inside the smaller tube. The heat transfer surface is considered as the outside surface of the inner pipe. The fluid film coefficient for the fluid inside the inner tube is determined the same as for any straight tube using Figures 10-46–10-52 or by the applicable relations correcting to the O.D. of the inner tube. For the fluid in the annulus, the same relations apply (Equation 10-47), except that the diameter, D, must be the equivalent diameter, D_e. The value of h obtained is applicable directly to the point desired — that is, the outer surface of the inner tube.[70]

$$D_e = \frac{D_2^2 - D_1^2}{D_1} = 4r_h = \frac{4(\text{flow area})}{(\text{wetted perimeter})} \tag{10-127}$$

where
D_1 = O.D. of inner tube, ft
D_2 = I.D. of outer pipe, ft
r_h = hydraulic radius, ft = (radius of a pipe equivalent to the annulus cross-section)

Approximation of Scraped Wall Heat Transfer

Little data is available for estimating the inside film coefficient for vessels or heat exchangers, heated externally by steam in a jacket and with a continuous moving inside wall scraper to clear away the heavy, viscous inside wall film. The exact heat transfer will vary with the unit design and speed of rotation of the scraper. The studies of Ramdas et al.[95] indicate that to some extent the mixing of the warmer and lower viscosity wall fluid with the cooler and higher viscosity is a significant part of the limitation of overall heat transfer to the fluid mass and that heat transfer by conduction in the bulk fluid is controlling. They conclude that slow scraping of the wall may be better than no scraping, but beyond a certain limit, the scraper speed provides little film heat transfer improvement. For laminar flow conditions, which quite often apply, the correlation developed is somewhat unique for the Votator design but should certainly establish a good guide as to what to expect from other designs.

$$N_u = 57Re_f^{0.059}\, Re_f^{0.113}\, P_r^{0.063}\, V_{isr}^{-0.018} \tag{10-128}$$

where (terms are all in consistent units)

$$\left.\begin{aligned} Nu &= h_s D_r/k \\ Re_r &= ND_t\,\rho/\mu \\ Re_f &= 4W/(\pi\mu)\ (D_t + D_s) \\ P_r &= C_p\,\mu/k \\ V_{isr} &= \mu/\mu w \\ &0.0016 < Re_f < 9.23 \\ &0.0164 < Re_r < 68.65 \end{aligned}\right\} \text{dimensionless groups}$$

h_s = film heat transfer coefficient from correlation
D_t = tube diameter
k = thermal conductivity of process fluid
N = shaft speed
ρ = density of process fluid
μ = bulk viscosity of process fluid
W = mass continuous throughput flow rate of process fluid
π = 3.1416
D_s = shaft diameter
C_p = specific heat of process fluid
μ_w = viscosity of process fluid at wall

DWG. NO. A ____

Item No. ____
Job No. ____
Charge No. ____

By ____

Date:- ____

EXCHANGER RATING

Apparatus *Cooler - Condenser E-C1* Plant *Gas*

Min. Req. Eff. O. S. Area Exposed in Shell, Sq. Ft. ____ Outside *870* Inside: *—*

Number of Units: Operating *One* Spares *none*

DESIGN DATA PER *Unit*		
UNIT DATA	**SHELL SIDE**	**TUBE SIDE**
Fluid	*Light Hydrocarbon with water*	*Sea water*
Fluid Flow Lbs./Hr.	*13,590*	*299,000 (598 GPM)*
Temperature In °F.	*250*	*90*
Temperature Out °F.	*104*	*96.7*
Operating Pressure PSIA	*34 psia at inlet*	*60 psig*
Density Lbs./CF	*Variable*	
Specific Heat *Dry Gas* Btu/Lb./°F.	*7.85*	
Latent Heat Btu/Lb.	*Variable*	
Therm. Cond. Btu/Hr./Sq. Ft./°F./Ft.	*0.0545 (Dry Gas) 0.0118 (H_2O)*	
Viscosity *Lbs/Ft/Hr* ~~Centipoise~~	*0.0409*	
Molecular Weight	*14 (Dry Gas)*	
No. of Passes	*1*	*4*
Pressure Drop PSI	Calc: *3.5* Used: *4*	Calc: Used: *ok*
Fouling Factor	*.0025*	*0.0015*

Heat Transferred - BTU/Hr: *2,005,550*

LMTD *82.8 & 26.7* °F. Overall U: Calc. *52.3 (Gas Cooling); 96.8 (Cond Cond)* Used *25 (Gas Cooling); 69.0 (Cooler Cond)*

CONSTRUCTION				
Max. Oper. Pressure	*50*	*Design 75* PSI	*60*	*Design 75* PSI
Max. Oper. Temperature	*300*	*Design 450* °F.	*115*	*125* °F.

Type of Unit *U - Bundle* Tube Pitch *1 1/4" Trian.* Joint *Roll & Bead*

Tubes - Material: *Duplex, Cu-Ni-Steel* No. (Approx.) *109* O.D. *1* BWG ____ Length *32' overall*

Shell - Material: *Steel* Diameter (Approx.): *25" I.D.*

Channel Material: *Steel* Supports Material: *Steel*

Tube Sheet Material: *Monel-Clad (1/4" Min) Steel* Baffle Material: *Steel*

Corrosion Allowance - Shell Side *1/16"* Tube Side *1/16"*

Connections - Shell In: *14"* Out *12"* Flange *150# ASA - RF*

Channel In: *6"* Out *6"* Flange *150# ASA - RF*

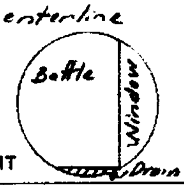
Others *①* Size *1"* Flange *3000 # Coupling*

Bolts: *Alloy steel* Gaskets: ____

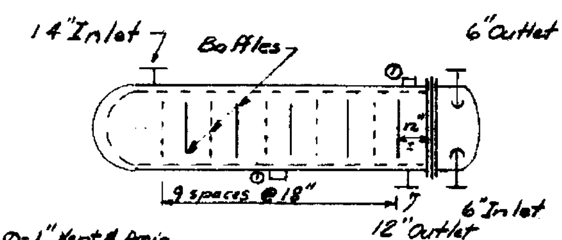
Code *TEMA - C* Stamp *Yes* X-Ray *—* SR *—*

Insulation *None* Class *—* Cathodic Protection *Yes*

Baffles are 25% Vertical Cut
U-Tubes are bent around horizontal centerline, use minimum radius for duplex tubes.
Cut horizontal drainage section from bottom of each baffle on centerline of bottom tube.



BAFFLE ARRANGEMENT



Horizontal Elevation
NOZZLE ARRANGEMENT

Remarks:- ** Tubes to be 18 BWG Cupronickel Inside, 18 BWG outside with Cupro-nickel ferrules on ends at tube sheet.*
Note: Design requires all gas and condensate removal out bottom. For top removal of gas-vapor the 12" outlet would be rotated 180°, and 2" drain added.

Checked ____ Date ____ Approved ____ Date ____

Rev. ____ By ____ Date ____ B/M No. ____

Figure 10-90. Exchanger rating for cooler-condenser.

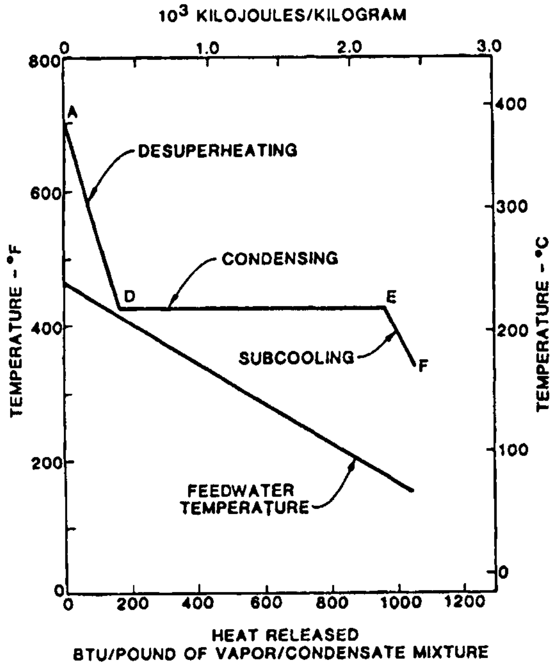


Figure 10-91A. High pressure boiler feedwater multizone condenser: heat release curve. Pressure drop is assumed to be negligible. Hot fluid (steam side) inlet at A. Condensing completed and submerged surface subcooling begins at E. Counter-current flow steam and condensate temperatures versus heat released. Steam 350 psig inlet, 700°F, 1365.5 Btu/lb; 350 psia saturated 431.72°F, 1203.9 Btu/lb. Condensate out 340°F. Desuperheating in the 268.3°F temperature range, 161.6 Btu/lb. Condensing at 431.72°F, 794.2 Btu/lb. Subcooling in the 91.72°F temperature range, 986.0 Btu/lb. Total heat removed 1054.4 Btu/lb. (Used by permission: Rubin, F. L. *Heat Transfer Engineering*, V. 3, No. 1, p. 49, ©1981. Taylor and Francis, Inc., Philadelphia, PA. All rights reserved.)

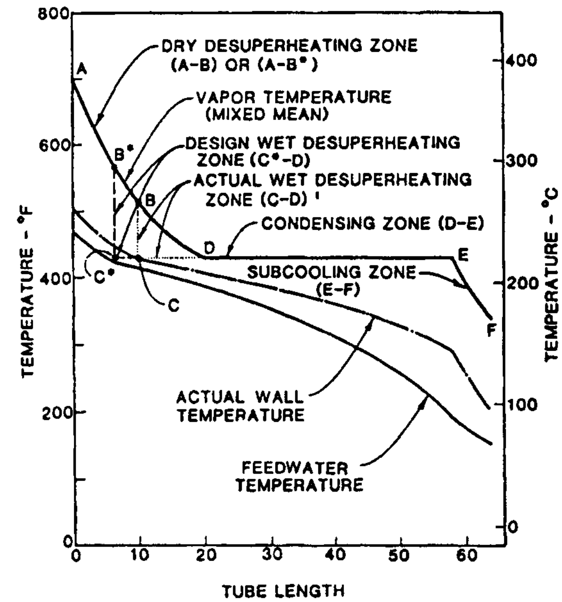


Figure 10-91B. Steam and condensate temperatures versus condenser length. Temperature distribution curve for the same multizone condenser as in Figure 10-91A. Points A, E, and F are the same. Point B is above C, which locates the start of the wet desuperheating zone on the tube surface. (Used by permission: Rubin, F. L. *Heat Transfer Engineering*, V. 3, No. 1, p. 49, ©1981. Taylor and Francis, Inc., Philadelphia, PA. All rights reserved.)

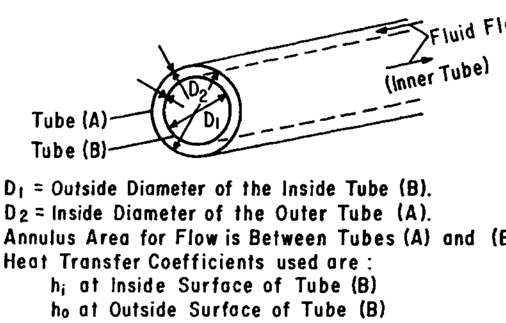


Figure 10-92. Double-pipe tube arrangement showing annulus area.

Interpreting the plotted data from the authors' tests indicates that heat transfer film coefficients at the scraped wall might be expected to range:

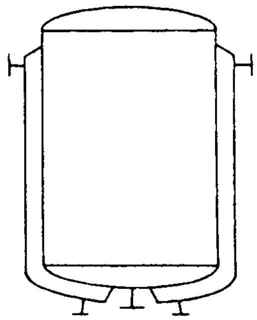
For heating: 20–40 Btu/hr/(ft^2) (°F)
For cooling: 10–30 Btu/hr/(ft^2) (°F)

The test data represented corn syrup, red oil, and golden oil, (API = 19.7 at 60°F)

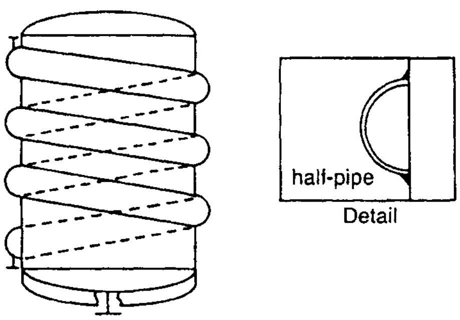
Heat Transfer in Jacketed, Agitated Vessels/Kettles

The heat transfer that is achieved in externally jacketed kettles used for reaction and/or mixing and heating/boiling/cooling varies considerably with the style of jacket. Jackets may be one piece open chambers surrounding the main shell of the vessel, or they may be coil style, usually of the half-pipe design (see Figures 10-93A and 10-93B). The half-pipes are continuously welded to the shell and may be grouped in segments or sections of the shell to allow for the rather rapid conversion of a section from external heating to external cooling, as is so often the required condition for some batch reaction processes. Expected heat transfer overall coefficients for *estimating* typical organic processes in the vessel with steam, water, or a cooling methanol-water mixture in the jackets are as follows:

	Overall, U, Btu/hr(ft^2) (°F)	
	Steam heating	**Cooling**
Open jacket	25–55	25–40
Coils, half-pipe	25–80	30–55



(a) External open jacket, minimum jacket baffling



(b) External half-pipe coils

Figure 10-93A. Typical vessel external jackets for heat transfer.

Vessel with Coils

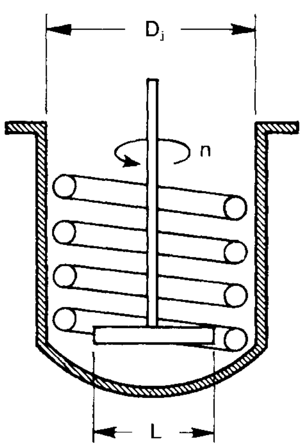


Figure 10-93B. Process vessel with internal coil and agitation to improve heat transfer. (Used by permission: *Engineering Manual Dowtherm™ Heat Transfer Fluids*, ©1971. The Dow Chemical Co.)

Baker and Walter[3] report tests performed on open jacketed agitated vessels and published some of the limited results for this type of equipment. These data indicate the effects of jetting the fluid at various velocities into the jacket.

Chapter 5, "Mixing of Liquids," in Volume 1, 3rd Edition, of this set provides more details and various vessel heat transfer mechanisms (outside jackets and/or half-pipes, internal vertical pipes, and internal coils). Jacketed vessels (external) are often used for chemical reaction temperature control and for the heating or cooling mixture being agitated in vessels. Internal vertical or horizontal coils in a vessel can also be used for temperature control. Gruver and Pike[163] recommend that the use of a single fluid in the jackets/coils is better than requiring complicated controls to switch types of fluids.

See Figures 10-93A and 10-93B as limited examples of reaction and other process vessels that require heat transfer for proper processing. Markovitz[202] reports improved heat transfer for the inside of jacketed vessels when the surface has been electropolished, which gives a fine, bright surface.

Heat transfer in *agitated* vessels with internal coils containing the heat transfer fluid (process on outside of coil) is expressed by the outside coefficient on coils[183]

$$\frac{h_c D_j}{k} = 0.87 \frac{(60nDa^2\rho)^{0.62}}{(\mu)} \frac{(c_p\mu)^{1/3}}{(k)} \frac{(\mu)^{0.14}}{(\mu_w)} \quad (10\text{-}129)$$

and as shown in Figure 10-94.

For heat transfer fluids inside reactor jackets or other process vessels with agitation to fluids in vessels (Figure 10-93A), the heat transfer is expressed[183] as

$$\frac{h_j D_j}{k} = 0.36 \frac{(60nDa^2\rho)^{2/3}}{(\mu)} \frac{(c_p\mu)^{1/3}}{(k)} \frac{(\mu)^{0.14}}{(\mu_w)} \quad (10\text{-}130)$$

and in Figure 10-94. Also see references 282 and 283.

where

h_c = average film coefficient, clean, Btu/(hr) (ft^2) (°F)
D_i or D_j = I.D. of vessel, ft
Da = diameter of agitator, ft
k = thermal conductivity of fluid processed, Btu/(hr) (ft^2) (°F/ft)
n = rev/min of agitator
L = length and thickness, ft, of coil or jacket
c_p = specific heat, Btu/(lb) (°F)
μ = viscosity = cps × 2.42, lb/(hr) (ft)
ρ = density, lb ft^3
r = fouling resistance, or tube resistance, (hr) (ft^2) (°F)/Btu

Subscripts:

w = wall
j = jacket side
c = clean
i = inside

Example 10-16. Heating Oil Using High Temperature Heat Transfer Fluid[183] (used by permission of The Dow Chemical Co., reference [183]©1971)

You want to heat 9,000 lb/hr of oil from 500°F (260°C) to 600°F (315.56°C). The oil is heat-sensitive and cannot be heated to more than 630°F (322.22° C). Condensing

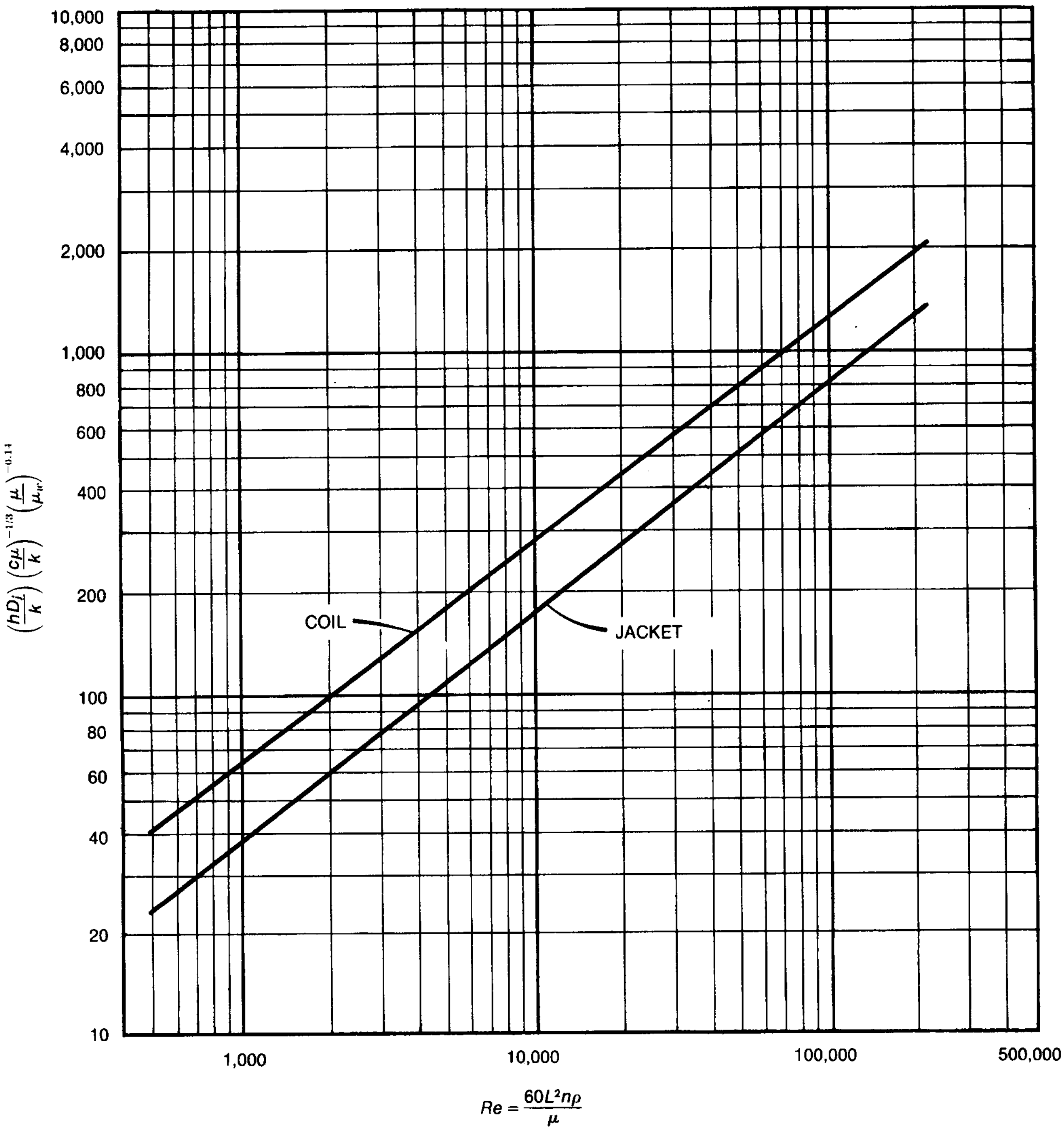


Figure 10-94. Heat transfer coefficients for jackets and coils with fluid agitation. (Used by permission: *Engineering Manual Dowtherm™ Heat Transfer Fluids*, ©1971. The Dow Chemical Co.)

DOWTHERM A will be used at 620°F (326.67°C) in a horizontal tubular heat exchanger.

Assumptions:

- The exchanger will have one shell pass and one tube pass.
- DOWTHERM A will be on the shell side of the exchanger, and no subcooling will take place.
- The film coefficient for the oil, h_i = 360 Btu/(hr) (ft^2) (°F).
- The fouling factors for the oil will be
  r_i = 0.003 (hr)(ft^2) (°F)/Btu, for DOWTHERM A
  r_o = 0.001 (hr)(ft^2) (°F)/Btu
- Tubes in the exchanger will be $^3/_4$-in. O.D., 16 BWG steel.
- Oil specific heat, c_p = 0.40 Btu/lb/°F.

1. *Heat Balance*

A. Oil

$$Q = mc_p\Delta t = (9{,}000)(0.4)(600 - 500) = 360{,}000 \text{ Btu/hr}$$

B. DOWTHERM A

$Q = (W)(\lambda)$ or $W = Q/\lambda$

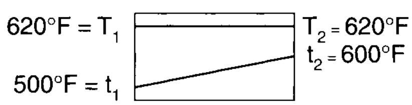
$Q_{\text{DOWTHERM}} = Q_{\text{Oil}} = 360{,}000$ Btu/hr

$\lambda_{620°F} = 111.3$ Btu/lb (physical property data)

$W = 360{,}000/111.3 = 3235$ lb/hr

g = condensate flow = 8.3 gal./min

2. *Log Mean Temperature Difference* (LMTD)



$$\text{LMTD} = \frac{\Delta t_1 - \Delta t_2}{2.3 \log \dfrac{\Delta t_1}{\Delta t_2}}$$

$\Delta t_1 = 620 - 500 = 120°F$

$\Delta t_2 = 620 - 600 = 20°F$

$$\text{LMTD} = \frac{120 - 20}{2.3 \log_{10}(120/20)}$$

$$= \frac{100}{(2.3)(0.778)} = 55.9°F$$

LMTD = 56 °F

3. *Clean Overall Heat Transfer Coefficient* (U_c)

$1/U = 1/h_i + 1/h_o + L/k + r_i + ro$

To determine a clean overall heat transfer coefficient, let r_i, r_o, and L/k equal zero. Thus,

$$\frac{1}{U_c} = \frac{1}{h_i} + \frac{1}{h_o} \tag{10-131}$$

A. Film Coefficient on Outside (h_o)

To calculate the outside film coefficient, you need to know the difference in temperature of the condensing vapor (T_v) and the pipe wall temperature (t_w). The pipe wall temperature is determined by trial-and-error calculations using the following equation:[70]

$$t_w = t_a + \left(\frac{h_o}{h_{io} + h_o}\right)(T_v - t_a) \tag{10-132}$$

where

t_w = wall temperature, °F
t_a = average temperature of cold fluid, °F
T_v = temperature of vapor, °F
$t_a = (500 + 600)/2 = 550°F$
$T_v = 620°F$

h_{io} . . . Basing the coefficient on the outside area, h_i may be converted from inside to outside as follows:

$$h_{io} = h_i \times \frac{D'_i}{D'_o} \tag{10-133}$$

$$\frac{D'_i}{D'_o} = \frac{0.620}{0.750} = 0.827$$ (tube properties, see Table 10-3)

h_i = 360 Btu/(hr) (ft²) (°F) (given)
$h_{io} = 360 \times 0.827 = 298$ Btu/(hr) (ft²) (°F)

To solve the preceding equation for t_w, assume h_o = 150 Btu/(hr) (ft²) (°F).
Thus,

$$t_w = 550 + \frac{(150)}{150 + 298}(620 - 550)$$

$t_w = 550 + (0.335)(70) = 573°F$

Now determine h_o from Figure 10-67B:

when

$\Delta t = T_v - t_w = 620 - 573 = 47°F$
$D' = 0.750$ in.
$(D')(\Delta t) = (0.750)(47) = 35.3$
From the graph, h_o = 270 Btu/(hr) (ft²) (°F)

Because the assumed value does not agree with the calculated value, assume h_o = 290, and repeat the calculations.

$$t_w = 550 + \frac{290}{290 + 298}(620 - 550)$$

$t_w = 550 + (0.49)(70) = 584°F$

and $\Delta t = T_v - t_w = 620 - 584 = 36°F$

$(D')(\Delta t) = (36.0)(0.750) = 27.0$ and

h_o = 290 Btu/(hr) (ft²) (°F)

This is the design value for h_o as the assumed value equals the calculated value.

B. Clean Overall Heat Transfer Coefficient (U_c)

$1/U_c = 1/h_{io} + 1/h_o$
$1/U_c = 1/298 + 1/290$
$1/U_c = 0.0036 + 0.00345 = 0.00681$
U_c = 147 Btu/(hr) (ft²) (°F)

Note: If the metal resistance (r_w) had been considered, it would equal 0.00024 (hr) (ft²) (°F)/Btu (Table 10-13A) and would have made U_c = 142 Btu/(hr) (ft²) (°F). The metal resistance is thus shown to be insignificant and may be neglected.

4. *Design Overall Heat Transfer Coefficient* (U_d)

$$\frac{1}{U_d} = \frac{1}{U_c} + r_{io} + r_o$$

$$r_{io} = r_i \times \frac{d_i}{d_o} = 0.003 \times 0.827$$

$$r_{io} = 0.00248\ (hr)\ (ft^2)\ (°F)/Btu$$

$$r_o = 0.001\ (hr)\ (ft^2)\ (°F)/Btu\ (given)$$

$$1/U_c = 1/147 = 0.00681\ (hr)\ (ft^2)\ (°F)/Btu$$

$$1/U_d = 0.00681 + 0.00248 + 0.00100$$

$$1/U_d = 0.01029\ (hr)\ (ft^2)\ (°F)/Btu$$

$$U_d = 97.2\ Btu/(hr)\ (ft^2)\ (°F)$$

or if read from Figure 10-39 at ($r_o + r_{io}$):

$$U_d = 97\ Btu/(hr)\ (ft^2)\ (°F)$$

5. *Surface Area*

$$Q = U_d A \Delta t_{LM}$$

$$A = Q/(U_d)(\Delta t_{LM}) = \frac{360{,}000}{(97.2)(55.9)} = 66.3\ ft^2$$

Pressure Drop

When a fluid flows over a stationary or moving surface, the pressure of the fluid decreases along the length of the surface due to friction. This is commonly called the pressure drop of the system. Of particular interest are the pressure drops in pipes (tubes) and in heat exchanger shells.

The Sieder and Tate equation for the pressure drop in tubes is

$$\Delta p = \frac{fG^2Ln'}{5.22(10)^{10}(D_i)(s)(\mu/\mu_w)^{0.14}}$$

The Sieder and Tate equation for the pressure drop in shells is

$$\Delta p = \frac{fG^2D_i(N+1)}{5.22(10)^{10}(D_e)(s)(\mu/\mu_w)^{0.14}}$$

Values of f versus Re number are given in Figure 10-140.

Pressure drops from DOWTHERM A heat transfer media flowing in pipes may be calculated from Figure 10-137. The effective lengths of fittings, etc., are shown in Chapter 2 of Volume 1. The vapor flow can be determined from the latent heat data and the condensate flow. With a liquid system, the liquid flow can be determined using the specific heat data.

In the design of all parts of a system, special consideration should be given to the large amount of flash vapor liberated on the reduction of pressure. Because of the high ratio of specific heat to latent heat, much more flash vapor is liberated with DOWTHERM A than with steam. Consequently, all constrictions that would cause high pressure drops should be avoided.

In addition to steam and controlled-temperature water, a number of different heat transfer fluids for a wide range of temperatures from 100–700°F are supplied by (a) the Dow Chemical Co., (b) Monsanto Chemical Co., (c) Multitherm Corp., (d) Union Carbide Corp., (e) Exxon Chemical Co., (f) Mobil Chemical Co., (g) Calfo division of Petro Canada, and (h) others with qualified products.

Falling Film Liquid Flow in Tubes

The liquid runs in a film-like manner by gravity down the inner walls of the vertical tubes in a falling film exchanger. The tubes do not run full, and, therefore, the film coefficient is greater than for the same liquid rate in a full tube by[81]

$$h(\text{film gravity}) = h(\text{full})\left(\frac{D_i^2}{4d'(D_i - d')}\right) \quad (10\text{-}134)$$

where

D_i = I.D. of tube, ft
d' = film thickness, ft

For water:[81]

$$h_m = 120\left(\frac{w}{\pi D_i}\right)^{1/3} = 120(G')^{1/3} \quad (10\text{-}135)$$

where

G' = mass flow rate/unit circumference, lb/(hr) (ft)
G' = w/(D
w = mass flow rate, lb/hr

For other liquids[81] in turbulent flow:

$$\frac{h_m}{\sqrt{k_f^3\rho^2 g/\mu_f^2}} = 0.01\left(\frac{c\mu}{k}\right)^{1/3}\left(\frac{4G'}{\mu_f}\right)^{1/3} \quad (10\text{-}136)$$

Properties are evaluated at the length mean average temperature.

where

μ_f = viscosity of liquid at film temperature, t_f, lb/(hr) (ft)
g = acceleration due to gravity = 4.17×10^8 ft/(hr)
ρ = lb/ft³

Table 10-24
Allowable Water Velocities in Tubes

Fluid	Tube Material	Minimum* Velocity, ft/sec	Maximum Velocity, ft/sec	Preferred Velocity, ft/sec
Sea water	70-3-Cupro-nickel; 0.5% Iron	2.5–3	12	6–8
Sea water	90-10-Cupro-nickel; 1.25% Iron	2.5–3	10	6–8
Sea water	Aluminum brass	2.5–3	8	5–6
Brackish water	Steel	2.5	5	4
Treated well water	Steel	2.5	8–10	5–6
Cooling tower recirculated water	Steel	2.5	8	6

*Do not design below these values.

The coefficient, h_m, is to be used with the length mean Δt.
In streamline flow, $4\,G'/\mu_f < 2{,}000$:

$$\frac{h_a}{\sqrt{k_f^3\rho^2 g/\mu_f^2}} = 0.67\left(\frac{c\mu_f^{5/3}}{k_a L\rho^{2/3} g^{1/3}}\right)^{1/3}\left(\frac{4G'}{\mu_f}\right)^{1/9} \tag{10-137}$$

where h_a = film coefficient based on arithmetic mean Δt

Table 10-24 is an experience guide for reasonable service using the types of water indicated inside tubes of the material listed.

Sinek and Young[160] present a design procedure for predicting liquid-side falling film heat transfer coefficients within 20% and overall coefficients within 10%.

Vaporization and Boiling

Boiling of liquids occurs as nucleate or as film boiling. Figures 10-95A and 10-95B illustrate a typical flux curve for water and hydrocarbons. In the region 1–2, the liquid is being heated by natural convection; in 2–3 the nucleate pool boiling occurs with bubbles forming at active sites on the heat transfer surface, natural convection currents set up, Q/A varies at Δt^n where n is 3–4, and the peak flux is at point 3 corresponding to the critical Δt for nucleate boiling; at 3 film boiling begins; and at 4–5–6 film boiling occurs. In film boiling heat is transferred by conduction and radiation through a film on the heating surface. Note that the rate of effective heat transfer decreases beyond point 3, and it is for this reason that essentially all process heating/boiling equipment is designed to operate to the left of point 3.

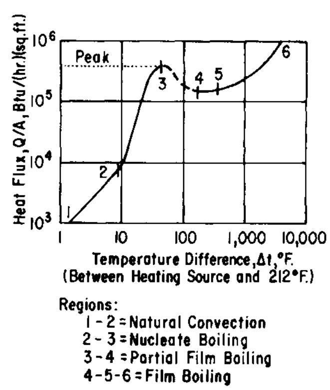


Figure 10-95A. Heat flux for boiling water at 212°F. (Used by permission: McAdams, W. H. *Heat Transmission,* 3rd Ed., ©1954. McGraw-Hill Book Co. All rights reserved.)

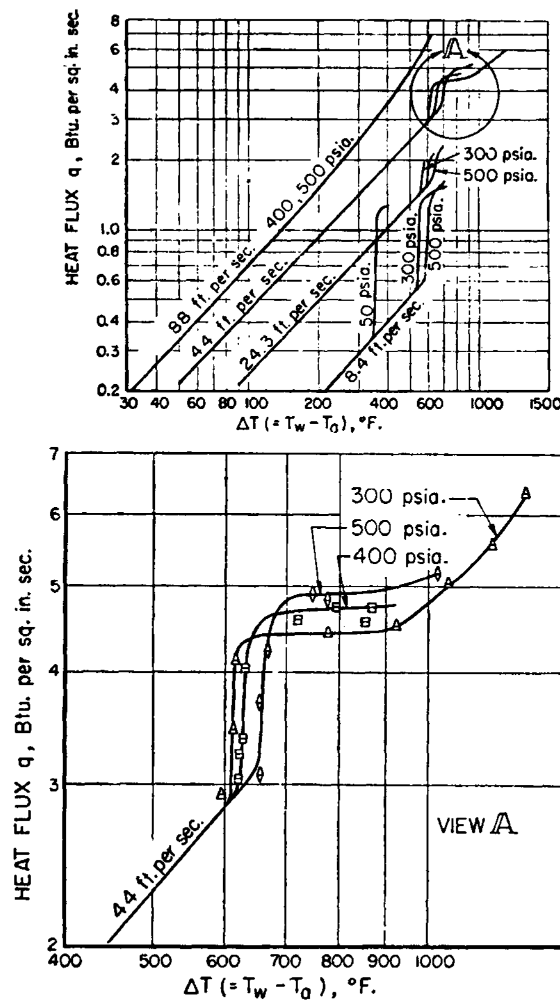


Figure 10-95B. Heat transfer behavior of a mixture of hydrocarbon fuels. (Used by permission: Jens, W. H. *Mechanical Engineering,* V. 76, Dec. 1954, p. 981. ©American Society of Mechanical Engineers. All rights reserved.)

Vaporizers, usually termed reboilers for chemical or petrochemical plant operations can be of several types:

- Horizontal
- Vertical
- Horizontal or vertical as shell and tube units operated by
  - Natural circulation, which includes thermosiphon action
  - Forced circulation or pump-through
- Horizontal kettle-type units

Table 10-25 provides a helpful breakdown of the major types and characteristics of vaporizers and reboilers used in the industry.

For horizontal thermosiphon/natural units the boiling fluid is almost always on the shell side, with the heating medium in the tubes. In the vertical units the reboiling of the fluid is in the tubes. For kettle units, the boiling is in the shell. Collins[185] suggests a "rule of thumb" that if the viscosity of the reboiler is less than 0.5 centipoise (cp), the vertical thermosiphon should be considered, but when the viscosity is more than 0.5 cp, the horizontal reboiler is probably more economical.

Because reboilers are used extensively with bottoms boiling of distillation columns, the horizontal units have some advantages.[185]

- High surface area.
- Process is on the shell side, with possibly less fouling, or easy access for cleaning the outside tubes.
- Tubes have easy access for cleaning on tube side, when fouled.
- Greater flexibility for operator handling high liquid rates.
- Lower boiling point elevation than vertical units.

Figures 10-96A–E illustrate horizontal and vertical thermosiphon reboiler flow arrangements.

For a distillation column bottoms heating, such as shown in Figures 10-96A and 10-96B, the bottoms liquid from the column flows under system pressure and liquid head into and through the shell side of the horizontal thermosiphon reboiler. The two-phase (liquid + vapor) mixture flows from the reboiler back into the distillation column either on the bottom tray or just under this tray into the column vapor space above the bottoms liquid, with the vapor passing upward into/through the bottom or first tray. The density difference between the liquid in the column and the two-phase mixture in the heat exchanger (reboiler) and the riser (outlet piping from the shell side of reboiler) cause the thermosiphon circulation through the reboiler.[186] According to Yilmaz,[186] the horizontal thermosiphon reboilers compared

Table 10-25
Types and Characteristics of Process Reboilers

Type	Advantages	Disadvantages
Vertical thermosiphon	Capable of very high heat transfer rates. Compact; simple piping required. Low residence time in heated zone. Not easily fouled. Good controllability.	Maintenance and cleaning can be awkward. Additional column skirt required. Equivalent to theoretical plate only at high recycle.
Horizontal thermosiphon	Capable of moderately high heat transfer rates. Low residence time in heated zone. Not easily fouled. Good controllability. Easy maintenance and cleaning.	Extra piping and space required. Equivalent to theoretical plate only at high recycle.
Once-through natural circulation	Capable of moderately high heat transfer rates. Compact; simple piping required. Low residence time in heated zone. Not easily fouled. Equivalent to theoretical plate.	Maintenance and cleaning can be awkward. Additional column skirt height required. No control over circulation rate. Danger of backup in column. Danger of excessive per-pass vaporization.
Flooded-bundle (kettle)	Easy maintenance and cleaning. Convenient when heating medium is dirty. Equivalent to theoretical plate. Contains vapor disengaging space.	Lower heat transfer rates. Extra piping and space required. High residence time in heated zone. Easily fouled.
Forced circulation*	Viscous and solid-containing liquids can be circulated. Enables an erosion-fouling balance. Circulation rate can be controlled.	Relatively expensive due to extra shell volume. Cost of pump and pumping. Leakage of material at stuffing box.

*Advantages and disadvantages will in general correspond to the type of reboiler to which forced circulation is applied. The advantages and disadvantages shown are in addition.

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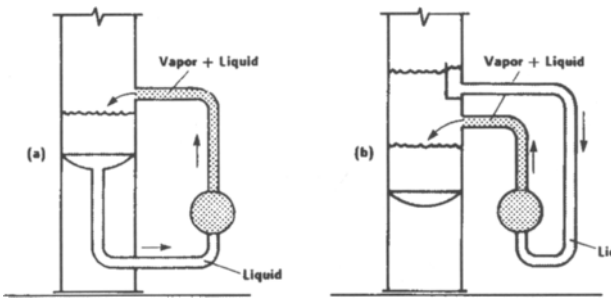

Figure 10-96A. Horizontal thermosiphon reboiler. a. Recirculating feed system. b. Once-through feed system. Both are natural circulation. (Used by permission: Yilmaz, S. B. *Chemical Engineering Progress,* V. 83, No. 11, ©1987. American Institute of Chemical Engineers. All rights reserved.)

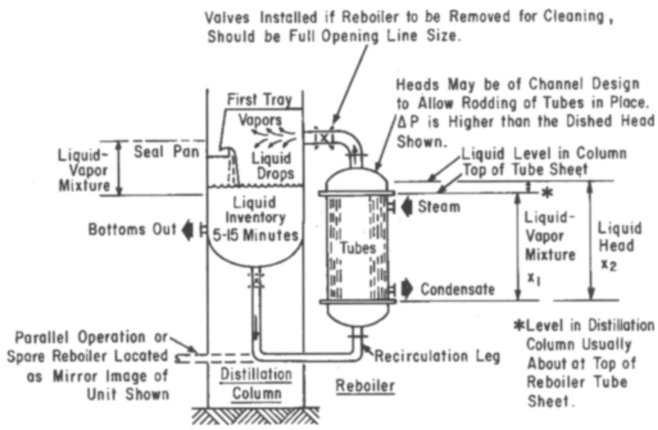

Figure 10-96D. Vertical recirculation thermosiphon reboiler.

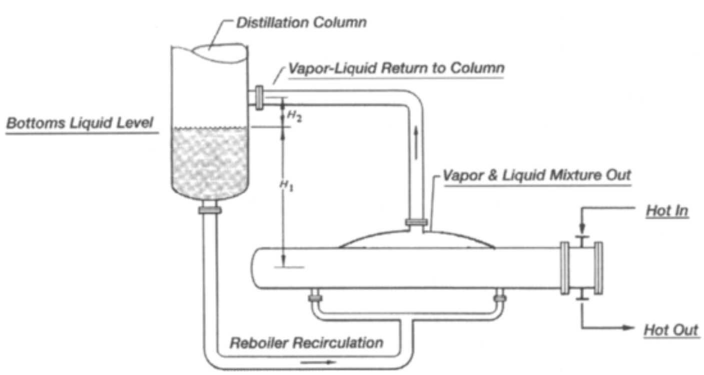

Figure 10-96B. Horizontal thermosiphon reboiler on distillation column: shell and tube design, not kettle. Boiling in shell.

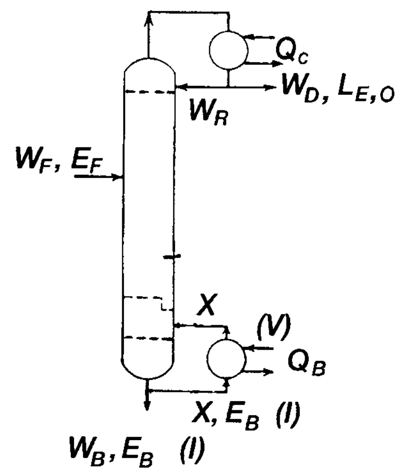

Figure 10-96E. Reboiler heat balance.

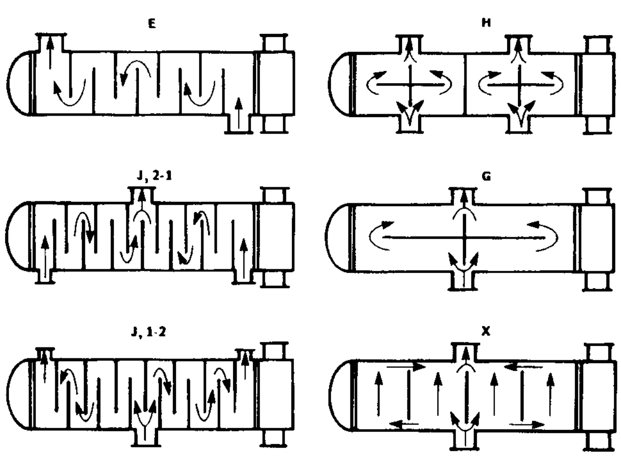

Figure 10-96C. Shell types selected for horizontal and thermosiphon reboilers, boiling in shell. (Used by permission: Yilmaz, S. B. *Chemical Engineering Progress,* V. 83, No. 11, ©1987. American Institute of Chemical Engineers. All rights reserved.)

to kettle reboilers are less likely to be fouled by the process due to their better circulation and lower percent vaporization. Vertical thermosiphon reboilers with process through the tubes, Figure 10-96D, are less suitable than horizontal units when heat transfer requirements are large due to mechanical considerations; that is, the vertical units may often determine the height of the first distillation tray above grade. Also per Yilmaz,[186] moderate viscosity fluids boil better in horizontal units than in vertical units. Low-finned tubing used in horizontal units can improve the boiling characteristics on the shell side. Due to the high liquid circulation rate for horizontal thermosiphon units, the temperature rise for the boiling process fluid is lower than for kettle reboilers (these are not thermosiphon). Ultimately this leads to higher heat transfer rates for the horizontal thermosiphon units.[186]

Hahne and Grigull[186] present a detailed study of heat transfer in boiling.

A number of obvious advantages and disadvantages exist for either the horizontal or vertical thermosiphon reboiler. For horizontal units, Yilmaz states that the TEMA types X, G, and H shown in Figure 10-96C are in more common usage, and types E and J are often used. The selection depends on the heat transfer, fouling, and pressure drop on the shell side. The X Shell is considered to have the lowest comparative pressure drop, then H, G, and J, with E having the best ΔP. The circulation through the thermosiphon loop described earlier depends on the pressure balance of the system, including the static pressure of the liquid level and the inlet pressure drop and exit two-phase pressure drops to and from the reboiler, plus the pressure drop through the unit itself.

Yilmaz[186] recommends that the maximum velocity in the exit from the horizontal thermosiphon reboiler be the work of Collins:[185]

$$V_{max} = 77.15\,(\rho_{tph})^{-0.5} \qquad (10\text{-}138)$$

where

V_{max} = maximum velocity in exit from reboiler, m/sec
ρ_{tph} = homogeneous two-phase density, kg/m³

The arrangement of baffle plates and nozzles, Figure 10-96C, are important to prevent (a) tube vibration, (b) maldistribution of the process boiling fluid, and (c) poor heat transfer coefficients due to uneven and stratified flow resulting in uneven and "dry spot" heat transfer from nonuniform tube wetting, and others.[186]

The work of Heat Transfer Research, Inc., has contributed much to the detailed technology; however, this information is proprietary and released only to subscribing member organizations.

Yilmaz[186] comments that several "unexpected" results have developed from the current horizontal reboiler research studies.

- These units provide higher heat fluxes at the same mean temperature difference.
- These units are superior in thermal performance to vertical tube thermosiphon units.
- These units are superior in thermal performance to kettle reboilers.

Figure 10-97A compares horizontal and vertical units in the same hydrocarbon boiling service at low pressures and shows that the horizontal units are more favorable in the same service than vertical units and even more so when the mean temperature difference is low. Figure 10-97B compares horizontal and vertical thermosiphon units with kettle reboilers when boiling the same hydrocarbon mixture; also see Fair,[46] Jacobs,[273] and Rubin.[278]

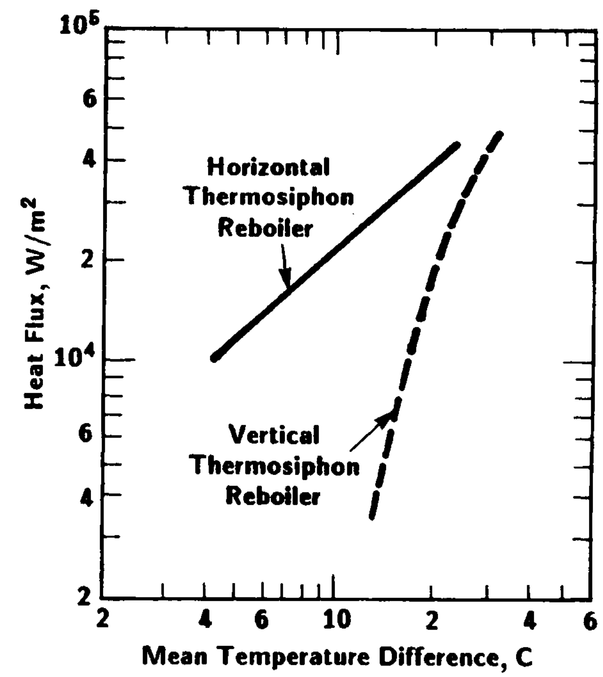

Figure 10-97A. Heat transfer data of reboilers boiling a pure hydrocarbon at low pressure in horizontal and vertical reboilers. (Used by permission: Yilmaz, S. B. *Chemical Engineering Progress,* V. 83, No. 11, p. 64, ©1987. American Institute of Chemical Engineers. All rights reserved.)

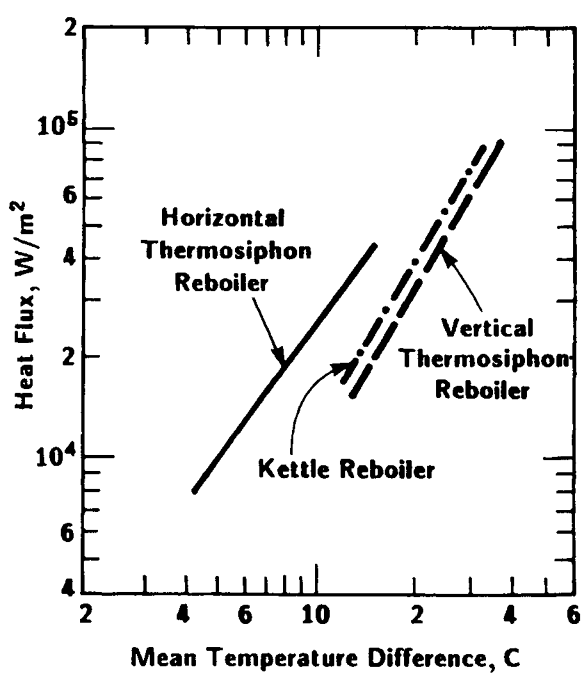

Figure 10-97B. Heat transfer data of reboilers boiling a hydrocarbon mixture in horizontal and vertical thermosiphon reboilers compared to a kettle reboiler. (Used by permission: Yilmaz, S. B. *Chemical Engineering Progress,* V. 83, No. 11, p. 64, ©1987. American Institute of Chemical Engineers. All rights reserved.)

Vaporization in Horizontal Shell; Natural Circulation

Kern[70] deserves a lot of credit for developing design methods for many heat transfer situations and in particular the natural circulation phenomena as used for thermosiphon reboilers and shown in part in Figures 10-96A–D.

The *horizontal natural circulation* systems do not use a kettle design exchanger, but rather a 1-2 (1 shell side, 2 tube-side passes) unit, with the vaporized liquid plus liquid not vaporized circulating back to a distillation column bottoms vapor space or, for example, to a separate drum where the vapor separates and flows back to the process system and where liquid recirculates back along with make-up "feed" to the inlet of the horizontal shell and tube reboiler. See Figures 10-96A–C.

A large portion of vaporization operations in industry are handled in the horizontal kettle unit. The kettle design is used to allow good vapor disengaging space above the boiling surface on the shell side and to keep tubesheet and head end connections as small as possible. Services include vaporizing (reboiling) distillation column bottoms for reintroducing the vapor below the first tray, vaporizing refrigerant in a closed system (chilling or condensing on the process steam side), and boiling a process stream at constant pressure. The tube side may be cooling or heating a fluid or condensation of a vapor.

Physically the main shell diameter should be about 40% greater than that required for the tube bundle only. This allows the disengaging action.

The kettle unit used in the reboiling service usually has an internal weir to maintain a fixed liquid level and tube coverage. The bottoms draw-off is from the weir section. The reboiling handled in horizontal thermosiphon units omits the disengaging space because the liquid-vapor mixture should enter the distillation tower where disengaging takes place. The chiller often keeps the kettle design but does not use the weir because no liquid bottoms draw off when a refrigerant is vaporized.

Pool and Nucleate Boiling—General Correlation for Heat Flux and Critical Temperature Difference

Levy[77] presented a correlation showing good agreement for pool boiling and nucleate boiling heat transfer flux (Q_b/A) below the critical Δt for subcooled and vapor-containing liquids. This covers the pressure range of sub- to above-atmospheric and is obtained from data from the inside and outside tube boiling.

$$\frac{Q_b}{A} = \frac{k_L c_L \rho_L^2 (\Delta T)^3 [1 - x]}{\sigma' T_s (\rho_L - \rho_v) B_L} \tag{10-139}$$

x = vapor quality of fluid = 0 for pool boiling and is a low fraction, about 0.1 to 0.3, for most nucleate boiling

This is represented in Figures 10-98 and 10-99.

where

k_L = thermal conductivity of saturated liquid, Btu/hr (°F/ft)
c_L = specific heat of liquid, Btu/lb (°F)
ρ_L = liquid, lb/ft^3
ρ_v = vapor, lb/ft^3
σ' = surface tension of liquid, Btu/ft^2; (dynes/cm) × (0.88 × 10^{-7}) = Btu/ft^2
ΔT = temperature difference = $T_w - T_s$, °R

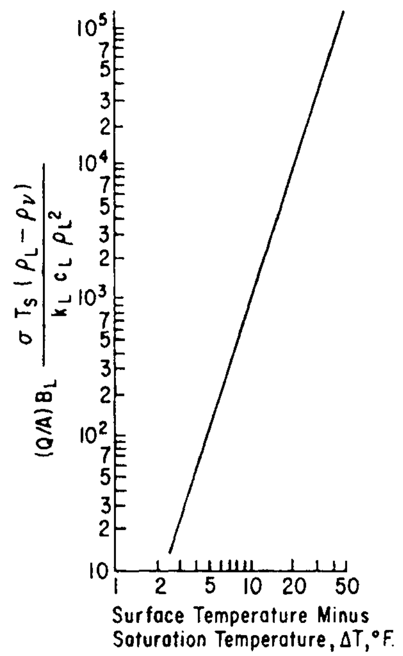

Figure 10-98. Levy correlation for boiling heat transfer equation. (Used by permission: Levy, S. ASME paper no. 58-HT-8, ©1958. American Society of Mechanical Engineers. All rights reserved.)

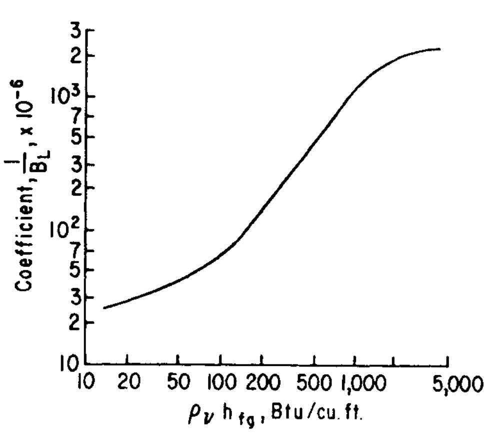

Figure 10-99. Coefficient B_L in Levy boiling heat transfer equation. (Used by permission: Levy, S. ASME paper no. 58-HT-8, ©1958. American Society of Mechanical Engineers. All rights reserved.)

T_w = temperature of tube heating surface, °R
T_s = saturation temperature of liquid, °R
B_L = coefficient of Figure 10-99
h_{fg} = latent heat of evaporation, Btu/lb

The value of this relation is that it serves as a maximum limit that may be expected from a designed unit when comparing design Q/A versus Equation 10-139.

Mikic and Rohsenow[86] present another boiling correlation for pool boiling, which includes the effects of the heating surface characteristics.

It is important to design at values of temperature difference below the critical value separating nucleate and film boiling. The work of Cichelli and Bonilla[27] has produced considerable valuable data, including Figure 10-100, which represents the maximum temperature difference between the fluid saturation temperature and the metal surface for nucleate boiling.[27] This is valuable in the absence of specific data for a system. Equipment should not be designed at greater than these ΔT values unless it is recognized that film boiling will be present and the performance will not be as efficient as if nucleate boiling were the mechanism. Figure 10-101 illustrates the effect of pressure on the nucleate boiling of ethyl alcohol.[27]

The maximum heat flux, Q/A, as a function of reduced pressure for a system before the transition begins to film type boiling is shown in Figure 10-102A and 10-102B. In general, a design should not exceed 90% of these peak values. Other correlations are available, some with special limitations and others reasonably general.[97]

Table 10-26 lists some values of maximum flux and critical ΔT for pool boiling. The values are very useful when quick data must be estimated or when guides and limits must be established. They are also applicable to natural circulation boiling in tubes.

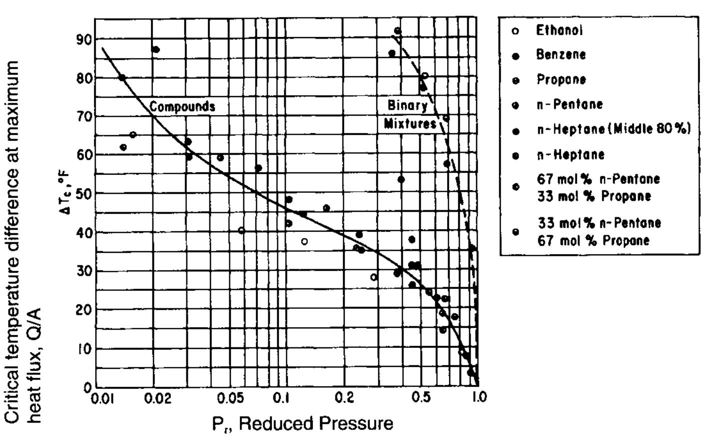

Figure 10-100. Maximum ΔT for nucleate boiling correlation. (Used by permission: Cichelli, M. T., and Bonilla, C. F. *Transactions,* AIChE, V. 41, No. 6, ©1945. American Institute of Chemical Engineers. All rights reserved.)

A well-recognized and often-used equation for determining a reasonable, even if preliminary, nucleate boiling coefficient is represented by the McNelly equation for boiling outside of tubes:

$$h_b = \phi(c_L G_v)\left[\frac{D_o G_v}{\mu_f}\right]^{-0.3}\left[\frac{c_f \mu_f}{k_L}\right]^{-0.6}\left[\frac{P_a^2}{\rho_L \sigma}\right]^{0.425} \qquad (10\text{-}140)$$

where

$$G_v = \frac{Q}{A_{nf}\lambda}\left[\frac{\rho_L}{\rho_v}\right] = \frac{V}{A_{nf}}\left[\frac{\rho_L}{\rho_v}\right] \qquad (10\text{-}141)$$

h_b = boiling side film coefficient, Btu/hr-ft²-°F
ϕ = surface condition factor:
For steel or copper = 0.001
St. steel, nickel = 0.0006
Polished surfaces = 0.0004
Teflon, plastics = 0.0004
c_L = liquid specific heat, Btu/lb-°F
ρ_L = liquid density, lb/ft³
ρ_v = vapor density, lb/ft³
μ_f = liquid viscosity, lb/ft-hr
D_o = tube O.D., ft
V = vapor rate, lb/hr
A_{nf} = surface area of tubes, outside, ft²
Q = heat duty, Btu/hr, or Q_b for boiling
k_L = liquid thermal conductivity, Btu/hr-ft-°F
P_a = absolute pressure of boiling fluid, lb/ft²
σ = surface tension of liquid, lb/ft
λ = latent heat of vaporization, Btu/lb

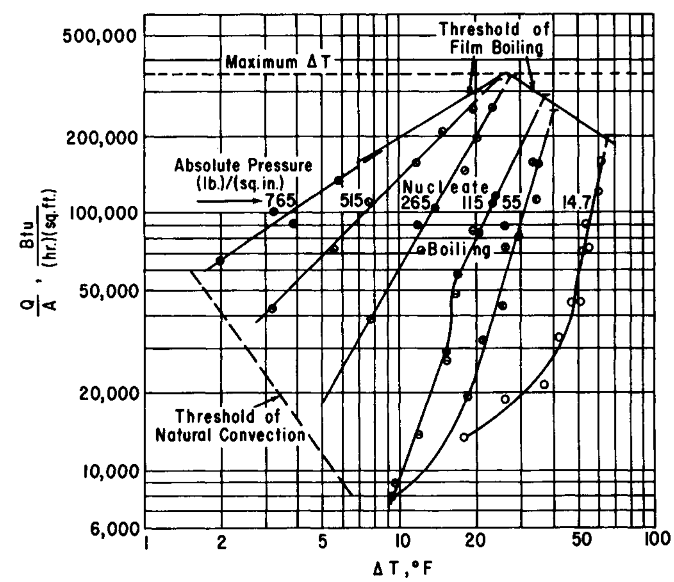

Figure 10-101. Maximum ΔT values occur at the indicated threshold of film boiling, a typical example using 100% ethyl alcohol from a clean surface. (Used by permission: Cichelli, M. T. and Bonilla, C. F. *Transactions.* AIChE, V. 41, No. 6, ©1945. American Institute of Chemical Engineers. All rights reserved.)

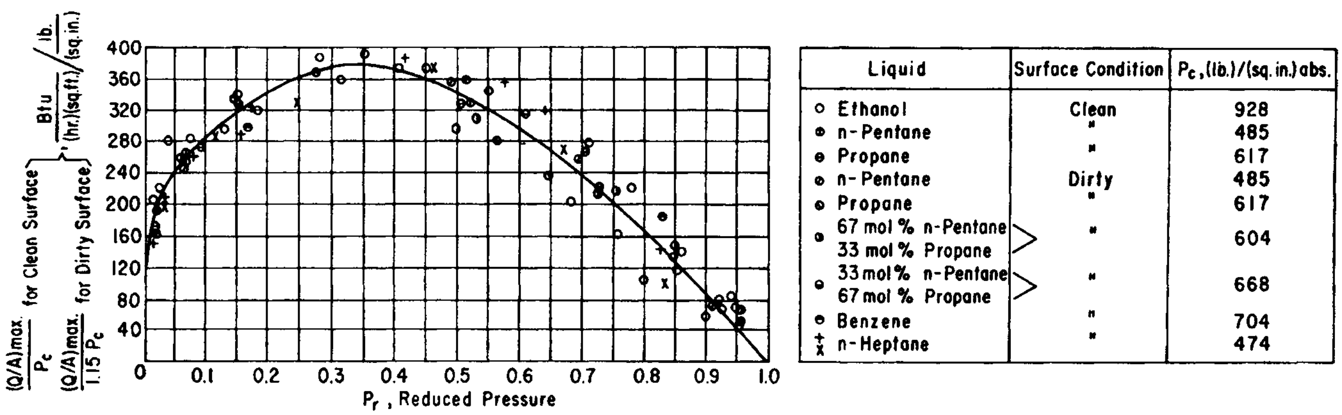

Figure 10-102A. Maximum heat flux (or burnout). (Used by permission: Cichelli, M. T. and Bonilla, C. E. *Transactions*. AIChE., V. 41, No. 6, ©1945. American Institute of Chemical Engineers. All rights reserved.)

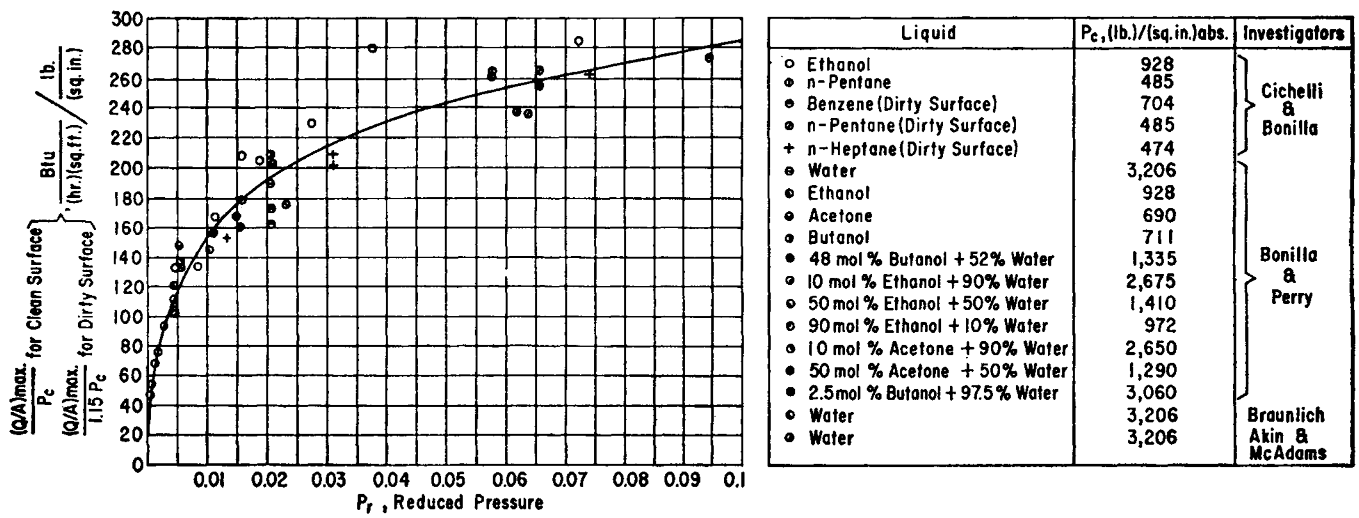

Figure 10-102B. Maximum boiling rate in the low pressure region. (Used by permission: Cichelli, M. T. and Bonilla, C. E. *Transactions*. AIChE, V. 41, No. 6, ©1945. American Institute of Chemical Engineers. All rights reserved.)

Film coefficients calculated by this equation, while useful, are quite strongly influenced by the effects of pressure of the system. In order to somewhat compensate for this, variable exponents for the pressure term of the equation are used as follows:

Pressure, psia	Exponent
Equation	2
Less than 10	2
10 – 30	1.9
30 – 300	1.8
>300	1.7

Experience suggests that the McNelly equation be used for the higher pressures and that the Gilmour[53] equation be used for low pressures of atmospheric and sub-atmospheric.

In kettle-type horizontal reboilers, often the bundle heat transfer film coefficients obtained may be higher than those calculated by most of the single-tube equations. This suggests the possibility of coefficient improvement by the rising agitation from the boiling liquid below. It is impossible to take this improvement into the design without more confirming data.

If the tube bundle is to be large in diameter, it is possible that the liquid head will suppress the boiling in the lower portion of the horizontal bundle; thereby actually creating a liquid heating in this region, with boiling above this. Under such situations, the boiling in the unit cannot be considered for the full volume; hence, there should be two shell-side coefficients calculated and the resultant areas added for the total.

Table 10-26
Maximum Flux at Critical Temperature Difference for Various Liquids Boiling in Pools Heated by Steam Condensing inside Submerged Tubes

	Tubes			
Liquid	Surface on Boiling Side	Liq. Temp. (°F)	Max. Flux Q/A	*Critical Δt, Δt_o (°F)
Ethyl acetate	Aluminum[d]	162[a]	42,000	80
Ethyl acetate	Slightly dirty copper[d]	162[a]	62,000	57
Ethyl acetate	Chrome-plated copper[d]	162[a]	77,000	70
Benzene	Slightly dirty copper[d]	177[a]	43,000	100
Benzene	Aluminum[d]	177[a]	50,000	80
Benzene	Copper[d]	177[a]	55,000	80
Benzene	Chrome-plated copper[d]	177[a]	69,000	100
Benzene	Copper[d]	177[a]	72,000	60
Benzene	Chrome-plated copper[d]	177[a]	70,000	100
Benzene	Steel[d]	177[a]	82,000	100
Carbon tetrachloride	Dirty copper[d]	170[a]	47,000	83
Carbon tetrachloride	Copper[d]	170[a]	58,000	79
Heptane	Copper[d]	209[a]	53,000	55
Ethanol	Aluminum[d]	173[a]	54,000	90
Ethanol	Copper	173[a]	80,000	66
Ethanol	Slightly dirty copper[d]	173[a]	93,000	65
Ethanol	Grooved copper[d]	173[a]	120,000	55
Ethanol	Chrome-plated copper[d]	173[a]	126,000	65
Propanol	Polished nickel-plated copper[d]	127	67,000	91
i-Propanol	Polished nickel-plated copper[d]	151	90,000	84
i-Propanol	Polished nickel-plated copper[d]	175[a]	110,000	96
Methanol	Slightly dirty copper[d]	149[a]	78,000	92
Methanol	Chrome-plated copper[d]	149[a]	120,000	110
Methanol	Steel[d]	149[a]	123,000	105
Methanol	Copper[d]	149[a]	124,000	115
n-Butanol	New nickel-plated copper[d]	173	79,000	83
n-Butanol	New nickel-plated copper[d]	207	92,000	79
n-Butanol	New nickel-plated copper[d]	241[a]	105,000	70
i-Butanol	Polished nickel-plated copper[d]	222[a]	115,000	85
Water	Polished nickel-plated copper[b]	131	115,000	53
Water	Chrome-plated copper[c]	110	150,000	...
Water	Chrome-plated copper[c]	130	175,000	65
Water	New nickel-plated copper[b]	155	190,000	...
Water	Chrome-plated copper[c]	150	220,000	64
Water	Chrome-plated copper[c]	170	243,000	64
Water	Polished nickel-plated copper[b]	171	250,000	72
Water	New nickel-plated copper[b]	191	260,000	...
Water	Chrome-plated copper[c]	190	300,000	70
Water	Chrome-plated copper[b]	212[a]	330,000	80
Water	New nickel-plated copper[b]	212[a]	360,000	68
Water	Polished nickel-plated copper[b]	212[a]	370,000	72
Water	Chrome-plated copper[c]	212[a]	390,000	72
Water	Steel[d]	212[a]	410,000	150

*The overall temperature difference Δt_o is defined as the saturation temperature of the steam less the boiling temperature of the liquid.

[a] Boiling at atmospheric pressure.

[b] Steam side was promoted with Benzyl Mercaptan.

[c] Steam side was promoted with Octyc Thiocyante.

[d] Steam probably contained a trace of Oleic acid.

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Reboiler Heat Balance

Because the reboiler is usually used in conjunction with distillation columns, the terminology and symbols used here will relate to that application. Assume a column with an overhead total condenser and a bottoms reboiler (see Figures 10-96D and 10-96E). Assuming all liquid feed, the heat balance is[70]

$$Q_R = (R + 1)\, W_D E_{D(V)} - R W_D E_{D(l)} + W_B E_{B(l)} - W_f E_{f(l)} \quad (10\text{-}142)$$

where

- R = reflux ratio, mol condensate returned to column/mol product withdrawn
- V_{RB} = vapor formed in reboiler, lb/hr
- W = flow rate, lb/hr
- E_D = enthalpy of overhead product removed from column., Btu/lb
- Q_c = heat load of overhead condenser (removed in condenser), Btu/hr
- Q_R = reboiler duty or heat added, Btu/hr
- L = latent heat of vaporization, Btu/hr
- E_B = enthalpy of bottoms, Btu/lb
- E_F = enthalpy of feed, Btu/lb

Subscripts:

- B = bottoms product
- c = condensing
- v = vapor
- l = liquid
- R = reboiler
- D = overhead distillate product
- F = feed

For a heat balance around the entire column[70] with the feed being liquid or vapor; that is, Heat In = Heat Out:

$$W_F E_{F(l\ \mathrm{or}\ V)} + Q_R = Q_C + W_B E_{B(l)} + W_D E_{D(l)} \quad (10\text{-}143)$$

Example 10-17. Reboiler Heat Duty after Kern[70]

See Figure 10-103.

Assume 25,000 lb/hr of a 50-50 mixture of light hydrocarbons to be separated to a 99.5% (wt) light HC overhead and bottoms of 5% (wt) heavier HC. The reflux ratio determined separately for the column is 3.0 mol reflux/mol of overhead distillate.

Overall material balance:

$$25{,}000 = W_D + W_B$$

HC #1 (more volatile): $25{,}000(0.50) = 0.995 W_D + .05\, W_B$

Then simultaneously:

$$(25{,}000)(0.05) = 0.995(25{,}000 - W_B) + 0.05(W_B)$$
$$12{,}500 = 24{,}875 - 0.995 W_B + 0.05(W_B)$$

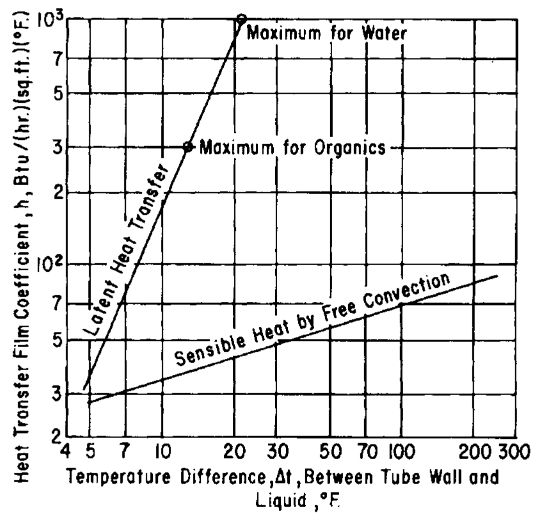

Figure 10-103. Kern correlation for natural circulation boiling and sensible film coefficients—outside and inside tubes. (Used by permission: Kern, D.Q. *Process Heat Transfer,* 1st Ed., ©1950. McGraw-Hill Book Company. All rights reserved.)

$$-1{,}237.5 = 0.945\, W_B$$

W_B = 13,095 lb/hr total bottoms

W_D = distillate total = 25,000 − 13,095 = 11,905 lb/hr

Assume using enthalpy and latent heat tables/charts.

$E_{B(l)}$ = 160 Btu/lb

$E_{F(l)}$ = 100 Btu/lb

$E_{D(v)}$ = 302 Btu/lb

L_v = 135 Btu/lb

Then, substituting in equation 10-142,

$$\begin{aligned} Q_R &= (3.0 + 1.0)(11{,}905)(302) - (3.0)(11{,}905)(95) + (13{,}095)(160) - 25{,}000(100) \\ &= 14{,}381{,}240 - 3{,}392{,}925 + 2{,}095{,}200 - 2{,}500{,}000 \\ &= 10{,}583{,}515 \text{ Btu/hr} \end{aligned}$$

Total vapor to be generated by reboiler:

= 10,583,315/ 135 = 78,394 lb/hr

Kettle Horizontal Reboilers

Kettle horizontal reboilers consist of either a U-bundle or a shell and tube bundle inserted into an enlarged shell. The enlarged shell provides disengaging space for the vapor outside and above the liquid, which is usually held by level control at the top level of the tube bundle or possibly a few in. below the top of the tubes. The heating medium is inside the tubes.

Internal reboilers are similar in concept and designed accordingly to the kettle units, but this style does not have a separate shell, as it is inserted into the circular shell of the sidewall of the distillation column or tank, for example, see Figure 10-111. It is inserted into the body of the liquid to be heated/boiled. The level of the liquid is also controlled as described earlier.

The mechanism of boiling is essentially nucleate pool boiling. In both styles of reboiler the liquid velocity is relatively low compared to thermosiphon units.[90, 188] Jacobs[188] provides an extensive comparison of advantages and disadvantages of essentially all the reboiler types used in industrial plants. Palen and Taborek[91] conducted extensive studies of available data and proposed nucleate boiling equations to correlate various data from the available 14 equations down to a selected 6 for detailed study. The study was limited to various hydrocarbons and hydrocarbon mixtures. Their conclusions after computer correlations of the results from several equations were as follows.

Palen[90] recommendation corrects single tube boiling data (outside) to the bundle effect in a horizontal reboiler by:

$$\text{Revised boiling coefficient, } h_b = h_{1t}\,(BCF)$$

This is limited by the maximum heat flux of approximately 12,000–25,000 Btu/(hr)(ft^2).

The bundle correction factor for vapor blanketing:

$$(BCF) = h_{ow}\left[0.714\left(p - D_o^{4.2(10-5)\bar{G}}\,[1/N_{vc}]_{vc}^{-0.24[1.75 + \ln(1/N)}\right)\right] \quad (10\text{-}144)$$

where

$$\bar{G} = \frac{a_o(U_1)(\Delta T)}{(\lambda)p - D_o} = \text{mass velocity of vapor} \quad (10\text{-}145)$$

U_1 is found by Equation 10-161 = overall coefficient for isolated single tube, Btu/(hr)(ft^2)(°F).

$\bar{G}$ = mass velocity of vapor from a bottom tube based on the $(p - D_o)$ spacing, lb/(hr) (ft^2)
a_o = tube outside heat transfer surface, ft^2/ft
p = tube pitch, ft
D_o = tube O.D., ft
N_{vc} = number of tubes in the center vertical row of bundle
λ = latent heat, Btu/lb
ΔT = mean temperature difference between the bulk of the boiling liquid and the bulk of the heating medium, °F
h_b = corrected boiling coefficient, Btu/(hr)(ft^2)(°F), for bundle
h_{1t} = nucleate boiling coefficient for an isolated single tube, Btu/(hr)(ft^2)(°F)

Maximum Bundle Heat Flux[91]

Recommended limiting maximum heat flux[91] for the tube density coefficient:

$$\phi = \frac{D_b(L)}{A} \quad (10\text{-}146)$$

$$= 0.359\left(\frac{p}{D_o}\right)\left[\frac{\sin\alpha}{N}\right]^{1/2} \quad (10\text{-}147)$$

$$\Psi = \rho_v\lambda\left[\frac{g\sigma(\rho_l - \rho_v)}{\rho_v^2}\right]^{0.25} \quad (10\text{-}148)$$

$$q_{max} = K\Phi\Psi \quad (10\text{-}149)$$

Gilmour's bundle correction is
$h_b = h\,(N_{rv})^{-0.185}\,(v_s)^{-0.358}$, improved to
$h_b = h\,(N_{rv})^{-0.185}\,(v_s/v_c)^{-0.385}$
N_{rv} = number of holes in vertical center row of bundle
v_s = superficial vapor velocity
v_c = maximum flux
$v_c = q_{max} / \lambda\rho_v$, where q_{max} comes from the Zuber equation discussed separately

The results for small bundles do not agree as well as the Palen and Taborek[91] equation.

Determine q_{max} from Figure 10-103A. Use a safety factor of 0.7 with Equation 10-149 per the recommendation of reference 91 for conservative results.

where
D_b = bundle diameter, ft
L = average bundle length, ft
A = bundle heat transfer surface, ft^2 (outside)
α = tube layout angle, degrees
N = number of tube holes/tubesheet. Note: U-tubes have 2 holes per tube, so $N = 2 \times$ number of tubes
ρ_v = vapor density, lb/ft^3
ρ_l = liquid density, lb/ft^3
g = acceleration of gravity, ft/(hr)(hr)
λ = latent heat, Btu/lb
σ = surface tension, lb (force)/ft
K = empirically determined constant used as 176 in the range of ϕ for bundles
q_{max} = maximum heat flux, Btu/(hr)(ft^2)
ψ = maximum flux physical property factor, Btu/(ft^3)(hr)
p = tube pitch, ft
D_o = tube O.D., ft

The original Zuber[191] equation for maximum heat flux as modified by Palen[90]

$$q_{max} = 25.8\,(\rho_v)\,(\lambda)\,[\sigma\,(\rho_l - \rho_v)\,g / \rho_v^2]^{0.25}\,[(\rho_v + \rho_l)/\rho_l]^{0.5},$$
$$\text{Btu/(hr)(ft}^3) \quad (10\text{-}150)$$

Symbols are as defined previously.

The tube density coefficient, ϕ, is given in Table 10-27.

The tube wall resistance cannot be ignored for reboilers. Based on the outside tube diameter,[91]

$$r_w = \frac{T_{wall}}{k_w}\left[\frac{a_o\ln(a_o/a_i)}{a_o - a_i}\right] = \frac{a_o\ln(D_o/D_i)}{2\pi k_w} \quad (10\text{-}151)$$

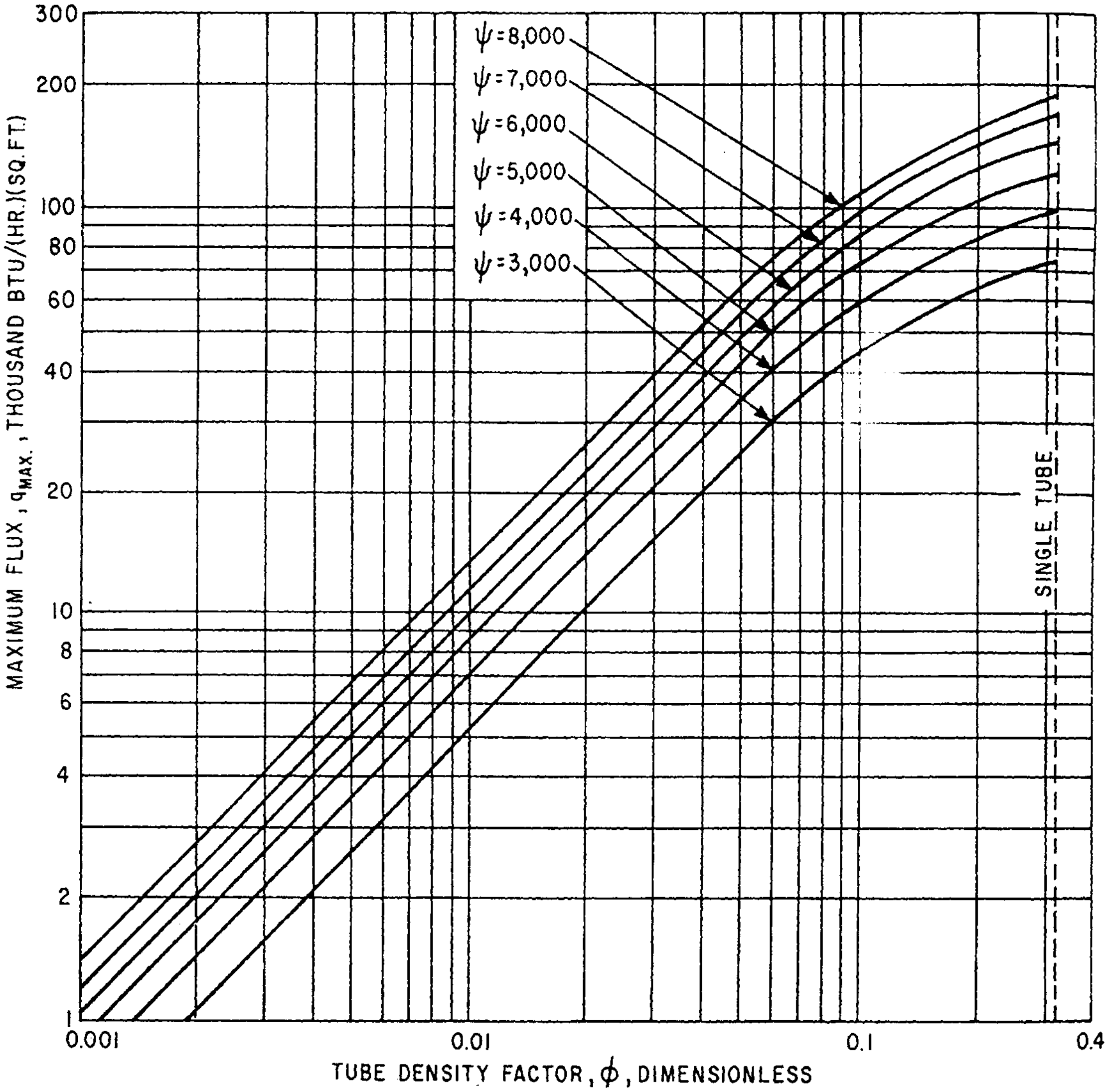

Figure 10-103A. Maximum heat flux: boiling outside horizontal tubes; kettle and internal reboilers. When using the estimate from this curve, a safety factor of 0.7 also should be used. (Used by permission: Palen, J. W., and Small, W. M. *Hydrocarbon Processing,* V. 43, No. 11, ©1964. Gulf Publishing Company, Houston, Texas. All rights reserved.)

Table 10-27
Tube Density Coefficient for 60° Triangular Pitch

ϕ = [Tube Density Coefficient] $[L/A]^{0.5}$		
	Coefficient for a given D_o	
p, in.	$^3/_4$ in.	1 in.
1	0.196	—
1.5	0.294	0.257
1.75	—	0.299

L=Bundle length, ft; bundle heat transfer surface, ft^2

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where

r_w = tube wall resistance, (hr)(ft)(°F)/Btu
a_i = tube inside heat transfer surface per ft, ft
a_o = tube outside heat transfer surface per ft, ft
T_w = tube wall thickness, ft
k = wall thermal conductivity, Btu/(hr)(ft)(°F)
D_o = tube O.D., ft
D_i = tube I.D., ft
π = pi = 3.1416

A. McNelly Equation,[90, 91, 189] overall deviation + 50% and −40%:

$$h_l = 0.225\, C_s\, (g\, c_l / \lambda)^{0.69}\, (144 P k_l/\sigma)^{0.31} [\rho_l/\rho_v) - 1]^{0.33} \quad (10\text{-}152)$$

where

C_s = surface factor, for clean copper and steel tubes = 1.0 and for clean chromium = 0.7
g = acceleration of gravity, ft/hr^2 = 4.17×10^8
λ = latent heat, Btu/lb
h_l = nucleate boiling coefficient for single isolated tube, Btu/(hr)(ft^2)(°F)
c_l = liquid specific heat, Btu/(lb)(°F)
σ = surface tension, lb/ft
P = reboiler operating pressure, lb/in.^{2}abs.

To account for tube and bundle geometry, Gilmour's[90, 91] equation is modified based on the single tube calculation,[91, 190]

$$h_b = h_{st}(N_{rv})^{-0.185}(v_s)^{-0.358} \tag{10-153}$$

v_s = superficial vapor velocity, ft/sec
N_{rv} = number of tubes in center vertical row of bundle
h_{st} = h = theoretical boiling film coefficient for a single tube, Btu/(hr)(ft^2)(°F)
h_b = heat transfer coefficient for a reboiler bundle, Btu/(hr)(ft^2)(°F)

B. Gilmour Equation[190] reportedly was used successfully in many reboilers and vaporizers:

$$h = 0.001\,(c_l\,G)\,(P^2/\rho_L\sigma)^{0.425}\,(k_l/c_l\mu_l)^{0.6}\,(\mu_l/D_oG)^{0.3} \tag{10-154}$$

where

G = (V/A) (ρ_l/ρ_v), mass velocity normal to tube surface, lb/(hr)(ft^2)
For all other factors constant: $h = (V/A)^{0.706}\lambda^{0.706}a^{0.416}$
h = film coefficient of boiling heat transfer, Btu/(hr)(ft^2)(°F)
V = vapor produced, lb/hr
A = surface area, ft^2
a = proportionality constant = 1.0
k = k_l = thermal conductivity of liquid, Btu/(hr)(ft)(°F)
c_l = specific heat of liquid, Btu/lb-°F
P = pressure of boiling liquid, lb/ft^2
D = D_o = O.D. of tube, ft
σ = surface tension of liquid, lb/ft
μ_l = viscosity of liquid, lb/(hr)(ft)
ρ_L = density of liquid, lb/ft^3
ρ_v = density of vapor, lb/ft^3

In tube bundles, if the disengaging space between the bundle and the kettle is small and insufficient to allow the vapor bubbles to "break-free" of the liquid and thus tend to blanket the upper tubes with gas, heat transfer will be restricted.[190] For best design the superficial vapor velocity should be in the range of 0.6–1.0 ft/sec to prevent the bubbles from blanketing the tube through the bundle and thereby preventing liquid contact with the tubes. When the maximum heat flux is approached, this condition can occur, so the 1.0 ft/sec vapor velocity is recommended.

Palen and Taborek[91] modified the Gilmour equation to better accommodate the effect of surface types and the effect of pressure, with the results agreeing with all the data +50% and −30%, which is better than other proposed correlations.

$$h = 9.0 \times 10^{-4} C_s(c_L^{0.4}k_L^{0.6}\rho_L^{0.275}/\mu^{0.3})\,[(q/\lambda)\rho_L]^{0.7}(P^2/\sigma^{0.425})(14{,}400/P)^m \tag{10-155}$$

$$m = 6.0\,e^{-0.0035}\,T_c$$

Maximum errors for data tested is +50% to −30%, restricted to hydrocarbons with $T_c > 600$°R.

where

T_c = critical temperature, °R
q = heat flux, Btu/(hr)(ft^2)
C_s = surface factor, noted with the McNelly equation cited earlier
h = theoretical boiling coefficient for a single tube, Btu/(hr)(ft^2)(°F)
P = pressure, lb/ft^2
λ = latent heat, Btu/lb
σ = surface tension, lb/ft
ρ_L = liquid density lb/ft^3
ρ_v = vapor density lb/ft^3
μ_L = liquid viscosity, lb/hr-ft
c_L = liquid specific heat, Btu/lb-°F
g = acceleration of gravity, ft/hr-hr
h = theoretical boiling coefficient for a single tube, Btu/(hr)(ft^2)(°F)
q = heat flux, Btu/(hr)(ft^2)
q_{max} = maximum heat flux, Btu/(hr)(ft^2)
m = coefficient for equation for h[91]

The maximum heat flux recommended by Zuber[191] and confirmed by Palen and Taborek:[91]

$$q_{max} = 25.8\,(\rho_v)\,(\lambda)\,[\sigma(\rho_L - \rho_v)\,g/\rho_v^2]^{0.25}[(\rho_v + \rho_L)/\rho_L]^{0.5} \tag{10-156}$$

where symbols are as listed earlier. To avoid confusion with subscripts: L = l (liquid) and V = v (vapor).

Examination of plant data by the authors[91] revealed that tubes spaced closely together tend to create a vapor blanketing effect and the consequence of lower heat flux than for wider-spaced tube pitches. This author's experience has been to spread out the tube spacing (pitch) from normal heat exchange design to ensure free boiling bubble movement to avoid the very problem expressed in reference.[91] To account for Gilmour's effect of the bundle on single tube calculations, see Equation 10-153.

Palen and Taborek[91] proposed as the best choice for circular tube bundles (as compared to square) the following film boiling coefficient after analyzing available data (their

Table 10-28
Error Comparison of Test Case from Reference 91*

	Existing Methods	Proposed Methods		
	Kern	Gilmour (Eq. 8)	Statistical Model (Eq. 10)	Tube-by-Tube-Model (Eq. 12)
Error with no safety factor				
1. Ave. % overdesign	61	15	17	15
2. Ave. % underdesign	0	40	19	10
Errors with safety factor included				
3. Area safety factor required	1.0	1.80*	1.25	1.25
4. Ave. % overdesign	61	48*	26	30
5. Max. % overdesign	140	110*	75	60

*Excluding case 15, which showed very high error.

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statistical analysis results). Also see Table 10-28 where the equation references are to their article reference,[91] not this current text.

$$h_b \approx 0.714\, h\, (p - d)^{4.2 \times 10^{-5} G}\, (1/N_{rv})^{-0.24\,(1.75 + \ln(1/N_{rv}))} \quad (10\text{-}157)$$

where
G is the single tube mass velocity through the (p-d) tube space, defined as:

$$G = \frac{a_t(\Delta t)U}{\lambda(p - d)}$$

where most symbols are as defined earlier, plus
- a_t = surface area, ft^2/ft of tube outside surface area
- p = tube pitch, ft
- d = tube O.D., ft
- h_b = heat transfer coefficient for reboiler bundle, Btu/hr-ft^2-°F
- U = overall heat transfer coefficient based on theoretical single tube, h

The fit of the data to the proposed equation is on average ± 30% overdesign, which is good in terms of boiling data and when compared to the Gilmour's bundle coefficient of +48% and Kern's +61%.[91]

- h = theoretical boiling coefficient, Btu/(hr)(ft^2)(°F)
- h_b = heat transfer coefficient for reboiler bundle, Btu/(hr)(ft^2)(°F)

The following is a calculation procedure suggested by Palen and Small[90] for the required boiling coefficient for a horizontal tube bundle:

1. Assume a value for h_l.
2. Calculate U_l from Equation 10-161.

- r_w = wall resistance, $[\text{Btu/(hr)(ft}^2\text{)(°F)}]^{-1}$
- r_i = inside fluid fouling resistance, $[\text{Btu/(hr)(ft}^2\text{)(°F}^{-1}\text{)}]$

3. Calculate h_l from McNelly or Gilmour equations.
4. Compare calculated h_l with assumed value, if difference is significant, use the calculated value and repeat from step 2 until convergence is acceptable
5. Calculate ΔT_b from:

$$\Delta T_b = (U_l/h_l)(\Delta T) \quad (10\text{-}158)$$

If ΔT_b is less than 8°F, free convection must be taken into account by a corrected $h_l = h_l'$:

$$h_l' = h_l + 0.53(k_L/D_o)\left[\frac{D_o^3 \rho_L^2 g \beta_L \Delta T_b c_L}{\mu_L k_L}\right]^{0.25} \quad (10\text{-}159)$$

where
- ΔT_b = mean temperature difference between the bulk of the boiling liquid and the tube wall, °F
- ΔT = mean temperature difference between the bulk of the boiling liquid and the bulk of the heating medium, °F
- h_l = nucleate boiling coefficient for an isolated single tube, Btu/(hr)(ft^2)(°F)
- h_l' = nucleate boiling coefficient for an isolated single tube corrected for free convection, Btu/(hr)(ft^2)(°F)
- β = coefficient of thermal expansion of liquid

Other symbols as cited previously.

6. Calculate the correction to the nucleate boiling film coefficient for the tube bundle number of tubes in vertical row, h_b. See previous discussion.
7. Using Equation 10-161 to determine the overall U for bundle.
8. Determine the area required using U of step 7.
9. Determine the physical properties at temperature: $T_b = \Delta T_b/2$.

Nucleate or Alternate Designs Procedure

The following nucleate or alternate designs procedure, suggested by Kern,[70] is for vaporization (nucleate or pool boiling) only. No sensible heat transfer is added to the boiling fluid.

Note: If sensible heat, Q_s, is required to bring the fluid up to the boiling point, this must be calculated separately, and the area of heat transfer must be added to that determined for the fluid boiling requirement, Q_b.

1. Evaluate the heat load for the unit, Q_b.
2. Determine the LMTD.
3. Assume or estimate a unit size (number and size of tubes, shell, etc.).
4. Determine the tube-side film coefficient for convection or condensation as required, by methods previously described.
5. Determine the shell-side coefficient
   a. Evaluate tube wall temperature
   b. Evaluate boiling coefficient from Equation 10-139 or Figure 10-103.

   Note: the use of Figure 10-103 is considered conservative. Many organic chemical and light hydrocarbon units have been successfully designed using it; however, it is not known whether these units are oversized or by how much.
6. Calculate the required area, based on the film coefficient of steps 4 and 5 together with fouling and tube wall resistances; $A = Q/U\,\Delta t$.
7. If the assumed unit does not have sufficient area, select a large size unit and repeat the preceding procedure until the unit is satisfactory (say 10–20% excess area).
8. Determine the tube side pressure drop.
9. Determine the shell-side pressure drop; however, it is usually insignificant. It can be evaluated as previously described for unbaffled shells.

Kettle Reboiler Horizontal Shells

See Figure 10-1F.

In order to properly handle the boiling-bubbling in a kettle unit, there must be disengaging space, and the velocities must be calculated to be low to reduce liquid droplets being carried out of the unit. Generally, no less than 12 in. of space should be above the liquid boiling surface to the top centerline of the reboiler shell. When vacuum operations are involved, the height should be greater than 12 in. Vapor outlet nozzle velocities must be selected to be low to essentially eliminate entrainment. The liquid boiling surface should not be greater than 2 in. above the top horizontal tube, and in order to reduce entrainment, it is often advisable to leave one or two horizontal rows of tubes exposed, i.e., above the liquid. This will tend to ensure that the liquid mist/droplets are vaporized and thereby reduce entrainment. As a *guide* to the relationship between tube bundle diameter and kettle shell diameter, the following can be helpful. Also, often the tube bundle is not completely circular; that is, the upper portions of circular tubes are omitted to leave a flat or horizontal tube row at the top of the bundle, at which level the liquid is often set.

Heat Flux, Boiling, Btu/(ft^2) (hr)	Ratio of Shell Diameter to Tube Bundle Diameter
20,000	1.9 to 2.5
15,000	1.8 to 2.1
12,000	1.5 to 1.7
8,000	1.3 to 1.6
Less than 8,000	1.2 to 1.5

Figure 10-104 is Palen and Small's[90] guide to selecting the kettle "larger" diameter for the design of horizontal kettle units.

Horizontal Kettle Reboiler Disengaging Space[90]

Palen[90] suggests that the distance from the centerline of the uppermost tube in a horizontal bundle to the top of the shell should not be less than 40% of the kettle shell diameter. To size the kettle shell:[90]

Allowable vapor load,

$$(VL) = 2{,}290\rho_v\left[\frac{\sigma}{6.86(10)^{-5}(\rho_L - \rho_v)}\right]^{0.5},\ \text{lb/hr-ft}^3 \qquad (10\text{-}160)$$

Vapor space, $S = V/(VL)$, ft^3
V = actual reboiler vapors, rate, lb/hr
σ = surface tension, lb (force)/ft
Dome segment area = (SA) = S/L, ft^2
L = average bundle length, ft

Kettel Horizontal Reboilers, Alternate Designs

Referring to the procedure of Palen and Small[90] and Palen and Taborek,[91] this is an alternate check on the previously suggested procedure. This technique should generally be restricted to single fluids or mixtures with narrow boiling ranges (wide boiling range gives too optimistic results). Mean temperature difference between the bulk boiling liquid temperature and tube wall should be greater than 8°F; the ratio of pitch to tube diameter should be 1.25 to 2.0; tube bundle diameters should be greater than 1.0 ft and less than 4 ft; and boiling must be less than 400 psia. Vacuum operations may give optimistic results and are not to be used on polar compounds.

Due to the development of the data, the method requires the use of a single tube boiling film coefficient. Using this to reach the overall bundle transfer: The overall U value is determined for a theoretical boiling coefficient of an unfouled tube (single) (this is an iterative procedure). See reference 90 also.

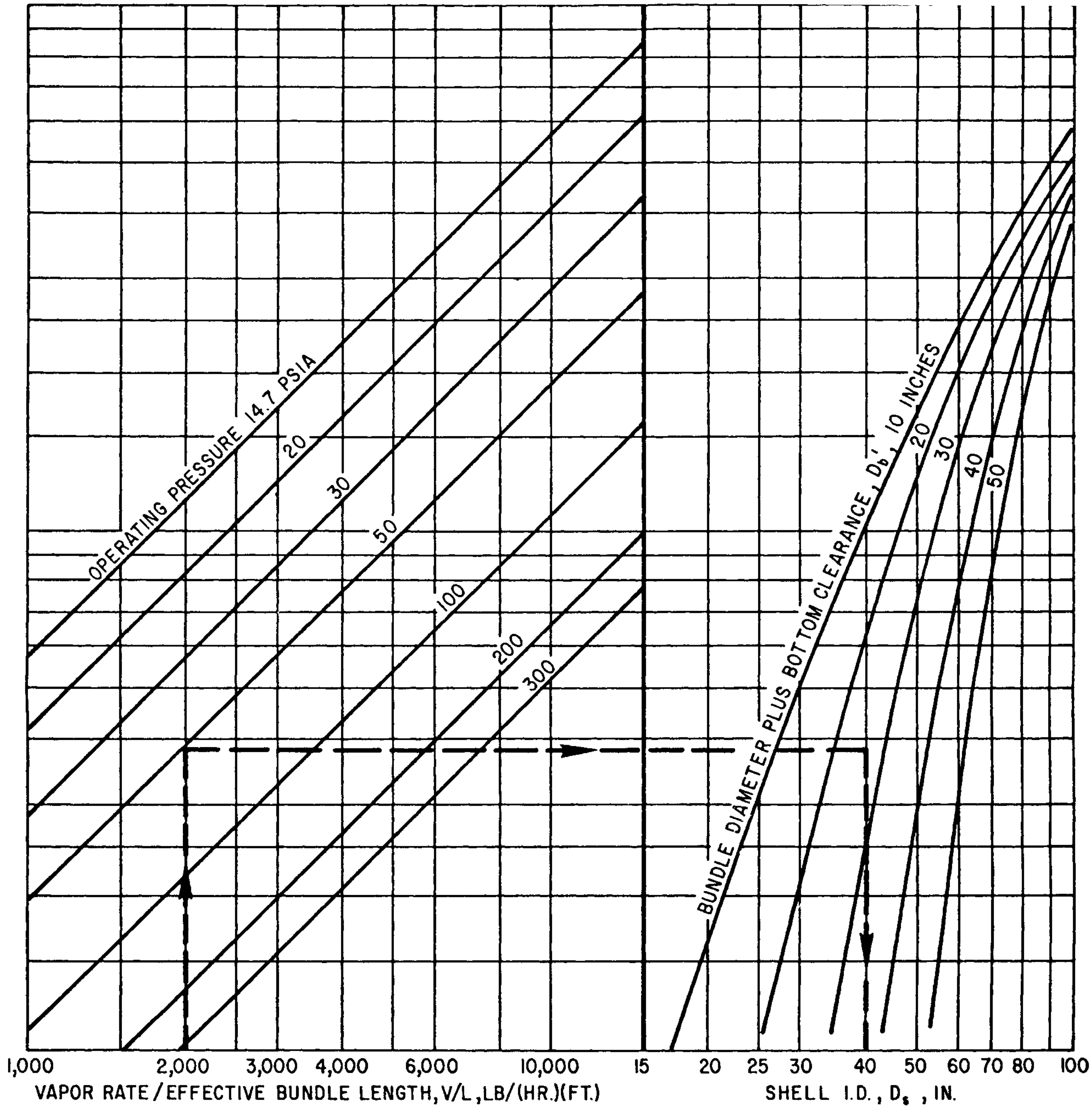


Figure 10-104. Kettle reboiler—estimate of shell diameter. Example: If the vapor rate is 50,000 lb/hr and the bundle is 25 ft, then V/L is 2,000 lb/(hr)(ft). Entering the curve at this V/L ratio, with an operating pressure of 50 psia and a bundle diameter of 24 in. gives an estimate for shell I.D. of 40 in. (Used by permission: Palen, J. W., and Small, W. M. *Hydrocarbon Processing,* V. 43, No. 11, ©1964. Gulf Publishing Company, Houston, Texas. All rights reserved.)

$$U_{(1)} = \frac{1}{\dfrac{1}{h_1} + r_w\left[\dfrac{A_o}{A_{Avg}}\right] + r_i\left[\dfrac{A_o}{A_i}\right] + \dfrac{1}{h_i}\left[\dfrac{A_o}{A_i}\right]} \quad (10\text{-}161)$$

where

$U_{(1)}$ = single tube overall heat transfer coefficient

h_1 = nucleate boiling coefficient for single tube, outside Btu/hr (ft) (°F)

h_i = heating side film coefficient, Btu/(hr) (ft²) (°F)

Note that h_1 must be assumed to solve this equation and later verified. An iteration on h_1 will possibly result in a balance.

For a single tube:

$$h_1 = 0.225C_s\left[\frac{U_1 \Delta T_{LM} C_L}{\lambda}\right]^{0.69}\left[\frac{144Pk_L}{\sigma}\right]^{0.31}\left[\frac{\rho_L}{\rho_v} - 1\right]^{0.33} \quad (10\text{-}162)$$

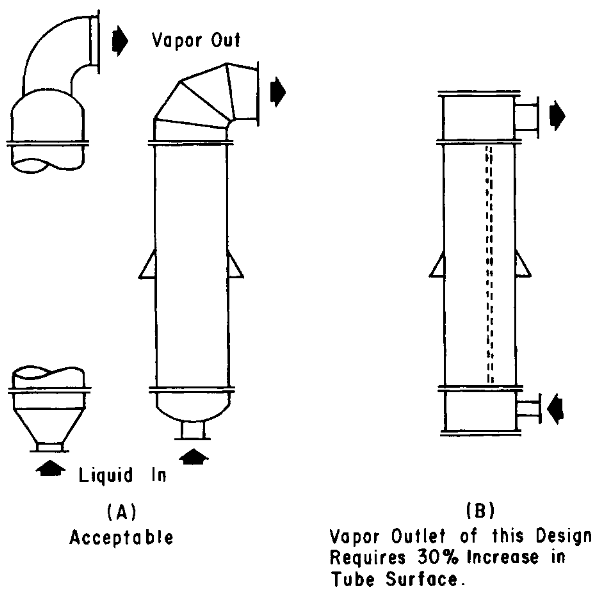


Figure 10-105. Nozzle connections for vertical thermosiphon reboilers.

where

ΔT_{LM} = log mean temperature difference between liquid and hot fluid, °F

C_s = surface condition constant
= 1.0 for commercial tubes
= 0.7 for highly polished tubes

For the entire bundle, the film boiling coefficient (assumes all tubes boiling, no liquid head effect so that some tubes are not boiling):

$$h_b = h_{on}[0.714(p_f - D_o)]^{4.2\times 10^{-5}E}\left[\frac{1}{N_{vc}}\right]^{-0.24[1.75+\ln(\frac{1}{N_{vc}})]} \quad (10\text{-}163)$$

where

p_f = tube pitch, ft
D_o = tube O.D., ft

$$E = \frac{A_o U' \Delta T_{LM}}{\lambda(p_f - D_o)}$$

A_o = surface area per ft of tube; ft^2/ft
N_{vc} = Number of tubes in vertical tier at centerline of bundle

$$= \frac{D_B}{2p_f \cos\frac{\theta}{2}}$$

D_B = bundle diameter, ft
θ = tube layout angle, degrees
= 90° for rotated square
= 60° for triangular

Now, the h_b coefficient can be used with the overall U equation, including shell-side fouling, to calculate a final overall coefficient for boiling.

The maximum flux equation of Zuber[128] is suggested as another check for kettle reboilers:

$$q_{max} = 176\left[\frac{D_b L}{A_n}\right]\rho_v \lambda\left[\frac{g\sigma(\rho_L - \rho_v)}{\rho_v^2}\right]^{0.25} \quad (10\text{-}164)$$

where

q_{max} = tube bundle maximum flux, Btu/hr-ft^2
D_b = tube bundle diameter, ft
L = length of tube bundle, straight tube, or average for U-bundle, ft
A_n = net effective total bundle outside tube surface area, ft^2
g = gravitational constant, 4.17×10^8 ft/hr^2

The usual range of q_{max} for organic fluids is 15,000–25,000 Btu/hr (ft^2). For aqueous solutions, the range is 30,000–40,000 Btu/hr (ft^2)

Example 10-18. Kettle Type Evaporator—Steam in Tubes

Evaporate 25,000 lb/hr of CCl_4 at 55 psia and saturation temperature on shell side of a kettle-type U-tube evaporator. Use steam as heating medium:

1. Determine heat load, Q.

B.P. of CCl_4 at 55 psia at 128°C ≅ 262°F
Heat of vaporization: l_v of CCl_4 at 128°C = 71.5 Btu/hr
Heat duty: Q = 25,000 (71.5) = 1,790,000 Btu/hr

2. Determine steam conditions (saturated), and Δt.
Use Δt of approximately 60°F (refer to Table 10-26 for guide).

Steam temperature = 262 + 60 = 322°F
Steam pressure = 92 psia
Latent heat (from steam tables)
l_v steam = 893.4 Btu/hr

$$\text{Steam required} = \frac{1,790,000}{893.4} = 2,000 \text{ lb/hr}$$

3. Estimate unit size.
Use maximum heat flux Q/A = 12,000 Btu/hr/ft^2. Note that Table 10-26 indicates this value is quite safe. You could use a higher allowable flux. Many designs are operating based upon this conservative value.

A = Q/(Q/A)

$$A = \frac{1,790,000}{12,000} = 149 \text{ ft}^2$$

Select a unit as follows:

$^3/_4$-inch O.D. × 14 BWG × 32 ft, 0 in. steel U tubes, assume effective length is 31 ft, 0 in. for preliminary calculation:

$$\text{Number of tubes} = \frac{149}{31 \times .196\ ft^2/ft} = 24.5$$

From standard tubesheet layout or Table 10-9, select a 10-in. I.D. shell with 30 tubes, 2 passes, for first trial.

4. Determine condensing coefficient tube loading, Figure 10-67A.

$$G''_o = \frac{2{,}000}{0.5 \times 31 \times 23} = 5.6\ \text{lb/lin ft (equivalent)}$$

Condensate properties at estimated t_w of 300°F
Sp. G. = 0.916
k_a = 0.455
μ = 0.19 centipoise

Because of the low tube loading and physical properties of condensate, the value of the film coefficient is beyond the range of the chart. Therefore, the use of a h_{io} of 1,500 is conservative.

5. Determine the boiling coefficient.
Add a "dirt" factor of 0.001 to h_{io}:

$$\frac{1}{h_{io}} = \frac{1}{h_{io}} + .001 = \frac{1}{1{,}500} + .001 = 0.00167$$

$h_{io} = 1/.00167 = 600$

Calculate tube wall temperature, t_w. In this example:

t_{steam} = 322°F $\quad$ t_c = 262°F

First try:

Assume h_o = 300

$$t_w = 262 + \frac{600}{600 + 300}(322 - 262)$$
$$= 262 + 0.667(60) = 262 + 40$$
$$= 302°F$$

Δt_w = 302 − 262 = 40°F
h_o for Δt_w of 40°F is greater than 300, Figure 10-103.
Use 300 maximum (Kern's recommendation).

6. Determine the required unit size.
Add a .001 dirt factor to h_o.

$$\frac{1}{U} = \frac{1}{600} + \frac{1}{300} + 100 + \frac{L_w}{k}$$
$$= .00167 + .00333 + .001 + .0018 = .00618$$

$$U = \frac{1}{.00618} = 162\ \text{Btu/hr}\ (ft^2)(°F)$$

$$\text{Area required} = \frac{1{,}790{,}000}{69\ (162)} = 184\ ft^2$$

From Figure 10-27B, the equivalent tube length = 31.2 ft
Area available = 30 (31.2) (0.196) = 188 ft^2

The originally assumed unit is satisfactory.

It is to be noted that only the steam-condensing coefficient will change (lower tube loading, increasing h_{io}). Because an arbitrary maximum value was used for coefficients, the overall U will not change nor will the Δt. Therefore, only the available area of the new sized unit needs to be checked against the previously calculated required area. For a 12-in. shell with 32 tubes available:

$$\text{Area available} = 32 \times 31 \times .196 = 194\ ft^2$$

A 12-in. shell will be satisfactory:

$$S.F. = \frac{194 - 184}{184} = \frac{10}{184} = 5.44\%\ \text{with .002 dirt factor}$$

Area "safety factor" (which may be interpreted as more allowance for fouling):

$$= \frac{188 - 149}{149}(100) = 26\%$$

For a small unit such as this, 26% over surface is not too uneconomical. A smaller unit might be selected; however, if the tubes are shortened and the shell diameter is enlarged, the unit will be more expensive. Note that 24-ft (total length) tubes will give 146 ft^2 of surface. The only safety factor is in the knowledge that the flux selected, Q/A, appears to be quite low. If it were doubled (and this could be done), the smaller unit would be a reasonable selection.

Boiling: Nucleate Natural Circulation (Thermosiphon) Inside Vertical Tubes or Outside Horizontal Tubes

Natural circulation reboilers are effective and convenient units for process systems operating under pressure. They are usable in vacuum applications but must be applied with care, because the effect of pressure head (liquid leg) on the boiling point of the fluid must be considered. The temperature difference between the heating medium and boiling point of the fluid may be so small as to be impractical, regardless of the tube length in a vertical unit.

The recommended tube length is 8 ft in vertical units, with 12 ft being a maximum. Of course, some designs operate with 4- and 6-ft tubes; however, these are usually in vacuum service and physically are very large in diameter when compared to an 8-ft tube unit (see Figures 10-96D and 10-105).

The method presented here requires that the majority of the heat load be latent, with a reasonably small percentage, say 10–20%, being sensible load.

Gilmour[51–54] has presented a boiling film relation, which is the result of the correlation of data covering a good range of organic materials and water from subatmospheric to above atmospheric pressure. This range has been the problem in most other attempts at correlation. The correlation is reported to have been successfully used on hundreds of vaporizers and reboilers by the author. Palen and Small[90] have examined data using Gilmour's equations. It has the advantage of avoiding trial-and-error approaches.

Gilmour Method[52, 53] Modified

This process is applicable to vertical tube side vaporization only and to vertical and horizontal shell-side vaporization.

1. Calculate the heat duty.
2. Estimate a unit based upon suggested values of U from Tables 10-15 and 10-18A and the known LMTD. Check to be certain that Δt_o does not exceed critical value between shell side and tube wall or the tube side temperatures (however expressed).
3. Calculate film coefficient, h_s, by

$$h_s = \frac{\phi(c)(G_{gb})}{\left(\frac{D'G_{gb}}{\mu}\right)^{0.3}}\left(\frac{c\mu}{k_a}\right)^{-0.6}\left(\frac{\rho_L\sigma}{\rho^2}\right)^{-0.425}(\alpha) \quad (10\text{-}165)$$

h_s = boiling side coefficient, Btu/hr (ft²) (°F)
ϕ = type metal factor
ϕ = 0.001 for copper and steel tubes
ϕ = 0.00059 for stainless steel and chromium-nickel
ϕ = 0.0004 for polished surfaces
α = surface condition factor
α = 1.0 for perfectly clean conditions, no pitting or corrosion
α = 1.7 for average tube conditions
α = 2.5 for worst tube conditions

Note: This is *not* fouling correction. Read Figure 10-106.

a. For shell-side vaporization:

$$G_{gb} = \frac{V}{(D'L)}\left(\frac{\rho_L}{\rho_v}\right) \quad (10\text{-}166)$$

Note that Gilmour[54] suggests that the correction for the number of vertical tube rows given in reference 53 be omitted due to the generous fouling factor recommended.

where

G_{gb} = mass velocity of liquid, lb/hr (ft²). For outside horizontal tubes, use projected area (diameter × length) of the tube, not the outside surface area. This assumes that only half of the tube is effective for bubble release. This does not apply to actual heat transfer area.
V = vapor rate, lb/hr
T_s = saturation temperature of liquid, °R
ρ_L = density of liquid, lb/ft³
T_w = temperature of heating surface, °R
ρ_v = density of vapor, lb/ft³
σ = surface tension of liquid, lb/ft
c = specific heat of liquid, Btu/lb (°F)
μ = viscosity of liquid, lb/(hr) (ft)
P = pressure at which fluid is boiling, lb/ft² abs
D' = tube diameter, ft (side where boiling takes place)
D_s' = shell I.D., ft
h_s = boiling side film coefficient, Btu/hr (ft²) (°F)
L_o = length of shell, or length of one tube pass, ft
A = surface area of tube, ft²
For outside horizontal tube, use outside tube surface area. For vertical tubes with inside boiling, use inside surface area of tube, A_i.

ϕ = proportionality constant for type of tube material
Δt_b = boiling temperature difference between boiling fluid and wall surface, on boiling side, °F
x = weight percent vapor in fluid stream, for nucleate boiling only

b. For tube-side vaporization:

$$G_{gb} = \frac{V}{A}\left(\frac{\rho_L}{\rho_v}\right), \text{ mass velocity of liquid, lb/hr (ft}^2)$$

4. For boiling in shell side of horizontal unit, a check point is

$6.665\ V/\rho_v D_s' L_o$ must not exceed 1.0 (10-166A)

This is concerned with the maximum vapor rate from a horizontal shell.

5. Check this second factor for condensate flooding in horizontal units with a condensable heating medium, such as steam, in the tubes:

$$\text{ft}^3\text{ condensate/(sec)(tube)} = \frac{W}{(3{,}600)\rho n'} = 9.43 D_o^{2.56} H_c \quad (10\text{-}166B)$$

n' = number of tubes per pass
W = condensate rate, lb/hr
ρ = density, lb/ft³
D_o = tube diameter, outside, ft
H_c = height of segment of circle divided by diameter

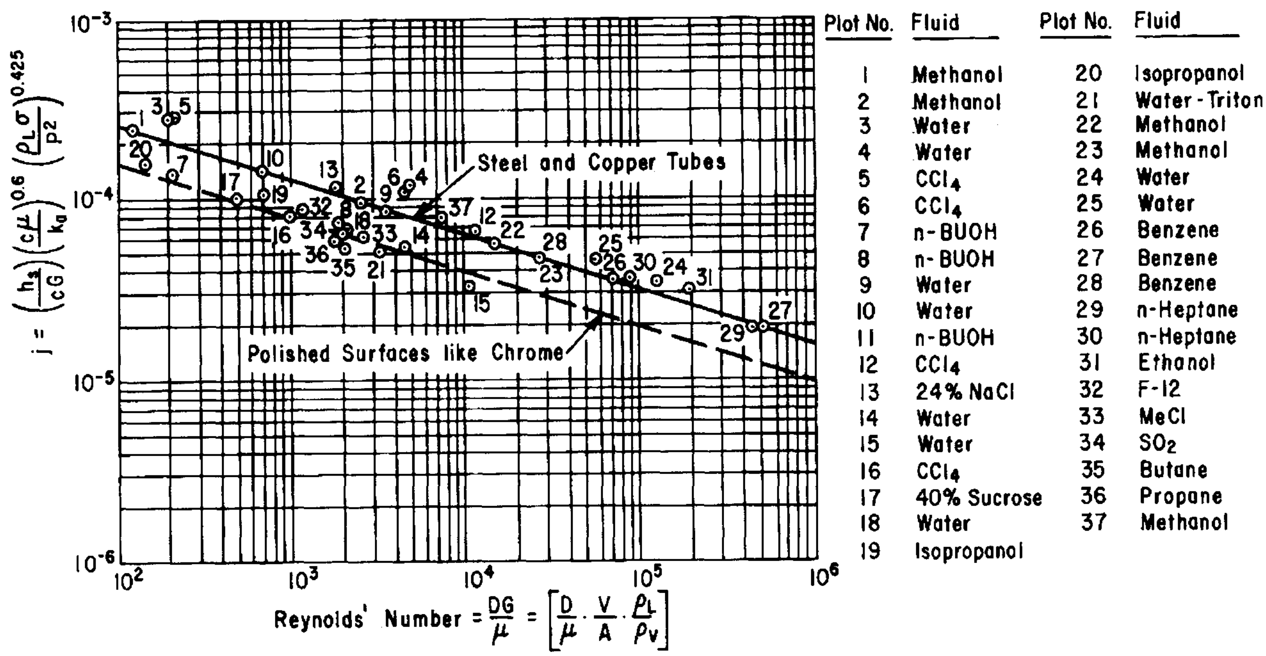


Figure 10-106. Gilmour correlation for nucleate boiling data. (Used by permission: Gilmour, C. H. *Chemical Engineering Progress,* V. 54, No. 10, ©1958. American Institute of Chemical Engineers. All rights reserved.)

Solve for H_c and then calculate the length of subtended arc. From the total circumference of the tube, the fraction of surface flooded can be calculated. If this fraction exceeds 0.3, recalculate the unit.

6. Calculate the film coefficient for fluid on side of tube opposite from the one associated with the boiling or vaporizing operation.
7. Use fouling factors for tube and shell side if known; otherwise use 0.002 for tubes 8–12 ft long and 0.001–0.002 for shorter tubes.
8. Calculate overall U.
9. Calculate Δt between boiling fluid and wall surface on boiling side.
10. Calculate surface area:

$$A = \frac{Q}{U\Delta t_b}$$

11. Compare the calculated and assumed areas. If acceptable, the design is complete from a thermal standpoint. If not, reassume the area of step 2 and repeat until a balance is achieved.
12. Pressure drop

    Boiling on shell side: usually negligible unless tubes are very small and close together. For preferred 45° rotated square pitch with 1.25 d_o, Δp_s will be low.

    Boiling in tubes: usually low, 3–9 in. fluid. Evaluate using two-phase flow.

13. Inlet and outlet nozzles for boiling-side fluid:
    a. Vertical thermosiphon units[54]

$$\text{Vapor out, } D_n = d_{it}\sqrt{N_t}, \text{ in.} \tag{10-160}$$

d_{it} = I.D. of tube, in.
N_t = number of tubes (preferable of 1–2 in. size)
Liquid mixture in, $D_n = {}^1/_2$ (vapor outlet nozzle) size

Note that the liquid inlet must be inline at bottom, and the vapor out must be inline at top (Figure 10-105). For a side outlet vapor nozzle, increase the heat transfer area by 30%.[53, 54]

    b. Horizontal or vertical shell-side boiling, size for low velocities and pressure drops.

Gilmour's basic correlation has been presented in graphical form by Chen,[26] in Figures 10-107A, 10-107B, 10-108, and 10-109. These charts are based on a metal wall factor, ϕ, of 0.001, and if other values are considered, multiply the calculated h value by the ratio of the new factor to 0.001. The use of the charts follows:

1. For a given fluid condition and assumed size of reboiler, evaluate physical property factor ϕ_1 from Figure 10-107A and physical property factor ϕ_2 from Figure 10-107B.
2. Read boiling coefficient, h, from Figure 10-108 using $\phi_x = (\phi_1)(\phi_2)$.

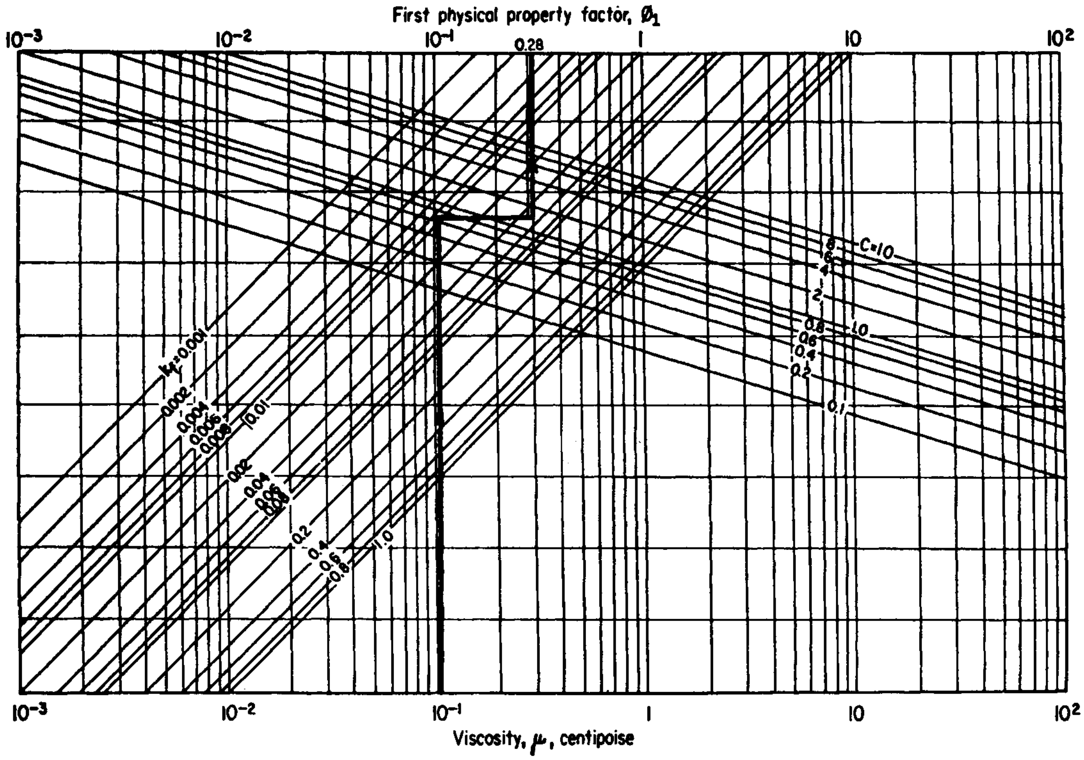


Figure 10-107A. First physical property factor for boiling coefficient. (Used by permission: Chen, Ning Hsing. *Chemical Engineering,* V. 66, No. 5, ©1959. McGraw-Hill, Inc. All rights reserved.)

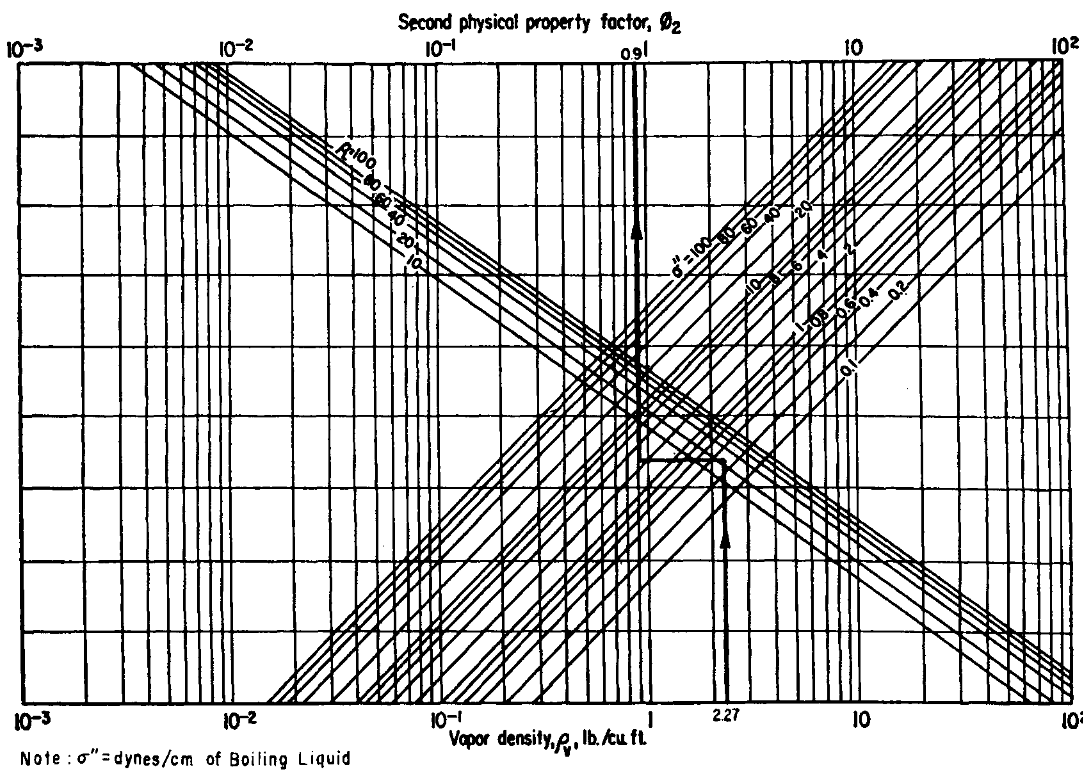
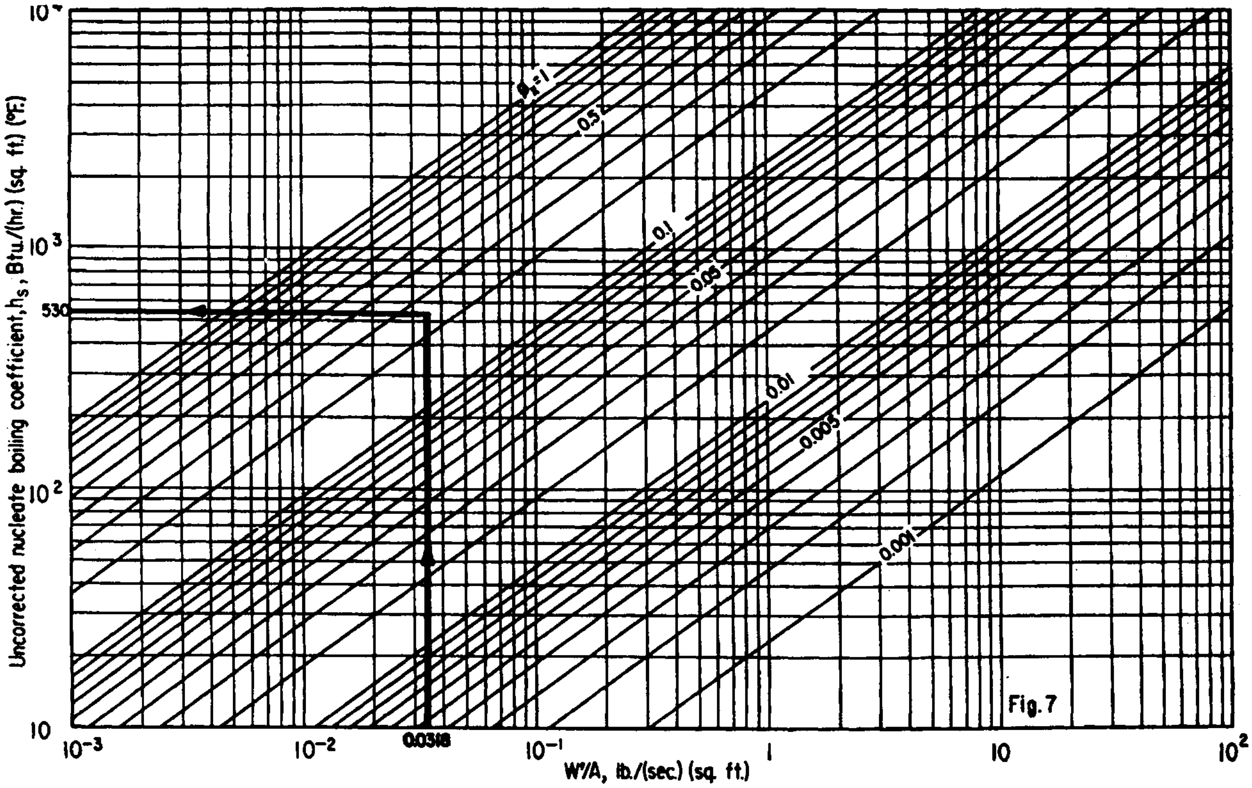


Figure 10-107B. Second physical property factor for boiling coefficient. (Used by permission: Chen, Ning Hsing. *Chemical Engineering,* V. 66, No. 5, ©1959. McGraw-Hill, Inc. All rights reserved.)



Tube-Size Correction Factors

Size	F_D'	Size	F_D'
⅝ in. O.D.	1.000	1 in.	0.866
¾	0.945	1¼	0.811

Figure 10-108. Uncorrected nucleate boiling coefficient. (Used by permission: Chen, Ning Hsing. *Chemical Engineering,* V. 66, No. 5, ©1959. McGraw-Hill, Inc. All rights reserved.)

3. Apply the tube size multiplier from table associated with Figure 10-108 and also the multiplier for pressure correction from Figure 10-109. Note that for high pressure systems a pressure can become quite large, and some designers limit it to an arbitrary value of about 3,000.

Suggested Procedure for Vaporization with Sensible Heat Transfer

1. Follow the general procedure for vaporization only.
2. Determine the sensible heat load separate from vaporization.
3. For organic liquids, evaluate the natural convection film coefficient from Figure 10-103. Equation 10-29 may be used for the inside horizontal tube by multiplying the right side of the equation by 2.25 $(1 + 0.010\ Gr_a^{1/3})/\log Re$.
4. Calculate the required area for sensible heat transfer.

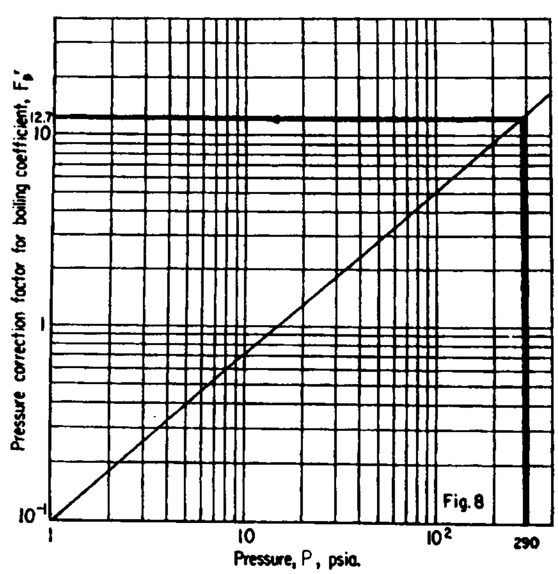


Figure 10-109. Pressure correction factor. (Used by permission: Chen, Ning Hsing. *Chemical Engineering,* V. 66, No. 5, ©1959. McGraw-Hill, Inc. All rights reserved.)

5. Add area requirements of sensible heat to the area required for vaporization to obtain the total area.
6. Follow steps 7 (Gilmour method), etc., of the procedure for *vaporization only.* If baffles are added for sensible heat (not assumed in free convection), then pressure drop will be affected accordingly. Gr_a is the Grashof number using properties at average fluid temperature, $= D_i^3 \rho g \beta' \Delta t/\mu^2$.

Procedure for Horizontal Natural Circulation Thermosiphon Reboiler

These units normally do not have a disengaging space but allow the vapor-liquid mixture to enter the distillation unit or other similar item of equipment. Feed is from the bottom with a split flow on the shell side by means of a shell-side baffle in the center being open at each end.

This unit is usually used as the reboiler for the distillation column and, in this service, operates by the thermosiphon action of the difference in static head in the column and in the vapor-liquid phase leaving the reboiler. When tied into the bottom chamber, the liquid is usually recirculated many times, vaporizing only 10–25% of the reboiler feed per pass; however, when used as a draw-off from the bottom tray seal pan, the feed to the reboiler is not recirculated flow. The basic operation is the same, however.

Kern Method[70]

1. Follow the procedure for steps 1, 2, 3, 4, and 5 of the earlier section, "Nucleate or Alternate Designs Procedure."
2. If sensible heat exchange exists, follow steps 2–6 of the "Suggested Procedure for Vaporization with Sensible Heat Transfer," previously in this chapter.
3. Pressure drops must be kept low through the piping to the reboiler and through the reboiler to avoid expensive elevation of the distillation equipment. Kern suggests 0.25 psi as preferable to 0.50 psi; however, the final economic balance of the system will determine the allowable pressure drop, because for the same system, either the piping or the exchanger must become larger if the pressure drop is to be reduced.

The length of the tubes should not be selected more than 4.5–5.5 times the shell diameter. This performance may be increased by placing two inlets in the bottom and two vapor outlets in the top, and at the same time adding shell-side longitudinal baffling to split the flow into four paths upon entrance. The paths recombine before leaving.

The recirculation ratio for a unit is the lb rate of liquid leaving the outlet compared to the lb rate of vapor leaving. The liquid recirculation flow rate entering the unit is set by the differential pressure driving the system.

Vaporization Inside Vertical Tubes; Natural Thermosiphon Action

The vertical thermosiphon reboiler is a popular unit for heating distillation column bottoms. However, it is indeed surprising how so many units have been installed with so little data available. This indicates that a lot of guessing, usually on the very conservative side, has created many uneconomical units. No well-defined understanding of the performance of these units exists. Kern's[70] recommended procedure has been found to be quite conservative on plant scale units; yet it has undoubtedly been the basis for more designs than any other single approach. For some systems at and below atmospheric pressure operation, Kern's procedure gives inconsistent results. The problem is in the evaluation of the two-phase gas-liquid pressure drop under these conditions.

For units that are vertical one-pass in tubes with liquid in the bottom entrance and a top exit of the liquid-vapor mixture, the separation is accomplished in the equipment to which it is attached, usually a distillation column (Figure 10-96D).

For services in which fouling is high or in which downtime cannot be tolerated, two reboilers may be installed on the same distillation column. These reboilers may each be half sized so that downtime will be limited to a half-capacity operation; each may be two-thirds sized; or each may be a full 100% spare. The latter is, of course, the most expensive from an equipment investment standpoint but may pay for itself in uptime.

The tubes are usually $1\ {}^{1}/_{4}$ -in. O.D. but never smaller than 1-in. O.D. because the flow contains vapor as well as liquid. The recirculation ratio; i.e., liquid-to-vapor ratio in the outlet, is seldom less than 5 and more often is 10–15, sometimes reaching 50.

Fair's Method[45]

This method for vertical thermosiphon reboilers is based on semi-empirical correlations of experimental data and is stated to predict heat transfer coefficients ±30 percent, which is about the same range of accuracy for most boiling coefficient data. The advantage of this method is that it has had significant design experience in the industry to support it. It is also adaptable to other types of reboilers used in the industry. See Figures 10-110 and 10-111.

Fair's[45] presentation provides the development of the design technique. The method recognizes two-phase flow in a vertical reboiler and points out that slug-type flow is most predominate and that mist-flow should be avoided.

This procedure develops stepwise calculations along the tube length, using increments of length or vaporization. The increments are chosen small enough so that average values of R_L, R_g, ϕ, X_{tt}, and h_t may be used in the difference

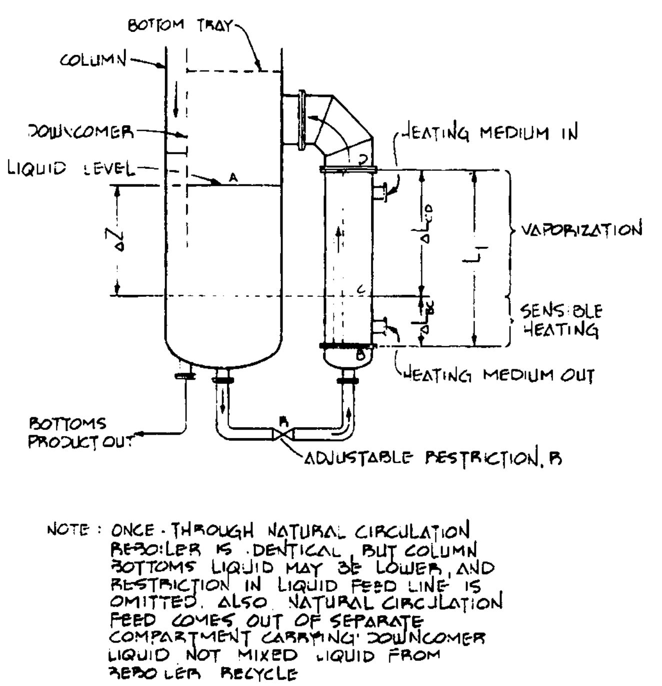


Figure 10-110. Typical vertical tube-side thermosiphon reboiler. (Used by permission: Fair, J. R. *Petroleum Refiner*, Feb. 1960, p. 105. ©Gulf Publishing Company. All rights reserved.)

equations. These calculations are well suited to computer application.

Select an increment of vaporization, beginning at the end of the sensible heat zone. Use an average value of x for the increment calculations. The circulation rate that must be already developed on the basis of average conditions should be used for the initial calculations.

Following Fair's[45] method outlined in the article, determine, select, or assume the following based on process requirements (reproduced by permission of the author and publisher, all rights reserved):

1. Boil-up rate.
2. Reboiler outlet temperature, pressure, and composition.
3. Physical properties at expected operating temperatures. See Figure 10-112 for temperature-pressure effects in vertical thermosiphon reboilers.

To facilitate design calculations, Figures 10-114–10-118 have been prepared to give the following information:

Figure 10-114—R_L values on the basis of Lockhart and Martinelli.

Figure 10-116—ϕ^2 values on the basis of Figure 10-113.

Figure 10-117—h_{tp}/h_L values on the basis of Equation 32 (Ref. 45), with modification at $1/X_{tt}$ values less than 0.2 as suggested by Dengler and Addoms.[12]

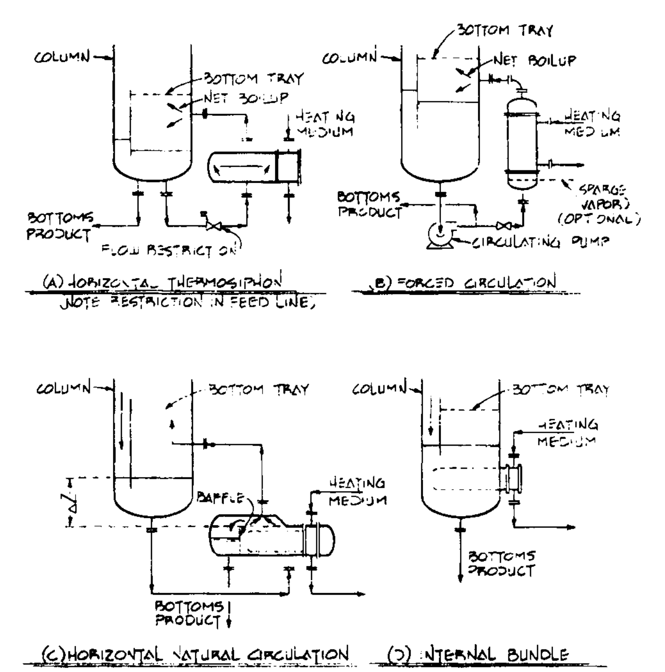


Figure 10-111. Typical reboiler arrangements. (Used by permission: Fair, J. R. *Petroleum Refiner*, Feb. 1960, p. 105. ©Gulf Publishing Company. All rights reserved.)

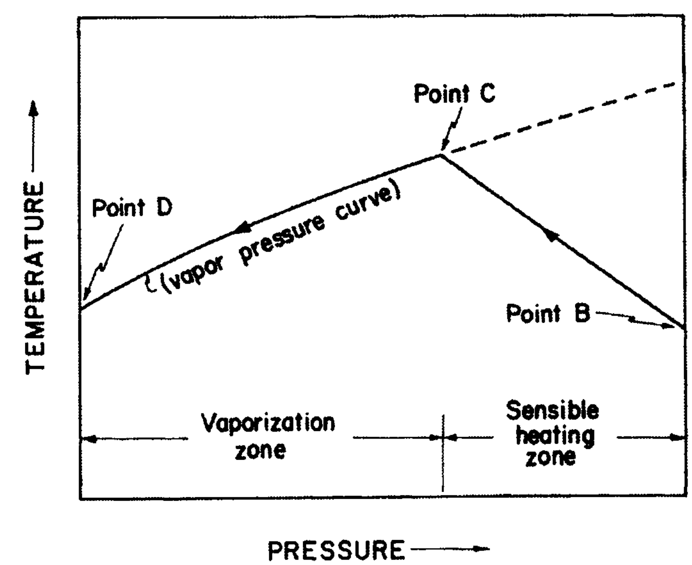


Figure 10-112. The temperature scale is accentuated to show the temperature-pressure effects in themosiphon reboilers. (Used by permission: Fair, J. R. *Petroleum Refiner*, Feb. 1960, p. 105. ©Gulf Publishing Company. All rights reserved.)

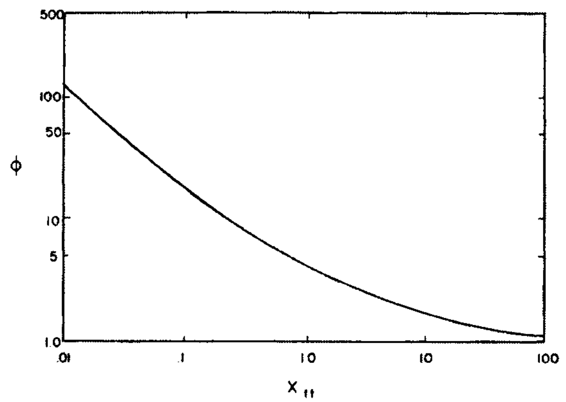


Figure 10-113. The factor φ for two-phase turbulent-turbulent flow.[28] Note: Reference number on chart is in Fair's article. (Used by permission: Fair, J. R. *Petroleum Refiner,* Feb. 1960, p. 105. ©Gulf Publishing Company. All rights reserved.)

Figure 10-115—$1/X_{tt}$ values for use in Figure 10-118.

Figure 10-118—α for correcting the nucleate boiling coefficient.

Flow pattern limits in Figure 10-118 are based on the data of Govier *et al.*,[15] Yoder and Dodge,[40] Dengler,[11] and Baker.[3]

The parameter ψ in Figures 10-114–10-117 is defined as

$$\psi = \left(\frac{\rho_g}{\rho_L}\right)^{0.5}\left(\frac{\mu_L}{\mu_g}\right)^{0.1} = \frac{X_{tt}}{(W_L/W_g)^{0.9}} \quad (10\text{-}168)$$

For cases in which a significant percentage change in pressure occurs across the reboiler tubes, ψ is not constant. In general, however, an average constant value may be assumed.

Design calculations may be made by one of two methods:

- Stepwise calculations along the tube length, using increments of length or vaporization. Increments are chosen small enough so that average values of R_L, R_g, φ, X_{tt}, h_t, etc., may be used in the difference equations.
- Simplified calculations using average values of variables for the overall tube length.

Sufficient information is given in this chapter to enable the more rigorous stepwise calculations. These calculations are ideally suited to digital computer solution and have been programmed by the author for a machine of the smaller type.

Emphasis in this section is given to the simplified method, because it is convenient to use and yet sufficiently reliable for most design cases. Thus, designers without ready access to a computer may quickly rate existing reboilers or design new ones.

Process Requirements

1. From fractionator calculations list
   a. Boilup rate.
   b. Reboiler outlet temperature, pressure, and composition.
2. Obtain physical property data:
   a. Liquid and gas (vapor) densities, ρ_L and ρ_g.
   b. Liquid and gas (vapor) viscosities, μ_L and μ_g.
   c. Liquid specific heat, c_L.
   d. Liquid thermal conductivity, k_L.
   e. Latent heat of vaporization, λ.
   f. Surface tension, σ.
   g. Slope of vapor pressure curve, $(\Delta t/\Delta p)_s$.

Preliminary Design

1. Select tubing material and dimensions.
2. Select heating medium.
3. Estimate overall coefficient U using resistance (Table 10-13B).
4. Calculate required surface and tube number.

Circulation Rate

1. Select flow loop geometry, i.e., type reboiler arrangement.
2. Assume exit fractional vaporization, x_E.
3. Evaluate sensible heating zone (Equation 10-169).
4. Obtain average values:
   a. Two-phase density, ρ_{tp}, at $x_E/3$.
   b. Pressure drop factor, φ, at 2 $x_E/3$.
5. Obtain ρ_{tp} and φ for exit conditions.
6. Calculate circulation rate (Equation 10-173).
7. Calculate boilup and check against required value.
8. Repeat calculations, adjusting flow loop geometry if necessary, until assumed x_E gives the proper boilup rate.

Heat Transfer—Stepwise Method

1. Choose an increment of vaporization, starting at the end of the sensible heating zone. Use the arithmetic average value of x for increment calculations. The circulation rate already obtained on the basis of average conditions should be used for initial calculations.
2. Calculate or obtain values for
   a. Two-phase density, ρ_{tp}.
   b. Pressure drop factor, φ.
   c. Convective transfer coefficient, h_{tp}.
   d. Boiling coefficient, h_b (Table 10-30 or other source, or Equation 10-185).
   e. Boiling coefficient correction factor, α (Figure 10-118).
   f. Combined film coefficient, h_v (Equation 10-186).

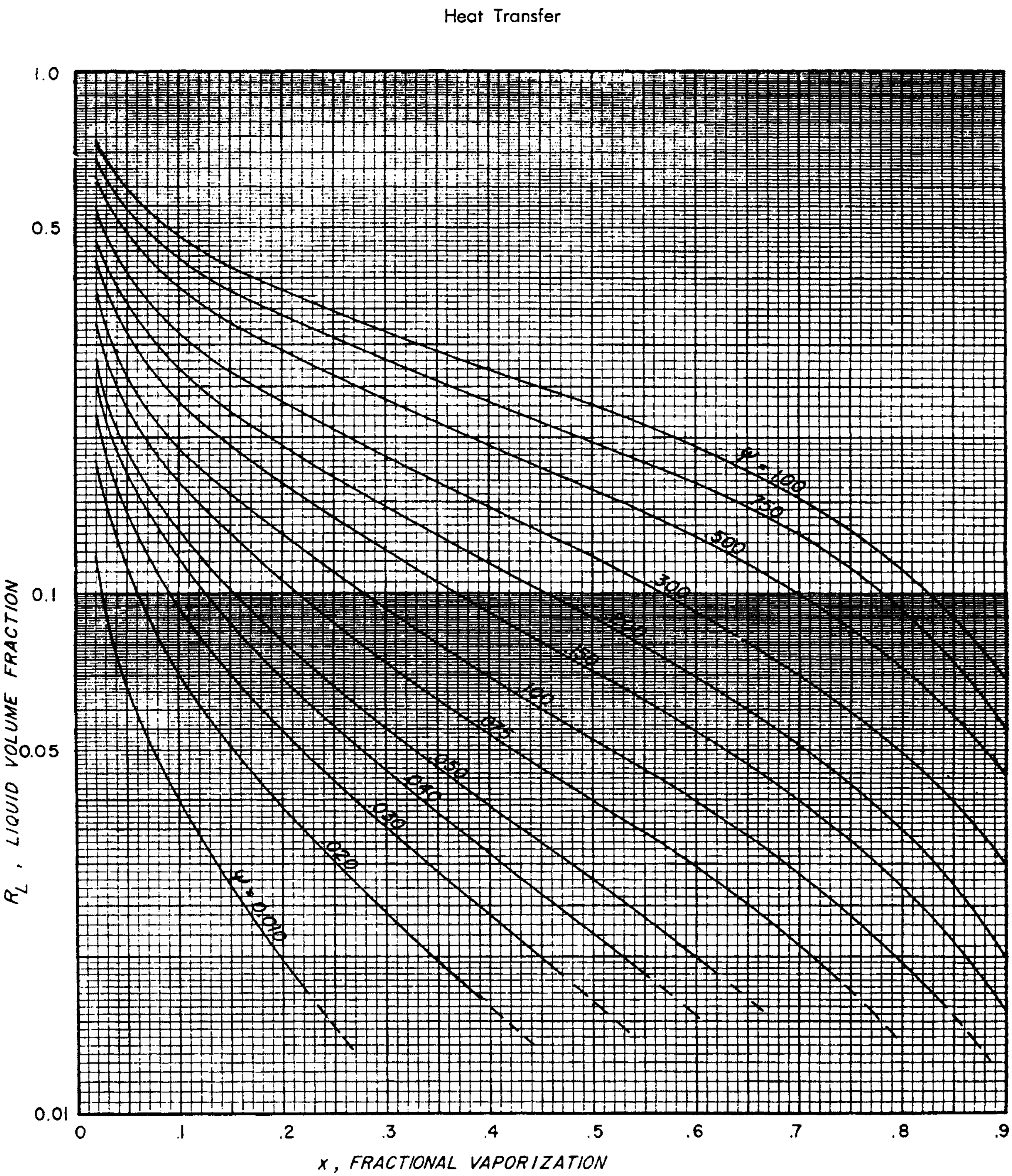


Figure 10-114. Design chart for liquid volume fraction using parameter defined in equation 10-168. (Used by permission: Fair, J. R. *Petroleum Refiner*, Feb. 1960, p. 105. ©Gulf Publishing Company. All rights reserved.)

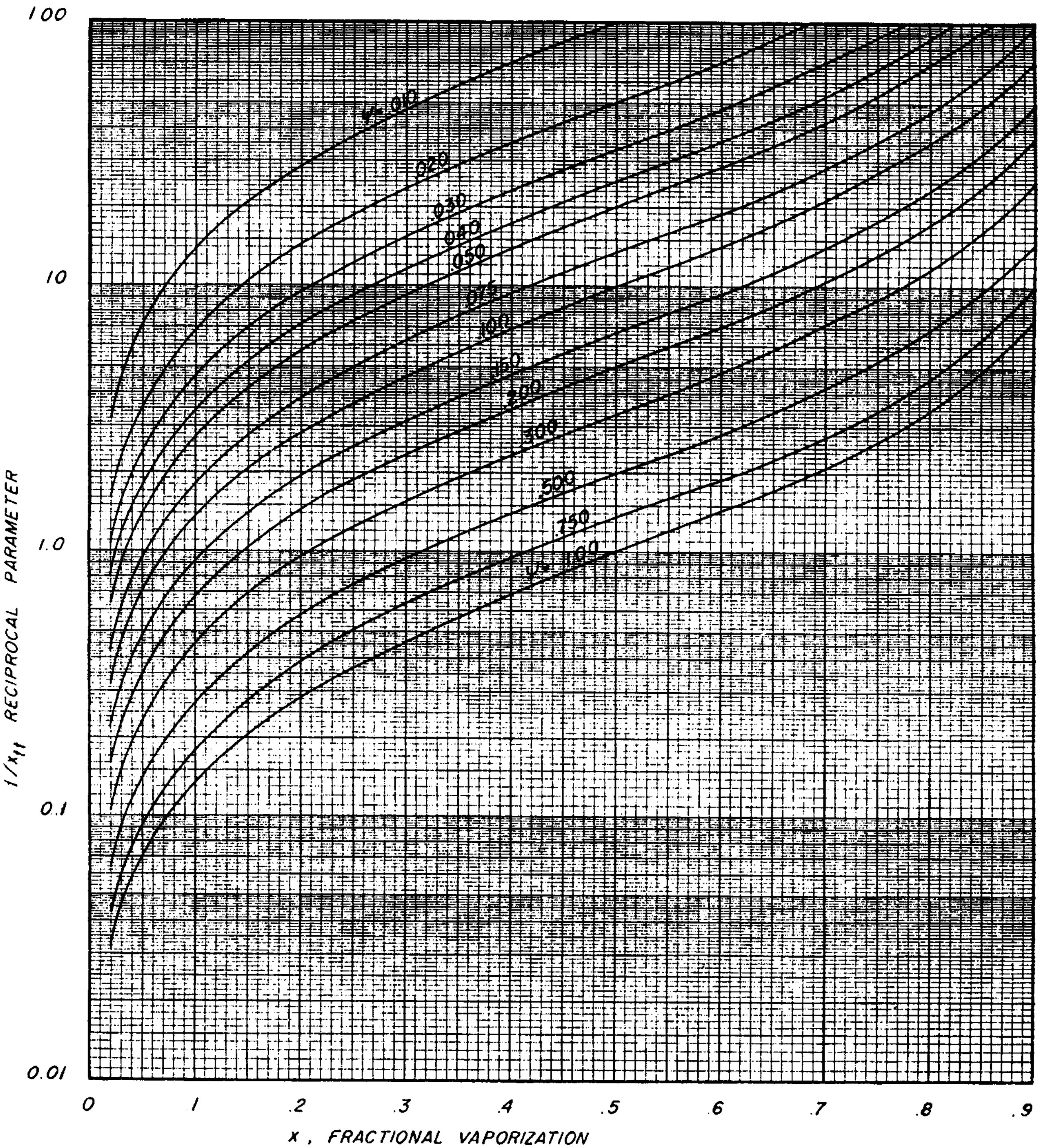


Figure 10-115. Design chart for correlating parameter X_{tt}. (Used by permission: Fair, J. R. *Petroleum Refiner,* Feb. 1960, p. 105. ©Gulf Publishing Company. All rights reserved.)

g. Length of increment, based on h_v.
h. Static pressure loss (Equation 10-172).
i. Frictional loss, Lockhart and Martinelli.[111]
j. Acceleration loss: the acceleration loss term of

$$-\Delta P_{acc} = \frac{1}{g_c}\int \rho_{tp} V_{tp} dV_{tp} = \frac{G_T}{g_c}(V_{tp,\,out} - V_{L,\,in}) \qquad (10\text{-}169)$$

In the usual case, liquid and gas do not issue from the reboiler with equal velocities.

k. Total pressure loss.

3. Continue stepwise calculations to end of tubes. After the pressure loss in exit piping is taken into account, the residual pressure should equal the fixed pressure below the bottom tray in the tower.

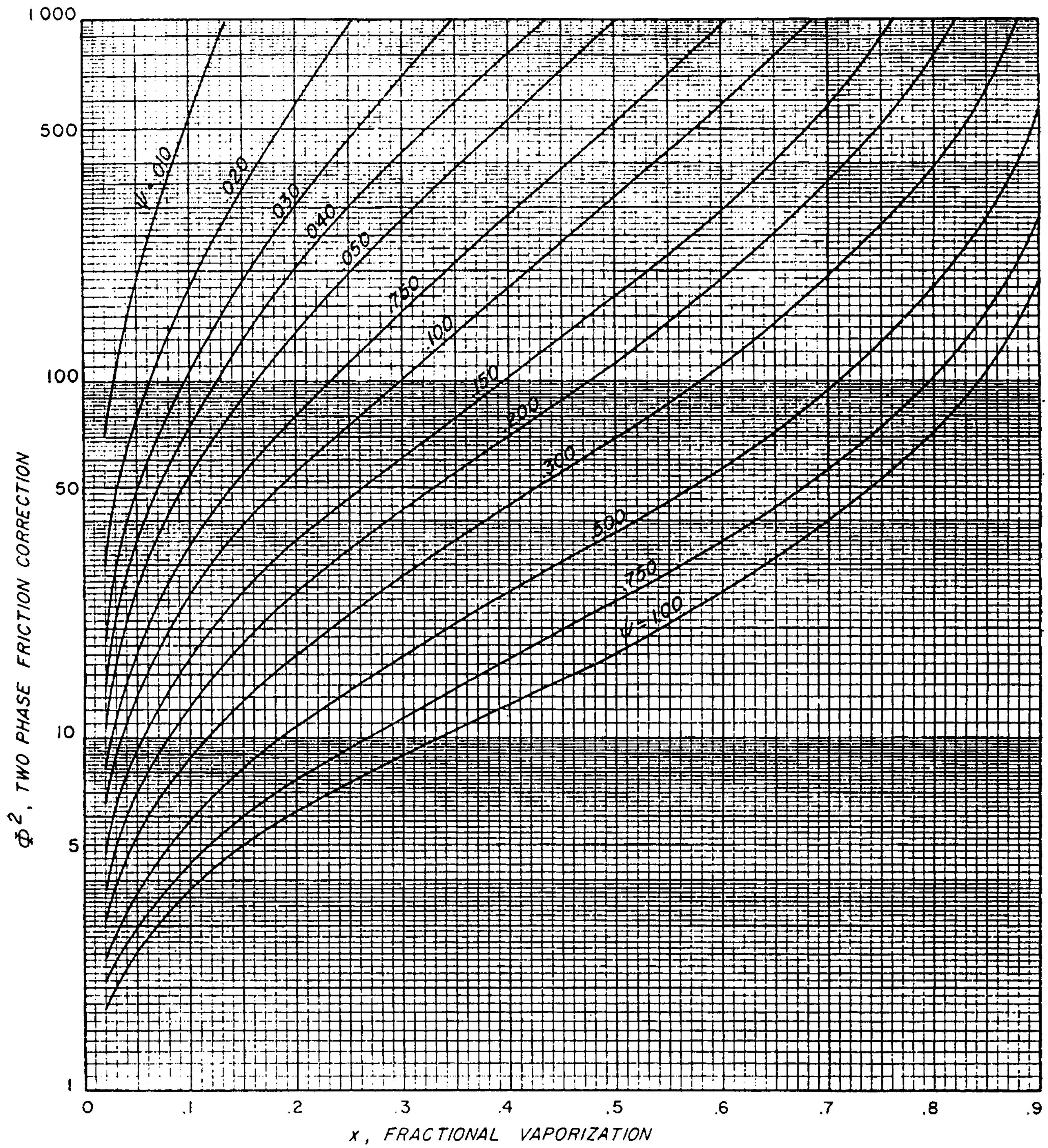


Figure 10-116. Design chart for ϕ^2 values. (Used by permission: Fair, J. R. *Petroleum Refiner*, Feb. 1960, p. 105. ©Gulf Publishing Company. All rights reserved.)

4. If the pressures do not match, the calculations are repeated for a different circulation rate. Alternately, the circulation rate may be kept constant, and pressure drop contributions (inlet line value, exit line diameter, etc.) can be adjusted.

5. After the proper pressure balance-heat transfer relationships are established, the calculations are summarized in connection with other heat flow resistances in the reboiler.

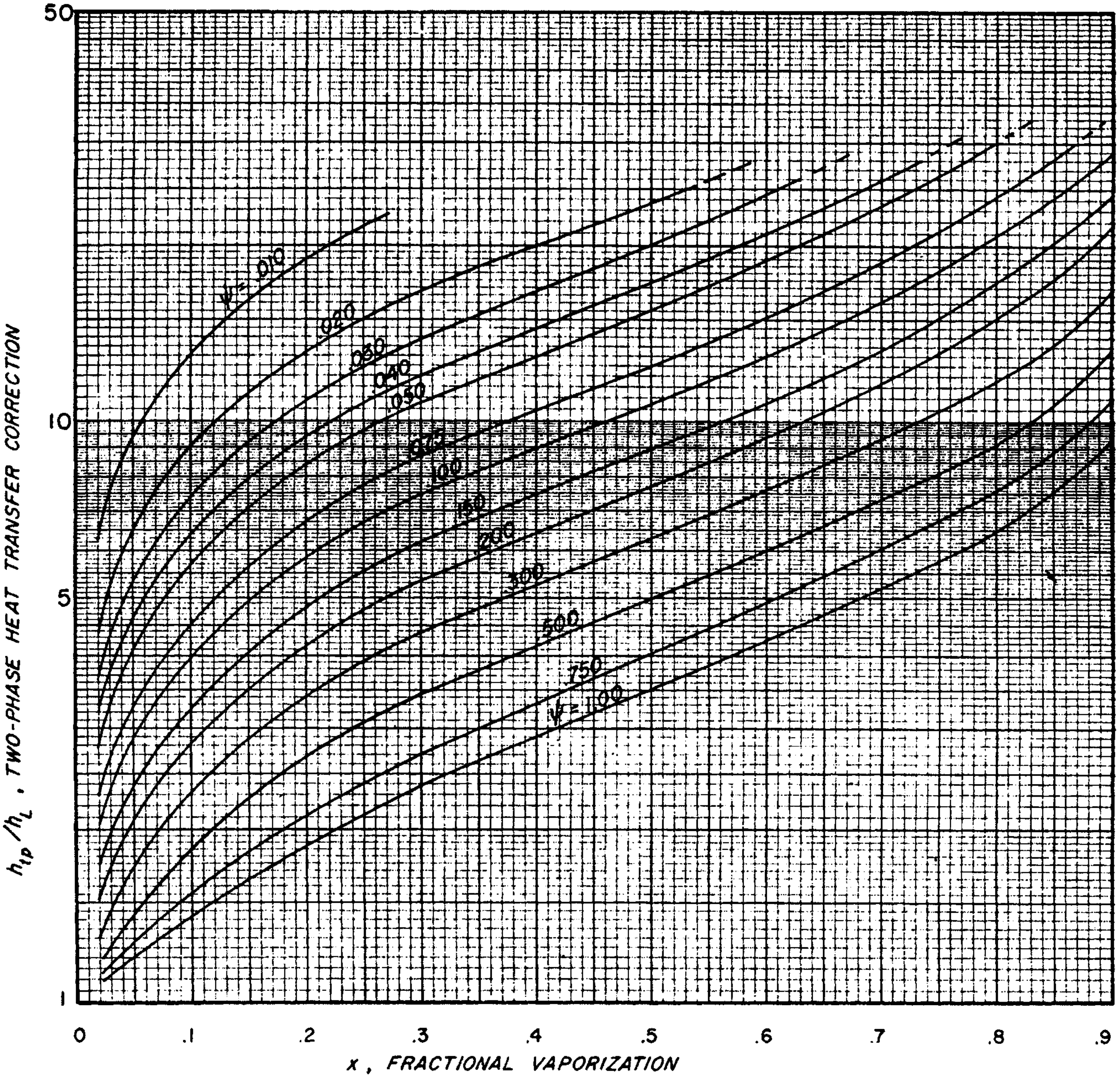


Figure 10-117. Design chart for two-phase heat transfer corrections. (Used by permission: Fair, J. R. *Petroleum Refiner*, Feb. 1960, p. 105. ©Gulf Publishing Company. All rights reserved.)

A. Circulation Rate

See Figure 10-110 and Table 10-29.

1. Select flow loop geometry, i.e., type of reboiler arrangement.
2. Assume exit fractional vaporization, x_E. Estimate range 15–40%.
3. Evaluate sensible heating zone by
   Fractional tube length devoted to sensible heating:[145]

$$\frac{p_B - p}{p_B - p_A} = \frac{(\Delta t/\Delta p)_s}{\dfrac{-\Delta t/\Delta L}{\Delta p/\Delta L} + \left(\dfrac{\Delta t}{\Delta p}\right)_s} \quad (10\text{-}170)$$

From a heat balance:

$$\Delta T/\Delta L = \frac{\pi D_t N_t h_l (t_w - t_l)}{3{,}600 W_T c_l} \quad (10\text{-}171)$$

h_l is from Dittus-Boelter Equation 10-175

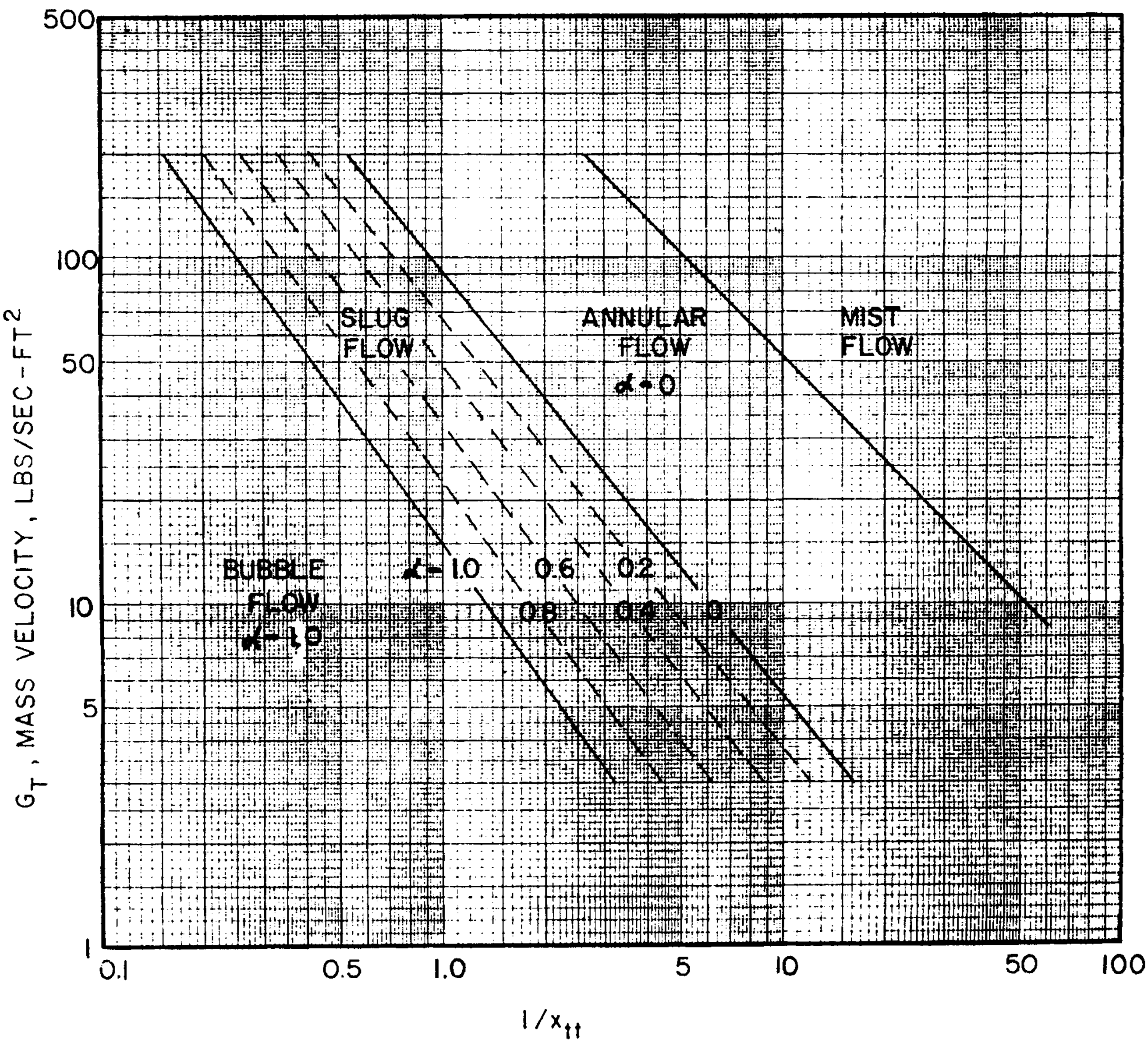


Figure 10-118. Design chart for α values. (Used by permission: Fair, J. R. *Petroleum Refiner,* Feb. 1960, p. 105. ©Gulf Publishing Company. All rights reserved.)

Table 10-29
Values for Short-Cut Calculation of Circulation Rate (Equation 10-181)

Value	Definition	Remarks
$\bar{x}$	Two-thirds of outlet fractional vaporization	$\bar{x} = 2x_E/3$*
$\bar{R}_L$	Liquid volume fraction based on one-third of outlet fractional vaporization	Figure 10-114 for $\bar{x} = x_E/3$*
$\bar{\rho}_{tp}$	Two-phase density based on $\bar{R}_L$	Equation 10-179
$\bar{\phi}^2$	Pressure drop ratio based on *two-thirds* the outlet fractional vapor	Figure 10-114 for $\bar{x} = 2x_E/3$*
ΔL_{CD}	Total tube length in which vaporization occurs	
ΔZ	Vertical distance in which two-phase flow occurs	Figures 10-110 and 10-111

*For sparged reboilers where entering $x \neq 0$, $\bar{x} = \dfrac{x_{ia} + 2x_E}{3}$, and $\bar{R}_L$ is based on $\bar{x} = \dfrac{2x_{ia} + x_E}{3}$

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This represents the fraction of the total available head between points A and B, which represents the sensible heating zone. This neglects liquid friction in the sensible zone and assumes the liquid level in the distillation column is maintained even with the top of the tubesheet. Equation 10-170 then gives the fractional tube length devoted to sensible heating. Refer to Figure 10-110 and note that:

p_B = total pressure at point B in flow loop, lb/in^2 (abs); (point at entrance to vertical reboiler)
p_A = total pressure at point A in flow loop, lb/in^2 (abs) (liquid level in distillation column bottoms)
p = total pressure, lb/in^2(abs)
Z = vertical height, ft
g = gravitational constant, 32.2 ft/sec^2
g_c = conversion factor, 32.2 $(lb)_m$ (ft)/$(lb)_f$ (sec^2)
π = 3.1416
D_i = I.D. of tube, ft
N_t = number of tubes
t_w = temperature of tube wall, °F
t_l = temperature of liquid phase, °F
W_T = mass rate, total flow, lb_m/sec
c_l = specific-heat, liquid phase, Btu/lb (°F)
h_l = heat transfer coefficient, liquid phase, Btu/hr-ft^2-°F
x_E = weight fraction of vapor or gas (quality), dimensionless at reboiler exit
$(\Delta t/\Delta p)_s$ = slope of vapor pressure curve. This may be calculated from Antoine type vapor pressure equation or obtain from a plot.
ρ_{tp} = two-phase density lb_m/ft^3

4. Obtain average values:
   a. two-phase density, ρ_{tp}, at $x_E/3$.
   b. pressure drop factor, ϕ, at 2 $x_E/3$.

where

ρ_{tp} = two-phase density, lb_m/ft^3
R_g = volume fraction of phase, dimensionless, gas phase
R_L = 1 − R_g; volume fraction of phase, dimensionless, liquid phase
$\rho_{tp} = \rho_g R_g + \rho_L R_L$
Δt = overall temperature difference, °F
Δp = pressure loss, lb/in^2
ΔP = pressure loss, lb/ft^2
p = total pressure, lb/in^2 abs.
P = total pressure, lb/ft^2 abs.
L = equivalent length of pipe, ft
BC = tube length for sensible heating, Figure 10-110, ft
CD = tube length for vaporization, ft, Figure 10-110
G = mass velocity, lb/(sec) (ft^2)
D_i = I.D. of tube, ft
G_t = mass velocity in tube, lb/(sec) (ft^2 of cross-section)
F_{B-C} = Friction loss from part B to part C in tubes, feet liquid
g = gravitational constant, 32.2 ft/(sec) (sec)
k = thermal conductivity, Btu/(hr) (ft^2) (°F/ft)
μ = viscosity, lb/(ft)(hr)
R = volume fraction of phase, dimensionless
x = weight fraction of vapor or gas (quality), dimensionless
X_{tt} = correlating parameter, dimensionless, for Figure 10-113
Z = vertical height, ft
ΔZ = vertical distance in which two-phase flow occurs, ft
ϕ = parameter for two-phase flow, dimensionless
$(\Delta t/\Delta p)_s$ = slope of vapor pressure curve
CD = tube length for vaporization (Figure 10-110)
D = point D in flow loop (Figure 10-110)
f = Force
fh = heating medium fouling
s = saturated
tp = two-phase mixture
T = total flow
v = vaporization
w = tube wall
a = cross-sectional area, ft^2
A = total (inside) surface for heat transfer, ft^2
c = specific heat, Btu/(lb) (°F)
d = differential operator
f = fanning friction factor, dimensionless
F = friction loss, (ft) $(lb_f)/lb_m$
g_c = conversion factor, 32.2 (lb_m) (ft)/(lb_f) (sec^2)
L = equivalent length of pipe, ft
q = heat transfer rate, Btu/hr
r = resistance to heat transfer, (hr) (ft^2) (°F)/Btu
r′ = composite resistance to heat transfer, (hr) (ft^2) (°F)/Btu
t = temperature, °F
T = absolute temperature, °R
U = overall heat transfer coefficient, Btu/(hr) (ft^2) (°F)
V = linear velocity, ft/sec
x = weight fraction of vapor or gas (quality), dimensionless
X_{tt} = correlating parameter, dimensionless

Greek letters

α = correction factor for nucleate boiling, dimensionless
β = correction factor for convective transfer, dimensionless
γ = acceleration loss group (Equation 10-169), dimensionless
ΔP = pressure loss, lb_f/ft^2
Δt = overall temperature difference, °F
λ = latent heat of vaporization, Btu/lb_m
μ = viscosity, lb_m/(ft) (hr)
π = constant, 3.1416 . . .
ρ = density, lb_m/ft^3
$\bar{\rho}$ = average density, lb_m/ft^3
σ = surface tension, lb_f/ft
$\bar{\phi}$ = average value of ϕ
ψ = parameter for two-phase physical properties, dimensionless

Subscripts

BC = tube length for sensible heating (Figure 10-110)
C = point C in flow loop (Figure 10-110)

fp = process-side fouling
F = friction
g = gas phase
h = heating medium
i = inlet (feed leg) piping system
m = mass
t = tube
l = L = liquid base
b = boiling
p = process on boiling side
tp = two phase
A = point A in flow loop, Figure 10-110
B = point B in flow loop, Figure 10-110
C = point C in flow loop, Figure 10-110
E = reboiler exit system

At pressures greater than about 100 psig, the slope of the vapor pressure curve, $(\Delta t/\Delta p)_s$ is low enough not to influence the sensible heating zone equation, as most of the tube is in vaporization. However, at low pressures and vacuum service, a large portion of the tube is in sensible heat.

The pressure at the inlet to the tubesheet at point B,[45] Figure 10-110:

$$p_B = \frac{(Z_A - Z_B)\rho_L g}{144 g_c} + p_A - \frac{\rho_L(\Delta F_{in})}{144} \quad (10\text{-}172)$$

Static pressure loss in outlet leg where density, ρ_{tp}, varies with vaporization:

$$-\Delta P_{static} = g/g_c \int \rho_{tp} dZ \quad (10\text{-}173)$$

where
ΔF = friction loss at inlet, ft of liquid
Z = vertical height, ft
p_B = boiling pressure, lb/in^2 abs
g = acceleration of gravity, ft/sec^2

Subscripts
A = point A in flow loop
B = point B in flow loop

5. The calculation of the term for Equation 10-170, Figure 10-110:

$$-\Delta p/\Delta L = \frac{\rho_l g_c}{144 g} + \frac{\Delta F_{B-C}}{\Delta L} \quad (10\text{-}174)$$

Fair[45] states that the second term in the preceding equation may be neglected. Fair reports that typical units show the sensible heating zone at 4–60% for ΔT = 20°F, and 4–49% for ΔT = 30°F for selected organics and also water. The values vary with pressure, Table 10-30.

6. The convection heat transfer rate inside the tubes is expressed by the Dittus-Boelter equation:[45, 82]

$$h_l = 0.023 \frac{k_l[3{,}600 D_i G_t]^{0.8}[c_l \mu_l]^{0.4}}{d_i[\mu_{l,b}][k_l]_b} \quad (10\text{-}175)$$

7. Select mechanical features of the vertical reboiler
   a. Tubes are preferably 1-in. minimum O.D., 1 $^1/_4$-in., 1 $^1/_2$–2-in. maximum.
   b. Vertical, with tube length preferable 6 ft to a maximum of 12 ft.
8. Determine average values
   a. Two-phase density, ρ_{tp}, at $x_E/3$.
   b. Pressure drop factor, ϕ, at $2x_E/3$; ϕ is from Figure 10-113.
   Obtain X_{tt} to use with Figure 10-113.
   Due to slippage effects of the gas phase past the liquid phase, the R_L is not a simple function of the weight fraction of vapor. Using the parameter:[45]

X_{tt} = Correlation parameter, dimensionless (tt refers to the turbulent-turbulent flow mechanism)

$$X_{tt} = \left(\frac{W_L}{W_g}\right)^{0.9}\left(\frac{\rho_g}{\rho_L}\right)^{0.5}\left(\frac{\mu_L}{\mu_g}\right)^{0.1} \quad (10\text{-}176)$$

$$= \left(\frac{1-x}{x}\right)^{0.9}\left(\frac{\rho_g}{\rho_L}\right)^{0.5}\left(\frac{\mu_L}{\mu}\right)^{0.1} \quad (10\text{-}177)$$

Often, approximately:

$$X_{tt} \cong \frac{W_L}{W_g}\left(\frac{\rho_g}{\rho_L}\right)^{0.5} \quad (10\text{-}178)$$

where
X_{tt} = correlation for turbulent-turbulent flow, dimensionless
W = mass flow rate, lb/sec

   c. Determine ρ_{tp} and ϕ at exit conditions.

tp = two phase
$$\rho_{tp} = \rho_g R_g + \rho_l R_l \quad (10\text{-}179)$$

and; $R_l = (1 - R_g)$, volumetric fraction of liquid at any point along the vertical tube. Read R_l using Figure 10-114 and 10-115.

R_l = volume fraction of liquid phase, dimensionless
R_g = volume fraction of vapor phase, dimensionless

Obtain ϕ^2 from Figure 10-116, for Equation 10-181 for both the average conditions and the exit conditions

9. $$\psi = \frac{(\rho_g)^{0.5}}{(\rho_l)}\frac{(\mu_l)^{0.1}}{(\mu_g)} = \frac{X_{tt}}{(W_l/W_g)^{0.9}} \quad (10\text{-}180)$$

Table 10-30
Nucleate Pool Boiling Data

		Equation Terms*				Boiling Coefficient at Designated Δt						
Material	Press. (psia)	m	n	Δt Range	Max. Δt (°F)	5°F	10°F	20°F	30°F	40°F	50°F	Heating Device
Propane	20–35	46	2.5	7–15	...	(515)	1,460	...	...	...	...	HT
Propane	170	87	2.0	15–25	50	...	(870)	1,740	(2610)	...	...	VT
Propane	245	110	2.0	10–30	40	...	1,100	2,200	3,300	...	...	VT
Propane	295	145	2.0	10–25	30	...	1,450	2,900	4,350	...	...	VT
Propane	375	205	2.0	10–25	25	...	2,050	4,100	6,150	...	...	VT
Propane	475	320	2.0	7–15	15	(1600)	3,200	...	...	...	...	VT
n-Butane	20–35	12	2.64	7–15	...	(170)	525	...	...	...	...	HT
n-Pentane	22	$4.5(10^{-4})$	4.70	30–60	60	...	...	...	130	385	850	VT
n-Pentane	59	$2.0(10^{-2})$	4.16	20–45	45	...	...	265	850	2,350	...	VT
n-Pentane	115	0.76	3.27	15–35	35	...	...	695	1,700	...	...	VT
n-Pentane	215	23.5	2.91	8–20	25	...	1,900	7,250	...	...	...	VT
n-Heptane	6.6	$3.4(10^{-3})$	3.85	50–80	80	...	...	...	...	...	240	VT
n-Heptane	14.7	0.60	2.90	30–60	60	...	...	...	380	660	1,010	VT
n-Heptane	50	2.25	2.90	25–40	40	...	...	...	1,450	2,480	...	VT
n-Heptane	115	9.0	2.75	20–35	35	...	...	1,710	3,400	...	...	VT
n-Heptane	215	107	2.20	15–25	25	...	(1,710)	3,900	...	...	...	VT
Kerosine	14.7	1.79	3.19	10–15	>15	(60)	280	...	...	...	...	HT
Benzene	14.7	$4.9(10^{-3})$	3.87	45–90	90	...	...	...	...	(195)	370	VT
Benzene	50	$3.4(10^{-3})$	3.87	25–50	50	...	...	...	580	1,350	2,580	VT
Benzene	115	0.77	3.27	15–40	40	...	...	690	1650	3,400	...	VT
Benzene	265	42	2.61	7–25	25	...	1,720	5,200	...	...	...	VT
Benzene	465	$1.0(10^{3})$	1.96	3–11	11	4,800	9,400	...	...	...	...	VT
Styrene	2.7	11.5	2.05	20–90	...	...	...	270	410	550	700	HP
Styrene	14.7	29.0	2.05	20–50	...	...	...	670	1,030	1,390	1,750	HP
Methanol	14.7	1.61	3.25	10–15	>15	(120)	290	...	...	...	...	HT
Ethanol	14.7	$2.4(10^{-2})$	3.73	40–60	60	...	...	...	...	570	1,020	VT
Ethanol	55	1.74	3.08	20–40	40	...	...	870	2,090	3,740	...	VT
Ethanol	114	21.5	2.67	10–30	35	...	1,010	3,160	6,350	...	...	VT
Isopropanol	14.7	6.0	2.40	10–60	60	...	150	400	690	1,050	1,440	VT
u-Butanol	14.7	0.40	3.21	15–30	>30	...	...	230	740	...	...	HT
Carbon tetrachloride	14.7	0.18	2.90	20–40	...	...	...	55	115	200	...	VT
Carbon tetrachloride	14.7	0.73	3.14	15–25	>25	...	(100)	440	...	...	...	HT
Acetone	14.7	$7.3(10^{-2})$	3.85	20–40	40	...	...	370	1,170	2,780	...	HP
Methyl ethyl ketone	14.7	70	1.84	20–50	50	...	...	850	1,200	1,550	1,850	W
Water	14.7	14.3	3.14	5–10	>10	450	2,000	...	...	...	...	HT
Water	14.7	52	2.35	10–35	35	...	1,200	2,780	5,100	...	...	VT
Water	14.7	$1.7(10^{-2})$	4.90	15–35	35	...	...	2,000	9,600	...	...	W
Water	383	605	2.82	5–30	30	1,130	4,000	14,200	30,000	...	...	W
Oxygen	14.7	4.8	2.47	6–10	>10	...	140	...	...	...	...	VT
Nitrogen	14.7	1.9	2.67	6–12	>12	...	90	...	...		...	VT
Freon-12	60	0.49	3.82	12–20	>20	...	(320)	2,250	...	...	...	HT

* Equation: $q/A = (m\ \Delta t)^{a}$ where $\Delta t = tw - tb$, °F
Notation: VT = vertical tube
HT = horizontal tube
HP = horizontal plate
W = wire
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ψ = parameter for two-phase physical properties, for use with Figures 10-114, 10-115, 10-116, and 10-117. Although ψ is not constant, in general an average constant value may be assumed.[45]

10. Calculate the circulation rate, W_T, by (see Table 10-29):

$$W_T^2 = \frac{g\, a_i^2 \rho_L[(\rho_L - \bar{\rho}_{tp})\Delta Z - \rho_L \Delta W_s]}{2f_{Li}\dfrac{\Delta L_i}{D_i} + 2f_{Lt}\left(\dfrac{a_i}{a_t}\right)^2 \cdots}$$

$$\frac{}{\cdots\left[\dfrac{\Delta L_{Bc}}{D_t} + \bar{\phi}^2(1-\bar{x})^2\dfrac{\Delta L_{cD}}{D_t}\right]\cdots}$$

$$\frac{}{\cdots + 2f_{LE}(1-x_E)^2\left(\dfrac{a_i}{a_E}\right)^2\phi_E^2\dfrac{\Delta L_{DA}}{D_E} + \gamma\left(\dfrac{a_i}{a_E}\right)^2} \tag{10-181}$$

where

ρ_{tp} = effective average two-phase density, lb/ft^3
$\bar{\rho}_{tp}$ = average density, lb/ft^3, two phase
ϕ^2 = effective average (two-phase)/liquid phase pressure drop ratio corresponding to effective average vaporization x
a_i/a_t = cross-sectional area ratio, inlet line/total tubes
a_i/a_E = cross-sectional areas ratio, inlet line/exit line
a_i = cross-sectional area, inlet feed pipe, ft^2
ΔZ = height of driving leg for thermal circulation, ft
W_T = mass rate, lb$_m$/sec, total flow, or W
W_s = shaft work done by system, ft liquid
Δ = change from one condition to another
ΔL_i = change in equivalent length of pipe, ft, inlet piping system
D_i = inlet diameter, feed pipe ft
f = fanning friction factor dimensionless
$\bar{\phi}$ = average value of ϕ
L = equivalent length of pipe, ft
γ = acceleration loss group, dimensionless

$$= \left[\frac{(1-x)^2}{R_L} + \frac{\rho_L}{\rho_g}\left(\frac{x^2}{R_g}\right) - 1\right]$$

(Evaluate at outlet values of x, R_L, and R_g)

$\bar{x}$ = average value of weight fraction of vapor or gas, dimensionless
E = subscript, exit reboiler vapor

This equation can be used to shorten the design by carefully selecting the overall average values. If specific data is not available, Fair[45] recommends using the guidelines in Table 10-28.

11. Calculate the boil-up and check against the required process balance value.
12. Calculate boil-up and check against the required balance for process balance.
13. Repeat calculations, adjusting flow as necessarv, until the assumed X_E (weight fraction of vapor in reboiler exit) produces the proper boil-up rate.

B. Heat Transfer: Simplified Method

Fair recommends this method rather than his stepwise method given in the article, because it avoids increments of calculations but is sufficiently reliable for most design cases. This procedure is duplicated by permission.[45]

See Table 10-29.

1. The previously established circulation rate is used, but the exchanger dimensions must be checked for heat transfer. Obtain additional overall average values:
   a. Heat transfer coefficient for liquid, h_L (equation following).
   b. Heat transfer coefficient for two-phase mixture, $\bar{h}_{tp}$, at $x = 0.4x_E$ (Figure 10-117).
   c. Boiling coefficient correction factors:

α' at $x = 0.4x_E$ and α_E at $x = x_E$

   Figure 10-118; $\bar{\alpha}$ from Equation 10-184.
   d. Nucleate boiling coefficient, h_b, from Table 10-30 or other source. (An estimate of the film temperature drop is required.)

2. Calculate the process side heat transfer coefficient, h_p, from Equations 10-185 and 10-186).
3. Calculate the total heat transferred to the process fluid check against the required value. The adjustments required may result in a new exchanger configuration and a new calculation of circulation rate.

Design Comments

These comments are directed to the inexperienced designer, and in general, amplify material previously presented. The comments are somewhat random in nature but are nonetheless important considerations in optimum design.

- The thermosiphon reboiler has inherent instabilities. A valve or other flow restriction in the inlet line helps overcome these instabilities. Adjustment possibilities of a valve also compensate for variations in reboiler duty as imposed by changes in operation of the fractionator.
- For once-through natural circulation reboilers, the liquid backup height is calculated from the pressure balance equation. If this height, plus an allowance for froth, reaches the bottom tray level, flooding of the tower will occur.
- Economic optimum design usually implies high circulation rates, although not high enough to give “mist” flow.
- The large fraction of tube length used for sensible heating in vacuum reboilers leaves little density difference for thermal circulation. This fact, plus the frequent

need for circulating viscous materials, points toward forced-circulation reboilers for vacuum fractionators.

- For steam distillation columns, it is desirable to sparge the steam uniformly to all reboiler tubes. Because this provides full tube length for two-phase flow, thermal circulation is permitted.
- In reboiler design, film boiling should be avoided. However, such rules-of-thumb as 10–12,000 Btu/(hr)(ft^2) maximum heat flux are frequently quite conservative.

For heating, Fair[45] suggests the Dittus-Boelter equation:

$$h_L = 0.023 \frac{k_l}{D_i} \frac{(3{,}600 D_t G_t)^{0.8}}{(\mu_l)} \frac{(c_l \mu_l)^{0.4}}{(k_l)} \tag{10-182}$$

See symbols previously listed.

The two-phase flow heat transfer coefficient is determined from:[45]

$$h_{tp} / h_L = 3.5\ (1/X_{tt})^{0.5} \tag{10-183}$$

For the short-cut calculations:

$$\bar{\alpha} = (\alpha_E + \alpha')/2, \text{ at average condition} \tag{10-184}$$

α_E is evaluated at exit conditions
α' is evaluated at 40% exit vaporization

Boiling coefficients: Determine from the McNelly, Gilmour, Kern, or Yilmaz equations previously given, or Fair's suggestion of the Bliss or Levy equations, which were not given due to constants not being available.

Process the side boiling heat transfer coefficient:

$$h_p = \frac{\Delta L_{BC} h_L + \Delta L_{CD} \bar{h}_v}{L_t}, \text{ see Figure 10-110 for lengths} \tag{10-185}$$

$$\bar{h}_v = \bar{\alpha}\, h_b + \bar{h}_{tp} \tag{10-186}$$

Evaluate h_b at average inside ΔT and h_{tp} at 40% of the exit vaporization.

The 40% value is based upon integration of the h_{tp}/h_L equation previously given.

Subscripts:

tp = two phase
v = vaporization
b = boiling
p = process side (boiling) coefficient
t = tube, tube side
α = average correction
¯ = bar over symbol = average value

Fair emphasizes some helpful points:

- Installation of a valve in the liquid circulation line as shown on the illustration can aid in overcoming instability and variations in reboiler duty.
- In the physical arrangement, make certain that the pressure balance level, plus an allowance for froth, establishes a height that is below the bottom tray of the column to avoid flooding the column. In addition, the estimated froth height on top of the liquid should still be below the level of the vapor return from the reboiler.
- "Mist" type flow usually does not occur in an economic design, even though the recirculation rates may be high.
- In vacuum service, the large fraction of the tube length used for sensible heating leaves little density difference for thermal circulation. This fact, plus the frequent need for circulating viscous materials, points towards forced-circulation reboilers for vacuum service.
- For steam distillation columns, it is desirable to sparge the steam uniformly into all reboiler tubes. This then provides full length for two-phase flow, and thermal circulation is permitted.
- Film boiling should be avoided; however, nucleate boiling often can be found at heat flux values greater than the rule-of-thumb values of 10–12,000 Btu/hr-ft^2. These are often conservative values. See Figure 10-119 from Fair.[46]

Examples from Fair[46] (by permission) include the following:

Example 10-19. C_3 Splitter Reboiler

A thermosiphon reboiler is to be designed for a fractionator that separates propane as the bottoms product. The conditions below the bottom tray are 401 psia and 164°F. A total of 17,600 lb/hr vapor is to be produced.

Physical data for propane:

ρ_L = 24.80 lb/ft^3
μ_g = 4.40 lb/ft ft^3
μ_L = 0.157 lb/(ft) (hr) or 0.065 cp
μ_g = 0.036 lb/(ft.) (hr) or 0.015 cp
$(\Delta t/\Delta p)_s$ = 0.24°F/psi
c_L = 0.85 Btu/(lb) (°F)
k_L = 0.068 Btu/(hr) (ft)(°F)
λ = 96.9 Btu/lb
σ = 1.24 (10^{-4}) lb/ft
ψ = 0.49

Preliminary Design

Boiling is to be inside $^3/_4$-in. 16 BWG steel tubes, 8 ft long. Condensing steam at 25 psig is available for heating. An overall coefficient U = 300 Btu/(hr) (ft^2)(°F) is expected. For the given duty and a total driving force of 20°F, the inside surface required is 285 ft^2 This is equivalent to 96 tubes.

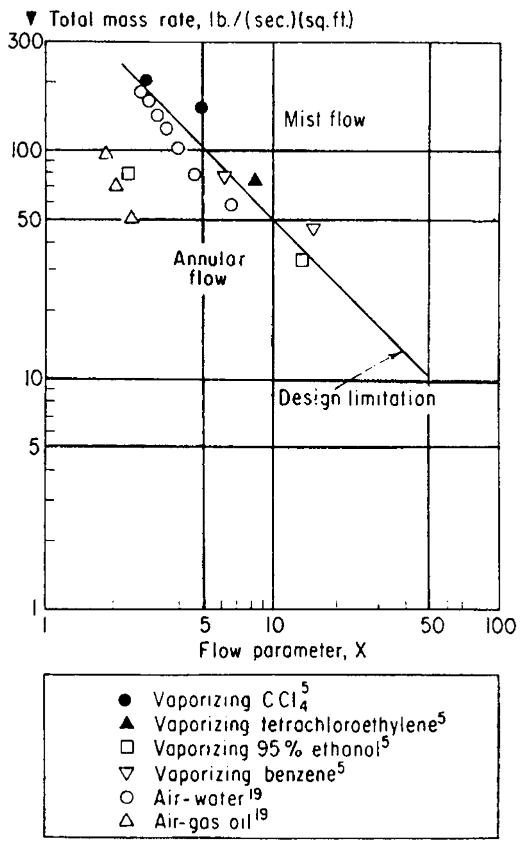


Figure 10-119. Dry-wall vapor-binding limitation. (Used by permission: Fair, J. R. *Chemical Engineering*, July 8, 1963. ©McGraw-Hill, Inc. All rights reserved.)

Circulation Rate

The inlet line consists of 50 equivalent ft of 4-in. standard pipe. The exit line consists of 50 equivalent ft of 6-in. standard pipe. No special flow restriction is in the inlet line. Preliminary calculations indicate that essentially the entire surface is in the vaporization zone.

To facilitate trial-and-error work, the following constant terms are calculated:

$$\frac{\Delta L_i}{D_i} = 149.3$$

$$\frac{\Delta L_{CD}}{D_t}\left(\frac{a_i}{a_t}\right)^2 = 5.79$$

$$\frac{\Delta L_E}{D_E}\left(\frac{a_i}{A_E}\right)^2 = 19.31$$

$$g\, a_i^2\, \rho_L\, \Delta Z = 49.5$$

$$h_L = 21.1\, W_T^{0.8}$$

For the first trial, assume 39% vaporization per pass. From Figures 10-114 and 10-116 and for $\psi = 0.49$,

x	R_L	ϕ^2	ρ_{tp}	γ
0.13	0.34	—	11.35	—
0.26	—	15	—	—
0.39	0.19	25	8.28	2.02

The circulation rate is calculated as shown in Equation 10-181:

$$W_T^2 = \{49.5\,(24.80 - 11.35)\}/$$

$$0.008)(149.4) + 0.012\,(0.55)(5.79)(15)$$

$$+ 0.008\,(0.37)(19.31)(65) + 0.442\,(2.02)\} = 162 \qquad (10\text{-}181)$$

$$W_T = 12.7 \text{ lbs./sec.}$$

Now, $(W_T)(x_E) = 12.7\ (0.39) = 4.95$ lb/sec vaporized, which is very close to the required vaporization rate of 17,600 lb/hr (4.89 lb/sec). No further adjustment is required at this point.

Heat Transfer Rate—Stepwise Method

Based on the calculated circulation rate of 12.7 lb/sec, stepwise calculations are carried out for increments of 5% vaporization ($\Delta x = 0.05$). A heat balance is taken over each increment based on the combined film coefficient for that increment.

The calculations are summarized in Table 10-31 and presented graphically in Figure 10-120. Several observations may be made:

1. The flow patterns are "bubble" and "slug."
2. The required vaporization is attained in the 8 ft tube length.
3. The pressure balance is off by about 21 lb_f/ft^2. In design, this can be corrected by a slight adjustment in liquid level or by adding pressure drop to the inlet line (e.g., a valve). It is assumed here that no further trial calculations are needed.

The results of the calculations are

1. Total duty = 17,600 (96.9) = 1,705,000 Btu/hr.
2. Average inside coefficient, based on design $t_w - t_b$ = 5.0°F = 1,200 Btu/(hr) (ft²) (°F).
3. Based on a steam film coefficient of 1,500, the *clean* overall coefficient is 658 Btu/(hr) (ft²) (°F) on an inside basis.
4. Including an inside fouling factor of 0.0010 (hr) (ft²) (°F)/Btu and an outside fouling factor of 0.0005

(hr) (ft²) (°F)/Btu, the service overall coefficient is 331 Btu/(hr)(ft²)(°F).

5. The overall temperature difference is 10–18°F, depending on fouling. Vacuum operation on the steam side is indicated.

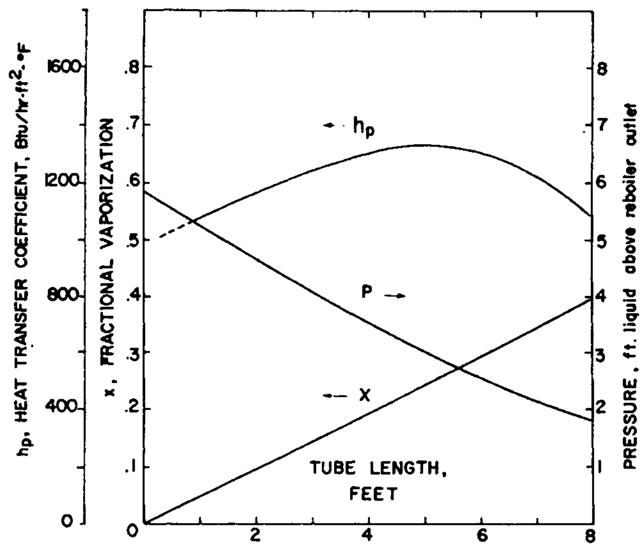


Figure 10-120. Summary of information for Example 10-19. (Used by permission: Fair, J. R. *Chemical Engineering,* July 8, 1963, Part 1. ©McGraw-Hill, Inc. All rights reserved.)

Heat Transfer Rate—Simplified Method

For W_T = 12.7 lb/sec circulation, the following values are obtained:

h_L = 162 Btu/(hr) (ft²) (°F)
h_{tp}/h_L = 2.4 (at x = 0.4, x_E = 0.16)
h_{tp} = 386 Btu/(hr) (ft²) (°F)
α' = 1.0, α_E = 0.5 } Figure 10-118, GT = 27.8 lb/(sec) (ft²)
α = 0.75
h_b = 900 Btu/(hr) (ft²) (°F) at $t_w - t_s$ = 5.0°F and data of Fair's reference 9.

$$hp = \frac{\Delta L_{BC} h_l + \Delta L_{cd} h_v}{L_t}$$

See Figure 10-110 for length designations.

$$h_p = h_v = 0.75(900) + 386 = 1061 \text{ Btu/(hr)(ft}^2\text{)(°F)}$$

For the inside surface of 285 ft², the vaporization is

$$\frac{h_pA(t_w - t_s)}{\lambda} = \frac{(1,061)(285)(5.0)}{(96.9)(3,600)} = 4.35 \text{ lb/sec}$$

Table 10-31
Stepwise Calculations for C_3 Splitter Reboiler, Example 10-19

Increment	1	2	3	4	5	6	7	8	Exit
Incr. vaporization, Δx	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.035	
Σ vaporization, x	0.05	0.10	0.15	0.20	0.25	0.30	0.35	0.385	0.385
$1\sqrt{X_{tt}}$	0.08	0.20	0.35	0.50	0.66	0.83	1.01	1.25	1.32
R_L	0.58	0.42	0.35	0.31	0.28	0.25	0.23	0.20	0.19
ρ_{tp}	16.25	12.98	11.54	10.74	10.08	9.40	9.00	8.60	8.28
ϕ^2	2.70	4.80	7.00	9.60	12.2	15.5	19.0	22.5	25.0
h_{tp}/h_L	1.23	1.65	2.07	2.50	2.90	3.28	3.60	3.95	...
h_{tp}	198	266	333	403	467	529	580	636	...
h_b	900	900	900	900	900	900	900	900	...
$\bar{\alpha}$	1.0	1.0	1.0	1.0	1.0	0.80	.070	0.50	...
h_p	1098	1166	1233	1303	1367	1249	1210	1085	...
Incr. length, ft	1.15	1.08	1.02	0.97	0.92	1.01	1.04	0.81	...
Σ length, ft	1.15	2.23	3.25	4.22	5.14	6.15	7.19	8.00	...
Σ Δp, psf	19.96	35.37	48.56	60.46	71.30	82.72	94.09	102.97	146.57

Pressure Balance

Outlet	psf	Inlet	psf
Static	89.99	Static (8.0 ft)	198.40
Friction, tubes	10.88	Friction, piping	−30.80
Friction, exit	36.30	(Required extra)	−21.03
Acceleration	9.40		146.57
	146.57		

Used by permission: Fair, J. R. *Petroleum Refiner,* V. 39, No. 2, p. 105, ©1960. Gulf Publishing Company. All rights reserved.

The value of the process side coefficieint, h_p = 1,061 Btu/hr (ft^2) (°F) is lower than the value calculated in a stepwise fashion. A slightly higher value of Δt would correct this.

Example 10-20. Cyclohexane Column Reboiler[45]

A thermosiphon reboiler is to be designed for a fractionator that separates cyclohexane as the bottoms product. The conditions below the bottom tray are 16.5 psia and 182°F. A total of 13,700 lb/hr vapor is to be produced.

Physical data for cyclohexane:

ρ_L = 45.0 lb/ft^3
ρ_g = 0.200 lb/ft^3
μ_L = 0.0208 lb/(ft) (hr) or 0.0086 cp
μ_g = 0.97 lb/(ft) (hr) or 0.40 cp
$(\Delta t/\Delta p)_s$ = 3.6 °F/psi
c_L = 0.45 Btu/(lb) (°F)
k_L = 0.086 Btu/(hr) (ft)(°F)
λ = 154 Btu/lb
σ = 1.24 (10^{-3}) lb/ft
ψ = 0.098

Preliminary Design

Boiling is to be inside 1-in. 12 BWG steel tubes, 8 ft long. Condensing steam at 50 psig is available for heating. An overall coefficient U = 300 Btu/(hr) (ft^2)(°F) is expected. For the given duty and a total driving force of 45°F, the inside surface required is 157 ft^2. This is equivalent to 96 tubes.

Circulation Rate

The inlet line consists of 100 equivalent ft of 6-in. standard pipe. The exit line consists of 50 equivalent ft of 10-in. standard pipe. No special flow restriction is in the inlet line.

For the first trial, assume 15% vaporization per pass. This gives a circulation rate of 91,300 lb/hr or 25.4 lb/sec. For this rate, the following apply:

h_L = 161 Btu/(hr) (ft^2) (°F)
$f_{L,i}$ = 0.0045
$f_{L,t}$ = 0.0065
$f_{L,E}$ = 0.0045

It will be assumed that the liquid level is maintained even with the top tubesheet. Neglecting inlet line friction, the sensible heating zone length may be estimated from Equation 10-176:

$$\Delta t/\Delta L = \frac{\pi D_t N_t h_L(t_w - t_L)}{3600\, W_T c_L} \qquad (10\text{-}187)$$

$$\frac{\Delta t}{\Delta L} = \frac{3.14(0.065)(218)(161)(30)}{3600(25.4)(0.45)} = 5.2°\text{F/ft.}$$

(assuming $t_w - t_L$ = 20°F)

From Equation 10-187A:

$$-\Delta p/\Delta L = \frac{\rho_L g_c}{144\, g} + \frac{\Delta F_{B-C}}{\Delta L} \qquad (10\text{-}187A)$$

$$\frac{-\Delta p}{\Delta L} = \frac{45.0(1.0)}{144} = 0.312 \text{ psi/ft}$$

(neglecting friction sensible zone)

h_l is from Equation 10-187.
Then, by Equation 10-170

$$\frac{p_B - p}{p_B - p_A} = \frac{3.6}{3.6 + 5.2} = 0.41$$

Hence, the length of the sensible zone = 0.41 (8.0) = 3.3 ft.

From Figures 10-114 and 10-116 and for ψ = 0.098, the following values are obtained:

x	R_L	ϕ^2	ρ_{tp}	γ
0.05	0.28	—	12.74	—
1.10	—	25	—	—
0.15	0.16	40	7.37	9.53

The circulation rate is calculated:

$$W_T^2 = (32.2)(0.181)^2(45.0)(45.0 - 12.74)(4.7)/\Big\{0.009\left(\frac{100}{0.336}\right) + 0.013\left(\frac{0.181}{0.320}\right)^2\left[\frac{3.3}{0.065} + 25\left(\frac{4.7}{0.065}\right)(0.90)^2\right] + 0.009\left(\frac{0.181}{0.548}\right)^2(40)\left(\frac{50}{0.85}\right)(.85)^2 + 9.53\left(\frac{0.181}{0.548}\right)^2\Big\}$$

$$= \frac{7{,}200}{2.68 + 6.35 + 1.68 + 1.04} = 615$$

$$W_T = 24.8 \text{ lb/sec}$$

This value is in reasonable agreement with the assumed value of 25.4 lb/sec.

Heat Transfer Rate—Simplified Method

For this calculation, the following are obtained:

h_L = 160 Btu/(hr) (ft^2)(°F)
$\bar{h}_{tp}/h_L$ = 3.2 (at x = 0.4 x_E = 0.060)
$\bar{h}_{tp}$ = 510 Btu/(hr) (ft^2)(°F)
α' = 0.2 } Figure 10-118
α_E = 0 } G_T = 66.0 lb/sec ft^2
$\bar{\alpha}$ = 0.1
h_b = 200 Btu/(hr) (ft^2)(°F) at $t_w - t_s$ = 30°F
h_v = 0.10 (200) + 510 = 530 Btu/(hr) (ft^2)(°F), from Equation 10-186

$$h_p = \frac{(3.3)(160) + (4.7)(530)}{8.0} = 377 \text{ Btu/(hr) (ft}^2\text{)(°F)},$$

from Equation 10-185

For clean conditions, the combined tube wall and steam-side resistance is calculated as 0.00090, corrected to inside dimensions. Thus, the calculated U is

$$U = \frac{1}{\frac{1}{377} + 0.00090} = 282 \text{ Btu/(hr) (ft}^2\text{)(°F)}$$

This is close to the assumed value of U. It does not, however, consider fouling in service. Additional trial designs may be used, or the temperature difference may be adjusted upward.

Kern's Method Stepwise[70]

In vacuum applications, use this method with caution and compare it with other methods.

1. Determine the heat requirements or duty for any sensible heat as well as the latent heat.
2. Assume a unit size, number and size of tubes, and area.
3. Evaluate sensible heat transfer inside tubes as previously outlined for in-tube transfer. Determine the area required.
4. Evaluate LMTD for isothermal boiling.
5. Trial 1: Estimate area, A, for maximum flux condition, limiting Q/A to 12,000 Btu/hr-ft^2 surface for organic materials. Experience has shown that a value of 6,000-8,000 is a good starting value for Q/A for organics.

$$A = \frac{Q}{Q/A}$$

6. Re-estimate the unit size assumed in Step 2, making the area the value of Step 5.
7. Evaluate an operating overall coefficient:

$$U_D = \frac{Q}{A(\Delta t)} \qquad (10\text{-}188)$$

Δt = LMTD

8. Assume a recirculation ratio of 4:1.
9. Determine the material balance around unit

Total weight of recirculated liquid = (4) (desired vapor rate, V)
Vapor = Desired vapor rate, V
Liquid = 4 (desired vapor rate, V)
Total = 5 (desired vapor rate, V) = W

10. Pressure balance across reboiler

Static pressure of boiler leg

$$\frac{L\rho_{(avg.)}}{144} = \frac{2.3L}{144(v_o - v_i)} \log \frac{v_o}{v_i}, \text{ psi} \qquad (10\text{-}189)$$

where L = length of reboiler tube, ft
v_o = specific volume of fluid at outlet of reboiler, ft^3/lb
v_i = specific volume of fluid at inlet to reboiler, ft^3/lb
log = log base 10

Friction resistance to flow inside tubes, flow rate into tubes (per tube):

$$\frac{W}{a} = \frac{W(144)(n)}{N_t a_i}, \text{ lb/hr-ft}^2 \text{ cross-section} \qquad (10\text{-}190)$$

n = 1 tube pass

where W = total flow rate, lb/hr into tubes
N_t = total number of tubes
a_i = cross-sectional flow area per tube, in^2
D_i = tube I.D., ft
Re = DG/μ, per tube
G = flow into tubes, lb/hr (ft^2 cross-section) with μ evaluated at the boiling temperature for the liquid.

Read friction factor, f, from Figure 10-137.
Calculate mean specific gravity in tube as average of inlet liquid and outlet vapor-liquid mixture.

$$\text{pressure drop} = \Delta p_t = \frac{fG^2Ln}{5.22 \times 10^{10} D_i s \phi_t} \qquad (10\text{-}191)$$

Let ϕ_t = 1.0

11. Total resistance to flow:

= (static pressure of reboiler leg) + (pressure drop through tubes) + (frictional resistance of inlet piping) + (frictional resistance of outlet piping) + (expansion loss) (10-192)

Note that for preliminary calculations, the frictional resistances in piping can be neglected but should be included in final calculations, particularly at high recirculation ratios.

12. Driving force

$$\frac{x_2 \rho_L}{144}, \text{ psi} \qquad (10\text{-}193)$$

where x_2 = height of liquid level in column above reboiler bottom tubesheet, ft
ρ_L = density of liquid, lb/ft^3

13. If the driving force, Step 12, does not equal or *slightly* exceed the total of resistances in Step 11, the unit should be rebalanced; that is, shorter tubes used to give less pressure drop, lower recirculation ratio used to give less pressure drop, or a larger number of total tubes used to give less pressure drop. The liquid can be raised

above the level of the tubesheet, but this is not recommended for differential levels greater than 6 in.

14. After a pressure drop balance has been obtained to ± 0.1–0.2 psi, compute the heat transfer coefficient as follows.

    Shell side: Usual procedure for condensing steam or for other heating, medium.

    Tube side: Determine heat transfer coefficient from Figure 10-46 (using tube-side curve) at Reynold's number calculated for pressure drop evaluation. If the h_i calculated exceeds 300 for organics (Figure 10-103), use a value of 300 and correct to outside coefficient, h_{io}.

15. Calculate the overall heat transfer coefficient by

$$U_C = \frac{h_o h_{io}}{h_{io} + h_o} \qquad (10\text{-}194)$$

U_D = calculated from assumed unit (corrected for final pressure balance) of Step 7.

Resistance of fouling and metal tube wall required for balanced operation of reboiler:

$$r = \frac{U_c - U_D}{U_c U_D} \qquad (10\text{-}195)$$

If the resistance seems too low for the service, then the unit must be redesigned to obtain the higher dirty coefficient, U_D.

Other Design Methods

Several other excellent presentations exist on the subject of reboiler design. Presenting all of the variations here is just not possible; designers should refer to Hughmark,[65, 66, 67] Palen and Taborek,[91] Palen and Small,[90] Frank and Prickett,[47] and Hajek.[60]

The article of Fair and Klip[193] presents a detailed analysis of the necessary design features and equations for horizontal kettle reboilers, horizontal thermosiphon reboilers, and vertical thermosiphon reboilers. Other useful references on reboilers are 185, 186, 188, 190, 192, 194, 195, 196, 197, and 201.

Example 10-21. Vertical Thermosiphon Reboiler, Kern's Method[70]

The design of a distillation column requires a reboiler operating at 2.23 psia (vapor space above bottom liquid). The heat duty is 1,528,600 Btu/hr. The properties of the acrylonitrile mixture have been calculated to be

Latent heat of vaporization = 285 Btu/lb
Average molecular weight = 63.4
Density of vapor = 0.0230 lb/ft^3
Density of liquid = 54.5 lb/ft^3
Thermal conductivity of liquid = 0.084 Btu/hr-ft.-°F
Average c_p = 0.425 Btu/lb-°F
Average viscosity, μ = 1.3389 c_p × (2.42)
= 3.24 lb/ft-hr

Design: Select a thermosiphon reboiler as the preferred operation, if design is acceptable.

Assume: flux, Q/A = 7,200 Btu/hr (ft^2)
U = 120 Btu/hr (ft^2)(°F)
Max. Δt = 60°F
(Steam temp. will be 113°F + 60 = 173°F. This is vacuum steam, 6.417 psia.)

Approximate surface area:

$$A = \frac{1{,}528{,}600}{7{,}200} = 212 \text{ ft}^2$$

Select: Fixed tube sheet vertical unit

1 1/4-in. tubes × 12 BWG steel for low pressure drop, easier cleaning on 1 1/2-in. triangular pitch, 4 ft long
Assume tube sheets, each 1 1/2 in. thick, then usable tube length = 3.75 ft.

$$\text{No. tubes required} = \frac{212}{3.75(0.3272 \text{ ft}^2/\text{ft})} = 173 \text{ tubes}$$

Shell size estimate:

23-in. I.D. shell contains 187 tubes; remove at least 3 for internal impingement baffle.
Available area = (187 − 3) (3.75) (0.3272) = 226 ft^2
Recirculation ratio: assume 20:1
Reboiler vapors required = 1,528,600/285 = 5,370 lb/hr
Liquid being recirculated = 20 (5,370) = 107,400 lb/hr
Total liquid flow at reboiler inlet = 107,400 + 5370
= 112,770 lb/hr
Specific volume of liquid into reboiler = 1/54.5 = 0.01835 ft^3/lb
Specific volume of vapor only at outlet = 1/0.023 = 43.5 ft^3/lb
Total volume of mixture out reboiler
107.400 (0.01835) = 1,980 ft^3/hr
5370 (43.5) = 233,500 ft^3/hr
Total volume = 235,480 ft^3/hr
Specific volume of mixture out reboiler = 235,480/112,770
= 2.09 ft^3/lb

Pressure balance across reboiler: Assume fluid in distillation column at reboiler tubesheet level.

Static pressure of reboiler leg:

$$= \frac{2.3L}{144(v_o - v_i)} \log_{10}\frac{v_o}{v_i}$$

$$= \frac{2.3(4)}{144(2.09 - 0.01835)} \log_{10}\frac{2.09}{0.01835}$$

$= 0.0308 \log 114 = 0.0308(2.057)$
$= 0.0633$ psi

Friction resistance to flow inside tubes:
flow rate into tubes

$$= \frac{112{,}700(144)(1)}{(187 - 3)(0.836 \text{ in}^2/\text{tube})}$$

$= 104{,}700$ lb/hr (ft^2 cross-section)

$$R_e = \frac{(1.03 \text{ in.})(104{,}700)}{(12)(3.24)} = 2{,}700 \text{ (from Figure 10-137)}$$

$f = 0.00038 \text{ ft}^2/\text{in.}^2$

$$\text{Mean Sp. Gr. for tube} = \frac{\left(\frac{54.5}{62.3}\right) + \frac{1}{2.09(62.3)}}{2} = 0.441$$

Tube pressure drop

$$= \Delta p_t = \frac{0.00038(112{,}770)^2(4)(1)}{(5.22)(10)^{10}(1.03/12)(0.441)(1)} = 0.0134 \text{ psi}$$

For the first try, neglect the piping friction into and out of reboiler. This should be designed with a minimum of pressure drop.

Total resistance to flow = 0.0633 + 0.0134 = 0.0767 psi

$$\text{Driving force} = \frac{x_1 \rho_L}{144} = \frac{4(54.5)}{144} = 1.51 \text{ psi}$$

This is not a satisfactory check.

Because the resistances are now considerably less than the available driving force, the new estimate for the recirculation ratio must be considerably greater than 20:1.

Final Trial: 24-in. O.D. shell, 6-in. inlet and 10-in.outlet nozzles.

After several trials to obtain a balance, try a recirculation ratio of 73:1 and an exchanger with 178, 12 BWG, 8-ft long tubes.

Liquid being circulated = 73 (5370) = 392,000 lb/hr
Liquid flow at reboiler inlet = 392,000 + 5,370 = 397,370 lb/hr
Total volume of mixture out reboiler:

(392,000) (0.01835) =	7,200
(5,370) (43.5) =	233,500
Total =	240,700 ft^3/hr

Specific volume of mixture = 240,700/397,370 = 0.606 ft^3/lb

Pressure balance:

Static pressure of reboiler leg

$$= \frac{2.3(8)}{144(0.606 - 0.01835)} \log \frac{0.606}{0.01835} = 0.331 \text{ psi}$$

Friction inside tubes:

Flow rate into tubes

$$= \frac{(397{,}370)(144)(1)}{(178)(0.836)} = 384{,}000 \text{ lb/hr (ft}^2)$$

$$R_e = \left[\frac{(1.03)}{(12)}\right] \frac{(384{,}000)}{(3.24)} = 10{,}200$$

$f = 0.00027$

$$\text{Arith. mean gravity} = \frac{\frac{54.5}{62.3} + \frac{1}{0.606(62.3)}}{2} = 0.451$$

However, for many cases, particularly when the difference in specific volume is large, the log mean average is much better.

Log mean average specific volume:

$$v_{lm} = \frac{v_o - v_i}{\ln(v_o/v_i)} \tag{10-196}$$

$$v_{lm} = \frac{\frac{1}{0.606} - 0.1835}{\ln\frac{1/.606}{0.1835}}$$

$= 0.0655$ ft^3/lb

$$\text{Log mean Sp. Gr.} = \frac{\frac{1}{0.0655}}{62.3} = 0.245$$

$$\Delta p_t = \frac{0.00027(384{,}000)^2(8)(1)}{5.22(10)^{10}(1.03/12)(0.245)(1.0)} = 0.289 \text{ psi}$$

Pressure loss in inlet piping (see Figure 10-96D). Assume piping consists of the following for an inlet pipe:

8-ft, 6-in. pipe
two 6-in. weld ells
one 6-in. open gate valve
Note: If a tee connection to spare reboiler is used, add tee to pipe list.

For equivalent feet pipe:

two 6-in. weld ells ≅ 2 (11 eqft see the "Fluid Flow" chapter in Vol. I.)	=	22 ft
one 6-in. open gate valve	≅	3.5 eqft
8-ft, 6-in. pipe	≅	8
Total eq ft	=	33.5 ft

From the "Fluid Flow" chapter in Vol. I.:

$$\Delta p_t = \frac{3.36 \times 10^{-6} f L W^2}{d^5 \rho}$$

A satisfactory average of "f" fitting from the Crane Co. Charts ("Fluid Flow" chapter, Vol. 1, 3rd Ed. this series) is

$$f = 0.0077 + \frac{0.555}{\sqrt[3]{\frac{DG}{\mu}}}$$

$\mu = 3.24$ lb/ft-hr

$D = 6$ in./12 = 0.5 ft

$$G = \frac{397{,}370 \text{ lb/hr}}{0.2006 \text{ ft}^2} = 1{,}928{,}000 \text{ lb/hr (ft}^2)$$

Cross-sectional area of 6-in. pipe = 0.2006 ft^2

$$f = 0.0077 + \frac{0.555}{\sqrt[3]{\frac{0.5(1{,}928{,}000)}{3.24}}}$$

$$f = 0.0077 + 0.00838 = 0.01608$$
$d = 6$ in.
$\rho = 54.5$ lb/ft^3
$W = 397{,}370$

$$\Delta p_t = \frac{3.36 \times 10^{-6}(0.01608)(33.5)(397{,}370)^2}{(6)^5(54.5)} = 67 \text{ psi}$$

Expansion losses in tubes due to vaporization (Kern[70] recommendation):

$$\Delta p_t = G^2/(144\, g\, \rho_{avg.})$$

$G = 384{,}000$ lb/hr (ft^2)

$g = 32.3$ ft/sec-sec $(3{,}600)^2 = 4.17\,(10)^8$ ft/hr-hr

$$\Delta p_t = \frac{(384{,}000)^2}{(144)(4.17 \times 10^8)(28.1)} = 0.0874 \text{ psi}$$

Loss in outlet piping: 10 in.
Assume:
one 10-in.- 90° elbow ≅ 17 ft pipe
3 ft pipe ≅ 3 ft
one 10-in. gate, open ≅ 6 ft
Total = 26 ft pipe
G = 397,370/0.5457 ft^2 = 726,000 lb/hr (ft^2 cross section)

$$f = 0.0077 + \frac{.555}{\sqrt[3]{\left(\frac{10}{12}\right)\left(\frac{726{,}000}{3.24}\right)}}$$

$f = 0.0077 + 0.00972$

$f = 0.0174$

This calculation is not accurate, because it does not account for two-phase flow.

$$\Delta p_t = \frac{3.36 \times 10^{-6}(0.0174)(26)(397{,}370)^2}{(10)^5(1/.606)} = 1.45 \text{ psi}$$

Total resistance to flow:

= 0.331 + 0.289 + 0.67 + 0.087 + 1.45 = 2.8 psi

$$\text{Driving force} = \frac{(8)(54.5)}{144} = 3.03 \text{ psi}$$

With the driving force being slightly greater than the total resistance to flow, the recirculation ratio will be greater than the trial value of 73. For purposes of design, this is a satisfactory basis to proceed.

Calculation of Tube Side Film Coefficient

From calculated Reynold's number = 10,200
Read Figure 10-46, $j_H = 45$

$$h_i = \frac{j_H(k_a)}{D_i}\left(\frac{c\mu}{k_a}\right)\left(\frac{\mu}{\mu_w}\right)^{-0.14}$$

$$h_i = \frac{(45)(0.084)}{\left(\frac{1.03}{12}\right)}\left[\frac{(0.425)(2.42)}{0.084}\right]^{1/3}(1)$$

$h_i = 94.2$ Btu/hr (ft^2) (°F)

$$h_{io} = 94.2\left(\frac{1.03}{1.25}\right) = 77.8$$

Assume: $h_o = 1{,}500$ for steam
Fouling: tube side = 0.001
Shell side = 0.001
Neglect tube wall resistance

$$U = \frac{1}{\frac{1}{77.8} + 0.001 + \frac{1}{1{,}500} + 0.001} = \frac{1}{0.0154}$$
$= 65$ Btu/hr (ft^2)(°F)

Required area for actual operation:

$$A = \frac{1{,}528{,}600}{65(60)} = 392 \text{ ft}^2$$

Actual area in assumed unit:

$$A = (178)(0.3272)(8 - 3/12) = 451 \text{ ft}^2$$

Factor of safety, or percent excess area:

$$\% = \frac{451 - 392}{392}(100) = 15\%$$

This is a satisfactory selection.
Overall coefficient information for operational guide:

$$\text{Clean: } U_c = \frac{1{,}500(77.8)}{77.8 + 1{,}500} = 74.0$$

Dirty based on total available surface:

$$U_D = \frac{1{,}528{,}600}{(451)(60)} = 56.3$$

Allowance in selected unit for fouling:

$$r = \frac{U_c - U_D}{U_c U_D} = \frac{74 - 56.3}{(74)(56.3)} = 0.00425$$

This compares to assumed value of 0.001 + 0.001 = 0.002

The Kern method is usually easier to handle for pressure systems than for vacuum systems. The recirculation ratio is higher and, therefore, requires more trials to "narrow-in" on a reasonable value for the low pressure systems. The omission of two-phase flow in pressure drop analysis may be a serious problem in the low pressure system, because a ratio on the high side may result, causing a high h_i value. In general, however, for systems from atmospheric pressure and above, the method usually gives conservative results when used within Kern's limitations.

Reboiler piping at the liquid inlet and at the vapor outlet can be summarized for convenience in pressure drop calculations into the suggested equivalent feet of Table 10-32. The inlet assumes piping to conveniently pipe the reboiler to a distillation column using welded fittings and a full open gate valve; the vapor outlet assumes short pipe connection, a 90° welded ell and a fully open gate valve. Reference 73 discusses exchanger piping layout.

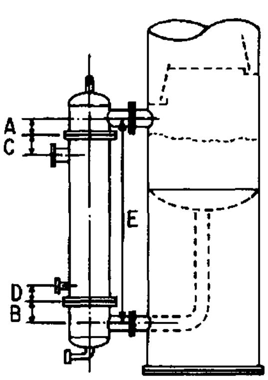
The outlet vapor line should be two nominal pipe sizes larger than the liquid inlet. Table 10-33 is helpful to relate exchanger size and pipe connections, although the same standards cannot apply to every design.

To illustrate the effect of the pressure drop on the reboiler selection, Table 10-34 compares several designs for the same heat load and quantity of reboiled vapor.

Table 10-32
Equivalent Feet of Inlet and Outlet Piping Vertical Thermosiphon Reboiler

	Eq ft	
Pipe Size, In.	**Inlet Liquid**	**Outlet Vapor**
1 1/2	21	10
2	25	12
3	35	17
4	44	21
6	65	32
8	84	40
10	103	49
12	123	58
14	140	65
16	159	74

Table 10-33
Typical Thermosiphon Reboiler Design Standards*



SHELL O.D., Inches	NO. TUBES	TUBE SHEET FACES	APPROX. AREA	NOZZLE SIZES				DIMENSIONS				
				VAPOR, Inches	LIQUID, Inches	STEAM, Inches	COND., Inches	A Inches	B Inches	C Inches	D Inches	E
16	108	4'-11 3/4"	132 □'	6	4	4	1 1/2	7 15/16	5 15/16	8	5 3/4	6'-1 3/4"
20	176	4'-11 3/4"	215 □'	8	6	4	2	8 15/16	7 15/16	8	5 3/4	6'-4 3/4"
24	272	4'-11 3/4"	333 □'	10	6	6	3	9 15/16	7 15/16	9	7 3/4	6'-5 3/4"
30	431	4'-11 3/4"	527 □'	12	6	6	3	11 7/16	7 15/16	9	7 3/4	6'-7 1/4
36	601	4'-11 3/4"	735 □'	16	8	8	4	13 3/16	9 15/16	10 3/4	8	6'-11"
24	272	6'-7 3/4"	448 □'	10	6	6	3	9 15/16	7 15/16	9	6 3/4	8'-1 3/4"
30	431	6'-7 3/4"	710 □'	12	6	6	3	11 7/16	7 15/16	9	6 3/4	8'-3 1/4"
36	601	6'-7 3/4"	990 □'	16	8	8	4	13 3/16	9 15/16	10 3/4	8	8'-7"
42	870	6'-7 3/4"	1,440 □'	16	10	8	4	17 11/16	10 15/16	10 1/16	6 13/16	9'-0 1/2"
30	431	9'-11 3/4"	1,065 □'	12	6	8	3	11 7/16	7 15/16	9	6 3/4	11'-7 1/4"
36	601	9'-11 3/4"	1,520 □'	16	8	8	4	13 3/16	9 15/16	10 3/4	6 3/4	11'-11"
42	870	9'-11 3/4"	2,180 □'	16	10	8	4	17 11/16	10 15/16	10 1/16	6 13/16	12'-4 1/2"

*Cross-section area of vapor nozzle off channel must be a minimum of 1.25 × total flow area of all tubes (author).

Used by permission: Lee, D. C., Dorsey, J. W., Moore, G. Z., and Mayfield, F. D. *Chemical Engineering Progress*, V. 52, No. 4, ©1956. American Institute of Chemical Engineers, Inc. All rights reserved.

Table 10-34
Vertical Thermosiphon Reboiler Comparison: Effect of Pressure Drop and Pipe Size on Selection

Tubes:	$^3/_4$ In. × 4 Ft Long	1 In. × 6 Ft Long	1 In. × 6 Ft Long	1 In. × 6 Ft Long
No. tubes	143	50	66	66
Inlet pipe size, in.	6	6	4	6
Outlet pipe size, in.	8	8	6	8
Shell I.D., in.	15.25	12	13.25	13.25
Recirc. ratio	42	40.5	42.1	55.7
ΔP tubes, psi	0.97	1.55	1.03	1.31
ΔP inlet pipe, psi	0.084	0.0789	0.41	0.137
ΔP outlet pipe, psi	0.085	0.0824	0.251	0.111
ΔP expansion, psi	0.00097	0.00209	0.00118	0.0011
U (calc.)	79.3	92.6	82.4	94.0
U (used)	66.1	92.4	70.0	70.0
Area, ft_2	105.3	75.2	99.3	99.3

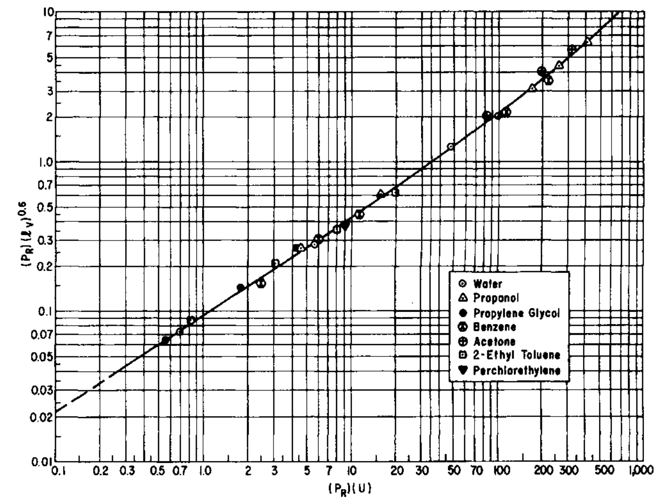


Figure 10-121. Hajek correlation thermosiphon reboilers overall coefficients at maximum flux. (Used by permission: Hajek, J. D. Private communication. Deceased.)

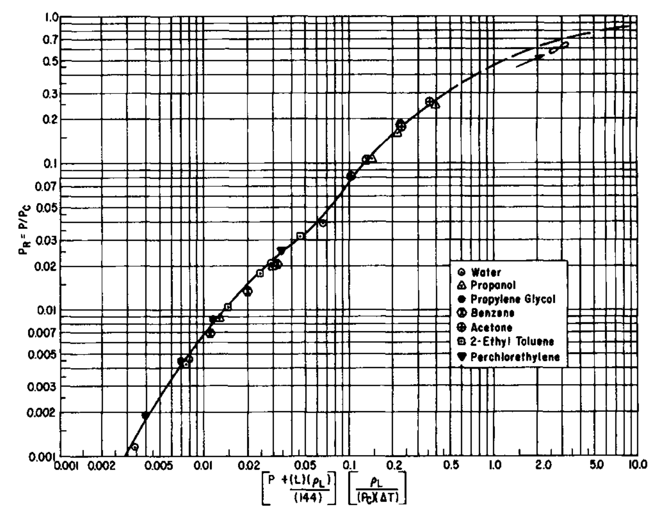


Figure 10-122. Overall ΔT at maximum flux; vertical thermosiphon reboilers. (Used by permission: Hajek, J. D. Private communication. Deceased.)

Simplified Hajek Method—Vertical Thermosiphon Reboiler

Although various correlations exist within thermosiphon reboiler data, each seems to have some limitation. The method of Hajek[60] recommended here has shown excellent correlation above and below atmospheric pressure. Operating data for water, acetic acid, benzene, styrene, ethyl benzene, and styrene-ethyl benzene mixtures agree well with predicted data at intermediate operating heat fluxes. Back calculated fouling factors were about 0.001 for the inside surface. The data of Lee, *et al.*,[75] is also included in the correlation. Data presented are good for vertical natural circulation thermosiphon reboilers with 1-in. 14 BWG Admiralty tubes 4–10 ft in length. Data are based on 10 ft long tubes. Maximum flux for the shorter tubes is greater. For 4, 6, and 8 ft tubes, add 17%, 9%, and 4%, respectively. Do not adjust intermediate flux values below the maximum flux. Column liquid level must be at the top tubesheet.

Design for no more than 90% of calculated maximum flux when using full shell diameter top outlet elbows. When using top-tee type side outlet vapor nozzles, use 76% and 60% of the calculated maximum flux for organics and inorganics, respectively.

Full shell diameter top outlet vapor nozzles are recommended. Top-tee outlet vapor nozzles usually cause the reboiler to fail before maximum flux is reached. If the latter is used, the vapor nozzle should be a minimum of two-thirds the shell diameter.

For 90° elbows off of top vapor outlet channels, the area of nozzle for flow must be *minimum* of 1.25 × the flow area (net) of all tubes.

Inlet nozzles should be sized for 2.5 gpm liquid per tube with the inlet line pressure drop not to exceed 1.5 psi per 100 equivalent ft of inlet piping for total gpm. Nozzles may, in all cases, come into the side of the bottom channel.

The design method is illustrated in Example 10-22 and uses Figures 10-121, 10-122, 10-123, and 10-124.

General Guides for Vertical Thermosiphon Reboilers Design

Nucleate vaporization rates usually run 15–40% (but can be as low as 2–10%) per pass through the unit for organic materials, averaging about 25–28% for many typical conditions. Aqueous solutions range from a low of 5% to 25–30%.

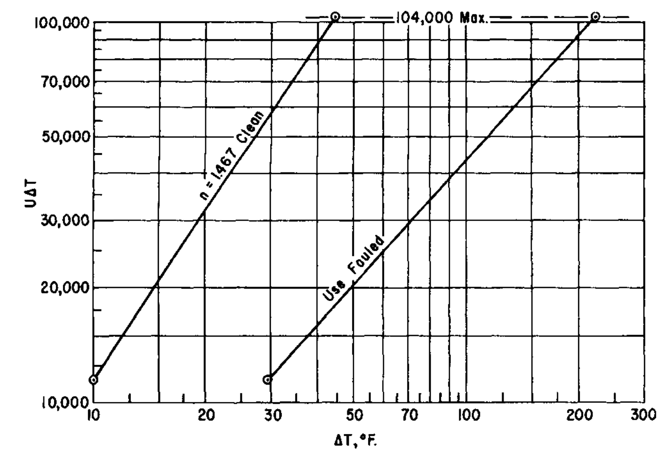


Figure 10-123. Vertical thermosiphon reboilers, UΔT versus ΔT for clean and fouled conditions. (Used by permission: Hajek, J. D. Private communication. Deceased.)

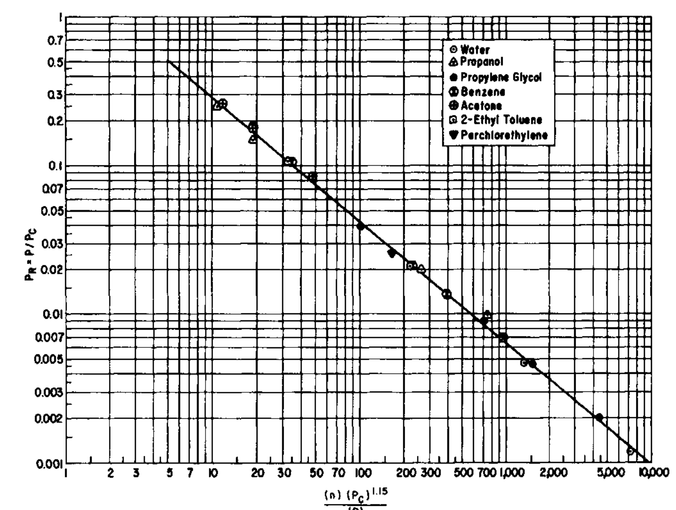


Figure 10-124. Vertical thermosiphon reboilers with a slope of log UΔT/logΔT for determination of ΔT at points below maximum flux. Note: n = slope. (Used by permission: Hajek, J. D. Private communication. Deceased.)

The temperature difference between the exiting vapor-liquid mixture and the inlet shell-side steam or hot fluid should not exceed 75–82°F, primarily due to fouling problems and possible conversion in the tube to inefficient film boiling in the upper section of the tubes.

Frequently used tubes in many vertical thermosiphons are 8 ft long, with 12 ft or 14 ft being maximums. The popular size is 1 $^1/_4$ in. O.D., with 1 in. being used sometimes. Sizes up to 2 in. O.D. are certainly acceptable, depending on the design criteria. Short tubes of 4 and 6 ft are used for special applications, including vacuum conditions.

The best designs provide for the percentage vaporization per pass to have been completed by the time the fluid mixture reaches the upper end of the tube and the mixture is leaving to enter the bottom chamber of the distillation column. In order to assist in accomplishing this, the initial reboiler elevation should be set to have the top tubesheet at the same level as the liquid in the column bottom section. A liquid-level control adjustment capability to raise or lower this bottoms level must exist to optimize the recirculation. Sometimes, the level in the bottom of the column may need to be 25–30% of the reboiler tube length above the elevation of the tubesheet. Therefore, the vapor nozzle return from the reboiler must enter at sufficient elevation to allow for this possibility.

Velocities of liquid entering the tubes should be in the range:

1–4.5 ft/sec when operating in atmospheric pressure and above.

0.4–1.0 ft/sec when operating in a vacuum.

A full opening valve or variable orifice should be able to restrict flows of liquid into the bottom of the reboiler so that the instability in the liquid in the column will not be directly introduced into the inlet of the reboiler. Generally, the liquid inlet nozzle size should be about 50% in the inlet tube flow cross-section area. A large line is sometimes used, but a restricting provision should be provided to to stabilize operations.

Example 10-22. Hajek's Method—Vertical Thermosiphon Reboiler

See Figure 10-125.

A fractionator stripping light ends from water is designed to operate at 80% of tray flooding. The heat load is 4,000,000 Btu/hr. Design the reboiler for full tower capacity or 5,000,000 Btu/hr. A base pressure of 50 psig is required to condense the overhead vapor with cooling water. Steam pressure downstream of the control valve can drop to 200 psig. Use 6-ft long Admiralty tubes, 1 in. O.D. by 14 BWG on 1 $^1/_4$ -in. triangular pitch. Inside and outside fouling resistances are 0.001 and 0.0005, respectively.

Physical data required

L = 6 ft tube length
ρ_L = 57.4 lb/ft^3, liquid density
l_v = 911.8 Btu/lb, heat of vaporization based on vapor to bottom tray
P = 64.7 psia, tower base vapor space pressure
P_c = 3,206 psia, critical pressure

Variables to be determined

U = overall heat transfer coefficient, Btu/hr (ft^2) (°F)
ΔT = over-all temperature difference, °F
n = slope = log UΔT/logΔT
P_R = P/Pc = reduced pressure

DWG. NO. A

Item No.

By

EXCHANGER RATING

Job No.

Date:-

Charge No.

Apparatus Reboiler Plant

Minimum Surface Area per Shell, Sq. Ft.: 1700 Outside Inside:

Number of Units: Operating Two Shells Spares None

DESIGN DATA PER Shell

UNIT DATA		SHELL SIDE	TUBE SIDE
Fluid		10 psig Steam	Hydrocarbon
Fluid Flow	Lbs./Hr.	3,140	
Temperature In	°F.	239.3	223
Temperature Out	°F.	239.3	223
Operating Pressure	PSIG	10.0	285 MM Hg. Abs.
Density	Lbs./CF		50.2
Specific Heat	Btu/Lb./°F.		
Latent Heat	Btu/Lb.	952.6	156.7
Therm. Cond.	Btu/Hr./Sq. Ft./°F-/Ft.		
Viscosity	Centipoise		
Molecular Weight			
No. of Passes		1	1
Pressure Drop	PSI	Calc: Used:	Calc: Used:
Fouling Factor		0.0005	0.0011

Heat Transferred - BTU/Hr: 3,000,000

LMTD 16.3 °F. Overall U: Calc. 119.2 Used 103.6

CONSTRUCTION

Max. Oper. Pressure	50 PSI	Full Vacuum
Max. Oper. Temperature	450 °F.	450 °F.

Type of Unit Vertical Fixed Tube Sheet Tube Pitch 1 1/4" Δ Joint 2 Groove Roll

Tubes - Material: Steel No. (Approx.) 1145 O.D. 1" BWG 12 Length 6'-0"

Shell - Material: Steel Diameter (Approx.):

Channel Material: Steel Supports Material:

Tube Sheet Material: Steel Baffle Material: Steel

Corrosion Allowance - Shell Side 1/16" Tube Side

Connections - Shell In: 6" Out 2" Flange 150# RF

Channel In: 12" Out 20" Flange 150# RF

Others Bottoms & Gage Size 3" Flg. & 3/4" cplg. Flange 150# RF & 6000# cplg.

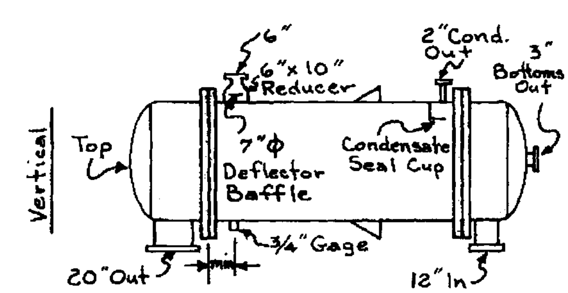
Bolts: Gaskets: JM-60

Code ASME Stamp Yes X-Ray SR

Insulation Yes Class G Cathodic Protection No

Baffles in shell to have maximum cut for tube support only.

BAFFLE ARRANGEMENT



NOZZLE ARRANGEMENT

Remarks:- Approximate diameter = 48"

Use 1" X 6000# cplgs. on all nozzles.

Checked Date Approved Date

Rev. By Date B/M No.

Figure 10-125. Thermosiphon reboiler specifications.

Determine Overall Coefficient at Maximum Flux

$$(P_R)(l_v)^{0.6} = \left(\frac{64.7}{3{,}206}\right)(911.8)^{0.6} = 1.20 \tag{10-197}$$

From Figure 10-121,

$$(P_R)(U) = 47$$

$$U = \frac{(47)}{(P_R)} = \left(\frac{47}{0.0202}\right) = 2{,}326 \text{ Btu/hr (ft}^2)(°\text{F}),$$

clean based on O.D. surface. (10-198)

Determine Overall ΔT at Maximum Flux

From Figure 10-122,

$$\text{For } P_R = 0.0202;\ \left[P + \frac{(L)(\rho_L)}{144}\right]\left[\frac{(\rho_L)}{(\Delta T)(P_c)}\right] = 0.0268 \tag{10-199}$$

$$\left[64.7 + \frac{(6)(57.4)}{(144)}\right]\left[\frac{(57.4)}{(3{,}206)(\Delta T)}\right] = 0.0268$$

$$\Delta T = \frac{(67.1)(0.0179)}{(0.0268)} = 44.8°\text{F, clean}$$

Maximum Flux

$$U\Delta T = (2{,}326)(44.8) = 104{,}000 \text{ Btu/hr-ft}^2$$

Maximum flux is the same for a clean or fouled tube (see Figure 10-123) with ΔT increasing as U decreases due to fouling.

Flux at Operating Levels Below Maximum

$U\Delta T$ plotted against ΔT on log-log paper gives a straight line.
See Figure 10-123.
Determine $U\Delta T$ at, say, 10°F clean ΔT to get another point other than at 44.8°F.

From Figure 10-124,

$$\text{For } P_R = 0.0202;\ \frac{(n)(P_c)^{1.15}}{(P)} = 244 \tag{10-200}$$

$$(P_c)^{1.15} = (3{,}206)^{1.15} = 10{,}760$$

$$n = \frac{(244)(64.7)}{(10{,}760)} = 1.467$$

$$U\Delta T \text{ for } 10°\text{F } \Delta T = (104{,}000)\left(\frac{10}{44.8}\right)^{1.467}$$

$$= 11{,}540 \text{ Btu/hr-ft}^2$$

$$U \text{ for } 10°\text{F } \Delta T = \frac{(11{,}540)}{(10)} = 1{,}154 \text{ Btu/hr (ft}^2)(°\text{F) clean}$$

Plot these data as shown in Figure 10-123 to get the flux versus clean surface ΔT.

Fouled ΔT at Maximum Flux

$$\frac{1}{U \text{ max clean}} = \frac{1}{2{,}326} = 0.00043$$

r_i = 0.0010 inside fouling; correct to outside surface

$$r_{io} = (0.0010)\left(\frac{\text{O.D.}}{\text{I.D.}}\right) = (0.0010)\left(\frac{1.00}{0.834}\right) = 0.0012$$

r_o = 0.0005 outside steam fouling resistance

$$U \text{ max dirty} = \left(\frac{1}{0.00043 + 0.0012 + 0.0005}\right)$$

$$= 469 \text{ Btu/hr (ft}^2)(°\text{F})$$

$$\Delta T \text{ dirty} = \frac{(U\Delta T)}{U \text{ dirty}} = \left(\frac{104{,}000}{469}\right) = 220°\text{F} \tag{10-201}$$

Plot this point as on Figure 10-123. Because the correlation is based on 14 BWG Admiralty tubes, no correction was made for tube wall resistance.

Fouled ΔT To Maintain Flux for 10°F Clean ΔT

$$\left(\frac{1}{U \text{ clean for } 10°\text{F } \Delta T}\right) = \left(\frac{1}{1{,}154}\right) = 0.000866$$

$$U \text{ dirty} = \left(\frac{1}{0.000866 + 0.0012 + 0.0005}\right)$$

$$= 390 \text{ Btu/hr (ft}^2)(°\text{F})$$

$$\Delta T \text{ dirty} = \left(\frac{11{,}540}{390}\right) = 29.6°\text{F for } U\Delta T = 11{,}540$$

Plot this point on Figure 10-123 and connect with the 222°F point to get the fouled flux line.

Analysis of Data on Figure 10-123

Boiling point of water in tower = 298°F
Assume the reboiler will have a full shell diameter top outlet elbow vapor nozzle. Thus, a flux of (104,000) (0.90) or 93,600 Btu/hr (ft²) is possible if steam temperature is adequate. At the latter flux, the fouled ΔT = 204°F. This will require a steam temperature of 298 + 204 or 498°F equivalent to 693 psia steam. Because only 200 psig steam is available downstream of the control valve at the chest, a lower flux must be used.
Steam at 215 psia has a temperature = 388°F
Available ΔT = 388 − 298 = 90°F fouled
From Figure 10-123 for the fouled condition at 90°F ΔT, a flux of only 38,600 Btu/hr (ft²) is available. Because this is less than 60% of maximum flux allowed for water and other inorganics when using a tee-type top vapor side outlet, the latter construction will be used to reduce investment slightly.

Surface Area Required

$$Q = 5{,}000{,}000 \text{ Btu/hr}$$

$$\text{Area} = \left(\frac{5{,}000{,}000}{38{,}600}\right) = 130 \text{ ft}^2$$

Use (130)(1.10) = 143 ft²

Tube length corrected for tubesheets = 5.73 ft

$$\text{Number of tubes} = \frac{(143)}{(0.2618)(5.73)} = 96 \text{ tubes}$$

A 16-in. O.D. shell is required.

Vapor Nozzle Diameter

(16) (0.667) = 10.7 in. approximately
Use a 10-in. vapor nozzle

Liquid Inlet Nozzle Diameter

gpm = (2.5/tube)(96 tubes) = 240 gpm
4-in. is too small; use 6 in.

Design Notes

When steam pressures in the chest are near atmospheric, condensate can rise in the shell and drastically reduce available surface—if the trap is too small to dump steam into the condensate return system or if the condensate return pressure is greater than the calculated chest pressure required. In these cases, the steam pressure will have to rise in the chest to overcome this error, if steam pressure is available. If not, the reboiler will not deliver design flux.

Proper condensate removal is important. An inverted split cup inside the shell, with the upper capped end above the nozzle and the lower open end $^3/_4$ -in. above the bottom tubesheet, should be used to cover the outlet nozzle. This can be made by splitting a pipe that is one size larger than the condensate outlet down the centerline. In this case, a 2-in. split is adequate. This cup must be fully seal welded (not tack welded) to force condensate down to the $^3/_4$ -in. clearance above the bottom tubesheet. A common error is to allow 6 in. or more above the tubesheet for the centerline of the condensate outlet. In this case, 6 in. of tube is 10% of the surface. If the cup is not used, add 10% more tubes to correct for the dead liquid space near the bottom. This is in addition to the 10% safety factor.

In the region of low Δt (less than about 10), the heat flux is about 10 times as great for water under forced convection (agitation) as for natural circulation.[82] This does not hold at the higher Δt values. The critical Δt is practically unaffected by agitation or increased velocity over the value at natural convection. In general, the heat flux is lower for a given Δt at lower pressures. Likewise, the peak Q/A is lower.

When a liquid is vaporized in horizontal tubes, the initial overall coefficient is several times the value for forced convection single-phase heat transfer. As the amount of vapor increases up to 100%, the coefficient falls off, down to a gas convection coefficient. The work of McAdams[84, 85] is some of the limited literature in this type of heat transfer.

Reboiler Piping

The mechanical design of thermosiphon reboiler piping must be carefully examined for (a) system pressures and (b) elevation relationship between the liquid level in the distillation column and the vertical or horizontal reboiler. Kern[199] provides an excellent presentation on this topic, including the important hydraulics. Abbot[200] also presents a computer program for this topic.

Film Boiling

Normally the designer does not try to establish film boiling conditions for the vaporizers or reboilers. However, for systems set by other controlling processing conditions, these film conditions may be imposed. In such cases, they should be recognized and handled accordingly. The principles of design for the equipment are the same as other such equipment, and only the actual value of the coefficient affected needs special attention.

Vertical Tubes, Boiling Outside, Submerged[14]

Conditions:	Above critical temperature difference for nucleate boiling
Tube sizes:	$^3/_8$ -in.–$^1/_2$ -in. O.D. (data of correlation)
Reynold's number:	800–5,000
Range of error:	14% (greatest single value in correlation data is 36%)

$$h_a' = 0.002 \left[\frac{4w}{\pi D_o \mu}\right]^{0.6} \left[\frac{2}{k_a^3 \rho_v(\rho_L - \rho_v)g}\right]^{-1/3} \quad (10\text{-}202)$$

where w = maximum vapor flow per tube, lb/hr
D_o = tube O.D., ft
μ = viscosity of vapor, lb/(ft) (hr)
k_a = thermal conductivity of vapor, Btu/hr (ft) (°F)
ρ_v = density of vapor, lb/ft³
ρ_L = density of liquid, lb/ft³
g = 4.17×10^8, ft/hr²
h_a' = average heat transfer coefficient over entire tube, Btu/hr (ft²) (F)

The data for organic fluids and low temperature nitrogen fit. However, methanol data gives coefficients many times higher.

Horizontal Tubes: Boiling Outside, Submerged[14]

Conditions:	Pressure to 500 psi, temperatures of saturated liquid to 700°F
Tubes:	0.025–0.75-in. O.D. (data of correlation)
Fluids:	Hydrocarbons, alcohols, benzene, chlorinated compounds, low-temperature nitrogen, and oxygen

$$h'_a = a'\left(\frac{1}{d_o} + 36.5\right)\left[\frac{k^3\rho_v(\rho_L - \rho_v)g\lambda'}{\Delta t_b \mu}\right]^{1/4} \quad (10\text{-}203)$$

Δt_b, °F	200	300	400	500
a, in./(ft)$^{1/4}$	0.0461	0.0450	0.0445	0.0441

$$\lambda' = h_{fg}\left(1 + 0.4\frac{c_p \Delta t_b}{\lambda'}\right)^2 \quad (10\text{-}204)$$

where d = tube O.D., in.
a' = constant, (in.)/(ft)$^{1/4}$
k_a = thermal conductivity of vapor at arithmetic average mean temperature, Btu/hr (°F)(ft)
ρ_v = density of vapor at its arithmetic mean temperature, lb/ft^3
ρ_L = density of saturated liquid, lb/ft^3
λ' = difference in enthalpy between vapor at its arithmetic temperature and saturated liquid, Btu/lb
μ = viscosity of vapor at mean temperature, lb/hr (ft)
Δt_b = temperature difference between heat transfer surface and boiling liquid, °F
h_a' = average film coefficient Btu/hr (ft^2) (°F)
h_{fg} = latent heat of vaporization, Btu/lb
c_p = specific heat at constant pressure of vapor at arithmetic mean temperature, Btu/lb (°F)

Horizontal Film or Cascade Drip-Coolers—Atmospheric

The film cooler, Figure 10-126, fabricated from pipe lengths, is popular and relatively inexpensive for some cooling and condensing applications. The principle of operation is also used in the cast iron sections of Figures 10-5A, 10-5B, and 10-127, which are particularly useful in handling sulfuric and similar acids. The same unit construction is sometimes submerged in cooling tanks or placed in the basins of cooling towers. Graphite-impregnated film coolers are used in hydrochloric acid cooling (Figure 10-128).

Design Procedure

1. Determine heat duty, Btu/hr.
2. Assume exit water temperature about 10–15°F (maximum) greater than inlet water, if possible.
3. Calculate water required with selected temperature rise. This rate should fall between 2–10 gpm per lin ft of plan coil length. Values greater than 10 gpm cause overflooding of tubes and a waste of water.
4. Determine LMTD if flows are counterflow and apply the correction factor of Bowman[8] *et al.*, established for this type of unmixed "shell-side" flow (Figure 10-129).[70]

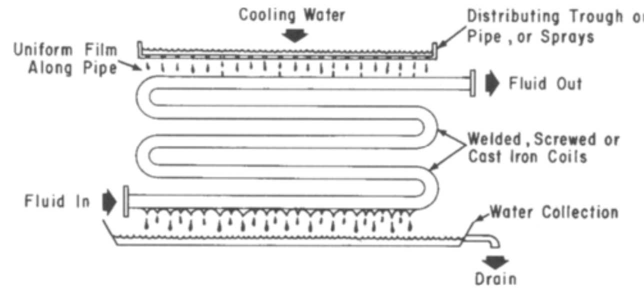


Figure 10-126. Horizontal film type cooler or condenser.

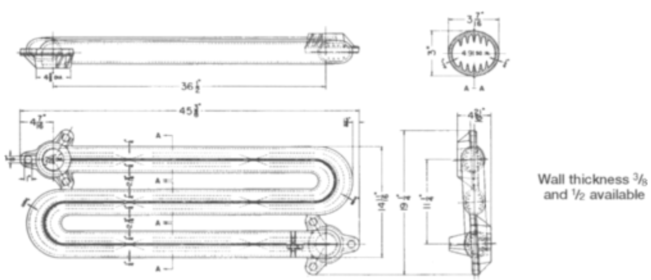


Figure 10-127. Typical cast iron cooling section.

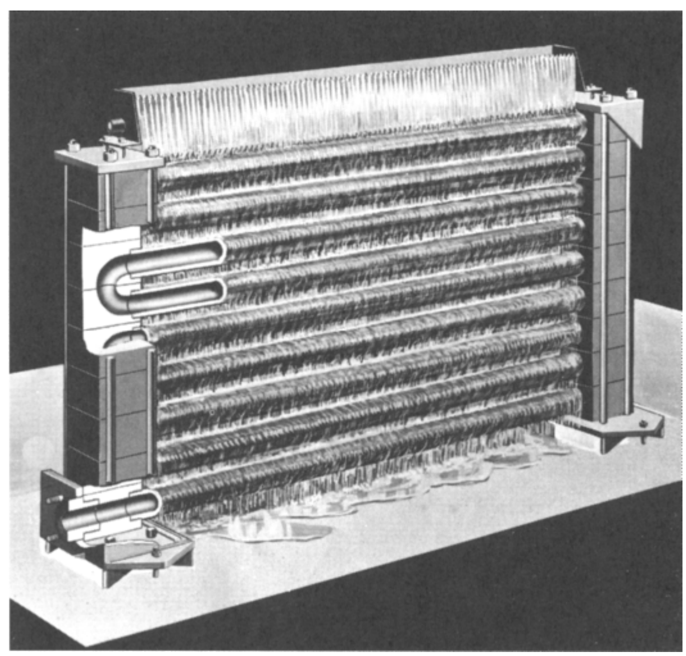


Figure 10-128. Graphite film coolers. (Used by permission: Bul. 537, Falls Industries, Inc. Research indicates that company went out of business, 1999.)

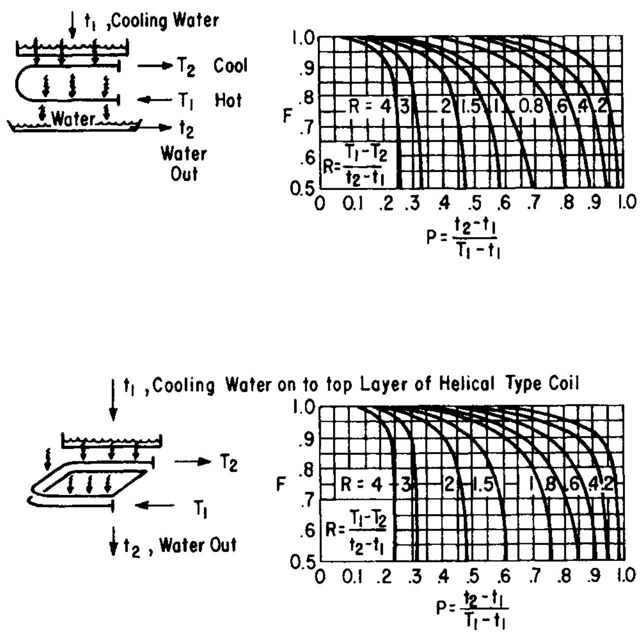


Figure 10-129. MTD correction factors for drip type coolers. (Used by permission: Bowman, R. A., Mueller, A. C., and Nagle, W.M. *Transactions of ASME,* No. 62, ©1940. American Society of Mechanical Engineers. All rights reserved.)

5. Assume a unit for the service or assume a pipe size and length and determine the number of lengths required.
6. Determine outside film coefficient for spray or drip cooling using the equation of McAdams[81] as presented by Kern.[70]

$$h_o = 65\left(\frac{G_d}{D_o}\right)^{1/3} \tag{10-205}$$

where h_o = outside film coefficient, ± 25%, Btu/hr (ft²) (°F)
G_d = W/2L, lb/hr (ft)
W = lb cooling water/hr flowing over length of tube
L = length of each pipe in bank, ft
D_o = O.D. of pipe, ft

For most purposes, an estimated value of h_o = 500 − 550 is conservative.

For a submerged unit, handle as natural convection on outside of pipes; values usually range from 40–130 Btu/hr (ft²) (°F) for h_o.

7. Determine the inside film coefficient using Equation 10-41 and Figure 10-46 for tube-side heat transfer. If two or more coils are in parallel, be certain that the flow rate per pipe is used in determining h_i. Correct h_i to outside of tube, giving h_{io}. Note that Figure 10-46 also applies to cast iron cooling sections.

Table 10-35
Typical Fouling Factors—Cast Iron Cooling Sections

Service	Inside Fouling	Outside Fouling
Concentrated sulfuric acid	0.002	—
Ammonia liquor	0.002	—
Clean water	0.001	0.005
Clean oil	0.002	—
Dirty oil	0.005	—
Tar	0.01	—
Dirty water		0.01
Sea water, brackish water		0.01–0.05

Used by permission: Cat. HT-23, National U.S. Radiator Corp. Existence of company not confirmed (1998). Value for sea water provided by this author.

Table 10-36
Wall Resistance Factors—Cast Iron Cooling Sections

Metal Thickness, in.	Wall Transfer, h_w, Btu/hr (ft²) (°F)
1/4	1,800
3/8	1,350
1/2	900

Used by permission: Cat. HT-23, National U.S. Radiator Corp. Existence of company not confirmed (1998).

8. Assume fouling factors. Inside tube factors can be selected from Table 10-12 or 10-13 or by referring to Table 10-15. Because the water rate is low over these coolers, they may develop salt crusts, scale, algae, etc.; therefore, the values of fouling will be high, see Table 10-35.
9. Determine overall coefficient U in the usual manner.

$$U = \frac{1}{\frac{1}{h_{io}} + r_i + \frac{1}{h_w} + r_o + \frac{1}{h_o}}$$

$L_w/k = 1/h_w$, wall resistance

For cast iron sections, see Tables 10-36 and 10-37.

10. Calculate area required:

$$A = \frac{Q}{U\ (\text{corrected LMTD})}$$

If this area is considerably different than the assumed unit, reassume a new unit and recalculate the preceding steps.

Table 10-37
Typical Cast Iron Section, Type B, Figure 10-127

Metal Thickness, in.	Cubic Contents, gal	Wt./ Section	Internal Surf., ft^2	External Surf., ft^2	Eq. Diam., in.
$^1/_4$	2.7	130	11.0	10.0	1.57
$^3/_8$	2.7	180	11.0	10.8	1.57
$^1/_2$	2.7	210	11.0	11.5	1.57

Note: Other types of sections are available to accomplish the same type of cooling.

Used by permission: Cat. HT-23, National U.S. Radiator Corp. Existence of company not confirmed (1998).

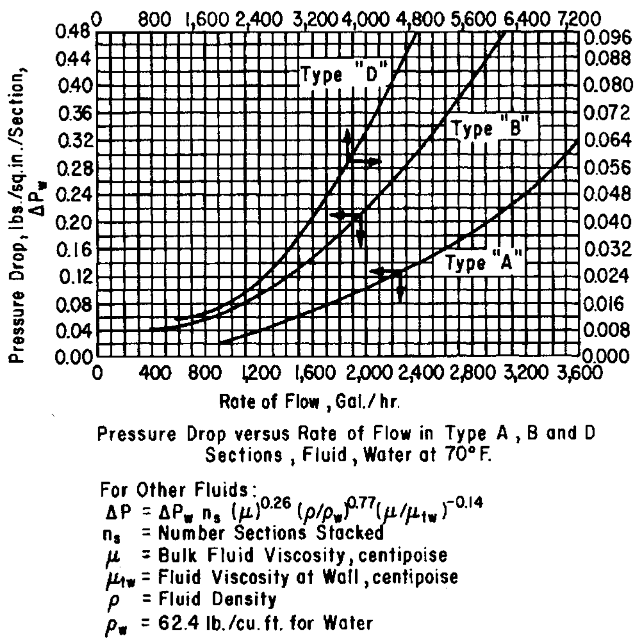


Figure 10-130. Pressure drop versus rate of flow for water at 70°F in cast iron cooling sections, similar to Figure 10-127.

If the calculations were started by assuming a pipe size and length, determine the number of lengths from the total area calculation and surface area per length of pipe selected.

$$\text{No. lengths} = \frac{\text{total surface A}}{\text{outside surface area/pipe length}} \quad (10\text{-}206)$$

Factor of safety or percent excess area should be at least 10–15%.

11. From a balanced design, determine the pressure drop for the entire series length of pipe in bank, including fittings. Use copyrighted chart in Reference 36, fluid flow principles, or Figure 10-130 for cast iron sections. If pressure drop is too high, reselect and redesign unit, making parallel units to reduce flow rate (and coefficient h_{io}), or select a larger pipe, reducing mass rate G, and hence h_{io}. Recalculate the pressure drop.

Impregnated graphite coolers, Figure 10-131 and Table 10-38, are used in acids and other corrosive liquids. The selection charts of Figures 10-132, 10-133, and 10-134 can be used to determine expected transfer coefficients and total external cooling surface for a typical style of unit. Although these charts are specific to the manufacturer's wall thicknesses and the thermal conductivity of the material, they are nevertheless convenient and generally acceptable. Exact selections should be obtained from the manufacturers by giving them the flow data and performance requirements. Pressure drops can be estimated from Figures 10-135 and 10-136.

Pressure Drop for Plain Tube Exchangers

A. Tube Side

Pressure loss through the inside of the tubes during heating or cooling in heat exchangers is given for liquids and gases by[70]

$$\Delta p_t = \frac{f\,G_t^2\,Ln}{2g\rho D_i \phi_t} = \frac{f\,G_t^2\,Ln}{5.22(10)^{10} D_i\, s\, \phi_t}, \text{ psi} \quad (10\text{-}207)$$

The friction factor, f, $ft^2/in.^2$, must be obtained from Figure 10-138. Because it is not a dimensional factor, the Δp_t relations take this into account.

$$\phi_t = \left(\frac{\mu}{\mu_w}\right)^{0.14} \text{ for } R_e > 2{,}100 \quad (10\text{-}208)$$

$$\phi_t = \left(\frac{\mu}{\mu_w}\right)^{0.25} \text{ for } R_e < 2{,}100 \quad (10\text{-}209)$$

For noncondensing gases and vapors in Equation 10-207 use the average of inlet and outlet gas density referenced to water at 62.4 lb/ft^3 for the value of s.

A convenient chart for water pressure drop in tubes is given in Figure 10-138.

A convenient chart for all fluids[38] including a 20% increase in pressure drop over theoretical smooth tubes is given in the copyrighted figure of Reference 36:

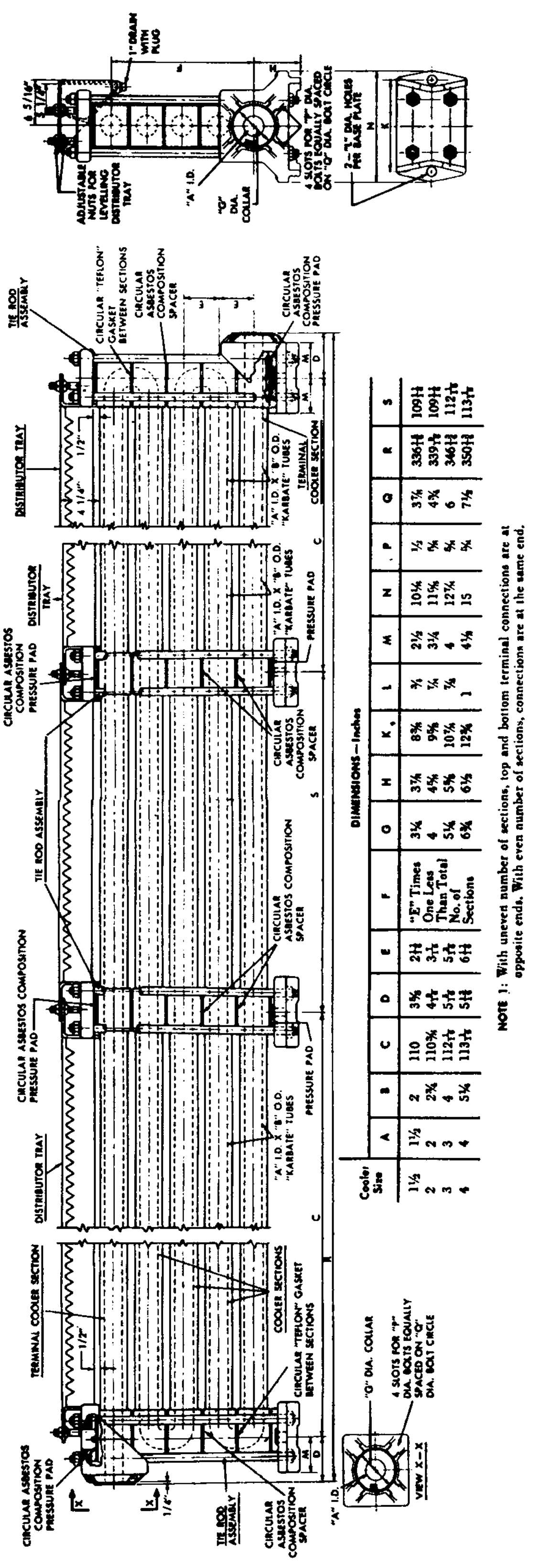
For streamline flow, $R_e < 2{,}100$:

$$f_f = \frac{16}{D_i G_t/\mu} \quad \text{(see note below regarding f)} \quad (10\text{-}210)$$

and this can be used in Equation 10-214.

The turbulent flow[38] $R_e > 2{,}100$:

$$f_f = 0.048/\left(\frac{D_i G_t}{\mu}\right)^{0.2} \quad (10\text{-}211)$$



DIMENSIONS—Inches

Cooler Size	A	B	C	D	E	F	G	H	K	L	M	N	P	Q	R	S
1½	1½	2	110	3⅝	2 11/16	"E" Times One Less Than Total No. of Sections	3¼	3⅞	8⅝	¾	2½	10¼	½	3⅞	336 13/16	109 13/16
2	2	2¾	110¾	4 5/16	3 1/16		4	4⅝	9⅝	⅞	3¼	11⅝	⅝	4¾	339 5/16	109 13/16
3	3	4	112 1/16	5 5/16	5 3/16		5¼	5⅝	10⅞	⅞	4	12¾	⅝	6	346 13/16	112 1/16
4	4	5¼	113 1/16	5 13/16	6 11/16		6¾	6½	12⅜	1	4½	15	⅝	7½	350 13/16	113 1/16

NOTE 1: With uneven number of sections, top and bottom terminal connections are at opposite ends. With even number of sections, connections are at the same end.

Figure 10-131. Typical sectional cooler using assembly of standardized components. (Used by permission: SGL Technic, Inc., Karbate® Division.)

Divide this f by 144 in order to use in Δp_t Equation 10-191 or 10-207.

Stoever[108, 109] presents convenient tables for pressure drop evaluation.

Pressure drop through the return ends of exchangers for any fluid is given as four velocity heads per tube pass[70]

$$\cong \frac{4nv^2}{2g'}, \text{ ft}$$

$$\Delta p_r = \frac{4nv^2 s}{(2g')}\left(\frac{62.5}{144}\right), \text{ psi} \tag{10-212}$$

This is given in Figure 10-139.

where Δp_r = return end pressure loss, including entrance losses, psi

n = no. of tubes passes per exchanger

g' = acceleration of gravity, 32.2 ft/(sec)^2

s = specific gravity of fluid (vapor or liquid) referred to water

v = tube velocity, ft/sec

$$\Delta pr = \frac{4n(G'')^2}{2g's}\left(\frac{1}{(62.5)(144)}\right)$$

G'' = mass velocity for tube side flow, lb/(sec) (ft^2 cross-section of tube)

Total Tube Side Pressure Drop

$$= \Delta p_t + \Delta p_r, \text{ psi} \tag{10-213}$$

Tube Side Condensation Pressure Drop

Kern[70] recommends the following conservative relation:

$$\Delta p_t = \frac{9.56(10)^{-12} f (G_t)^2 Ln}{D_i s}, \text{ psi} \tag{10-214}$$

This is one-half the values calculated for straight fluid drop, based on inlet flows; f is from Figure 10-137.

B. Shell Side

Pressure losses through the shell side of exchangers are subject to much more uncertainty in evaluation than for tube side. In many instances, they should be considered as approximations or orders of magnitude. This is especially true for units operating under vacuum less than 7 psia. Very little data has been published to test the above-atmospheric pressure correlations at below-atmospheric pressures. The losses due to differences in construction, baffle clearances, tube clearances, etc., create indeterminate values for exact correlation. Also see the short-cut method of reference 279.

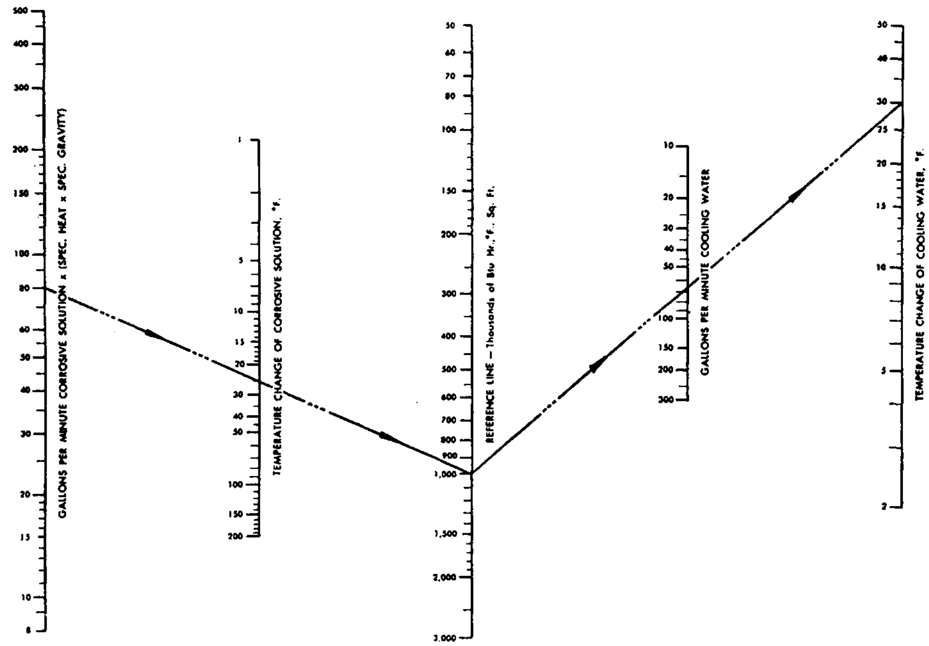
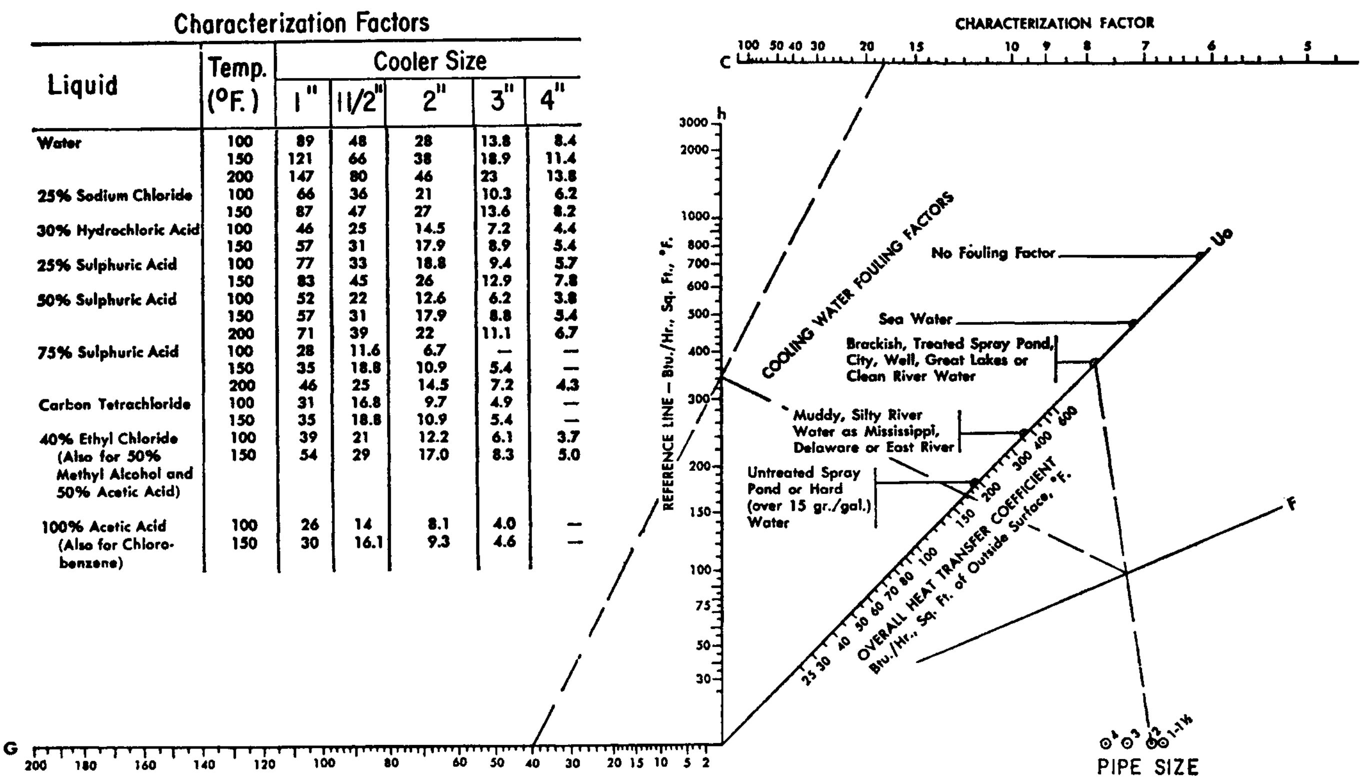


Figure 10-132. Cooling water requirements for cooler of Figure 10-131. (Used by permission: SGL Technic, Inc., Karbate® Division.)



Characterization Factors

Liquid	Temp. (°F.)	Cooler Size 1"	1 1/2"	2"	3"	4"
Water	100	89	48	28	13.8	8.4
	150	121	66	38	18.9	11.4
	200	147	80	46	23	13.8
25% Sodium Chloride	100	66	36	21	10.3	6.2
	150	87	47	27	13.6	8.2
30% Hydrochloric Acid	100	46	25	14.5	7.2	4.4
	150	57	31	17.9	8.9	5.4
25% Sulphuric Acid	100	77	33	18.8	9.4	5.7
	150	83	45	26	12.9	7.8
50% Sulphuric Acid	100	52	22	12.6	6.2	3.8
	150	57	31	17.9	8.8	5.4
	200	71	39	22	11.1	6.7
75% Sulphuric Acid	100	28	11.6	6.7	—	—
	150	35	18.8	10.9	5.4	—
	200	46	25	14.5	7.2	4.3
Carbon Tetrachloride	100	31	16.8	9.7	4.9	—
	150	35	18.8	10.9	5.4	—
40% Ethyl Chloride (Also for 50% Methyl Alcohol and 50% Acetic Acid)	100	39	21	12.2	6.1	3.7
	150	54	29	17.0	8.3	5.0
100% Acetic Acid (Also for Chlorobenzene)	100	26	14	8.1	4.0	—
	150	30	16.1	9.3	4.6	—

Figure 10-133. Overall heat transfer coefficient for Karbate® impervious graphite cascade cooler. (Used by permission: SGL Technic, Inc., Karbate® Division.)

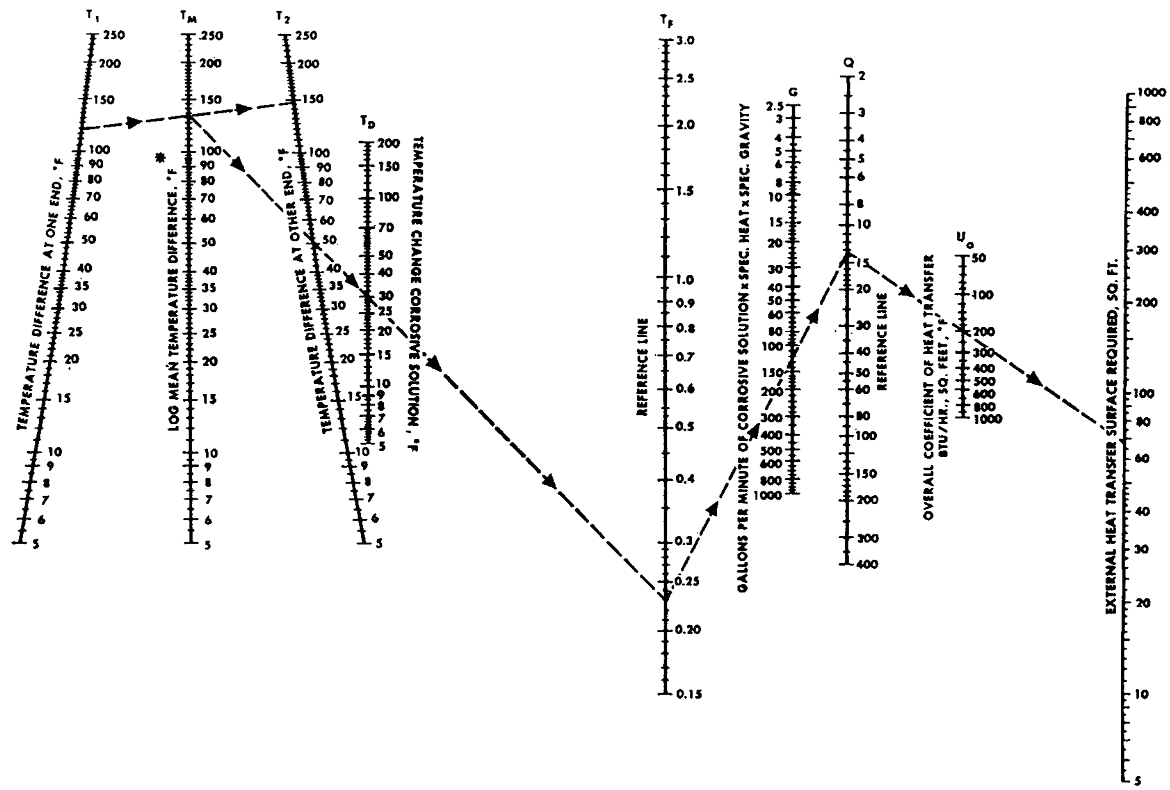


Figure 10-134. Required cooling surface. (Used by permission: SGL Technic, Inc., Karbate® Division.)

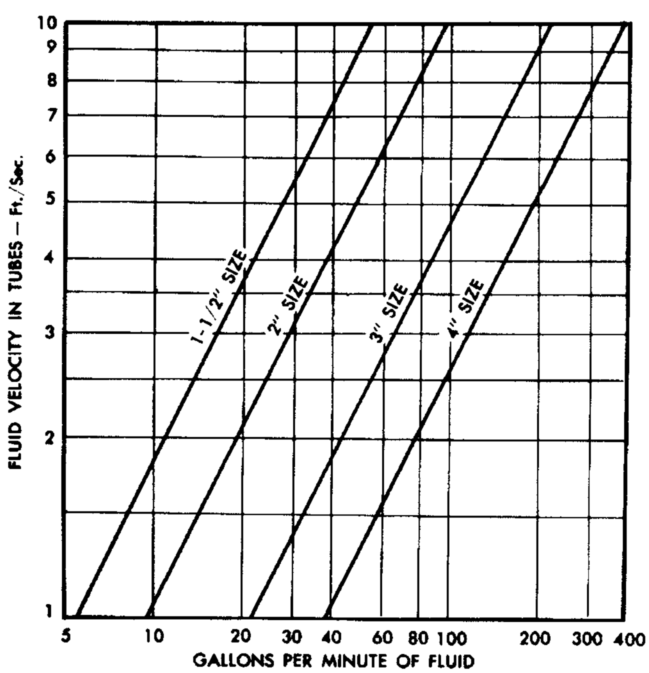


Figure 10-135. Tube-side fluid velocity for cascade cooler. (Used by permission: SGL Technic, Inc., Karbate® Division.)

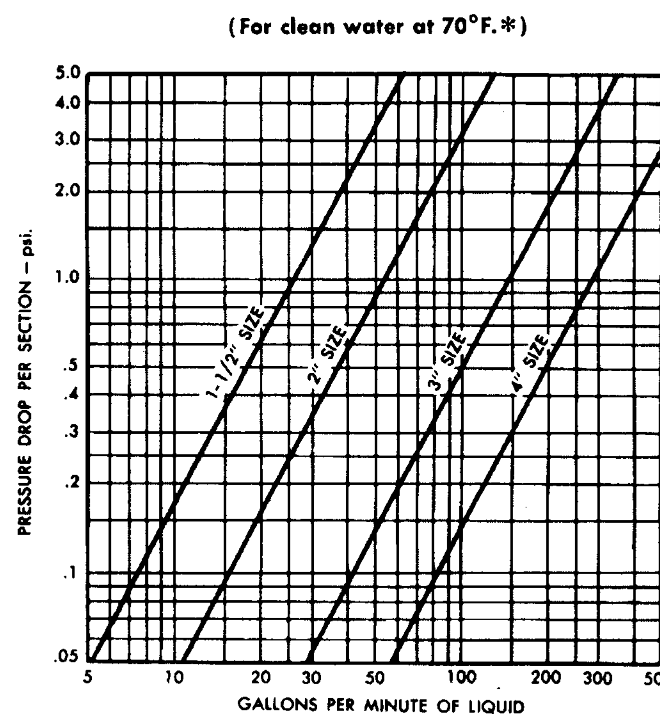


Figure 10-136. Tube-side liquid pressure drop for cascade cooler. For nonwater liquids, multiply pressure drop by $(\mu')^{0.140}(s)^{0.86}$ (Used by permission: SGL Technic, Inc., Karbate® Division.)

Table 10-38
Data for Coolers of Figure 10-131

Cooler Size, in.	Pipe I.D. (Nom.), in.	Pipe O.D. (Nom.), in.	Inside Cross-section, ft²	Effective Inside Area Per Section, ft²	Effective Outside Area Per Section, ft²	Max. No. of Sections Per Cooler	Total Effective Outside Area for Max. No. of Sections	Weight Each Cooler Section, lb	Weight Set of Tie Rod Assemblies, lb
1 1/2	1 1/2	2	0.01227	10.60	14.14	26	367.6	40	182
2	2	2 3/4	0.0218	14.14	19.44	20	388.8	65	299
3	3	4	0.0491	21.21	28.27	13	367.6	180	381
4	4	5 1/4	0.0873	28.27	37.11	10	371.1	285	541

Used by permission: Cat. S-6820, ©1953. National Carbon Co. Existence of company not confirmed (1998).

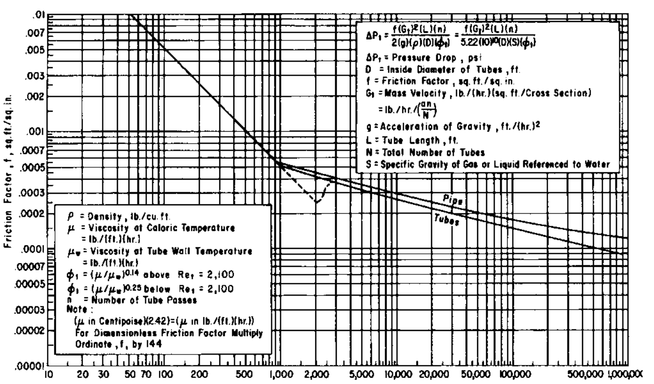


Figure 10-137. Heating and cooling in tube bundles—tube-side friction factor. (Used by permission: Kern, D. Q. *Process Heat Transfer,* 1st Ed., p. 836, ©1950. McGraw-Hill, Inc. All rights reserved. Using nomenclature of Standards of Tubular Exchanger Manufacturers Association.)

Unbaffled Shells

For short exchangers with no shell-side baffles, pressure drop is usually negligible. Allowances should be made for nozzle entrances and exits if the pressure level of the system warrants this detail.

For longer units requiring support plates for the tubes, the pressure drop will still be very small or negligible and can be estimated by Figure 10-140 using the appropriate baffle cut curve to match the tube support cut-out of about 50%. Kern[70] recommends that the flow be considered similar to an annulus of a double pipe and treated accordingly.

Equivalent shell-side diameter for pressure drop, D_e':

$$= \frac{4\text{ (flow area of space between shell and tubes)}}{\text{wetted perimeter of tubes + wetted perimeter of shell I.D.}} \tag{10-215}$$

$$D_e' = \frac{4\left[\dfrac{\pi D_s^2/4 - N\pi d_o^{\,2}/4}{144}\right]}{\dfrac{N\pi d_o}{12} + \pi\dfrac{D_s}{12}} \tag{10-216}$$

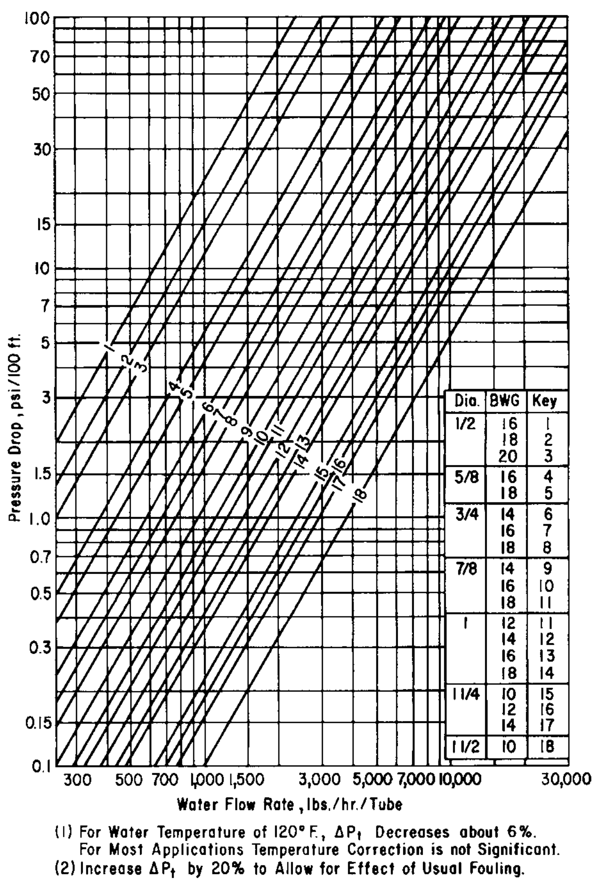


Figure 10-138. Pressure drop for water in smooth tubes at 68°F. (Used by permission: *Scovill Heat Exchanger Tube Manual,* 3rd Ed. Scovill Manufacturing Co.)

where D_e' = equivalent diameter for pressure drop of bundle in shell, ft
D_s = shell I.D., in.
N = number of tubes
d_o = tube O.D., in.

$$\Delta p_r = \frac{4nv^2 s}{2g'}\left(\frac{62.5}{144}\right), \text{ psi}$$

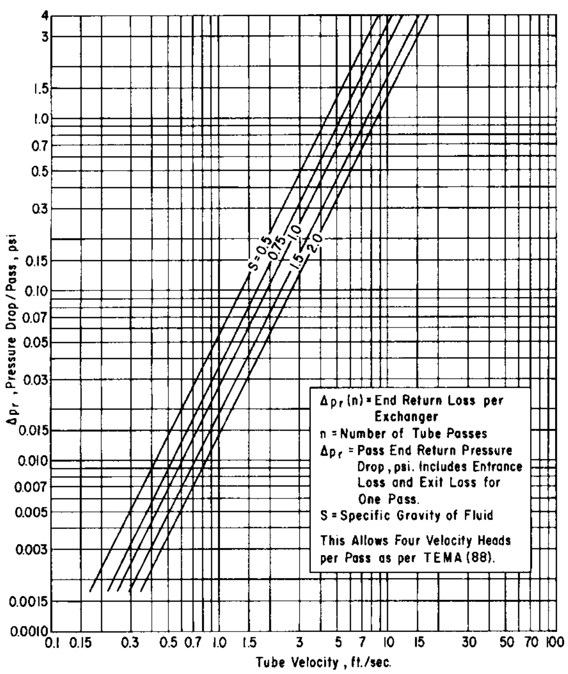


Figure 10-139. Tube side end return pressure drop per tube pass; viscosity close to water.

The friction factor, f_s, is determined using Figure 10-140 for shell-side pressure drop with D_e, used in determining R_e. For bundles with bare tubes (plain tubes), $f_s = f/1.2$ (see Figure 10-140), calculate pressure drop:

$$\Delta p_s = \frac{f_s G_s^2 L N_c}{5.22(10)^{10} D_e' s \phi_s} \text{ psi} \tag{10-217}$$

where N_c = 1 for single-pass shell, no baffles
$\phi_s = (\mu/\mu_w)^{0.14}$
Δp_s = shell side pressure drop with no baffles, psi

Segmental Baffles in Shell

Figure 10-140 is used for determining the friction factor (dimensional) for segmental type baffles. The loss across the tube bundle and through the baffle "window" is represented in the combined factor, f, which is to be used with the equation for pressure drop.[70]

$$\Delta p_s = \frac{f_s G_s^2 D_s'(N_c + 1)}{5.22(10)^{10} D_e s \phi_s}, \text{ psi} \tag{10-218}$$

$$\text{also, } \Delta p_s = \frac{f_s G_s^2 D_s'(N_c + 1)}{2 g \rho D_e \phi_s} \tag{10-219}$$

where f_s = friction factor from Figure 10-140, for plain bare tubes, $f_s = f/1.2$ (from Figure 10-140), shell side
G_s = mass velocity, lb/hr (ft^2 of flow area)
D_e = equivalent diameter of tubes, ft. See Figure 10-54 or Table 10-21.
D_s' = I.D. of shell, ft
N_c = number of baffles
$(N_c + 1)$ = number of times fluid crosses bundle from inlet to outlet
$g = 4.17 \times 10^8$
s = specific gravity of gas or liquid referenced to water
$\phi_s = (\mu/\mu_w)^{0.14}$, subscript w refers to wall condition
μ = viscosity, lb/hr (ft) = (centipoise) (2.42)

For values of specific gravity for noncondensing gases and vapors use the average density at inlet and outlet conditions referenced to water at 62.4 lb/ft^3.

Alternate: Segmental Baffles Pressure Drop

$$\Delta p_s = \Delta p_b + \Delta p_c \tag{10-220}$$

a. Baffle Window Pressure Drop, Δp_b, psi
This drop is usually very small unless the baffle cut has been limited to a low value.[36]

$$\Delta p_b = \frac{2.9(10)^{-13}(G_b)^2(N_c)}{s}, \text{ psi} \tag{10-221}$$

Donohue[36] reports agreement of ± 36% in turbulent flow conditions.

where Δp_s = total shell-side pressure drop, psi
Δp_b = pressure drop across window opening of segmental baffles, total for all baffles, psi
Δp_c = pressure drop across the bundle in cross-flow, psi
s = specific gravity of gas or liquid referenced to water
N_c = number of baffles
G_b = flow rate, lb fluid/(hr) (ft^2 of flow cross-section area through window opening in baffle)

b. Bundle Cross-flow Pressure Drop, Δp_c, psi, Williams[126]

$$\Delta p_c = \left(\frac{c_b f_f (G_c)^2}{10^9 g \rho (\mu/\mu_w)^{0.14}}\right)(n_c)(N_c + 1) \tag{10-222}$$

f_f = (f, from Figure 10-140) (144)
Note: f from Figure 10-140 must be divided by 1.2 when plain bare tubes are used.
c_b = 1.07 for bare tubes
= 1.2 for low finned tubes

As an alternate, the equation of Chilton and Generaux:[28, 82]

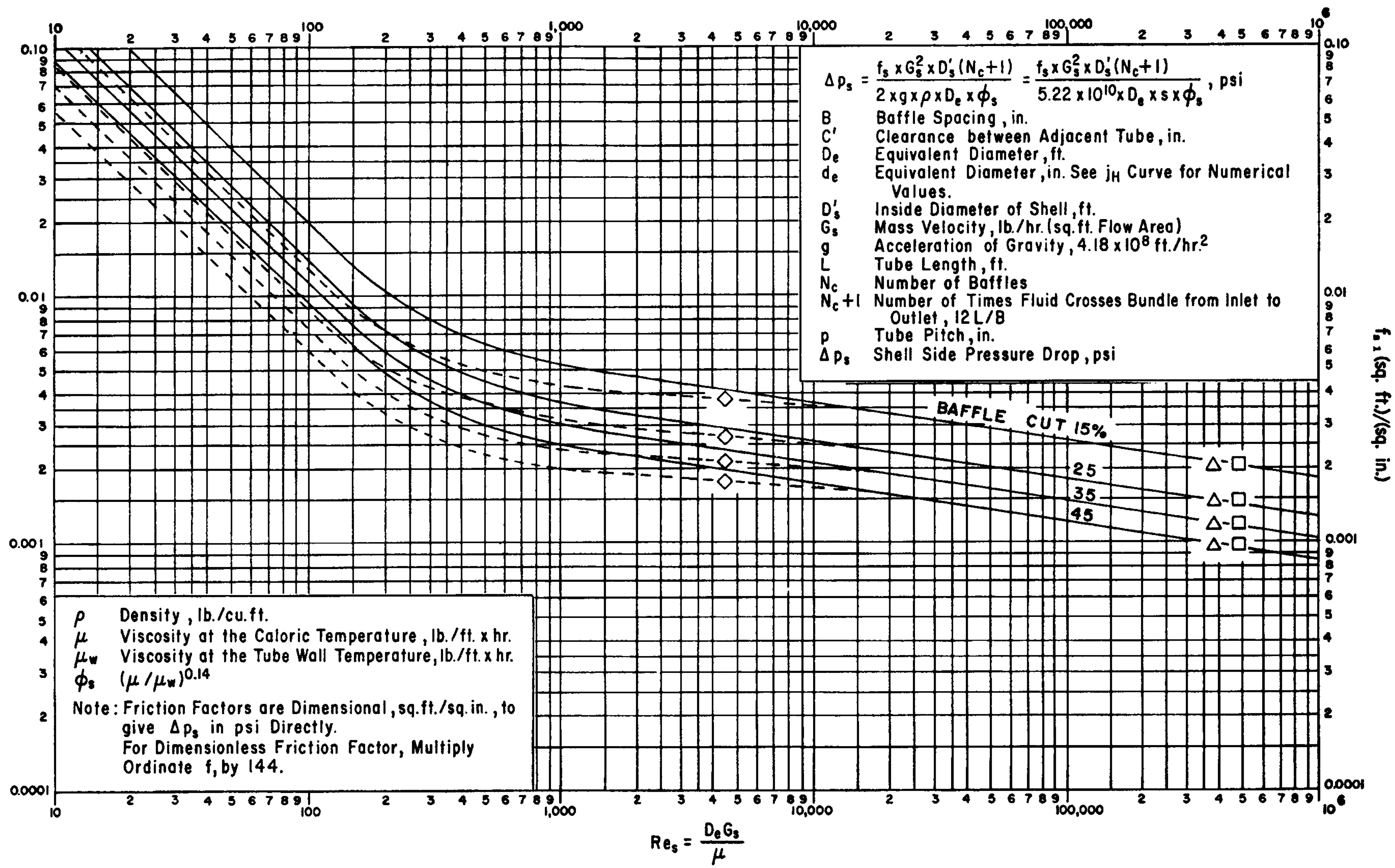


Figure 10-140. Shell-side friction factors for low-finned and plain tubes. (Used by permission: *Engineering Data Book,* © 1960. Wolverine Tube, Inc.)

$$\Delta p_c = 4 f_s'' n_c G_{max}^2/(2g'\rho)(144) \quad (10\text{-}223)$$

For triangular pitch:[57, 58, 82]

r_t from 1.5 to 4.0

$$f_s'' = \left[0.25 + \frac{0.1175}{(r_t - 1)^{1.08}}\right]\left(\frac{D_o G_{max}}{\mu_f'}\right)^{-1.16} \quad (10\text{-}224)$$

For square or in-line pitch:[57, 58, 82]

r_t from 1.5 to 4.0

$$f_s'' = \left(0.044 + \frac{0.08 r_l}{(r_t - 1)^a}\right)\left(\frac{D_o G_{max}}{\mu_f'}\right)^{-0.15} \quad (10\text{-}225)$$

$$a = 0.43 + \frac{1.13}{r_l} \quad (10\text{-}226)$$

where c_b = constant
f_f = dimensionless friction factor for shellside cross-flow
G_c = mass flow, lb/(hr) (ft^2 of cross section at minimum free area in cross-flow)
G_{max} = mass flow, lb/sec (ft^2 of cross section at minimum free area in cross-flow)
ρ = fluid density, lb/ft^3
g' = acceleration constant 32.2 ft/(sec)^2
μ/μ_w = viscosity ratio of fluid at bulk temperature to that at wall temperature
μ_f' = absolute viscosity, lb/sec (ft), μ_f' = (centipoises) (0.000672)
n_c = minimum number of tube rows fluid crosses in flowing from one baffle window to one adjacent.
N_c = number of baffles
Δp_c = bundle cross-flow pressure drop, psi

$$r_t = \frac{\text{Tube pitch, in.}}{\text{Tube O.D., in.}} \quad \text{transverse to fluid flow, dimensionless}$$

$$r_l = \frac{\text{Tube pitch, in.}}{\text{Tube O.D., in.}} \quad \text{longitudinal value in direction of fluid flow, dimensionless}$$

McAdams[82] points out that at r_t of 1.25, the pressure drop may deviate high as much as 50% and is high for $r_t < 1.5$ and > 4.

Streamline flow shell-side cross-flow; modified Donohue:[38]

$$\Delta p_c = 3.02(10)^{-5}\,\frac{(n_c)G_c\,\mu'}{s(p - d_o)},\ \text{psi} \quad (10\text{-}227)$$

s = specific gravity of fluid referenced to water
p = tube pitch, in.
d_o = tube O.D., in.
μ' = viscosity, centipoise, at average temperature

Shell Side Pressure Drop in Condensers

Kern[70] recommends Equation 10-228 as being conservative:

$$\Delta p_s = \frac{9.56(10)^{-12}(f_s)G_s^2 D_s'(N_c + 1)}{D_e's},\ \text{psi} \quad (10\text{-}228)$$

This equation gives values that are half of those calculated as total gas flow for the shell side by using friction factors from Figure 10-140. (Note that f_s for plain or bare tubes = f/1.2 (with f from Figure 10-140)).

The method of Buthod[22] has given unusually good checks with data from industrial units. In general this method appears to give results that are slightly higher than field data but not as high as the other methods presented previously. For shell-side pressure drop:

$$\Delta p_s\,(\text{total}) = \Delta p_{long.} + \Delta p_c \quad (10\text{-}229)$$

1. Calculate loss due to longitudinal flow through tube bundle; use Figure 10-141.

$$G(\text{longitudinal}) = \frac{0.04 W_s}{\frac{\pi}{4}(D_s^2 - N d_o)B_{ca}},\ \text{lbs/sec (ft}^2) \quad (10\text{-}230)$$

where W_s = shell-side flow, lb/hr
D_s = shell I.D., in.
d_o = tube O.D., in.
N = number of tubes in bundle
B_{ca} = baffle cut area, expressed as fraction, representing opening as percent of shell cross-section area.

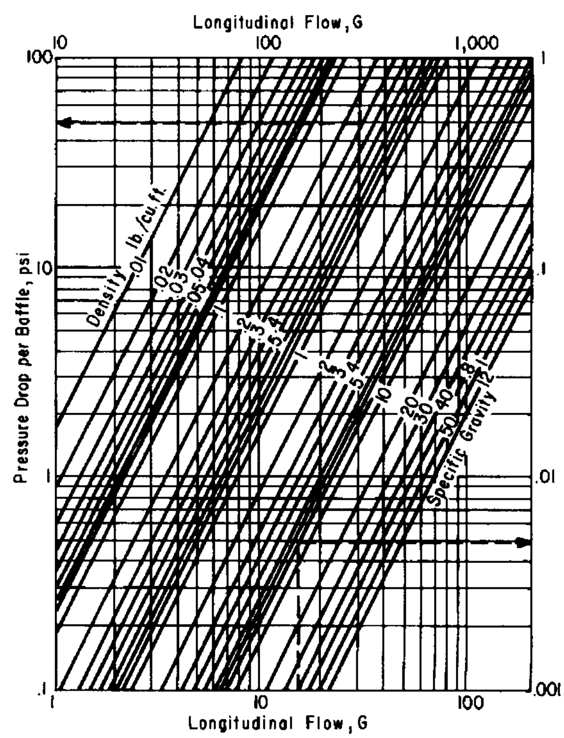


Figure 10-141. Pressure drop in exchanger shell due to longitudinal flow. (Used by permission: Buthod, A. P. *Oil & Gas Journal*, V. 58, No. 3, ©1960. PennWell Publishing Company. All rights reserved.)

From the chart, read Δp_{long} per baffle, as psi. To obtain total longitudinal drop, multiply by the number of baffles.

2. Calculate loss due to cross-flow through the tube bundle; use Figure 10-142.

$$G(\text{cross-flow}) = (0.04\, W)/(B)\,(M),\ \text{lb/sec}\,(\text{ft}^2) \qquad (10\text{-}231)$$

where B = baffle pitch or spacing, in.
M = net free distance (sum) of spaces between tubes from wall to wall at center of shell circle, in.

B is held to a 2-in. minimum or $^1/_5$ shell diameter (I.D.) and is 26 in. maximum for $^3/_4$-in. tubes and 30 in. for 1-in. tubes. Refer to TEMA for tube support and baffle spacing recommendations.

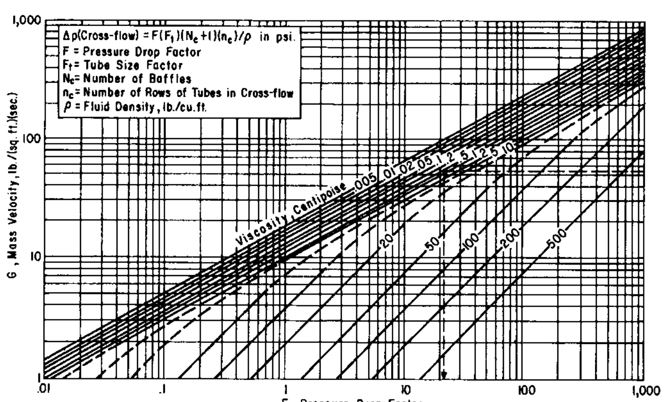
Read pressure drop factor, F_p, from Figure 10-142.

$$\Delta p_c(\text{cross-flow}) = \frac{(F_t)(Fp)(N_c + 1)(n_c)}{\rho},\ \text{psi} \qquad (10\text{-}232)$$

where F_t = tube size factor, from table on Figure 10-142
F_p = pressure drop factor, Figure 10-142
N_c = number of baffles
n_c = number of rows of tubes in cross-flow
ρ = density of fluid, lb/ft^3

The number of tube rows that will be crossed as the fluid flows around the edge of one baffle and then across and over to the next baffle is used as for conventional designs.

$$n_c = 0.9\ (\text{total tube rows in shell at center line})$$



Values of F_t	Tube Size (inches)	Pitch	Viscous Flow	Turbulent Flow
	5/8	13/16 in. Δ	0.008	0.01
	3/4	15/16 in. Δ	0.01	0.01
	3/4	1 in. □	0.0025	0.0042
	3/4	15/16 in. □	0.0044	0.0044
	1	1 1/4 in. Δ	0.0075	0.0095
	1	1 1/4 in. □	0.0033	0.0042

Figure 10-142. Pressure drop in fluid flowing across tube banks with segmental baffles. (Used by permission: Buthod, A. P. *Oil & Gas Journal,* V. 58, No. 3, ©1960. PennWell Publishing Company. All rights reserved.)

Finned Tube Exchangers

The procedures for designing exchangers using the finned tubes are generally specific to the types of fins under consideration. The 16 and 19 fins-per-in. low fin tubes (Figure 10-10A and 10-10B) are uniquely adaptable to the conventional shell and tube exchanger[16, 127] (see Table 10-39) and are the type of tubes considered here.

These low-fin tubes can be installed and handled in the same manner as plain tubes. The larger diameter fins (5 or more per in.) are usually used in services with very low outside coefficients of heat transfer and require a unit design to accommodate the tube's installation.

Other finned tube configurations are shown in Figures 10-10A, 10-10E, 10-10G, and 10-10H and represent increased external finning possibilities. Internal ribs, Figures 10-10K and 10-10M, can certainly help the film transfer coefficient, provided fouling is not a prominent factor. Other finned designs (number of fins/in.) are available from most manufacturers, and in order to use them in heat transfer designs, specific data needs to be available from the manufacturer. The literature cannot adequately cover suitable design data for each style of tube. Pase and O'Donnell[203] present the use of finned titanium in corrosive services. One of the outstanding books by Kern and Kraus[206] covering the entire topic of *Extended Surface Heat Transfer* includes detailed theory and derivations of relations plus practical applied problems for finned and compact heat exchangers.

The longitudinal finned tube usually is adapted to double pipe exchangers but is used in the conventional bundle design with special considerations.

Other finned tube references of interest are Hashizume[208] and Webb.[209]

Low Finned Tubes, 16 and 19 Fins/In.

This tube has a ratio of outside to inside surface of about 3.5 and is useful in exchangers when the outside coefficient is poorer than the inside tube coefficient. The fin efficiency factor, which is determined by fin shape and size, is important to final exchanger sizing. Likewise, the effect of the inside tube fouling factor is important to evaluate carefully. Economically, the outside coefficient should be about $^1/_5$ or less than the inside coefficient to make the finned unit look attractive; however, this break-even point varies with the market and designed-in features of the exchanger.

Process applications are primarily limited to low-finned tubing, although the high-finned tubes fit many process gas designs that require special mechanical details. This test limits the presentation to the low-finned design.

Table 10-39
Approximate Estimating Physical Data for Low-Finned Tubing for Use in Design Calculations

Nominal Size O.D.	Plain Section Dimensions O.D.	Plain Section Dimensions Wall Thk.	Finned Section Root Dia.	Finned Section Wall Thickness	Finned Section I.D.	Finned Section d_e	Outside Area ft^2 per lin ft	Surface Area Ratio a_o/a_i	I.D. Cross Sectional Area, in.^{2}	Approx. wt/ft lb (Copper)
19 Fins Per In.										
$^5/_8$	.625	.042	.500	.028	.444	.535	.405	3.48	.155	.275
		.049		.035	.430			3.60	.145	.316
		.058		.042	.416			3.72	.136	.368
		.065		.049	.402			3.85	.127	.408
		.072		.065	.370			4.18	.108	.444
$^3/_4$	.750	.049	.625	.028	.569	.660	.496	3.33	.254	.344
		.049		.035	.555			3.41	.242	.376
		.058		.042	.541			3.50	.230	.449
		.065		.049	.527			3.60	.218	.490
		.082		.065	.495			3.84	.192	.612
		.095		.083	.459			4.13	.166	.695
$^7/_8$	.875	.054	.750	.035	.680	.785	.588	3.30	.363	.483
		.058		.042	.666			3.37	.349	.530
		.065		.049	.652			3.44	.334	.589
		.082		.065	.620			3.62	.302	.727
		.095		.083	.584			3.85	.268	.829
1	1.000	.058	.875	.042	.791	.910	.678	3.27	.492	.612
		.065		.049	.777			3.33	.474	.680
		.082		.065	.745			3.48	.436	.841
		.095		.083	.709			3.65	.395	.965
16 Fins per in.										
$^5/_8$	.625	.082	.500	.065	.370	.540	.368	3.80	.108	.497
$^3/_4$	.750	.082	.625	.065	.495	.665	.438	3.38	.192	.612
		.095		.083	.459			3.63	.166	.695
$^7/_8$	.875	.082	.750	.065	.620	.790	.520	3.20	.302	.727
		.095		.083	.584			3.40	.268	.829
1	1.000	.082	.875	.065	.745	.917	.598	3.07	.436	.841
		.095		.083	.709			3.22	.395	.965

ALLOY	wt/ft Conversion Factor (wt/ft of Copper × Conv. Factor wt/ft of Alloy)
Copper	1
Admiralty (type C)	.9531
Admiralty (types B & D)	.9531
$^{85}/_{15}$ red brass	.9780
Aluminum brass (type B)	.9319
1100 aluminum	.3032
3003 aluminum	.3065
Nickel	1
$^{70}/_{30}$ cupro-nickel	1
$^{90}/_{10}$ cupro-nickel	1
Monel	1
Low carbon steel	.8761
Stainless steel	.8978

Note: Units are in., except as noted.
Used by permission: *Engineering Data Book*, Section 2, ©1959. Wolverine Tube, Inc.

Finned Surface Heat Transfer

Rohsenow and Hartnett[166] recommend the Briggs and Young[205] convection film coefficient relation for externally finned tubes.

$$\frac{h_{fo}D_r}{k} = 0.134\frac{(D_rG_{max})^{0.681}}{(\mu)}\frac{(c_p\mu)^{1/3}}{(k)}\frac{(s)^{0.2}}{l}\frac{(s)^{0.113}}{t} \quad (10\text{-}233)$$

where h_{fo} = mean outside finned surface heat transfer (usually gas) coefficient, Btu/(hr) (°F) (ft² external)
D_r = root diameter of tube (external), ft
d_n = root diameter of tube, external, in.
k = thermal conductivity of gas, Btu/(hr) (ft²) (°F/ft)
G_{max} = gas mass velocity at minimum cross-section, through a row or tubes normal to flow, lb/(hr) (ft²)
G_m = mass velocity at minimum cross-section through a row of tubes normal to flow, lb/(hr) (ft²)
g_c = acceleration of gravity, 4.18×10^8, ft/(hr) (hr)
n = number of rows in direction of flow
μ = gas/vapor viscosity at bulk temperature, lb/(hr) (ft)
c_p = specific heat, Btu/(lb) (°F)
s = distance between adjacent fins, in.
l = fin height, in.
t = fin thickness, in.
P_t = transverse pitch between adjacent tubes in same row, in.
P_l = longitudinal pitch between adjacent tubes in different rows measured on the diagonal, in.
ΔP = static pressure drop, lb/ft²
ρ = density of gas, lb/ft³
f = mean friction factor, this is the "small" or fanning friction factor. Note: $f = \Delta P\, g_c\, \rho/(n\, G_m^2)$

Pressure drop across finned tubes:[166]

$$\Delta p = 18.93\frac{(G_mD_r)^{-0.316}}{(\mu)}\frac{(P_t)^{-0.927}}{(D_r)}\frac{(P_t)^{0.515}}{(P_l)}\frac{(G_m^2 n)}{(g_c\rho)} \quad (10\text{-}234)$$

The equations provide reasonable estimates per Rohsenow, [166] who suggests using with caution, only when performance on the system is not available. Ganapathy[204] offers simplified equations and nomographs to solve these relations.

Table 10-40 provides a suggested range of overall heat transfer coefficients, U_o, for actual finned heat exchangers.

Economics of Finned Tubes

Figure 10-143 is useful in *roughly* predicting the relative economic picture for adapting low finned tubes to the heat or cooling of oil on the shell side of conventional shell and tube units. This is *not* a design chart.

Figures 10-144 and 10-145[126] also indicate the relative advantage regions for the finned unit, for the average water-cooled exchanger of 150 psi design. For example, for a plain tube with an overall fouling coefficient of 125, inside fouling of 0.0015, and outside fouling of 0.002, the finned tube unit would be more economical. The fouling lines, r, on the charts are the limit border lines of the particular economics, which assumed equal costs for the finned and bare tube exchangers. Again, these are not to be used for specific exchanger design, but merely in deciding the region of applicability.

Table 10-40
Comparison of Calculated, Designed, and Operating U_o Values; $^3/_4$-in., 19 Fins/in. Finned Tubes

Service	Calc'd. U_o	Designed U_o	Operating U_o	Comments
Propane condenser (66°F H_2O)		35	47.4	
Ethylene cross exchanger (liquid to gas)	9.9	9.5	14.8	
Ethylene compressor intercooler (67°F H_2O)	21	18	28.7	
Ethylene compressor aftercooler (67°F H_2O)	21	18.3	16.3	Possibly fouled by oil.
Propane compressor intercooler (67°F H_2O)	21.6	20	23.8	
Propane cross exchanger (liquid to gas)	14.2	8.2	11.6 & 9.1	Lower flow rate than used in calculations.
Gas cooler (67°F H_2O)	17.6	13.3	14.6	Lower heat duty & inlet gas temperature than used in calculations.
Gas heater (400 lb sat'd. steam)	22.7	15	22.5	
Ethylene compressor intercooler (68°F H_2O)	21.0	11.5	13.9	Lower flow rate than used in calculations.
Methane gas-Ethylene liquid cross exchanger	25	20	26.2	U_o drops to 10 after fouling with hydrate ice.
Methane gas-propane liquid cross exchanger	25	17.9	19.7	U_o drops to 13 after fouling with hydrate ice.

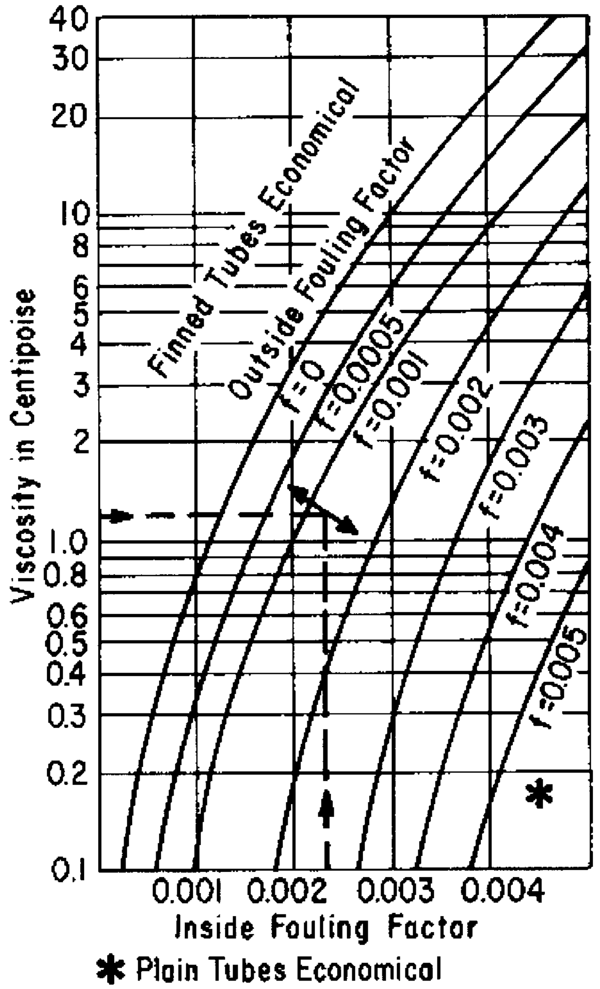


Figure 10-143. Estimating relationship for selection of low-finned units in oil heaters or coolers; for reference only (1950 costs). (Used by permission: Williams, R. B. and Katz, D. L. *Petroleum Refiner,* V. 33, No. 3, ©1954. Gulf Publishing Company. All rights reserved.)

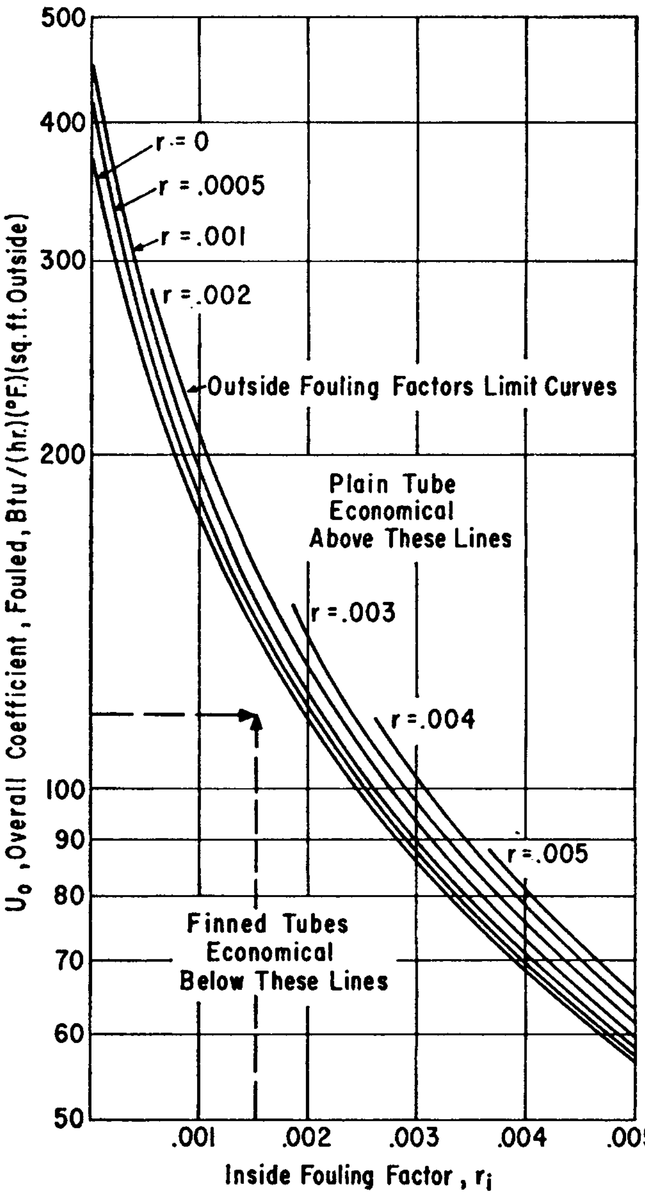


Figure 10-144. Approximate relationship of the overall coefficient fouled, and the fouling factor of inside tubes for predicting the economical use of finned tubes in shell and tube units. (Used by permission: Williams, R. B., and Katz, D. L. "Performance of Finned Tubes and Shell and Tube Heat Exchangers," ©1951. University of Michigan. Note: For reference only, 1950 costs.)

Wolverine[21] has presented evaluations of the cost comparisons for various types of exchangers and tube materials.

Figure 10-146 gives a rough indication as to the possible advantages of a finned tube unit when referenced to a specific design. If the film coefficients and fouling factors based on plain tubes are known, the reduction in the number of finned tubes for the same length and service can be approximated roughly. The significant saving arises when a reduction in shell diameter can be effected, based on the estimated reduction in the number of tubes. The results of this graph should indicate whether a detailed comparison in design is justified; keep in mind that the curve is based on an average set of conditions.

Tubing Dimensions, Table 10-39

For finned tube efficiencies, see Figure 10-147.

The fin efficiency is defined by Kern and Kraus[206] as the "ratio of the actual heat dissipation of a fin to its ideal heat dissipation if the entire fin surface were at the same temperature at its base." Figure 10-147 provides a weighted fin

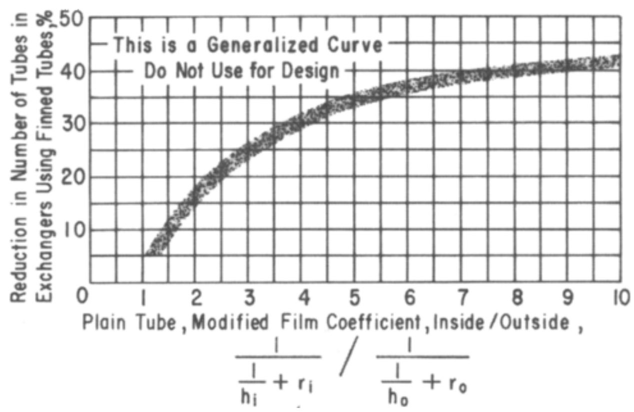


Figure 10-145. Generalized design evaluation of low-finned tubes and fluid heat exchangers. (Used by permission: "An Opportunity." Wolverine Tube, Inc.)

efficiency. Weighted fin efficiency is expressed by Kern and Kraus[206] as:

$$\eta_w = \frac{\eta_f s_f'' + s_o''}{s_f'' + s_o''} \tag{10-235}$$

where s_f'' = fin surface per ft of pipe length
s_o'' = plain pipe surface per ft length
η_w = weighted fin efficiency, fraction, from Figure 10-147
η_f = fin efficiency, fraction = [(tan h) (mb)]/(mb)
tan h = hyperbolic tangent
m = fin performance factor = $[(2h)/k_m\delta_o]^{1/2}$, ft^{-1}
b = fin height, ft
h = heat transfer coefficient, Btu/(hr) (ft^2)(°F)
k_m = metal thermal conductivity, Btu/(ft) (hr)(°F)
δ_o = fin width, ft, at fin base

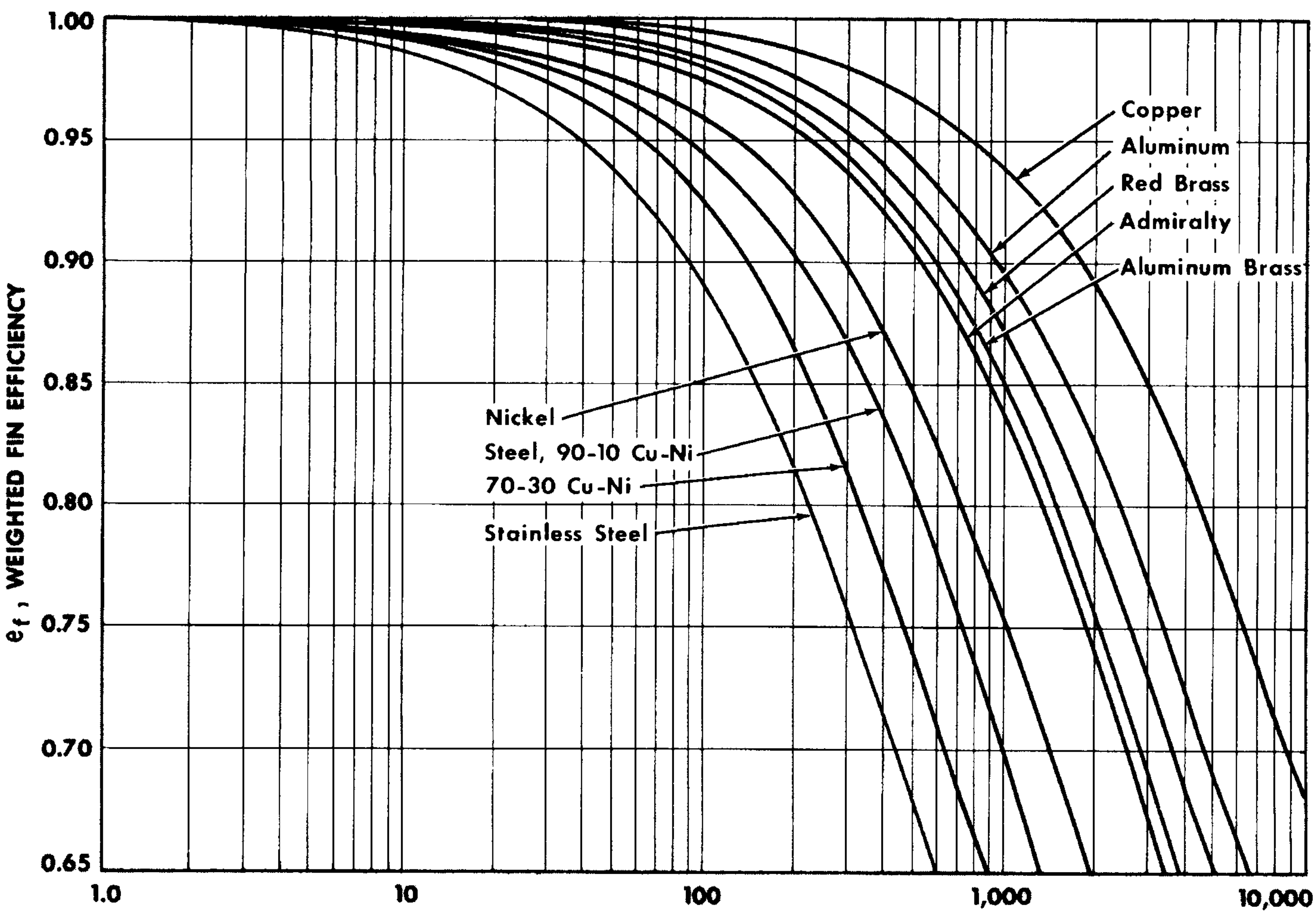


Figure 10-146. Weighted efficiencies of low-finned tubing of 11, 16 and 19 fins per in. length, $^{1}/_{16}$-in. high, radial. (Used by permission: *Engineering Data Book,* 2nd Ed., ©1960. Wolverine Tube, Inc.)

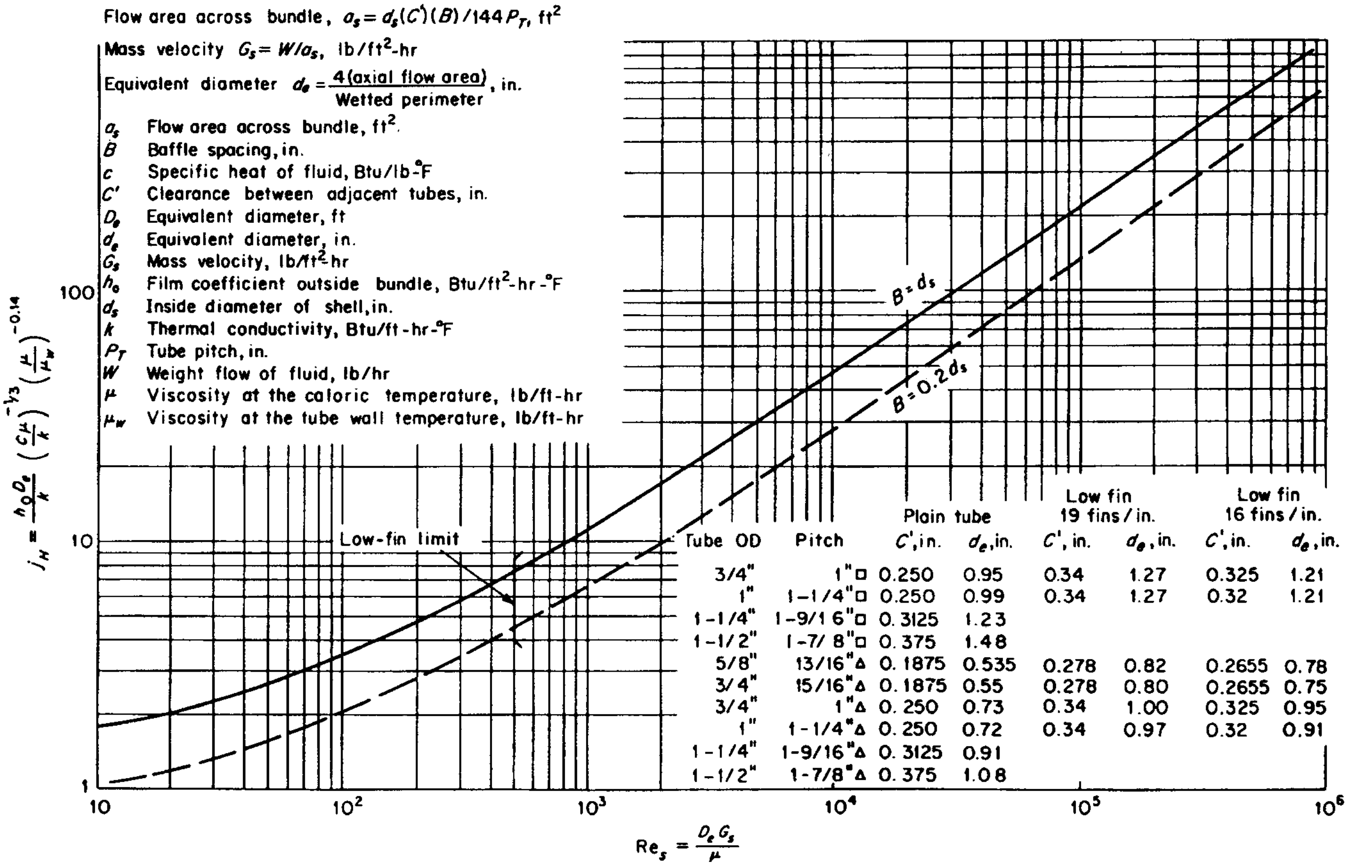


Figure 10-147. Shell-side j_H factors for bundles. One sealing strip per 10 rows of tubes and TEMA clearances. (Source: *Engineering Data Book*, 2nd Ed., ©1960. Wolverine Tube, Inc. Used by permission: Kern, D. Q., and Kraus, A. D. *Extended Surface Heat Transfer*, p. 506, ©1972. McGraw-Hill, Inc. All rights reserved.)

Design for Heat Transfer Coefficients by Forced Convection Using Radial Low-Fin tubes in Heat Exchanger Bundles

Kern and Kraus[206] reference the ASME-University of Delaware Cooperative Research Program on Heat Exchangers by Bell[207] and later work by Bell and Tinker. The Kern[206] recommendation is based on the Delaware work and the TEMA details of construction.

Heat Transfer Coefficient, Shell Side

$$h_o = j_H\,[k/D_e]\,\frac{(c_p\mu)^{1/3}}{k}\,(\mu/\mu_w)^{0.14} \qquad (10\text{-}236)$$

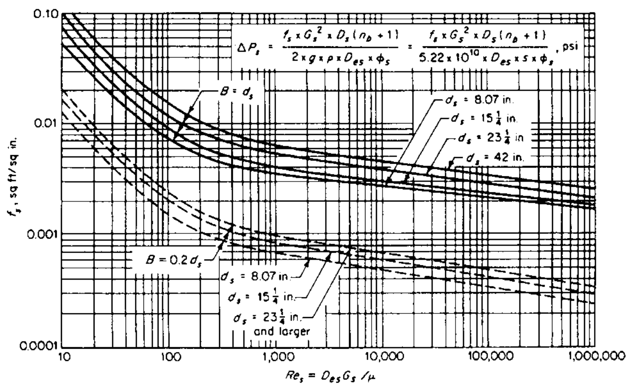
See Figure 10-147.

where D_e = shell-side equivalent diameter outside tubes, ft, see Figure 10-56
c_p = specific heat of shell-side fluid, Btu/(lb-°F)
k = thermal conductivity of fluid, Btu/(ft) (hr)(°F)
μ = viscosity of shell-side fluid (at bulk temperature) lb/(ft) (hr)
μ_w = viscosity of shell-side fluid at tube wall temperature, lb/(ft)(hr)
j_H = heat transfer factor, dimensionless
h_o = heat transfer coefficient for fluid outside tubes based on tube external surface, Btu/(hr) (ft²)(°F)
Re_s = Reynolds Number, shell side, dimensionless
G_s = mass velocity (cross-flow), lb/(hr) (ft²)

The baffle used in the preceding equation has 20% segmental cuts. Shell-side cross-flow velocity:[206]

$$\text{Cross-flow area, } a_s = \frac{d_s C' B}{144p} \qquad (10\text{-}237)$$

where a_s = cross-flow area in a tube bundle, ft²
d_s = shell-side I.D., in.
p = tube pitch, in., see Figures 10-56 and 10-148
C' = clearance between low-fin tubes, $(p - d_e')$, or for plain tubes, $(p - d)$, in., see Figure 10-148.
B = baffle pitch, in.



B = baffle spacing, in.
C' = clearance between adjacent tubes, in.
D_{es} = equivalent diameter, ft
d_{es} = equivalent diameter, in. See j_H curve for numerical values
D_s = shell diameter, ft
d_s = shell diameter, in.
G_s = mass velocity, lb/ft^2-hr
g = acceleration of gravity, 4.18×10^8 ft/hr^2
L = tube length, ft
n_b = number of baffles
$n_b + 1$ = number of times fluid crosses bundle from inlet to outlet
ΔP_s = shell side pressure drop, psi
ρ = density lb/ft^3
μ = viscosity at the caloric temperature, lb/ft-hr
μ_w = viscosity at the tube-wall temperature, lb/ft-hr
$\phi_s = (\mu/\mu_w)^{0.14}$

Note: Friction factors are dimensional, sq ft/sq in. to give ΔP_s in psi directly

Figure 10-148. Shell-side friction factors for bundles with 20%-cut segmental baffles, one seal strip per 10 rows of tubes, and TEMA clearances. These factors can be used for plain or low-finned tubes with the appropriate values of D_{es} or d_{es}. (Source: *Engineering Data Book*, ©1960. Wolverine Tube, Inc. Used by permission: Kern, D. Q., and Kraus, A. D. *External Surface Heat Transfer*, p. 511, ©1972. McGraw-Hill Book Co., Inc. All rights reserved.)

Figure 10-147 allows for the correction for the by-pass area between the outer tube limit of the bundle and the shell I.D., or as an alternative, see Figure 10-54.

Referring to Figure 10-147, the marking "low-fin limit"[206] at Re = 500 is explained by Kern;[206] because the low-fin tube is somewhat more inclined to insulating itself with liquids of high viscosity, when a low shell-side Re number is the result of a high mass velocity and high viscosity as compared to a low mass velocity at low viscosity, caution is suggested.[206]

Pressure Drop in Exchanger Shells Using Bundles of Low-Fin Tubes

The Delaware[207] work is considered[206] the most comprehensive (up to its date of preparation), taking into account the individual detailed components that make up the flow and pressure loss components of a total exchanger operation.

Figure 10-148 presents a recommended pressure drop correlation[206] for low-fin tubes in shells and is based on clean tube pressure drop with no dirt sealing the leakage clearances between tubes and baffle holes or baffle-to-shell clearances. A fouled condition pressure drop may be an indeterminate amount greater. The authors[206] state that this University of Delaware correlation has some factors built in that limit the deviations to a relatively small range. Figure 10-148 has allowances built in for entrance and exit losses to the shell and leakage at baffles.[206] The suggested pressure drop for shell-side heating or cooling, including entrance and exit losses is

$$\Delta P_s = \frac{f\, G_c^2\, D_s (n_b + 1)}{(5.22 \times 10^{10})(D_e\, s\, \phi_s)}, \text{ psi} \tag{10-238}$$

where f = friction factor, dimensional, ft^2/ in.^{2}
P_s = shell-side pressure drop, psi
f = friction factor, ft^2/ in.^{2}
G_c = cross-flow mass velocity, lb/(ft^2) (hr)
D_s = shell I.D., ft
n_b = number of baffles
D_e = D_{es} = equivalent O.D. of tubes, ft, see earlier discussion on this topic.
d_e = d_{es} = equivalent O.D. of tubes, in., see Figures 10-147 or 10-148 for numerical values.
s = specific gravity, dimensionless
ΔP_s = pressure drop of fluid, heated or cooled, including entrance and exit losses, lb/in.^{2}
ϕ_s = viscosity correction = (μ/μ_w), dimensionless
μ_w = viscosity of fluid at wall of tube, lb/(ft-hr)
μ = viscosity of fluid in bulk at caloric temperature, lb/(ft-hr)
ρ = fluid density, lb/ft^3
d_s = shell diameter, in.
B = baffle spacing, in.
Re_s = shell-side Reynolds Number

Note that this figure can be used for plain or low-fin tubes when the appropriate value of D_e is used.[206]

Tube-Side Heat Transfer and Pressure Drop

Because finned tubes of the low-fin design are standard tubes, the inside heat exchange and pressure drop performance will be the same as determined for "plain" or "bare" tubes. Use the appropriate information from earlier design sections.

Design Procedure for Shell-Side Condensers and Shell-Side Condensation with Gas Cooling of Condensables, Fluid-Fluid Convection Heat Exchange

Follow the procedures outlined for bare tube equipment, substituting the characteristics of finned tubes where appropriate. The presentation of Wolverine[41] recommends this technique over previous methods.[16] The methods of reference 16 have proven acceptable in a wide number of petrochemical hydrocarbon systems. Figure 10-150 is an example unit in summary form.

Vertical Condensation on Low Fin Tubes

Follow the same procedure as for horizontal tubes but multiply outside film coefficient, h_o, by a factor of 0.7 and try for balance as previously outlined.

DWG. NO. A ______

Item No. ______

Job No. ______

Charge No. ______

EXCHANGER RATING

By ______

Date:- 1-9-59

Apparatus: Inter-cooler for Compressor C-59 Plant ______

Min. Req. Eff. O. S. Area Exposed In Shell, Sq. Ft. ______ Outside 1003 Inside: —

Number of Units: Operating One Spares None

DESIGN DATA PER Unit

UNIT DATA		SHELL SIDE	TUBE SIDE
Fluid		Air with water vapor	Brackish bay water
Fluid Flow	Lbs./Hr.	36,000	118,000 (236 gpm)
Temperature In	°F.	290	90
Temperature Out	°F.	110	105
Operating Pressure	PSIG	180	50
Density at 200°F	Lbs./CF	0.785	62.3
Specific Heat	Btu/Lb./°F.	0.25	1.0
Latent Heat	Btu/Lb.		
Therm. Cond.	Btu/Hr./Sq. Ft./°F./Ft.	0.0183	
Viscosity	Centipoise	0.021	
Molecular Weight		28.8	
No. of Passes		1	4
Pressure Drop	PSI	Calc: 2.53 Used: 3.0	Calc: 8.19 Used: 9.0
Fouling Factor		0.001	0.0015

Heat Transferred - BTU/Hr: 1,771,000

LMTD 69.1 °F. Overall U: Calc. 26.4 Used 25.5

CONSTRUCTION

	Shell side	Tube side
Max. Oper. Pressure	250 PSI	65 PSI
Max. Oper. Temperature	315 °F.	120 °F.

Type of Unit Fixed Tube Sheet Tube Pitch 1" Joint Roll and Flare

*Tubes - Material: 90-10 Cupro-Nickel No. (Approx.) 264 O.D. 3/4" BWG 16 Length 8'-0"

Shell - Material: Steel Diameter (Approx.): 21.25"

Channel Material: Steel, monel clad Supports Material: Admiralty or Brass

Tube Sheet Material: Monel-clad steel, 1/4" Min. Clad Baffle Material: Admiralty or Brass

Corrosion Allowance - Shell Side 1/16 Tube Side 1/16" on any exposed steel

Connections - Shell In: 10" Out 10" Flange 300#ASA Raised Face

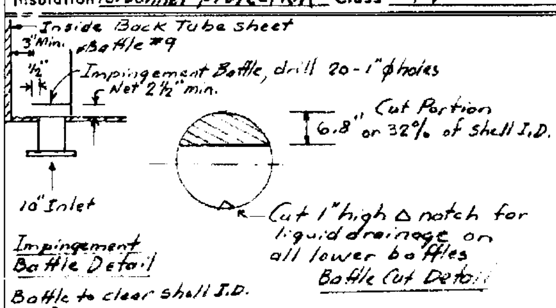
Channel In: 4" Out 4" Flange 150#ASA Raised Face

Others Vents and drains Size 3/4" Flange 3000 Coupling

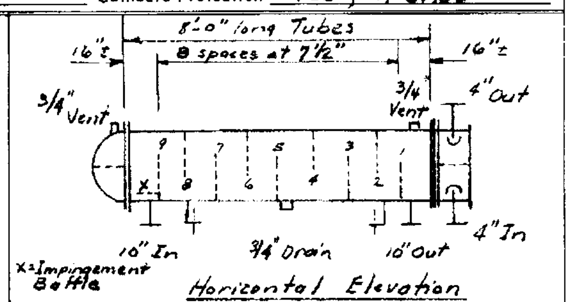
Bolts: Alloy steel Gaskets: Impregnated asbestos

Code TEMA - Class C Stamp Yes X-Ray — SR —

Insulation Personnel protection Class PP Cathodic Protection Yes, in ends



BAFFLE ARRANGEMENT



NOZZLE ARRANGEMENT

Remarks:- * Low fin tubes, 19 fins/inch.

Checked ______ Date ______ Approved ______ Date ______

Rev. ______ By ______ Date ______ B/M No. ______

Figure 10-149. Exchanger rating for intercooler, using low-fin tubes. Note: specifications here are for illustration purposes. The design as developed represents more conservative surface area than substantiated by current data.

Nucleate Boiling Outside Horizontal or Vertical Tubes

Nucleate boiling is boiling at the tube surface at a temperature difference between outside tube surface temperature and the fluid body, less than the critical temperature difference. At and beyond the critical temperature difference, metastable and film boiling take place. These produce lower transfer coefficients as the temperature difference increases.

The nucleate region is the one of interest in most plant design as previously described for plain tube boiling. The critical temperature difference curves have been determined experimentally for a reasonable number of fluids and should be used whenever possible.

Design Procedure for Boiling, Using Experimental Data

The method suggested by Katz, *et al.*,[69] is logical when using experimental data:

1. Determine the heat duty by the usual procedures and define the boiling temperature on the shell side.
2. Determine the arithmetic average of tube side temperature, t_a.
3. Determine overall temperature drop, Δt_o, from average tube side temperature, t_a, to shell-side boiling temperature, t_s.

$$\Delta t_{oa} = t_a - t_s \tag{10-239}$$

4. Calculate the tube-side film coefficient for finned tube, h_t. If water, use Figure 10-50A or 10-50B; if other fluid, use Equation 10-44 or 10-47. Use an assumed or process determined tube-side velocity or other "film fixing" characteristic.
5. Assume fouling resistances for shell side and tube side.
6. Calculate the overall resistance, less shell-side film resistance:

$$B = R_t - \frac{1}{h_b} = \frac{A_o}{A_i h_t} + \frac{A_o}{A_i} r_i + r_o \tag{10-240}$$

Note: Subscripts s and t represent shell and tube side, respectively; b and i represent boiling outside and inside tube.

7. *Assume* a temperature drop across the boiling film, Δt_b. Neglect tube wall resistance.
8. From Figure 10-150, read an expected film coefficient, h_b, at the value of Δt_b. The shell-side boiling temperature should be within 15–20°F of the values given on the graph in order to be reasonably close. Calculate $1/h_b$ from the chart.
9. Substitute the graph value of h_b in

$$R_t - 1/h_b = B \tag{10-241}$$

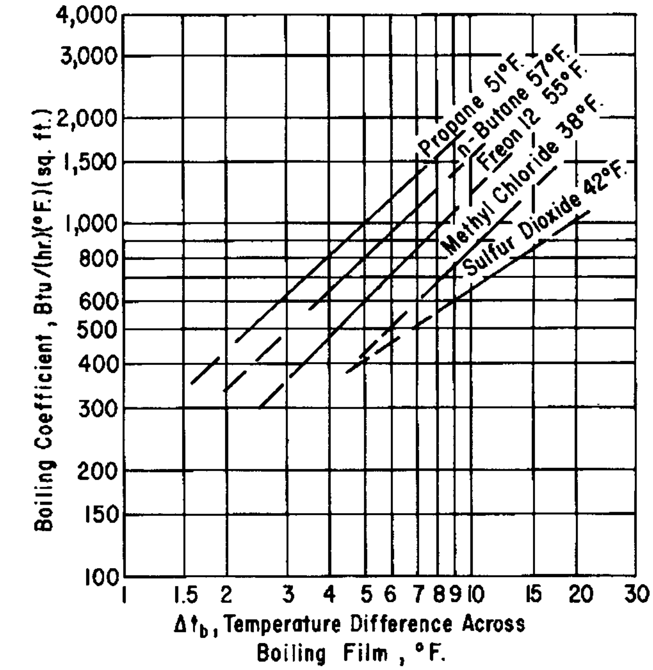


Figure 10-150. Boiling coefficients for low-finned tubes. (Used by permission: Katz, D. L., Meyers, J. E., Young, E. H., and Balekjian, G. *Petroleum Refiner*, V. 34, No. 2, ©1955. Gulf Publishing Company. All rights reserved.)

Then, R_t

$$R_t = B + 1/h_b$$

10. Calculate Δt_b:

$$\Delta t_b = \Delta t_o \left(\frac{1}{h_b}\right)\left(\frac{1}{R_t}\right) = \Delta t_o \left(\frac{r_b}{R_t}\right) \tag{10-242}$$

If the calculated value agrees with the assumed value, proceed to finish the design; if not, reassume the Δt_b in Step 7 and repeat until an acceptable check is obtained.

11. Calculate the overall coefficient, U_o.

$$U_o = \frac{1}{R_t} \text{ Btu/hr (ft}^2\text{)(°F)} \tag{10-243}$$

12. Determine the length of tubing required:

$$U_L = U_c (A_o), \text{ Btu/hr (lin ft)(°F)} \tag{10-244}$$

$$Q = U_L L_t \Delta t_{oa}, \text{ Btu/hr} \tag{10-245}$$

$$L_t = \frac{Q}{U_L \Delta t_o}, \text{ f (total for exchanger)} \tag{10-246}$$

13. Check to determine that the maximum flux, Q/A, and critical temperature difference, Δt_o, are not exceeded,

or even approached too closely. This is to avoid a film boiling condition, rather than nucleate boiling.

14. Determine the number of tubes:

$$\text{No. tubes} = L_T/l \quad (10\text{-}247)$$

where l = assumed length of tube, ft.

Remember to keep a standard length if possible and maintain a tube-side pass condition to realize the film conditions established in Step 4. U-tubes are a good selection for this type of service, and a kettle-type shell is usually used.

15. Determine the pressure drops in the usual manner.

In general, at low boiling temperature film drops, the finned tubes give considerably higher coefficients than plain tubes, but in the general region of a 10–12°F boiling film temperature difference, the two tubes become about the same.

where l = assume length of one tube, ft
L_t = total tube length, ft
L_f = total finned tube length, ft
L_p = total plain tube length, ft
R_t = total resistance to heat transfer, (hr) (°F) (ft²)/Btu
r_b = r_o = outside (tube) fouling factor, (hr) (°F) (ft²)/Btu
r_i = inside (tube) fouling factor, (hr) (°F) (ft²)/Btu
A_o = outside tube surface area, ft²/ft
A_i = inside tube surface area, ft²/ft
Δt_o = overall Δt between average tube-side bulk temperature, °F, and evaporating (boiling) side fluid
Δt_b = temperature drop across boiling film, °F
U_o = overall coefficient of heat transfer, Btu/(hr) (ft²) (°F)
U_L = overall coefficient of heat transfer per ft of tube length, Btu/(hr) (ft of tubing) (°F)
h_b = boiling film coefficient, Btu/(hr) (ft²) (°F)
$h_t = h_w$ = inside water film coefficient, Btu/(hr) (ft²) (°F)
A_o = outside tube surface area, ft²/ft
Q = total heat duty, Btu/hr

Example 10-23. Boiling with Finned Tubes

See Figure 10-151.

A direct evaporation water chiller is to use Freon 12 on the shell side, cooling 187 gpm of water for a closed system from 82°F to 40°F. Because the Freon is to come from an already existing system, operating at 30°F evaporator temperature, this same condition will be used to avoid compressor suction pressure problems. Note: Care must be given to avoid water freezing on tubes. Keep the evaporating temperature slightly above the freezing point of fluid.

Tubes are copper 1-in. nominal O.D. × 14 BWG (0.083-in. thick at finned section) × 19 fins/in. Wolverine Trufin® (standard tube (unfinned) wall thickness = 0.095 in.). Finned surface area/ft length = 0.678 ft²/ft. Plain tubes are 0.5463 ft²/ft.

1. Heat duty:

$$Q = (187)(8.33)(60)(82 - 40) = 3{,}935{,}000 \text{ Btu/hr}$$

Shell-side boiling temperature = 30°F

2. Arithmetic average tube-side temperature,

$$t_a = \frac{82 + 40}{2} = 61°F$$

3. Overall shell temperature drop:

$$\Delta t_o = t_a - t_s = 61 - 30 = 31°F$$

4. Tube-side film coefficient. Assume minimum water velocity of 5 ft/sec, using 1-in. × 14 BWG tubes.

From Figures 10-50A and 10-50B, h_i = 1,215 Btu/hr (ft²) (°F)
Correction = 0.925

In terms of outside surface:

$$h_{io} = h_t = 1{,}215(0.925)\frac{1}{3.66} = 310 \text{ Btu/hr (ft}^2\text{)(°F)}$$

5. Assumed fouling resistances:

Tube side = 0.002
Shell side = 0.001

6. $$B = R_{oa} - \frac{1}{h_s} = \frac{1}{310} + (3.66)(0.002) + 0.001 = 0.01144$$

7. Assume temperature drop across film, Δt_b = 5°F.
8. From Figure 10-150, h_s = 620 Btu/hr (ft²) (°F)
9. R_{oa} = 0.01144 + 1/620 = 0.01144 + 0.00162 = 0.01306
10. Calculate,

$$\Delta t_b = 31\left(\frac{1/620}{0.01306}\right) = 3.95°F$$

Not a check.

7a. Reassume: Δt_b = 4.5°F.
8a. h_s = 550
9a. $R_{oa} = 0.01144 + {}^1/_{550} = 0.01326$
10a. Calculated,

$$\Delta t_b = 31\left(\frac{1/550}{0.01326}\right) = 4.25°F$$

This is close enough.

11. Overall coefficient:

$$U_o = \frac{1}{R_{oa}} = \frac{1}{0.01326} = 75.5 \text{ Btu/hr (ft}^2\text{)(°F)(outside)}$$

DWG. NO. A ____

Item No. E - 1

By ____ **EXCHANGER RATING** Job No. ____

Date:- ____ Charge No. ____

Apparatus ____ Plant ____

Min. Req. Eff. O. S. Area Exposed in Shell, Sq. Ft. ____ Outside 1950 finned Inside: ____

Number of Units: Operating One Spares None

DESIGN DATA PER Unit		
UNIT DATA	SHELL SIDE	TUBE SIDE
Fluid	Freon - 12	Chilled Water
Fluid Flow Lbs./Hr.	75,400	93,500 (187 GPM)
Temperature In °F.	30	82
Temperature Out °F.	30	40
Operating Pressure PSIG	28.5	85
Density Lbs./CF	81.4 (Liquid); 1.065 (Vapor)	
Specific Heat Btu/Lb./°F.		
Latent Heat Btu/Lb.	53.15	
Therm. Cond. Btu/Hr./Sq. Ft./°F./Ft.		
Viscosity Centipoise		
Molecular Weight		
No. of Passes	1	6
Pressure Drop PSI	Calc: None Used: 0.1	Calc: 1.6 Used: 3
Fouling Factor	0.001 (max. expected)	0.002 (conservative)

Heat Transferred - BTU/Hr: 3,935,000

~~LMTD~~ Δt = 31 °F. Overall U: Calc. 75.5 Used 65

CONSTRUCTION			
Max. Oper. Pressure 50	Design Mechanical: 150 PSI	85	Design: 100 PSI
Max. Oper. Temperature 80	Design Mechanical: 200 °F.	100	Design: 150 °F.

Type of Unit Fixed Tube Sheet Tube Pitch 1¼" Triangular Joint Roll and Flare

Tubes - Material: * Steel ASTM A-179 No. (Approx.) 226 O.D. 1" BWG 14 Length 12'-0"

Shell - Material: Steel ASTM A-185 Gr. C. Diameter (Approx.): 31" I.D.

Channel Material: Steel, A-185 or equal Supports Material: Steel

Tube Sheet Material: Steel, A-185 or equal Baffle Material: Steel

Corrosion Allowance - Shell Side None Tube Side 1/16" on ends

Connections - Shell In: 3" Out 12" Flange 150# R.F.

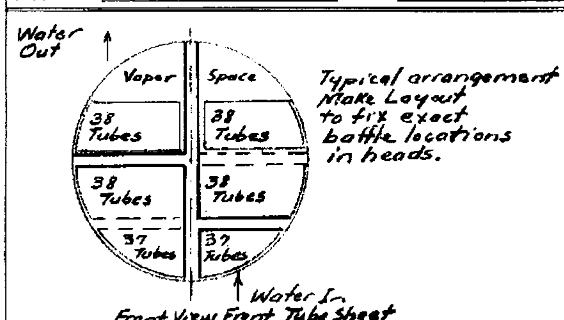
Channel In: 4" Out 4" Flange 150# R.F.

Others See sketch Size ____ Flange 150# R.F & 3000# Couplings

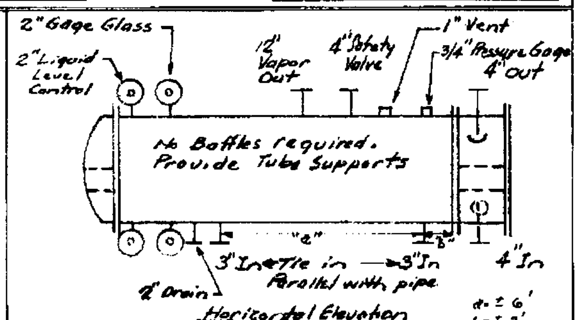
Bolts: Alloy Gaskets: Asbestos

Code TEMA - C Stamp Yes X-Ray No SR No

Insulation Yes Class "REF" for 40°F Cathodic Protection No



BAFFLE ARRANGEMENT



NOZZLE ARRANGEMENT

Remarks:- * Tubes: Finned 19 Fins/Inch, ends plain for rolling, Wolverine Trufin S/T (or equal)

Checked ____ Date ____ Approved ____ Date ____

Rev. ____ By ____ Date ____ B/M No. ____

Figure 10-151. Exchanger rating for refrigerant vaporizer–water chiller.

Total tubing length: $A_o = 0.678\ ft^2/ft$
$U_L = U_o(A_o$, finned tubes) = (75.5) (0.688, updated Table 10-39)
= 51.9 Btu/(hr) (°F) (ft^2) tubing)

$$L_t = \frac{Q}{U_L(\Delta t_o)} = \frac{3{,}935{,}000}{(51.9)(31)} = 2{,}445 \text{ ft finned tubing}$$

Water flow = (187 gpm) (8.34 lb/gal) = 1559.6 lb/min

12. Number of tubes to give 5 ft/sec water velocity

$$= \frac{(187 \text{ gpm})(0.00288)}{(0.519 \text{ in}^2/\text{tube I.D.})(1/144 \text{ in}^2/\text{ft}^2)(3.9 \text{ ft/sec})}$$
= 30.3 tubes/pass, use 30 tubes

Number of 12-ft tubes required, total = 2,445/12
= 203.9, use 204 tubes
Number of tube passes = 203/30 = 6.76 passes, use 6 passes

$$L_T = \frac{Q}{U_L \Delta t_o} = \frac{3{,}935{,}000}{(51.2)(31)} = 2445 \text{ ft}$$

13. Number of tubes:

$$\text{Tube velocity} = 5 = \frac{(187)(0.813)}{\text{No. tubes/pass}}$$

No. tubes/pass = 30.4, say 30
Number of 12-ft tubes, total = 2,470/12 = 206
Number tube passes = 206/30 = 6.85, use 6 passes

14. For 6 passes:

Use 226 tubes on 1 $^1/_4$-in. triangular pitch
Shell I.D. = 25 in. (Table 10-9)
No. tubes/pass = 226/6 = 37.7
Use: Passes of 38-38-38-38-37-37 tubes
Use shell diameter larger in order to have vapor disengaging space; a diameter of 29 in. or 31 in. I.D. will be satisfactory—the latter being the better choice.

Outside surface area, net (finned):

$$= (226)(0.688)(12 \text{ ft} - 4 \text{ in.}/12) = 1{,}814 \text{ ft}^2$$

$$\text{Actual U} = \frac{3{,}935{,}000}{(31)(1{,}814)} = 69.9 \text{ Btu/hr (ft}^2)(°\text{F})$$

Nozzles for water inlet and outlet:

Flow = 187 gpm
Use 4-in. nozzle, velocity = 4.79 ft/sec
(Cameron Table, Fluid Flow Chapter)
Liquid Freon-12 in:
80°F and 84 psig (from condenser)
Enthalpy of liquid = 26.28 Btu/lb
At 30°F and 28.46 psig evaporation condition
Enthalpy of the vapor = 81.61 Btu/lb
Enthalpy change = 81.61 − 28.46 = 53.15 Btu/lb
Freon flow = 3,935,000/53.15 = 74,035 lb/hr
At 80°F and 84 psig, liquid = 0.0123 ft^3/lb

$$\text{gpm} = \frac{74{,}035(0.0123)(7.48)}{(60)} = 113.5$$

Design gpm = (113.5) (1.25) = 141.8
Use 3-in. nozzle, velocity = 6.2 ft/sec

Vapor Freon-12 Out:

From Figure 10-63, max. allowable vapor
Velocity = (28) (1.3) = 36 ft/sec
To reduce entrainment, use 25–30 ft/sec
At 30°F, vapor = 0.939 ft^3/lb

$$\text{Total flow} = \frac{(74{,}035)(0.939)}{(3{,}600)} = 19.31 \text{ ft}^3/\text{sec}$$

For 12-in. nozzle, area of cross-section = 0.777 ft^2
Capacity at 25-ft/sec = (0.777) (25) = 19.5 ft^3/sec

This nozzle O.K.

Double Pipe Finned Tube Heat Exchangers

To properly rate and design this type of unit, the process data should be submitted to the manufacturer, because adequate published correlation literature is not available.

Figures 10-4A, 10-4B, 10-4C, and 10-4D illustrate the usual construction of finned-tube heat exchangers with the fins running parallel to the length of the tube. These are usually, but not always, installed with a tube or pipe outer shell. Typical fins are shown in Figure 10-152. Tube may be fabricated with fins attached by resistance welding rather than imbedding in the tube as shown in Figure 10-152B. The I.D. of the internal finned pipe usually ranges from $^3/_4$ –1 $^1/_2$ in., and the outside surrounding pipe shell can be 2 $^1/_2$ in., 3 in., and 3 $^1/_2$ in. nominal standard pipe size. The number of fins range from 18 for the $^3/_4$-in. pipe, 24 or 32 for the 1 $^1/_2$-in. pipe, and 16 or 32 for the 1 $^1/_2$-in. pipe with $^1/_2$-in. fin height, per manufacturer Griscom-Russell/Ecolaire Corp. The fins of Figure 10-152B are imbedded longitudinally in groves "plowed" into the tube's outer surface. The displaced metal is squeezed back against the imbedded fin base to form a tight metal–metal bond. This bond is not affected by changes in temperature.

Except for fluid conditions of possible galvanic corrosion, the fins can be any selected material, not necessarily the same as the tube. Some usable fin and/or tube materials are steel, aluminum, aluminum bronze, stainless steel, admiralty, copper, copper-nickel, monel, and chrome moly alloy.

This longitudinal fin style unit can be used in cross-exchange, kettle-type reboilers, chillers, and condensers. The rating/design of longitudinal finned tubes is presented by Brown Fintube Co. in an unnumbered bulletin, reference 211. The double pipe finned tube, Figure 10-4A, is often applicable for gas, viscous liquids, or small volumes, and the economics favor high operating pressure due to the small diameter shell.[211] They operate well in dirty or somewhat fouling conditions due to the ease of cleaning. Units can be fabricated with more than one finned tube in a larger shell. The fins are more effective or beneficial when the fin-side film coefficient is lower than the inside tube coefficient; therefore, the poorest heat transfer fluid conditions are best used on the finned side of the tube.

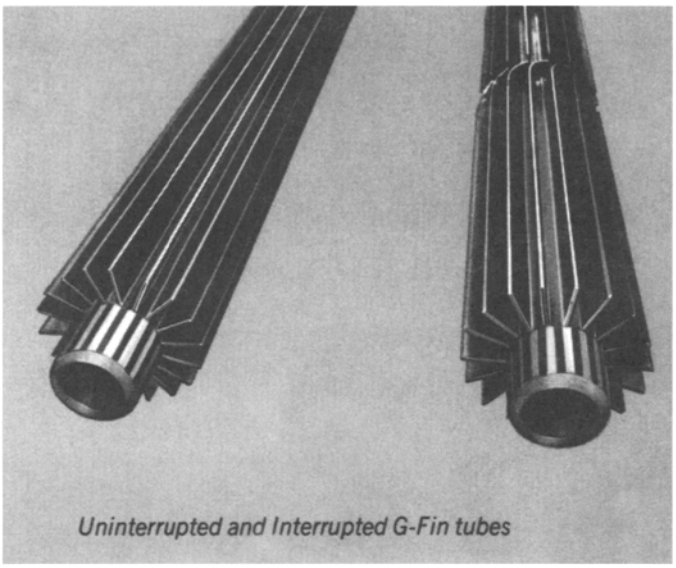


Figure 10-152A. Typical longitudinal finned tubes. Uninterrupted and Interrupted G-Fin® Tubes. (Used by permission: Griscom-Russell, Ecolaire Corp.) (Also see Figure 10-4A(3).)

Finned Side Heat Transfer

For a double pipe exchanger (one finned tube in each of two shells), see Figure 10-4A, the heat flow resistances are[211]

a. Film resistances on outside of the tube, h_o.
b. Metal tube wall resistance, R_m
c. Film resistance on inside of tube, h_i.
d. Note that fouling resistance on tube finned side and inside tube must be added.

$$1/U_o = 1/h_o + 1/h_i + R_m \tag{10-248}$$

where U_o, h_o, h_i = Btu/(hr) (ft^2)(°F)
R_m = (hr)(ft^2) (°F)/BTU

e. See Table 10-41 for suggested overall U coefficients and Table 10-42 for mechanical data.

Figure 10-46 gives the usual Sieder-Tate chart and equation for tube-side, bare-tube heat transfer. For the finned shell-side heat transfer, see Figures 10-153A, 10-153B, 10-153C[211] or the recommendation of Kern and Kraus,[206] Figure 10-154.

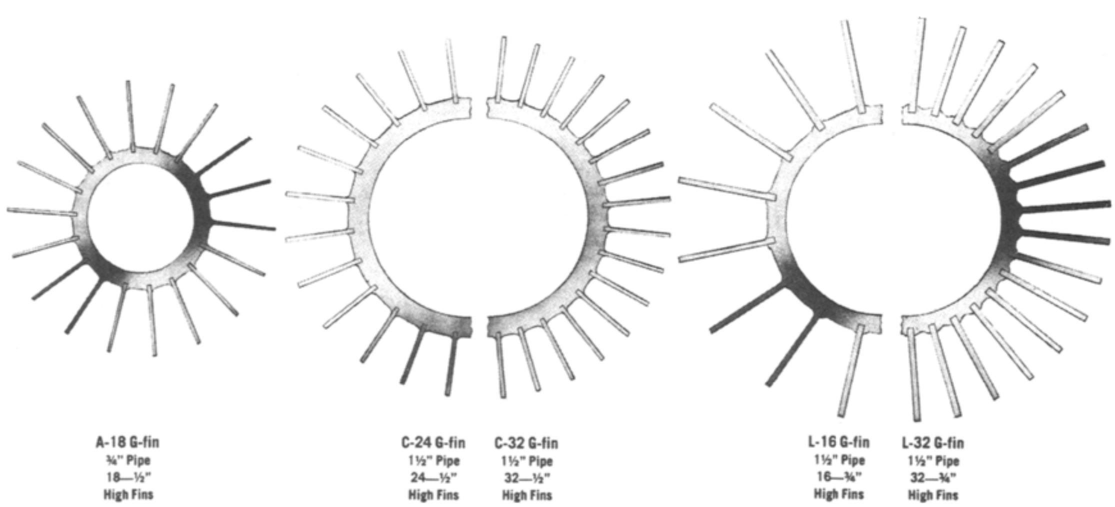


Figure 10-152B. Typical longitudinal finned tubes. Relative pipe sizes and number of longitudinal fins. (Used by permission: Griscom-Russell, Ecolaire Corp.) (Also see Figure 10-4A(3).)

Table 10-41
Estimating Overall Heat Transfer Rates, U_o, for Longitudinal Finned Heat Exchangers

With water for cooling or steam for heating, these are estimated values for preliminary study only.

Process		Estimated Overall Rates "U_o"
Heating viscous materials		
Double pipe—cut & twist fins		12
Multitube bare tubes		15
Medium HC viscosity 3–15 cp avg.		
Heating—double pipe w/fins		15
Multitube bare tube		25
Cooling—double pipe w/fins		12
Multitube bare tube		20
Light HC viscosity < 1 cp		
Double pipe w/fins		25
Multitube w/fins		40
Multitube bare tube		75
Condensing & vaporizing—bare tube		150
Very light HC—bare tubes		150
Gases		
0 psig w/$^1/_2$ psi ΔP	bare tube	25
100 psig w/1 psi ΔP	fin tube	15
Water to water—bare tubes		200
Glycol to glycol		
Double pipe w/fins		10
Multitube bare tube w/turbulators		30

Many factors affect heat transfer rates for example velocity, tube wall temperature and pressure drop. These rates listed do not represent the limit, but are suggested values for study and estimating.

Used by permission: Bul. "Application and Design Estimating of Double Pipe and Hairpin Exchangers." © Brown Fintube Co., A Koch® Engineering Co., Houston, Texas.

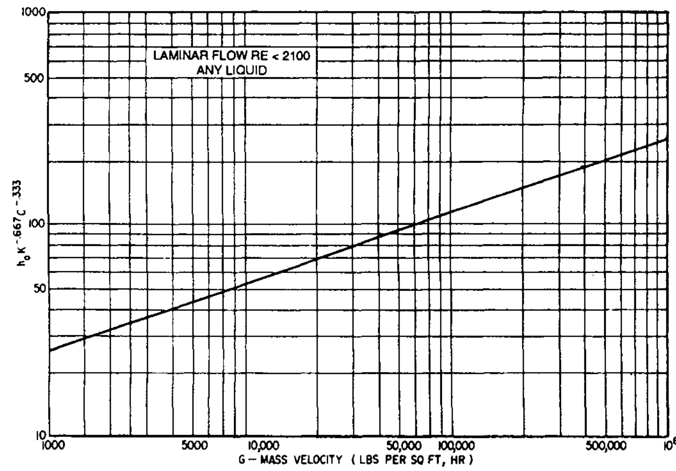


Figure 10-153A. For determination of h_o, shell-side (finned side) film coefficient $h_oK^{-0.667}C^{-0.333}$ for longitudinal fins, flow laminar. h_o must be corrected for fin efficiency using Figure 10-154 and mechanical data as Table 10-42. (Used by permission: Bul. "How to Design Double Pipe Finned Tube Heat Exchangers." © Brown Fintube Company, A Koch® Engineering Company, Houston, Texas.)

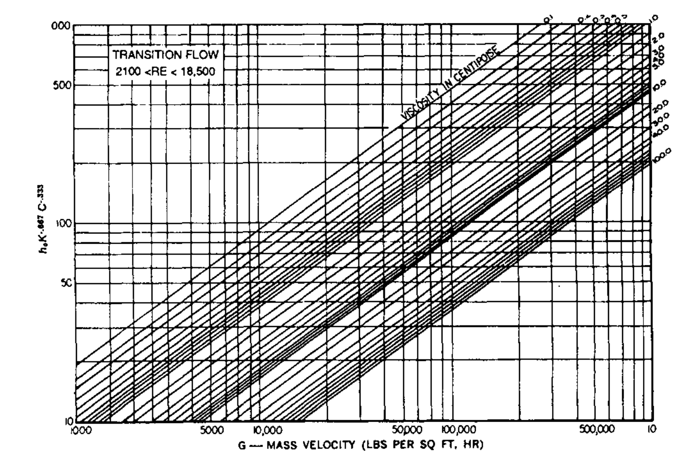


Figure 10-153B. Shell-side film coefficient for longitudinal fins, transition flow. See Figure 10-153A for applicable details. (Used by permission: Brown Fintube Company, A Koch® Engineering Company, Houston, Texas.)

Table 10-42
Brown Fintube's Typical Mechanical Design Data for Fintube Sections As Needed for Design Calculations

BFT Section Type	No. Tubes	No. Fins	Tube O.D. & Wall Thick (in.)	Shell Size Sch. 40 IPS	Fin Height in.	Net Free Area, in.^{2}	D_e Equiv. Dia., in.	A_f,A_o	A_o,A_i	Nominal Surface, Ft^2 A_o Nominal Length of Section 5 ft	10 ft	15 ft	20 ft	25 ft
X51	1	24	1.900 ×	3 in.	$^1/_2$	4.11	.415	.801	5.93	25	50	76	101	126
		36	.145			3.89	301	.858	8.3	35	71	106	141	177
X53	1	24	1.900 ×	4 in.	1	9.03	.542	.889	10.67	45	91	136	182	227
		36	.145			8.60	.379	.923	15.42	66	131	197	262	328

Notes:
1. Fin thickness equals 0.035 in. (narrow web).
2. A_f/A_o ratio of fin surface to total external heated surface.
3. A_o/A_i ratio of total external heated surface to inside tube surface.

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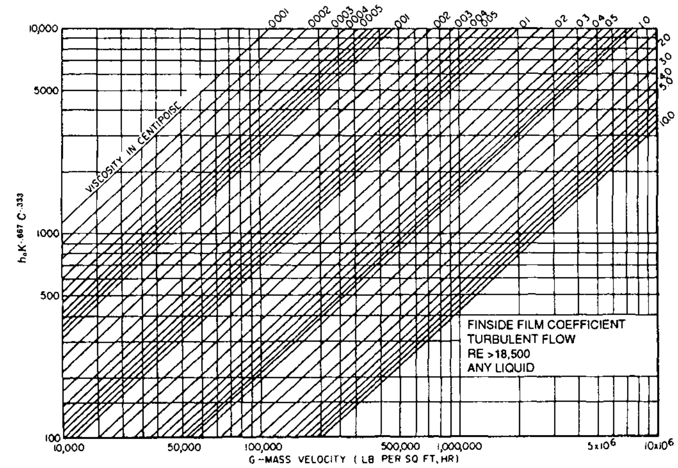

Figure 10-153C. Shell-side film coefficient, h_o, for longitudinal fins, flow turbulent. See Figure 10-153A and mechanical data from Table 10-42 for applicable details. The value of h_o must be corrected using Figure 10-154 and data of Table 10-42. (Used by permission: Brown Fintube Co., A Koch® Engineering Company, Houston, Texas.)

The needed equivalent diameter, D_e, is determined:[211]

$$D_e = \frac{4NFA}{\pi(D_s + D_t) + 2N(l)} \tag{10-249}$$

where NFA = net free area, in.^{2} from typical manufacturer's data as Table 10-42. The denominator is the wetted perimeter.

D_s = shell I.D., in.

D_t = tube O.D., in.

N = number of fins per tube

l = fin height, in.

After determining the h_o from the preceding figures, the film coefficient must be corrected for fin efficiency using Figure 10-154.

where E = 100 (TanH X)/X

X = $L(H_F/6KT)^{0.50}$

L = fin height, in.

H_F = fin film coefficient

K = conductivity of fin material, Btu/(hr) (ft^2)(°F/ft)

E = % fin efficiency

T = fin thickness, in.

Conductivity values*

Mat'l	K
Monel	15.0
18-8 st.stl	9.5**
C.steel	25.0
Low chrom stl	17.0
Nickel	35.0
Adm.% brass	65.0
Al	100
CU	200

*Average K values for temperatures 100–600°F. For temperatures beyond this range, see literature.

**Use E chart = 0.70 for design efficiency for this material.

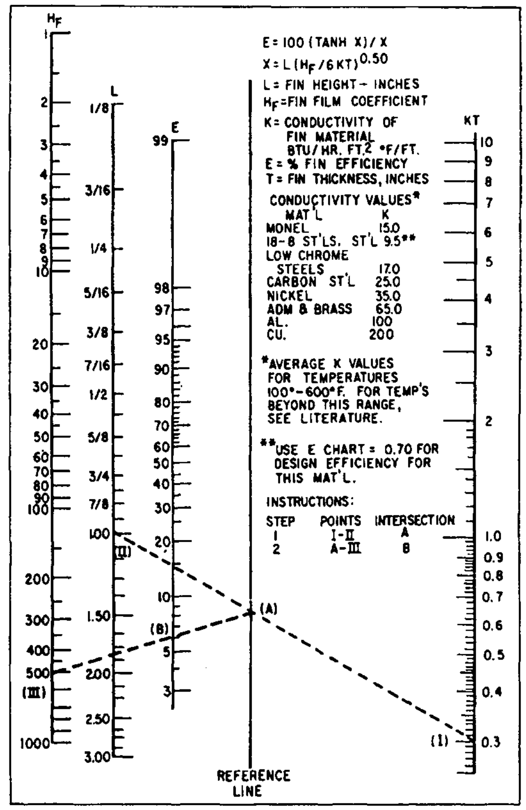

Figure 10-154. Finned transfer efficiency is never as great per unit area as the bare pipe; therefore, fin efficiency must be calculated to arrive at correct h_o, shell-side heat transfer coefficient. (Used by permission: Technical paper. © Brown Fintube Co., A Koch® Engineering Company, Houston, Texas.)

The total surface area, A_o, in the annulus is the sum of the extended surface area and the bare pipe surfaces not covered by fins. See Table 10-40. The fin efficiency, η_w, e_f or E, from Figure 10-154 is corrected for the percent surface that is finned. The corrected value, η_w, is the effective surface efficiency.

$$\eta_w = (E/100)(A_f/A_o) + (1 - A_f/A_o) \tag{10-250}$$

where A_f/A_o = fraction finned area

$(1 - A_f/A_o)$ = fraction bare or unfinned tube area

The net effective surface true film heat transfer rate is obtained by correcting the coefficient for the bare surface; thus,[211] fouling is excluded:

$$h_{bare} = (h_o K^{-0.667} c_p^{-0.333})(K^{0.667} c_p^{0.333}),$$

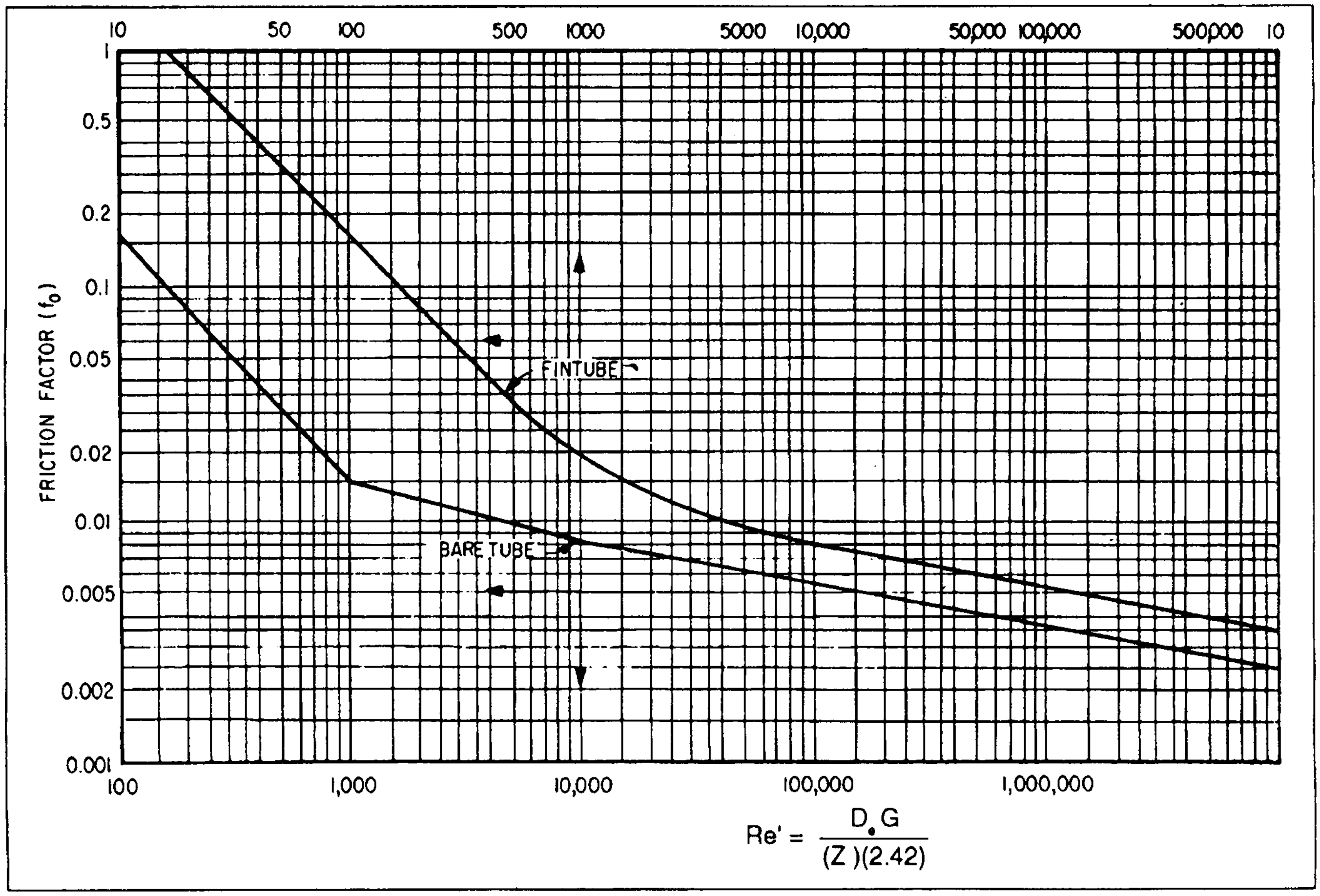

Figure 10-155. Shell-side friction factor, f_o, for pressure drop calculation is determined from plot vs. Reynolds Number. z = viscosity at average flowing temperature, centipoise. (Used by permission: Brown Fintube Co., A Koch® Engineering Company, Houston, Texas.)

using Figure 10-153A, 10-153B, or 10-153C. (10-251)

with r_o = shell-side fouling resistance

$$h_f = \frac{1}{(1/h_{bare}) + r_o}, \text{ Btu/(hr)(ft}^2\text{)(°F)}$$

$h_{of} = \eta_w(h_f)$, outside film coefficient with fouling, Btu/(hr) (ft^2)(°F)

Tube Wall Resistance

The pipe wall resistance to heat transfer is[211]

$$R_m = (O.D._{tube}/2K_m)(\ln[O.D._{tube}/I.D._{tube}]) \quad (10\text{-}252)$$

where K = thermal conductivity of tube metal, Btu/(hr) (ft^2)(°F/ft)
R_m = wall resistance, (hr) (ft^2)(°F)/Btu

Tube-Side Heat Transfer and Pressure Drop

Refer to the earlier section in this chapter, because tube-side pressure drop and heat transfer are subject to the same conditions as other tubular exchangers.

Fouling Factor

(See the earlier discussion in this chapter for more information on this topic.) Fouling factors require a lot of data, judgment, and experience. Ruining a design is easy to do by allowing for too large a fouling factor and actually creating a unit so large that the needed design velocities for heat transfer film coefficients cannot be attained.

The double-pipe longitudinal finned exchanger is designed by adding the fouling factor to each respective *film* coefficient before calculating the overall U_o.[211]

Finned Side Pressure Drop

Brown[211] recommends:

$$\Delta P = \frac{(0.000432)(f_o)(G')^2 L}{(D_e)(Z/Z_w)^{0.14}(\rho)} \tag{10-253}$$

Use Figure 10-156 to determine f_o.

$$Re = \frac{D_e G}{(Z)(2.42)} \tag{10-254}$$

where D_e = equivalent annulus diameter, ft; (see earlier calculation)
G = flow, lb/(ft^2)(hr) = 3,600 (G′)
G' = flow, lb/(ft^2)(sec)
Z = viscosity, average, centipoise
ρ = fluid density, lb/ft^3
L = equivalent length of travel, including bend factor, ft
D = tube I.D., ft.

After designing an approximate unit area requirement, it is important to review the final design performance details with a qualified exchanger manufacturer. See Table 10-42.

Miscellaneous Special Application Heat Transfer Equipment

It is necessary to work with the manufacturer in sizing and rating these special units, because sufficient public data/correlation of heat transfer does not exist to allow the design engineer to handle the final and detailed design with confidence.

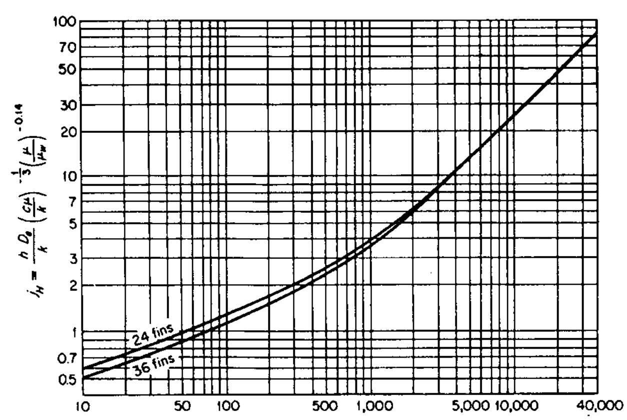

Figure 10-156. Heat-transfer curve for annuli with longitudinal fins. (Adapted from DeLorenzo, B., and Anderson, E. D. *Trans ASME*, V. 67, No. 697, ©1945. The American Society of Mechnical Engineers) (Used by permission: Kern, D. Q., and Kraus, A. D. *Extended Surface Heat Transfer*, p. 464, ©1972. McGraw-Hill, Inc. All rights reserved.)

A. Plate and Frame Heat Exchangers

Figures 10-7, 10-7A, 10-7B, and 10-7C illustrate the general arrangements of most manufacturers, although several variations of plate flow pattern designs are available to accomplish specific heat transfer fluids' temperature exchanges. Also, the gasket sealing varies, and some styles are seal welded (usually laser) to prevent cross-contamination. Note that Figure 10-7C has no interplate gaskets and is totally accessible on both sides, yet easy to clean.

The construction materials for the plates include most corrosion-resistant metals, usually 304SS, 316SS, titanium, Incoloy 825®, Hastelloy®, and others, plus nonmetallic fused graphite, and fluoroplastic Diabon F®. Typical gaskets between the plates include nitrile rubber, butyl, and EPDM elastromers, Hypalon® and Viton®, based on the various manufacturers' literature.

A heat transfer comparison is made in Figure 10-157. The plate and frame designs are used in convection, condensing, and some evaporation/boiling applications.

This type of exchanger usually provides relatively high heat transfer coefficients and does allow good cleaning by mechanically separating the plates, if back-flushing does not provide the needed cleanup. An excellent discussion on the performance and capabilities is presented by Carlson.[210] To obtain a proper design for a specific application, it is necessary to contact the several manufacturers to obtain their recommendations, because the surface area of these units is proprietary to the manufacturer.

B. Spiral Heat Exchangers

1. The spiral design heat exchangers, Figures 10-9A, 10-9B, 10-9C, and 10-9D are conveniently adaptable to many process applications. The true spiral units (Figure 10-9A and 10-9B) are usually large and suitable for higher flow rates, and the Heliflow®-style, Figure 10-9C, can be fabricated into small sizes, suitable for many "medium" (but not limited) process and sample cooler applications.
   The spiral units are used as cross-flow interchangers, condensers, and reboilers. These units can often be conveniently located to reduce space requirements. They are suitable for vacuum as low as 3mm Hg, because the pressure drops can be quite low. Bailey[214] identifies temperature limits of −30 to +1,500°F, pressure limits of 0 to 350 psia, maximum flow rate per shell of 3,000 gpm, and a heat transfer area of 4,000 ft^2. Trom[213] discusses a wide variety of process-related applications.
2. The Heliflow® is a tubular version of the spiral plate heat exchanger, Figures 10-9C and 10-9D, and has a high efficiency and counter-flow operation with a wide range of applications while occupying a limited space. The applications include vent condensing, sample coolers, instantaneous water heating, process heating and cooling, reboilers and vaporizers, cryogenic coolers,

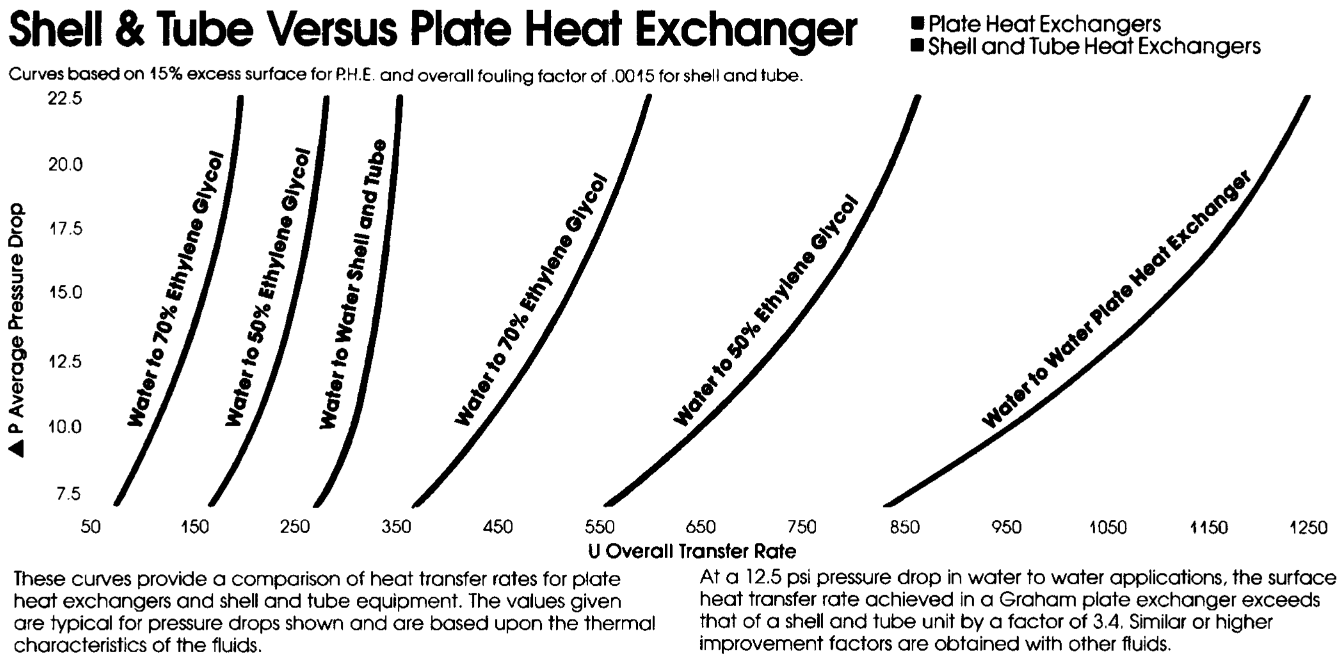

Figure 10-157. Convection heat transfer comparison for shell and tube and plate and frame exchangers. (Used by permission: Bul. PHE 96-1 6/96. ©Graham Manufacturing Company, Inc.)

interchangers, steam generators, process condensers, pump seal coolers, high temperature and high pressure exchangers, and others.[212] The heat transfer design and pressure drop should be referred to the manufacturer to obtain proper unit surface and casing size selection. The company has a bulletin providing charts to aid in preliminary size selection by the engineer. Also see Minton[268] for heat transfer calculations.

This unit can be fabricated of a wide range of ferrous, stainless steels, and nonferrous corrosion resistant metals and alloys.

C. Corrugated Tube Heat Exchangers

Figures 10-10I, 10-10J, and 10-10K indicate the process flow patterns for single tube units and for multiple corrugated tubes in a single plain shell. These units are suitable for heating or cooling process fluids containing high pulp or fiber content or suspended particulates. The heat transfer coefficients are improved when compared to plain tubes as the turbulence improves the performance. The units can be arranged in multiple shells for parallel or series flow. The manufacturers should be contacted for details.

D. Heat Transfer Flat (or Shaped) Panels

Heat transfer panels are generally used to fit onto a process vessel shape and to transfer heat from the panel through a good heat transfer cement and into the wall of the process vessel, or can be used to create a physically tight fit without the cement. Then the fluid in the vessel is heated or cooled or "held at temperature" by the heat/cooling from/into the panel. The shapes of these panels are versatile and can be used individually to submerge in tanks or vessels and to wrap around cylindrical vessels to serve a wide range of applications.

Generally, two styles and techniques of fabrication are used, but may vary between manufactures, see Figures 10-158A and 10-158B. Note the importance of good flow distribution in between the heat transfer plates/panels, which suggests the specific style depending on whether the heat is to be transferred to only one side of the plate pair or to both sides, as in submerged applications. Note that to improve heat transfer (internally), the fluid velocity may be designed to increase the film coefficient by use of series or parallel zones.

A few application arrangements are given in Figures 10-159A, 10-159B, 10-160, 10-161, and 10-162.

The heat transfer calculations are presented by the several manufacturers, and due to the proprietary nature of the surface areas, are available for various arrangements. It is advisable to obtain specific help.

It is important to recognize any galvanic corrosion between the heat transfer surface and any metal to which it is attached or connected. This can depend on many factors that must be recognized in the selection of construction

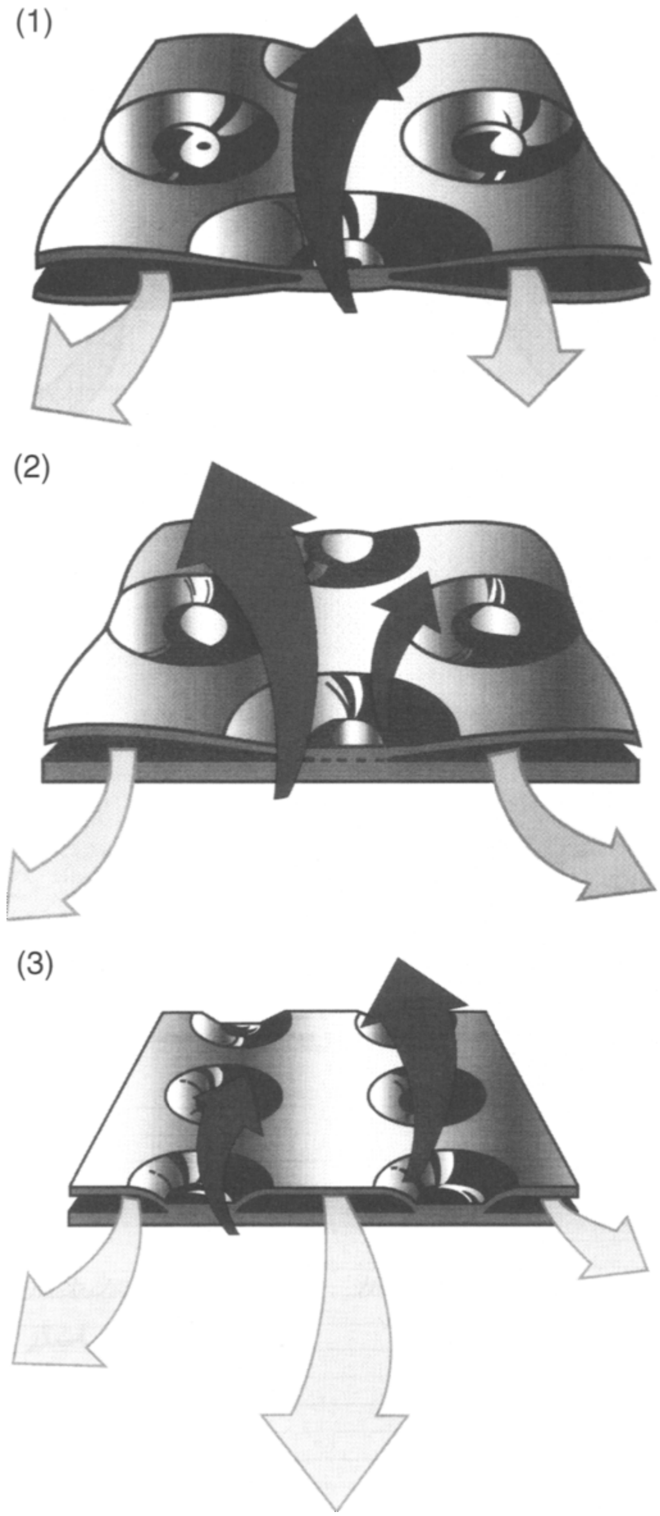

Figure 10-158A. Styles of Mueller Temp-Plate® heat transfer plates. (1) Double-embossed surface, inflated both sides. Used in immersion applications, using both sides of the heat transfer plate. (2) Single-embossed surface, inflated one side, used for interior tank walls, conveyor beds. (3) Dimpled surface (one side), available MIG plugwelded or resistance spot welded. Used for interior tank walls, conveyor belts. (Used by permission: Bul. TP-108-9, ©1994. Paul Mueller® Company.)

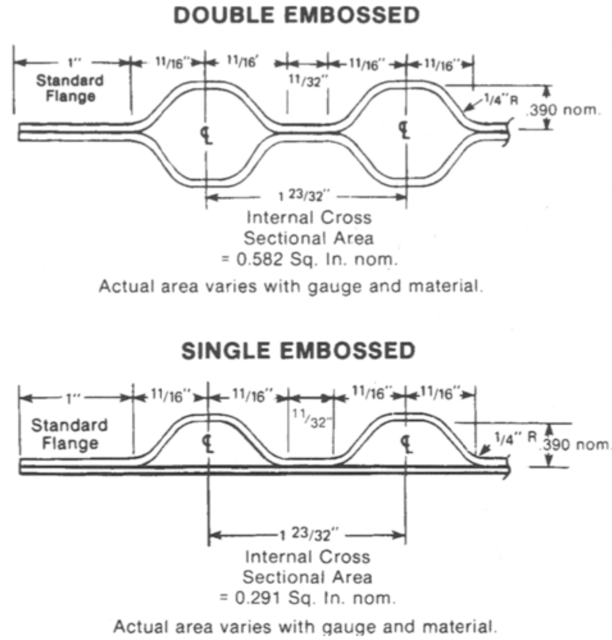

Figure 10-158B. Platecoil® double- and single-embossing designs for standard units. The Platecoil® is fabricated using resistance, spot, seam, and Tungsten Inert Gas (TIG) and/or Metal Inert Gas (MIG) welding techniques in order to hold and seal the two plates together. (Used by permission: Cat. 5-63, ©1994. Tranter®, Inc.)

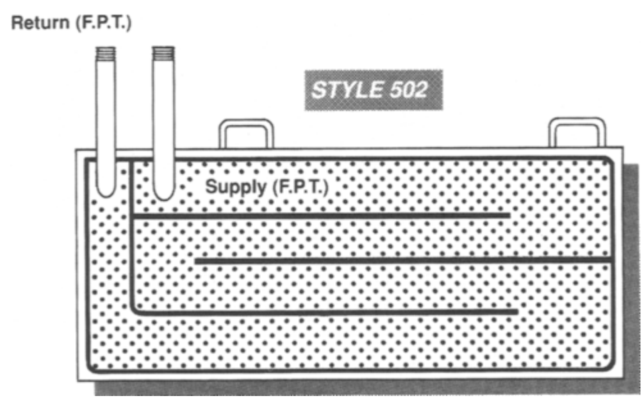

Figure 10-159A. Used as an immersion plate with liquids, the serpentine flow path increases the heat transfer rate. (Used by permission: Cat. "Heat Transfer Equipment." DEC International, Engineered Products Group.)

materials as well as pure corrosion of the metal by the chemical environment. Likewise, the thermal expansion of the heat transfer surface must be accounted for by the manner in which it is attached, fastened, or connected to the equipment to be heated or cooled.

E. Direct Steam Injection Heating

This system is used for heating liquids for process and utility services.[217] Using proper controls, the temperature of the

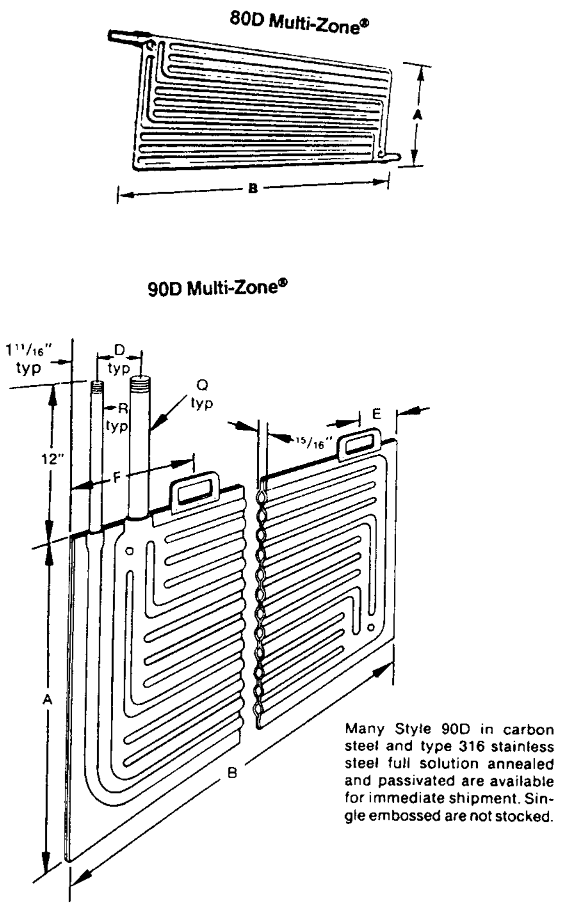

Figure 10-159B. Typical styles of Platecoil®. Other styles include vertical and serpentine. (Used by permission: Cat. PCC-1-25M-RLB-1290, ©1990. Tranter®, Inc.)

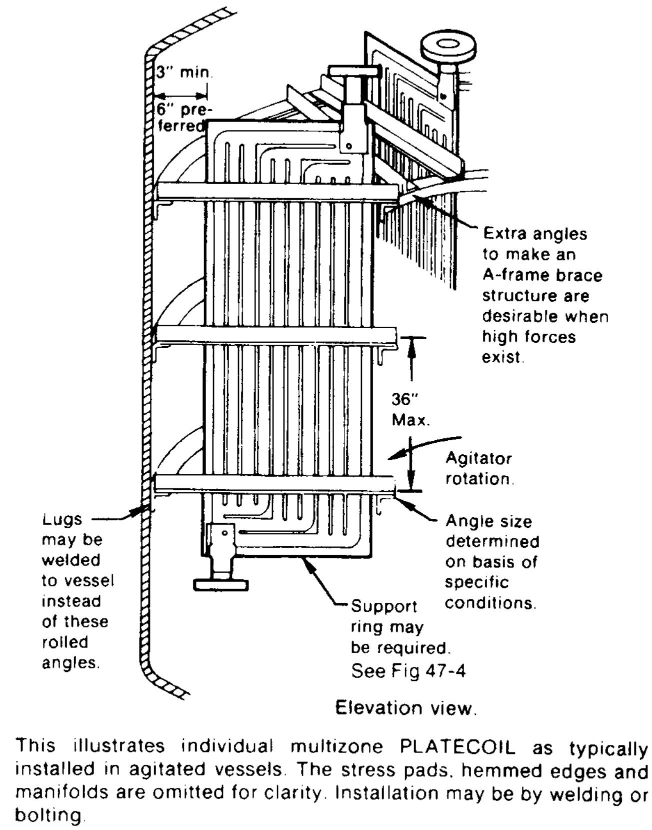

Figure 10-161. This figure illustrates an individual multizone Platecoil® as typically installed in agitated vessels. The stress pads, hemmed edges, and manifolds are omitted for clarity. Installation may be completed by welding or bolting. (Used by permission: Cat. 5-63, Sept. 1994. ©Tranter®, Inc.)

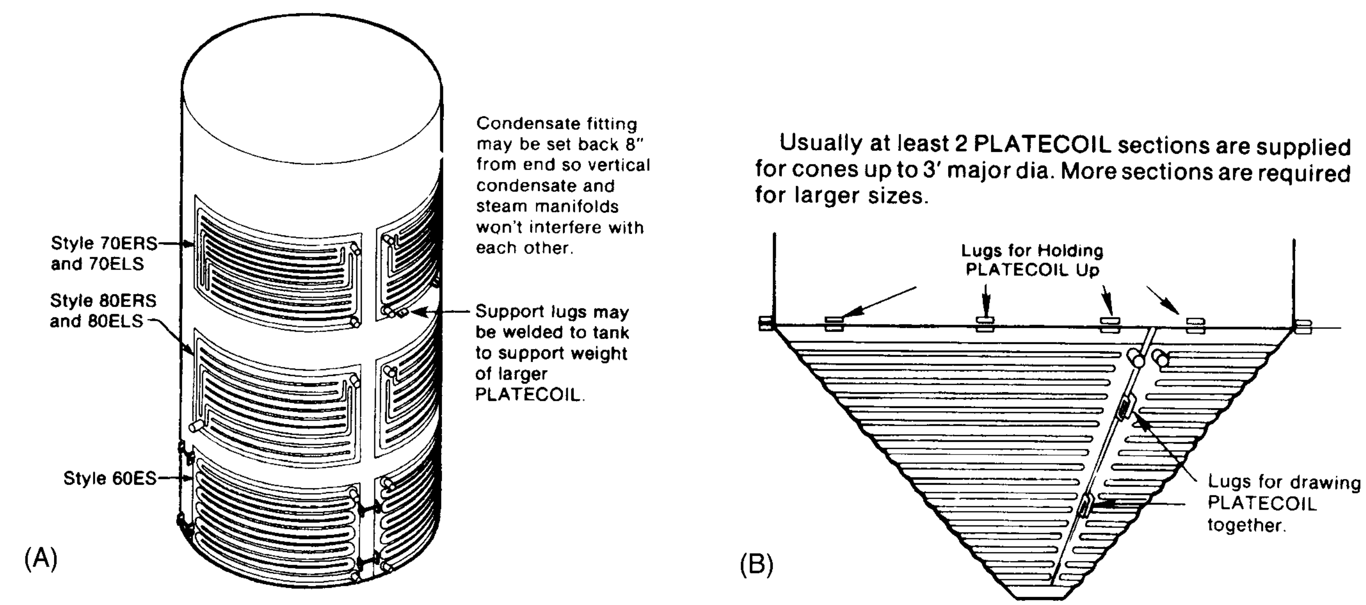

Figure 10-160. Platecoils® on tank walls and cone bottoms. Note: See Figure 10-163 for use of heat transfer mastic between vessel and heat transfer coils/plates. (Used by permission: Bul. 5-63, ©1994. Tranter®, Inc..)

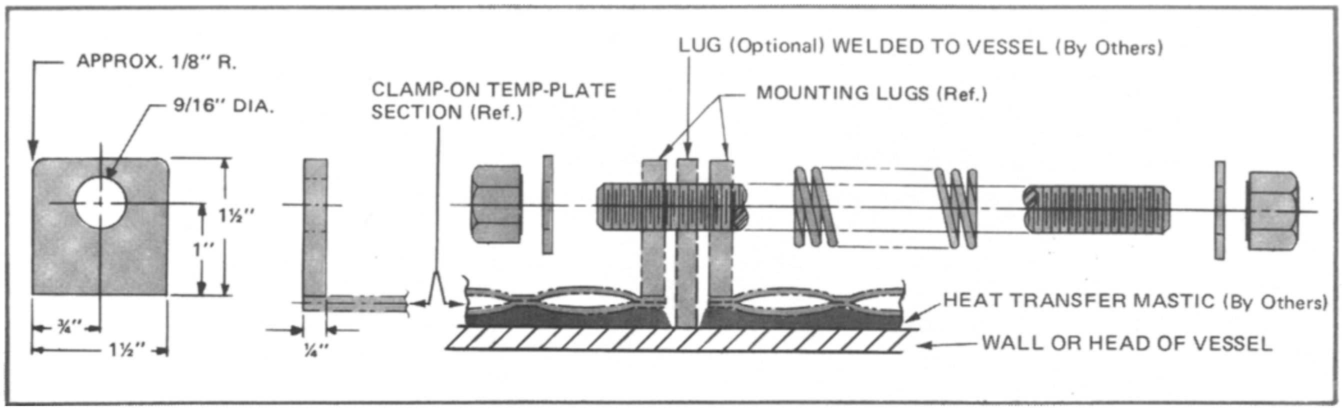

Figure 10-162. Typical heat transfer cement/mastic sealing between vessel and heat transfer plates/coils/Temp-Plates® using spring-loaded assembly. (Used by permission: Cat. TP-108-9, ©1994. Paul Mueller® Co.)

HOW THE PICK "CONSTANT FLOW" HEATER WORKS:

1 Set pneumatic controller to any desired outflow temperature. This temperature will be maintained within 3°F regardless of variations in inlet liquid temperature.

2 Modulating steam control valve, activated by the temperature controller ❶ admits the exact amount of steam needed to maintain the desired outflow temperature.

3 Water (or water-miscible liquid) to be heated enters mixing chamber here.

4 Steam is injected into the liquid through hundreds of very small orifices in the injection tube. The fine "bubbles" of steam are instantly absorbed by the liquid, resulting in 100% transfer of heat energy. The spring-loaded piston rises or falls as more or less steam is required. This arrangement prevents pressure equalization between steam and water pressures, thus eliminating steam/water hammer. Helical flights on the chamber wall ensure thorough mixing of steam and water for total and immediate energy transfer to the liquid.

5 Hot water/liquid outlet.

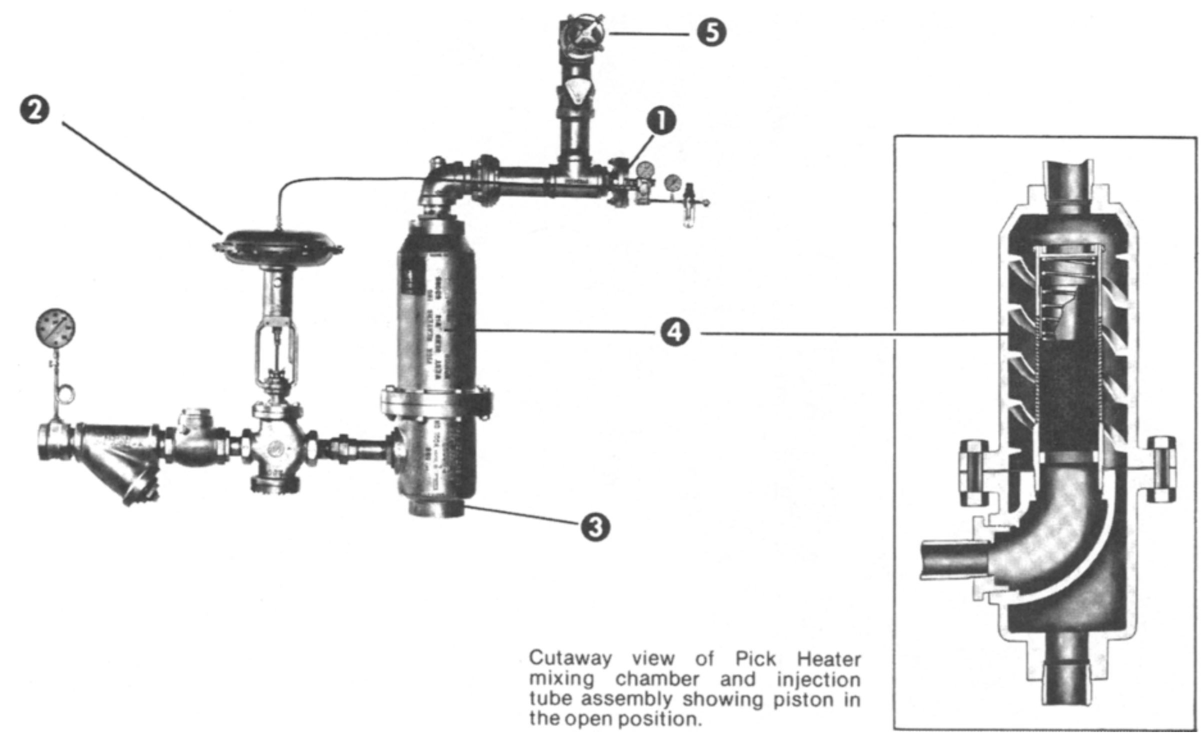

Figure 10-163. Constant flow direct steam heater (variable flow also available.) (Used by permission: Cat. CF-5924 P. Pick™ Heaters, Inc.)

resulting mixture can be set for the desired temperature for direct mixing, heating jackets of vessels, and similar requirements, see Figures 10-163 and 10-164.

F. Bayonet Heat Exchangers

Bayonet heat exchangers are modified shell and tube types. The tubes are concentric with the outer tube, being sealed closed at one end, although the shell in its entirety is not always used or needed, see Figure 10-165. A helpful article describing this type of unit is by Corsi.[216]

A useful application is for tank and vessel heating, with the heater protruding into the vessel. Bayonet heat exchangers are used in place of reactor jackets when the vessel is large and the heat transfer of a large mass of fluid through the wall would be difficult or slow, because the bayonet can have considerably more surface area than the vessel wall for transfer. Table 10-43 compares bayonet, U-tube, and fixed-tubesheet exchangers.[216]

The outer and inner tubes extend from separate stationary tube sheets. The process fluid is heated or cooled by heat transfer to/from the outer tube's outside surface. The overall heat transfer coefficient for the O.D. of the inner tube is found in the same manner as for the double-pipe exchanger.[70] The equivalent diameter of the annulus uses the perimeter of the O.D. of the inner tube and the I.D. of the inner tube. Kern[70] presents calculation details.

G. Heat-Loss Tracing for Process Piping

The two basic types of systems for maintaining and/or heating process piping temperature conditioning are (1) steam tracing or jacketing and (2) electric tracing. For most systems requiring extensive pipe lengths of heat maintenance, it is advisable to make an economic cost comparison for both capital and operating costs between the two applicable systems. For electric tracing see pg. 245.

1. Steam Tracing

See Figures 10-166A and 10-166B.

To maintain a desired temperature in the process pipe, it may be necessary to use 1, 2, or 3 tracer tubes (small pipes) located symmetrically around the pipe and running parallel to the pipe; however, at valves and fittings, the tracing needs to be so placed as to provide protection uniformly to the surface. Some designers recommend arranging the tracing in the lower half of the pipe.

2. Bare Tracer

See Figure 10-166A.

The bare tracer is usually copper tubing, or sometimes carbon or stainless steel tubing, usually of $^3/_8$-in., $^1/_2$-in., or $^3/_4$-in. nominal size.

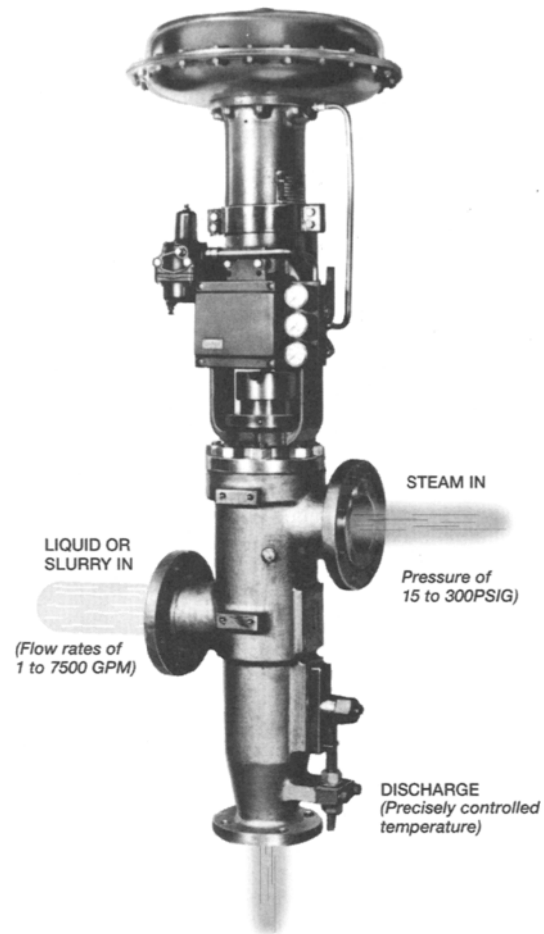

Figure 10-164. Direct steam heating of liquids with internal temperature control using variable orifice steam nozzle. (Used by permission: Bul. H 150. Hydro-Thermal Corp.)

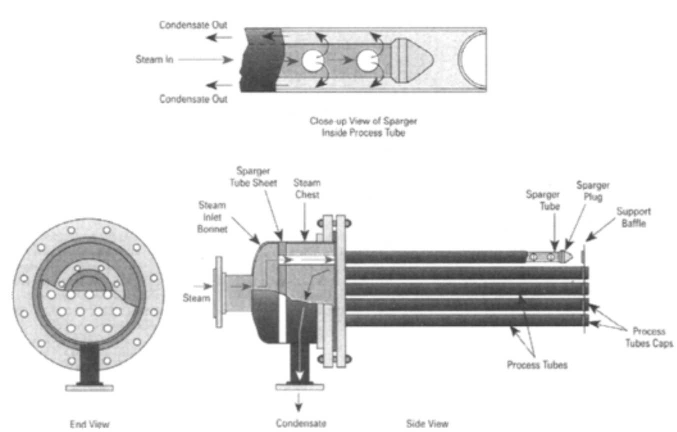

Figure 10-165. Typical bayonet type heat exchanger, showing the key sparger arrangement internally as a part of each tube. (Used by permission: Corsi, R. *Chemical Engineering Progress,* V. 88, No. 7, ©1992. American Institute of Chemical Engineers. All rights reserved.)

Table 10-43
Comparison of Bayonet, U-Tube, and Fixed Tubesheet Heat Exchangers

Design	Advantages	Limitations	Applications and Notes
Bayonet	Removable tube bundle permits easy internal cleaning. Design allows free expansion of tubes in high-temperature service. Needs no expansion joint if shell is used.	Double tubesheet increases initial cost.	Commonly used for heating or cooling very corrosive fluids that require expensive corrosion-resistant materials. Less economical than U-tube design for in-tank heating.
U-tube	Elimination of one tubesheet reduces initial cost. Tube bundle is removable for inspection and cleaning. Full tube bundle minimizes shell-side bypassing. U-bends permit each tube to expand and contract individually. Tube bundle expansion is independent of shell; no expansion diaphragm is required.	Bends make mechanical cleaning of tube interiors difficult. Also, only a few outer bends can be replaced, so retubing usually involves replacement of all tubes.	Recommended for high-pressure (>600 psi), high-temperature applications. Tube shape allows extreme temperature differences ($\Delta T > 250°F$) across the bundle. Often used as integral column bottom reboiler and as tank suction heater to preheat product before pumping. Tube side cannot be made single-pass.
Fixed-tubesheet	Lower cost per ft^2 of heat-transfer surface. Replaceable straight tubes allow for easy internal cleaning. Full tube bundle minimizes shell-side bypassing. No packed joints or internal gaskets, so hot and cold fluids cannot mix due to gasket failure.	Differential expansion must be accommodated by an expansion joint. Gasket failure can allow tube-side fluid to escape to the atmosphere.	Almost universal application unless a removable tube bundle is required for exterior inspection and cleaning, which may be avoided by running the fouling fluid on the accessible tube side. Completely closed shell side eliminates gasket leakage. Excellent for high-vacuum work. Also available in double-tubesheet design to eliminate cross-contamination.

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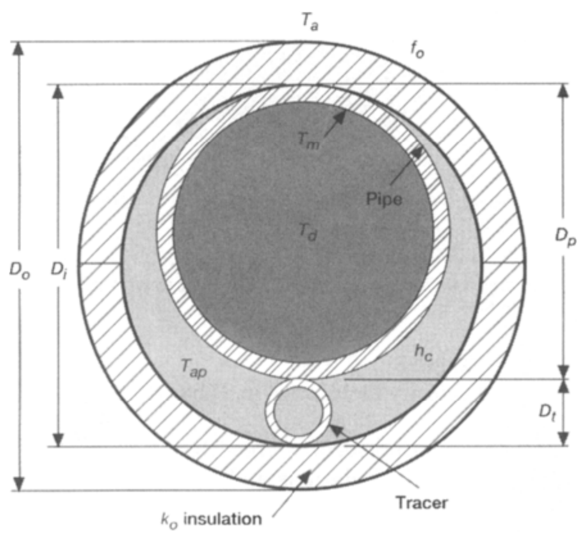

Figure 10-166A. Cross-sectional view of pipe with bare single tracer. Requirements may dictate 2 or 3 tracer pipes/tubes strapped to pipe at generally equal spacing around circumference, then insulated. (Used by permission: Foo, K. W. *Hydrocarbon Processing,* V. 73, No. 1, Part 1, ©1994. Gulf Publishing Company.)

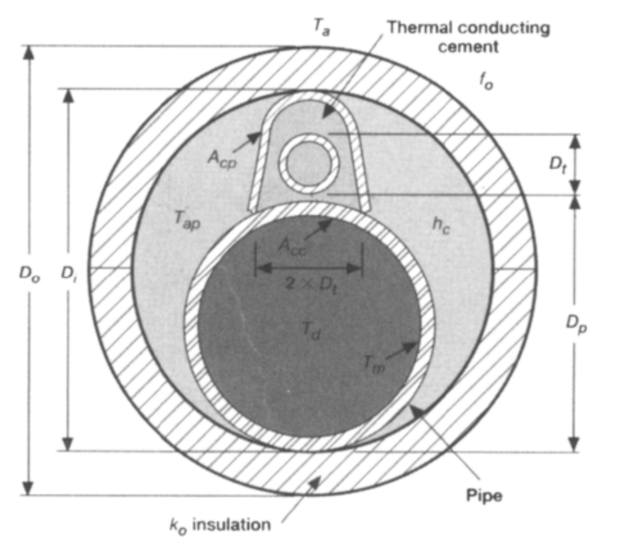

Figure 10-166B. Cross-sectional view of pipe and tracer with thermal conducting cement. (Used by permission: Foo, K. W. *Hydrocarbon Processing,* V. 73, No. 1, Part. 1, ©1994. Gulf Publishing Company.)

a. From reference Foo,[223] heat loss through the insulation to the ambient air is

$$Q_{ia} = U_o \frac{\pi D_o}{12}(T_m - T_a) \tag{10-255}$$

The overall heat transfer coefficient for the insulation and the ambient air is[223]

$$\frac{1}{U_o} = \frac{(D_o/2)\ln(D_o/D_i)}{k_o} + \frac{1D_o}{h_c D_i} + \frac{1}{f_o} \tag{10-256}$$

For f_o, see Table 10-44.

The overall transfer coefficient for the tracer annulus space is

$$\frac{1}{U_t} = \frac{1}{h_c} + \frac{1}{h_s} \tag{10-257}$$

For condensing steam, the heat transfer coefficient, h_s, is approximately 2,000 Btu/(hr) (ft²)(°F), and the preceding equation approximates to

$$U_t = h_c = 0.45\frac{(T_s - T_{ap})^2}{D_t} \tag{10-258}$$

b. Heat loss when tracer is surrounded by thermally conducting cement and insulated (otherwise same as (a), see Figure 10-167):

$$Q_{ia} = U_o\pi(D_o/12)(T_{ap} - T_a) \tag{10-259}$$

Annulus space temperature,

$$T_{ap} = T_m - \frac{nq_t A_{cc}(T_s - T_m)}{h_c(A_p - nA_{cc})}, \text{°F} \tag{10-260}$$

Table 10-44
Wind Velocity Factor, f_o@dT = 150°F Mean

Wind velocity, mph	f_o
0	2.5
5	3.8
10	4.8
15	5.5
20	6
25	6.5
30	7
35	7.3
40	7.7
45	8

Note: $dT = T_{wall} - T_a$

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Pipe temperature, 223

$$T_m = \frac{T_a\left[\frac{a}{(a+b+c)}\right] + T_s\left[\frac{c}{(a+b+c)} + d\right]}{1 + d - \frac{b}{(a+b+c)}} \tag{10-261}$$

where $a = U_o A_o$
$b = h_c(A_p - nA_{cc})$
$c = h_c A_{cp}$
$d = nq_t A_{cc}$
$A_p = \pi D_p/12$, superficial area of pipe, ft²/ft
$A_{cp} = 0.23357 D_t$, cement channel superficial area, ft²/ft
$A_{cc} = 2 D_t/12$, cement contact area, ft²/ft
$A_o = \pi D_o/12$, external superficial area of insulation, ft²/ft
$Q_{ia} = Q_{pa} + Q_{ta}$
$Q_{tp} = Q_{pa}$
D_o = O.D. of insulation, in.
D_t = O.D. of tracer, in.
T_s = steam temperature, °F
T_a = ambient temperature, °F
h_c = average of the horizontal and vertical transfer film coefficients by convection in still air.

$$U_t = h_c = 0.45\frac{(T_s - T_{ap})^2}{D_t} \tag{10-262}$$

Assume $T_{ap} = T_m$, annulus space temperature = pipe temperature, °F.

Annulus space temperature, T_{ap};

$$T_{ap} = T_m - \frac{nq_t A_{cc}(T_s - T_m)}{h_c(A_p - nA_{cc})} \tag{10-263}$$

Then the pipe temperature, T_m is[223]

$$T_m = \frac{T_a\left[\frac{a}{(a+b+c)}\right] + T_s\left[\frac{c}{(a+b+c)} + d\right]}{1 + d - \frac{b}{a+b+c}} \tag{10-264}$$

The heat transfer from annulus space through insulation to air:

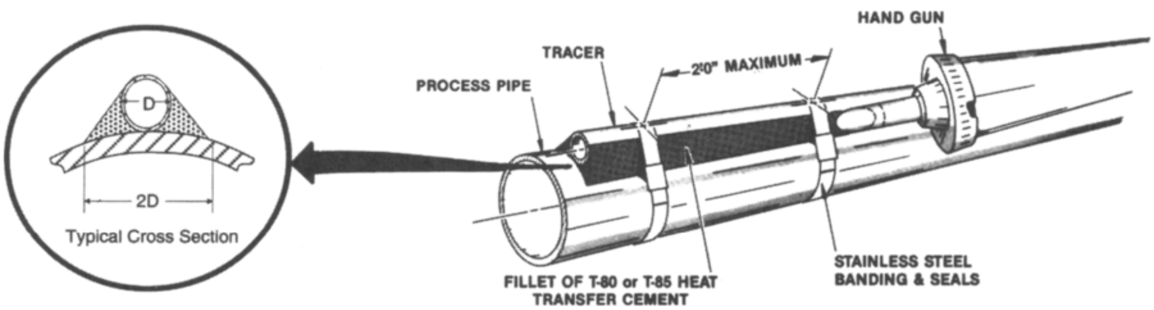
Q_{ia} = heat transfer from annulus space through insulation to air, Btu/hr/ft pipe
Q_{pa} = heat transfer from process pipe to annulus space, Btu/hr/ft pipe
Q_{ta} = heat transfer from tracer to annulus space, Btu/hr/ft pipe
Q_{tp} = heat transfer from tracer to process pipe, Btu/hr/ft pipe
Q_{ap} = heat transfer from annulus space to pipe, Btu/hr/ft pipe
n = number of tracers

$$Q_{ia} = U_o \frac{nD_o}{12}(T_{ap} - T_a) \quad (10\text{-}265)$$

Table 10-45 presents types of insulation material. Foo[223] gives insulation thermal conductivity, k, at 100°F mean as:

Calcium silicate	0.38 Btu/(hr)(ft)(F)
Foam glass	0.40
Mineral wool	0.28

Heat transfer cements are quite useful for transferring the heat from an external tracing when attached outside of the process pipe, Figures 10-167 and 10-168. To determine the number of heat transfer steam tracers, it is important to contact the manufacturer of the heat transfer cement. The illustrations here should be considered preliminary for approximating purposes. The information/data that follows is used with permission from Thermon® Manufacturing Co./Cellex Div. Except for specific conditions, most applications represent the requirements to *maintain* a pipe (or vessel) system temperature, not to raise or lower the temperature.



Installation of Tracing on Straight Runs of Pipe: Tracers are to be run parallel and in direct contact with the process pipe where possible. Tracer location on pipe is to be where most accessible. If more than two tracers are used, they should be equally spaced circumferentially around the pipe.

Figure 10-167. Tracer placement on pipe using heat transfer cement. (Used by permission: Bul T-109M ©1994. Thermon® Manuifacturing Co./Cellex Div.)

Table 10-45
Insulation Material and Thickness

Temp. Ranges, °F and Recommended Insulation Thicknesses, in.							
	Mineral Wool & Calcium Silicate					Foam Glass	
Pipe Size NPS	100–199	200–399	400–599	600–699	700–799	100–390	Up to 390
						normal	fire protn.
1	1	1 1/2	2	2	2 1/2	1 1/2	3
1 1/2	1	1 1/2	2	2	2 1/2	1 1/2	3
2	1 1/2	1 1/2	2 1/2	2 1/2	3	1 1/2	3
3	1 1/2	1 1/2	2 1/2	3	3	1 1/2	3
4	1 1/2	1 1/2	2 1/2	3	4	1 1/2	3
6	1 1/2	1 1/2	2 1/2	3	4	1 1/2	3
8	2	2	2 1/2	3	4	2	3
10	2	2	2 1/2	3	5	2	3
12	2	2	3	3	5	2	3
14	2	2	3	4	5 1/2	2	3
16	2	2	3	4	5 1/2	2	3
18	2	2	3	4	5 1/2	2	3
20	2	2	3	4	5 1/2	2	3
24	2	2	3	4	5 1/2	2	3
30	2	2	3	4	5 1/2	2	3
36	2	2	3	4	5 1/2	2	3

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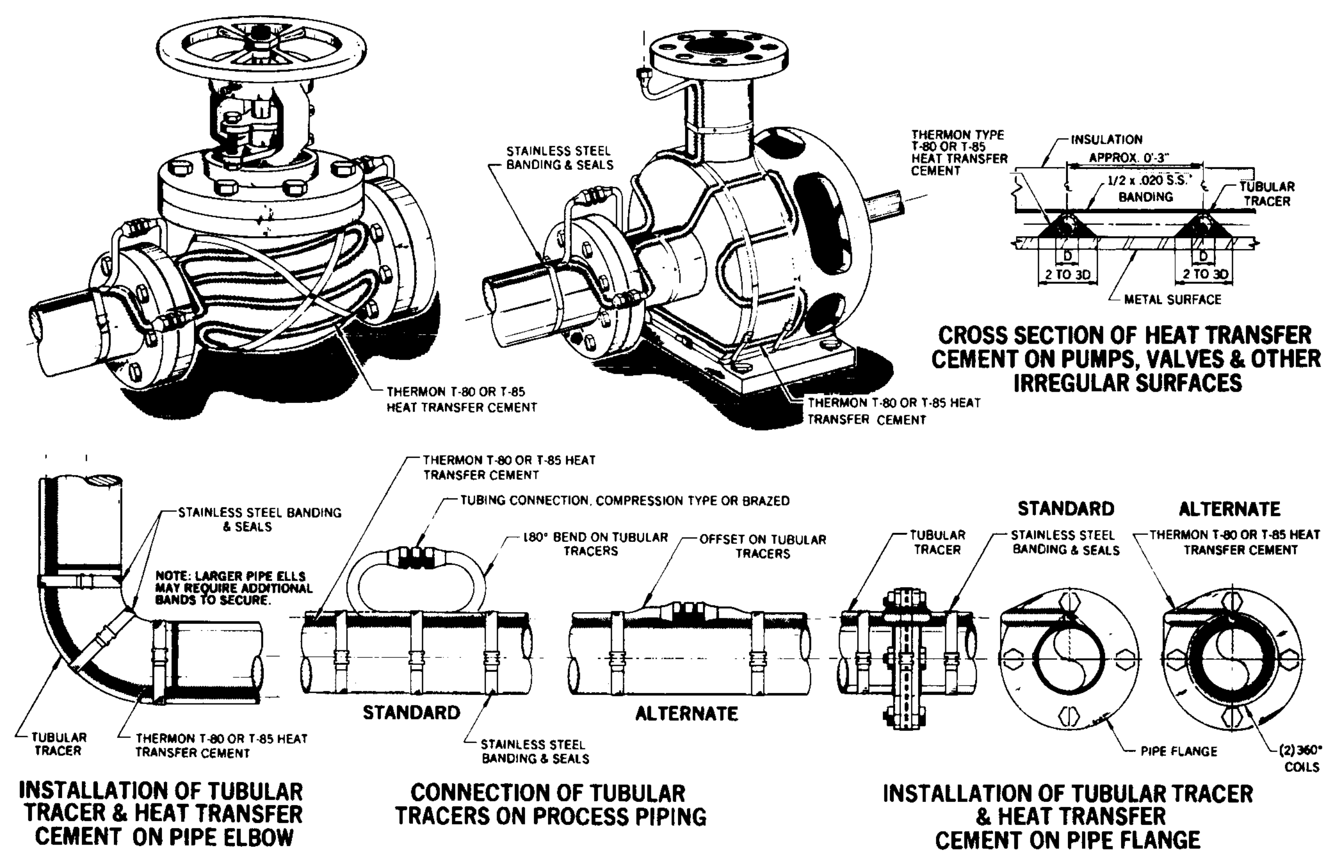

Figure 10-168. Installation of heat transfer cement with tracing on valves, pumps, and pipe. (Used by permission: Bul.T-109-M, ©1994. Thermon® Manufacturing Co./ Cellex Div.)

In design considerations for Thermonized® process lines, temperatures may be determined by the "Stagnation Method." The calculations involved in this method are based on static conditions where process fluid flow is not present, and are independent of the viscosity, density and thermal conductivity of the process fluid. The process temperature may be calculated from the following relationship:

$$R = \frac{T_p - t_a}{T_s - T_p} \tag{10-266}$$

$$T_p = \frac{RT_s + t_a}{1 + R} \tag{10-267}$$

where T_p = process temperature, °F
t_a = ambient temperature, °F
T_s = steam temperature, °F
R = factor from Table 10-46

Example 10-24. Determine the Number of Thermonized® Tracers to Maintain a Process Line Temperature

Used by permission of Thermon® Manufacturing Co./Cellex Div.

Assume a 3-in. line. Design process temperature: 320°F (T_p). Insulation: 1 $^1/_2$-in. thick calcium silicate. Steam temperature: 366°F (T_s). Ambient temperature: 0°F (t_a).

Required: The number (N) and size of Thermonized® tracers required to maintain a 320°F process temperature (T_p) under the preceding conditions.

Solution: Calculate the R factor and determine the tracer requirements from Table 10-46.

$$R = \frac{T_p - t_a}{T_s - T_p} = \frac{320 - 0}{366 - 320} = 6.96$$

From Table 10-46 it can be determined that the calculated R factor of 6.96 is less than that of 7 shown for one $^3/_8$-in. O.D. tracer on a 3-in. line using 1 $^1/_2$-in. insulation. Thus, a single $^3/_8$-in. O.D. tracer is satisfactory.

The overall heat transmittance from tracer through heat transfer cement to process pipe, q_t, in Btu/(hr) (ft^2) (°F) is given in Table 10-47.[223] From the detailed articles of Foo,[223] the following nomenclature applies:

Table 10-46
R-Factors for Thermonized® Process Lines

Process Line Size I. P. S. (Schedule 40) / TRACER TUBING SIZE Number of Parallel Tracers or Ft. of Tracing Per Ft. of Pipe	3/8" O.D. Tubing								1/2" O.D. Tubing							
	1	2	3	4	5	6	7	8	1	2	3	4	5	6	7	8
1"	29 / 32	— / —	— / —	— / —	— / —	— / —	— / —	— / —	32 / 36	— / —	— / —	— / —	— / —	— / —	— / —	— / —
2"	9.5 / 10.6	29 / 32	— / —	— / —	— / —	— / —	— / —	— / —	10.9 / 12.1	32 / 36	— / —	— / —	— / —	— / —	— / —	— / —
3"	6.3 / 7.0	13.2 / 14.7	29 / 32	— / —	— / —	— / —	— / —	— / —	7.3 / 8.1	15.1 / 16.8	32 / 36	— / —	— / —	— / —	— / —	— / —
4"	4.7 / 5.3	9.5 / 10.6	17.6 / 19.6	29 / 32	— / —	— / —	— / —	— / —	5.4 / 6.0	10.9 / 12.1	19.8 / 22	32 / 36	— / —	— / —	— / —	— / —
6"	3.1 / 3.5	6.3 / 7.0	9.5 / 10.6	13.2 / 14.7	18.9 / 21	29 / 32	— / —	— / —	3.6 / 4.0	7.3 / 8.1	10.9 / 12.1	15.1 / 16.8	21 / 24	32 / 36	— / —	— / —
8"	— / 2.7	— / 5.3	— / 8.2	— / 10.6	— / 13.9	— / 18.7	— / —	— / —	— / 3.1	— / 6.0	— / 9.3	— / 12.1	— / 15.9	— / 21	— / —	— / —
10"	— / 2.1	— / 4.2	— / 6.4	— / 8.9	— / 10.6	— / 13.1	— / —	— / —	— / 2.4	— / 4.8	— / 7.3	— / 10.1	— / 12.1	— / 15.0	— / —	— / —
12"	— / 1.7	— / 3.5	— / 5.3	— / 7.0	— / 8.9	— / 10.6	— / 12.4	— / 14.7	— / 1.9	— / 4.0	— / 6.0	— / 8.1	— / 10.1	— / 12.1	— / 14.1	— / 16.8
14"	— / 1.5	— / 3.1	— / 4.6	— / 6.2	— / 7.9	— / 9.4	— / 10.6	— / 12.6	— / 1.7	— / 3.5	— / 5.2	— / 7.1	— / 9.0	— / 10.7	— / 12.1	— / 14.4
16"	— / 1.3	— / 2.7	— / 3.9	— / 5.3	— / 6.9	— / 8.2	— / 9.3	— / 10.6	— / 1.5	— / 3.0	— / 4.4	— / 6.0	— / 7.8	— / 9.7	— / 10.6	— / 12.1
18"	— / 1.1	— / 2.3	— / 3.5	— / 4.8	— / 6.1	— / 7.3	— / 8.4	— / 9.4	— / 1.2	— / 2.6	— / 4.0	— / 5.5	— / 7.0	— / 8.3	— / 9.6	— / 10.7
20"	— / 1.0	— / 2.1	— / 3.1	— / 4.2	— / 5.3	— / 6.4	— / 7.6	— / 8.6	— / 1.1	— / 2.4	— / 3.5	— / 4.8	— / 6.1	— / 7.3	— / 8.7	— / 9.8
24"	— / .8	— / 1.7	— / 2.6	— / 3.5	— / 4.5	— / 5.3	— / 6.3	— / 7.4	— / .9	— / 1.9	— / 2.9	— / 3.9	— / 5.1	— / 6.0	— / 7.2	— / 8.4

Note: The upper figure is based on 1-in. insulation, the lower on $1^1/_2$ inch. This data is to be used for temperature maintenance only.

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A_{cc} = cement contact area, ft^2/ft
A_{cp} = cement channel superficial area, ft^2/ft
A_o = external superficial area of insulation, ft^2/ft
A_p = superficial area of pipe, ft^2/ft
D_i = I.D. of insulation, in.
D_o = O.D. of insulation, in.
D_p = O.D. of pipe, in.
D_t = O.D. of tracer, in.
f_o = wind velocity factor, Btu/hr-ft²-°F
h_c = convective heat transfer coefficient, Btu/hr-ft²-°F
h_s = steam, heat transfer coefficient, Btu/hr-ft²-°F
k_o = thermal conductivity of insulation, Btu/hr-ft-°F
L = length of pipe, ft
n = number of tracers
q_t = overall heat transmittance from tracer through cement to process pipe, Btu/hr-ft²-°F
Q_{ap} = heat transfer from annulus space to pipe, Btu/hr/ft pipe
Q_{ia} = heat transfer from annulus space through insulation to air, Btu/hr/ft pipe
Q_{ta} = heat transfer from tracer to annulus space, Btu/hr/ft pipe
Q_{tp} = heat transfer from tracer to process pipe, Btu/hr/ft pipe
T_a = ambient temperature, °F
T_{ap} = annulus space temperature, °F
T_d = desired holding temperature, °F
T_m = pipe temperature, °F
T_s = steam temperature, °F
U_o = overall outside heat transfer coefficient from insulation to air, Btu/hr-ft²-°F
U_t = overall heat transfer coefficient from tracer to annulus space, Btu/hr-ft²-°F

Other useful references to steam and electrical tracing include 232, 233, 234, 235, 236, 237, 238, 239, 240.

The electric heat tracer systems require good temperature control. A self-regulating system is shown in Figure 10-169. The manufacturers should be consulted to prepare proper temperature control systems.

Table 10-47
Heat Transmittance from Tracers through Heat Transfer Cement to Process Pipe[4]

NPS	q_t
1	34.3
1.5	34.3
2	32.6
2.5	32.6
3	29.1
4	26.9
6	23.8
8	21.5
10	18.4
12	14.6
14	12.2
16	9.8
18	9.8
20	9.8

[4]Note: Reference 4 is to Foo's article's literature citation. Symbols: NPS = nominal pipe size, in.; q_t = Btu/(hr)(ft^2)(°F).

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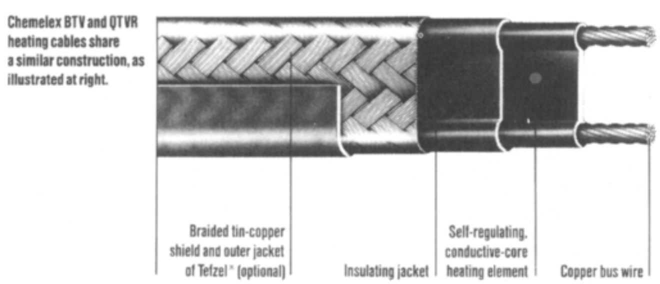


Chemelex® heating systems consist of insulated, electric heating cables with voltage applied to two parallel bus wires. Because of this parallel construction, all Chemelex® heating cables can be cut to any length and spliced and "teed" in the field.

Figure 10-169. Self-regulating heat tracer for pipe and vessels. Some simpler designs have temperature monitoring and power control. (Used by permission: Bul. (P6909) H53398 4/94. ©Raychem Corporation, Chemelex® Division.)

H. Heat Loss for Bare Process Pipe

Table 10-48 presents a tabulation of heat loss from the outside surface of bare standard pipe.

Heat loss through wall of uninsulated pipe:[70]

$$q = \frac{2\pi k(t_i - t_o)}{2.3\log(D_o/D_i)},\ \text{Btu/lin ft} \qquad (10\text{-}268)$$

where D = pipe diameter, in.
t = temperature, °F
q = heat loss through wall, Btu/lin ft
k = thermal conductivity of pipe wall, Btu/(hr) (ft^2)(°F/ft)
i = inside wall pipe
o = outside wall surface of pipe

Heat loss from fluid inside pipe through exterior insulation to outside air.[70] Combined convection and radiation:

$$q = \frac{\pi(t_s - t_a)}{(2.3/2k_c)\log(D_1/D_s'') + 1/(h_a D_1)},\ \text{Btu/(hr)(lin ft)} \qquad (10\text{-}269)$$

where s = inside surface of pipe
h_a = surface coefficient of heat transfer, Btu/(hr) (ft^2)(°F/ft)
k = thermal conductivity of insulation, Btu/(hr) (ft^2)(°F/ft)
D = pipe O.D., ft
D_1 = insulation O.D., ft
a = bulk fluid outside insulated pipe
q = heat loss per linear foot of pipe, Btu/(hr) (lin ft)

Selected Values for k, Thermal Conductivity of Insulation*

Material	k, Btu/(hr) (ft^2) (°F/ft)
Mineral wool	0.28
Foam glass	0.43
Calcium silicate	0.38
Magnesia, 85%	0.38
Glass	0.59-0.79
Glass wool	0.022

*Compiled from references 284 and 223.

Heat loss through the walls of the insulation is[221]

$$q = k\,\Delta t_i/X = h\,\Delta t_o \qquad (10\text{-}270)$$

For heat loss from bare standard NPS pipe, see Table 10-48.[220]

For pipe insulation,

heat flow between the inside surface of pipe insulation and the outside air at outside surface of pipe insulation:[248]

Rate of heat transfer,

$$q_s = \frac{t_o - t_a}{[r_s \log_e(r_1/r_o)]/k_1 + [r_s \log_e(r_s/r_1)]/k_2 + R_s} \qquad (10\text{-}271)$$

where q_s = rate of heat transfer per ft^2 of outer *surface* of *insulation*, Btu/(hr) (ft^2)
k = thermal conductivity of insulation at mean temperature, Btu/(hr) (ft^2)(°F/in.)
r_o = inside radius of pipe insulation, in.
r_s = outside radius of pipe insulation, in.

Table 10-48
"Q" Heat Loss from Bare NPS Pipe, Btu/(lin ft) (hr)

Ambient Air Temperature 70°F, Natural Circulation

Pipe Temperature, °F (English Units)

NPS Pipe	100	200	300	400	500	600	700	800	900	1,000	1,100	1,200	Pipe dia. mm
1/2	13	75	165	287	444	649	901	1,218	1,602	2,075	2,644	3,317	**21.3**
3/4	16	93	204	353	547	801	1,113	1,508	1,984	2,576	3,282	4,122	**26.7**
1	20	114	250	433	674	989	1,379	1,865	2,462	3,194	4,080	5,123	**33.4**
1 1/4	24	141	312	541	843	1,237	1,728	2,342	3,091	4,010	5,123	6,433	**42.2**
1 1/2	27	159	352	613	955	1,403	1,960	2,661	3,514	4,568	5,841	7,345	**48.2**
2	33	196	432	753	1,176	1,732	2,423	3,295	4,355	5,665	7,251	9,133	**60.3**
2 1/2	40	233	516	899	1,408	2,077	2,907	3,956	5,235	6,817	8,732	11,005	**73.0**
3	48	280	620	1,083	1,695	2,505	3,511	4,784	6,337	8,259	10,582	13,344	**88.9**
3 1/2	55	317	701	1,226	1,922	2,841	3,987	5,434	7,201	9,390	12,039	15,189	**101.6**
4	61	354	784	1,370	2,149	3,179	4,467	6,094	8,079	10,545	13,513	17,049	**114.3**
6	87	503	1,122	1,976	3,105	4,604	6,479	8,858	11,769	15,366	19,740	24,909	**168.3**
8	111	644	1,436	2,530	3,988	5,927	8,356	11,433	15,211	19,895	25,568	32,293	**219.1**
10	136	791	1,769	3,114	4,918	7,312	10,325	14,147	18,812	24,621	31,679	40,051	**273.0**
12	159	930	2,076	3,664	5,809	8,627	12,194	16,721	22,248	29,133	37,501	47,429	**323.3**
14	174	1,009	2,258	3,989	6,316	9,404	13,301	18,236	24,279	31,844	40,965	51,856	**355.6**
16	197	1,142	2,560	4,529	7,160	10,697	15,124	20,769	27,663	36,257	46,689	59,089	**406.4**
18	221	1,282	2,873	5,074	8,032	12,010	16,992	23,346	31,109	40,788	52,539	66,456	**457.2**
20	244	1,416	3,168	5,618	8,897	13,270	18,777	25,810	34,432	45,147	58,159	73,691	**500.8**
24	289	1,683	3,772	6,679	10,595	15,825	22,375	30,836	41,112	53,958	69,536	88,221	**609.6**
30	351	2,042	4,642	8,242	13,103	19,606	27,758	38,299	51,107	67,125	86,558	109,872	**762.0**
36	421	2,450	5,570	9,890	15,724	23,527	33,309	45,959	61,328	80,550	103,860	131,846	**914.4**
	38	**93**	**149**	**205**	**260**	**315**	**371**	**423**	**482**	**539**	**593**	**649**	

Pipe Temperature, °C (Metric Units)

Used by permission: Turner, W. C., and Malloy, J. F. *Handbook of Thermal Insulation Design Economics for Pipe and Equipment*, ©1980. R. E. Krieger Publishing Company. Joint edition with McGraw-Hill Book Company, Inc. All rights reserved.

r_1 = outside radius of *any* (if used) intermediate layer of insulation, in.
R_s = outside surface resistance, (°F) (hr) (ft^2)/Btu
Δt = temperature difference ($t_1 - t_{ave}$) between inside surface of pipe insulation and average outside air temperature, °F
L = thickness of insulation, in.
t_a = temperature of ambient air, °F
t_o = temperature of inner surface of insulation, °F
t_s = temperature of outer surface of insulation, °F
X = insulation thickness, ft

Note: For k, subscript 1 = first (inner) layer of insulation. If more than one has a different k value, subscript 2 = second layer of insulation if different than first layer.

Heat flow per ft^2 of *pipe* surface $q_o = q_s(r_s/r_o)$, Btu/(hr) (ft^2)

I. Heat Loss through Insulation for Process Pipe

An alternate presentation of Chapman and Holland[227] is useful.

Heat transfer from the surface of an insulated or uninsulated pipe in air involves convection and radiation. In still air more heat is lost by radiation than convection. The heat loss from an insulated or bare pipe is, in Btu/hr:

$$Q = h_a' A_a (T_s - T_a) \qquad (10\text{-}272)$$

where T_s = surface temperature of insulated or bare pipe in contact with air, °F
h_a' = heat transfer film coefficient between the insulated or bare pipe and air. See Figure 10-170, assume ϵ = 0.90 and ambient air temperature = 70°F
$h'_a = h_c + \epsilon h_r$, Btu/(hr) ($ft^2$)(°F)

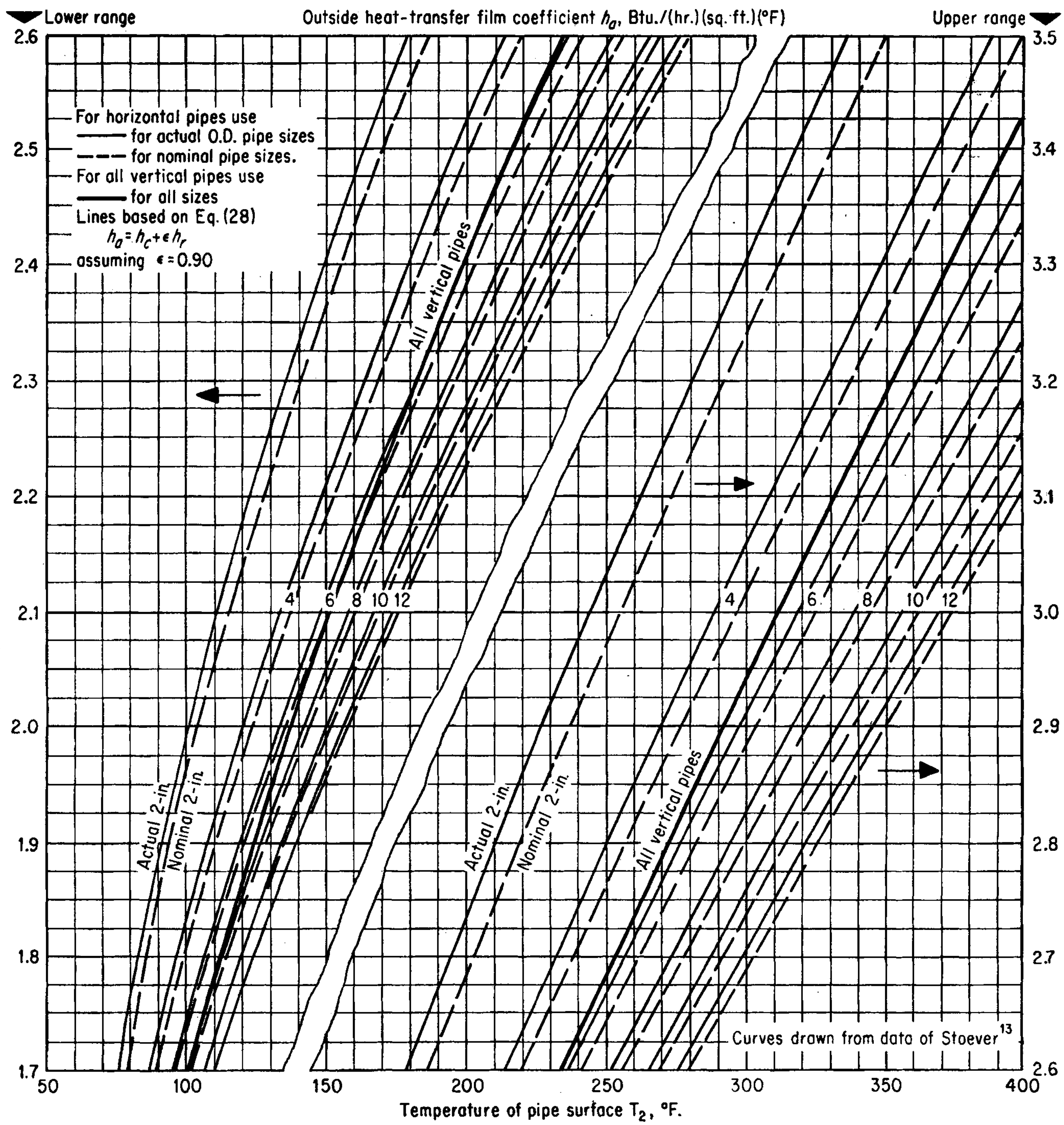


Figure 10-170. Outside heat-transfer film coefficient as function of pipe temperature and O.D. (Used by permission: Chapman, F. S., and Holland, F. A. *Chemical Engineering*, Dec. 20, 1965, p. 79. ©McGraw-Hill, Inc. All rights reserved.)

h_c = convection heat transfer film coefficient, Btu/(hr) (ft^2)(°F)

h_r = radiation heat transfer film coefficient, Btu/(hr) (ft^2)(°F)

ϵ = emissivity of the outside surface of insulated or bare pipe

T_a = ambient temperature, °F

D = O.D. of the insulated or bare pipe, whichever is being studied.

A_a = area of heat transfer between the insulation or bare pipe and air, ft^2

T′ = °R (degrees Rankine)

The authors[227] point out that the emissivity, ϵ, for many pipe surfaces ranges from 0.87–0.92 at approximately 70°F for highly polished aluminum, ϵ = 0.23–0.28.

For pipe exposed to wind velocities other than "calm," use Figure 10-171 to determine a value for h_c, which can be much greater than the "calm" values of 1.8–2.1 Btu/(hr)(ft^2)(°F).

To calculate h_r, per Kern:[70]

$$q = (\epsilon)\ (\text{area/lin ft})[[T_s'/100]^4 - [T_s'/100]^4], \quad \text{Btu/hr(lin ft)} \qquad (10\text{-}273)$$

σ = Stefan-Boltzmann constant = 0.173×10^{-8}, Btu/(hr) (ft^2)((R^4)

$$h_r = q/A, \text{ft}^2/\text{lin ft})\ (T_s - T_r), \text{Btu/(hr) (ft}^2\text{)(°F)} \qquad (10\text{-}274)$$

Because pipe heat loss can be an expensive cost for many process plants, Figure 10-172 illustrates a rapid solution to many situations. Ganapathy[218] summarizes his analysis by use of this figure.

Example 10-25. Determine Pipe Insulation Thickness[218]

Used by permission of Ganapathy, V.[218] (Follow dotted line on Figure 10-171.)

Determine the thickness of insulation to limit heat loss to 60 Btu/ft^2-hr in a 3-in. NPS pipe. Pipe temperature t_l is 580°F; t_a, the ambient temperature, is 80°F. Insulation K value is 0.5 Btu/ft^2-hr-°F/in., and outside film coefficient is 2.0 Btu/ft^2-hr-°F. (See table that follows.) What is the surface temperature of the insulation?

For the solution, connect $(t_l - t_a)$ = 500 with Q = 60 and extend to cut line 1 (dashed lines) at A. Connect f = 2.0 with point A and extend to cut line 2 at B. Connect B with K = 0.5 to cut line 3 at C. The horizontal from C and the vertical from pipe size 3 intersect the curve corresponding to t = 2.5 in. Therefore, the solution is 2.5 or the next standard size of insulation.

From the equation, $(t_s - t_a) = Q/f$, $(t_s - t_a) = 60/2 = 30$. Therefore, $t_s = 30 + 80 = 110$°F.

The effect of using a different thickness of insulation and the corresponding heat loss can easily be calculated. For example, using 2-in. thick insulation, we see that (solid lines) Q = 78 Btu/ft^2-hr, and surface temperature increases to (78/2 + 80) = 119°F.

Values of f*

Still air	1.2–1.8
7.5 mph wind	2.0–4.0
15 mph wind	3.5–5.0

*Btu/(ft^2) (hr)(°F); for preliminary estimates, use 2.0.

where f = outside film coefficient, Btu/(hr) (ft^2)(°F)
Q = heat loss, Btu/(ft^2) (hr), from pipe insulation
t_a, t_s, t_l = ambient, surface and pipe temperatures, °F
k = thermal conductivity of insulation, Btu/(ft^2) (hr)(°F/in.)

Heilman[219] presents a thorough discussion of heat loss from bare and insulated surfaces.

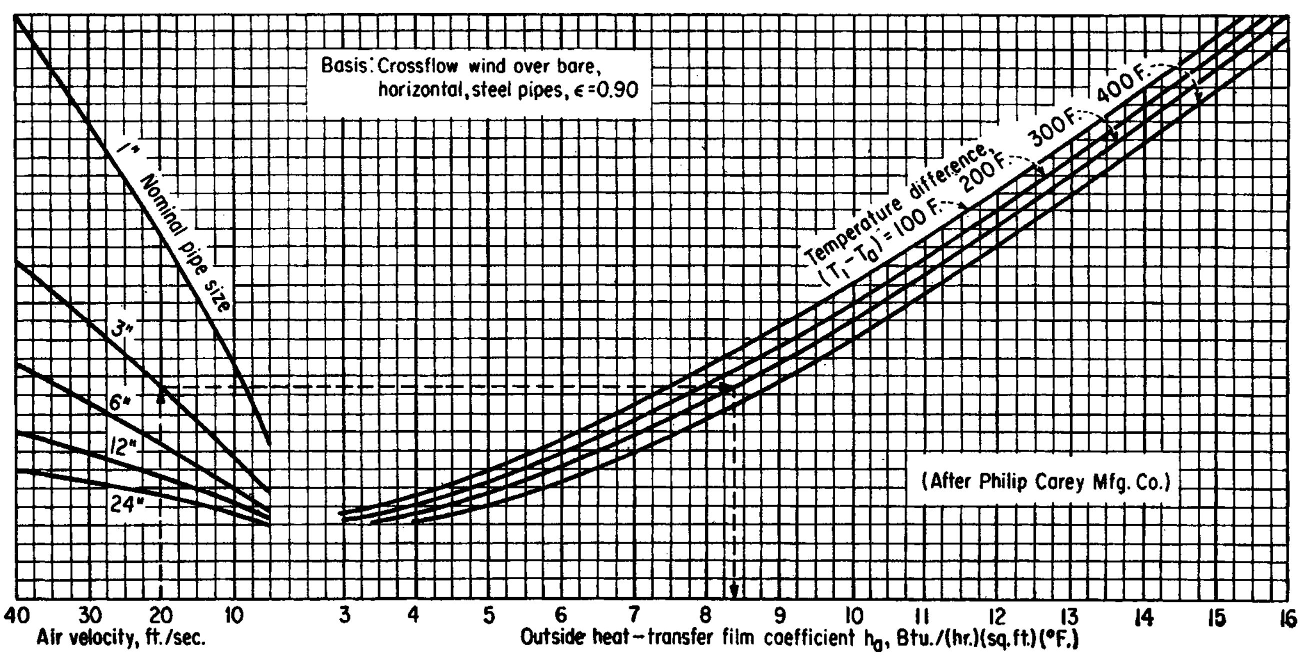


Figure 10-171. How air velocity over heated pipe increases heat transfer through forced convection. (Used by permission: Chapman, F. S., and Holland, F. A. *Chemical Engineering*, Dec. 20, 1965, p. 79. ©McGraw-Hill, Inc. All rights reserved.)

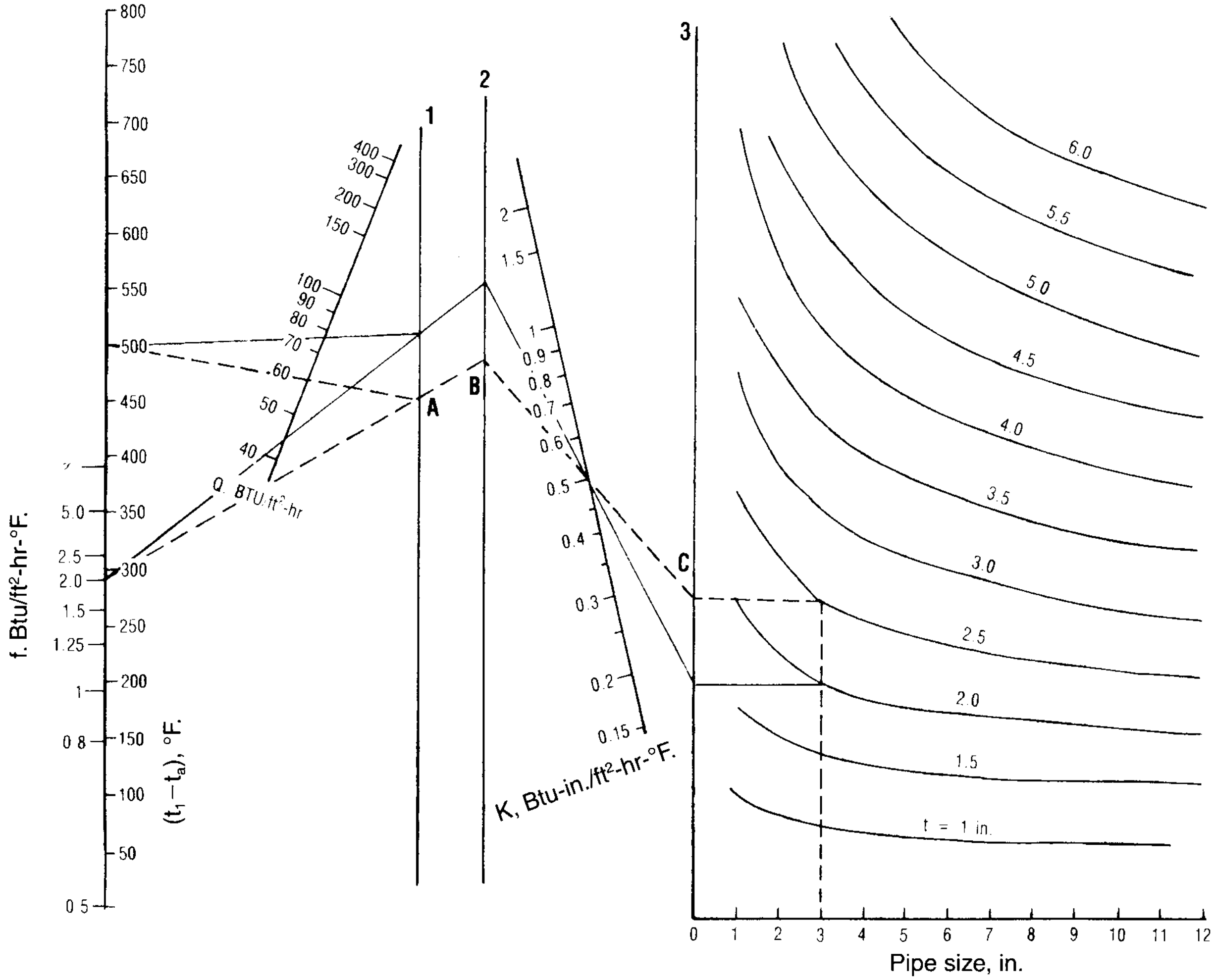


Figure 10-172. Heat loss through process pipes and insulation. (Used by permission: Ganapathy, V. *Oil and Gas Journal*, Apr. 25,1983, p. 75. ©PennWell Publishing Company. All rights reserved.)

J. Direct-Contact Gas-Liquid Heat Transfer

The direct counter-current contact of a hot gas with a cool immiscible liquid is effectively used in certain hydrocarbon cracking processes for the quenching of hot gases/vapors. Sometimes, the liquid used is oil and followed by water quench, as is typical in ethylene plants cracking naphtha or other hydrocarbon as feed stock.

The three primary devices used in this service are (a) open spray columns,[242, 245] (b) packed columns,[242, 243, 244, 245] and (c) tray columns, which are perforated plates, baffle trays, etc.[242, 245, 246, 247, 248, 249]

Fair[242] reports that the data for mass transfer in spray, packed, and tray columns can be used for heat-transfer calculations for these columns. The pressure drop in these types of columns is usually quite low.

A. Spray Columns

Data correlated by Fair[242] provides an empirical relationship for heat transfer:

$$h_{ga} = \frac{0.015 G^{0.82} L^{0.47}}{Z_{sp}^{0.38}} \tag{10-275}$$

where Z_{sp} = height of a single zone of spray contact, which most likely is the space between spray nozzles when the full area coverage is achieved, ft.

G = superficial gas mass velocity, lb/(hr) (ft²)

L = superficial liquid mass velocity, lb/(hr) (ft²)

h_{ga} = gas phase volumetric heat transfer coefficient, Btu/(hr) (ft³)(°F)

The liquid phase resistance, h_{la}, is considered low when compared to the overall resistance; therefore, the h_{ga} should give a reasonable approximation to the overall resistance for the system,[242, 247] because $1/U_a = 1/h_{ga} + 1/h_{la}$.

B. Random Packed Columns

Fair[242] recommends the correlating relations from Huang[250] as shown in Table 10-49, which satisfies the relation.

Coefficient = $h_g a$, or $h_l a$, or $U_a = C_l\, G^m\, L^n$

where $h_g a$ = volumetric gas-phase heat transfer coefficient, Btu/(hr) (ft^3)(°F)
$h_l a$ = liquid-phase heat transfer coefficient, Btu/(hr) (ft^3)(°F)
U_a = volumetric overall heat transfer coefficient, Btu/(hr) (ft^3)(°F)
G = superficial gas mass velocity, lb/(hr) (ft^2)
L = superficial liquid mass velocity, lb/(hr) (ft^2)

Table 10-49
Heat Transfer Coefficients for Packed Columns

Coefficient* = $C_l G^m L^n$					
Packing	**System**	**Coefficient**	**C_l**	**m**	**n**
RR–1 in.	Air/water	$h_l a$	0.774	0.51	0.63
	Air/water	$h_g a$	0.230	1.10	0.02
	Air/oil	U_a	0.00026	1.69	0.51
RR–1.5 in.	Air/water	$h_l a$	0.738	0.48	0.75
	Air/water	$h_g a$	0.008	1.45	0.16
	Air/oil	U_a	0.0016	1.49	0.38
IS–1 in.	Air/water	$h_l a$	2.075	0.20	0.84
	Air/water	$h_g a$	0.095	1.01	0.25
	Air/oil	U_a	0.0045	1.32	0.43
IS–1.5 in.	Air/water	$h_l a$	6.430	0.20	0.69
	Air/water	$h_g a$	0.019	1.38	0.10
	Air/oil	U_a	0.003	1.44	0.36
PR–1 in.	Air/water	$h_l a$	0.296	0.45	0.87
	Air/water	$h_g a$	0.019	1.12	0.33
	Air/oil	U_a	0.0013	1.47	0.46
PR–1.5 in.	Air/water	h_{la}	1.164	0.31	0.80
	Air/water	h_{ga}	0.011	1.28	0.26
	Air/oil	U_a	0.027	1.07	0.36

*$h_g a$ or $h_l a$ or U_a, Btu/(hr-ft^3-°F)

Symbols

RR 1 in., RR 1.5 in. } Ceramic Raschig rings, 1-in. and 1.5-in. nominal size

IS 1 in., IS 1.5 in. } Ceramic Intalox saddles, 1-in. and 1.5-in. nominal size

PR 1 in., PR 1.5 in. } Metal Pall rings, 1-in. and 1.5-in. normal size

Used by permission: Fair, J. R. ASME Solar Energy Division Conference, April 1989. ©American Society of Mechanical Engineers, San Diego, CA.

α = Ackerman correction factor, dimensionless, source unknown.

For little or no condensation in the system:

$$1/U_a = 1/h_g a + 1/h_l a$$

For condensation:

$$1/U_a = 1/\alpha h_g a + (1/h_l a)(Q_s/Q_T)$$

Sc = Schmidt number, dimensionless
Pr = Prandtl number, dimensionless
c_g = gas specific heat, Btu/lb-°F
a = interfacial area, ft^2/ft^3
Q_s = sensible heat transfer duty, Btu/hr
Q_T = total heat transfer duty, Btu/hr

C. Sieve Tray Columns

The thesis of Stewart[249] indicates that the overall liquid film and mass transfer coefficients were functions of the gas flow rate and the column pressure and are independent of the liquid flow rate and inlet air temperature. The gas film heat transfer coefficient was found to be a function only of the air flow rate.

From Fair[242] the gas phase coefficient is

$$h_g a = \frac{c_g G (Sc_g)^{2/3}}{H_{g,d}(Pr_g)} \tag{10-276}$$

and the heat transfer efficiencies range from 60–100%. Based on the gas phase, the height of a transfer unit, H_g, is[242]

$$H_{g,d} = \frac{G}{k_g\, a\, M_g\, P} \tag{10-277}$$

For nitrogen data:[242] $U_a = 0.213G^{1.0}$
For helium data:[242] $U_a = 1.05G^{1.0}$

D. Baffle Tray Column[242]

The contacting counterflow action provides a dependence on the liquid rate, similar in concept for packed columns:

$$H_g a = C_l G^m L^n \tag{10-277A}$$

where C_l = coefficient which depends on the system used, for example, C_l = 2.058 for nitrogen/absorption oil
h_g = heat transfer coefficient, J/m^3sk
a = interfacial area, n^2/m^3, or ft^2/ft^3
c = specific heat, Btu/(lb) (°F)
G = superficial gas mass velocity, lb/(hr) (ft^2)
h = heat transfer coefficient, Btu/(hr) (ft^2)(°F)

h_ga = volumetric gas phase coefficient, Btu/(hr) (ft^3)(°F)
$H_{g,d}$ = height of a gas phase mass transfer unit, ft
$H_{l,d}$ = height of a liquid phase mass transfer coefficient, ft
k_g = gas phase mass transfer coefficient, lb-mol/(hr)(ft^2)(atm)
L = superficial liquid mass velocity, lb/(hr) (ft^2)
M = molecular weight
m = exponent in baffle tray columns
= 1.18, experimental value for system studied
n = exponent in baffle tray columns = 0.44
P = pressure, atm
Pr = Prandtl number, dimensionless
Q = heat transfer duty, Btu/hr
Sc = Schmidt number, dimensionless
U_a = volumetric overall heat transfer coefficient, Btu/(hr) (ft^3)(°F)
U = overall heat transfer coefficient, Btu/(hr) (ft^2)(°F)
Z = height, ft
Z_{sp} = height of individual spray zone, ft
ρ = density, lb/ft^3

Subscripts
d = diffusional
g = gas
l = liquid

Smith[248] presents a design for this type of tray direct contact column, summarized as shown in Figure 10-173. Also see Vol. 2, 3rd Ed., Chapter. 8, of this series for design details.

When vapor stream has lower heat capacity than liquid stream ($\Delta T_v > \Delta T_L$), use[248]

$$H_v = \Delta T_v/\Delta T_L \tag{10-278}$$

$$H_v^* = (H_v^{n+1} - H_v)/(H_v^{n+1} - 1) = \Delta T_v/\Delta T_{v,max} \tag{10-279}$$

$$H_v^* = (H_v^{n+1} - H_v)/(H_v^{n+1} - 1.0), \text{ solve for n, number of equilibrium stages} \tag{10-280}$$

When liquid stream has lower heat capacity than vapor stream ($\Delta T_L > \Delta T_v$) use[248]

$$H_L = \Delta T_L/\Delta T_v \tag{10-281}$$

$$H_L^* = (H_L^{n+1} - H_L)/(H_L^{n+1} - 1) = \Delta T_L/\Delta T_{L,max} \tag{10-282}$$

$$H_L^* = (H_L^{n+1} - H_L)/(H_L^{n+1} - 1.0), \text{ solve for n.} \tag{10-283}$$

Example 10-26. Determine Contact Stages Actually Required for Direct Contact Heat Transfer in Plate-Type Columns

Used by permission: Smith, J. H. *Hydrocarbon Processing*, V. 58, No. 1, ©1979.

How many theoretical contact stages are required for a side reflux system on an atmospheric crude tower? The vapor is to be cooled from 500°F to 440°F; the circulating distillate is to be heated from 325°F to 475°F. Because $\Delta T_L > \Delta T_v$, use Equations 10-281 and 10-282.

$$H_L = (475 - 325)/(500 - 440) = 2.50$$
$$H^*_L = (475 - 325)/(500 - 325) = 0.857$$
$$0.857 = (2.5^{n+1} - 2.5)/(2.5^{n+1} - 1.0)$$
$$n = 1.665$$

About 65% efficiency is to be expected in this service, requiring three actual trays.

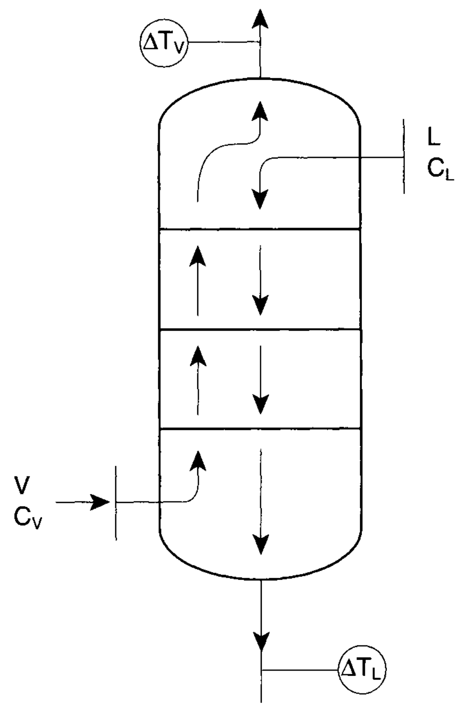


Figure 10-173. Direct contact tray column for heat transfer. This could be a baffle tray, sieve type tray, bubble or other contact device, or open spray or random packed column. (Symbols only used by permission: Smith, J. H. *Hydrocarbon Processing*, Jan. 1979, p. 147. ©Gulf Publishing Company. All rights reserved.)

where H_v = heat transfer factor, vapor limiting
H_L = heat transfer factor, liquid limiting
H_L^* = heat transfer efficiency, equals ratio of actual liquid temperature rise to maximum possible rise
H_v^* = heat transfer efficiency, equals ratio of actual vapor temperature decrease to maximum possible decrease
n = number of equilibrium contact stages
ΔT_v = actual vapor temperature decrease
$\Delta T_{v,max}$ = maximum possible vapor temperature decrease (to liquid inlet temperature)
ΔT_L = actual liquid temperature rise
$\Delta T_{L,max}$ = maximum possible liquid temperature rise (to vapor inlet temperature.)

E. Baffle Tray Column (or, Termed Shower Deck, No Holes, Caps, or Other Contact Devices)

For counter flow, gas flowing up a column through a falling shower film of liquid, Fair's correlation[242] of collected data is to be used as a guide:

$$U_a = 0.011\ G^{1.04}\ L^{0.3} \qquad (10\text{-}284)$$

See Fair's reference given previously for nomenclature.

For baffle trays, the coefficient equation given under packed columns, the values of $m = 1.18$ and $n = 0.44$ with C_1 depending on the system. For example, for a nitrogen/absorption oil system, $C_1 = 0.00250$. See the reference and Table 10-48 for more details.

Air-Cooled Heat Exchangers

Air-cooled heat exchangers are very seldom, if ever, finally designed by the user company (or engineering design contractor), because the best final designs are prepared by the manufacturers specializing in this unique design and requiring special data. This topic is presented here to aid the engineer in understanding the equipment and applications, but not to provide methods for preparing final fabrication designs.[106, 206, 251, 252, 253, 254, 255, 256, 257, 258, 259, 260, 261, 262, 263, 264, 265] Standard 661, 3rd Ed., American Petroleum Institute, "Air Cooled Heat Exchangers for General Refinery Services" is a good basic reference.

Air-cooled exchangers use atmospheric air on the outside of high-finned tubes (except bare tubes are used in a few applications) to cool or condense fluids flowing through the inside of the tubes.

This type of exchanger is used to reject heat from a fluid inside the tubes (and associated headers) directly to ambient air.[251] To be effective, the air must flow in forced convection to develop acceptable transfer coefficients. Figures 10-174, 10-175, and 10-176 illustrate the two types, designated by the type of air movement, induced draft or forced draft.

The advantages and disadvantages of forced and induced draft fan operation on the performance of the unit as presented by Hudson Products Corp.[251] are used by permission in the following discussions.

Induced Draft.

Advantages:

1. Better distribution of air across the bundle.
2. Less possibility of hot effluent air recirculating into the intake. The hot air is discharged upward at approximately 2.5 times the intake velocity, or about 1,500 ft per min.
3. Better process control and stability because the plenum covers 60% of the bundle face area, reducing the effects of sun, rain, and hail.

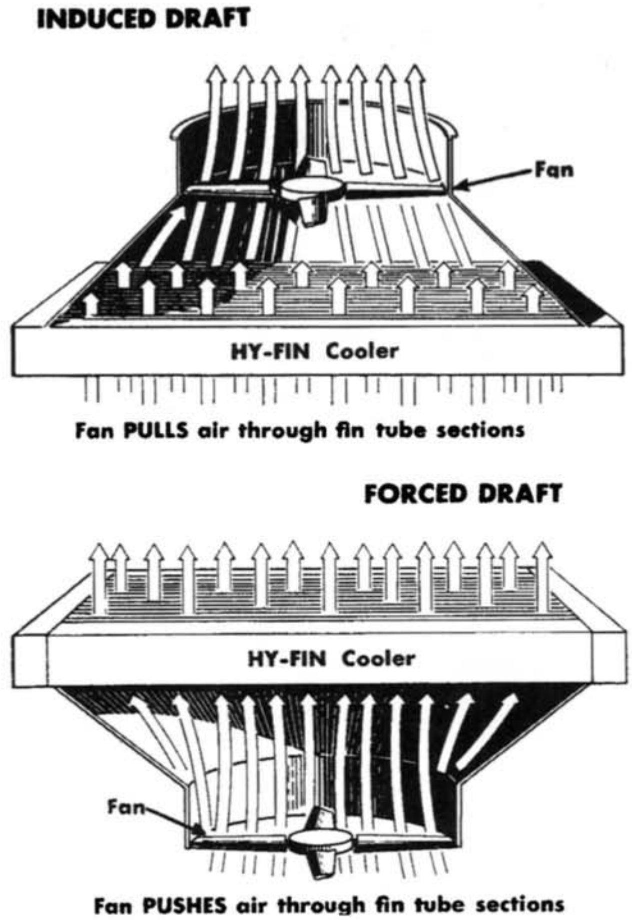


Figure 10-174. Two types of air-cooled heat exchangers. (Used by permission: © Hudson Products Corporation.)

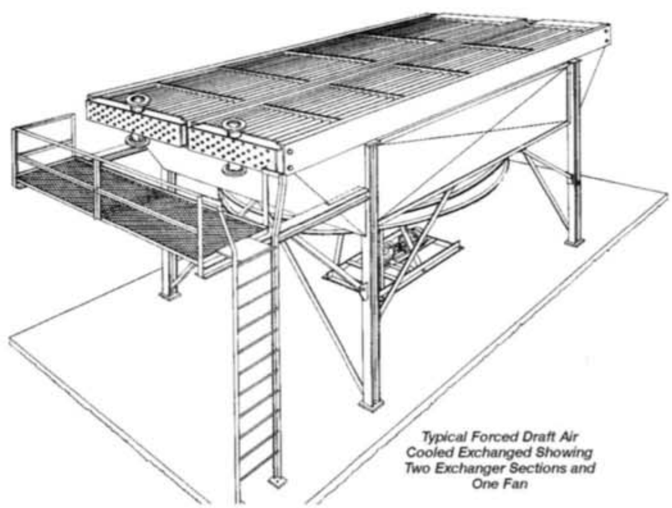


Figure 10-175. Typical forced draft air-cooled exchanger showing two exchanger sections and one fan. (Used by permission: Yuba Heat Transfer Division of Connell Limited Partnership.)

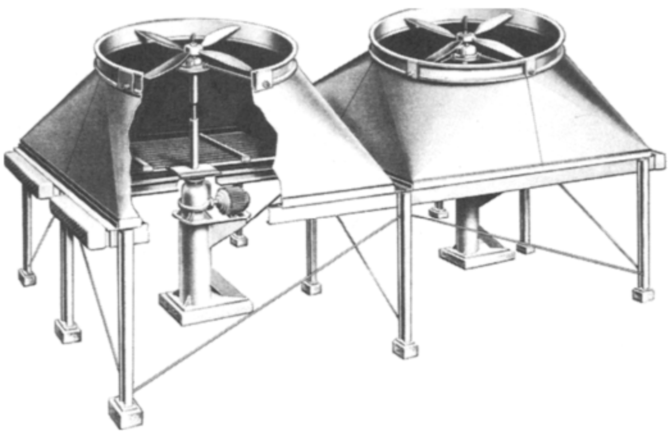


Figure 10-176. Typical induced draft air-cooled exchanger showing two exchanger sections and two fans. (Used by permission: Griscom-Russell/Ecolaire Corporation, Easton, PA.)

4. Increase capacity in the fan-off or fan-failure condition, because the natural draft stack effect is much greater.

Disadvantages and limitations:

1. Possibly higher horsepower requirements if the effluent air is very hot.
2. Effluent air temperature should be limited to 220°F to prevent damage to fan blades, bearing, or other mechanical equipment in the hot airstream. When the process inlet temperature exceeds 350°F, forced draft design should be considered because high effluent air temperatures may occur during fan-off or low air flow operations.
3. Fans are less accessible for maintenance, and maintenance may have to be done in the hot air generated by natural convection.
4. Plenums must be removed to replace bundles.

Forced Draft.
Advantages:

1. Possibly lower horsepower requirements if the effluent air is very hot. (Horsepower varies inversely with the absolute temperature.)
2. Better accessibility of fans and upper bearings for maintenance.
3. Better accessibility of bundles for replacement.
4. Accommodates higher process inlet temperatures.

Disadvantages:

1. Less uniform distribution of air over the bundle.
2. Increased possibility of hot air recirculation, resulting from low discharge velocity from the bundles, high intake velocity to the fan ring, and no stack.
3. Low natural draft capability on fan failure.
4. Complete exposure of the finned tubes to sun, rain, and hail, which results in poor process control and stability.

Hudson[251] states that the advantages of the induced draft design outweigh the disadvantages.

Although most units are installed horizontally, inclined, Figure 10-177, and vertical units are also in service. Figures 10-178 and 10-179 show typical assemblies for tube bundles with fabricated or cast end headers and also with flanged cover plates.

The tube bundle is an assembly of tubes rolled into tubesheets and assembled into headers. See Figures 10-175, 10-176, 10-178, 10-179 and 10-180. The usual headers are plug and cover plate but can accommodate U-bend types if the design so dictates.

The headers may be

1. Cast box type, with shoulder or other plugs opposite every tube. The shoulder plug is generally considered best for most services. The hole of the plug provides access to the individual tubes for (a) cleaning, (b) rerolling to tighten the tube joint, and (c) plugging the tube in case of singular tube leaks.
2. Welded box type, same features as (1).
3. Coverplate type using flat or confined gasket. This type provides complete access to all tubes upon removal of bolted coverplate. This is used for fouling or plugging services where frequent cleaning is necessary.
4. Manifold type, which is used in high pressure and special applications.[16, 18]

For heat transfer performance, horizontal baffles to isolate tube-side passes in horizontal bundles are preferred over vertical baffles that isolate groups of tubes in vertical columns. The expansion of capacity by adding more tube bundles or sections in parallel is easier, and the MTD is better with the horizontal pass plates. The fan drive may be by any of the available means, including:

1. Direct electric motor or with belts.
2. Two-speed electric motor with belts or gears, gear or fluid coupling.
3. Steam turbine direct or with gear or fluid coupling.
4. Gasoline engine with belt, gear, or fluid coupling.
5. Hydraulic drive (see Figure 10-181).

Gears should be specified as American Gear Manufacturer's Association (AGMA) requirements for cooling tower service in order to ensure an adequate minimum service factor rating of 2.0. The spiral bevel type is probably used a little more often than the worm gear. It is also cheaper. When gears are used with induced draft applications, the

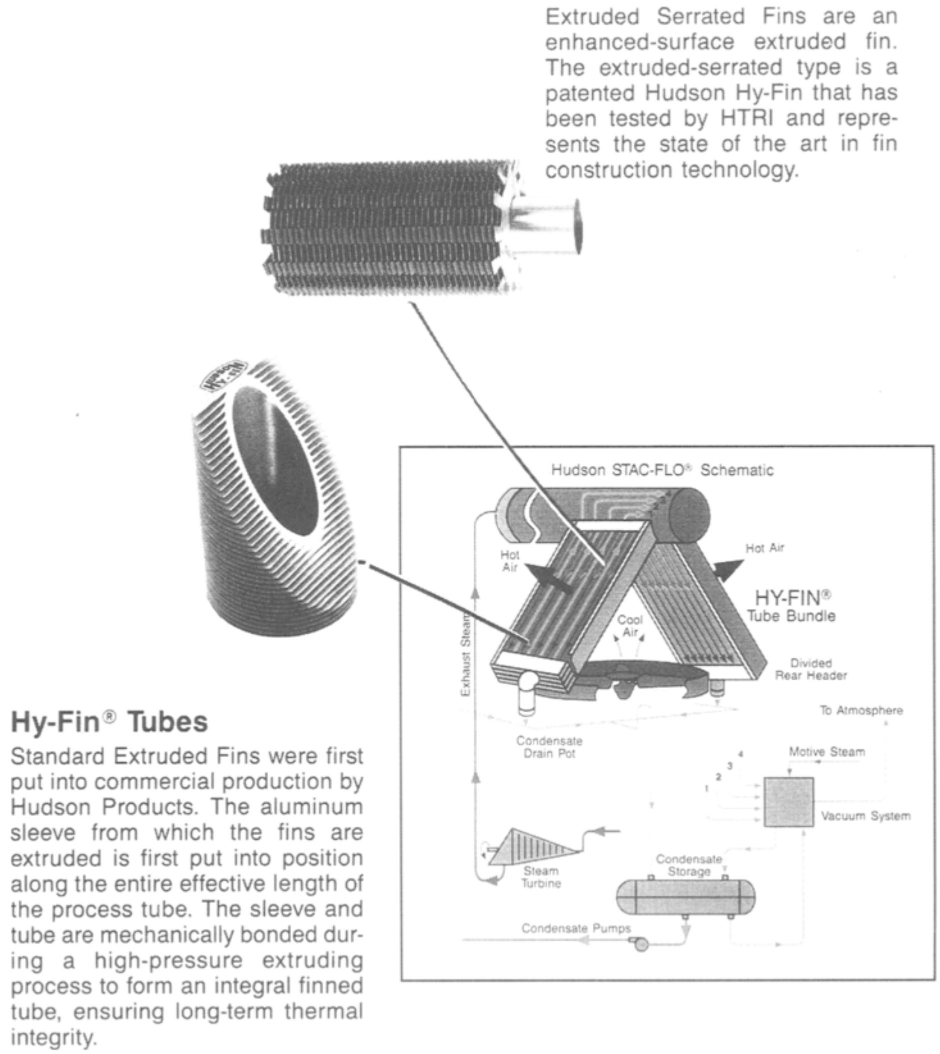


Figure 10-177. Air-cooled Stac-Flo® steam condensers illustrating process system. Representative types of tubes are illustrated. (Used by permission: Bul. M-390621 10/90. ©Hudson Products Corporation)

maximum temperature of the exit air must either be limited by specification, or the gears must be rated at the expected air temperature surrounding the case.[88] Remote lubrication should be provided for gears, bearings, etc., to prevent shutdown of the unit.

For V-belt drive, the type of belt section and maximum number of belts may be specified, as well as the minimum number —usually 3. B-sections are most common. V-belts are not considered for drives over about 50–60 hp, and a minimum service factor of 1.4 should be specified for continuous duty. Belts should not be used in any conditions where the surrounding temperature is greater than 160°F, with or without fans operating. This is of particular importance in induced draft conditions where belts might be in the exit air stream.

For general service, the fans are axial flow, propeller type with 2–20 blades per fan which force or induce the air across the bundle. Four blades are considered minimum, and an even number of blades (2–20) are preferable to an odd number (for emergency removal of blades to obtain balance for continued partial operation.)

Fan diameters range from 3–60 ft. The blades may be solid or hollow construction,[251] with the hollow design being the most popular.

The blades are usually fixed pitch up to 48-in. diameter with applications for adjustable pitch above this size. Fixed pitch is used up to 60-in. diameter with aluminum fan blades when direct-connected to a motor shaft. Variable pitch is used with belts, gears, etc., between the fan shaft and the driver to allow for the possibilities of slight unbalance between blades due to pitch angle variation. Aluminum blades are used up to 300°F, and plastic is limited to about 160°–180°F air stream temperature.

Air noise is usually less with multibladed fans (4 or more) than with 2 or 3 blades. In general, noise is not a real problem when associated with other operating machinery and when the frequency level is low and nonpenetrating. When

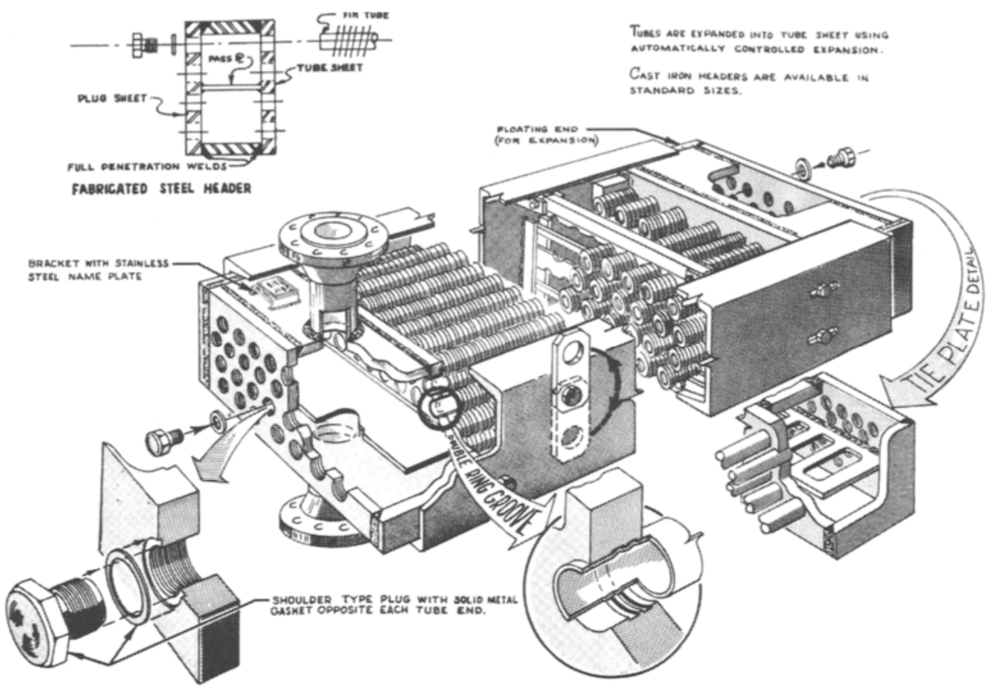


Figure 10-178. Typical tube bundle using fabricated or cast end headers. (Used by permission: Yuba Heat Transfer Division of Connell Limited Partnership.)

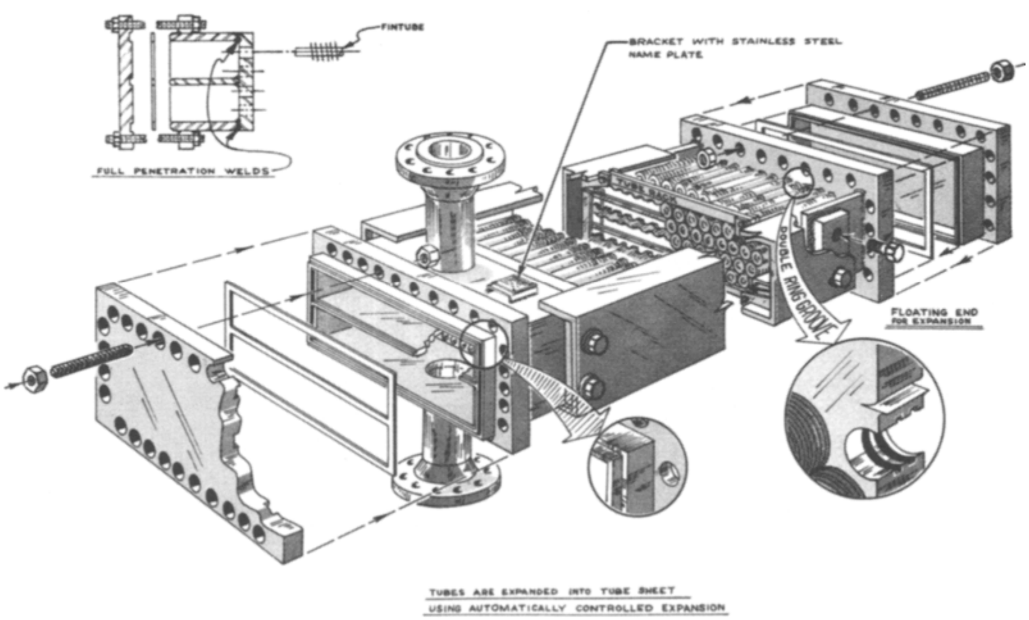


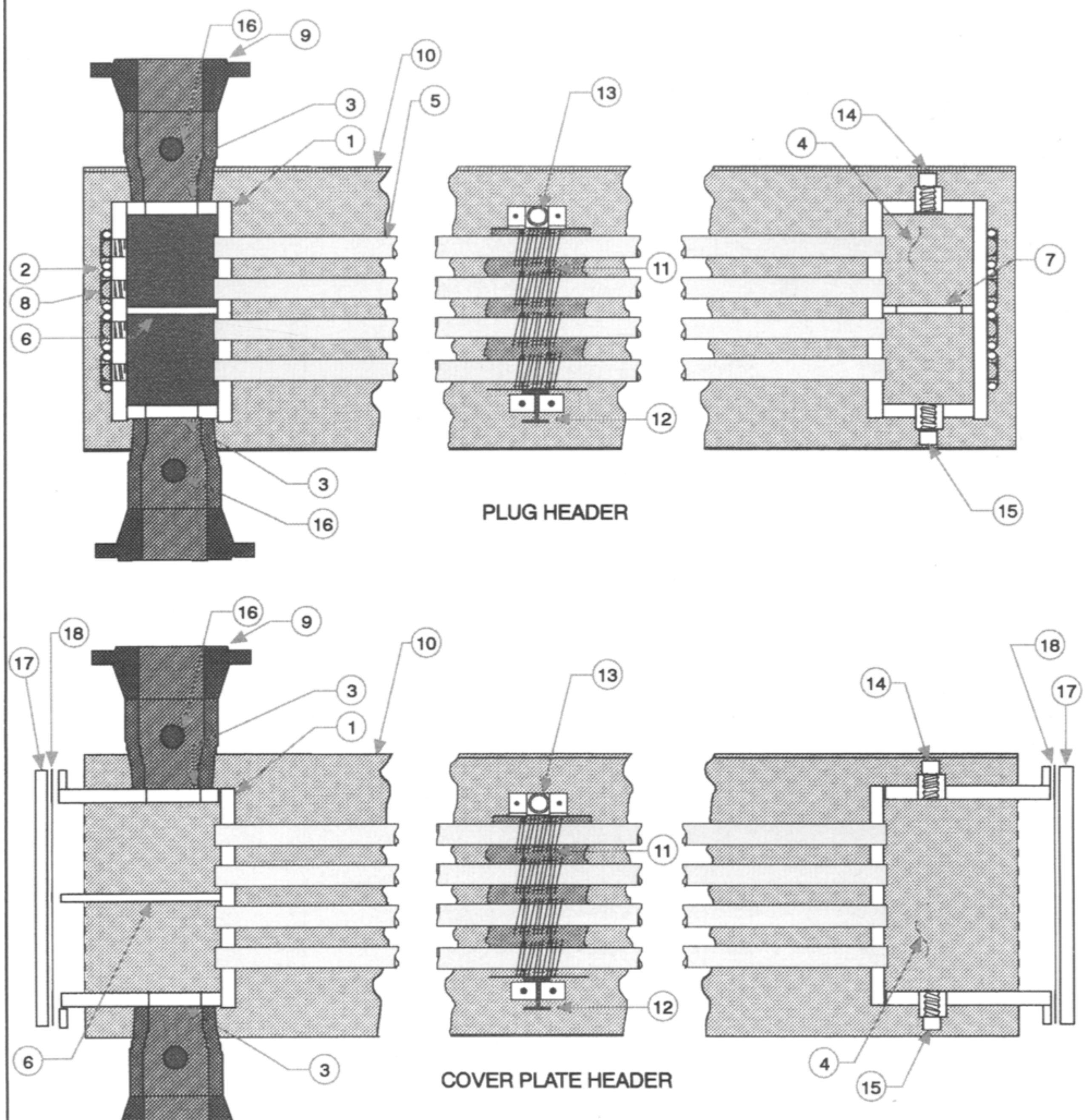
Figure 10-179. Typical tube bundle using flanged end cover plates. (Used by permission: Yuba Heat Transfer Division of Connell Limited Partnership.)

these units are isolated, the associated noise would be immediately noticeable but not objectionable unless confined between buildings or structures where reverberation could take place. The noise level is usually limited to 75 decibels maximum at 50 ft from the fan, and the blade tip speed is limited to 11,000–12,000 ft per min (= $\pi \times$ blade dia. in ft $\times$ rpm). This may run higher for units below 48-in. dia.

Figure 10-175 illustrates the assembly of a typical forced draft unit with electric motor and gear drive. Note that walkways and access ladders are necessary to reach the exchanger connections where valves are usually installed. If designs require a pipe inlet or outlet at each end of the tube bundle, walkways may be required at each end. Pipe layout studies are necessary when multiple sections (exchanger bundles) are placed in the same service.

The structural parts can be galvanized or pickled and painted to prevent rusting of the steel. The specifications will depend upon local requirements and experience.

Hail guards of stiff hardware cloth mounted in a removable frame are used to prevent hail damage to the relatively soft fins in hail-susceptible areas. If damaged just slightly, the performance is not impaired.



1. Tube Sheet
2. Plug Sheet
3. Top and Bottom Plates
4. End Plate
5. Tube
6. Pass Partition
7. Stiffener
8. Plug
9. Nozzle
10. Side Frame
11. Tube Spacer
12. Tube Support Cross-member
13. Tube Keeper
14. Vent
15. Drain
16. Instrument Connection
17. Cover Plate
18. Gasket

Figure 10-180. Typical construction of tube bundles with plug and cover plate headers. (Used by permission: Bul. M92-3003MC 10/94. ©Hudson Products Corporation.)

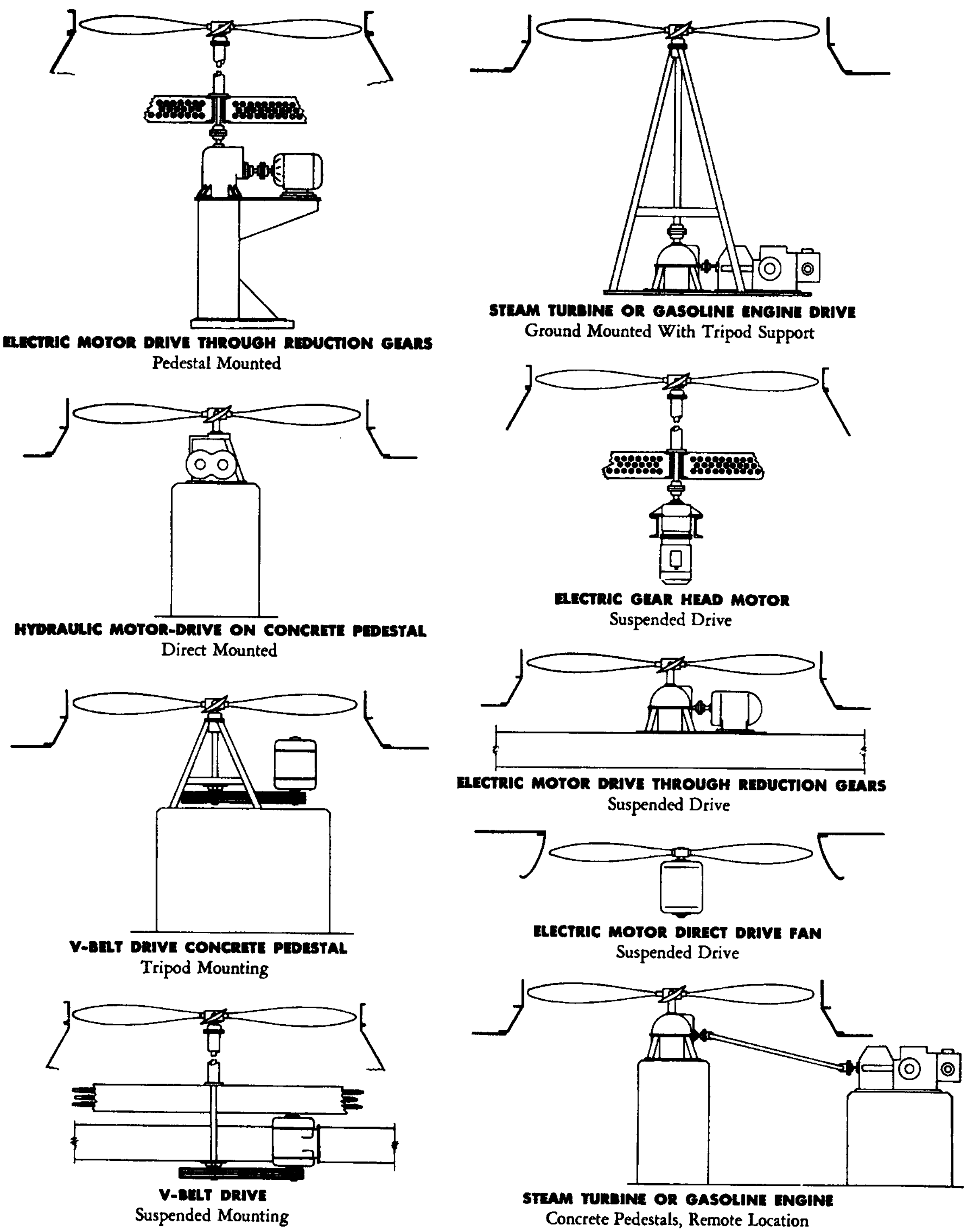


Figure 10-181. Typical drive arrangements for air-coolers. (Used by permission: Griscom-Russell/Ecolaire Corporation.)

Fan guards of wire grating or hardware cloth are mounted below the fan to prevent accidental contact with the moving blades and to keep newspapers, leaves, and other light objects from being drawn into the fan. The use of a wire fence around the entire unit is good to keep unauthorized individuals away from all of the equipment; however, a close fan guard, Figure 10-182, will prevent blade contact by the operators.

Tubes, Figure 10-183A and 10-183B, are usually finned with copper, aluminum, steel, or a duplex combination of steel inside with copper or aluminum fins outside. Other combinations are used to suit the service with the ratio of

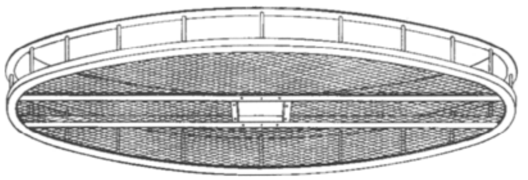


Figure 10-182. Fan blade guard mounted directly below blades. Note that drive shaft connects through the opening. (Used by permission: Bul. 107. SMITHCO Engineering, Inc.)

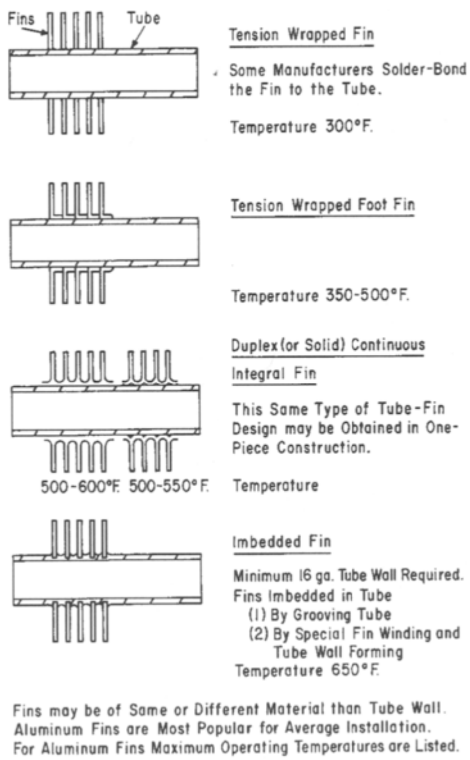


Figure 10-183A. Fin designs for use with air-cooled exchangers.

finned to bare tube surface of 15:1–20:1. Common sizes are $^3/_4$ -in. and 1-in. O.D. with $^1/_2$ -in. to $^5/_8$ -in. high fins, although 1 $^1/_2$-in. O.D. as well as small sizes are available for a specific design.

The minimum number of the tube rows recommended to establish a proper air flow pattern is 4, although 3 rows can be used.[265] The typical unit has 4–6 rows of tubes, but more can be used. Although more heat can be transferred by increasing the number of tubes, the required fan horsepower will be increased; however, this balance must be optimized for an effective economical design. Tubes are laid out on transverse or longitudinal patterns; however, the transverse is usually used due to the improved performance related to pressure drop and heat transfer.[265] The tube pitch is quite important for best air-side performance. A typical representative tube arrangement for design optimization is for bare-tube O.D., finned-tube O.D., and tube pitch:[256]

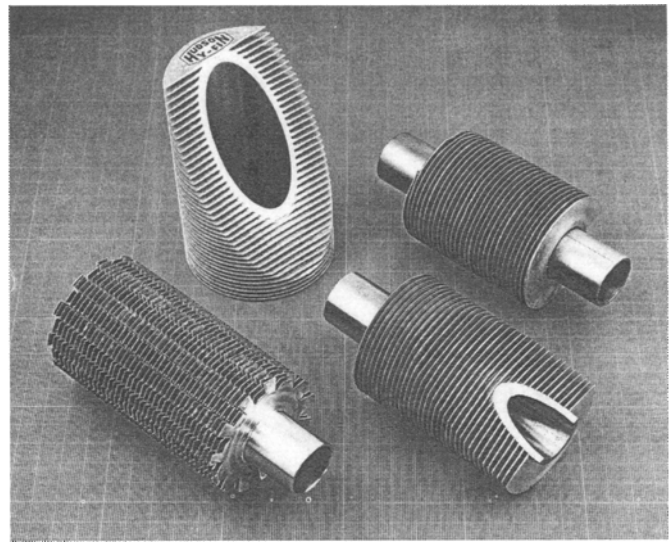


Figure 10-183B. Illustrations of actual fin construction. (Used by permission: Bul. B589-455, 6/89. ©Hudson Products Corporation.)

Counterclockwise from top: (1) Extruded fins offer high performance, reliability, and economy. (2) Hy-Fin extruded-serrated fins represent the state-of-the-art in fin tube construction technology. (3) Imbedded fins are recommended for applications involving high process temperatures. (4) L-base wrap-on fins offer low initial cost for applications involving low process temperatures.

1-in. / 2-in. / 2.375 in.
1-in. / 2.25 in./ 2.625 in.

For 1-in./2-in. (bare tube O.D./finned tube O.D.) the usual range for tube pitch is 2.125–2.5. For a 1-in./2.25-in tube, the pitch range would be 2.375–2.75. Reference 265 presents an interesting comparison of the effects of tube pitch on the heat transfer coefficient and pressure drop.

Tube lengths vary from 5 ft to more than 30 ft. Units for some heavy lube oils have been installed without fins due to the poor heat transfer inside the tube, i.e., the fins could not improve the overall coefficient above plain tubes. Economical tube lengths usually run 14–24 ft and longer. The performance of the tubes is varied for a fixed number of tubes and number of tube rows by varying the number of fins placed per lin in. on the bare tube. The usual number of fins/in. ranges from 7–11, with the lower number giving less total finned surface, ft^2 per lin ft of tube. Available extended or finned surface may be increased by changing the height of the fins from the usual $^1/_2$ -in. to $^5/_8$ -in.

When the fluid in the tubes yields a low film coefficient, the amount of finned surface area is adjusted, as suggested, to provide an economical and compatible area. A high ratio of outside finned surface to bare tube surface is of little value when the outside air and inside fluid coefficients are about the same. The tubes are usually on 2-in. or $^1/_2$ -in. triangular (60°) spacing. Fin thickness usually varies from

0.016–0.014 in. The effect of mechanical bond on heat transfer resistance is discussed by Gardner.[50]

It is helpful to the manufacturer for the purchaser to specify any conditions that are peculiar to the plant's warehouse stock of tubes or process controlled preferences:

1. Preferred bare tube O.D. and gage, giving minimum average wall thickness.
2. Seamless or resistance welded base tube.
3. Fin material preferred from atmospheric corrosion standpoint.

General Application

Air-cooled units have been successfully and economically used in liquid cooling for compressor engine and jacket water and other recirculating systems, petroleum fractions, oils, etc., and also in condensing service for steam, high boiling organic vapors, petroleum still vapors, gasoline, ammonia, etc. In general, the economics of application favors service allowing a 30–40°F difference between ambient air temperature and the exchange exit temperature for the fluid. These units are often used in conjunction with water-cooled "trim" coolers, i.e., units picking up the exit fluid from the air-cooled unit and carrying it down to the final desired temperature with water. In some situations, the air-cooled unit can be carried to within 20–25°F of the dry bulb air temperature if this is the desired endpoint rather than adding a small trim cooler. Kern[72] has studied optimum trim cooler conditions. As the temperature approach to the ambient air decreases, the power consumption increases rapidly at constant exchanger surface. This balance of first cost vs. operating cost is one of the key comparisons in evaluating these units.

Because surface area affects the first cost much more than the normally required horsepower (driver), the selection of the proper unit is a function of the relative change in these two items for a fixed heat duty. The optimum design gives the lowest total costs (first, operating, and maintenance) over the life of the unit, taken in many instances as 15 years or longer. Fan horsepower runs 2–5 hp per 10^6 Btu/hr.[63] First costs range from 25–150% of cooling tower systems with an average indicated at greater than 30%.[115]

Although these units find initial application in areas of limited water, they have not been limited to this situation. In many instances they are more economical than cooling tower systems and have been successfully applied in combination with cooling towers (see Figure 10-184). Economic comparisons should include such items as tower costs, basin, make-up facilities, water treatment, pumps for circulation, power supply, blow down, piping, etc. For small installations of air-cooled units, they should be compared

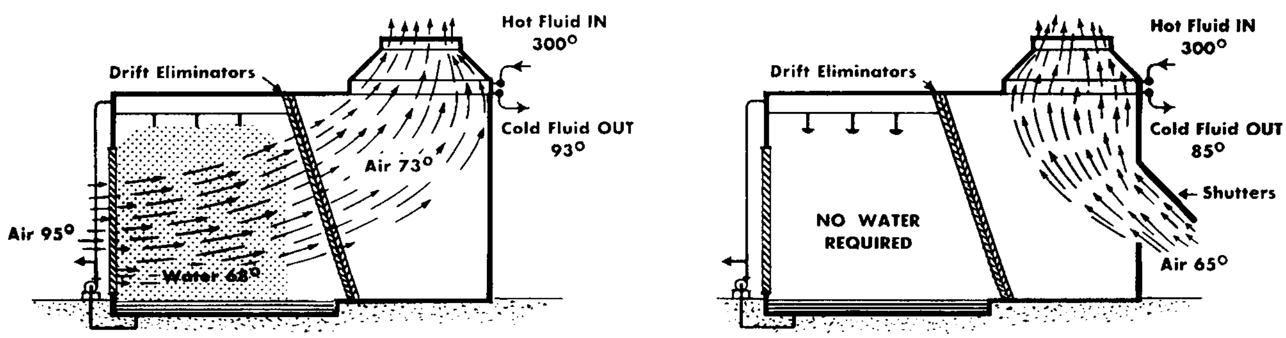


ILLUSTRATION OF SUMMER AND WINTER OPERATION OF THE COMBIN-AIRE (R)

SUMMER

1. **Relatively small quantity of water required, with no treatment necessary. Salt water may be used.**
2. **Cooled water may be used for other cooling purposes.**
3. **Because of elevated temperature, air leaving Combin-aire is under-saturated with water vapor, thus preventing spray carryover or misting.**

WINTER

1. **No water required.**
2. **Shutters may be made automatically responsive to air temperature, thus automatically controlling percentage of air pre-cooled.**
3. **No possibility of icing as encountered in winter operated cooling towers.**

(R) TRADEMARK, HUDSON ENGINEERING CORP.

Figure 10-184. Combined system using cooling tower and air-cooler units. (Used by permission: Hudson Products Corporation.)

with the prorata share of such cooling facilities unless the specific plant account of costs dictates otherwise.

The overall economics of an air-cooled application depends upon the following:

1. Quantity and quality of available water.
2. Ambient air and water temperature.
3. Fluid inlet as well as exit temperatures.
4. Operating pressure.
5. First costs.
6. Maintenance and operating costs.
7. Physical location and space requirements.

Mukherjee[265] presents an interesting examination of factors that can influence operating problems with air-cooled heat exchangers.

Advantages—Air-Cooled Heat Exchangers

1. Generally simple construction, even at relatively high pressure and/or high temperatures. Amount of special metals often is reduced.
2. No water problems, as associated with corrosion, algae, treating, scale, spray, etc.
3. Excellent for removing high level temperatures, particularly greater than 200°F.
4. Maintenance generally claimed to be $^1/_3$ or less than water coolers. Clean fins by compressed air and brushes, sometimes while operating.
5. Lower operating costs under many conditions, depending upon the type of water system used for comparison.
6. Ground space often $\leq$ cooling towers; can also serve dual purpose by mounting air-cooled units above other equipment or on pipe ways or roofs of buildings. Vibration is no problem.

Disadvantages

1. Rather high limitation on outlet fluid temperature.
2. Generally most suitable only for liquids or condensing vapors in tubes, with limited application for gas cooling due to low inside coefficient.
3. First capital costs may range from only 25–125% above water-cooled equipment for same heat load. Each situation must be examined on a comparative basis.
4. Fire and toxic vapor and liquid hazard, if leaks occur to atmosphere.
5. Not too suitable for vacuum services due to pressure drop limitations but are used in application.

Chase[24] lists these factors affecting the overall costs:

1. Exchanger Sections
   a. Tube material and thickness.
   b. Fin material size, shape.
   c. Fin bond efficiency.
   d. Header type and pressure.
   e. Type of piping connections.
2. Air Moving Equipment
   a. Power source (electricity, gas, etc.).
   b. Power transmission to fan (direct, gear, belt, etc.).
   c. Number of fans.
   d. Fan material and design.
3. Structure
   a. Slab or pier foundation.
   b. Forced or induced draft.
   c. Structural stability.
   d. Ladders, walkways, handrails.
   e. Type of construction.
   f. Belts, reducing gears, shaft and fan guards.
4. Controls
   a. Temperature control instruments.
   b. Power.
   c. Louvers, rolling doors.
   d. Mixing valves.

Factors to consider in evaluating the selection between induced and forced draft include the following:[24]

1. Induced Draft
   a. Recirculation of air is less (exit air velocity 2–3 times forced draft).
   b. Air distribution over exchanger is better.
   c. Sections are closer to ground and easier to maintain, provided driver mounted below cooler.
   d. Maximum weather protection for finned tubes (rain, hail, freezing).
   e. Few walkways needed, mounting easier overhead.
   f. Connecting piping usually less.
2. Forced Draft
   a. Mechanical equipment more easily accessible.
   b. Isolated supports for mechanical equipment.
   c. Simpler structure.
   d. Easier to adapt to other than motor drives.
   e. Fan horsepower less for same performance (due to difference in air density).
   f. Exchangers are easier to remove for repairs.

Bid Evaluation

Manufacturer's specification sheets, Figure 10-185, are important for proper bid evaluation, and purchaser's specifications may be offered on a form as in Figure 10-186.

Optimum design is not often achieved in all respects; however, the fundamentals and application cost factors of Nakayama[87] are of real value in selecting goals and design features.

In addition to the items listed on the specification sheets and in other paragraphs of this section, it is important for

Revision	No.	Date

AIR COOLED EXCHANGER SPECIFICATION SHEET

		Item No.
Customer		Date
Address	Plant Location	Proposal No.
		Cust. Ref. No.
Service O.H. CONDENSER		Sheet 1 of 1
Size 12-244 Type 2NPVS	Induced Draft	No. of Units 2
Surface/Item ——————— External 48,376	Bare Tube 2,872	Sq. Ft.
Heat Exchange —————— BTU/Hr 5,205,000		Effective MTD 35.3 °F
Transfer Rate ——— External Surface 3.03	Bare Tube Surface — Service 51.3	— Clean 54.5 BTU/Hr. Sq. Ft °F

PERFORMANCE DATA

TUBE SIDE

Fluid Circulated	Hydrocarbons + Air	Temperature In °F 147
Total Fluid Entering Lbs/Hr	32,185	Temperature Out °F 130
Vapor	32,150	Inlet Pressure PSIG 200 mm Hg
Liquid		Gravity—Liquid .833 @ 130°F
Steam		Viscosity .44 cp @ 130 °F
Non-Condensables	35 (Air)	Viscosity @ °F
Vapor Condensed	29,353	Molecular Weight
Steam Condensed		Specific Heat BTU/Lb °F
Density Vapor Lbs/Cu. Ft.	0.504	Latent Heat BTU/Lb 171.7
Conductivity BTU Ft/Hr. Sq. Ft. °F		Allowable Press. Drop PSI 15 mm Hg
Fouling Resistance T. S. Hr. Sq. Ft. °F/BTU	0.001	Design Pressure Drop PSI 15 mm Hg

AIR SIDE

Air Quantity/Item SCFM	345,600 | Face Velocity 600 SFM	Temperature In °F 95
Air Quantity/Fan ACFM	92,875	Temperature Out °F 109.2
Actual Static Pressure In. Water	0.427	Altitude Ft. Sea Level

CONSTRUCTION

Design Pressure 30 & Full Vac. PSI	Test Pressure 45 PSI	Design Temperature 250 °F
SECTION	HEADER	TUBE**
Size 6 Ft. 0 In. X 24 Ft. X 4 Rows	Type Fabricated Box	Material Steel SEAMLESS WELDED
No./Unit 2	Material Carbon Steel	OD 1 In. 14 BWG. Avg. Min. Wall
Arrangement:	No. Passes 1 | Slope 1/4 In./Ft.	No./Section 114
Sections in Parallel 2 in Series –	Plug — Design Shoulder Material CS	Length 24 Ft.
Units in Parallel 2 in Series –	Gasket Material Soft Iron	Pitch 2-3/8 In. △
Section Side Frames Galv. Steel	Corrosion Allowance 1/8 In.	FIN**
MISC.	Size Inlet Nozzle 12 In.	Material Aluminum
Structure Galv Stl | Ladder –	Size Outlet Nozzle 10 In.	OD 2-1/4 In.
Hood Galv Stl | Walkway –	Rating 150 RF	No./In. 8
Shutters No MANUAL AUTO | Vibration Switch No	Code — ASME — Stamp	Type Extruded

MECHANICAL EQUIPMENT

FAN**	DRIVER	SPEED REDUCER
Mfr.	Type Electric Motor	Type V-belt
No./Unit 2 | HP/Fan 11.8	No./Unit 2 | HP/Driver 15	No./Unit 2
Diameter 10 Ft. | RPM 382	RPM 1750	Model –
No. Blades 4 | Pitch ADJUSTABLE	Enclosure Cl.I Gr.D Exp. Proof	AGMA HP Rating –
Blade Material Plastic	Specials –	Ratio 4.58/1 /1
Hub Material Cast Iron	Mfr. Optional	Mfr. –

NOTES: The Following Items are Located in One Common Structure:

Plot Area	Proposal Drawing No.	Shipping Weight Lbs.

Figure 10-185. Specification sheet for air-cooled exchangers. (Used by permission: Air-Cooled Exchangers Manufacturers Association, New York (no longer in existence, 1999); Hudson Engineering Corporation, now Hudson Products Corporation.)

AIR COOLED EQUIPMENT SPECIFICATION FORM

SERVICE ____ EQUIP. SERIAL NO. ____ EQUIP. IDENTIFICATION NO. ____
DRAFT TYPE ____ *MFR. TYPE & DESIGNATION ____ *MFR. JOB NO. ____
DUTY ____ Btu/hr. *TOTAL SURFACE (bare tube) ____ sq. ft. *TOTAL FINNED SURFACE ____ sq. ft.
*LMTD (eff.) ____ *TRANSFER RATE: Service ____ Clean ____ Overall (referred to fin. surface) ____

TUBE SIDE

FLUID ____
FLUID: Fouling ____ Corrosive ____
CORROSIVE COMPOUNDS ____
FOULING FACTOR ____
TOTAL FLUID ENTERING ____ lb/hr.
VAPOR (except steam) ____ lb/hr.
MOLECULAR WEIGHT ____
LIQUID ____ lb/hr.
GRAVITY ____ deg. API @ 60°F.
VISCOSITY (in & out) ____ CP
STEAM ____ lb/hr.
NON-CONDENSABLE GAS ____ lb/hr.
MOLECULAR WEIGHT ____
VAPOR CONDENSED ____ lb/hr.
MOLECULAR WEIGHT ____
GRAVITY ____ deg. API @ 60°F.
STEAM CONDENSED ____ lb/hr.
TEMPERATURE (in & out) ____ °F.
DEG. CONTROL: Product Outlet Temp. ____
PRESSURE DROP (clean) ____ psi
*CALC. PRESSURE DROP ____ psi
NORMAL OPERATING PRESSURE ____ psig
COND. CURVE DATA ____

DESIGN – MATERIALS – CONSTRUCTION

DESIGN PRESSURE & TEMP. ____ psig @ ____ °F.
*NO. PASSES ____ * TUBE PITCH ____
*NO. ROWS (each bundle) ____
*CONNECTED (series/parallel) ____
CORR. ALLOW., HEADER & PARTITIONS ____
MATL: Header ____ Hdr. Cover ____ Plugs ____
MATL: Partitions ____ Tubes ____ Fins ____
NOZZLES: Rating ____ Facing ____ Line Size (in & out) ____
HEADER TYPE ____
*TUBE TYPE ____
TUBE: OD ____ X Min. Wall ____ X Length ____
*FIN: Height ____ *Spacing ____
*SIZE: Bundle ____ *No. Bundles/Section ____
*NO. SECTIONS ____ *TOTAL AREA ____ PLOT AREA ____
*MAX. BUNDLE WT. ____ *TOT. SHIPPING WT. ____
PACKAGING ____ WALKWAYS: No. Sides ____ No. Ladders ____

AIR SIDE

ALTITUDE ____ LOWEST WINTER TEMP. ____ °F.
DESIGN AIR TEMP. IN ____ °F. * AIR TEMP. OUT ____ °F.
*TOTAL FLOW ____ SCFM *STAT PRESS ____ in. H_2O
*CORRECTED QUANT. AT FAN ____ CFM
*FACE VELOCITY ____ ft./min. Std Air

MECHANICAL EQUIPMENT

COUPLING: *Mfr. ____ *Type ____
FANS: *Mfr. & Model ____
*Quan. & Diam. ____
*No. Blades/Fan ____ *RPM ____
Blade Material ____
*bhp/fan: design ____ *Winter ____
*Blade Angle Setting (design cond.) ____
HUBS: Type ____
No. (for auto. variable control) ____
*Material ____ *Mfr. ____
Operational Test Req'd ____
LOUVERS: *Type ____ *Mfr. ____
*Material ____
MOTORS: Type enclosure ____
Volts ____ Cycles ____ Phase ____
*Mfr. ____
*Quan. ____ *RPM ____
*hp/each ____ *Total hp ____
TURBINES: *Mfr. ____ *Model ____
*Quan. ____ *RPM ____
*hp/each ____ *Total hp ____
*Steam (in & out) ____ °F. & psi
*Water Rate ____ lb/hp-hr.
GEARS: *Mfr. & Model ____
*Quan. ____ *Ratio ____
*AGMA Rating ____ hp @ RPM ____

DRAWINGS AND INSTRUCTIONS

DESCRIPTION	NO. REQ'D.	DATE REQ'D.
PRELIM. OUTLINE & EL DWG.		W/proposal
APPROVAL DWGS.		
SPARE PARTS LIST		W/approval dwgs.
FINAL DWGS. & FORMS 1 & 2		
OPERATING INST.		

* If not Specified, Data to be Furnished by Exchanger Manufacturer.

Figure 10-186. Air-cooled equipment specifications form. (Used by permission: Segel, K. D. *Chemical Engineering Progress,* V. 55, ©1959. American Insistute of Chemical Engineers. All rights reserved.)

the process engineer to evaluate the manufacturer's bids for air cooled units with the following points in mind:[88]

1. The dollars/ft^2 of finned surface or dollars/ft^2 of bare tube surface in a finned unit do not necessarily give the only important factor.
2. Determine whether parallel or counter flow exists inside tubes.
3. For condensing problems, determine whether apparent weighted mean temperature difference is used, and which is applicable.
4. Determine fouling factors.
5. Determine tube metal resistance.
6. Determine net free flow area for air across bundle, and determine air linear velocity. Compare air side coefficients for same linear velocities.
7. Determine required fan horsepower (bhp) per million Btu transferred.
8. Determine total dollars per ft^2 of finned surface including standard (or specified) support structure, ladders, etc.

From such items and others pertinent to a specific situation will emerge the conclusions:

1. The lowest dollar value based on complete structure, including the important finned surface.
2. The best dollar value considering amount of basic surface, type of fans, etc.

These two may not be the same. In some instances, high-finned surface area but low bare tube surface means that a lot of tall (sometimes less efficient) fins are crowded onto the tube. In this case, horsepower might be expected to be higher.

Bid evaluations must include a study of the peculiar costs expected to be associated with a given unit, and these include first cost of equipment, power (or driver) operating costs, maintenance for entire unit, foundations, special structural limitations, pipe layout, and perhaps others.

To simplify the evaluation, it is to the advantage of the purchaser to advise the manufacturer of the dollar cost per installed horsepower in his plant and the operating costs for power. The manufacturer can select, from a wide combination of units, the size and number that are the most economical. Otherwise, the bids should be requested as based on "lowest operating cost" or "lowest capital cost," neither being the best in itself except for certain purposes.

Design Considerations (Continuous Service)

The air-cooled heat transfer exchanger is like other exchangers in that the basic heat transfer equation must be statisfied:[265]

$$A = \frac{Q}{(U)(MTD)} \tag{10-285}$$

or,[251]

$$Q = (U)(A)(T - t)_{mean}$$

$$\frac{1}{U} = \frac{1}{htc_a} + \frac{1}{htc_t} + r_{f,t} + r_{f,a} + r_w \tag{10-286}$$

where A = total bare tubeheat transfer area, ft^2
htc_a = airside heat transfer coefficient, Btu/(ft^2) (hr) (°F)
htc_t = tube-side heat transfer coefficient, Btu/(ft^2) (hr) (°F)
MTD = mean temperature difference, °F
Q = heat transfer duty, Btu/hr
$r_{f,t}$ = tube-side fouling resistance, (hr.-ft^2-°F)/Btu
$r_{f,a}$ = air-side fouling resistance, (hr-ft^2-°F)/Btu
r_w = wall resistance, (hr.-ft^2-°F)/Btu
t = air temperature, °F
T = hot fluid temperature, °F
U = overall heat transfer coefficient, Btu/(hr.-ft^2-°F)
CMTD = corrected mean temperature difference, °F
LMTD = log mean temperature difference, °F

And,

$$(T - t)_{mean} = CMTD = (LMTD)\ (F) \tag{10-287}$$

$$= \frac{[(T_1 - t_2) - (T_2 - t_1)][F]}{\ln \frac{[(T_1 - t_2)]}{[(T_2 - t_1)}}$$

F = MTD correction factor, dimensionless, corrects log mean temperature difference for any deviation from true counter-current flow.

In air-cooled heat exchangers, the air flows upward umixed across the finned tubes/bundle, and the tube-side process fluid can flow back and forth and downward as established by the pass arrangements. At 4 or more passes, the flow is considered counter-current, and the "F" factor = 1.0.[215] The other fewer-passes correction factors are given in Figures 10-187A, 10-187B, 10-187C.

Referring to Hudson Products Corporation,[251] used by permission:

1. Hot fluid heat capacity rate = $C_h = C_{tube} = (Mc_p)_{tube} = Q/(T_1 - T_2)$ (10-288)
2. Cold fluid heat capacity rate = $C_c = C_{air} = (Mc_p)_{air} = Q/(t_2 - t_1)$ (10-289)
3. Number of heat transfer units = Ntu $= (A)(U)/C_{min}$ (10-290)
4. Heat capacity rate ratio = $R = C_{min}/C_{max}$ (10-291)
5. Heat transfer effectiveness = E

$$E = \frac{C_h(T_1 - T_2)}{C_{min}(T_1 - t_1)} = \frac{C_c(t_2 - t_1)}{C_{min}(T_1 - t_1)} \tag{10-292}$$

$$E = \frac{1 - e^{-NTU(1-R)}}{1 - Re^{-NTU(1-R)}} \tag{10-293}$$

MTD Correction Factors / 1 Pass-Cross Flow

HUDSON PRODUCTS CORPORATION
Houston, Texas, USA

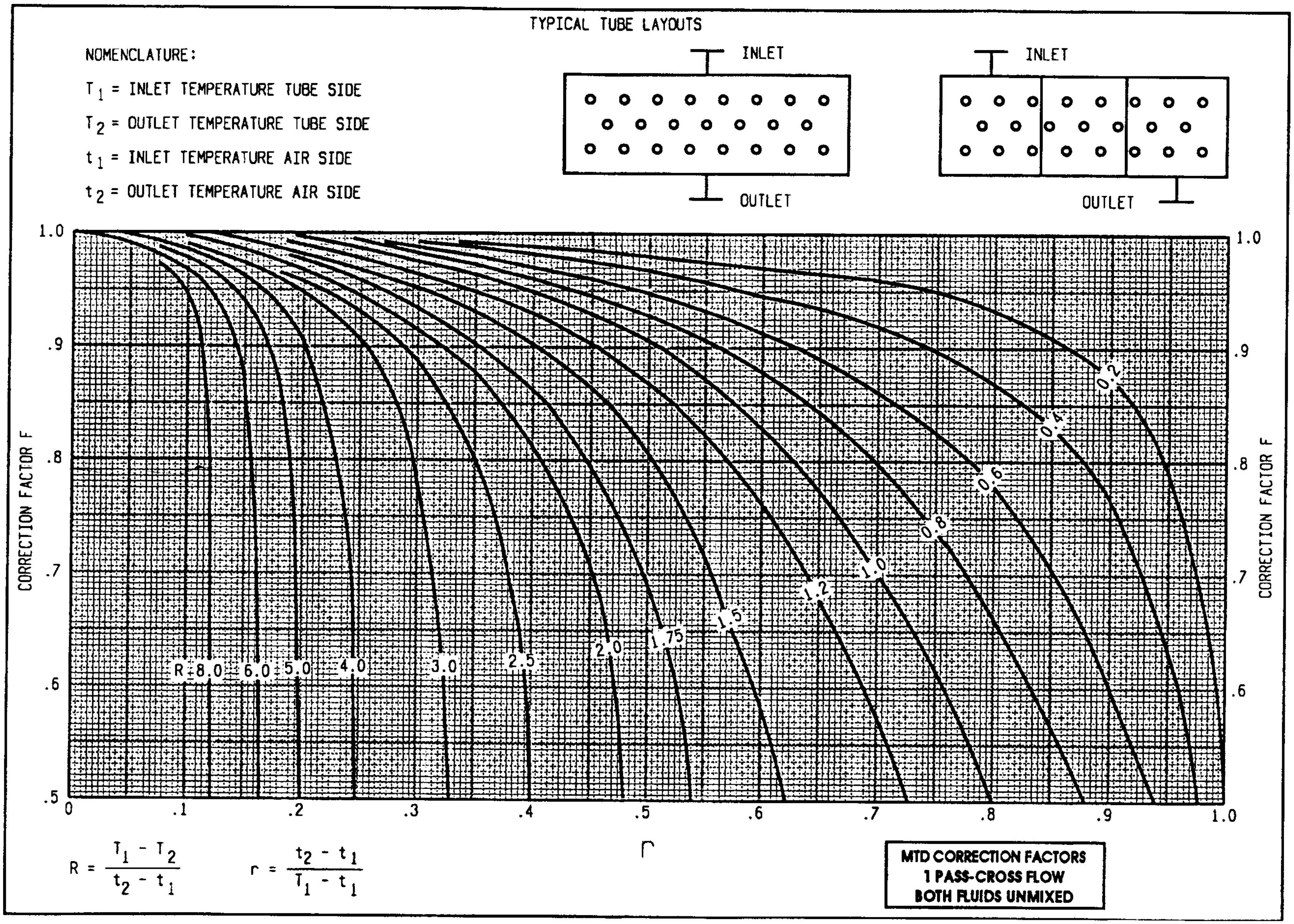


Figure 10-187A. MTD correction factors/1 pass, cross flow. (Used by permission: Bul. M92-300-3M C (10/94). ©Hudson Product Corporation)

MTD Correction Factors / 2 Pass-Cross Flow

HUDSON PRODUCTS CORPORATION
Houston, Texas, USA

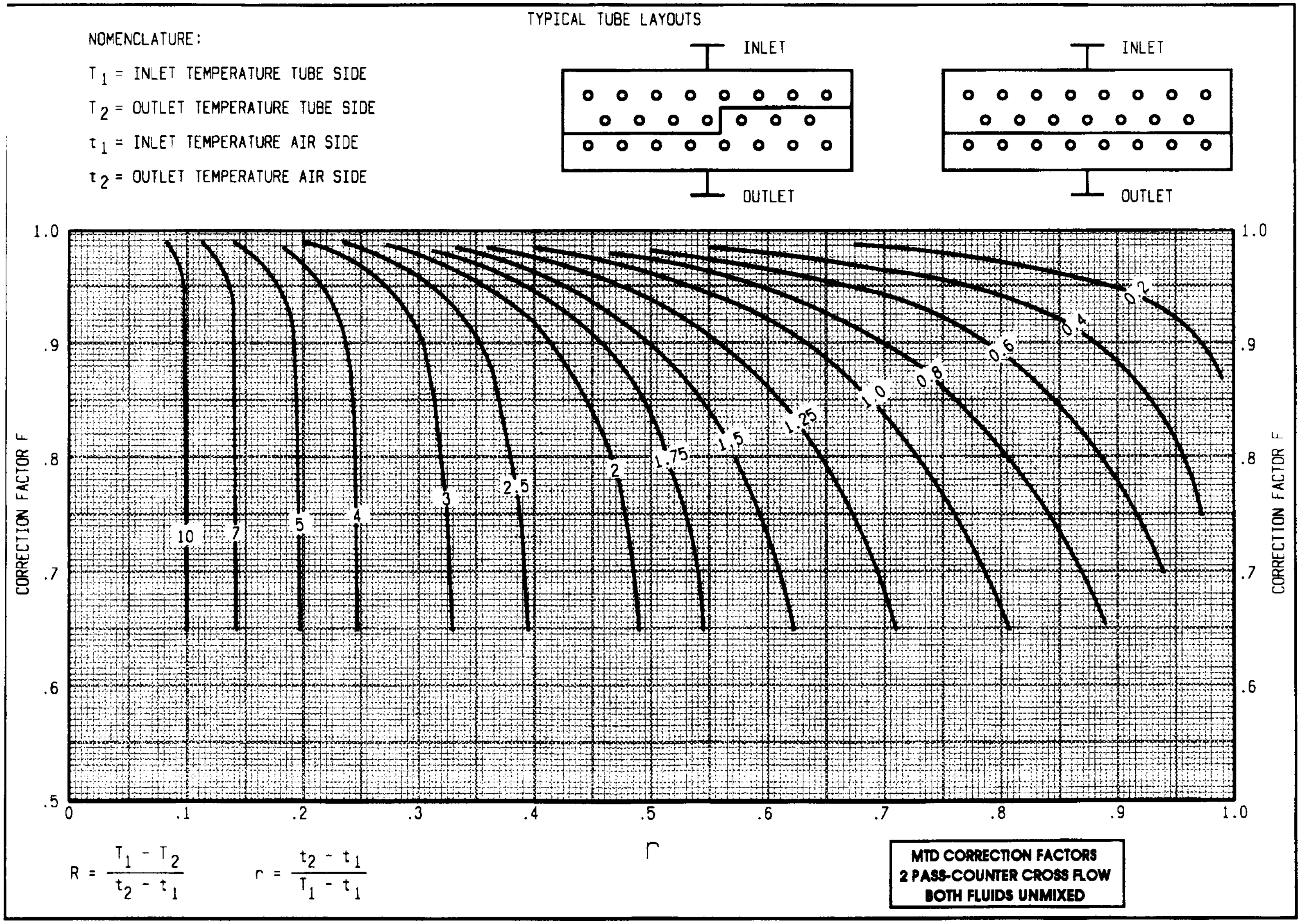


Figure 10-187B. MTD correction factors/2 pass, cross flow. (Used by permission: Bul. M92-300-3M C (10/94). ©Hudson Product Corporation.)

MTD Correction Factors / 3 Pass-Cross Flow

HUDSON PRODUCTS CORPORATION
Houston, Texas, USA

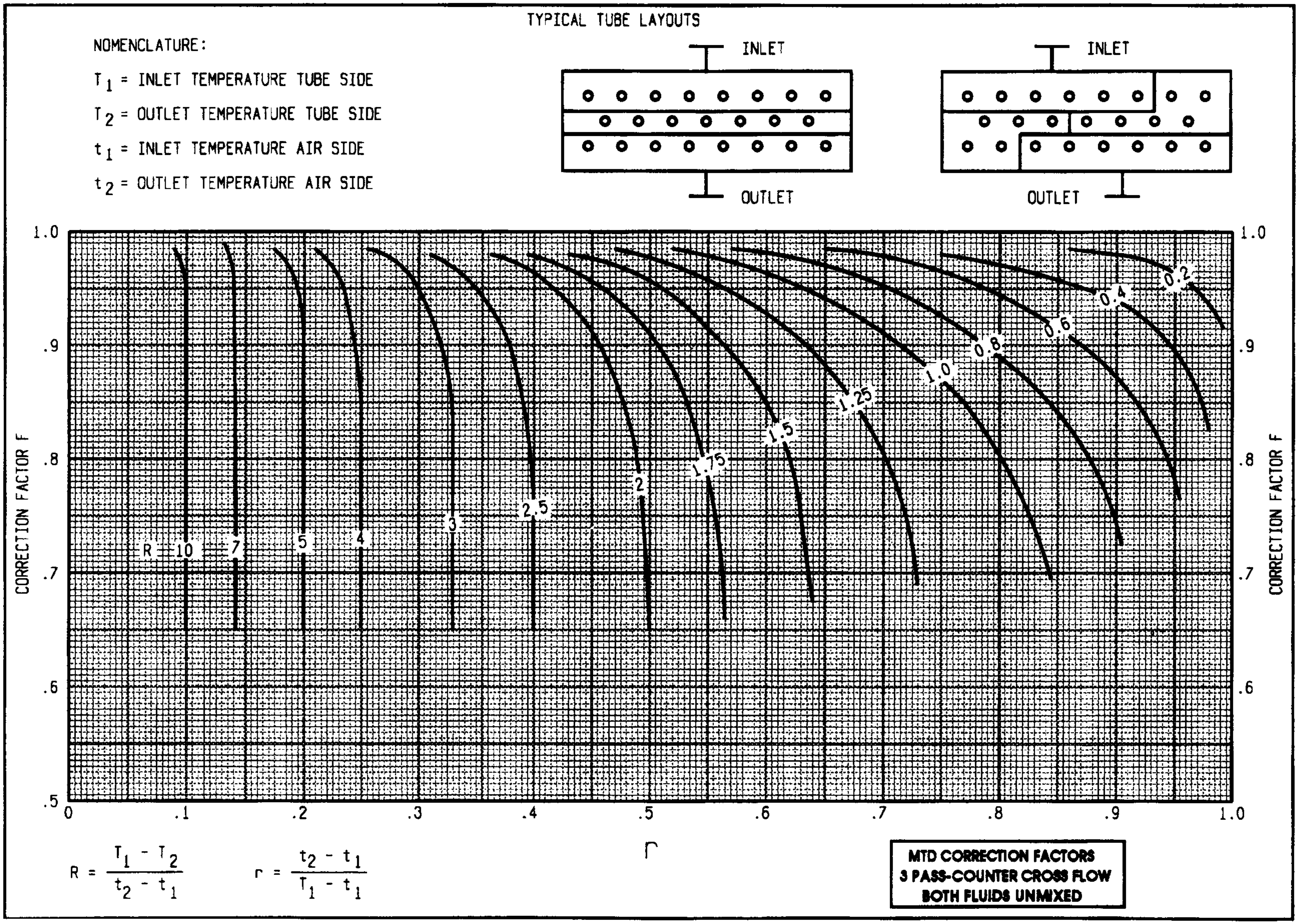


Figure 10-187C. MTD correction factors/3 pass, cross flow. (Used by permission: Bul. M92-300-3M C (10/94). ©Hudson Product Corporation.)

$$R = \frac{C_{min}}{C_{max}} = \frac{C_{air}}{C_{hot}} = \frac{(scfm)(1.08)}{[Q/(T_1 - T_2)]} \tag{10-294}$$

$$= (FV)L\,W(1.08)(T_1 - T_2)/Q$$

$$W = \text{width of exchanger, ft} = \frac{QR}{1.08(FV)L(T_1 - T_2)} \tag{10-295}$$

$$Nta = \frac{AU}{C_{min}} = \frac{AU}{C_{air}} = \frac{nNaWLU}{1.08WLFV} \tag{10-296}$$

$$= \frac{nNa}{1.08(FV)(r_i + r_{air} + r_f + r_m)}$$

$$t_2 = \frac{T_1 - T_2}{R} + t_1 \tag{10-297}$$

Air flow is expressed as standard ft^3 per min (scfm). It is determined by the effective width of the exchanger, W, times the length, L, times the face velocity, FV, in standard ft per min (sfm). From reference 251:

FV(ft/min)	Rows of Tubes
650	4
600	5
550	6
400–450	8–10

Because it is not practical for the design engineer to expect to specify all fabrication features (including size, number of tubes, etc.) the foregoing provides an exposure to the topic, but relies on contact with a competent design/manufacturing firm for the final details.

where (from reference 251 by permission)
- a = heat transfer surface area per unit length of tube, ft^2/ft
- A = total exchanger bare tube heat transfer surface, ft^2
- c_p = specific heat, Btu/(lb)(°F)
- t = air temperature, °F
- T = hot fluid temperature, °F
- U = overall heat transfer coefficient (rate), Btu/(hr)(ft^2)(°F)

Subscripts
- air = air side
- cold = cold fluid = air
- f = tube-side fouling
- hot = hot fluid = tube-side fluid
- i = inside tube
- max = maximum
- min = minimum
- m = tube metal
- 1 = inlet
- 2 = outlet

The lower outside film coefficient (air side) makes use of finned tubes beneficially, while the inside (usually liquid) film coefficient is greater, thus "approaching" a balance for the two sides. The film coefficients on the tube side are calculated in the same manner as described in an earlier topic for conventional exchangers. Mukherjee[265] suggests pressure drop ranges in tubes:

a. For gases and condensers, allowable pressure drop is 0.7–2.84 psi.
   Lower pressure systems require lower pressure drops.
b. For liquids, allowable pressure drop is 7.11–9.95 psi, except when viscosity is high requiring higher pressure drops. The air-side calculations require the specific data of the manufacturer and can be estimated or approximated only by some published data.

$$C_{air} = C_{cold} = Q/\Delta t = Q/(t_2 - t_1)$$
$$= \text{air side heat capacity rate, Btu/(hr) (°F)} \tag{10-298}$$
$$= 1.08(FV)(L)(W)$$
$$C_{tube} = C_{hot} = Q/\Delta T = Q/(T_1 - T_2)$$
$$= \text{tube-side heat capacity rate} = \text{Btu/(hr)(°F)} \tag{10-299}$$
$$= Mc_p$$

- C_{min} = minimum heat capacity rate, Btu/(hr) (°F)
- C_{max} = maximum heat capacity rate, Btu/(hr) (°F)
- CMTD = corrected mean temperature difference = °F(LMTD), °F
- E = exchanger thermal effectiveness, dimensionless

$$= \frac{C_{hot}(T_1 - T_2)}{C_{min}(T_1 - t_1)};$$

Note, see previous information; $C_{hot} = C_{tube}$ (10-300)

$$= \frac{C_{cold}(t_2 - t_1)}{C_{min}(T_1 - t_1)} \tag{10-301}$$

- F = MTD correction factor, dimensionless
- FA = face area, ft^2
- FV = standard air face velocity, sfm
- G = mass velocity, lb/(sec) (ft^2)
- h = individual heat transfer coefficient, Btu/(hr)(ft^2)(°F)
- k = parameter = nNa/[(1.08)(FV)(I/U)]
- LMTD = log mean temperature difference, °F
- M = mass flow rate, lb/hr
- Ntu = number of heat transfer units, dimensionless
- N = number tubes/row in direction of air flow
- n = number tubes/row, per ft of exchanger width, 1/ft
- Q = total exchanger heat load (duty), Btu/hr
- R = C_{min}/C_{max} = heat capacity ratio, dimensionless

Mean Temperature Difference

These units are pure cross-flow and require the use of specific data not found in the TEMA Standards,[266] but are available in references 251 and 206. See Figures 10-187A, 10-187B, and 10-187C.

Table 10-50
Typical Temperature Study for Design Air Temperature Determination

Location	Maximum Dry Bulb Temp., °F	Dry Bulb Temp., °F; % of Annual Hr Stated Temp. Is Exceeded			Annual Average Dry Bulb Temp., °F	Suggested Design Temp., °F
		1%	2%	3%		
Beaumont, Texas	102	93	91	90	69	91
Victoria, Texas	110	89	96	95	71	96
Parkersburg, W. Va.	106	90	87	86	55	87
New Orleans, La.	102	92	91	89	70	91
Wilmington, Del.	106	88	85	84	55	85
Grand Rapids, Mich.	99	83	80	78	47	80

Note: 1% = 88 hr; 2% = 175 hr; 3% = 263 hr

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1. Design maximum ambient air temperature should be selected so that it will not be exceeded more than 2–5% of the time. Lower figures mean a smaller exchanger, but they also indicate a question on performance during the hottest weather. Daily temperature charts as well as curves showing the number of hours and time of year any given temperature is exceeded are valuable and often necessary in establishing an economical design air temperature. Collins and Mathews discuss this in detail.[32] Also see Table 10-50.

2. Units preferably should be placed in the open and at least 75–100 ft from any large building or obstruction to normal wind flow. If closer, the recirculation from downdrafts may require raising the effective inlet air temperature 2–3°F or more above the ambient selected for unobstructed locations. If wind velocities are high around congested areas, the allowance for recirculation should be raised to greater than 3°F.

3. Units should not be located near heat sources. Cook[33] cautions that units near exhaust gases from engines can raise inlet air 15°F or more above the expected ambient.

4. The effect of cold weather on the freezing of tube-side fluids and increasing horsepower due to increased air density can not be overlooked. Usual practice is to reduce fan output by using a two-speed motor, louvers on variable-pitch fans, or drivers.[33]

5. Fouling on the outside of finned surfaces is usually rather small, but must be recognized. Values of 0.0001–0.0015 usually satisfy most fin-side conditions. Finned surfaces should be cleaned periodically to avoid excessive build-up of dust, oil films, bugs, etc.

6. When inquiring or designing, the range of expected temperature operations should be stated, as well as maximums only. If any particular temperature is "key" or critical to the system, it should be so identified.

7. When processing (tube side) coefficients referred to the bare outside tube are less than 200 Btu/hr (ft^2) (°F), the total surface of the air-cooled unit usually compares favorably cost-wise with a water-cooled unit.[80]

8. For a specific service of desuperheating Freon 11 (180°F) and condensing at 115°F, ambient air at 70°F, total $Q = 31.6 \times 10^6$ Btu/hr, Smith[106] points out that for three comparative designs with a threefold reduction in fan horsepower, a 35% increase occurs in first cost, a 30% increase in surface, and a 75% increase in plan area. In general this trend will apply to all comparisons on design parameters; of course it is influenced to a greater or lesser degree by specific conditions, which reflect the sensitivity of changes in flow quantities on heat transfer coefficients.

9. Nakayama[87] suggests essentially the same procedure except recommending that specific manufacturers' data for various units be assembled and correlated for use in the detailed design of film coefficients and pressure drop.

10. In general, tube-side pressure drops less than one psi per pass should not be specified for economical designs.[80] Drops as low as 15 mm Hg. have been specified, and designs obtained which were competitive with cooling tower installations. For viscous materials, pressure drop limitations can markedly influence a design and its economics.

Required fan driver horsepower based on material from Hudson Products Corp.,[251] motor shaft horsepower output to fan is:

$$= \frac{[\text{Actual ft}^3/\text{min (at fan inlet)}]\,[\text{Total pressure loss (in. water) through air-cooled outside fins}]}{6{,}356\ [\text{fan (system) efficiency}]\,[\text{speed reducer efficiency}]} \quad (10\text{-}302)$$

Volume of fan

$$= \frac{(\text{standard volume of air, scfm})\ [\text{Air Std. density, } 0.075\ \text{lb/ft}^3]}{[\text{density of air at fan inlet, lb/ft}^3]}$$

Total pressure difference across fan

= velocity pressure for fan diameter
+ static pressure loss through air-cooled bundle (from manufacturer's data for a specific exchanger)
+ other losses in the air system.

Fans usually result in velocity pressure of approximately 0.1-in. water.

System efficiency is influenced by the air plenum chamber and fan housing. Industrial axial flow fans in proper system design will have efficiencies of approximately 75% based on total pressure. Poor designs can run 40%.[251] Speed reducers are about 75% mechanically efficient.

Then, motor (driver) input power

$$= \frac{\text{motor shaft hp to fan}}{\text{motor efficiency, fraction*}} \quad (10\text{-}303)$$

*See the chapter on Drivers, this volume.

Table 10-51
Typical Transfer Coefficients for Air-Cooled Exchangers Based on Bare Tube Surface

Condensing Service	U
Amine reactivator	100–120
Ammonia	105–125
Refrigerant 12	75–90
Heavy naphtha	70–90
Light gasoline	95
Light hydrocarbons	95–105
Light naphtha	80–100
Reactor effluent Platformers, Hydroformers, Rexformers	80–100
Steam (0–20 psig)	135–200
Gas cooling service	
Air or flue gas @ 50 psig (ΔP = 1 psi)	10
Air or flue gas @ 100 psig (ΔP = 2 psi)	20
Air or flue gas @ 100 psig (ΔP = 5 psi)	30
Ammonia reactor stream	90–110
Hydrocarbon gasses @ 15–50 psig (ΔP = 1 psi)	30–40
Hydrocarbon gasses @ 50–250 psig (ΔP = 3 psi)	50–60
Hydrocarbon gasses @ 250–1500 psig (ΔP = 5 psi)	70–90
Liquid cooling service	
Engine jacket water	130–155
Fuel oil	20–30
Hydroformer and Platformer liquids	85
Light gas oil	70–90
Light hydrocarbons	90–120
Light naphtha	90
Process water	120–145
Residuum	10–20
Tar	5–10

Coefficients are based on outside bare tube surface for 1-in. O.D. tubes with 10 plain extruded aluminum fins per in., $^5/_8$ in. high, 21.2:1 surface ratio.

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Design Procedure for Approximation

Specific designs are best obtained from manufacturers offering this type of equipment or from specific curves applicable to the units under study.

A suggested inquiry specification sheet is shown in Figure 10-186. It serves to define the known factors at the time of inquiry and then to summarize the exact specifications as proposed by a specific vendor.

The method summarized is essentially that of Smith.[106]

1. Determine heat duty for the exchanger from process fluid temperatures.
2. Select design ambient air temperature, t_1.
3. Select design pressure on tube side, tube material, tube size, and gage.

Table 10-52
Overall Transfer Rates for Air-Cooled Heat Exchangers

Service	*Stab Transfer Rate	**Suggested No. of Tube Layers
Cooling Service		
Engine jacket water	6–7	4
Light hydrocarbons	4–5	4 or 6
Light gas oil	3–4	4 or 6
Heavy gas oil	2.5–3	4 or 6
Lube oil	1–2	4 or 6
Bottoms	0.75–1.5	6 or more
Flue gas @ 100 psig & 5 psi ΔP	2–2.5	4
Condensing Service		
Steam	7–8	4
Light hydrocarbon	4–5	4 or 6
Reactor effluent	3–4	6
Still overhead	2.75–3.5	4 or 6

*Transfer rate, Btu/(hr) (ft^2)(°F), based on **outside fin tube surface** for 1-in. O.D. tubes with $^5/_8$ in. high aluminum fins spaced 11 per in.

**The suggested number of tube layers cannot be accurately predicted for all services. In general coolers having a cooling range up to 80°F and condensers having a condensing range up to 50°F are selected with 4 tube layers. Cooling and condensing services with ranges exceeding these values are generally figured with 6 tube layers.

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4. From Table 10-51 select overall U for exchanger service. Note that Table 10-52 gives transfer rates based on outside finned surface.
5. Calculate,

$$\frac{T_1 - t_1}{U(\text{bare tube})} \tag{10-304}$$

and from Figure 10-188, read optimum bundle tube row depth.

6. From Table 10-53, select (a) typical standard air face velocity, (b) ratio of surface area to face area, and (c) ratio of weight to face area.
7. Determine surface requirements by trial and error:
   a. Assume air temperature rise, $t_2 - t_1$.
   b. Solve for total face area required:

$$FA = \frac{Q}{(t_2 - t_1)(FV)(1.08)} \tag{10-305}$$

   c. Calculate LMTD using t_1, t_2, T_1, T_2
   Neglect correction to LMTD unless outlet air temperature, t_2, is considerably greater than the required outlet tube-side temperature, T_2.
   d. Calculate *bare* or plain tube surface required:

$$A = \frac{Q}{U(\text{LMTD})}$$

Table 10-53
Estimating Factors, 1-in. O.D. Tube × 2 $^3/_8$-in. Δ Spacing

Depth, tube rows	4	6	8	10	12
Typical standard FV, ft/min**	595	540	490	445	405
Ft^2 surface/ft^2 face area	5.04	7.60	10.08	12.64	15.20
Weight lb/ft^2 face area	75	88	115	131	147

**FV = face velocity.

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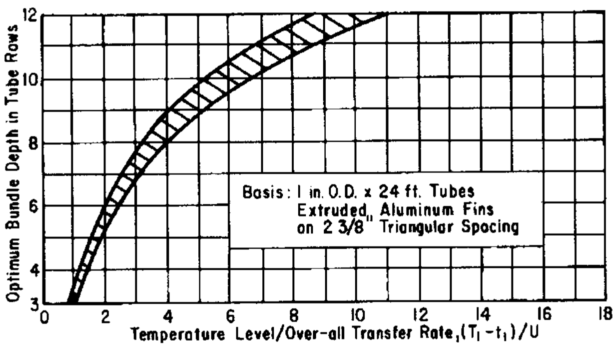


Figure 10-188. Optimum bundle depth. (Used by permission: Smith, E. C. *Chemical Engineering,* V. 65, Nov. ©1958. McGraw-Hill, Inc. All rights reserved.)

This can be converted to finned surface by ratio of finned/bare surface areas.

e. Calculate face area, FA_2:

$$FA_2 = \frac{A}{\left(\dfrac{\text{surface area}}{\text{face area}}\right)} = FA_1\text{, (from Table 10-53)} \qquad (10\text{-}306)$$

Where subscript 2 refers to second or check calculation, and 1 refers to original trial.

f. If $FA_2 = FA_1$, proceed with detailed design.
If $FA_2 \neq FA_1$, reassume new air outlet temperature and repeat from (a).

g. For detailed check:
Assume tube-side passes and calculate h_i, h_{io} in the usual manner.
From air velocity, ft/min, calculate quantity and film coefficient considering fin efficiency.
Figure 10-189 may be used directly to obtain effective outside fin coefficient, h_o, based on the *bare* tube surface. Recalculate overall U. If this value differs greatly, unit should be calculated until balance is reached.

h. Calculate tube-side pressure drop in usual manner, including loss in headers.

i. Determine unit plan size:

$$\text{Width} = \frac{\text{face area}}{\text{assumed tube length (usually 4 ft, 6 in. min. through 30 ft)}} \qquad (10\text{-}307)$$

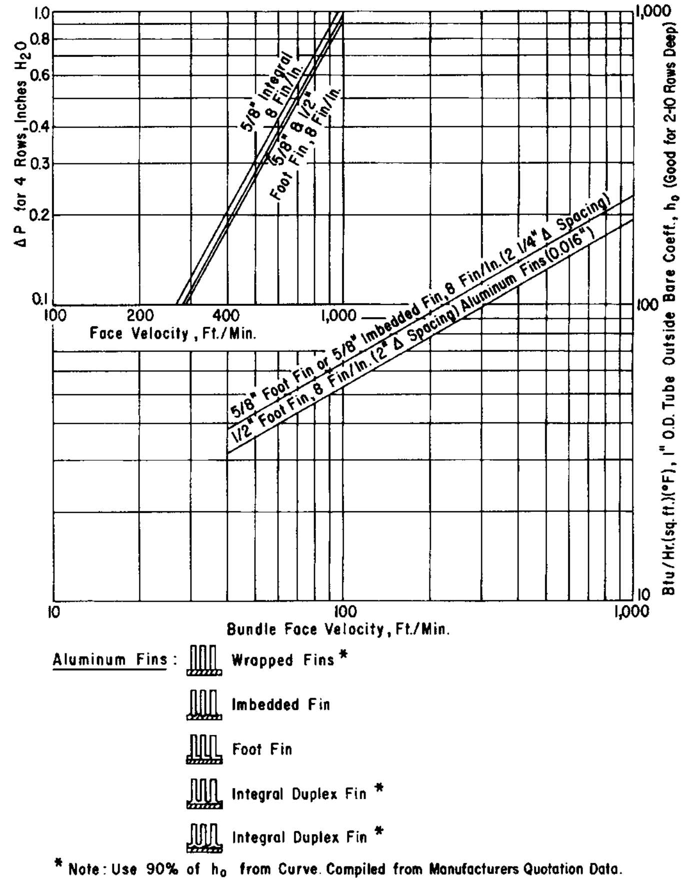


Figure 10-189. Outside fin film, coefficient for air-fin exchangers. (Used by permission: Hajek, J. D. Compiled from manufacturer's data, private communications, now deceased.)

Balance these to obtain practical or standard size units. Bundle widths are usually 4 ft, 6 in. and 7 ft, 6 in.

j. Calculate hp requirements from Figure 10-190; read surface area/hp or read Figure 10-189 for pressure drop for certain tube arrangements.

$$HP = \frac{\text{total surface, A}}{\text{surface area/hp}} \qquad (10\text{-}308)$$

$$\text{Also: } HP = \frac{(ACFM)(p_t)}{(6{,}356)(e_r)(e_d)} \qquad (10\text{-}309)$$

p_v = 0.1 in. water, usually
p_s = 0.2 to 0.25 in. water at ≅ 500 ft/min FV for each 3 rows of tubes
p_t = $p_v + p_s$, in. water
e_r = 0.65 usually
e_d = 0.95 usually

k. Approximate weight:

$$= \text{(face area) (weight/face area)} \qquad (10\text{-}310)$$

p_v = velocity pressure, in. water
p_s = static pressure, in. water
p_t = total pressure, in. water
ACFM = actual CFM at fan intake
FA = face area of air cooled exchanger tube bundle, length × width, ft^2

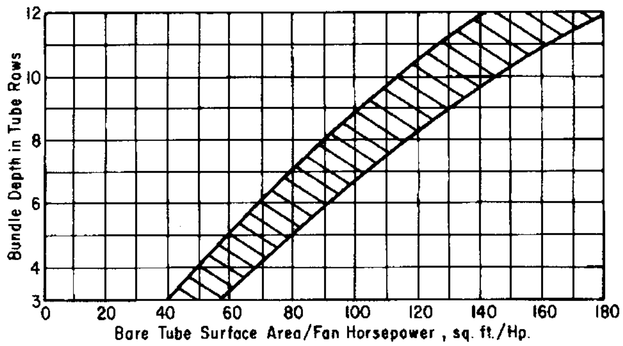


Figure 10-190. Surface per fan hp. (Used by permission: Smith, E. C. *Chemical Engineering,* V. 65, Nov. 1958. ©McGraw-Hill, Inc. All rights reserved.)

t_1, t_2 = inlet and outlet air temperature of fin unit
T_1, T_2 = inlet and outlet tube-side fluid temperature of fin unit
FV = face velocity, ft/min, entering face area of air cooled unit
U = overall heat transfer rate based on bare tube O.D., Btu/hr (ft^2)(°F)

Tube-Side Fluid Temperature Control

The tube-side fluid responds quickly to changes in inlet air temperature. In many applications this is of no great consequence as long as the unit has been designed to take the maximum. For condensing or other critical service, a sudden drop in air temperature can create pressure surges in distillation or other process equipment, and even cause flooding due to changes in vapor loading. Vacuum units must have a pressure control that can bleed air or other inerts into the ejector or vacuum pump to maintain near-constant conditions on the process equipment. For some units the resulting liquid subcooling is not of great concern.

Depending upon the extent of control considered necessary, the following systems are used (Figures 10-191 and 10-192):

1. By-pass control of inlet fluid with downstream mixing to desired final temperature.
2. Manual (for seasonal changes only) or automatic pitch control operated by air-motor on fan blades.
3. Variable speed drive (motor, turbine, hydraulic).
4. Fixed two-speed drive (usually for day and night operation).
5. Louvers on air off exchanger.
6. Shut-down of fans (one or more) when multiple fans are used in the same process service.

When only one fan and/or exchanger exists per process service, it may be advisable to control with an automatic variable pitch fan, unless a single- or two-speed drive is considered adequate. If the process service consists of several exchanger sections or tube bundles per cell (groups of bundles) and multiple fans are used, see Figure 10-193. If single fans are used per cell, see Figure 10-194. If several

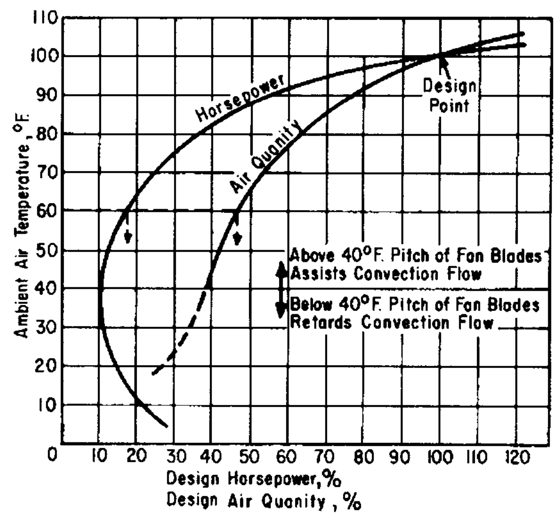


Figure 10-191. Temperature control and horsepower savings with automatic variable pitch fans. (Used by permission: Hudson Products Corporation.)

cells are used per process service, some of the fans should be considered for automatic variable pitch control (if continuous variable speed not used), some for two-speed control with or without louvers, and the remainder set on constant speed. Various combinations can be developed to suit the process needed, taking into account the change in air flow with speed and its effect on film coefficients. The manufacturer can supply this after the type of control is established.

For winter operation, it is important to consider the effect of cold temperatures on process fluid (gelling, freezing, etc.), and this may dictate the controls. In addition, tarpaulins or other moveable barriers may be added to reduce air intake or discharge. This can be a serious control problem.

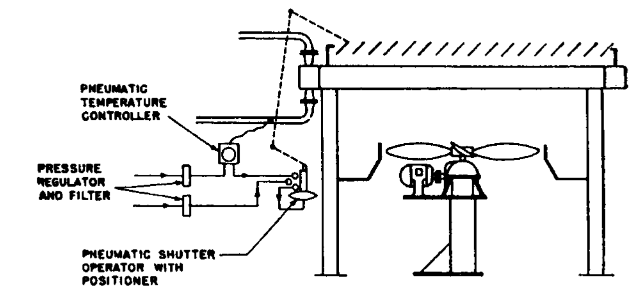
When a two-speed motor is reduced to half-speed, the air capacity will be cut 50%, and the required horsepower consumption will drop to $^1/_8$ of full speed power.

Heat Exchanger Design with Computers

Several descriptions have been presented[43, 48, 92, 111, 125] to bring out the usefulness of electronic computers in various phases of heat exchanger design. Although any medium-sized digital computer can handle the decisions and storage capacity, a large investment must be made in programming time required to achieve a good flexible program. Often several months are required to polish the program; but when completed, it can save many hours of calculation time. It is usually better to create programs specific to the types of exchanger performance, such as convection, condensing, thermosiphon reboiling, condensing in the presence of noncondensable gases, etc., rather than creating an overall program to attempt to cover all types.

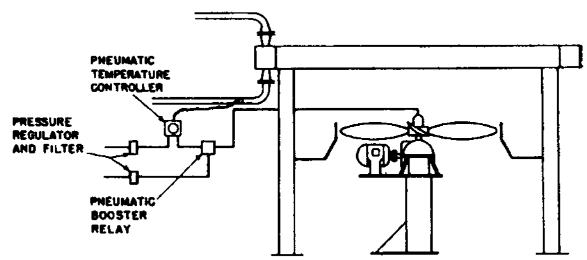
Adjustable Shutters

Shutters mounted above the cooling sections serve to protect them from overhead wind, snow, ice, and hail. In addition, they are also used to regulate, either manually or automatically, the flow of air across the finned tubes; thus, they control the process fluid outlet temperature.



Controllable Pitch Fan

The controllable pitch fan provides an infinitely variable air delivery across the K-fin sections through automatic changes in the fan blade angle. The temperature can be closely controlled to meet the varying demands of operating conditions and fluctuating atmospheric temperatures with appreciable power savings under low load conditions. For certain control applications, the ability of this fan to pump air backwards with negative blade angles is used.



Combination Controls

The combination of adjustable shutters with variable speed drive or with controllable pitch fan is frequently used for close fluid temperature control. This system is particularly useful during start-up and shut-down procedures in which fluids are subject to freezing in cold ambient temperatures. It is also well-adapted to fluid temperature control while operating under high wind and freezing conditions. This diagram shows a two-speed electric motor with automatically adjustable shutters. The arrangement lends itself to many cooling services while effecting a horsepower savings through the use of the two-speed motor.

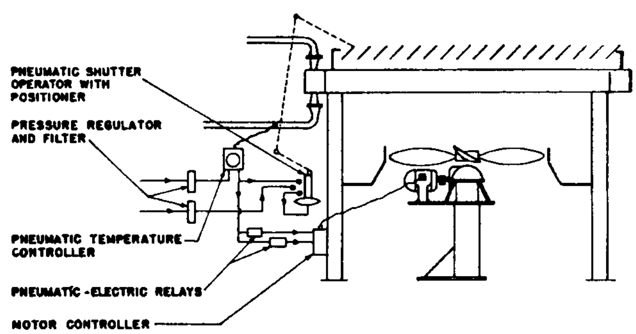


Figure 10-192. Schemes for temperature control of air-coolers. (Used by permission: Griscom-Russell/Ecolaire Corporation.)

No. Cells	A Sections Per Cell	B Nom. Lg. Tubes	C Fan Dia.	D Length	E Width ℄ to ℄ Col's. per Cell	Total Width
	2	240"	8'-0"	18'-3 1/2"	10'-8 1/4"	
	2	288"	8'-0"	22'-3 1/2"	10'-8 1/4"	
	2	288"	10'-0"	22'-3 1/2"	10'-8 1/4"	
	2	360"	10'-0"	28'-3 1/2"	10'-8 1/4"	
	2 1/2	240"	8'-0"	18'-3 1/2"	13'-4 5/16"	
	2 1/2	288"	10'-0"	22'-3 1/2"	13'-4 5/16"	
	2 1/2	360"	10'-0"	28'-3 1/2"	13'-4 5/16"	

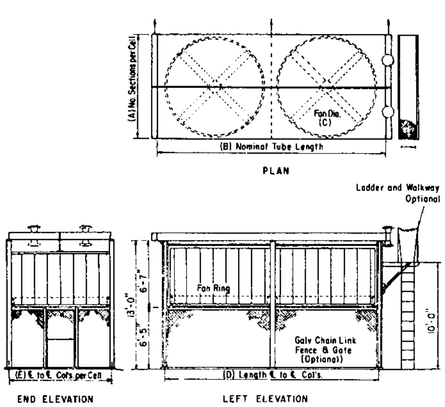


Figure 10-193. Typical dimensions for air-coolers with two fans. (Used by permission: Griscom-Russell/Ecolaire Corporation.)

No. Cells	A Sections Per Cell	B Nom. Lg. Tubes	C Fan Dia.	D Length ℄ to ℄ Col's.	E Width ℄ to ℄ Col's. per Cell	Total Width
	2	120"	8'-0"	8'-3 1/2"	10'-8 1/4"	
	2	180"	10'-0"	13'-3 1/2"	10'-8 1/4"	
	2	240"	10'-0"	18'-3 1/2"	10'-8 1/4"	
	2 1/2	180"	10'-0"	13'-3 1/2"	13'-4 5/16"	
	2 1/2	240"	12'-0"	18'-3 1/2"	13'-4 5/16"	

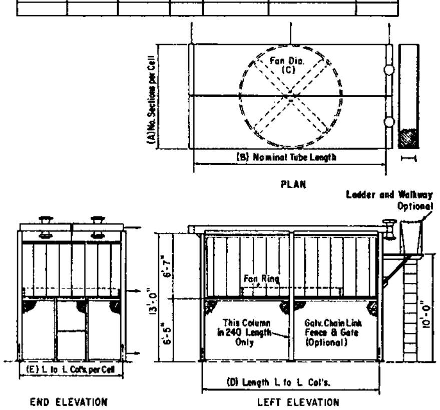


Figure 10-194. Typical dimensions for air-coolers with one fan. (Used by permission: Griscom-Russell/Ecolaire Corporation.)

Nomenclature

A = total exchanger bare tube heat transfer, ft^2; or, net external surface area of tubes exposed to fluid heat transfer, ft^2; or, area available for heat transfer, ft^2 (for conduction heat transfer, A is a cross-sectional area, taken normally in the direction of heat flow, ft^2).
A' = area of heat transfer, from emitting to or absorbing radiation source, ft^2.
A_a = area of heat transfer between the insulation or bare pipe and air, ft^2.
A_b = net free area of flow through baffle window, ft^2.
A_c = net free area of cross-flow between baffles on shell side of tube bundle, ft^2.
A_{cc}, A_{cp}, A_o, A_p = special symbols for insulation equation.
A_g = geometric mean free flow area on shell side of tube bundle, ft^2.
A_i = inside area of unit length of tube, ft^2/ft (or, where indicated).
A_n = net effective outside tube surface area for plain or bare tubes, ft^2.
A_{nf} = net effective outside tube surface area for finned tubes, ft^2.
A_o = outside area of unit length of tube, ft^2; or, required effective outside heat transfer surface area based on net exposed tube area, ft^2.
ACFM = actual ft^3 per min of air at fan intake.
a = cross-section area of tube for flow through inside, ft^2. May be used for an individual tube or a group of tubes, depending on the intended use; or, bulk fluid outside insulated pipe; or, interfacial area ft^2/ft^3; or, heat transfer surface area per unit length of tube, $ft^2/ft/$(air cooler).
a' = constant in Bromley equation, $(in./ft)^{1/4}$.
a'' = exponent.
a_i = cross-sectional flow area per tube, $in.^2$; or, cross-sectional flow area, inlet feed pipe, ft^2.
a_o = tube outside heat transfer surface/ft, ft.
a_p = flow area per pass inside tubes, ft^2.
a_s = flow area across tube bundle, ft^2 (cross-flow area).
a_t = surface area of tube outside, ft^2/ft.
a_x = projected area of cross-section of finned tube at the fin, $in.^2$
B = overall resistance to heat transfer less shell-side resistance (for finned tube calculations), 1/Btu (hr)(ft^2)(°F).
B or B_s = baffle pitch or spacing, in.
B_s' = baffle spacing at ends of bundle, in.
B_{ca} = baffle cut areas, expressed as fraction, representing opening as percent of shell cross-section area.
B_L = coefficient in Levy boiling coefficient equation, Figure 10-99.
BC = tube length for sensible heating, ft, Figure 10-110.
BCF = bundle correction factor, Equation 10-144.
b = fin height, ft.
C = capacity constant for tubes, Table 10-3, (except ratio in caloric temperature).
C' or c' = clearance between tubes, measured along the tube pitch, in.; Figure 10-56.
CD = tube length for vaporization, ft, Figure 10-110.
c or c_p = heat capacity or specific heat at constant pressure, Btu/lb(°F); or, heat capacity of condensate, pcu/lb(°C)
c_b = constant in pressure drop equation.
C_s or c_s = surface condition constant for Equation 10-162.
CMTD = corrected mean temperature difference = (°F)(LMTD), °F.
C_N = turbulence correction factor.
D = I.D. of tube, ft; also used as D_1 (inside), D_2 (outside); or, impeller diameter, ft; or, pipe O.D., ft; or, O.D. of insulated or bare pipe, whichever is being studied.
D' = tube diameter, side where boiling takes place, ft, Equation 10-165.
D_a = diameter of agitator, ft.
D_b = tube bundle diameter, ft.
D_e or D_{es} = equivalent tube outside diameter, ft; or, equivalent inside tube diameter, in.; or, equivalent diameter of annulus (D_1 = O.D. of inner tube, ft; D_2 = inside diameter of outer pipe, ft).
D_e' = equivalent diameter of tube bundle in shell, for pressure drop, ft.
D_{eg} = an equivalent diameter of a finned tube for calculation of condensing coefficient, ft.
D_H = diameter of helix of coil, in.
D_i = I.D. of tube, ft; or, insulation O.D., ft.
D_j = diameter of inside of vessel, ft.
D_H = nozzle size, diameter, in.
D_o = O.D. of tube, ft; or, outside diameter of insulation, in.
D_o' or D_t = tube O.D., in.
D_s = shell I.D., in.
D_s' = shell I.D., ft.
D_t = see D_o'; or, O.D. of tracer, in.
d_e or d_s = shell-side equivalent tube diameter, in.
d_e' = equivalent diameter, finned, in.
d_i or d_{it} = I.D. of tube, in.
d' = film thickness, ft.
d_o = tube O.D., in. (or ft, as specified).
d_r = root diameter of finned tube, in.
d_s = shell-side I.D., in.
d_t = tube O.D., ft.
E = superficial liquid entrainment, $lb/hr/ft^2$; or, heat transfer effectiveness.
E_B = enthalpy of bottoms, Btu/lb.
E_D = enthalpy of overhead product removed from column, Btu/lb.
E_F = enthalpy of feed, Btu/lb.

e_d = efficiency of driver, fraction.
e_f = finned tube efficiency; or, efficiency of fan, fraction.
F = LMTD correction factor as read from charts; or, friction loss, (ft) (lb)/(lb).
F_{B-C} = friction loss from part B to part C in tubes, ft liquid.
F_1, F_2 = correction factors, flooding equation.
F_l = total lin ft of tube, ft.
F_c = calorie fraction, dimensionless.
F_p = pressure drop factor, dimensionless.
F_p' = pressure correction factor, boiling, dimensionless.
F_t = dimensionless tube size factor, Buthod's pressure drop method.
F_w = tube size correction factor for water film, dimensionless.
FA = face area of air cooled exchanger bundle; length and width, ft^2
FV = face velocity of air entering face area of air cooled exchanger, ft/min.
ΔF = friction loss at inlet, ft liquid.
f = friction factor, $ft^2/in.^2$; or, outside film coefficient, Btu/(hr) (ft^2) (°F).
f_f = dimensionless friction factor.
f_s = friction factor, $ft^2/in.^2$, in Figure 10-140.
f_s'' = friction factor for shell-side cross flow.
G = fluid mass velocity, lb/hr(ft^2 tube); or, lb/(sec) (ft^2) (cross-section flow area) for tube side, or lb/hr(ft^2) of shell-side flow area for shell side; or, velocity normal to tube surface; or, superficial gas mass velocity, lb/(hr) (ft^2); or, mass velocity, lb/(hr) (ft^2). (Used by permission: Brown Fintube Company.)
G' = mass flow rate per unit tube inside circumference =w/(πD), lb/(hr) (ft^2); or, condensate loading for vertical tubes, lb/(hr) (ft), Figure 10-67A; or, mass velocity, lb/(sec) (ft^2). (Used by permission: Brown Fintube Company.)
G''' = condensate mass flow rate inside horizontal tubes, lb/(hr) (lin ft).
G_o' = condensate mass flow rate per unit tube outside circumference, vertical tubes, lb/(hr) (ft).
G'' = mass velocity for tube flow, lb/sec(ft^2 cross-section of tube); or, units as, lb/hr(ft).
G_o'' = condensate mass flow rate outside (shell side) for horizontal tubes, lb/(hr) (lin. ft).
G_b = mass flow rate through baffle "window," lb fluid/(hr) (net ft^2 of flow cross- section area through the "window" opening in baffle).
G_b' = mass velocity through baffle opening, lb/sec(ft^2).
G_c = mass flow, lb/hr(ft^2 of cross-section at minimum free area in cross-flow).
G_c' = maximum bundle cross-flow mass velocity, lb/sec(ft^2).
G_d = liquid loading for tubular drip type coolers, lb./hr(lin ft).
G_e = equivalent liquid mass velocity for Akers, *et al.* equation, lb./hr(ft^2 cross-section flow area).
G_e' = geometric mean mass velocity, lb/sec(ft^2).
G_g = geometric mean mass velocity through shell side, lb/hr(ft^2).
G_{gb} = boiling equation mass velocity of liquid, lb/hr(ft^2). For outside tubes, use projected area (diameter × tube length).
G_s = mass velocity, lb/hr(ft^2); or, mass rate of flow on shell side of exchanger, lb/(hr) (ft^2 of flow area); *also*, cross-flow on shell side.
G_t = mass flow rate in tubes, lb/hr(ft^2 of cross-section flow area); or, lb/(sec) (ft^2).
G_w = weighed or geometric mean mass velocity, lb/hr(ft^2).
$\bar{G}$ = mass velocity of vapor from a bottom tube on (p-D_o) spacing, lb/(hr) (ft^2), Equation 10-145.
$\bar{G}_g$ = arithmetic average vapor flow, inlet to outlet, for vapor flowing inside tubes, lb vapor/hr(ft^2 flow cross-section).
$\bar{G}_L$ = arithmetic average liquid flow, inlet to outlet, inside tube, lb condensate/hr(ft^2 of flow cross-section).
G_{max} = mass flow, lb/sec(ft^2 of cross-section at minimum free area in cross-flow).
Gr = Grashof number.
Gz = Graetz number.
g or G = acceleration due to gravity, 4.17×10^8, $ft/(hr)^2$; or, gravitational constant, 32.2 ft/(sec) (sec).
g' = acceleration of gravity, 32.2 $ft/(sec)^2$.
H = heat transfer coefficient ratio, h_M/h_{Nu}.
H_c = height of segment of circle divided by diameter.
$H_{g,d}$ = height of a gas phase mass transfer unit, ft.
$H_{l,d}$ = height of a liquid phase mass transfer unit, ft.
Hp = horsepower, usually as brake horsepower.
h = heat transfer coefficient, Btu/(hr) (ft^2) (°F).
h_1 = nucleate boiling film coefficient, Btu/(hr) (ft^2) (°F).
h' = outside film coefficient based on total outside fin tube area uncorrected for fin efficiency, Btu/(hr) (ft^2 outside surface).
h_a' = average film coefficient entire tube, Btu/(hr) (ft^2) (°F); or, heat transfer film coefficient between the insulated or bare pipe and air; see Figure 10-171; assume ϵ = 0.90 and ambient air = 70°F.
h_a = surface coefficient of heat transfer, Btu/(hr) (ft^2) (°F/ft); or, film coefficient based on arithmetic mean temperature, Btu/(hr) (ft^2) (°F).
h_b = boiling film coefficient, corrected coefficient for bundle, Btu/(hr) (ft^2) (°F).

h_c = heat transfer coefficient for outside of coil, Btu/(hr)(ft^2)(°F); or, average condensing coefficient on outside of tube, Btu/(hr)(ft^2)(°F); or, U_t = average of horizontal and vertical transfer film coefficients in still air; or, convection heat transfer film coefficient, Btu/(hr)(ft^2)(°F).
h_{cm} = average value of condensing film coefficient, for vertical rows of horizontal tube, Btu/(hr)(ft^2)(°F).
h_{fg} = latent heat of vaporization or evaporation, Btu/lb.
h_i = film coefficient of fluid on inside of tube, Btu/(hr)(ft^2)(°F).
h_{io}, h_{io}' = inside film coefficient referred to outside of tube surface (plus including condensing film coefficient clean basis for Colburn-Hougen calculations), Btu/(hr)(ft^2)(°F).
h_l = h_{lt} = nucleate boiling coefficient for an isolated tube, Btu/(hr)(ft^2)(°F).
h_{ga} = volumetric gas phase coefficient, Btu/(hr)(ft^3)(°F).
h_{la} = liquid phase heat transfer coefficient, Btu/(hr)(ft^2)(°F).
h_M = effective heat transfer film coefficient, Btu/(hr)(ft^2)(°F).
h_m = film coefficient evaluated at length mean Δt for vertical tube, Btu/(hr)(ft^2)(°F).
h_{Nu} = condensing film coefficient by Nusselt equation, Btu/(hr)(ft^2)(°F).
h_o = dry gas coefficient on shell side for partial condensers, Btu/(hr)(ft^2)(°F); or, (for bare or plain tubes) film coefficient outside of tube in bundle, Btu/(hr)(ft^2)(°F); or, (for fin tubes) outside film coefficient corrected to base of fin, Btu/(hr)(ft^2 of outside area)(°F).
h_r = radiation heat transfer film coefficient, Btu/(hr)(ft^2)(°F).
h_s = film coefficient for boiling fluid, Gilmour equation, Equation 10-165, Btu/(hr)(ft^2)(°F); or, subcooling film coefficient, pcu/(hr)(ft)(°F).
h_{st} = theoretical boiling film coefficient for single tubes, Btu/(hr)(ft^2)(°F).
htc_a = air-side heat transfer coefficient, Btu/(hr)(ft^2)(°F).
hct_c = tube-side heat transfer coefficient, Btu/(hr)(ft^2)(°F).
J = Colburn factor.
J_4 = Colburn factor for equation proposed by Pierce.
j_H = factor in heat transfer equations, dimensionless.
K = thermal conductivity of fluid, Btu/(hr)(ft)(°F). (Used by Permission: Brown Fintube Company.)
K_g = diffusion coefficient, lb mol/(hr)(ft^2)(atm).
K_L = thermal conductivity of saturated liquid, Btu/(hr)(°F/hr).
K_1, K_2, K_3, K_4 = constants in tube count equation.
K or k = thermal conductivity of material of tube, or other, wall, Btu/(hr)(ft^2)(°F/in.).
k = k_l = thermal conductivity of liquid, Btu/(hr)(°F/ft); or, thermal conductivity of insulation, Btu/(hr)(ft^2)(°F/ft); or, when specified, Btu/(hr)(ft^2)°F/(in.).
k or k_a = thermal conductivity of fluid at average of temperatures, t_1 and t_2, Btu/(hr)(°F/ft); or, Btu/(hr)(ft^2)(°F/ft).
k_f = thermal conductivity at film temperature, Btu/(hr)(ft^2)(°F/ft).
k_g = gas phase mass transfer coefficient, lb mol/(hr)(ft^2)(atm).
k_d = diffusivity, ft^2/hr.
k_L = thermal conductivity of saturated liquid, Btu/(hr)(°F/ft).
k_w = thermal conductivity of material of wall, (Btu-ft)/[(hr)(ft^2)(°F)].
L = length of straight tube for heat transfer (or, nominal tube length), ft; or, length of path, ft; or, length or thickness of coil or jacket, ft; or, superficial liquid mass velocity, lb/(hr)(ft^2); or, equivalent length of pipe, ft; or, thickness of insulation, in.
L_c = thickness, ft.
L_e = net effective tube length, ft.
L_f = finned length of each tube, ft.
L_o = length of shell; or, length of one tube pass, ft.
L_t = total length of tubing required, ft.
L_w = thickness of tube wall, in.; or, ft as consistent with other units.
L_v = weighted average latent heat of reboiler vapor, Btu/lb.
LMTD = log mean temperature difference, °F.
l = length of each individual tube, ft; or, fin height, in.
l_v = latent heat of vaporization, Btu/lb.
M = net free distance (sum) of space between tubes from wall to wall at center of shell circle, in.; or, molecular weight.
M_a = average molecular weight.
M_v = average molecular weight of vapor.
m = exponent for baffle tray columns.
N or N_t = total number of tubes in bundle.
N = number of tube holes per tubesheet; or, number of fins per in. on tube; or, agitator or impeller speed, revolutions/hr; or, number of tubes in bundle; or, number tubes/row in direction of air flow.
N_a = average number of tubes in a row in circular shell of multitube condenser.
N_c = number of baffles.
N_{nv} = number of holes in vertical center row of bundle.
N_t' = number of tubes used, for condensing.
N_v = number of rows of tubes in a vertical tier.
N_{vc} = number of rows of tubes in a vertical tier at centerline of exchanger.

N_{Pr} = Prandtl number.
N_{Re} = Reynolds number.
Nu = Nusselt number.
n = number of tube passes; *also*, in Hajek reboiler relation, $n = \log U\Delta T/\log\Delta T$; or, agitator revolutions/min; or, number of horizontal tubes in a vertical bank; or, number of equilibrium contact stages; or, exponent for baffle tray columns; or, number of tubes/row, per ft of exchanger width, 1/ft.
n' = number of tubes per pass; or, effective surface efficiency; or, number of tracers.
n'' = number of rows of tubes between centers of gravity of two adjacent segmental baffles, for finned tubes.
n_b = number of baffles.
n_c = minimum number of tube rows fluid crosses in flowing from one baffle window to an adjacent baffle.
n_n = number of tubes in a row at the exchanger centerline normal to the direction of flow.
n_s = number of fluid streams in shell side of condenser.
n_w = number of tubes passing through baffle window.
NPS = nominal pipe size.
P = absolute pressure, lb/in.^{2} abs; or, total pressure lb/ft^2 abs. See Appendix A-4 and A-8 for additional information; *also*, P = factor in LMTD correction = $(t_2 - t_1)/(T_1 - t_1)$.
p_A = total pressure at point A in flow loop, psi, abs.
P_B = boiling pressure, lb/in.^{2}, abs.
P_c = critical pressure, psi abs.
P_r = reduced pressure = absolute pressure/absolute critical pressure.
P_f or p = tube pitch, in. or ft, consistent units.
p = total pressure, psi abs; or, tube pitch, in.
p_a = number of tube passes in exchanger.
p_B = total pressure at point B in flow loop, lb/in.^{2} abs.
p_c = partial pressure of vapor in condensate film, atm.
p_{fg} = log mean pressure difference of the inert gas between p_g and p_g', atm.
p_g = partial pressure of inert gas in the main gas body, atm = $(P - p_v)$, psi abs or atm.
p_g' = partial pressure of one inert at the condensate film, atm or psi, abs.
p^o = partial pressure inert gas at condensate film, atm or psi, abs.
p_s = static pressure, in. water.
p_t = total pressure, in. water.
p_t' = total pressure, atm.
p_v = velocity pressure, in. water; or, = Colburn-Hougen calculation; or, vapor pressure of condensate at t_g, psia or atm; or, partial pressure condensing vapor, psia, or atm.
P_V' = partial pressure of vapor in gas body, atm.
ΔP = pressure loss, lb/ft^2.
Δp = pressure loss, lb/in.^{2}
Δp_b = pressure loss across or through the "window" opening of segmental baffles, lb/in.^{2}
Δp_c = pressure loss across the tube bundle in cross flow, lb/in.^{2}
$\Delta p_{long.}$ = shell-side pressure drop due to longitudinal flow, lb/in.^{2}
Δp_r = pressure loss through return ends or channels of tube side of exchanger, lb/in.^{2}
ΔP_s = pressure drop of fluid, heated or cooled, including entrance and exit losses, lb/ft^2.
Δp_s or p = shell-side pressure drop, lb/in.^{2}
Δp_t = pressure drop through tubes, lb/in.^{2}
Δp_{tt} = tube side of exchanger total pressure drop, lb/in.^{2}
Pe = Peclet number.
Q = total heat load, or transferred, Btu/hr (see q'); or, heat loss from pipe insulation, Btu/(hr)(ft^2)
Q_{ap}, Q_{ia}, Q_{pa}, Q_{ta}, Q_{tp} = heat transfer related to pipe, annulus and pipe tracer.
Q_b = heat duty for boiling, Btu/hr.
Q_c = heat load of overhead condenser (removed in condenser), Btu/hr.
Q' = total rate at which a black body emits heat radiation of all wave lengths, Btu/hr.
Q_R = reboiler duty, or heat added, Btu/hr.
Q_s = sensible heat transfer duty, Btu/hr.
Q_T = total heat transfer duty, Btu/hr.
q = heat transferred, Btu/hr, usually used as identification for individual heat quantity (see Q); or, heat loss per lin ft of pipe, Btu/(hr)(lin ft); or, heat loss through wall, Btu/lin ft.
q or q_{max} = tube bundle maximum heat flux for boiling, Btu/(hr)(ft^2).
q' = sum of latent and sensible heat duty/load, Btu/hr.
q_s = rate of heat transfer per ft^2 of outer surface of insulation, Btu/(hr)(ft^2).
q_t = heat transfer, Btu/(hr)(ft^2)(°F).
R = factor in LMTD correction = $(T_1 - T_2)/(t_2 - t_1)$; or, factor from Table 10-46; or, mean radius of bend, in.; or, reflux ratio, mol condensate returned/mol product withdrawn; or, volume fraction of phase, dimensionless.
R* = asymptotic value for fouling resistance, (hr)(ft^2)(°F)/Btu; see Figure 10-43C.
R_e = Reynold's number, expressed in units suitable for application.
R_f = fouling resistance, (hr)(ft^2)(°F)/(Btu × 10^4).
R_g = volume fraction of gas phase, dimensionless.
R_L or R_l = $(1 - R_g)$, volume fraction of liquid phase, dimensionless.
R_s = outside surface resistance, (°F)(hr)(ft^2)/Btu.
R_t = total resistance to heat transfer, (hr)(°F)(ft^2)/Btu.

R_{oa} = overall resistance to heat transfer, (hr) (ft^2) (°F)/Btu.
r_b or r_o = fouling resistance (factor) with fluid on outside of tube, (hr) (ft^2) (°F)/Btu.
r_h = hydraulic radius, ft.
r_i or r_t = fouling resistance (factor) associated with fluid on inside of tube, (hr) (ft^2) (°F)/Btu.
$r_{f,t}$ = tube-side fouling resistance, (hr) (ft^2) (°F)/Btu.
$r_{f,a}$ = air-side fouling resistance, (hr) (ft^2) (°F)/Btu.
r_l = longitudinal tube pitch, in./tube O.D., in. for cross-flow pressure drop, in direction of fluid flow; or, outside radius of *any* (if used) intermediate layer of insulation, in.
r_o = inside radius of pipe insulation, in.
r_s = outside radius of pipe insulation, in.
r_t = transverse tube pitch, in./tube O.D., in.; for cross-flow pressure drop, transverse to direction of flow.
r_w = resistance of tube wall, L_w/k_w; (hr) (ft^2) (°F)/Btu.
Sc = Schmidt number.
S_e = weighted flow area = [(cross-flow area) (baffle window area)]$^{1/2}$, ft^2.
St = Stanton number.
s = specific gravity of fluid referred to water; or, inside surface of pipe, ft^2.
s_f'' = fin surface per ft of length.
s_o'' = plain pipe/tube surface per ft of length.
T = temperature of hot fluid, °F; or, absolute temperature, °R; or, tank diameter, ft.
T_{ap} = annulus space temperature, °F.
T_k = temperature absolute, °Kelvin.
T_m = pipe temperature, °F.
T_p = process temperature, °F.
T_r = absolute radiation temperature, °R = °F + 460.
T_s = saturation temperature of liquid, °R; or, steam temperature, °F; or, surface temperature of insulated or bare pipe in contact with air, °F.
T_w = temperature of heating surface, °R; or, wall temperature of tube, °F.
T_1 = inlet temperature of hot fluid, °F; or, inlet fluid to tubes of air cooled exchanger, °F.
T_2 = outlet temperature of hot fluid, °F; or, outlet process fluid temperature from air-cooled exchanger, °F.
T′ = °R (degrees Rankine).
t = temperature , °F; or, tube wall thickness, in.; or, temperature of cold fluid, °F; or, temperature of a fluid or material, °F; or, air temperature.
t_a or T_a = ambient temperature, °F; or, arithmetic average tube-side fluid temperature, °F.
t_b = bulk temperature of fluid, °F.
t_c = temperature of condensate film, °F; or, caloric temperature of cool fluid, °F; or, temperature of condensate film in partial condenser calculations, °F.
t_g = temperature of dry gas (inerts), °F.
t_h = caloric temperature of hot fluid, °F.
t_i = inside tubes fluid temperature, °F.
t_o = fluid outside tubes temperature, °F; or, temperature of inner surface of insulation °F.
t_s = boiling fluid temperature, °F; or, temperature of outer surface of insulation, °F.
t_{sv} = temperature of saturation or dew point, °F.
t_v = temperature of vapor, °F.
t_w = temperature of tube wall, °F.
t_w' = temperature of inlet water, °F.
t_{wo} = temperature of outside tube wall, °F.
t_1 = inlet dry bulb air temperature to air-cooled exchanger, °F.
t_2 = outlet dry bulb air temperature from air-cooled exchanger, °F
ΔT = overall temperature difference, °F; or, mean temperature difference between bulk of boiling liquid and bulk of heating medium, °F; or, temperature difference of fluid, (end points),°F; or, ($T_w - T_s$), °R, in Levy Boiling equation.
ΔT_b = mean temperature difference between bulk of boiling liquid and tube wall, °F; or, temperature drop across boiling film, °F.
Δt = temperature difference, ($t_2 - t_1$),°F; or, log mean temperature difference based on limits (unless defined otherwise), °F; or, temperature difference ($t_1 - t_{avg}$) between inside surface of pipe insulation and average outside air temperature, °F.
Δt_b = temperature drop across boiling film, between boiling fluid and wall surface on boiling side, °F.
ΔT_L = actual liquid temperature rise, °F.
$\Delta T_{L,max}$ = maximum possible liquid temperature rise (to vapor inlet temperature).
Δt_o = overall Δt between average tube-side bulk temperature, °F, between average tube-side bulk temperature and boiling film, °F; or, critical temperature differential in boiling, °F.
Δt_m = mean temperature difference, ($t_v - t_w$)/2, °F; or, corrected mean temperature difference, °F.
Δt_{ca} = overall temperature drop from average tube-side temperature to shell boiling temperature, °F.
$\Delta T_{V,max}$ = Maximum possible vapor temperature decrease (to liquid inlet temperature).
U, U_o = overall heat transfer coefficient corrected for fouling conditions, Btu/(hr) (ft^2) (°F).
U_a = volumetric overall heat transfer coefficient, Btu/(hr) (ft^3) (°F).
U_L = overall coefficient of heat transfer per ft of length, Btu/(hr) (lin ft) (°F).
U_l = single tube overall heat transfer coefficient, Btu/(hr) (ft^2) (°F).

V = vapor rate, lb/hr; *also*, used as molecular volume for component in diffusivity equation; or, linear velocity, ft/sec.
V_f = superficial flooding velocity of vapor, ft/sec.
V_{max} = maximum vapor velocity in exit from reboiler, m/sec.
V_{RB} = vapor formed in reboiler, lb/hr.
v = tube-side velocity, ft/sec; or, ft/hr.
v_i = specific volume of fluid at inlet to reboiler, ft^3/lb.
v_o = specific volume of fluid at outlet of reboiler, ft^3/lb.
v_s = superficial vapor velocity, usually, ft/sec.
W = mass rate of flow, lb/hr condensate for condensers; or, lb/sec; or, width of air-cooled exchanger, ft.
W′ = flow rate, lb/sec.
W″ = tube loading for condensing, lb/hr (ft of finned tube length).
W_s = shaft work done by system, ft liquid.
W_T = total vapor condensed in 1 tube, lb/hr; or, total mass flow rate, lb/sec.
w = weight rate of fluid flow per tube, lb/hr/tube; or, rate of condensation per tube from lowest point of vertical tube(s), lb/(hr)(tube); or, = fin height, in.
X = $N_{Re}/1000$.
X_{tt} = correlating parameter, dimensionless for turbulent-turbulent flow mechanism, Figure 10-113.
x = vapor quality in fluid, weight fraction, nucleate boiling; or distance film has fallen; or, weight fraction of vapor or gas, dimensionless.
x_2 = height of liquid level in column above reboiler bottom tubesheet, ft.
x_c = distance from top (effective) of tube, ft.
y = mol vol % noncondensables in bulk stream.
Z = distance or length, or vertical height, ft or in. as consistent; or, viscosity at flowing temperature, centipoise. (Used by permission: Brown Fintube Company.)
Z_{sp} = height of individual spray zone, ft.
ΔZ = height (vertical) of driving leg for thermal circulation, ft.

Greek Symbols

α = constant, Gilmour boiling equation; or, tube layout angle, degree = coefficient of thermal expansion of fluid, %/°F; or, correction factor for convective heat transfer, dimensionless; or, correction factor for nucleate boiling, dimensionless; or, Ackerman correction factor, dimensionless.
β′ = coefficient of expansion = $1/T_r$, with T_r in °Rankine = °F + 460.
Γ = mass rate of flow of condensate from lowest point on condensing surface divided by breadth (unit perimeter), lb/(hr)(ft).
γ = acceleration loss group, dimensionless; see Equation 10-181.
Δ = difference.
$(\Delta t/\Delta p)_s$ = slope of vapor pressure curve.
ε = emissivity of a surface, ratio of total rate at which a gray surface emits radiation to the total rate at which a black body at the same temperature would emit radiation of all wave lengths, dimensionless; or, emissivity of the outside surface of insulated or bare pipe.
δ = fin width at fin base, ft.
η_f = fin efficiency, fraction.
η_w = e_f, weighted fin efficiency, fraction.
θ = tube taper angle (to horizontal), degrees.
θ^2 = effective average (two-phase)/liquid phase pressure drop ratio corresponding to effective average vaporization.
λ = latent heat of vaporization, Btu/lb.
μ = viscosity of fluid, lb/(hr)(ft); or, centipoise (as specified); or, viscosity at the caloric temperature, lb/(ft)(hr).
μ = viscosity of fluid at film temperature, lb/(hr)(ft) = (centipoise) × 2.42 = lb/(hr)(ft).
μ′ = viscosity, centipoise.
μ_f' = absolute viscosity, lb/sec (ft) = (centipoise) (0.000672).
μ_s = viscosity of fluid at film temperature of heat transfer surface, lb/(hr)(ft).
μ_w = viscosity of fluid at wall temperature, lb/(hr)(ft); or, centipoise.
π = 3.1416.
ρ = density, lb/ft^3.
ρ_t = πd_o for vertical tube perimeter, ft.
ρ_{tp} = effective average two-phase density, $lb.ft^3$
ρ_{tph} = homogeneous two-phase density, kg/m^3.
σ = surface tension, lb/ft; or, dynes/cm as specified.
σ′ = surface tension of liquid, Btu/ft^2; or, (dynes/cm)(0.88×10^{-7}) = Btu/ft^2.
σ″ = surface tension of liquid, dynes/cm.
σ_s or σ = Stefan-Boltzmann constant, 0.173×10^{-8} $Btu/(hr)(ft^2)(°R)^4$.
ϕ = constant, Gilmour boiling equation; or, surface condition factor, Equation 10-140; or, tube density coefficient, Table 10-27.
$\bar{\phi}$ = average value of ϕ.
$\underline{\phi}$ = physical property factor; or, parameter for two-phase flow, dimensionless.
ϕ_p' = physical property factor for shell-side heat transfer.
ϕ_s = $(\mu/\mu_w)^{0.14}$, subscript $_w$ refers to wall condition.
ψ = maximum flux physical property factor, $Btu/(ft^3)(hr)$, Equation 10-149.

Subscripts

$\bar{\ }$ = bar over symbol = average value.
1 = inlet condition.
2 = outlet condition.
A = point A in flow loop, Figure 10-110.
B = point B in flow loop, Figure 10-110.
A, B, etc. = component identification.
A_v or avg. = average of limits of function, inside and outside tube area for unit length, ft^2/ft.
D = dirty or under operating conditions; or, overhead distillate.
E = exit reboiler vapor.
F = friction.
L = liquid.
T = total.
b = boiling condition; or, bottoms material.
c = clean, or cold end; or, correction; or, condensing; or, critical condition.
d = diffusional.
f = film conditions; or, force.
f_p or f = process-side fouling; or, tube-side fouling.
g = gas.
h or hot = hot end; or, hot fluid; or, heating medium.
i = inside
l or L = liquid.
lm or Lm = log mean.
m = mass.
o = outside
p = process on boiling side.
m = mean; or, metal.
max = maximum.
min = minimum.
r = radiation.
s = steam; or, shell-side of exchanger; or, surface; or, saturation temperature.
sv = saturated vapor.
t = tube; or, total.
tp = two phase.
v = vapor.
w = wall or surface.
air = air side.
cold = cold side, may = air.

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Chapter

11

Refrigeration Systems

Process refrigeration is used at many different temperature levels to condense or cool gases, vapors, or liquids. Refrigeration is necessary when the process requires cooling to a temperature not reliably available from the usual water service or other coolant source, including Joule-Thompson, or polytropic expansion of natural gas or process system vapors.

In general, auxiliary refrigeration is used for temperature requirements from 80–85°F to as near absolute zero as the process demands. The usual petrochemical and chemical range does not go much below −200°F. This section does not include low-temperature air separation for oxygen, nitrogen, argon, etc., or the separation of process gases at liquid air temperatures.

A valuable technical presentation of refrigeration is given in the ASHRAE handbook.[2]

Types of Refrigeration Systems

The three most used systems are as follows:

System	Approx. Temperature Coolant Range, °F	Refrigerant
Steam jet†	35 to 70°F	Water
Absorption		
Water-Lithium Bromide	40 to 70°F	Lithium Bromide Solution (water*)
Ammonia	−40 to +30°F	Ammonia*-water
Mechanical compression (Reciprocating centrifugal or rotary screw)	−200° to +40°F	Ammonia, halogenated hydrocarbons, propane ethylene, and others
Plus:		
Cryogenics	−150° to −200°F	Liquefaction of gases, and power/temperature recovery from natural gas

* refrigerant

†Vacuum system, discussed in detail, Chapter 6, Vol. 1, 3rd Ed., this text series.

The most common light hydrocarbon refrigerant cooling temperature ranges are (evaporation temperature):

Methane	−200° to −300°F
Ethylene and ethane	−75° to −175°F
Propylene and propane	+40° to −50°F

Mehra[8–11] has developed a valuable series of working charts for the common industrial refrigerants along with application examples for ethylene, propylene, ethane, and propane.

Terminology

Ton of refrigeration: The heat equivalent to melting 2,000 lb (one ton) of ice in 24 hours. One ton equals 12,000 Btu/hr or 200 Btu/min. To be comparative, refrigeration equipment must have the refrigerant level (or evaporation temperature) specified.

Selection of a Refrigeration System for a Given Temperature Level and Heat Load

In general, the simplest system is selected for any specific refrigeration requirement, because it should be the least expensive from first cost and operating cost viewpoints. Factors that are weighed in arriving at the process-directed refrigeration system include the following, in addition to the purchase and operating costs:

1. Temperature Level of Evaporating Refrigerant

Refrigerant temperatures greater than 32°F suggest the steam jet or lithium bromide absorption system. Between 30°F and −40°F, the ammonia-water absorption or a mechanical compression system is indicated. At less than −40°F, a mechanical compression is used, except in special desiccant situations. The economics of temperature level selection will depend on utility (steam, power) costs at the point of installation and the type of pay-out required, because in some tonnage ranges, the various systems are competitive based on first costs.

In most process systems, the evaporation of the refrigerant is carried out in shell and tube heat exchange equipment, and allowance must be made for a reasonable temperature approach between the process fluid and the evaporating refrigerant. The process fluid always leaves the evaporator at a higher temperature (by 3°–15°F) than the refrigerant.

2. Suction of Absorbing Pressure of Refrigerant

When the process circulating coolant and required refrigerant evaporator temperature level are established, the suction pressure to the compressor of a mechanical machine or the absorbing pressure of an absorption system is set. Keep the low pressure point of the system at atmospheric pressure or above in order to avoid air in-leakage that would later have to be purged. This is not possible for some systems; however, the issue is important and must be recognized, as explosive mixtures can be formed with some refrigerants. Moisture also enters the systems with air in-leakage.

3. Discharge or Condensing Pressure

In mechanical systems, the temperature of the available water (or coolant) to condense the refrigerant from the compressor determines the pressure level of this part of the system. Generally speaking, it is less expensive to operate at as low a pressure level on the discharge as is consistent with the suction pressure and with the physical characteristics of the refrigerant. Sometimes the cost of the refrigerant and the cost of its replacement on loss dictate that the optimum situation is not determined by the system and refrigerant's physical properties.

4. Refrigerant Characteristics

Available refrigerants for various levels or conditions of operation may be toxic, flammable, irritating on exposure, hydroscopic, and expensive. These characteristics cannot be ignored, as large systems contain large quantities of refrigerant, and a leak or other failure can release a potentially serious condition into a building or process area.

The thermodynamic properties of the refrigerant determine the suitability for a given condition of operation, particularly when compared with the same requirements or other refrigerants. The quantity of refrigerant needed for a particular level of evaporation is a function of its latent heat, except when using steam jet refrigeration, because the use of its chilled water involves only sensible heat transfer to process fluids.

5. System Maintenance

The maintenance requirements for operation of the different types of refrigeration systems vary somewhat and should be evaluated along with the particular performance.

Steam Jet Refrigeration

In steam jet refrigeration, water is the refrigerant being evaporated at low pressures created by the steam jets. These units have a barometric direct contact condenser, Figure 11-1, or a surface condenser as in Figure 11-2. The former system wastes or loses the steam in the water; whereas the latter allows the steam to be condensed and reused.

This type of system may be used to cool water on a once-through basis where the water is wasted, or the water may be in a closed system, and the refrigeration unit used to remove the heat taken up by the circulation to another part of the process. The warm water enters the active compartment(s), depending upon the refrigeration load, and flashes due to the reduced pressure maintained by the steam jets. The water boils at this low pressure, and the vapor is drawn into the booster jet where it is compressed for condensation in the direct contact or surface condenser. The cooled or chilled water is pumped from the compartment with a pump capable of handling the boiling water *at this pressure.* Figure 11-3 indicates the expected water temperature and corresponding pressure in the active or flash compartment.

In general, the number of boosters determines the operational flexibility of the unit with respect to the refrigeration load. A single booster unit operates continuously, regardless of load. A two booster unit can operate at 50% load by shutting off one unit; at lower load levels it uses a pressure controller on the steam actuated by the condenser pressure. Because jets are not usually very flexible with respect to steam consumption and vacuum, load control may be in increments as compared to continuous variation. If a 100-ton unit is expected to operate an appreciable portion of the time at 25% of load, it may prove economical to install a four-booster unit and to operate only one for this period. Auxiliary ejectors remove uncondensed water vapor and air from the main condenser.

Low-head jet units without a barometric leg use a pump to withdraw the water from the barometric condenser. This pump must be carefully selected as it must operate under vacuum suction conditions.

Advantages of Steam Jet Units

Assuming water greater than 35°F is needed and that water and steam costs are reasonable:[5]

1. No moving parts exist, except water pumps.
2. Refrigerant (water) is nonhazardous and has a low cost.
3. System pressures are low.
4. Unit can be installed outdoors, if desired, and is self supporting.
5. Physical arrangement is flexible and can be made to fit odd spaces.
6. Steam and cooling water requirements can be adjusted to reasonable economical balance.

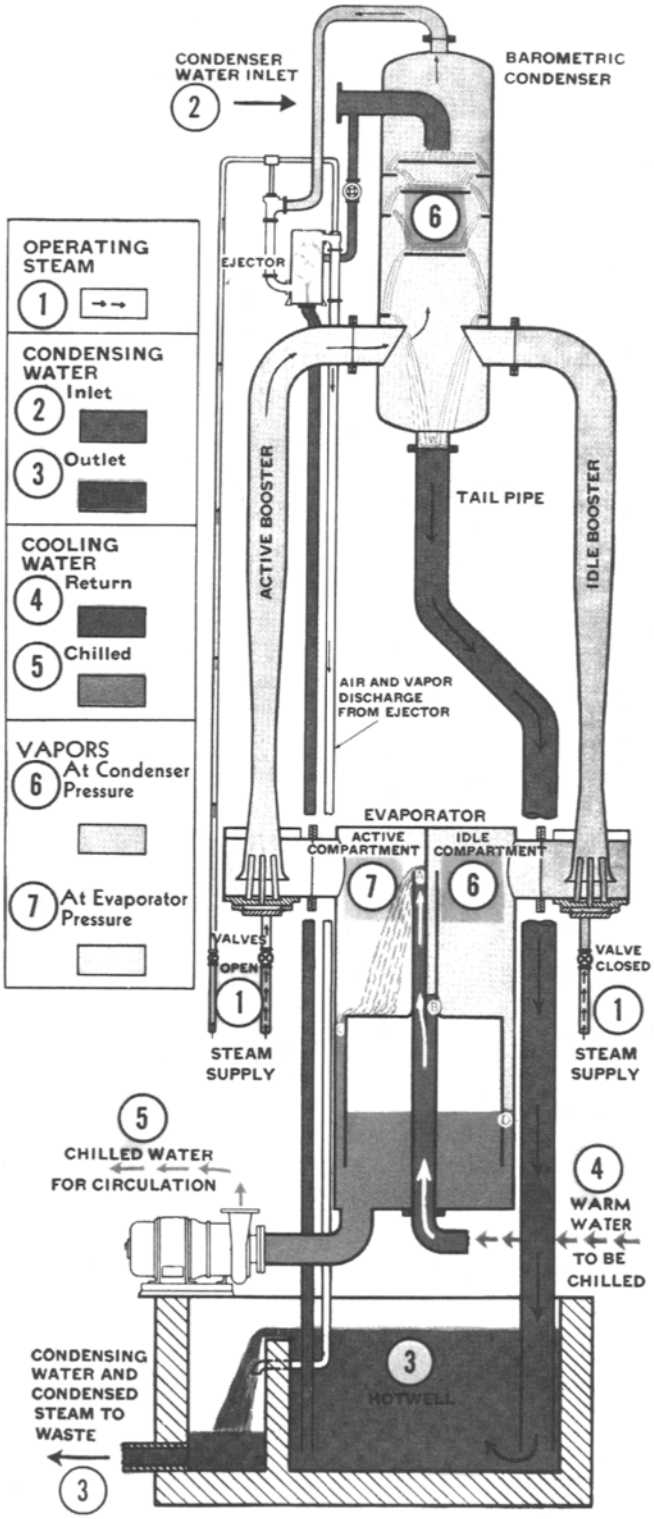

Figure 11-1. Multibooster barometric refrigeration unit with barometric condenser. (Used by permission: ©Ingersoll-Rand Company.)

7. Steam condensate can be recovered in surface type units.
8. Refrigeration tonnage can be varied as to amount and temperature level.
9. Start up and operation are simple.
10. Barometric units can use dirty, brackish, or waste water.
11. Cost per ton of refrigeration is relatively low.

Materials of Construction

The main fabricated parts of the units are carbon steel, with suitable corrosion allowance for the conditions of the chilled and condensing water. When brackish or sea water is used in a barometric condenser, steel construction with a $^1/_4$-in. to $^3/_8$-in. corrosion allowance is suggested, and minimum wall plates of $^1/_2$-in. to $^3/_4$-in. may be justified. Internal splash plates should be $^1/_2$-in. to $^3/_4$-in. minimum, because the atmosphere of water vapor-air is very corrosive. Alloy construction is not justified except in exceptional cases.

For surface condensers, the tubes, tubesheets, and shell should be consistent with experiences in heat exchanger construction. In sea or brackish water, one of the cupronickels or aluminum brass may be a good choice for tubes. The water boxes may be vertically divided to allow half of the unit to operate while the other half is being opened for repair or inspection.

The booster ejectors are usually of steel plate (or cast) with Monel steam nozzles.

The air ejectors are usually of cast iron with Monel nozzles. The associated inter- and after-condensers are usually of cast iron shell and water boxes with Admiralty tubes (unless sea or brackish water) with Muntz metal tubesheets. Some inter- and after-condensers may also be barometric rather than tubular.

These units usually come complete with interconnecting piping, valves, strainers, control valves, level controls, gages, water pumps, etc. The specifications should state how much of this is desired by the purchaser, as well as delineating each detail peculiar to the system, such as the use of sea or brackish water, special materials of construction for condenser water, steam, chilled water, etc.

The dimensions of Figure 11-4 for barometric-type units and Figure 11-5 for surface-type units are a general guide as to space requirements.

Performance

The two basic types of steam jet units as shown in Figure 11-6 are

- Recirculated water
- Once-through water

These steam jet units refer to the water that is being chilled for process use. This use may be direct-process injection or

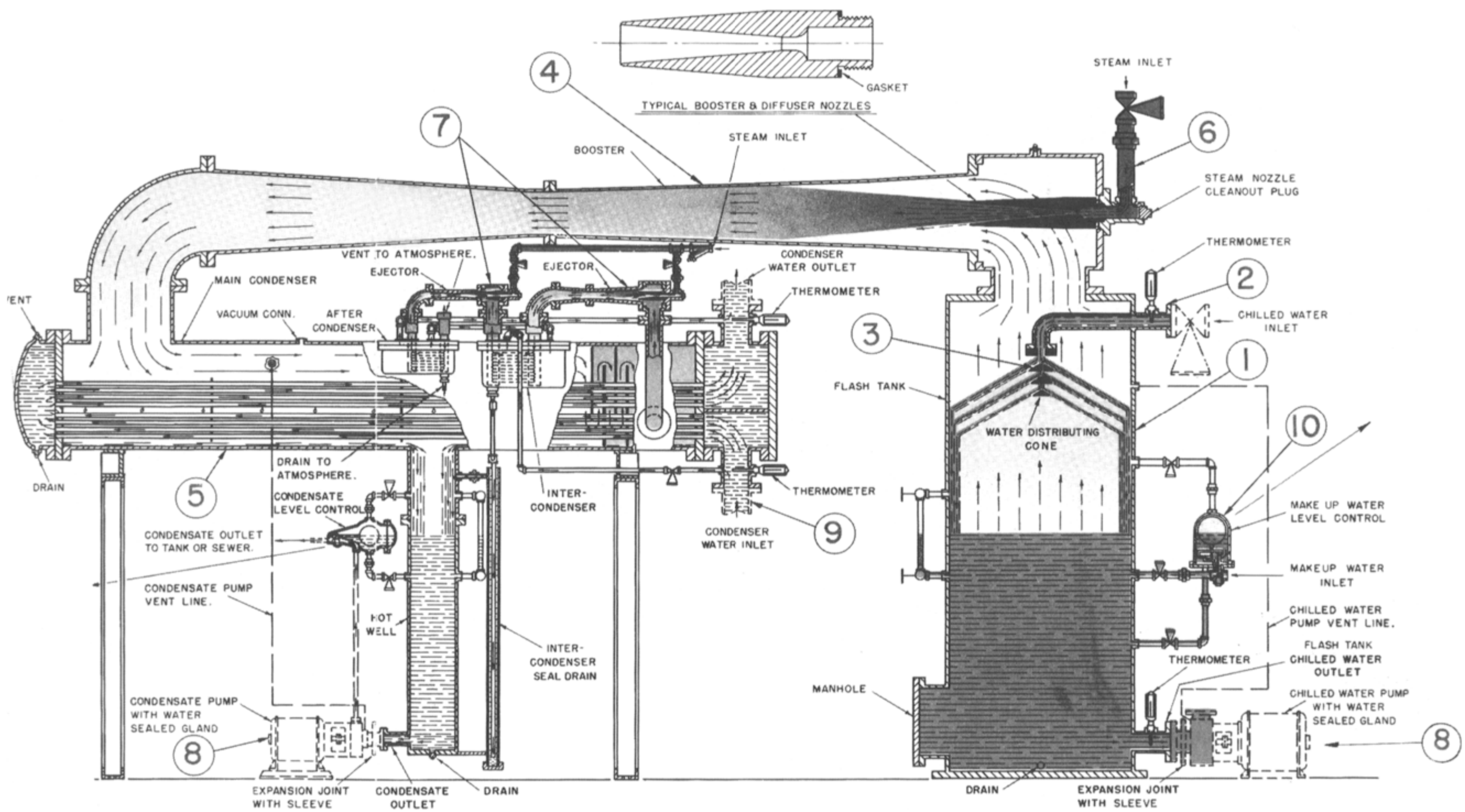

Figure 11-2. Surface condenser steam jet refrigeration unit. (Used by permission: ©Graham Manufacturing Company, Inc.)

indirect for heat exchangers or similar equipment. Water for a once-through system need only be as free from impurities as its use and reasonable corrosion conditions require; whereas the recirculated water is usually condensate (with condensate make-up) with blow-down or treatment to avoid solids or other contaminant build up. The principles of design and selection are the same for the two types.

Capacity

$$\text{Tons of refrigeration required} = \frac{\text{Heat load of process per hr}}{12{,}000 \text{ Btu/hr.}} \tag{11-1}$$

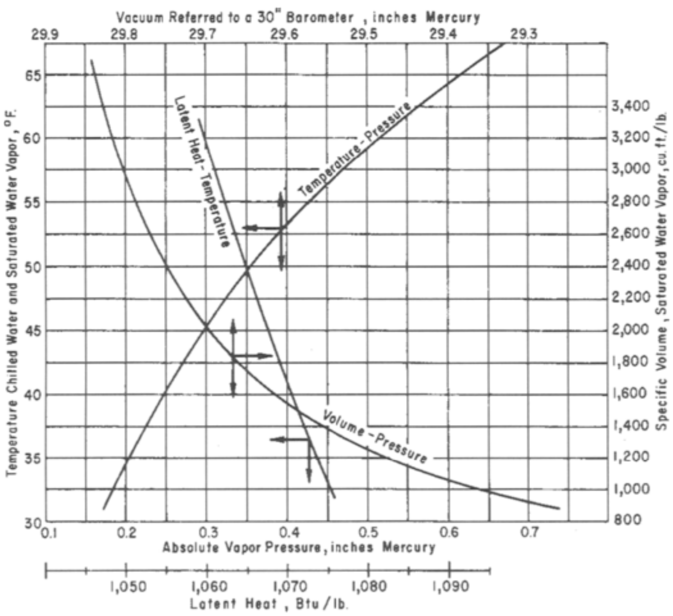

Figure 11-3. Water-vapor relationship in steam jet refrigeration unit.

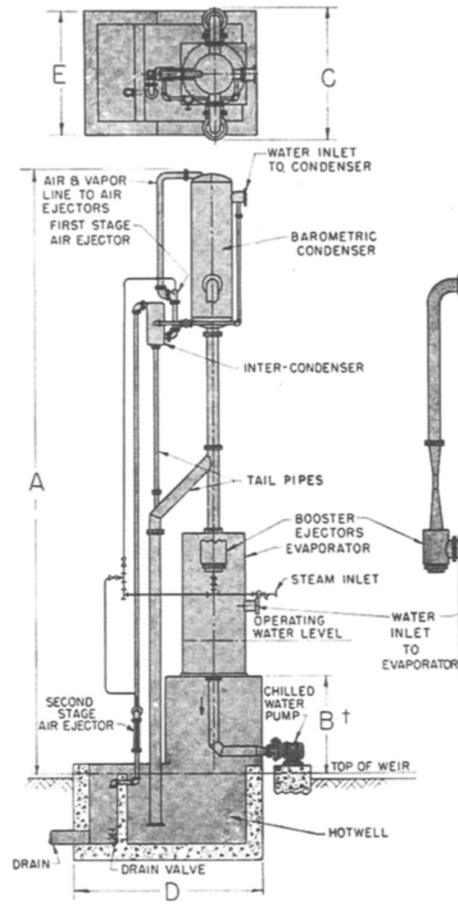

Figure 11-4. (Above and below) Barometric-type condenser, steam jets with two boosters. (Used by permission: ©Ingersoll-Rand Company.)

†Based on required N.P.S.H. of pump. May be reduced by pump pit or by pump requiring less N.S.P.H.

No. of Boosters	Tons* of Refrig. at 50°F	Approximate Dimensions				
		A	B	C	D	E
1	30	46 ft-0 in.	4 ft-6 in.	6 ft-0 in.	8 ft-6 in.	5 ft-0 in.
1	40	48 ft-6 in.	4 ft-6 in.	6 ft 6 in.	9 ft-0 in.	5 ft-0 in.
1	50	48 ft-6 in.	4 ft-6 in.	6 ft 6 in.	9 ft-0 in.	5 ft-6 in.
2	60	50 ft-0 in.	4 ft-6 in.	9 ft 6 in.	9 ft-6 in.	6 ft-0 in.
2	80	51 ft-0 in.	4 ft-6 in.	9 ft 6 in.	10 ft-6 in.	6 ft-6 in.
2	100	52 ft-0 in.	5 ft-6 in.	10 ft 6 in.	11 ft-0 in.	7 ft-0 in.
2	125	53 ft-0 in.	5 ft-6 in.	12 ft 0 in.	12 ft-0 in.	7 ft-6 in.
2	160	54 ft-0 in.	5 ft-6 in.	13 ft 0 in.	13 ft-0 in.	8 ft-0 in.
3	200	56 ft-0 in.	5 ft-6 in.	12 ft 6 in.	14 ft-6 in.	9 ft-0 in.
3	250	57 ft-6 in.	6 ft-0 in.	13 ft 6 in.	15 ft-6 in.	10 ft-0 in.
4	320	60 ft-0 in.	6 ft-0 in.	13 ft 0 in.	17 ft-6 in.	11 ft-6 in.
4	400	62 ft-6 in.	7 ft-0 in.	14 ft 0 in.	18 ft-6 in.	12 ft-0 in.

*See Figure 11-7 for estimating capacities at other temperatures.

No. of Boosters	Tons* of Refrig. at 50°F	A	B	C	D	E*	F
		Approximate Dimensions					
1	30	5 ft-0 in.	16 ft-0 in.	14 ft-0 in.	2 ft-6 in.	5 ft-3 in.	4 ft-0 in.
1	40	5 ft-6 in.	17 ft-0 in.	14 ft-0 in.	2 ft-10 in.	5 ft-3 in.	4 ft-0 in.
1	50	6 ft-0 in.	18 ft-0 in.	14 ft-6 in.	3 ft-0 in.	5 ft-3 in.	4 ft-0 in.
2	60	12 ft	12 ft-0 in.	14 ft-0 in.	4 ft-6 in.	5 ft-3 in.	4 ft-0 in.
2	80	13 ft	13 ft-6 in.	14 ft-0 in.	5 ft-0 in.	5 ft-3 in.	4 ft-0 in.
2	100	15 ft	14 ft-0 in.	15 ft-6 in.	5 ft-6 in.	6 ft-3 in.	5 ft-0 in.
2	125	15 ft	15 ft-0 in.	16 ft-0 in.	6 ft-0 in.	6 ft-3 in.	5 ft-0 in.
2	160	15 ft	17 ft-0 in.	16 ft-0 in.	7 ft-0 in.	6 ft-3 in.	5 ft-0 in.
3	200	16 ft	16 ft-0 in.	16 ft-0 in.	9 ft-0 in.	6 ft-3 in.	5 ft-0 in.
3	250	16 ft	17 ft-6 in.	17 ft-0 in.	10 ft-0 in.	6 ft-9 in.	5 ft-6 in.
4	320	18 ft	18 ft-0 in.	17 ft-0 in.	13 ft-0 in.	6 ft-9 in.	5 ft-6 in.
4	400	20 ft	20 ft-0 in.	18 ft-0 in.	15 ft-0 in.	7 ft-9 in.	6 ft-6 in.

*See Figure 11-7 for estimating capacities at other temperatures.

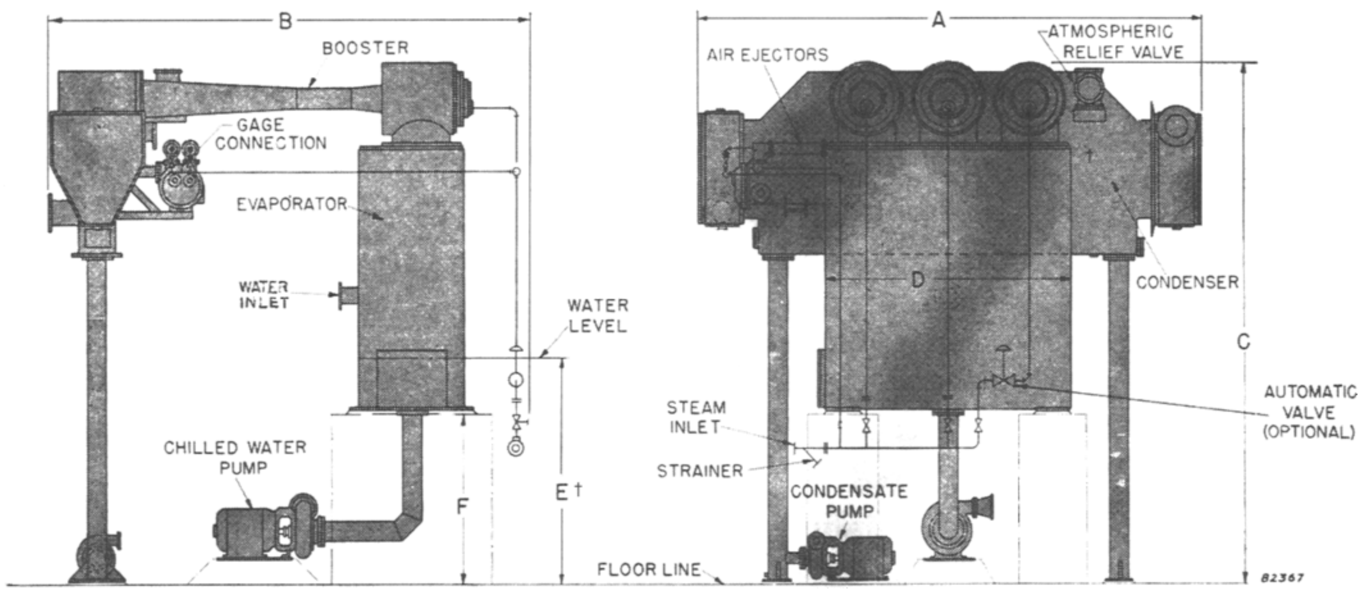

Figure 11-5. Surface-type condenser, steam jets with three boosters. (Used by permission: ©Ingersoll-Rand Company.)

†Based on required N.P.S.H. of pump. May be reduced by pump pit or by pump requiring less N.P.S.H.

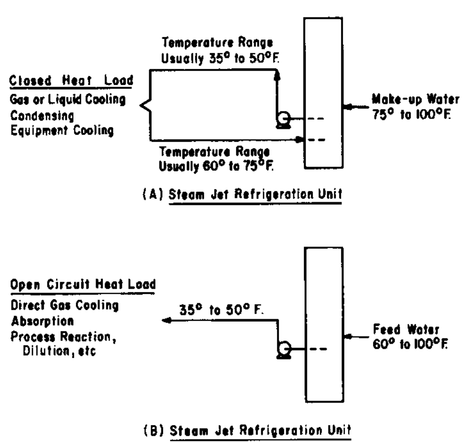

Figure 11-6. Steam jet refrigeration systems.

Tons of steam jet refrigeration

$$= \frac{(\text{gpm water cooled})(\Delta t \text{ water drop °F})}{24} \tag{11-2}$$

By means of this relation, the effect of water-flow rate and the required temperature drop in the refrigeration unit can be visualized. Thus, if the water requirement increases, but is at a smaller Δt, it is possible that an existing unit may be capable of handling the load.

Figure 11-7 indicates the effect of temperature level on capacity of a given unit. The dotted line indicates that 50°F is the reference chilled water temperature from the unit at 100% capacity. Any other temperature may be used as a reference and the percentage change in tonnage noted for change in water-off temperature. A unit producing 100 tons at 50°F chilled water will also produce 115 tons when the chilled water is used at 55°F or will produce only 70 tons when the chilled water off the unit is 40°F.

The thermal efficiency of the units is approximately:

Operating Steam Pressure, psig	Thermal Efficiency, %
15	30
100	60
400	80

Operation

At the low absolute pressure of the flash chamber, the entering water partially evaporates and in so doing absorbs heat from the bulk of the water in the compartment. The latent heat of steam (greater than 1,000 Btu/lb) at the evaporator pressure is removed and the water in the compartment is cooled an equivalent amount. Figure 11-8 indicates the conditions for one system.

Although it is usually not desired, water may be cooled to the freezing point, and ice has been formed in units. Chilling water less than 40°F becomes expensive.

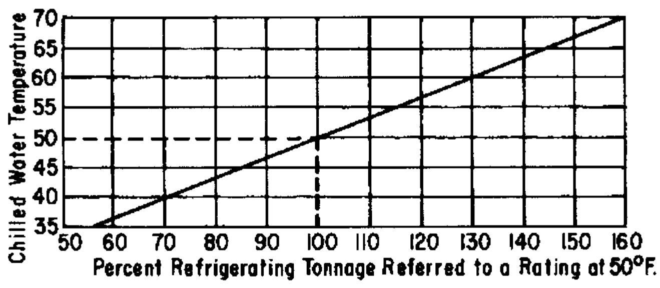

Figure 11-7. Variation in tonnage with water temperature for steam jet refrigeration systems. (Used by permission: Havermeyer, H. R. *Chem. Eng.*, Sept. 1948. ©McGraw-Hill, Inc., New York. All rights reserved.)

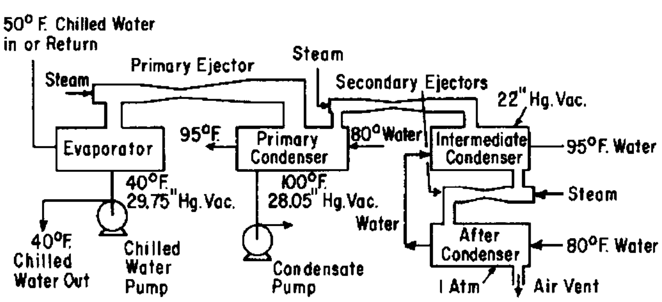

Figure 11-8. Operation conditions for a steam jet refrigeration system. (Used by permission: Rescorla, C. J. *Chem. Eng.*, June 1953. ©McGraw-Hill, Inc., New York. All rights reserved.)

Units are usually operated with 50–200 psig steam, although pressures down to 2 psig are possible, and 30 psig units are giving economical performance for their specific situation.

In transporting this water through insulated pipes to the process equipment, it is good practice to allow a 2°F temperature rise when planning heat transfer calculations.

Cooling water must be specified at maximum expected temperature, otherwise the unit cannot condense in hot weather and still maintain full load.

Utilities

The utilities required for steam jet refrigeration operation often determine the selection of these units, between manufacturers and between types of refrigeration. As the chilled water temperature off of the unit approaches 32°F, the cost of the basic unit and its steam and cooling water requirements rise rapidly.

Figures 11-9, 11-10, 11-11, and 11-12 present estimating operating utility requirements for barometric type units when referenced to a given chilled water temperature and 100 psig motivating steam. These curves allow the designer and operator to vary conditions to suit the relative costs of steam and cooling water (utility, not the chilled water) and still maintain the tonnage from the unit. The performance of specific units can be improved usually over the values of the curves. Higher pressures are some advantage to a maximum of 12%. When the steam pressure to the boosters is reduced from 100 psig to about 30–50 psig, the quantity required will increase by a factor of 2 for 40°F chilled water to only about 1.5 for 55°F chilled water.

Referring to the curves for 40°F water a unit initially designed for 90°F cooling water with a steam consumption of 30 lb of 100 psig steam per ton of refrigeration will use 10.9 gpm of the 90° water per ton of refrigeration. In the winter when cooling water temperatures drop to a maximum of 75°F, the steam consumption will drop to 15.5 lb/hr/ton when the water rate is maintained at 10.9 gpm/ton. If 45°F chilled water is needed in place of the 40°F chilled water, and if the steam is the most expensive utility, then 10.9 gpm/ton of cooling water (utility) will require only 22.7 lb/hr of steam.

Make up water is about 1% of the water circulated in a closed system.[13] For water being chilled from 50° to 40°F and used at 40°F:

1 lb water through 10°F $\Delta t \cong 10$ Btu

At 40°F, the latent heat is 1,071 Btu/lb (Figure 11-3).

Lb water evaporated/lb water recirculated = 10/1,071 = 0.0093

Approximate % make up $\cong$ 0.0093 (100)/1 $\cong$ 0.93 $\cong$ 1.0%

Typical performance of a 150-ton unit, using 100 psig steam is as follows for a barometric refrigeration unit:

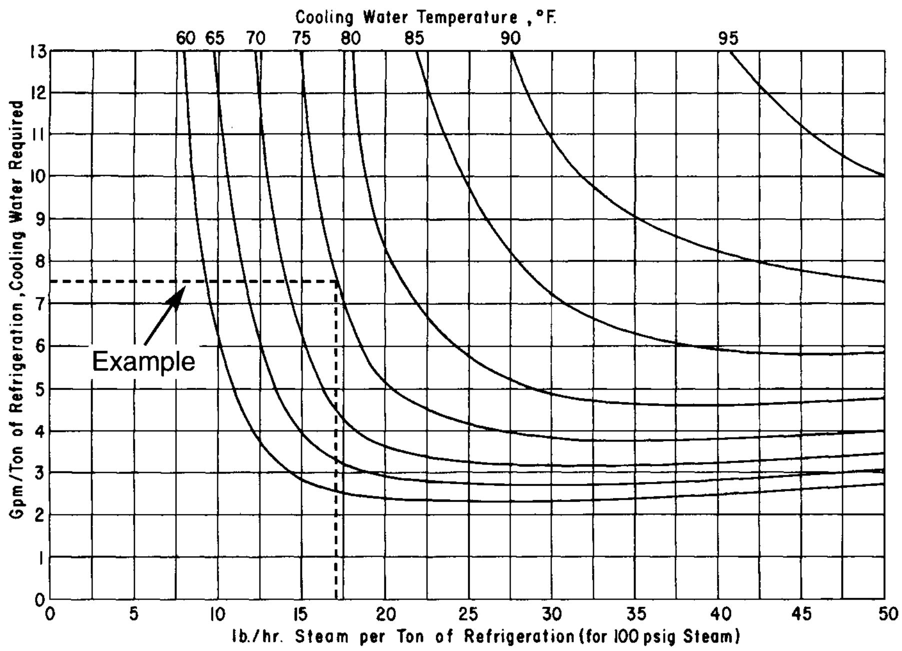

Figure 11-9. Steam jet utilities requirements when producing 40°F chilled water temperature. (Used by permission: ©Croll Reynolds Company, Inc.)

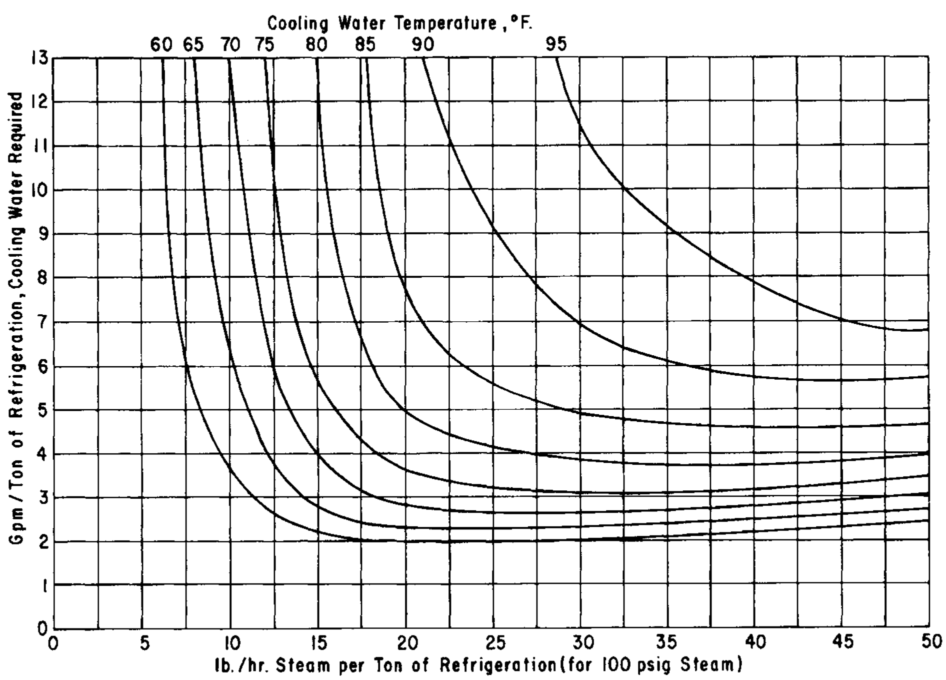

Figure 11-10. Steam jet utilities requirements when producing 45°F chilled water temperature. (Used by permission: ©Croll-Reynolds Company, Inc.)

Barometric Refrigeration Unit

Tons refrigeration	= 150	Gpm condensing water, air ejector	= 10
Chilled water temperature, °F	= 45	Total gpm	= 880
Return water temperature, °F	= 59.4	Minimum steam pressure for boosters, psig	= 100
Gpm chilled water	= 250	Minimum steam pressure for air ejector, psig	= 100
Maximum temperature condenser water in, °F	= 85	Steam consumption, booster, lb/hr	= 4,150
Condensed water out, °F	= 99.9	Steam consumption, air ejector, lb/hr	= 100
Gpm condensing water, booster condenser	= 870	Total steam, lb/hr	= 4,250

Note: System must be free of air leaks; steam must be dry; and condenser tubes clean.

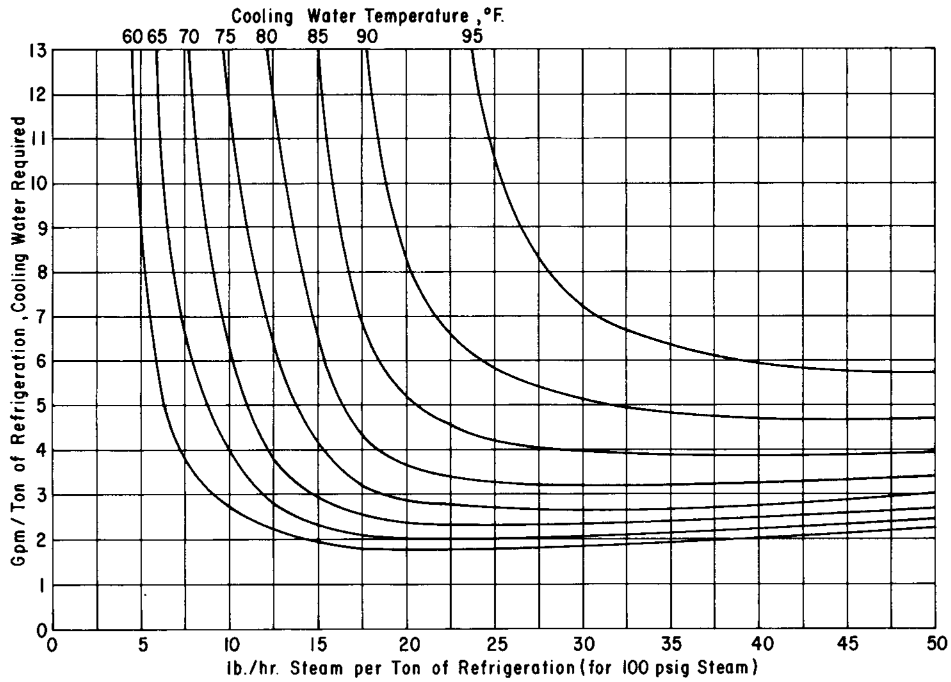

Figure 11-11. Steam jet utilities requirements when producing 50°F chilled water temperature. (Used by permission: ©Croll-Reynolds Company, Inc.)

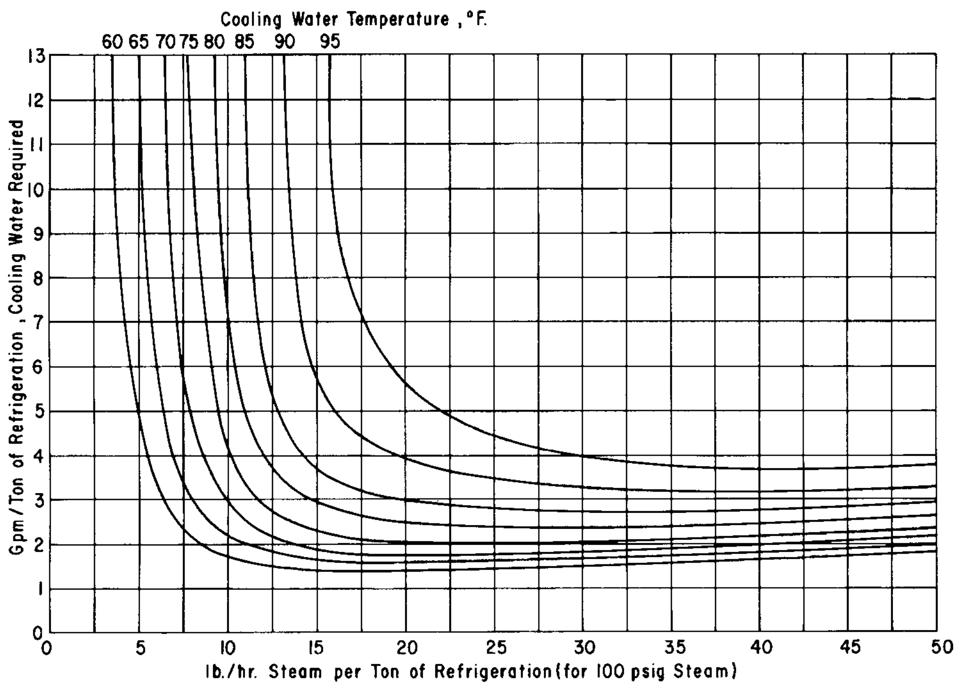

Figure 11-12. Steam jet utilities requirements when producing 60°F chilled water temperature. (Used by permission: ©Croll-Reynolds Company, Inc.)

Specification

Figure 11-13 is convenient for preparing an initial specification for competitive inquiry, as well as for summarizing the final accepted performance and specifications. In using the specification sheet, supplemental notes should be included to state:

1. Flexibility of operation desired as this determines the number of booster jets.
2. Type of condenser: barometric, surface, or low-level jet.
3. Temperature extremes of cooling water for condenser and duration of time for these temperatures.
4. Closed or open circuit chilled water.
5. Discharge heads on chilled water pumps and condensate pumps, if manufacturer is to furnish.

303-296 10-58

SPEC. DWG. NO.

A-

Job No. ____________________

____

B/M No. ____________________

Page of Pages
Unit Price
No. Units
Item No.

CHILLED WATER
REFRIGERATION SPECIFICATIONS

SERVICE CONDITIONS

Type: Surface Barometric: Capacity__________ Tons. Manufacturer ______________________ Model ________

Chilled Water:__________ Gpm. @ ________°F. Return Water ________ Gpm. @________°F

Make-Up Water: (Condensate) (River Water) (Sea Water) ______________Gpm @ ________°F

Booster Jet Steam: (a) ______ lbs/hr @ ______Psig and ______°F. (b) ______ lbs/hr @ ______ Psig and ____°F

Secondary Jet Steam: (a) ______ lbs/hr @ ______Psig and ______°F. (b) ______ lbs/hr @ ______ Psig and ____°F

Cooling Water: River Water / Sea Water__________ Psig @________°F Primary Condenser ________Gpm @______°F Exit Secondary Condenser ________ Gpm @______°F Exit

Water Fouling Factor ____________________

SPECIFICATIONS

Flash Tank Chilled Water:	Connection - ______ Inlet,	______ Outlet.	Flange -	______ Rating,	______ Face
Make-Up Water:	Connection - ______ Inlet,	______ Outlet	Flange -	______ Rating,	______ Face
Primary Condenser Steam:	Connection - ______ Inlet,	______ Outlet	Flange -	______ Rating,	______ Face
Secondary Condenser Steam:	Connection - ______ Inlet,	______ Outlet.	Flange -	______ Rating,	______ Face
Prim. Cond. Cooling Water:	Connection - ______ Inlet,	______ Outlet.	Flange -	______ Rating,	______ Face
Sec. Cond. Cooling Water:	Connection - ______ Inlet,	______ Outlet.	Flange -	______ Rating,	______ Face

Steam Hand Valves: Supplied by (Owner) (Vendor)

Level Controls:

Pressure Gauges: ½" Connections

Condenser Tube Water Velocity Limits: ______________ Ft./Sec. to ______________ Ft./Sec.

MATERIALS OF CONSTRUCTION

Flash Tank: ____________________ Corr. Allow ________ . Barometric Leg ____________________ Corr. Allow ______

Booster Ejector Nozzle ____________________ . Secondary Ejector Nozzles ____________________

Primary Surface Condenser - Tubes ____________________ , O.D.______ , BWG______ , Length______ Ft., No.______ ,

Pitch ______________ . Tube Sheet ______________ Channel ______________

Surface After Condenser - Tubes ____________________ , O.D. ______ , BWG ______ , Length ______ Ft., No. ______ ,

Pitch ______________ . Tube Sheet ______________ Channel ______________

Barometric Primary Condenser:

Shell ____________________ Corr. Allow. ______

Baffles ____________________ Corr. Allow. ______

Spray Noz. ____________________ No ______

Barometric After Condenser:

Shell ____________________ Corr. Allow. ______

Baffles ____________________ Corr. Allow. ______

Spray Noz. ____________________ No. ______

REMARKS

Vendor is to Specify:

1. Make and Model for Flow, Temperature and Pressure Control Valves, Hand Valves, Thermometers, Steam Traps and Strainers, Electrical and other miscellaneous equipment.

PURCHASE ORDER NUMBER

By	Chk'd.	App.	Rev.	Rev.	Rev.
Date					

P.O. To: ____________________

Figure 11-13. Chilled water refrigeration specifications.

6. Space limitation, if any.
7. Utility costs for steam and cooling water if manufacturer is to make economical compromise of requirements for these services.
8. Type and nature of cooling water for condenser.

Example 11-1. Barometric Steam Jet Refrigeration

A process has the following requirements for chilled water refrigeration:

1. One condenser at 55°F water, 110 gpm
2. One gas cooler at 55°F water, 65 gpm
3. One direct gas absorber-cooler at 40°F water, 223 gpm

Make up water for these refrigeration units is at 80°F, and feed water for the gas cooler unit is available at 90°F. Barometric water is 90°F. Items 1 and 2 are for a closed-circuit operation with return water at 68°F and total 175 gpm. Note that in order to consolidate the temperature levels of water, it is economical to establish a temperature, such as 55°F, which satisfies the bulk of the requirement, and then design the other phases of the plant process to also use this water temperature. Item 3 is an open-circuit operation because the water is sent to waste after absorbing certain corrosive vapors and cooling the bulk of the gas.

Closed Circuit

$$\text{Tons refrigeration} = \frac{(175)(68 - 55)}{24} = 94.8 \text{ tons}$$

Open Circuit

$$\text{Tons refrigeration} = \frac{(223)(90 - 40)}{24} = 465 \text{ tons}$$

Recommended Tonnage for Purchase

Closed circuit: 100 tons
Open circuit: 500 tons

Approximate Utility Requirements

100 psig steam is high in cost, and cooling water very cheap.

Closed Circuit. Approximate values between those in Figures 11-11 and 11-12 for 55°F chilled water temperature and 90°F cooling water.

For high-cost steam, reduce its requirements at the sacrifice of water: lb/hr steam/ton = (13.5 + 18.8)/2 = 16.1 approx. Steam required = (16.1)(94.8) = 1,530 lb/hr at a barometric cooling water rate of 10.1 gpm/ton; then this water required = (94.8)(10.1) = 958 gpm. Make-up process side water = (0.01)(175 gpm) = 1.75 gpm.

Open Circuit. Refer to Figure 11-9.

For 40°F chilled water, 100 psig high-cost steam, 90°F cooling water to the barometric:

Selecting cooling water at 11 gpm/ton:
Total = (11)(465) = 5,110 gpm
Then, the corresponding steam rate is 29.9 lb/hr/ton:
Total = (29.9)(465) = 13,900 lb/hr
Make-up water: 100% (once through)

Because this is a fairly large unit, consider two, 250-ton units. This configuration would be justified only if the tonnage load was not steady at the 465-ton rate, but fluctuated for reasonably long periods down to 250 tons. Then one unit could be shut down and perhaps some utilities. The 500-ton unit is a good economical size, and with multiboosters can operate at $^1/_2$ (two boosters) or $^1/_4$ (four boosters) load. The economics should be investigated.

Hot Well

The hot well is the sump where the barometric leg is sealed. It must be designed to give adequate cross-section below the seal leg and for upward and horizontal flow over a seal dam or weir. At sea level the hot well must be a minimum of 34.0 ft below the base of the barometric condenser. For safety to avoid air in-leakage, a value of 35–36 ft is used. For an altitude corresponding to a 26-in. Hg. barometer, the theoretical seal height is 29.5 ft; actual practice still uses about 34 ft.

Absorption Refrigeration

The two most common industrial absorption-type refrigeration systems are (1) aqua ammonia and (2) lithium bromide-water, with ammonia and water respectively the refrigerant for each system.

Ammonia System

In general the principle of operation depends upon the capability of water to absorb large quantities of ammonia vapor, which can be released from the solution by the direct application of heat.[18]

A typical pictorial flow diagram for the ammonia system is given in Figure 11-14 for a single-stage system and in Figure 11-15 for a two-stage ammonia system.

As far as refrigeration is concerned, this system produces ammonia liquid as the refrigerant that is evaporated in the process unit requiring the refrigeration. In principle this is the same operation at the evaporator as if the system were one of compression. The method of removal of the heat of the ammonia vapor and then the reliquefaction are the points of difference between the systems.

In the absorption system, as the ammonia is vaporized by the process heat load being cooled or condensed in the

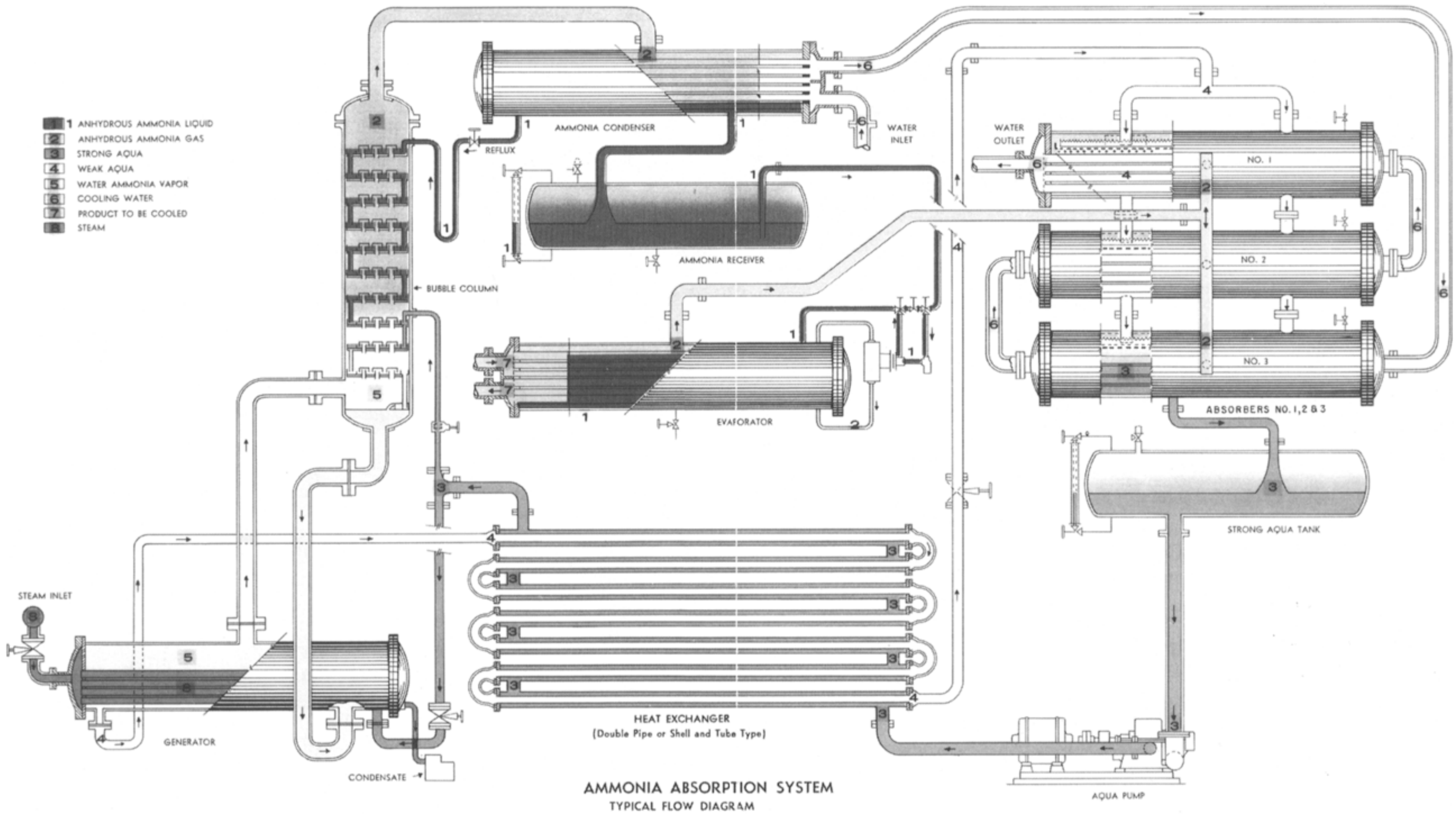

Figure 11-14. Ammonia absorption system typical flow diagram. (Used by permission: Frick®, Div. of York International.)

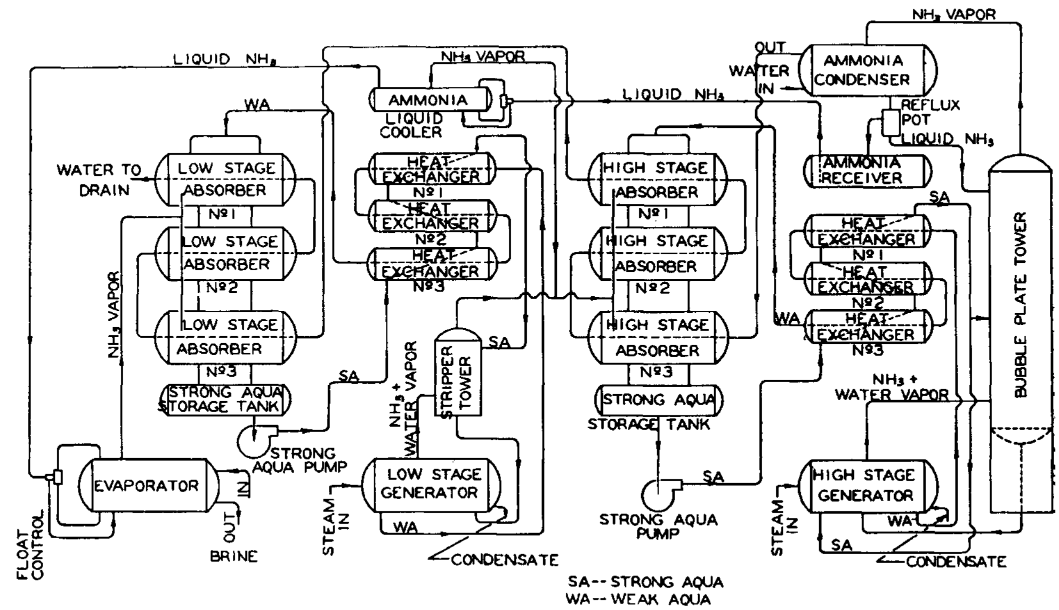

Figure 11-15. Schematic diagram of two-stage absorption plant. (Used by permission: Rescorla, C. L. *Refrigerating Engineering*, ©March 1953; now merged and used by permission: American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc. All rights reserved.)

evaporator, the ammonia vapor is drawn into the absorber by the affinity of ammonia for aqua-ammonia solutions. The heat of absorption and the latent heat of condensation is removed by cooling water flowing through the tubes. The aqua concentration increases until an equilibrium is reached with the temperature and pressure maintained in the absorber. Aqua tables give the conditions to be expected.[6] The pressure is usually 15–25 psig. The strong aqua is cross-exchanged with hot stripper column bottoms to preheat it for feed to the distillation column. Here, the water is removed by taking the ammonia overhead as refrigerant grade (99.99 wt %) and recirculating the weak aqua bottoms back to the absorbers.

The condensed pure ammonia is now ready to go to the evaporator under reduced pressure and perform its refrigeration function. The evaporator usually operates with the ammonia on the shell side of a tubular unit and with process vapor or liquid on the tube side being cooled or condensed, or the evaporator may use an indirect scheme by interposing secondary coolant/brines, such as inhibited ethylene and propylene glycols (pure or aqueous solutions); methylene chloride; dichlorodifluoromethane; and trichloroethylene, trichloromonofluoromethane, and dichlorodifluoromethane; and calcium chloride and sodium chloride brines. These secondary coolants are cooled to the desired temperature by the primary evaporating refrigerant, and the secondary coolant in turn is used to cool/condense the process application. Then it recirculates back to the evaporator for re-establishing the level of cold temperatures required for the process. For data on most of the listed coolants, see reference 2, and see Figure 11-16 for freezing points for aqueous solutions for ethylene and propylene glycols. This chilled coolant is then used to condense or cool the process material. The latter scheme is usually termed *indirect refrigeration.* It is less efficient than direct evaporation, due to the losses in heat transfer. The ammonia vapors pass back to the absorbers to contact the weak aqua.

General Advantages and Features

This type of system can be adapted to steam, direct-fired natural gas or oil, waste heat, or another heat source for the distillation column reboiler. The only moving parts are the strong aqua pumps. This generally affords low routing maintenance. The process cycle is quite adaptable to automatic controls and can be kept in balance at very reduced loads, particularly by applying a false load to the absorbers. Generally speaking, these absorption systems are economically competitive down to 200 tons of refrigeration. In special cases, they have been found advantageous to 50 tons.

Capacity

Range of plants available: 50 to 2,000–5,000 tons.
Average size: 500–1,000 tons.

Performance

The performance balance of this type of system is a function of the specific conditions established or required for

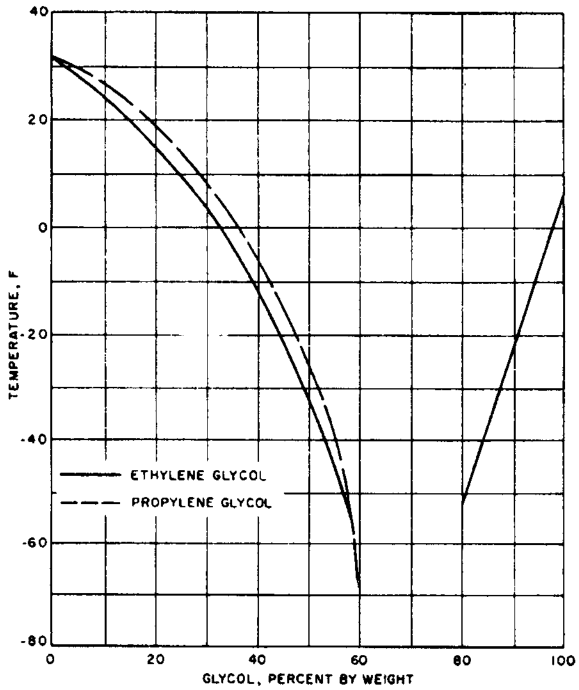

Figure 11-16. Freezing points of aqueous solutions of ethylene and propylene glycol. (Used by permission: *1977 ASHRAE Handbook, I-P Ed., Fundamentals*, 1979 *ASHRAE Handbook and Product Directory*, ©1979, 1980 2nd printing, American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc. All rights reserved.)

the various components, such as ammonia evaporating pressure, absorber pressure, cooling water temperature, and temperature level of the heat to the distillation column vapor generator. The calculations are reasonably straightforward when the fundamentals of the cycle are understood. Figure 11-14 is a single-stage system, and Figure 11-15 is for a two-stage system. The latter is adaptable to locations having low-temperature heat for the operation of the generator.

Table 11-1 presents a few of the many possible operating conditions for these systems. The same refrigerating plant may operate through a wide range of pressures and temperatures. Table 11-2 presents a summary of utilities required for absorption plants. The steam indicated can be substituted by direct gas, waste heat, or other stream at an equivalent temperature level.

Table 11-1
Representative Single-Stage Absorption Plant Operating Conditions

Tons Refrigeration	(1) 500	(2)* 650	(3)* 800
Temperature, °F			
Brine to evaporator (or cooler)	...	a	−16
Brine from evaporator	...	a	−20
Evaporator	−14.8	29	−30.8
Weak aqua to absorbers	112	126	102
Strong aqua from absorbers	89.6	104	90
Water to absorbers	87.8	86.7	70
Water from absorbers	99	94.1	79
Strong aqua from heat exchangers	224.5	178	248
Weak aqua to heat exchangers	271	210	283
Liquid ammonia from condenser	85.1	93	92
Water to condenser	77	81	79
Water from condenser	87.8	86.7	87.3
Steam in generator	272[b]	222	298
Pressure, Psig			
Evaporator	6	43.8	2.3 in. Hg.
Absorbers	6	43.8	2.3 in. Hg.
Condenser	175	174.9	171.9
Steam	b	3.2	50
Surface, Ft²			
Evaporator	...	**	12,800
Absorbers	6,660	5,580	6,390
Heat exchangers	2,550	1,675	8,400
Generator	b	3,100	5,410
Condenser	5,540	5,661	5,270
Cooling water, gpm	...	3,600	4,000
Aqua pump, hp	...	40	50
Steam rate, lb/hr/ton	...	24.6	36.1

*Data for columns (2) and (3) from Reference (13).

**Not available.

[a]Pump recirculation system on air-cooling coils.

[b]Gas fired generator with aqua ammonia on shell side.

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Example 11-2. Heat Load Determination for Single-Stage Absorption Equipment

Design Basis

Ammonia refrigeration tonnage required for process = 400
Evaporator (process unit using ammonia refrigerant) temperature = +5°F
Evaporator pressure = 19.6 psig
Water temperature to absorbers = 88°F
Distillation column condenser pressure = 214.2 psig

Ammonia Flow for 400 Tons Refrigeration

From thermodynamic properties of ammonia greater than a datum of −40°F:
Enthalpy of vapor at +5°F = 613.3 Btu/lb
Pressure = 19.6 psig or 34.27 psia
Enthalpy of liquid from condenser at 214.2 psig = 161.1 Btu/lb

$$\text{Lb of ammonia/min} = \frac{(400)(200\ \text{Btu/min/ton})}{(613.3 - 161.1)} = 177$$

Table 11-2*
Data for Determining Utilities Required for Absorption Plants

Single-Stage

Evap. Temp., °F	Steam Sat. Temp., °F Req. in Generators	Btu/min Req. in Generator per Ton Refrig. (200 Btu/Min)	Steam Rate lb/hr/ton Refrig.	Water Rate through Cond. (7.5°F Temp. Rise), gpm/ton
50	210	325	20.1	3.9
40	225	353	22.0	4.0
30	240	377	23.7	4.1
20	255	405	25.7	4.3
10	270	435	28.0	4.6
0	285	467	30.6	4.9
−10	300	507	33.6	5.4
−20	315	555	37.3	5.9
−30	330	621	42.5	6.6
−40	350	701	48.5	7.7
−50	370	820	57.8	9.5

Two-Stage

Steam Sat. Temp., °F, Req. in Generators	Btu/min Req. in Generator per Ton Refrig.	Steam Rate, lb/hr/ton Refrig.	Water Rate through Cond. (7.5°F Temp. Rise), gpm/ton
175	595	35.9	4.3
180	625	37.8	4.5
190	655	40.0	4.6
195	690	42.3	4.9
205	725	44.7	5.3
210	770	47.5	5.7
220	815	50.6	6.3
230	865	54.0	6.9
240	920	58.0	7.8
250	980	62.3	9.0
265	1050	67.5	11.0

Note: Water to condenser at 85°F, with 100°F condensing temperature. Water from condenser used in absorbers.

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Absorber Design

The range of temperature difference in the absorber between strong aqua from the absorber and cooling water entering is 9–15°F. Select 13°F for this case. Then strong aqua temperature from absorber is 90°F + 13°F = 103°F. Assume an absorber pressure of 18.0 psig. This is 1.6 psig lower than the evaporator pressure of 19.6 psig.

Absorber Aqua Conditions (Using Chart, Figure 11-17)

Liquid	Temp., °F	Psig	Wt% NH_3	Liquid Enthalpy, Btu/lb
Strong	103	18	35.8	0
Weak	267	214.2**	25.8*	190

*Assumed difference in aqua concentration of 10%; this may range from 6–15%, depending upon conditions, and is often varied to suit a particular capacity situation.

**Neglects effect of distillation column pressure drop.

Using the chart, Figure 11-17, the 103°F and 18 psig determine the 35.8% ammonia and 0 Btu/lb. Then the 25.8% as determined previously, together with the column pressure of 214.2 psig, determine the 267°F and 190 Btu/lb.

When the weak aqua enters the absorber, it flashes or is expanded through a control valve from about a column pressure of 214.2 psig to the absorber pressure of 18 psig. At equilibrium for this 25.8% ammonia solution and at 18 psig, the temperature is 138°F (Figure 11-17), and the liquid enthalpy is 49 Btu/lb.

The weak aqua leaving the heat exchanger ahead of the absorbers is assumed to be cooled to about 6–11°F (sometimes more) less than this equilibrium temperature of 138°F. Then the weak liquid entering the absorbers is taken as 138°F − 10°F = 128°F. This may have to be corrected later if proper balance is not reached. Figure 11-17A is also a useful reference.

Strong Aqua Leaving Absorber

lb/min = (1 − a/(b − a) (lb NH_3/min)
where: a = weight fraction concentration of weak aqua
b = weight fraction concentration of strong aqua

$$\text{lb/min} = \frac{1 - 0.258}{0.358 - 0.258}(177) = 1{,}312$$

gpm = 1,312/7.18 lb/gal = 183

Weak Aqua Entering Absorber

lb/min = 1,312 − 177 = 1,135
gpm = 1,135/7.42 lb/gal (approx.) = 153

Heat Exchanger Before Absorbers

Weak aqua temperature range: 267°F → 128°F
Strong aqua temperature range: 103°F → ?
The temperature of the strong aqua must be evaluated.

1. Heat in exchanger (weak aqua) = (1,135)(189)
   = 215,000 Btu/min
2. Heat in (strong aqua) = (1,312)(0) = 0
3. Heat out (weak aqua) = (1,135)(49) = 55,700 Btu/min
4. Enthalpy of strong aqua liquid out of exchanger:

$$= \frac{(215{,}000 - 55{,}700) + 0}{1{,}312} = 121.3 \text{ Btu/lb}$$

Strong aqua temperature from Figure 11-17 at 35.8% ammonia and 121.3 Btu/lb = 211°F. (temp. out of exchanger). The equilibrium value at 35.8% and column pressure of 214.2 psig is 226°F, and the ammonia vapor concentration is 93.3 wt%.

For exchangers, the strong aqua temperature range is 103°F → 211°F.

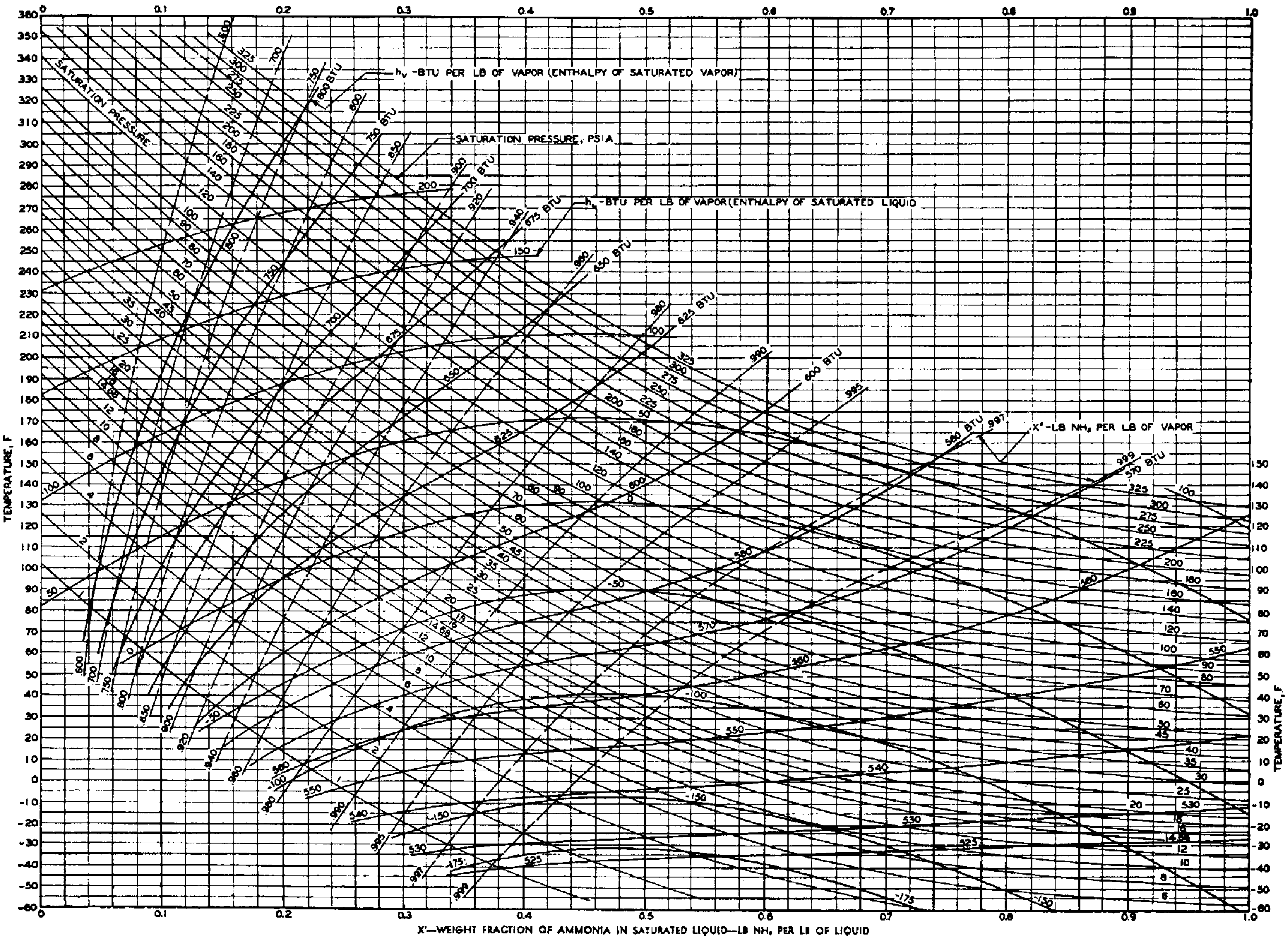


Figure 11-17. Weight fraction of ammonia in saturated liquid versus temperature. (Used by permission: Kohloss, F. H., Jr. and Scott, G. L., *Refrigerating Engineering,* V. 58, No. 10, ©1950; now merged with and used by permission: American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc. All rights reserved.)

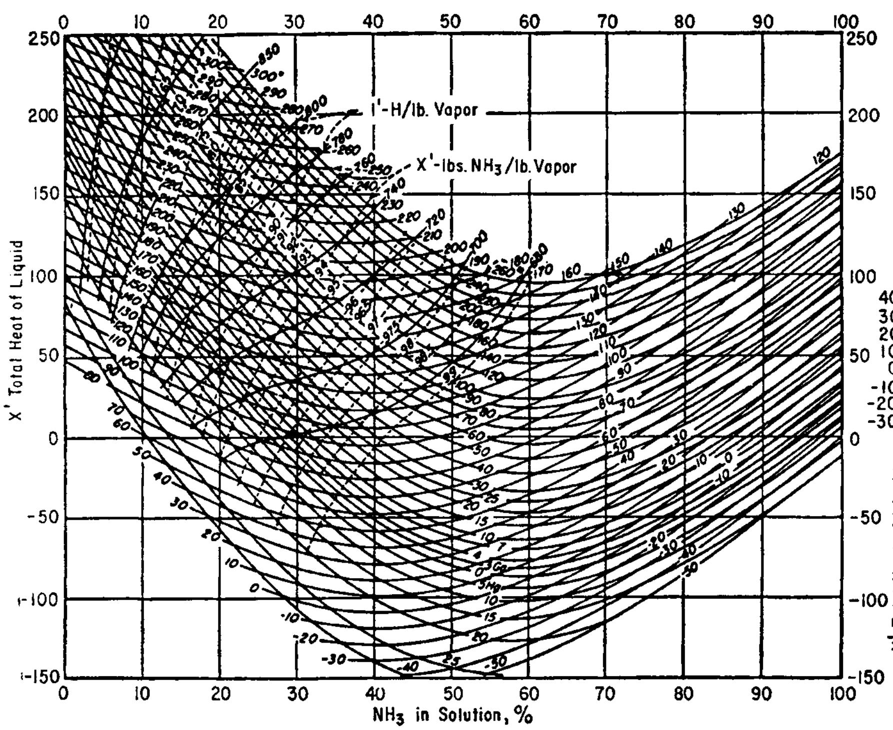


Figure 11-17A. Thermal properties of ammonia-water solutions. (Used by permission: Rescorla, C. L. and Miller, D. K. ©*Plant Engineering Handbook*, 2nd Ed., Figure 14-7. McGraw-Hill, Inc. All rights reserved.)

Note: At constant pressure, the quantity of ammonia in a solution decreases as the temperature increases.

An overall heat transfer coefficient, U, for the usual operating range is 200–230 Btu/hr (ft^2)°F, with liquid velocities in shell between 2–4 ft/sec.

Heat load for exchanger = 215,000 − 55,700 = 159,300 Btu/min.

(Note that the heat content of the strong aqua in, item (2), is used for enthalpy calculations only. It is accidental that the strong aqua enthalpy is 0 in this case).

Absorber Heat Load

Inlet: ammonia vapor from evaporator	108,700 BTU/min.
= (177)(613.3) =	55,700
weak aqua = (49)(1,135) =	164,400
Out: heat of strong aqua = (0)(1,312) =	−0
Heat removed in absorber =	164,400 Btu/min
Cooling water: 88°F ⟶	100°F (assumed)

Absorber inlet: (138 + 128)/2 = 132°F (avg.)
Absorber outlet: 103°F
Overall transfer coefficient for absorbers = 100 Btu/hr (ft^2)°F

Distillation Column

The column is designed as an ammonia rectifier-stripper using fundamental design techniques. A 48-in. diameter column will handle at least 500 tons of refrigeration system load for the above temperature range, using 10 bubble cap trays with 32, 4-in. pressed steel caps per tray (slot area = 7.81 in.^{2}/cap; riser area 4.83 in.^{2}/cap: 3 ft 0 in. weir length). Tray No. 4 from the bottom is usually a satisfactory feed tray. An L/D, reflux-to-ammonia product withdrawn to between 0.5–1.0 is usually satisfactory.

Overhead Condenser

Design using usual methods, handling total overhead vapors from column, as anhydrous ammonia.

Vapor Generator

This unit may use steam, natural gas (fired), or waste gas as the heat source to reboil the bottoms aqua.

Heat duty = Heat out overhead vapors of column
+ heat out in weak aqua (215,000 Btu/min for this example)
− heat of strong aqua into column [(121.3 Btu/lb) (1,312) = 159,000 Btu/min for this example], Btu/min.

The design of the heater depends upon the type required.

Lithium Bromide Absorption for Chilled Water

In this system, the capability of lithium bromide to absorb water vapor is used to evaporate and cool water in the system. Process water for chilling is circulated from the process application, where it is warmed from a normal low of 40°F

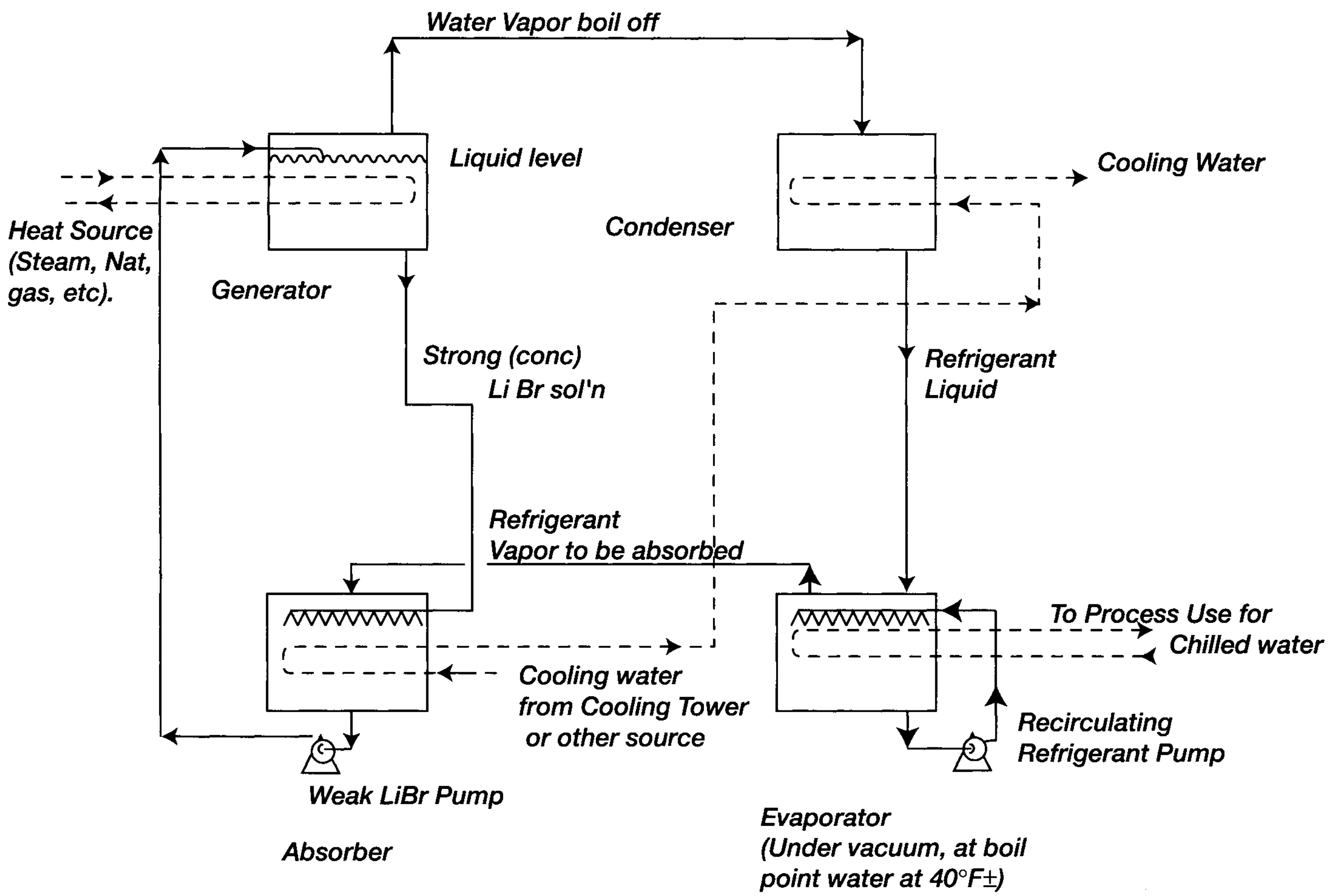


Figure 11-18. Lithium bromide absorption refrigeration system concept; water is the refrigerant. Actual commercial and industrial process flows reflect various heat recovery arrangements.

with a maximum temperature rise of 10–60°F. Then the process water is circulated back to the refrigeration system evaporator coils for cooling back down to the system working temperature of 40–45°F. See Figure 11-18 for a conceptual flow arrangement.

Nominal tonnage capacities range from 60–1,700 tons of refrigeration producing chilled water at 45°F. One unique feature of this system is the use of low pressure steam for the operation of the generator.

This type of system is similar in basic principle to the ammonia absorption, and some of the advantages are the same:

1. Few moving parts are used.
2. No damage occurs from capacity overload or freezing.
3. Salt solution is nontoxic and does not need replacement unless spilled but does require a corrosion inhibitor.
4. Refrigerant water is easily handled.
5. Maintenance can be kept low.
6. Compact arrangements can be designed.
7. Very little operator attention is needed.

Figure 11-19 indicates the basic operating system involved, and Figure 11-20 shows a sectional view of a typical compact unit.

A description of the absorption cycle of Figure 11-19 is used by permission from Carrier Corporation, Bul. 521-606:

"The 16DF direct-fired, double effect, absorption chiller/heater consists of an evaporator, absorber, condenser, high- and low-stage generators, separator, solution heat exchangers, refrigerant/solution pumps, burner and gas train assembly, purge, controls and auxiliaries. Water is used as the refrigerant in vessels maintained under low absolute pressure (vacuum). In the cooling mode, the chiller operates on the principle that under vacuum, water boils at a low temperature. In this case water boils at approximately 40 F (4.4 C), thereby cooling the chilled water circulating through

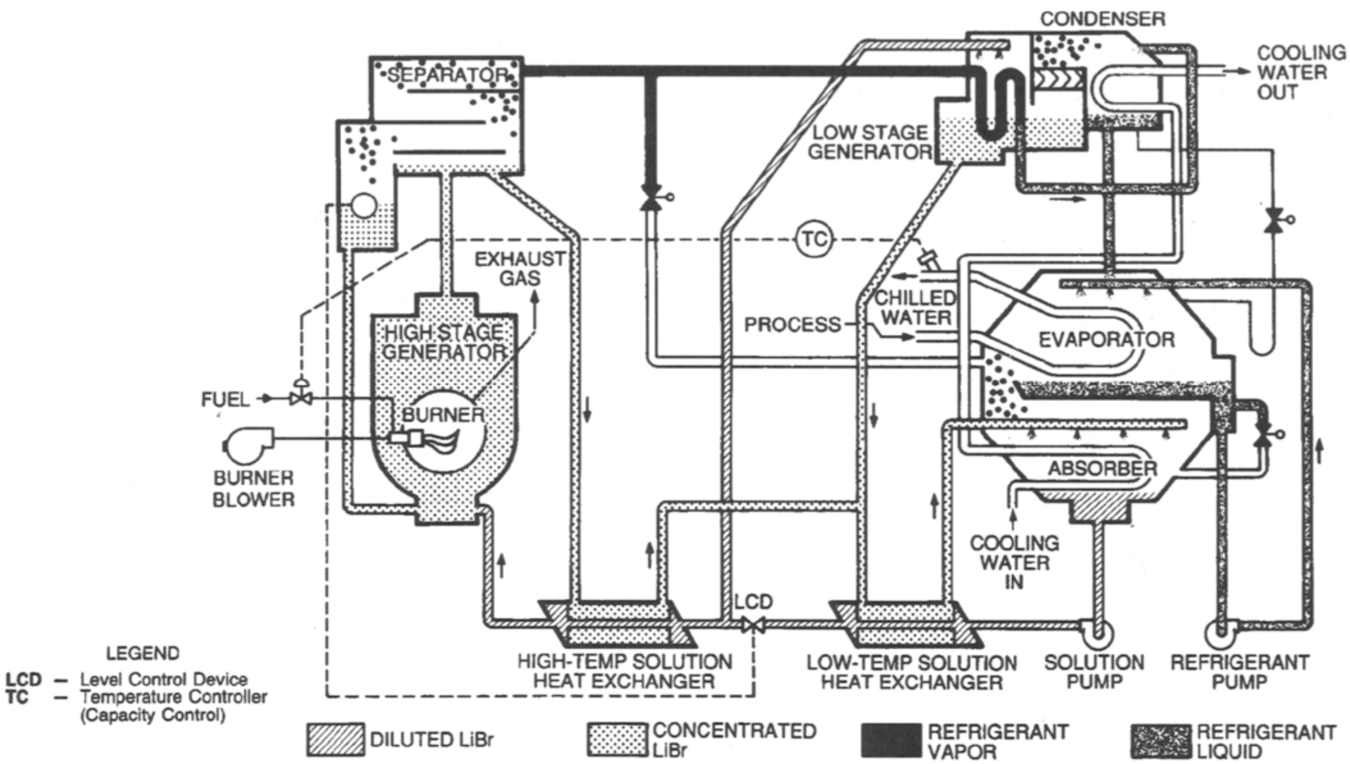


Figure 11-19. Lithium bromide hermetic absorption refrigeration system, double effect, liquid chiller/heater. As shown in chilling mode, water is the refrigerant under low absolute pressure (boiling at 40°F) (Used by permission: Cat. 521-606, form 16DF-1PD, ©1994. Carrier Corporation, a United Technologies Company.)

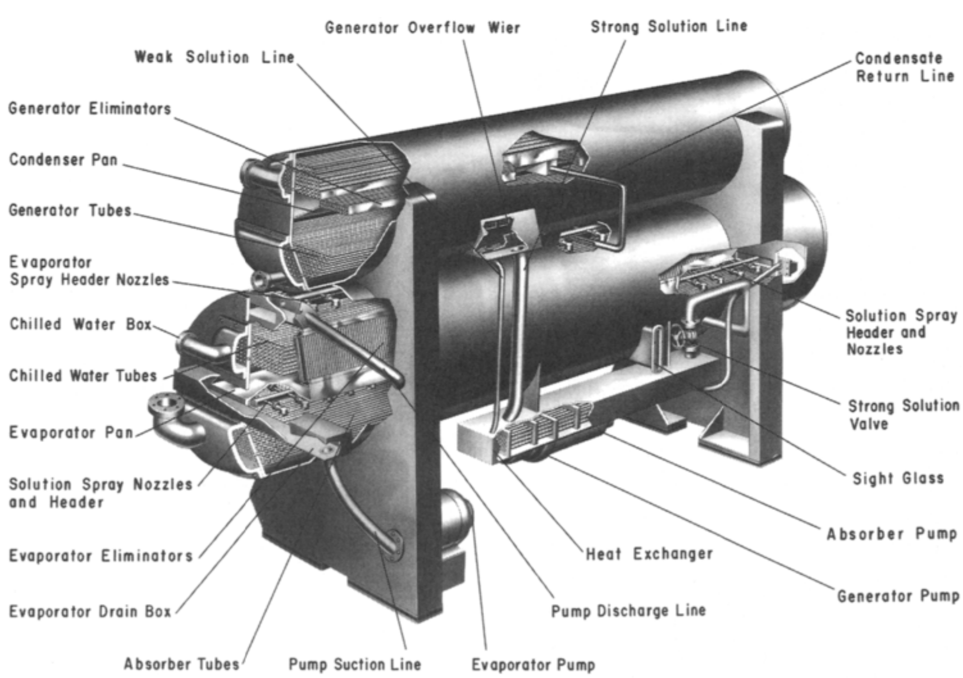


Figure 11-20. Sectional view of lithium bromide absorption refrigeration industrial unit. (Used by permission: ©1956. Carrier Corporation, a United Technologies Company.)

the evaporator tubes. A refrigerant pump is used to circulate the refrigerant water over the evaporator tubes to improve heat transfer.

To make the cooling process continuous, the refrigerant vapor must be removed as it is produced. To accomplish this, a lithium bromide solution (which has a high affinity for water) is used to absorb the water vapor. As this process continues, the lithium bromide becomes diluted, reducing its absorption capacity. A solution pump then transfers this weak (diluted) solution to the generators where it is reconcentrated in 2 stages to boil off the previously absorbed water. A solution flow control valve automatically maintains optimum solution flow to the generators at all operating conditions for maximum efficiency. Approximately half of the diluted solution is pumped to the high-stage generator where it is heated and reconcentrated by the heat from the combustion of natural gas or No. 2 oil. The other half of the weak solution flows to the low-stage generator where it is heated and reconcentrated by the high temperature water vapor released from the solution in the high-stage generator. Since the low-stage generator acts as the condenser for the high-stage generator, the heat energy first applied in the high-stage generator is used again in the low-stage generator thus reducing the heat input by approximately 45% as compared to an absorption chiller with a single stage of reconcentration. The water vapor released in the shellside of the low-stage generator, in addition to the now condensed water vapor from the tubeside of the low-stage generator, enters the condenser to be cooled and returned to a liquid state. The refrigerant water then returns to the evaporator to begin a new cycle.

To remove heat from the machine, relatively cool water from a cooling tower or other source is first circulated through the tubes of the absorber to remove the heat of vaporization. The water is then circulated through the heat tubes of the condenser. The strong (reconcentrated) solution from the high- and low-stage generator flows back to the absorber to begin a new cycle. For efficiency reasons, the strong solution from the high-stage generator is passed through the high-temperature solution heat exchanger to pre-heat the weak solution, while pre-cooling the strong solution. This strong solution is now combined with the strong solution from the low-stage generator and is passed through the low-temperature solution heat exchanger to preheat/precool the solution before being returned to the absorber.

The 16DF direct-fired, double effect, absorption chiller/heater can also be operated in a non-simultaneous heating (only) mode to provide 140 F (60 C) hot water for-space heating or other purposes without any additional components. In this mode, the cycle follows a different vapor flow path than that undertaken for cooling and does not use the absorption process."

$$\text{Chilled water gpm} = \frac{(\text{tons refrigeration})(24)}{\text{chilled water range, } \Delta t, \text{ °F}}$$

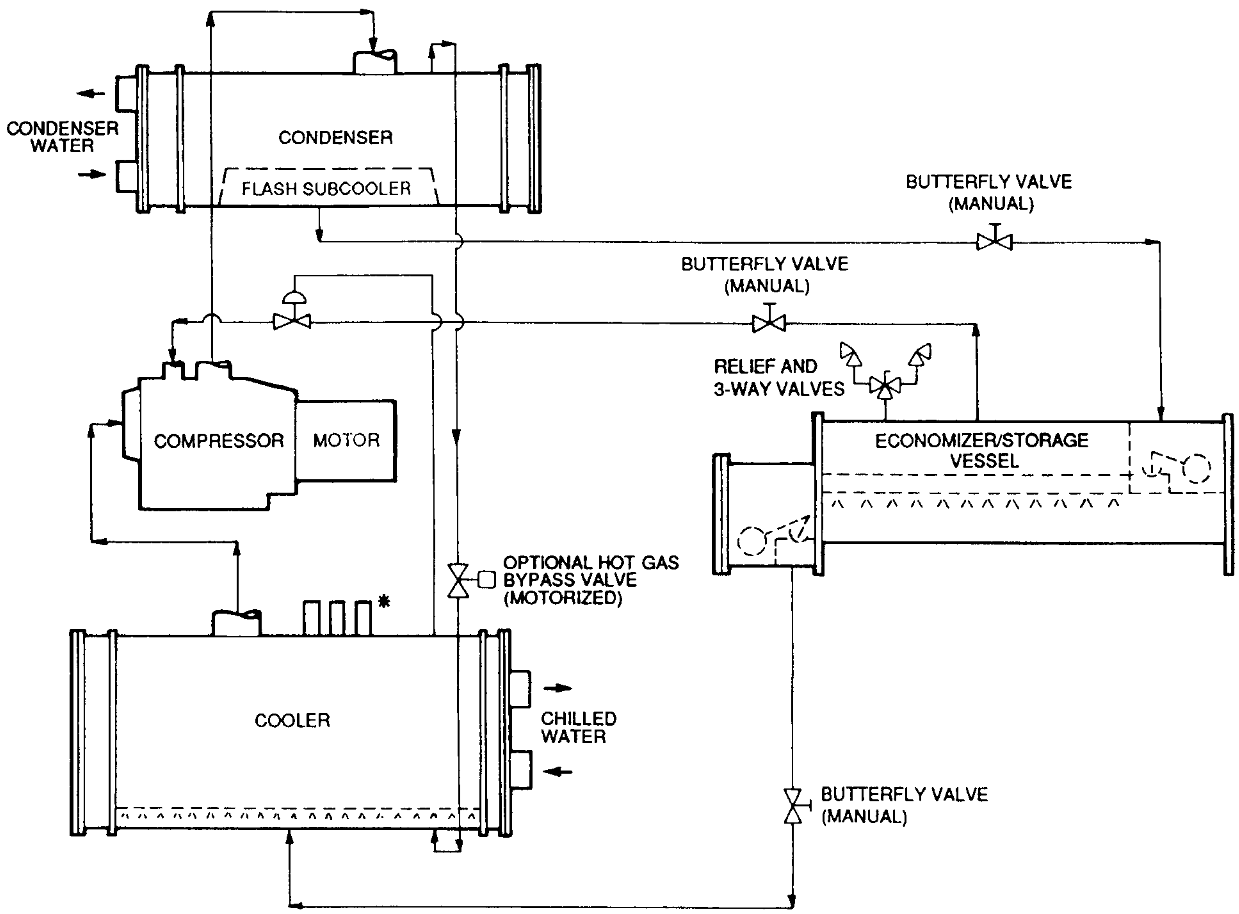
Manufacturers should be consulted for specific performance data for a given situation.

Mechanical Refrigeration

Mechanical systems may use reciprocating, screw, twin screw, or centrifugal compressors to move the refrigerant from the low- to high-pressure operating conditions. Some units up to 1,100–2,500 tons may be a compact "unitized" assembly of the compressor, condenser, piping, and controls. See Figures 11-21A, 11-21B, 11-21C, and 11-21D. The hermetically sealed centrifugal or reciprocating compressor has the compressing system and drive seal in a single case or housing. This eliminates the shaft seal and some lubricating problems. For special and large installations, the same basic equipment is involved, but the size and/or conditions require that the equipment be arranged separately.

Figure 11-22 is a schematic flow diagram of the basic mechanical refrigeration cycle. This simple cycle is in use, as are many modifications designed to improve heat or refrigeration efficiency. The process evaporator may be "direct"—that is, the refrigerant evaporates directly against the process fluid in the tube side (usually)—or it may be "indirect"—a brine coolant solution, usually sodium or calcium chloride, inhibited ethylene, propylene glycol, or methylene chloride is cooled by the refrigerant evaporation, and it in turn is used as a cold fluid for heat removed from other process equipment. Avoid brine whenever possible because it imposes an inefficiency in refrigerant use as far as heat transfer is concerned. Also, the brine is somewhat corrosive and adds to the maintenance of the system, unless inhibited glycol solutions or methanol solutions are used.

The refrigeration or cooling is the result of evaporating the refrigerant. For "direct" refrigeration, the liquid refrigerant is vaporized under reduced pressure through an expansion valve (thus producing cool vapor) against the process that is usually in the tubes while the refrigerant boils on the outside. (See Chapter 10, "Heat Transfer," of this volume). This vapor then passes to the suction side of the compressor where the pressure is raised to a temperature suitable for condensing the vapor against cooling water (or a secondary liquid, even from another refrigeration chiller system). Liquid refrigerant is produced in the shell, and this passes to a receiver under essentially compressor discharge pressure for the system. From here the liquid passes through the expansion valve noted previously to the evaporator unit, and the vapor returns to the compressor to complete the cycle; see Figure 11-22.



*The number of relief valves will vary depending upon the machine.

Figure 11-21A. Typical centrifugal compressor with economizer packaged refrigeration system. Note that screw-type or even reciprocating compressors can be used in such a system. (Used by permission: Cat. 521-727, ©1995. Carrier Corporation, a United Technologies Company.)

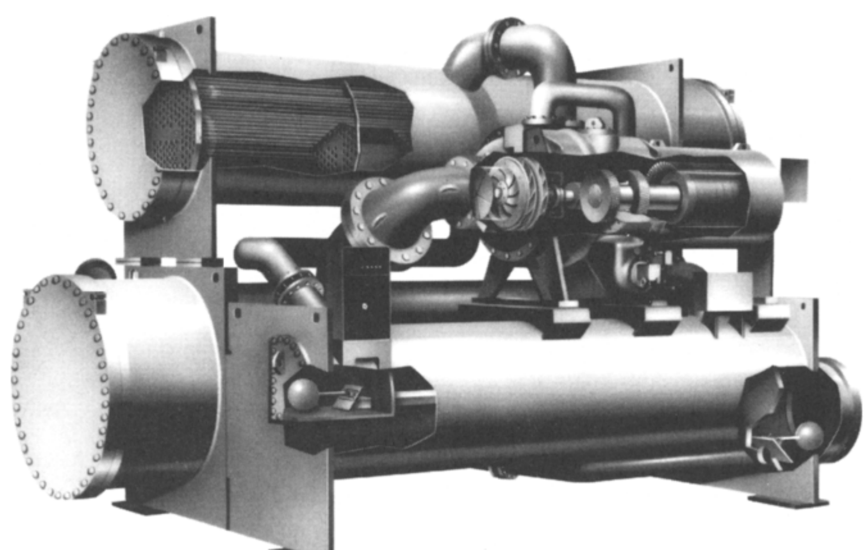
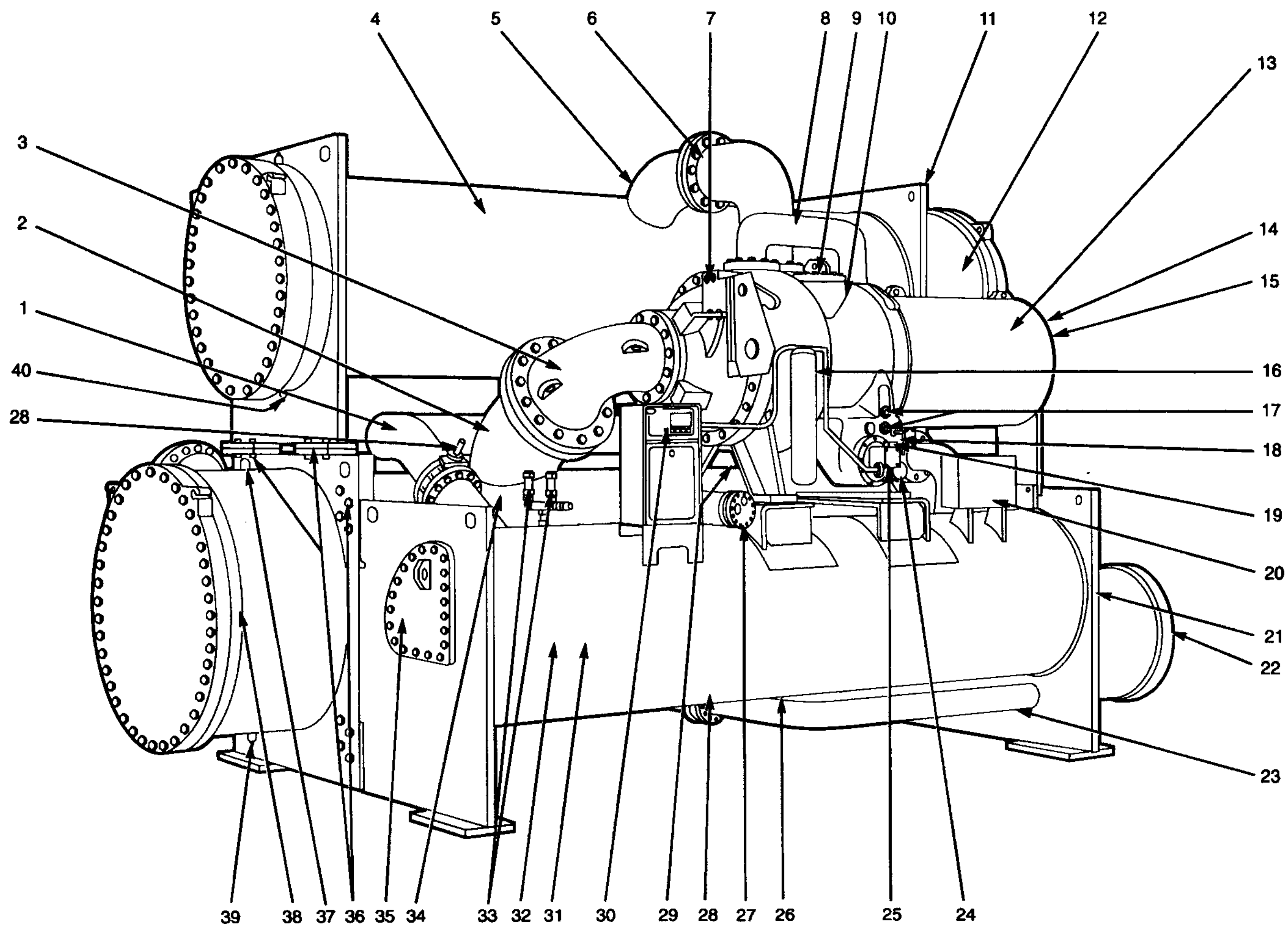


Figure 11-21B. Packaged hermetic open drive two-stage centrifugal liquid chiller with flash economizer. (Used with permission: Cat. 521-727, ©1995. Carrier Corporation, a United Technologies Company.)



19EX

LEGEND

1 — Refrigerant Liquid Line to Economizer/Storage Vessel
2 — Cooler Suction Pipe
3 — Compressor Suction Elbow
4 — Condenser
5 — Condenser Discharge Pipe
6 — Compressor Discharge Elbow
7 — Guide Vane Actuator
8 — Economizer Gas Line to Compressor
9 — Gear Inspection Cover
10 — 2-Stage Hermetic Compressor
11 — Condenser Waterbox Vent (Not Shown)
12 — Condenser Marine Waterbox
13 — Hermetic Compressor Motor
14 — Compressor Motor Terminal Box (Not Shown)
15 — Motor Sight Glass (Not Shown)
16 — Oil Filter
17 — Oil Level Sight Glasses (2)
18 — Cooler Relief Valves (Not Shown)
19 — Oil Heater (Not Shown)
20 — Auxiliary Power Panel (Field Wiring Terminals)
21 — Pumpdown Unit (Not Shown)
22 — Low-Side Float Box Cover
23 — Refrigerant Liquid Line to Cooler
24 — Oil Drain and Charging Valve
25 — Oil Pump
26 — Refrigerant Charging/Service Valve 10 (Not Shown)
27 — Oil Cooler
28 — Isolation Valves (Not Shown)
29 — Refrigerant Filter Drier
30 — Local Interface Display Control Panel
31 — Economizer/Storage Vessel
32 — Rigging Guide (Not Shown)
33 — Economizer/Storage Vessel Relief Valves
34 — Cooler
35 — High-Side Float Box Cover
36 — Take-Apart Connections
37 — Cooler Waterbox Vent
38 — Cooler Marine Waterbox
39 — Cooler Waterbox Drain
40 — Condenser Waterbox Drain

Figure 11-21C. Parts legend for centrifugal chiller refrigeration unit. (Used by permission: Cat. 521-727, ©1995. Carrier Corporation, a United Technologies Company.)

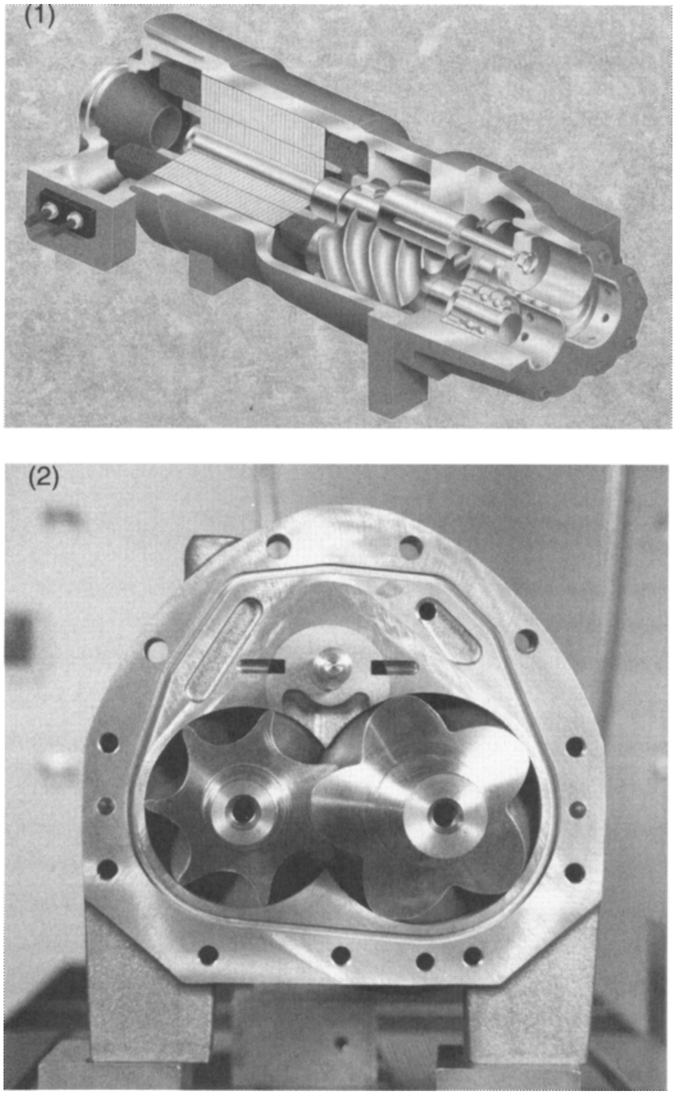
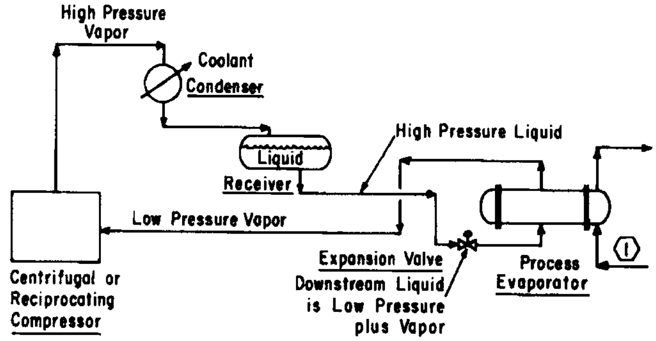


Figure 11-21D. Helical rotors refrigerant compressors. (1) Cutaway of a 100-ton intermediate compressor. The "intermediate" Helirotor® compressor has only three moving parts: the two rotor assemblies and the capacity controlling slide valve. The "general purpose" Helirotor® compressor has only four moving parts: two rotor assemblies, the variable unloader valve, and the step unloader valve. Unlike reciprocating compressors, the Trane Helirotor® compressor has no pistons, connecting rods, suction and discharge valves, or mechanical oil pump.

(2) End view showing male and female rotors and slide valve on an 85-ton intermediate compressor. The robust design of the Series R compressor can ingest amounts of liquid refrigerant that would severely damage reciprocating compressor valves, piston rods, and cylinders. (Used by permission: Cat. RLC-DS-2, ©Aug. 1995. The Trane Company.)

Compressors

The reciprocating, centrifugal and rotary-screw compressors described in the chapter covering this equipment design and selection are also used in refrigerant service. Several standardized designs fit each refrigerant, and this affords some economy in purchasing units developed for the application. Drivers for centrifugal compressors are usually steam turbine or electric motor through gears, and for reciprocating compressors, the electric motor through gears or belts is frequently used; see Reference 31.



Stream (I) may be Process Fluid being Cooled or Condensed, Or it may be Brine being Cooled for Use in Process Coolers, Condensers and Miscellaneous Equipment.

Figure 11-22. Basic features of mechanical refrigeration cycle.

Condensers

Condensers are usually designed to use water as the coolant for condensing the refrigerant. In some cascade or very low-temperature systems, one refrigerant may be used to condense a second refrigerant. The units are designed with good heat transfer principles. To improve heat transfer area characteristics of the tubes, finned tubes are helpful for the chloro-fluoro-refrigerants.

Process Evaporator

This unit handles the evaporating refrigerant on the shell side (for the usual case) and is designed in accordance with the principles for this type of heat transfer. The shell side of a wet unit must have vapor disengaging space above the tubes to allow for free surface boiling. To keep tubes clean, refrigerant level is maintained an inch or two above the topmost layer of tubes. If some superheating action of the top layer or two of tubes is desired to knock-down liquid droplets, then the liquid level is kept a few inches below the top tubes. Because oil and/or water may accumulate in the evaporator, it must be removed by purging or draining at intervals; otherwise, this may foul the shell-side tubes as well as affect the boiling characteristics.

In a dry evaporator the inlet expansion of refrigerant is controlled at a rate to have essentially no liquid in the unit.

Most industrial units are of the wet type.

Purge

The purge device or system removes noncondensables from the system at a minimum loss of refrigerant.

Process Performance

Refrigerants

Many materials are suitable for refrigerant purposes, and each usually has some special characteristics that allow it to serve a particular application better than some of the others. Before selecting a refrigerant, it is important to evaluate its flammability and toxicity data, pressure-temperature-volume relationships, enthalpy, density, molecular weight, boiling and freezing points, and various effects on gaskets, metals, oils, etc.[16]

ANSI/ASHRAE Standard 34-1992, "Number Designation and Safety Classification of Refrigerants"

The two purposes of ASHRAE Standard 34-1992 are

1. "... to establish a simple means of referring to common refrigerants. It also establishes a uniform system for assigning reference numbers and safety classifications to refrigerants." [from Section 1][20, 27]
2. "This standard provides an unambiguous system for numbering refrigerants and assigning composition-designating prefixes for refrigerants. Safety classifications based on toxicity and flammability data are included." [from Section 2][20, 27]

The ASHRAE Standard cited here (with addenda through 1994) provides refrigerant Safety Group Classifications related to toxicity and flammability. Tables 11-3A, 11-3B, and 11-3C are the official ASHRAE Refrigerant Data and Safety Classifications of all refrigerants used. Note that all the latest replacement/new refrigerants are not included in the 1992 addenda through 1994, because addenda have been issued through 1995 as of this writing. Most of the newest and replacement refrigerants are presented in manufacturer's data to follow. ANSI/ASHRAE Standard 15-1994, "Safety Code for Mechanical Refrigeration," responds to the rapid development of new refrigerants and refrigerant mixtures for use in new and existing equipment.[20]

Toxicity is referenced to the "Threshold Limit Value®—Time Weighted Average" established for each refrigerant. This is defined in ASHRAE Standard 15-1994[20] as (refer to the manufacturer's product data for more complete detail):

"the refrigerant concentration in air for a normal 8-hour work day and a 40-hour work week, to which repeated exposure, day-after-day, will cause an adverse effect in most persons" from Section 3.[20,27]

Two classes for refrigerants are "A" and "B" and can be identified as follows:

"Class A signifies refrigerants for which toxicity has not been identified at concentrations less than or equal to 400 ppm, based on data used to determine Threshold Limit Value-Time Weighted Average (TLV-TWA) or consistent indices" from Section 6.1.2.[20,27]

"Class B signifies refrigerant for which there is evidence of toxicity at concentrations below 400 ppm, based on data used to determine TLV-TWA on consistent indices" from Section 6.1.2.[20,27]

In identifying toxicity, its class is followed by a number designating flammability. The identifying numbers are

- One (1) for refrigerants with no flame propagation potential.
- Two (2) for refrigerants with low flame propagation potential.
- Three (3) for refrigerants with high flame propagation potential.

A summary from ASHRAE Standard 34-1992, prepared by The Trane Co.,[27] is used with permission of ASHRAE 34-1992, American Society of Heating, Refrigerating, and Air Conditioning Engineers, Inc., ©1992:

"Class 1 indicates refrigerants that do not slow flame propagation when tested in air at 101 kPa (14.7 psi) and 18° C (65°F).

"Class 2 signifies refrigerants having a lower flammability limit (LFL) of more than 0.10 kg/m^3 (0.00625 lb/ft^3) at 21°C and 101 kPa (70°F and 14.7 psia) *and* a heat of combustion of less than 19,000 kJ/kg (8,174 Btu/lb). The heat of combustion shall be calculated assuming that combustion products are in the gas phase and in their most stable state (e.g., C, N, S give CO_2, N_2, SO_3; F and CI give HF and HCL if there is enough H in the molecule, otherwise they give F_2 and CI_2; excess H is converted to H_2O).

"Class 3 indicates refrigerants that are highly flammable, as defined by an LFL of less than or equal to 0.10 kg/m^3 (0.00625 lb/ft^3) at 21° C and 101 kPa (70°F and 14.7 psia) *or* a heat of combustion greater than or equal to 19,000 kJ/kg (8,174 Btu/lb). The heat of combustion is calculated as explained above in the definition of a Class 2 category." from Section 6.1.3

ANSI/ASHRAE Standard 15-1994, "Safety Code for Mechanical Refrigeration," should be studied, examined, and complied with by the design engineer.

(Text continues on page 317)

Table 11-3A
Standard Designation of Refrigerants (ASHRAE *Standard 34*)

Refrigerant Number	Chemical Name or Composition (% by mass)	Chemical Formula
Methane Series		
10	tetrachloromethane (carbon tetrachloride)	CCl_4
11	trichlorofluoromethane	CCl_3F
12	dichlorodifluoromethane	CCl_2F_2
12B1	bromochlorodifluoromethane	$CBrClF_2$
12B2	dibromodifluoromethane	CBr_2F_2
13	chlorotrifluoromethane	$CClF_3$
13B1	bromotrifluoromethane	$CBrF_3$
14	tetrafluoromethane (carbon tetrafluoride)	CF_4
20	trichlorormethane (chloroform)	$CHCl_3$
21	dichlorofluoromethane	$CHCl_2F$
22	chlorodifluoromethane	$CHClF_2$
22B1	bromodifluoromethane	$CHBrF_2$
23	trifluoromethane	CHF_3
30	dichloromethane (methylene chloride)	CH_2Cl_2
31	chlorofluoromethane	CH_2ClF
32	difluoromethane (methylene fluoride)	CH_2F_2
40	chloromethane (methyl chloride)	CH_3Cl
41	fluoromethane (methyl fluoride)	CH_3F
50	methane	CH_4
Ethane Series		
110	hexachloroethane	CCl_3CCl_3
111	pentachlorofluoroethane	CCl_3CCl_2F
112	1,1,2,2-tetrachloro-1,2-difluoroethane	CCl_2FCCl_2F
112a	1,1,1,2-tetrachloro-2,2-difluoroethane	CCl_3CClF_2
113	1,1,2-trichloro-1,2,2-trifluoroethane	CCl_2FCClF_2
113a	1,1,1-trichloro-2,2,2-trifluoroethane	CCl_3CF_3
114	1,2-dichloro-1,1,2,2-tetrafluoroethane	$CClF_2CClF_2$
114a	1,1-dichloro-1,2,2,2-tetrafluoroethane	CCl_2FCF_3
114B2	1,2-dibromo-1,1,2,2-tetrafluoroethane	$CBrF_2CBrF_2$
115	chloropentafluoroethane	$CClF_2CF_3$
116	hexafluoroethane	CF_3CF_3
120	pentachloroethane	$CHCl_2CCl_3$
123	2,2-dichloro-1,1,1-trifluoroethane	$CHCl_2CF_3$
123a	1,2-dichloro-1,1,2-trifluoroethane	$CHClFCClF_2$
124	2-chloro-1,1,1,2-tetrafluoroethane	$CHClFCF_3$
124a	1-chloro-1,1,2,2-tetrafluoroethane	CHF_2CClF_2
125	pentafluoroethane	CHF_2CF_3
133a	2-chloro-1,1,1-trifluoroethane	CH_2ClCF_3
134a	1,1,1,2-tetrafluoroethane	CH_2FCF_3
140a	1,1,1-trichloroethane (methyl chloroform)	CH_3CCl_3
141b	1,1-dichloro-1-fluoroethane	CCl_2FCH_3
142b	1-chloro-1,1-difluoroethane	$CClF_2CH_3$
143a	1,1,1-trifluoroethane	CF_3CH_3
150a	1,1-dichloroethane	$CHCl_2CH_3$
152a	1,1-difluoroethane	CHF_2CH_3
160	chloroethane (ethyl chloride)	CH_3CH_2Cl
170	ethane	CH_3CH_3
Propane Series		
216ca	1,3-dichloro-1,1,2,2,3,3-hexafluoropropane	$CClF_2CF_2CClF_2$
218	octafluoropropane	$CF_3CF_2CF_3$
245cb	1,1,1,2,2-pentafluoropropane	$CF_3CF_2CH_3$
290	propane	$CH_3CH_2CH_3$
Cyclic Organic Compounds		
C316	1,2-dichloro-1,2,3,3,4,4-hexafluorocyclobutane	$C_4Cl_2F_6$
C317	chloroheptafluorocyclobutane	C_4ClF_7
C318	octafluorocyclobutane	C_4F_8
Zeotropic Blends (% by mass)		
400	R12/114 (must be specified)	
401A	R22/152a/124 (53/13/34)	
401B	R22/152a/124 (61/11/28)	
401C	R22/152a/124 (33/15/52)	
402A	R125/290/22 (60/2/38)	
402B	R125/290/22 (38/2/60)	
403A	R290/22/218 (5/75/20)	
Zeotropes *(Continued)*		
403B	R290/22/218 (5/56/39)	
404A	R125/143a/134a(44/52/4)	
405A	R22/152a/142b/C318 (45/7/5.5/42.5)	
406A	R22/600a/142b (55/4 /41)	
407A	R32/125/134a (20/40/40)	
407B	R32/125/134a (10/70/20)	
407C	R32/125/134a (23/25/52)	
407D	R32/125/134a (15/15/70)	
408A	R125/143a/22 (7/46/47)	
409A	R22/124/142b (60/25/15)	
409B	R22/124/142b (65/25/10)	
410A	R32/125 (50/50)	
410B	R32/125 (45/55)	
411A	R1270/22/152a (1.5/87.5/11.0)	
411B	R1270/22/152a (3/94/3)	
412A	R22/218/142b (70/5/25)	
Azeotropic Blends (% by mass)		
500	R12/152a (73.8/26 .2)	
501	R22/12 (75.0/25.0)*	
502	R22/115 (48.8/51.2)	
503	R23/13 (40.1/59.9)	
504	R32/115 (48.2/51.8)	
505	R12/31 (78.0/22.0)*	
506	R31/114 (55.1/44 .9)	
507A	R125/143a (50/50)	
508A	R23/116 (39/61)	
508B	R23/116 (46/54)	
509A	R22/218 (44/56)	
Miscellaneous Organic Compounds		
Hydrocarbons		
600	butane	$CH_3CH_2CH_2CH_3$
600a	2-methyl propane (isobutane)	$CH(CH_3)_3$
Oxygen Compounds		
610	ethyl ether	$C_2H_5OC_2H_5$
611	methyl formate	$HCOOCH_3$
Sulfur Compounds		
620	(Reserved for future assignment)	
Nitrogen Compounds		
630	methyl amine	CH_3NH_2
631	ethyl amine	$C_2H_5NH_2$
Inorganic Compounds		
702	hydrogen	H_2
704	helium	He
717	ammonia	NH_3
718	water	H_2O
720	neon	Ne
728	nitrogen	N_2
732	oxygen	O_2
740	argon	Ar
744	carbon dioxide	CO_2
744A	nitrous oxide	N_2O
764	sulfur dioxide	SO_2
Unsaturated Organic Compounds		
1112a	1,1-dichloro-2,2-difluoroethene	$CCl_2{=}CF_2$
1113	1-chloro-1,2,2-trifluoroethene	$CClF{=}CF_2$
1114	tetrafluoroethene	$CF_2{=}CF_2$
1120	trichloroethene	$CHCl{=}CCl_2$
1130	1,2-dichloroethene (trans)	$CHCl{=}CHCl$
1132a	1,1 difluoroethene (vinylidene fluoride)	$CF_2{=}CH_2$
1140	1-chloroethene (vinyl chloride)	$CHCl{=}CH_2$
1141	1-fluoroethene (vinyl fluoride)	$CHF\text{-}CH_2$
1150	ethene (ethylene)	$CH_2{=}CH_2$
1270	propene (propylene)	$CH_3CH{=}CH_2$

*The exact composition of this azeotrope is in question.

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Table 11-3B
Refrigerant Data and Safety Classifications

The following tables replace Tables 1 and 2 of ANSI/ASHRAE 34-1992. The details as to which addendum contained which changes are outlined in the foreword.

Refrigerant Number	Chemical Name[a,b]	Chemical Formula[a]	Molecular Mass[a]	Normal Boiling Point[a] (°C)	(°F)	Safety Group
Methane Series						
10	tetrachloromethane (carbon tetrachloride)	CCl_4	153.8	77	171	B1
11	trichlorofluoromethane	CCl_3F	137.4	24	75	A1
12	dichlorodifluoromethane	CCl_2F_2	120.9	−30	−22	A1
12B1	bromochlorodifluoromethane	$CBrClF_2$	165.4	−4	25	
12B2	dibromodifluoromethane	CBr_2F_2	209.8	25	77	
13	chlorotrifluoromethane	$CClF_3$	104.5	−81	−115	A1
13B1	bromotrifluoromethane	$CBrF_3$	148.9	−58	−72	A1
14	tetrafluoromethane (carbon tetrafluoride)	CF_4	88.0	−128	−198	A1
20	trichloromethane (chloroform)	$CHCl_3$	119.4	61	142	
21	dichlorofluoromethane	$CHCl_2F$	102.9	9	48	B1
22	chlorodifluoromethane	$CHClF_2$	86.5	−41	−41	A1
22B1	bromodifluoromethane	$CHBrF_2$	130.9	−15	5	
23	trifluoromethane	CHF_3	70.0	-82	−116	A1
30	dichloromethane (methylene chloride)	CH_2Cl_2	84.9	40	104	B2
31	chlorofluoromethane	CH_2ClF	68.5	-9	16	
32	difluoromethane (methylene fluoride)	CH_2F_2	52.0	−52	−62	A2
40	chloromethane (methyl chloride)	CH_3Cl	50.5	−24	−12	B2
41	fluoromethane (methyl fluoride)	CH_3F	34.0	−78	−108	
50	methane	CH_4	16.0	−161	−259	A3
Ethane Series						
110	hexachloroethane	CCl_3CCl_3	236.8	185	365	
111	pentachlorofluoroethane	CCl_3CCl_2F	220.3	135	275	
112	1,1,2,2-tetrachloro-1,2-difluoroethane	CCl_2FCCl_2F	203.8	93	199	
112a	1,1,1,2-tetrachloro-2,2-difluoroethane	CCl_3CClF_2	203.8	91	196	
113	1,1,2-trichloro-1,2,2-trifluoroethane	CCl_2FCClF_2	187.4	48	118	A1
113a	1,1,1-trichloro-2,2,2-trifluoroethane	CCl_3CF_3	187.4	46	115	
114	1,2-dichloro-1,1,2,2-tetrafluoroethane	$CClF_2CClF_2$	170.9	4	38	A1
114a	1,1-dichloro-1,2,2,2-tetrafluoroethane	CCl_2FCF_3	170.9	3	37	
114B2	1,2-dibromo1,1,2,2-tetrafluoroethane	$CBrF_2CBrF_2$	259.9	47	117	
115	chloropentafluoroethane	$CClF_2CF_3$	154.5	−39	−38	A1
116	hexafluoroethane	CF_3CF_3	138.0	−78	−109	A1
120	pentachloroethane	$CHCl_2CCl_3$	202.3	162	324	
123	2,2-dichloro-1,1,1-trifluoroethane	$CHCl_2CF_3$	153.0	27	81	B1*
123a	1,2-dichloro-1,1,2-trifluoroethane	$CHClFCClF_2$	153.0	28	82	
124	2-chloro-1,1,1,2-tetrafluoroethane	$CHClFCF_3$	136.5	−12	10	A1*
124a	1-chloro-1,1,2,2-tetrafluoroethane	$CClF_2CHF_2$	136.5	−10	14	
125	pentafluoroethane	CHF_2CF_3	120.0	−49	−56	A1*
133a	2-chloro-1,1,1-trifluoroethane	CH_2ClCF_3	118.5	6	43	
134a	1,1,1,2-tetrafluoroethane	CF_3CH_2F	102.0	−26	−15	A1*
140a	1,1,1-trichloroethane (methyl chloroform)	CCl_3CH_3	133.4	74	165	
141b	1,1-dichloro-1-fluoroethane	CCl_2FCH_3	117.0	32	90	
142b	1-chloro-1,1-difluoroethane	$CClF_2CH_3$	100.5	−10	14	A2
143a	1,1,1-trifluoroethane	CF_3CH_3	84.0	−47	−53	A2
150a	1,1-dichloroethane	$CHCl_2CH_3$	99.0	57	135	
152a	1,1-difluoroethane	CHF_2CH_3	66.0	−25	−13	A2
160	chloroethane (ethyl chloride)	CH_3CH_2Cl	64.5	12	54	
170	ethane	CH_3CH_3	30.0	−89	−128	A3

[a]The chemical name, chemical formula, molecular mass, and normal boiling point are not part of this standard.

[b]The preferred chemical name is followed by the popular name in parentheses.

[c]Unclassified refrigerants indicate either insufficient data to classify or no formal request for classification.

[d]Held open for future use, formerly used as an indicator of the provisional status of safety classifications.

[e]Sublimes.

*Indicates removal of provisional status of the classification.

(Continued on pages 315 and 316)

Table 11-3B (continued)
Refrigerant Data and Safety Classifications

Refrigerant Number	Chemical Name[a,b]	Chemical Formula[a]	Molecular Mass[a]	Normal Boiling Point[a] (°C)	Normal Boiling Point[a] (°F)	Safety Group
Propane Series						
216ca	1,3-dichloro-1,1,2,2,3,3-hexafluoropropane	$CClF_2CF_2CClF_2$	221.0	36	97	
218	octafluoropropane	$CF_3CF_2CF_3$	188.0	−37	−35	A1
245cb	1,1,1,2,2-pentafluoropropane	$CF_3CF_2CH_3$	134.0	−18	0	
290	propane	$CH_3CH_2CH_3$	44.0	−42	−44	A3
Cyclic Organic Compounds						
C316	1,2-dichloro-1,2,3,3,4,4-hexafluorocyclobutane	$C_4Cl_2F_6$	233.3	60	100	
C317	chloroheptafluorocyclobutane	C_4ClF_7	216.5	26	79	
C318	octafluorocyclobutane	C_4F_8	200.0	−6	21	A1
		See Table 2 for Blends				
Miscellaneous Organic Compounds						
Hydrocarbons						
600	butane	$CH_3CH_2CH_2CH_3$	58.1	0	31	A3
600a	2-methyl propane (isobutane)	$CH(CH_3)_3$	58.1	−12	11	A3
Oxygen Compounds						
610	ethyl ether	$C_2H_5OC_2H_5$	74.1	35	94	
611	methyl formate	$HCOOCH_3$	60.0	32	89	B2
Sulfur Compounds						
620	(Reserved for future assignment)					
Nitrogen Compounds						
630	methyl amine	CH_3NH_2	31.1	−7	20	
631	ethyl amine	$C_2H_5NH_2$	45.1	17	62	
Inorganic Compounds						
702	hydrogen	H_2	2.0	−253	−423	A3
704	helium	He	4.0	−269	−452	A1
717	ammonia	NH_3	17.0	−33	−28	B2
718	water	H_2O	18.0	100	212	A1
720	neon	Ne	20.2	−246	−411	A1
728	nitrogen	N_2	28.1	−196	−320	A1
732	oxygen	O_2	32.0	−183	−297	
740	argon	Ar	39.9	−186	−303	A1
744	carbon dioxide	CO_2	44.0	−78*	−109*	A1
744A	nitrous oxide	N_2O	44.0	−90	−129	
764	sulfur dioxide	SO_2	64.1	−10	14	B1
Unsaturated Organic Compounds						
1112a	1,1-dichloro-2,2-difluoroethene	$CCl_2{=}CF_2$	133.0	19	66	
1113	1-chloro-1,2,2-trifluoroethene	$CClF{=}CF_2$	116.5	−28	−18	
1114	tetrafluoroethene	$CF_2{=}CF_2$	100.0	−76	−105	
1120	trichloroethene	$CHCl{=}CCl_2$	131.4	87	189	
1130	1,2-dichloroethene (trans)	$CHCl{=}CHCl$	96.9	48	118	
1132a	1,1-difluoroethene (vinylidene fluoride)	$CF_2{=}CH_2$	64.0	−82	−116	
1140	1-chloroethene (vinyl chloride)	$CHCl{=}CH_2$	62.5	−14	7	B3
1141	1-fluoroethene (vinyl fluoride)	$CHF{=}CH_2$	46.0	−72	−98	
1150	ethene (ethylene)	$CH_2{=}CH_2$	28.1	−104	−155	A3
1270	propene (propylene)	$CH_3CH{=}CH_2$	42.1	−48	−54	A3

[a]The chemical name, chemical formula, molecular mass, and normal boiling point are not part of this standard.

[b]The preferred chemical name is followed by the popular name in parentheses.

[c]Unclassified refrigerants indicate either insufficient data to classify or no formal request for classification.

[d]Toxicity classification is based on recommended exposure limits provided by chemical suppliers. This rating is provisional and will be reviewed when toxicological testing is completed.

[e]Sublimes.

Table 11-3B (continued)
Refrigerant Data and Safety Classifications

Refrigerant Number	Composition (Wt%)	Azeotropic Temperature (°C)	(°F)	Molecular Mass[a]	Normal Boiling Point[a] (°C)	(°F)	Safety Group
Zeotropes							
400	R-12/114 (must be specified)	none	none				A1/A1
401A	R-22/152a/124 (53/13/34)[e]						A1/A1*
401B	R-22/152a/124 (61/11/28)[e]						A1/A1*
401C	R-22/152a/124(33/15/52)[e]						A1/A1*
402A	R-125/290/22 (60/2/38)[f]						A1/A1*
402B	R-125/290/22 (38/2/60)[f]						A1/A1*
403A	R-290/22/218 (5/75/20)[g]						A1/A1
403B	R-290/22/218 (5/56/39)[g]						A1/A1
404A	R-125/143a/134a (44/52/4)[l]						A1/A1*
405A	R-22/152a/142b/C318 (45/7/5.5/42.5)[h]						A1/A1
406A	R-22/600a/142b (55/4/41)[i]						A1/A2
407A	R-32/125/134a (20/40/40)[j]						A1/A1
407B	R-32/125/134a (10/70/20)[j]						A1/A1
407C	R-32/125/134a (23/25/52)[o]						A1/A1
408A	R-125/143a/22 (7/46/47)[f]						A1/A1
409A	R-22/124/142b (60/25/15)[k]						A1/A1
409B	R-22/124/142b (65/25/10)[k]						A1/A1
410A	R-32/125 (50/50)[f]						A1/A1
410B	R-32/125 (45/55)[n]						A1/A1
411A	R-1270/22/152a (1.5/87.5/11.0)[m]						A1/A2
411B	R-1270/22/152a (3/94/3)[m]						A1/A2
412A	R-22/218/142b (70/5/25)[k]						A1/A2
Azeotropes[b]							
500	R-12/152a (73.8/26.2)	0	32	99.3	−33	−27	A1
501	R-22/12 (75.0/25.0)[c]	−41	−42	93.1	−41	−42	A1
502	R-22/115 (48.8/51.2)	19	66	112.0	−45	−49	A1
503	R-23/13 (40.1/59.9)	88	126	87.5	−88	−126	
504	R-32/115 (48.2/51.8)	17	63	79.2	−57	−71	
505	R-12/31 (78.0/22.0)[c]	115	239	103.5	−30	−22	
506	R-31/114 (55.1/44.9)	18	64	93.7	−12	10	
507A[p]	R-125/143a (50/50)	−40	−40	98.9	−46.7	−52.1	A1
508A[p]	R-23/116 (39/61)	−86	−122	100.1	−86	−122	A1
508B	R-23/116 (46/54)	−45.6	−50.1	95.4	−88.3	−126.9	A1/A1
509A[p]	R-22/218 (44/56)	0	32	124.0	−47	−53	A1

[a]The molecular mass and normal boiling point are not part of this standard.
[b]Azeotropic refrigerants exhibit some segregation of components at conditions of temperature and pressure other than those at which they were formulated. The extent of segregation depends on the particular azeotrope and hardware system configuration.
[c]The exact composition of this azeotrope is in question, and additional experimental studies are needed.
[d]Held open for future use, formerly used as an indicator of the provisional status of safety classifications.
[e]Composition tolerances are (±2/+0.5,−1.5/±1).
[f]Composition tolerances are (±2/±1/±2).
[g]Composition tolerances are (+ 0.2,−2.0/±2.0/±2.0).
[h]Composition tolerances for the individual components are (±2/±1/±1/±2) and for the sum of R-152a and R-142b are (+0,−2).
[i]Composition tolerances are (±2/±1/±1).
[j]Composition tolerances are (±1/±2/±2).
[k]Composition tolerances are (±2/±2/±1).
[l]Composition tolerances are (+0.5,−1.5/+1.5,−0.5).
[m]Composition tolerances are (+0,−1/+2,−0/+0,−1).
[n]Composition tolerances are (±1/±1).
[o]Composition tolerances are (±2/±2/±2).
[p]R-507, R-508, and R-509 are allowed alternative designations for R-507A, R-508A, and R-509A due to a change in designations after assignment of R-500 through R-509. Corresponding changes were not made for R-500 through R-506.
*Indicates removal of provisional status of the classification.
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Table 11-3C
Refrigerant/Refrigerant Blend Data and Safety Classifications for Common Refrigerants

Prefix: "R" or ...	No.	Chemical Name	Chemical Formula	Normal Boiling Point		Safety Group	Quantity of Refrigerant per Occupied Space[a] lb per 1,000 ft³ [b]	Ppm by vol	g/m³ [b]
Methane Series									
CFC	11	trichlorofluoromethane	CCl_3F	24°C	75°F	A1	1.6	4,000	25
CFC	12	dichlorodifluoromethane	CCl_2F_2	−30°C	−22°F	A1	12.0	40,000	200
HCFC	22	chlorodifluoromethane	$CHClF_2$	−41°C	−41°F	A1	9.4	42,000	150
Ethane Series									
CFC	113	1,1,2-trichlorotrifluoroethane	CCl_2FCClF_2	48°C	118°F	A1	1.9	4,000	31
HCFC	123	2,2-dichloro-1,1,1-trifluoroethane	$CHCl_2CF_3$	27°C	81°F	B1[c]	0.4	1,000	6.3
HFC	134a	1,1,1,2-tetrafluoroethane	CH_2FCF_3	−26°C	−15°F	A1[c]	16	60,000	250
HFC	152a	1,1-difluoroethane	CH_3CHF_2	−25°C	−13°F	A2	1.2	7,000	20

Refrigerant No.	Composition (Wt%)	Normal Boiling Point		Safety Group	Quantity of Refrigerant Blend per Room Volume[a] lb per 1,000 ft³ [b]	ppm by vol	g/m³ [b]
Azeotropes[d]							
500	CFC-12/HFC-152a (73.8/26.2)	−33°C	−27°F	A1	12	47,000	200
502	HCFC-22/CFC-115 (48.8/51.2)	−45°C	−49°F	A1	19	65,000	300

[a]Quantities are to be used only in conjunction with Section 7 of ASHRAE Standard 15-1994. The basis for the values shown in this table is a single event in which a complete discharge of any refrigerant system into the occupied space occurs. The quantity of refrigerant is the most restrictive of a minimum oxygen concentration of 19.5% or as follows:

Group A1, 80% of the cardiac sensitization level for R-11, R-12, R-22, R-113, R-134a, R-500, and R-502; others are limited by levels in which oxygen deprivation begins to occur.

Group A2 and A3, approximately 20% of LFL (Lower Flammability Limit).

Group B1, 100% of the measure consistent with the IDLH (Immediately Dangerous to Life or Health) value for R-123.

Group B2 and B3, 100% of IDLH value or 20% of LFL, whichever is lower.

[b]To correct for height above sea level, multiply values given in table by $[1 - 2.42 \times 10^{-6} H]$ where H is measured in feet, or by $[1 - 7.94 \times 10^{-2} h]$ where h is measured in kilometers.

[c]Per ASHRAE Standard 34-1992, this toxicity classification is based on recommended exposure limits provided by chemical suppliers. These ratings are provisional and will be reviewed when toxicological testing is completed.

[d]Per ASHRAE Standard 34-1992, azeotropic refrigerants exhibit some segregation of components at conditions of temperature and pressure other than those at which they were formulated. The extent of segregation depends on the particular azeotrope and hardware system configuration.

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(Text continued from page 312)

See Table 11-3A (ASHRAE) regarding numbers and names of refrigerants.

See Figure 11-23 for a diagram of refrigerant safety group classification. The manufacture and use of some specific refrigerants have been canceled and/or restricted due to the detrimental effect on the ozone layer.

On September 6, 1987, the European Economic Community and the United States signed a phase-out agreement for the manufacture and use of specific refrigerants containing chlorine and bromine in the hydrocarbon molecule because of the effects on the atmosphere's ozone layer.[20] See Reference 20, p. 18.1, for a more detailed history of this

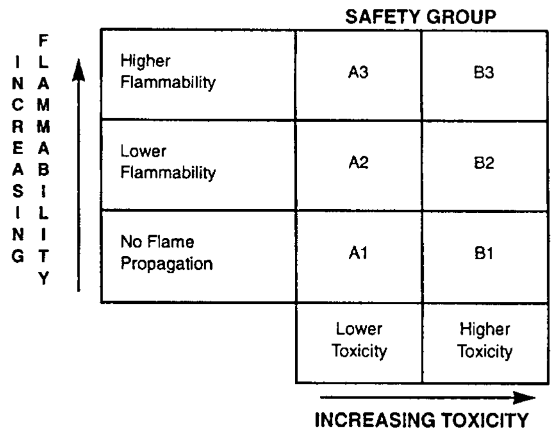

Figure 11-23. Refrigerant safety group classification, per ANSI/ASHRAE® Standard 34-1992, also see Table 11-3B. Used by permission: ANSI/ASHRAE® Standard 34-1992 including Addenda 34a–o and 34q–x, p. 5, ©1992, 1996. American Society of Heating, Refrigerating, and Air Conditioning Engineers, Inc. All rights reserved.

important agreement. The agreement limited the 1988 production of certain CFCs and the levels of certain other halogenated compounds were "frozen" at 1986 levels. In 1994, the Copenhagen Amendment called for the production of CFCs to stop by January 1, 1996, and the production of halogens to stop by January 1, 1994. The hydrofluorocarbon (HFC) refrigerants and their mixtures are not regulated by the agreements. For other specific details not outlined in this text, see Reference 20, Figure 11-24, and Tables 11-6, 11-7, and 11-8.

Table 11-4 presents tabulations of the safety of important refrigerants, but this list does not include all available refrigerants. Table 11-5 summarizes a limited list of comparative hazards to life of refrigerant gas and vapor. The current more applicable refrigerants from the major manufacturers of the CFC and HCFC refrigerants and their azeotropes/blends/mixtures are included, but the list excludes the pure hydrocarbons such as propane, chlorinated hydrocarbons such as methyl chloride and others, inorganics, ammonia, carbon dioxide, etc. See Table 11-6. The CFC compounds have a longer and more serious ozone depletion potential than the HCFC compounds, because these decompose at a much lower atmospheric level and have relatively short atmospheric lifetimes; therefore, they do less damage to the ozone layer.[28] Table 11-7 summarizes alternate refrigerants of the same classes as discussed previously. Table 11-8 correlates DuPont's SUVA® refrigerant numbers to the corresponding ASHRAE numbers.

Figure 11-24 provides a graphical representation of the phase-out of the prominent CFC and HCFC refrigerants and the timing for phasing in the availability of the respective replacements.

Tables 11-9, 11-10, and 11-11 give useful comparative data for most of the common refrigerants.

Pressure, temperature, and enthalpy or total heat values may be obtained from tables or diagrams covering each particular refrigerant. Table 11-12 presents a few comparative values of boiling points (evaporator temperature) and corresponding pressures as taken from such data.

System Performance Comparison

Table 11-13 is a study of the physical properties of several refrigerants indicating a common level of temperature operation of 0°F evaporator operation and 110°F condensing temperature on the high pressure side. The comparison includes an approximate evaluation of the centrifugal and reciprocating applications.

Note that several CFC refrigerants are included, although they are being phased-out and replaced by more environmentally safe refrigerants. These are left in the table at this time because they have been such common/prominent refrigerants in industrial applications.

From Table 11-13, refrigerants no. 114, 11**, and 113 operate below atmospheric pressure in the evaporator and hence at the suction side of the compressor.[12] In general this is not a good condition as it is likely to cause the in-leakage of air and moisture. Refrigerant 114 might be used in order to apply a centrifugal machine to a relatively low tonnage system, as shown in column (H). Refrigerants 12 (soon to be phased-out, see Figure 11-24 [R-11]) and 114 have low condensing pressures, requiring less expensive condensers. In column (B) the refrigerant may be selected based on boiling temperature at 14.7 psia. This indicates an operating pressure that will prevent in-leakage of air. Actually, the suction pressure at the compressor flange will be below atmospheric pressure unless proper allowances are made for the suction line pressure drop. This must be done if air in-leakage is to be avoided. Then the temperature at the evaporator will be increased by an amount corresponding to the temperature equivalent of the pressure drop for the particular refrigerant. (Figure 11-25 illustrates this point for R-12 refrigerant.)

According to Table 11-7 and Figure 11-24, the refrigerant R-11 was to have been phased-out by 1996. In principle the same concept applies to other refrigerant applications as just described. Note that Figure 11-25 for Freon R-12 is used for illustrative purposes, because R-12 was also to be phased-out of availability in 1996 (production); however, similar useful charts can be constructed for other refrigerants.

Compression ratios of column (C) are considered as they affect the limitations on the number of stages in a reciprocating machine or the number of wheels of a centrifugal machine. The molecular weight is a rough guide as far as centrifugal compressor application is concerned, because the higher molecular weight gases require fewer stages of

**Soon to be replaced by R-123.

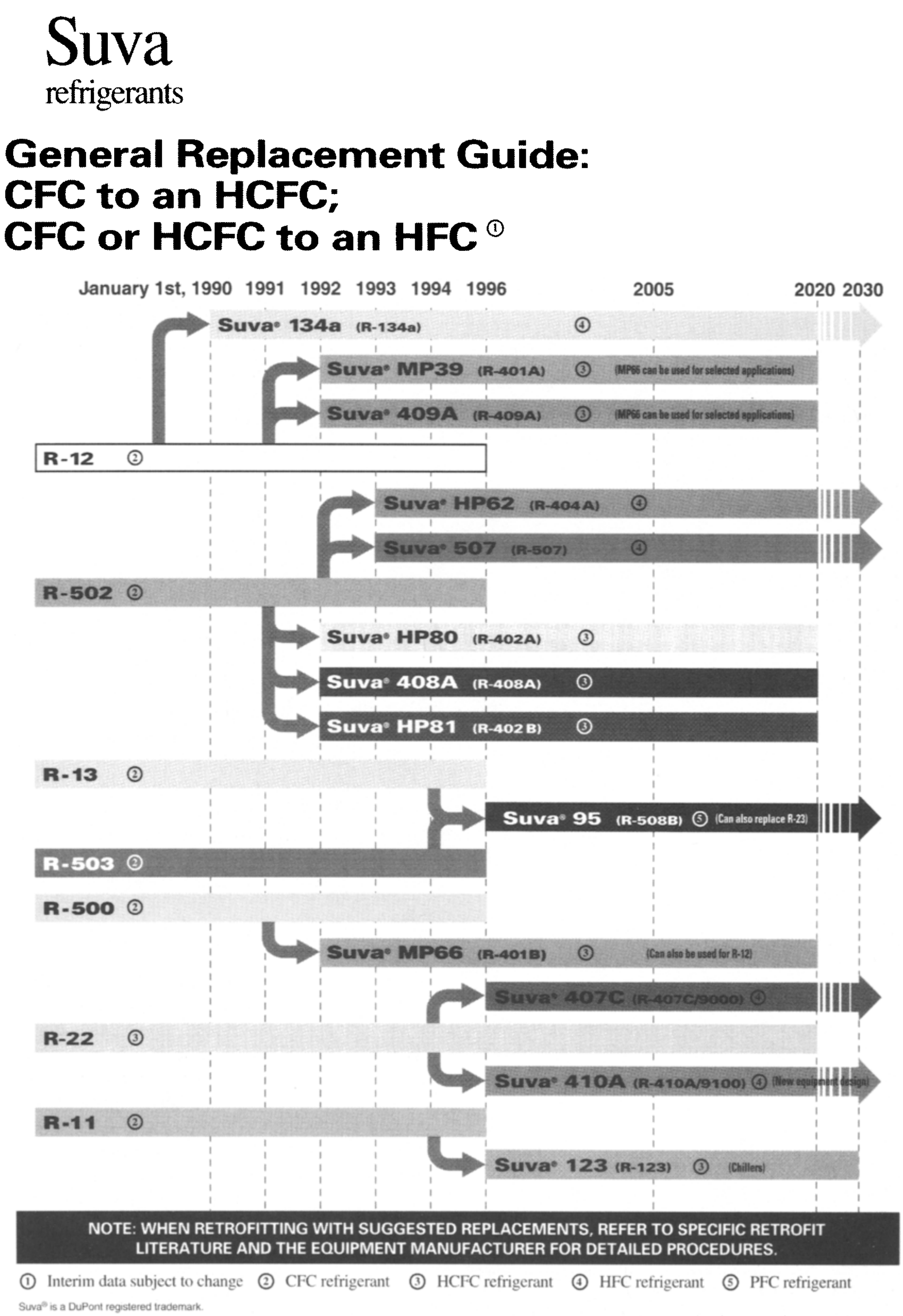

Figure 11-24. General replacement guide for refrigerant phaseout: Suva® Refrigerants. (Used by permission: DuPont Company, Fluoroproducts, SUVA® Refrigerants, Wilmington, DE.)

Table 11-4
Comparison of Safety Group Classifications in ASHRAE Standard 34-1989 and ASHRAE Standard 34-1992

Refrigerant Number	Chemical Formula	Safety Group Old	Safety Group New
10	CCl_4	2	B1
11	CCl_3F	1	A1
12	CCl_2F_2	1	A1
13	$CClF_3$	1	A1
13B1	$CBrF_3$	1	A1
14	CF_4	1	A1
21	$CHCl_2F$	2	B1
22	$CHClF_2$	1	A1
23	CHF_3		A1
30	CH_2Cl_2	2	B2
32	CH_2F_2		A2
40	CH_3Cl	2	B2
50	CH_4	3a	A3
113	CCl_2FCClF_2	1	A1
114	$CClF_2CClF_2$	1	A1
115	$CClF_2CF_3$	1	A1
116	CF_3CF_3		A1
123	$CHCl_2CF_3$		B1
124	$CHClFCF_3$		A1
125	CHF_2CF_3		A1
134a	CF_3CH_2F		A1
142b	$CClF_2CH_3$	3b	A2
143a	CF_3CH_3		A2
152a	CHF_2CH_3	3b	A2
170	CH_3CH_3	3a	A3
218	$CF_3CF_2CF_3$		A1
290	$CH_3CH_2CH_3$	3a	A3
C318	C_4F_8	1	A1
400	R12/114 (must be specified)	1	A1/A1
500	R12/152a (73.8/26.2)	1	A1
501	R22/12 (75.0/25.0)*	1	A1
502	R22/115 (48.8/51.2)	1	A1
507A	R125/143a (50/50)		A1
508A	R23/116 (39/61)		A1
508B	R23/116 (46/54)		A1/A1
509A	R22/218 (44/56)		A1
600	$CH_3CH_2CH_2CH_3$	3a	A3
600a	$CH(CH_3)_3$	3a	A3
611	$HCOOCH_3$	2	B2
702	H_2		A3
704	He		A1
717	NH_3	2	B2
718	H_2O		A1
720	Ne		A1
728	N_2		A1
740	Ar		A1
744	CO_2	1	A1
764	SO_2	2	B1
1140	$CHCl{=}CH_2$		B3
1150	$CH_2{=}CH_2$	3a	A3
1270	$CH_3CH{=}CH_2$	3a	A3

*The exact composition of this azeotrope is in question.

Used by permission: *ANSI/ASHRAE Standard 34-92* ©1992–1994; *1997 ASHRAE Handbook Fundamentals, I-P Ed.* Table 9, p. 18.9, ©1997. American Society of Heating, Refrigerating, and Air Conditioning Engineers, Inc. All rights reserved.

Table 11-5
Underwriters' Laboratories Classification of Comparative Hazard to Life of Gases and Vapors

Group	Definition	Examples
1	Gases or vapors that in concentrations of about $^1/_2$–1% for durations of exposure of about 5 min are lethal or produce serious injury.	Sulfur dioxide
2	Gases or vapors that in concentrations of about $^1/_2$–1% for durations of exposure of about $^1/_2$ hr are lethal or produce serious injury.	Ammonia Methyl bromide
3	Gases or vapors that in concentrations of about 2–2 $^1/_2$ % for durations of exposure of about 1 hr are lethal or produce serious injury.	Carbon tetrachloride Chloroform Methyl formate
4	Gases or vapors that in concentrations of about 2–2 $^1/_2$ % for durations of exposure of about 2 hr are lethal or produce serious injury.	Dichloroethylene Methyl chloride Ethyl bromide
Between 4 and 5	Appear to classify as somewhat less toxic than Group 4. Much less toxic than Group 4 but somewhat more toxic than Group 5.	Methylene chloride Ethyl chloride Refrigerant 113
5a	Gases or vapors much less toxic than Group 4 but more toxic than Group 6.	Refrigerant 11 Refrigerant 22 Carbon dioxide
5b	Gases or vapors that available data indicate would classify as either Group 5a or Group 6.	Ethane Propane Butane
6	Gases or vapors that in concentrations up to at least 20% by volume for durations of exposure of about 2 hr do not appear to produce injury.	Refrigerant 12 Refrigerant 114 Refrigerant 13B1

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compression but require larger amounts of gas per ton of refrigeration. The reciprocating compressor operates on the reverse condition; the lower molecular weight gases allow more gas to be pumped in a particular size cylinder.

Adiabatic head, column (E), is a direct measure of the number of stages of centrifugal compression. In actual rating, the polytropic head must be used. As a guide, 8,000–10,000 ft of head are developed per stage of centrifugal compression, depending upon speed.

The lb/min/ton of refrigeration, column (G), is an indication of the latent heat of the refrigerant. The greater the latent heat, the lower the flow rate per ton. The flow rate, cfm/ton of refrigeration, is an important guide, because refrigerants with low cfm/ton are the best for reciprocating compressor application. For centrifugal compressor application, the low cfm/ton refrigerants are better for the large tonnage requirements, and the high cfm/ton are better for the small tonnage loads.

The approximate minimum tons for centrifugal applications, column (H), is a rough guide based upon 2,000–3,000 cfm at inlet conditions being an efficient minimum capacity. Some designs can be efficient at lower cfm values, depending upon the particular manufacturer's equipment. The tons of refrigeration is actually a function of the evaporator level and condenser temperature, and therefore, the cfm must actually be considered for each particular condition. Refrigerants 11 and 113 are probably not good for this application due to the very low suction pressure condition.

The approximate number of stages for a centrifugal compressor, column (I), is a function of the adiabatic (and actually the polytropic) head and varies with the efficiency and physical properties of the gas.

The minimum recommended saturated suction temperature on single-stage reciprocating applications, column (J), is based on a compression ratio of about 9 to 1. The refrigerants 114, 11 (soon to be replaced by R-123), and 113 are not included due to the large cfm/ton.

The minimum recommended saturated suction temperature on series multistage reciprocating-centrifugal applications, column (K), represents an approximate reasonable limit on suction temperature. The temperatures shown correspond to suction pressures below atmospheric.

Hydrocarbon Refrigerants

The use of methane, ethane, ethylene, propylene, and propane pure light hydrocarbons as refrigerants is quite common, practical, and economical for many hydrocarbon processing plants. Examples include ethylene manufacture from cracking some feedstock, ethylene or other hydrocarbon recycle purification plants, gas-treating plants, and petroleum refineries.

Commonly used hydrocarbon refrigerants and their cooling temperatures are:[30]

Methane	−200 to −300°F
Ethane	−75 to −175°F
Ethylene	−75 to −175°F
Propane	+40 to −50°F
Propylene	+40 to −50°F

Methane is not used frequently in industrial plants for this service, due to mechanical sealing and safety related problems. Due to the danger of air being drawn into hydrocarbon systems, a positive pressure should always be maintained.

Although these hydrocarbons have good refrigerant properties for many applications, it is important to avoid internal pressures in the systems that are below atmospheric pressure because of the danger of air in-leakage and possible explosion of an air-hydrocarbon mixture. Mehra[8-11] presents useful charts for designing and comparing these hydrocarbon refrigerants. Methane is not included because of its somewhat special handling requirements.

Frequently, some plants use mixtures of some of the hydrocarbon refrigerants because of local convenience. In such cases it is important to develop the appropriate mixture's physical property and enthalpy charts for design, because the properties of only one of the components cannot define the mixture.

To specify the system performance requirement, the following must be defined: (1) lowest refrigerant temperature, taking into account the loss in heat transfer ΔT (may be estimated at first) that can occur in the evaporator and (2) condensing temperature of the refrigerant, again taking into account the heat transfer ΔT based on the coolant circulating to accomplish the refrigerant condensing. From these initially established values, system pressures can be defined or established from the thermodynamic charts.

To design hydrocarbon refrigeration, it is necessary to have available accurate Mollier diagrams, vapor pressure charts, etc. (see Figures 11-26 through 11-33[8-11, 15]). By using the convenient estimating charts and excellent presentation of Mehra[8-11, 30] (Figures 11-34 through 11-46) or some other equivalent convenience charts, the performances of various refrigeration systems can be examined and approximately optimized. These charts assume equal ratios of compression per stage for centrifugal compressors with a polytropic efficiency of 0.77. A pressure drop of 1.5 psi has been allowed at the suction to the compressor, a 5 psi drop across the refrigerant condenser for ethylene and ethane, and a 10 psi drop for propylene and propane.[8-11] See Example 11-3 and 11-4 and Figure 11-47.

(Text continues on page 328)

Table 11-6
Genetron® CFC and HCFC Types of Refrigerants Indicating Substitution "Phasing-Out" of Selected Compounds with Newer Replacement Compounds

Selected Physical Data	CFC 113	HCFC 141b	HCFC 123	CFC 11	CFC 114	HCFC 142b	HCFC 124
	Trichlorotrifluoroethane ($C_2Cl_3F_3$) Used in low capacity centrifugal chiller packaged units. Operates with very low system pressures, high gas volumes. Also used as an intermediate in the manufacture of specialty lubricants.	Dichlorofluoroethane (CCl_2FCH_3) The leading substitute blowing agent for CFC-11 in rigid foam insulation applications such as: construction (commercial, residential, and public), appliances, and transport vehicles.	Dichlorotrifluoroethane ($CHCl_2CF_3$) A very low ozone depleting compound that serves as a replacement for CFC-11 in centrifugal chillers.	Trichlorofluoromethane (CCl_3F) A blowing agent for rigid foam insulation applications such as construction (commercial, residential, and public), appliances, and transport vehicles. Refrigerant for centrifugal chillers.	Dichlorotetrafluoroethane ($C_2Cl_2F_4$) Intermediate in pressure and displacement. Principally used with chillers for higher capacities or for lower evaporator temperature process type applications.	Difluorochloroethane (CH_3CClF_2) An effective replacement for CFC-12 in rigid polyurethane, polystyrene, and polyethylene foam insulation applications. Uses include both residential and commercial construction and process piping.	Chlorotetrafluoroethane ($CHClFCF_3$) A potential medium pressure refrigerant for chiller applications. It is designed to replace CFC-12 as a diluent in sterilizing gas. A potential replacement for CFC-11 and -12 in rigid foam insulation applications.
Substitutes (see legend)		□	□			◩	◩
ASHRAE number	R-113	R-141b	R-123	R-11	R-114	R-142b	R-124
Molecular weight	187.4	116.95	152.91	137.4	170.9	100.5	136.5
Boiling point @ 1 atm, (°F)	117.6	89.7	82.2	74.9	38.8	14.4	10.3
Freezing point @ 1 atm, (°F)	−31	−154.3	−160.6	−168	−137	−204.4	−326
Critical temperature (°F)	417	410.4	363.2	388	294	278.8	252
Critical pressure (psia)	499	673.0	533.1	640	473	598	525
Saturated liquid density @ 86°F, (lb/ft^3)	96.8	76.31	90.41	91.4	89.8	68.48	83.6
Specific heat of liquid @ 86°F, (Btu/lb•°F)*	0.22	0.28	0.24	0.21	0.24	0.32	0.27
Specific heat of vapor @ constant pressure* (Cp), @ 86°F and 1 atm, (Btu/lb•°F)	0.15[1]	0.18[1]	0.17	0.14	0.17	0.199	0.17
Flammable range, % vol in air (based on ASHRAE Standard 34 with match ignition)††	None	7.6−17.7†	None	None	None	7.1−18.6†	None
ANSI/ASHRAE Standard 34-1992 safety group classifications	A1	N.C.†††	B1	A1	A1	A2	N.C.†††
	[1]@ 0.2 atm pressure	[1]@ 0.2 atm pressure					

LEGEND:

- □ **CFC 11** Substitutes
- ◩ **CFC 12** Substitutes
- ■ **R-502** Substitutes
- ◧ **CFC 13/R-503** Substitutes
- ◨ **HCFC 22** Substitutes

*Preliminary information based on estimated properties.

▲: Bubble point temperature.

†: Upper and lower vapor flammability (vol %).

††: ASTM E681-85 match ignition ambient conditions.

†††:N.C. not classified.

Ω: @ −30°F.

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(Continued on pages through 325)

Table 11-6 (continued)
Genetron® CFC and HCFC Types of Refrigerants Indicating Substitution "Phasing-Out" of Selected Compounds with Newer Replacement Compounds

Tetrafluoroethane (CF_3CH_2F) A refrigerant to replace CFC-12 in auto air conditioning and in residential, commercial, and industrial refrigeration systems. Also used as a blowing agent in rigid foam insulation.	Dichlorodifluoromethane (CCl_2F_2) A widely used refrigerant in reciprocating and rotary type equipment and in some centrifugal designs. Also used as a diluent in a sterilant gas and as a blowing agent in rigid foam applications.	Chlorodifluoromethane Difluoroethane Chlorotetrafluoroethane ($CHClF_2$/CH_3CHF_2/$CHClFCF_3$) An interim replacement for CFC-12 in medium-temperature commercial refrigeration systems. Contains HCFC-22/HFC-152a/HCFC-124.	Azeotrope (CCl_2F_2/CH_3CHF_2) An azeotropic mixture that has slightly higher vapor pressures and provides higher capacities from the same compressor displacement.	Chlorodifluoromethane Difluoroethane Chlorotetrafluoroethane ($CHClF_2$/CH_3CHF_2/$CHClFCF_3$) An interim replacement for CFC-12 in low-temperature commercial refrigeration systems. Contains HCFC-22/HFC-152a/HCFC-124.	Chlorodifluoromethane Chlorotetrafluoroethane Chlorodifluoroethane ($CHClF_2$/CF_3CHClF/CH_3CClF_2) An interim replacement for CFC-12 in refrigeration systems. Contains HCFC-22/HCFC-124/HCFC-142b.	Chlorodifluoromethane ($CHClF_2$) As a refrigerant, operates with higher system pressures but low compressor displacement. Popular in residential, commercial, and industrial applications. Also used as an intermediate to produce fluoropolymers and as a blowing agent in rigid foam applications.
HFC 134a	**CFC 12**	**Blend MP39**	**Azeotrope 500**	**Blend MP66**	**Blend 409A**	**HCFC 22**
◩		◩		◩	◩	◩ ■
R-134a	R-12	R-401A	R-500	R-401B	R-409A	R-22
102.03	120.9	94.4	99.3	92.9	97.4	86.5
−15.1	−21.6	−27.7▲	−28.3	−30.4▲	−31.6▲	−41.4
−141.9	−252	—	−254	—	—	−256
214	234	228.7	222	226.4	228.9	205
589.8	597	600.0	642	596.1	673.1	722
74.17	80.7	73.8	71.1	73.7	75.2	73.3
0.34	0.24	0.31	0.30	0.30	0.29	0.31
0.21	0.15	0.17	0.18	0.17	0.17	0.16
None	None	None	None	None	None	None
A1	A1	A1/A1	A1	A1/A1	A1/A1	A1
			NOTE: 500 is an azeotropic mixture consisting of CFC 12 (CCl_2F_2), 73.8% by weight and HFC 152a (CH_3CHF_2), 26.2% by weight.			

Table 11-6 (continued)
Genetron® CFC and HCFC Types of Refrigerants Indicating Substitution "Phasing-Out" of Selected Compounds with Newer Replacement Compounds

Selected Physical Data	Blend 407C	Blend 408A	Azeotrope 502	Blend 404A	Blend HP81	Azeotrope AZ-50
	Difluoromethane Pentafluoroethane Tetrafluoroethane ($CH_2F_2/CHF_2CF_3/CF_3CH_2F$) A long-term, non-ozone-depleting replacement for HCFC-22 in various air-conditioning applications, as well as in positive displacement refrigeration systems. It is a ternary blend of HFC-32/HFC-125/HFC-134a.	Chlorodifluoromethane Pentafluoroethane Trifluoroethane ($CHClF_2/CHF_2CF_3/CH_3CF_3$) A interim replacement for retrofitting low- and medium-temperature commercial refrigeration systems.	Azeotrope ($CHClF_2/CClF_2CF_3$) An azeotropic mixture used in low- and medium-temperature applications.	Pentafluoroethane Trifluoroethane Tetrafluoroethane ($CHF_2CF_3/CH_3CF_3/CF_3CH_2F$) A long-term, non-ozone-depleting replacement for R-502 in low- and medium-temperature commercial refrigeration systems.	Chlorodifluoromethane Pentafluoroethane Propane ($CHClF_2/CHF_2CF_3/C_3H_8$) An interim replacement for R-502 used mainly for ice machines and soft ice cream machines.	Azeotrope (CHF_2CF_3/CH_3CF_3) AZ-50 is a non-ozone-depleting azeotropic mixture of HFC-125 and HFC-143a. It has been primarily designed to replace R-502 in low- and medium-temperature commercial refrigeration applications such as supermarket display cases and ice machines.
Substitutes (see legend)	◨	■		■ ◨	■	■ ◨
ASHRAE number	R-407C	R-408A	R-502	R-404A	R-402B	R-507
Molecular weight	86.2	87.7	111.6	97.6	94.7	98.9
Boiling point @ 1 atm, (°F)	−46.4▲	−49.0	−49.8	−51.0▲	−52.5	−52.1
Freezing point @ 1 atm, (°F)	−256	—	—	—	—	−178
Critical temperature (°F)	189.1	201.1	180	162.3	180.7	160
Critical pressure (psia)	699.1	736.7	591	535.0	644.6	550
Saturated liquid density @ 86°F, (lb/ft^3)	70.5	64.87	74.4	63.5	69.73	63.8
Specific heat of liquid @ 86°F (Btu/lb•°F)*	0.37	0.34	0.30	0.37	0.32	0.35
Specific heat of vapor @ constant pressure* (Cp), @ 86°F and 1 atm, (Btu/lb•°F)	0.18	0.19	0.17	0.21	0.18	0.22
Flammable range, (based on ASHRAE Standard 34 with match ignition)††	None	None	None	None	None	None
ANSI/ASHRAE Standard 34-1992 safety group classification	A1/A1	A1/A1	A1	A1/A1	A1/A1	A1
			NOTE: 502 is an azeotropic mixture consisting of HCFC-22 ($CHClF_2$), 48.8% by weight and CFC 115 ($CClF_2CF_3$), 51.2% by weight.			NOTE: AZ-50 is an azeotropic mixture consisting of HFC-125 (CHF_2CF_3), 50% by weight and HFC-143a (CH_3CF_3), 50% by weight. U.S. Patent 5,211,867 AlliedSignal Inc.

LEGEND:

- □ **CFC 11** Substitutes
- ◩ **CFC 12** Substitutes
- ■ **R-502** Substitutes
- ◧ **CFC 13/R-503** Substitutes
- ◨ **HCFC 22** Substitutes

*Preliminary information based on estimated properties.

▲: Bubble point temperature.

†: Upper and lower vapor flammability (vol %).

††: ASTM E681-85 match ignition ambient conditions.

†††:N.C. not classified.

Ω: @ −30°F.

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Table 11-6 (continued)
Genetron® CFC and HCFC Types of Refrigerants Indicating Substitution "Phasing-Out" of Selected Compounds with Newer Replacement Compounds

Blend HP80	**HFC 125**	**Azeotropic Mixture AZ-20**	**CFC 13**	**HFC 23**	**Azeotrope 503**	**Azeotrope 508B**
Chlorodifluoromethane Pentafluoroethane Propane ($CHClF_2/CHF_2CF_3/C_3H_8$) An interim replacement for retrofitting low- and medium-temperature commercial refrigeration systems.	Pentafluoroethane (CHF_2CF_3) A candidate substitute for use in low temperature refrigerant applications. Low critical temperature may limit use as a stand-alone fluid.	Azeotropic Mixture (CH_2F_2/CHF_2CF_3) AZ-20 is an azeotropic mixture of HFC-32 and HFC-125. It has been designed to replace HCFC-22 in air conditioning and refrigeration applications.	Chlorotrifluoromethane ($CClF_3$) A specialty low-temperature refrigerant used in the low stage of cascade systems to provide evaporator temperatures in the range of −75°C.	Trifluoromethane (CHF_3) A specialty low-temperature refrigerant that may be used to replace CFC-13 and R-503 in the low stage of cascade systems.	Azeotrope ($CHF_3/CClF_3$) An azeotropic mixture used in the low stage of cascade type systems where it provides gains in compressor capacity in low-temperature capability.	Azeotrope Trifluoromethane Hexafluoroethane (CHF_3/C_2F_6) A non-ozone-depleting azeotrope of HFC-23 and FC-116 used to replace CFC-13 and R-503 in the low stage of cascade systems.
■	■	◨		◧		◧
R-402A	R-125	R-410A	R-13	R-23	R-503	R-508B
101.6	120.0	72.6	104.5	70.0	87.5	95.4
−54.8▲	−55.8	−62.9	−114.6	−115.7	−126.1	−126.9
−153	—	−247	−294	−247	—	—
168.3	151	163	84	78	67	57.2
615.0	525	720	561	701	632	568.5
69.3	72.3	64.8	82.4Ω	74.7Ω	78.5Ω	81.12Ω
0.33	0.35	0.41	0.24Ω	0.33Ω	0.28Ω	0.32Ω
0.18	0.19	0.21	0.13Ω	0.16Ω	0.14Ω	0.16Ω
None	None	None	None	None	None	None
A1/A1	N.C.†††	A1/A1	A1	N.C.†††	N.C.†††	A1/A1
		NOTE: AZ-20 is an azeotropic mixture consisting of HFC-32 (CH_2F_2), 50% by weight and HFC-125 (CHF_2CF_3), 50% by weight. U.S. Patent 4,978,467 AlliedSignal Inc. European Patent 533,673			NOTE: 503 is an azeotropic mixture consisting of HFC-23 (CHF_3), 40.1% by weight and CFC-13 ($CClF_3$), 59.9% by weight. U.S. Patent 3,640,869 Allied Chemical Corp.	NOTE: 508B is an azeotrope of HFC-23 (CHF_3), 46% by weight and FC-116 (C_2F_6), 54% by weight.

Table 11-7
Alternate Refrigerants

Low- and Medium-Temperature Commercial Refrigeration
Long-Term Replacements

ASHRAE#	Trade Name	Manufacturer	Replaces	Type	Lubricant[a]	Applications	Comments
R-507 (125/143a)	AZ-50 507	AlliedSignal DuPont	R-502 & HCFC-22	Azeotrope	Polyol ester	New equipment & retrofits	Close match to R-502; higher efficiency than 404A; higher efficiency than R-22 at low temperature.
R-404A (125/143a/134a)	404A HP62	AlliedSignal Elf Atochem Dupont	R-502 & HCFC-22	Blend (small glide)	Polyol ester	New equipment & retrofits	Close match to R-502; higher efficiency than R-22 at low temperature.
R-407D (32/125/134a)	407D	ICI	R-500 R-12 low-temp.	Blend (moderate glide)	Polyol ester	New equipment & retrofits	Slightly higher capacity. Higher capacity.

Low- and Medium-Temperature Commercial Refrigeration
Interim Replacements[b]

ASHRAE#	Trade Name	Manufacturer	Replaces	Type	Lubricant[a]	Applications	Comments
R-402A (22/125/290)	HP80	AlliedSignal DuPont	R-502	Blend (small glide)	Alkylbenzene or polyol ester	Retrofits	Higher discharge pressure than R-502.
R-402B (22/125/290)	HP81	AlliedSignal DuPont	R-502	Blend (small glide)	Alkylbenzene or polyol ester	Ice machines	Higher discharge temperature than R-502.
R-408A (125/143a/22)	408A	AlliedSignal Dupont Elf Atochem	R-502	Blend (small glide)	Alkylbenzene or polyol ester	Retrofits	Higher discharge temperature than R-502.

Very Low Temperature Commercial Refrigeration
Long-Term Replacements

ASHRAE#	Trade Name	Manufacturer	Replaces	Type	Lubricant[a]	Applications	Comments
R-23	HFC-23	AlliedSignal DuPont ICI	R-13	Pure fluid	Polyol ester	New equipment & retrofits	Higher discharge temperature than R-13.
R-508B (23/116)	508B 95	AlliedSignal DuPont	R-13 & R-503	Azeotrope	Polyol ester	New equipment & retrofits	
R-508A (23/116)	508A	ICI	R-13 & R-503	Azeotrope	Polyol ester	New equipment & retrofits	

Medium Temperature Commercial Refrigeration Long-Term Replacements

ASHRAE#	Trade Name	Manufacturer	Replaces	Type	Lubricant[a]	Applications	Comments
R134a	HFC-134a	AlliedSignal DuPont Elf Atochem ICI	CFC-12	Pure fluid	Polyol ester	New equipment & retrofits	Close match to CFC-12.

Medium-Temperature Commercial Refrigeration Interim Replacements[b]

ASHRAE#	Trade Name	Manufacturer	Replaces	Type	Lubricant[a]	Applications	Comments
R-401A	MP39	AlliedSignal DuPont	CFC-12	Blend (moderate glide)	Alkylbenzene or polyol ester or some cases mineral oil[d]	Retrofits	Close to CFC-12 Use where evap. temperature = −10°F or higher.

(Continued on page 327)

Medium-Temperature Commercial Refrigeration Interim Replacements[b]

ASHRAE#	Trade Name	Manufacturer	Replaces	Type	Lubricant[a]	Applications	Comments
R-401B (22/152a/124)	MP66	AlliedSignal DuPont	CFC-12	Blend (moderate glide)	Alkylbenzene or polyol ester	Transport[c] Refrigeration retrofits	Close to CFC-12. Use where evap. temperature below -10° F.
			R-500		Alkylbenzene or polyol ester or in some cases mineral oil[d]	Retrofits including air conditioners & dehumidifiers	
R-406A (22/142b/600a)	GHG	Peoples Welding Supply	CFC-12	Blend (high glide)	Mineral oil or alkylbenzene	Retrofits	Can segregate to flammable composition.
R-409A (22/124/142b)	409A	AlliedSignal DuPont Elf Atochem	CFC-12	Blend (high glide)	Alkylbenzene or polyol ester or in some cases mineral oil[d]	Retrofits[c]	Higher capacity than CFC-12. Similar to MP66.
R-414A (22/124/142b/600a)	Autofrost	Peoples Welding Supply	CFC-12	Blend (high glide)	Alkylbenzene or polyol ester or in some cases mineral oil [d]	Retrofits	Similar to 409A.
R-414B (22/124/124b/600a)	Hot Shot	ICOR International	CFC-12	Blend (high glide)	Alkylbenzene or polyol ester or in some cases mineral oil [d]	Retrofits	Similar to 409A.
R-416A (124/134a/600)	FRIGC FR-12	Intercool Energy	CFC-12	Blend (small glide)	Polyol ester	Retrofits	Lower pressure than 134a at high ambient conditions.

Commercial and Residential Air-Conditioning Long-Term Replacements

ASHRAE#	Trade Name	Manufacturer	Replaces	Type	Lubricant[a]	Applications	Comments
R-123	HCFC-123	AlliedSignal DuPont Elf Atochem	CFC-11	Pure fluid	Alkylbenzene or mineral oil	Centrifugal chillers	Lower capacity than R-11. With modifications, equivalent performance to CFC-11.
R-134a	HFC-134a	AlliedSignal DuPont Elf Atochem ICI	CFC-12	Pure fluid	Polyol ester	New equipment & retrofits	Close match to CFC-12.
			HCFC-22	Pure fluid	Polyol ester	New equipment	Lower capacity than HCFC-22, larger equipment needed.
R-410A (32/125)	AZ-20 9100 410A	AlliedSignal Dupont Elf Atochem	HCFC-22	Azeotropic mixture	Polyol ester	New equipment	Higher efficiency than HCFC-22. Requires equipment redesign.
R-407C (32/125/134a)	407C	AlliedSignal Elf Atochem ICI	HCFC-22	Blend (high glide)	Polyol ester	New equipment & retrofits	Lower efficiency than HCFC-22; close capacity to HCFC-22.
	9000	Dupont					

[a]Check with the compressor manufacturer for their recommended lubricant.
[b]Interim replacement, contains HCFC-22, which is scheduled for phase-out under the Montreal Protocol.
[c]Not recommended for automotive air-conditioning.
[d]For more information on when to use mineral oil, see Applications Bulletin GENAP1, "Are Oil Changes Needed for HCFC Blends?"
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Note: Company disclaimer applies.

Table 11-8
Correlation of DuPont SUVA® Refrigerant Number with ASHRAE Replacement Numbers

Product Name	Replaces	Applications
SUVA® *123*	CFC-11	Centrifugal chillers
SUVA® 124	CFC-114	A/C and refrigeration applications; marine chillers
SUVA® *134a*	CFC-12	Centrifugal and reciprocating chillers; medium-temperature refrigeration; appliances New automotive A/C and automotive service refrigerant
SUVA® MP39 (R-401A)	CFC-12	Service refrigerant for medium-temperature commercial refrigeration; appliances
SUVA® MP66 (R-401B)	CFC-12 R-500	Service refrigerant for low-temperature commercial and transport refrigeration
SUVA®HP62 (R-404A)	R-502	All commercial refrigeration
SUVA® HP80 (R-402A)	R-502	Service refrigerant for all commercial refrigeration
SUVA® HP81 (R-402B)	R-502	Ice machines and other medium-temperature equipment
SUVA® 9000 (R-407C)	HCFC-22	An equivalent pressure replacement for HCFC-22 with 0 ozone depletion potential for use in commercial and residential air conditioners and heat pumps. Suva® 9000 provides the closest match to HCFC-22 performance in existing HCFC equipment design.
SUVA®9100 (R-410A)	HCFC-22	A high pressure replacement for HCFC-22 with 0 ozone depletion potential in new HCFC equipment
SUVA® 95 (R-508B)	R-503 CFC-13	Non-ozone-depleting replacement for R-503 and CFC-13 in very low-temperature applications (less than −40° F)

For more information, call 1-800-235-SUVA.

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(Text continued from page 321)

Example 11-3. Single-Stage Propane Refrigeration System, Using Charts of Mehra[30]

Determine the approximate horsepower and condenser duty for a single-stage propane refrigeration system (see flow diagram of Figure 11-43).

Refrigeration load of process cooling: 35 MM Btu/hr
Refrigeration evaporating temperature: −13°F
Temperature of process fluid: −10°F
Condensing temperature of refrigerant (propane): 110°F
Referring to Figure 11-43 at a condensing temperature of 110°F and −13°F evaporator temperature: 222 hp/10^6 Btu/hr of refrigeration duty
Condenser duty: 1.57 MM Btu/hr/(10^6Btu) (hr) of refrigerant duty.
Then, gas hp = 35(222) = 7,770 hp
Condenser duty = 35(1.57) = 54.95 MM Btu/hr

Example 11-4. Two-Stage Propane Refrigeration System, Using Charts of Mehra[30]

Determine the approximate horsepower and condenser duty for the two-stage system of Figure 11-47. Note: A simpler system omits the sub-cooler and smaller evaporator (see diagram on Figure 11-37).

Refrigeration load of processing cooling:
(25 + 10 + 4) MM Btu/hr
Refrigerant evaporating temperature: Consider the 25MM Btu/hr @ −40°F independently as a simple two-stage system. Thus, condensing at +120°F
Duty = 1.71 MM Btu/hr/(10^6Btu/hr), see Figure 11-46. Using Figure 11-45, read the gas hp/10^6 Btu/hr at evaporator of −40°F and refrigerant condensing temperature of +120°F = 279 hp/(10^6Btu) (hr)
Combining the second refrigeration load of 10 MM Btu/hr and +25°F and the refrigerant subcooler duty of 4 MM Btu/hr as a single-stage system using Figure 11-43: 143.5 hp/10^6Btu/hr refrigerant duty condensing at 120°F. From Figure 11-44 at an evaporator temperature of +25°F and a condensing temperature of +120°F,

Condenser duty = 1.366 MM Btu/hr/(10^6Btu hr).

The total horsepower and condenser duty are the sums for each stage of the system. Then,

Gas hp = (25)(279) + (10 − 4)(143.5) = 7,836 hp
Condenser duty = (25)(1.71) + (10 − 4)(1.366)
= 50.95 MM Btu/hr

Hydrocarbon Mixtures and Refrigerants

Many gas processing and hydrocarbon cracking and petroleum processing plants have large quantities of hydrocarbon mixtures (such as propane-butane, propane propylene, ethylene-ethane, ethylene-propylene, etc.) available and these mixtures can be used as refrigerants in properly designed systems. For proper operation of the compressor, condenser, and evaporator in these systems, it is extremely important to maintain uniform compositions (avoiding leaks and random make-up additions of hydrocarbon) and to use only two hydrocarbons in the refrigerant mix, rather than three or more, as the operation can become quite complicated otherwise. Even for a mixture high in composition

(Text continued on page 333)

Table 11-9
Physical Properties of Selected Refrigerants[a]

No.	Refrigerant: Chemical Name or Composition (% by mass)	Chemical Formula	Molecular Mass	Boiling Pt. (NBP) at 14.693 psia, °F	Freezing Point, °F	Critical Temperature, °F	Critical Pressure, psia	Critical Volume, ft³/lb	Refractive Index of Liquid[b,c]
704	Helium	He	4.0026	−452.1	None	−450.3	33.21	0.2311	1.021 (NBP) 5,461 Å
702p	Hydrogen, para	H_2	2.0159	−423.2	−434.8	−400.3	187.5	0.5097	1.09 (NBP)[f]
702n	Hydrogen, normal	H_2	2.0159	−423.0	−434.5	−399.9	190.8	0.5320	1.097 (NBP) 5,791 Å
720	Neon	Ne	20.183	−410.9	−415.5	−379.7	493.1	0.03316	—
728	Nitrogen	N_2	28.013	−320.4	−346.0	−232.4	492.9	0.05092	1.205 (83 K) 5,893 Å
729	Air	—	28.97	−317.8	—	−220.95	548.9	0.0530	
						−221.1	546.3	0.05007	
740	Argon	Ar	39.948	−302.55	−308.7	−188.48	704.9	0.0301	1.233 (84 K) 5,893 Å
732	Oxygen	O_2	31.9988	−297.332	−361.8	−181.424	731.4	0.03673	1.221 (92 K) 5,893Å
50	Methane	CH_4	16.04	−258.7	−296	−116.5	673.1	0.099	
14	Tetrafluoromethane	CF_4	88.01	−198.3	−299	−50.2	543	0.0256	
1150	Ethylene	C_2H_4	28.05	−154.7	−272	48.8	742.2	0.070	1.363(−148)[1]
744A[2]	[2]Nitrous oxide	N_2O	44.02	−129.1	−152	97.7	1,048	0.0355	
170	Ethane	C_2H_6	30.7	−127.85	−297	90.0	709.8	0.0830	
503	R23/13 (40.1/59.9)	—	87.5	−127.6	—	67.1	607	0.0326	
23	Trifluoromethane	CHF_3	70.02	−115.7	−247	78.1	701.4	0.0311	
13	Chlorotrifluoromethane	$CClF_3$	104.47	−114.6	−294	83.9	561	0.0277	1.146 (77)[4]
744	Carbon dioxide	CO_2	44.01	−109.2[d]	−69.9[e]	87.9	1,070.0	0.0342	1.195 (59)
13B1	[1]Bromotrifluoromethane	$CBrF_3$	148.93	−71.95	−270	152.6	575	0.0215	1.239 (77)[4]
504	R32/115 (48.2/51.8)	—	79.2	−71.0	—	151.5	690.5	0.0324	
32	Difluoromethane	CH_2F_2	52.02	−61.1	−213	173.14	845.6	0.03726	
125	Pentafluoroethane	C_2HF_5	120.03	−55.43	−153.67	151.34	526.57	—	—
1270	Propylene	C_3H_6	42.09	−53.86	−301	197.2	670.3	0.0720	1.3640 (−58)[1]
502[5]	R22/115 (48.8/51.2)	—	111.63	−49.8	—	179.9	591.0	0.0286	
290	Propane	C_3H_8	44.10	−43.76	−305.8	206.1	616.1	0.0726	1.3397 (−43)
22	Chlorodifluoromethane	$CHClF_2$	86.48	−41.36	−256	204.8	721.9	0.0305	1.234 (77)[4]
115	Chloropentafluoroethane	$CClF_2CF_3$	154.48	−38.4	−159	175.9	457.6	0.0261	1.221 (77)[4]
500	R12/152 a (73.8/26.2)	—	99.31	−28.3	−254	221.9	641.9	0.0323	
717	Ammonia	NH_3	17.03	−28.0	−107.9	271.4	1,657	0.068[d]	1.325 (61.7)
12	Dichlorodifluoromethane	CCl_2F_2	120.93	−21.62	−252	233.6	596.9	0.0287	1.288 (77)[4]
134a	Tetrafluoroethane	CF_3CH_2F	102.03	−15.08	−141.9	214.0	589.8	0.029	—
152a	Difluoroethane	CHF_2CH_3	66.05	−13.0	−178.6	236.3	652	0.0439	
40[2]	Methyl chloride	CH_3Cl	50.49	−11.6	−144	289.6	968.7	0.0454	
124	Chlorotetrafluoroethane	$CHClFCF_3$	136.47	8.26	−326.47	252.5	530.84	—	—
600a	Isobutane	C_4H_{10}	58.13	10.89	−255.5	275.0	529.1	0.0725	1.3514 (−13)[1]
764[6]	Sulfur dioxide	SO_2	64.07	14.0	−103.9	315.5	1,143	0.0306	
142b	Chlorodifluoroethane	$CClF_2CH_3$	100.5	14.4	−204	278.8	598	0.0368	
630[6]	Methyl amine	CH_3NH_2	31.06	19.9	−134.5	314.4	1,082		1.432 (63.5)
C318	Octafluorocyclobutane	C_4F_8	200.04	21.5	−42.5	239.6	403.6	0.0258	
600	Butane	C_4H_{10}	58.13	31.1	−217.3	305.6	550.7	0.0702	1.3562 (5)[1]
114	Dichlorotetrafluoroethane	$CClF_2CClF_2$	170.94	38.8	−137	294.3	473	0.0275	1.294 (77)
21[7]	Dichlorofluoromethane	$CHCl_2F$	102.92	47.8	−211	353.3	750	0.0307	1.332 (77)[4]
160[2]	Ethyl chloride	C_2H_5Cl	64.52	54.32	−216.9	369.0	764.4	0.0485	
631[6]	Ethyl amine	$C_2H_5NH_2$	45.08	61.88	−113	361.4	815.6		
11	Trichlorofluoromethane	CCl_3F	137.38	74.87	−168	388.4	639.5	0.0289	1.362 (77)[4]
123	Dichlorotrifluoroethane	$CHCl_2CF_3$	152.93	82.17	−160.87	362.82	532.87	—	—
611[6]	Methyl formate	$C_2H_4O_2$	60.05	89.2	−146	417.2	870	0.0459	
141b	Dichlorofluoroethane	CCl_2FCH_3	116.95	89.6	—	399.6	616.4	—	
610[6]	Ethyl ether	$C_4H_{10}O$	74.12	94.3	−177.3	381.2	523	0.0607	1.3526 (68)
216ca	Dichlorohexafluoropropane	$C_3Cl_2F_6$	220.93	96.24	−193.7	356.0	399.5	0.0279	
30[6]	Methylene chloride	CH_2Cl_2	84.93	104.4	−142	458.6	882		1.4244 (68)[3]
113	Trichlorotrifluoroethane	CCl_2FCClF_2	187.39	117.63	−31	417.4	498.9	0.0278	1.357 (77)[4]
1130[8]	Dichloroethylene	CHCl=CHCl	96.95	118	−58	470	795		
1120[6]	Trichloroethylene	$CHCl=CCl_2$	131.39	189.0	−99	520	728		1.4782 (68)[3]
718[6]	Water	H_2O	18.02	212	32	705.18	3200	0.0498	

[a] Data from ASHRAE *Thermodynamic Properties of Refrigerants* (Stewart *et al.* 1986) or from McLinden (1990), unless otherwise noted.

[b] Temperature of measurement (°F, unless Kelvin is noted) shown in parentheses. Data from CRC *Handbook of Chemistry and Physics* (CRC 1987), unless otherwise noted.

[c] For the sodium D line.

[d] Sublimes.

[e] At 76.4 psia.

[f] Dielectric constant data.

References:

[1] Kirk and Othmer (1956).

[2] *Matheson Gas Data Book* (1966).

[3] Electrochemicals Department, E.I. duPont de Nemours & Co.

[4] *Bulletin* B-32A (duPont).

[5] *Bulletin* T-502 (duPont 1980).

[6] *Handbook of Chemistry* (1967).

[7] *Bulletin* G-1 (duPont).

[8] CRC *Handbook of Chemistry and Physics* (CRC 1987).

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Table 11-10
Comparative Refrigerant Performance per Ton of Refrigeration[a]

Refrigerant No.	Chemical Name or Composition (% by mass)	Evaporator Pressure, psia	Condenser Pressure, psia	Compression Ratio	Net Refrigerating Effect, Btu/lb$_m$	Refrigerant Circulated, lb$_m$/min	Liquid Circulated, in.^{3}/min	Specific Volume of Suction Gas, ft^3/lb$_m$	Compressor Displacement, cfm	Power Consumption, hp	Coefficient of Performance	Comp. Discharge Temp., °F
170	Ethane	236.410	674.710	2.85	69.27	2.88704	289.1266	0.5344	1.543	1.733	2.72	123
744	Carbon dioxide	332.375	1045.360	3.15	57.75	3.46320	158.5272	0.2639	0.914	1.678	2.81	156
13B1	Bromotrifluoromethane	77.820	264.128	3.39	28.45	7.02901	129.7814	0.3798	2.669	1.134	4.16	104
125	Pentafluoroethane	58.870	228.110	3.87	37.69	5.30645	126.8148	0.6281	3.333	1.283	3.67	108
1270	Propylene	52.704	189.440	3.59	123.15	1.62401	90.7048	2.0487	3.327	1.035	4.56	108
290	Propane	42.37	156.820	3.70	120.30	1.66251	95.0386	2.4589	4.088	1.031	4.57	98
502	R-22/115 (48.8/51.2)	50.561	191.290	3.78	44.91	4.45305	103.3499	0.8015	3.569	1.067	4.42	98
22	Chlorodifluoromethane	42.963	172.899	4.02	69.90	2.86144	67.6465	1.2394	3.546	1.011	4.67	128
717	Ammonia	34.170	168.795	4.94	474.20	0.42177	19.6087	8.1790	3.450	0.989	4.77	210
500	R-12/152a (73.8/26.2)	31.064	127.504	4.10	60.64	3.29834	80.1925	1.5022	4.955	1.005	4.69	105
12	Dichlorodifluoromethane	26.505	107.991	4.07	50.25	3.97981	85.2280	1.4649	5.830	0.992	4.75	100
134a	Tetrafluoroethane	23.790	111.630	4.69	64.77	3.08785	71.8199	1.9500	6.021	1.070	4.41	108
124	Chlorotetrafluoroethane	12.960	64.590	4.98	50.93	3.92696	81.1580	2.7140	10.658	1.054	4.47	90
600a	Isobutane	12.924	59.286	4.59	113.00	1.76991	90.0059	6.4189	11.361	1.070	4.41	80
600	Butane	8.176	41.191	5.04	125.55	1.59299	77.7772	10.2058	16.258	0.952	4.95	88
114	Dichlorotetrafluoroethane[b]	6.747	36.493	5.41	43.02	4.64889	89.5631	4.3400	20.176	1.015	4.65	86
11	Trichlorofluoromethane	2.937	18.318	6.24	67.21	2.97592	56.2578	12.2400	36.425	0.939	5.02	110
123	Dichlorotrifluoroethane	2.290	15.900	6.94	61.19	3.26829	62.3495	14.0800	46.018	0.974	4.84	94
113	Trichlorotrifluoroethane[b]	1.006	7.884	7.83	52.08	3.84047	68.5997	26.2845	100.945	1.105	4.27	86

[a]Based on 5°F evaporation and 86°F condensation.
[b]Saturated suction except R-113 and R-114. Enough superheat was added to give saturated discharge.
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Table 11-11
Comparative Refrigerant Performance per Ton at Various Evaporating and Condensing Temperatures

Refrigerant No.	Chemical Name or Composition (% by mass)	Suction Temp., °F	Evaporator Pressure, psia	Condenser Pressure, psia	Compression Ratio	Net Refrigerating Effect, Btu/lb$_m$	Refrigerant Circulated, lb$_m$/min	Specific Volume of Suction Gas, ft^3/lb$_m$	Compressor Displacement, cfm	Power Consumption, Hp
−130°F Saturated Evaporating, 0°F Suction Superheat, −40°F Saturated Condensing										
1150	Ethylene	−130	30.887	210.670	6.82	142.01	1.40835	3.8529	5.426	1.756
170	Ethane	−130	13.620	112.790	8.28	156.58	1.27730	8.3575	10.675	1.633
13	Chlorotrifluoromethane	−130	9.059	88.037	9.72	45.82	4.36529	3.6245	15.822	1.685
23	Trifluoromethane	−130	9.06	103.03	11.37	79.38	2.51953	5.4580	13.752	1.753
−100°F Saturated Evaporating, 0°F Suction Superheat, −30°F Saturated Condensing										
170	Ethane	−100	31.267	134.730	4.31	157.76	1.26775	3.8671	4.903	1.118
13	Chlorotrifluoromethane	−100	22.276	106.290	4.77	46.23	4.32581	1.5631	6.762	1.153
125	Pentafluoroethane	−100	3.780	27.760	7.34	56.43	3.54403	8.3900	29.734	1.101
22	Chlorodifluoromethane	−100	2.380	19.629	8.25	90.75	2.20397	18.5580	40.901	1.074
23	Trifluoromethane	−100	23.74	125.99	5.31	79.37	2.51984	2.219	5.592	1.178
−76°F Saturated Evaporating, 0°F Suction Superheat, 5°F Saturated Condensing										
1150	Ethylene	−76	109.370	416.235	3.81	116.95	1.71021	1.1617	1.987	1.478
170	Ethane	−76	54.634	235.440	4.31	322.65	0.61987	2.2906	1.420	0.566
23	Trifluoromethane	−76	45.410	237.180	5.22	69.60	2.87356	1.2030	3.457	1.394
13	Chlorotrifluoromethane	−76	40.872	192.135	4.70	39.42	5.07389	0.8801	4.465	1.382
13B1	Bromotrifluoromethane	−76	13.173	77.820	5.91	37.8	5.29128	2.0329	10.757	1.253
125	Pentafluoroethane	−76	8.210	58.870	7.17	50.62	3.95101	4.0720	16.089	1.277
290	Propane	−76	6.150	42.367	6.89	147.39	1.35699	14.8560	20.159	1.196
22	Chlorodifluoromethane	−76	5.438	42.963	7.90	84.24	2.37425	8.5925	20.401	1.195
717	Ammonia	−76	3.18	34.26	10.79	540.63	0.37	75.7838	28.04	1.247
12	Dichlorodifluoromethane	−76	3.277	26.501	8.09	58.61	3.41219	10.2448	34.957	1.191
134a	Tetrafluoroethane	−76	2.3	23.77	10.32	78.1	2.561	17.3038	44.315	1.182
−40°F Saturated Evaporating, 0°F Suction Superheat, 68°F Saturated Condensing										
744	Carbon dioxide	−40	145.770	830.530	5.70	77.22	2.59000	0.6128	1.587	2.208
23	Trifluoromethane	−40	103.030	597.900	5.80	45.67	4.37924	0.5448	2.386	2.442

(Continued on page 331)

Refrigerant No.	Refrigerant Chemical Name or Composition (% by mass)	Suction Temp., °F	Evaporator Pressure, psia	Condenser Pressure, psia	Compression Ratio	Net Refrigerating Effect, Btu/lb$_m$	Refrigerant Circulated, lb$_m$/min	Specific Volume of Suction Gas, ft^3/lb$_m$	Compressor Displacement, cfm	Power Consumption, hp
13B1	Bromotrifluoromethane	−40	31.855	207.854	6.53	28.81	6.94155	0.8915	6.189	1.855
125	Pentafluoroethane	−40	21.840	175.100	8.02	37.44	5.34188	1.6250	8.681	1.962
290	Propane	−40	16.099	121.560	7.55	119.33	1.67602	6.0829	10.195	1.670
22	Chlorodifluoromethane	−40	15.268	131.997	8.65	70.65	2.83106	3.2805	9.287	1.606
717	Ammonia	−40	10.4	124.31	11.95	486.55	0.411	25.1436	10.334	1.576
500	R12/152a (73.8/26.2)	−40	10.959	96.948	8.85	60.24	3.31989	3.9895	13.245	1.583
12	Dichlorodifluoromethane	−40	9.304	82.295	8.84	49.44	4.04572	3.8868	15.725	1.596
134a	Tetrafluoroethane	−40	7.42	83.0	11.19	63.17	3.166	5.7899	18.331	1.597
−10°F Saturated Evaporating, 0°F Suction Superheat, 100°F Saturated Condensing										
123	Dichlorotrifluoroethane	−10	1.48	20.8	14.07	55.64	3.594	21.1405	75.979	1.436
11	Trichlorofluoromethane	−10	1.92	23.37	12.2	61.82	3.235	18.1691	58.777	1.398
124	Chlorotetrafluoroethane	−10	8.950	80.920	9.04	44.99	4.44543	3.8410	17.075	1.649
134a	Tetrafluoroethane	−10	16.62	138.98	8.36	56.57	3.535	2.7114	9.585	1.589
12	Dichlorodifluoromethane	−10	19.197	131.720	6.86	44.89	4.45563	1.9803	8.823	1.606
717	Ammonia	−10	23.73	211.96	8.93	461.25	0.434	11.6774	5.068	1.494
22	Chlorodifluoromethane	−10	31.231	210.670	6.75	64.07	3.12173	1.6757	5.231	1.602
502	R22/115 (48.8/51.2)	−10	37.256	230.890	6.20	39.05	5.12177	1.0727	5.494	1.904
125	Pentafluoroethane	−10	43.320	276.950	6.39	31.09	6.43294	0.8459	5.442	2.172
−10°F Saturated Evaporating, 75°F Suction Superheat (Not Included in Refrigeration Effect), 100°F Saturated Condensing										
123	Dichlorotrifluoroethane	65	1.48	20.8	14.07	55.64	3.594	24.8022	89.139	1.678
11	Trichlorofluoromethane	65	1.92	23.37	12.2	61.82	3.235	21.2804	68.842	1.632
124	Chlorotetrafluoroethane	65	8.950	80.920	9.04	44.99	4.44543	4.5310	20.142	1.919
134a	Tetrafluoroethane	65	16.62	138.98	8.36	56.57	3.535	3.2359	11.439	1.906
12	Dichlorodifluoromethane	65	19.197	131.720	6.86	44.89	4.45563	2.3597	10.514	1.914
717	Ammonia	65	23.73	211.96	8.93	461.25	0.434	13.7281	5.958	1.742
22	Chlorodifluoromethane	65	31.231	210.670	6.75	64.07	3.12173	2.0121	6.281	1.924
502	R22/115 (48.8/51.2)	65	37.256	230.890	6.20	39.05	5.12177	1.3015	6.666	2.310
125	Pentafluoroethane	65	43.320	276.950	6.39	31.09	6.43294	1.0280	6.613	2.573
−10°F Saturated Evaporating, 75°F Suction Superheat (Included in Refrigeration Effect), 100°F Saturated Condensing										
123	Dichlorotrifluoroethane	65	1.48	20.8	14.07	67.3	2.972	24.7971	73.697	1.387
11	Trichlorofluoromethane	65	1.92	23.37	12.2	71.88	2.783	21.2763	59.212	1.403
124	Chlorotetrafluoroethane	65	8.950	80.920	9.04	57.33	3.48857	4.5310	15.807	1.506
134a	Tetrafluoroethane	65	16.62	138.98	8.36	71.25	2.807	3.2358	9.083	1.513
12	Dichlorodifluoromethane	65	19.197	131.720	6.86	55.83	3.58251	2.3597	8.454	1.539
717	Ammonia	65	23.73	211.96	8.93	498.44	0.401	13.7506	5.514	1.612
22	Chlorodifluoromethane	65	31.231	210.670	6.75	75.95	2.63326	2.0121	5.298	1.623
502	R22/115 (48.8/51.2)	65	37.256	230.890	6.20	51.23	3.90362	1.3015	5.081	1.761
125	Pentafluoroethane	65	43.320	276.950	6.39	45.13	4.43164	1.0280	4.556	1.773
20°F Saturated Evaporating, 0°F Suction Superheat, 80°F Saturated Condensing										
125	Pentafluoroethane	20	78.400	209.270	2.67	41.47	4.82276	0.4735	2.284	0.831
290	Propane	20	55.931	144.330	2.58	128.39	1.55775	1.8873	2.940	0.721
22	Chlorodifluoromethane	20	57.786	158.360	2.74	73.12	2.73512	0.9334	2.553	0.707
717	Ammonia	20	48.19	153.06	3.18	497.1	0.402	6.0498	2.432	0.677
500	R12/152a (73.8/26.2)	20	41.936	116.620	2.78	64.15	3.11784	1.1294	3.521	0.702
12	Dichlorodifluoromethane	20	35.765	98.850	2.76	53.22	3.75827	1.1045	4.151	0.701
134a	Tetrafluoroethane	20	33.13	101.49	3.06	67.91	2.945	1.4088	4.149	0.693
124	Chlorotetrafluoroethane	20	18.290	58.410	3.19	54.67	3.65831	1.9640	7.185	0.710
600a	Isobutane	20	17.916	53.907	3.01	121.45	1.64677	4.7361	7.799	0.706
600	Butane	20	11.557	37.225	3.22	134.18	1.49054	7.3947	11.022	0.686
123	Dichlorotrifluoroethane	20	3.48	14.07	4.04	64.75	3.089	9.5073	29.368	0.656
11	Trichlorofluoromethane	20	4.33	16.17	3.74	69.78	2.866	8.5213	24.422	0.649
40°F Saturated Evaporating, 0°F Suction Superheat, 100°F Saturated Condensing										
125	Pentafluoroethane	40	111.710	276.950	2.48	37.10	5.39084	0.3312	1.785	0.860
290	Propane	40	78.782	189.040	2.40	114.96	1.73974	1.3563	2.360	0.750
22	Chlorodifluoromethane	40	83.246	210.670	2.53	68.71	2.91091	0.6557	1.909	0.696
717	Ammonia	40	73.3	211.96	2.89	480.33	0.416	4.0841	1.699	0.653
500	R12/152a (73.8/26.2)	40	60.722	155.790	2.57	60.54	3.30344	0.7920	2.616	0.692
12	Dichlorodifluoromethane	40	51.705	131.720	2.55	50.50	3.96024	0.7784	3.083	0.689
134a	Tetrafluoroethane	40	49.77	138.98	2.79	63.72	3.139	0.9522	2.989	0.679
124	Chlorotetrafluoroethane	40	27.890	80.920	2.90	52.06	3.84172	1.3180	5.063	0.698
600a	Isobutane	40	26.750	73.364	2.74	115.83	1.72667	3.2564	5.623	0.693
600	Butane	40	17.679	51.683	2.92	129.22	1.54775	4.9754	7.701	0.669
11	Trichlorofluoromethane	40	6.99	23.37	3.34	68.04	2.939	5.4546	16.031	0.624
123	Dichlorotrifluoroethane	40	5.79	20.8	3.59	62.82	3.184	5.9212	18.853	0.653
113	Trichlorotrifluoroethane	47	2.695	10.494	3.89	54.14	3.69433	10.7059	39.551	0.710

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Table 11-12
Comparison of Evaporator Temperature and Pressure for Common Refrigerants

	Pressure, psia														
Evaporator Temp., °F	−160	−140	−100	−80	−60	−40	−30	−20	−10	0	5	10	20	30	40
Ammonia			1.24	2.74	5.55	10.41	13.90	18.30	23.74	30.42	34.27	38.51	48.21	59.74	73.32
Ethane*				50.	78.	113.	135.8	160.	186.	220.	235.	260.	290.	338.	388.
Ethylene		22.4				210.4									
Propane				5.65	9.72	16.2	20.3	25.4	31.4	38.2	41.9	46.0	55.5	66.3	78.0
Propylene			3.85	7.21	12.6	20.6	25.8	32.1	39.4	47.9	52.6	57.7	68.9	81.7	96.3
N-Butane									5.67	7.30	8.2	9.2	11.6	14.4	17.7
Iso-Butane							6.08	7.50	9.28	11.6	13.1	14.6	18.2	22.3	26.9
+R-11								1.42	1.92	2.56	2.93	3.34	4.34	5.56	7.03
+R-12				2.88	5.36	9.3	11.99	15.26	19.19	23.84	26.48	29.35	35.73	43.14	51.67
+R-13	3.10	6.45	22.2	36.9	58.1	87.4	105.6	126.4	150.1	176.8		206.8	240.4	277.9	319.6
+R-21						1.36	1.89	2.58	3.46	4.58	5.24	5.97	7.69	9.79	12.32
+R-22		0.43	2.38	4.78	8.86	15.3	19.7	25.0	31.3	38.8	43.0	47.6	57.98	69.93	83.72
+R-113										0.84	0.98	1.14	1.53	2.03	2.65
+R-114						1.87	2.56	3.44	4.56	5.96	6.77	7.67	9.75	12.25	15.22
Methyl chloride				1.95	3.79	6.9	9.0	11.7	14.96	18.9	21.15	23.6	29.1	35.7	43.33
Methylene chloride												1.38	1.92	2.56	3.38
Sulfur dioxide						3.14	4.33	5.88	7.86	10.35	11.81	13.42	17.18	21.70	27.1
Carbon dioxide					94.7	145.8	177.9	215.0	257.4	305.7		360.4	421.8	490.6	567.3

All values from Section 32, *Air Conditioning Refrigerating Data Book*, 10th Ed., ASHRAE (1957) except

R-11 Trichloromonofluoromethane
R-12 Dichlorodifluoromethane
R-13 Monochlorotrifluoromethane
R-21 Dichloromonofluoromethane
R-22 Monochlorodifluoromethane
R-113 Trichlorotrifluoroethane
R-114 Dichlorotetrafluoroethane

* Approximate value.
+ These refrigerants already have been or are being phased out of industrial usage; therefore, the values have no current significance.
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Table 11-13
Physical Property Study of Various Refrigerants

Datum Plane: 0°F Suction, 110°F Condensing[a]

Refrigerants	(A) Pressure in psia 0°F	(A) Pressure in psia 110° F	(B) Boiling Point at 14.7 psia, °F	(C) Compression Ratio	(D) Molecular Weight	(E) Adiabatic Head, ft[b]	(F) Flow Rate lb/min/ton	(G) Flow Rate cfm/TR	(H) Approx. Min. Tons, Centrifugal[c]	(I) Approx. No. of Stages Centrifugal	(J) Min. Recommended Sat. Suction Temp. on Single-Stage Reciprocating[c]	(K) Min. Recommended Sat. Suction Temp. on Multi-Stage Reciprocating/Centrifugal[c]
Ammonia	30.42	247	−28	8.12	17.03	107,800	.446	4.07	700	11	− 3	− 90
Propylene	48	258	−53.86	5.48	42.08	48,600	1.825	4.11	600	6	−20	−125
Propane	38.2	212	−43.73	5.55	44.09	25,100	1.91	5.17	400	4	−20	−125
Refrigerant 22	38.8	243.4	−41.4	6.29	86.48	14,800	3.3	4.54	400	3	−16	−125
Refrigerant 12	23.8	151.1	−21.6	6.34	120.93	11,050	4.57	7.35	250	3	−15	−125
Refrigerant 114	5.96	54.4	38.4	9.12	170.9	11,900	5.46	26.0	100	3	e	0
Refrigerant 11	2.55	28.1	74.8	11.1	137.38	13,200	3.24	45.2	50	3	e	20
Refrigerant 113	0.838	12.76	117.6	15.2	187.39	12,850	4.2	131.4	20	3	e	40

Note: [a]Based on no intermediate flashing or subcooling of liquid.
[b]For approximation, polytropic head used in actual design.
[c]Based on 2,000–3,000 cfm at suction conditions as minimum for efficient selection. This is approximate only. Columns (J) and (K) are not referring to the 0°F and 110°F conditions.
[d]Refrigerant number of column one corresponds to the A.S.R.E. standard designation, which agrees with previous designations for the chloro-fluoro hydrocarbon type refrigerants.
[e]Generally not recommended for reciprocating compressors.

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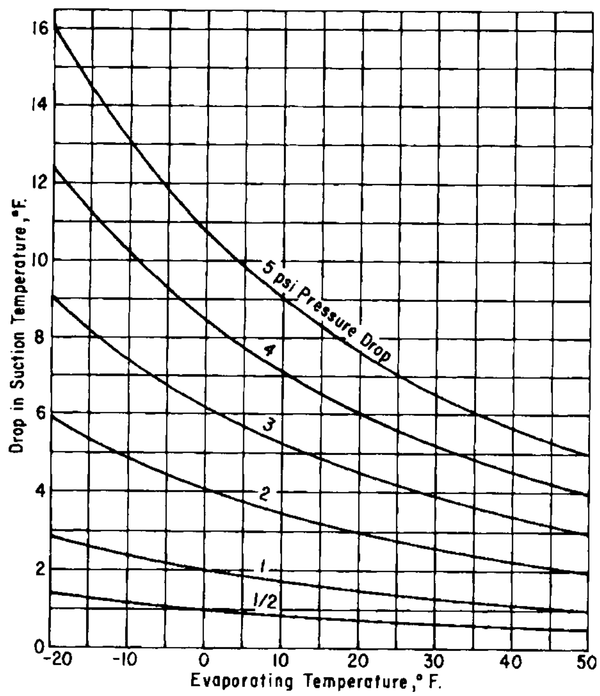

Figure 11-25. Piping practices—Freon; Freon-12 line sizing. (Used by permission: Dresser-Rand Company.)

(Text continued from page 328)

of one component, say propane at 90 vol% and 10% butane, it is not possible to assume that the thermodynamic data for the propane will be satisfactory for a "single" component design of the system. It just won't work.

Example 11-5. Use of Hydrocarbon Mixtures as Refrigerants[29] (Used by Permission of the Carrier Corporation.)

A mixture of propane and butane is to be used as a refrigerant and charged to the system as a liquid. From the specification listing that follows determine the evaporator, condenser, and compressor for this application. This refrigerant mixture requires the use of Mollier Diagrams for propane and butane.

Thus, if a gas mixture exerts 100 psia total pressure and is composed of 20% by volume (mol%) propane and 80% by volume butane, the partial pressures are 20 and 80 psia for propane and butane, respectively. The liquid in equilibrium with this mixture of vapors would have a lower percentage of propane and a higher percentage of butane. If this mixture is used as a refrigerant, the low-boiling component (propane) reaches equilibrium with a higher concentration in the condenser (as liquid) and increases the total pressure in the condenser. This requires more head and more horsepower at the compressor.

The problem is difficult because when the system is in equilibrium, the composition of the initial liquid charge in the evaporator has changed, and the composition of the vapor in equilibrium with it is not known. Vapor and liquid compositions in the condenser are also not known, but three important facts can be established:

1. By material balance, the composition of the vapor entering the condenser is the same composition as the liquid leaving the condenser (with no bleed off).
2. The condensed liquid at the top of the condenser is in equilibrium with the vapor composition entering the condenser, which is also the composition of the vapor leaving the evaporator.
3. The mixture (liquid and vapor) in the total system must have the same overall composition as the initial charge.

The problem is one of trial and error and illustrates the procedure.

Liquid and Vapor Equilibrium

See Chapter 12 for Mollier Diagrams.

Given

$$\text{Initial charge} = \left.\begin{array}{l}79 \text{ mol\% propane}\\ 21 \text{ mol\% butane}\end{array}\right\} \text{ as a liquid}$$

Average cond. temp. = 100°F
Average evap. temp. = 0°F
Load = 1,000 tons of refrigeration

Required

Evaporator and condenser pressure, composition of compressed vapor, size of compressor, and weight of charge required.

Solution

Step 1. Assume the composition of the liquid in the evaporator at equilibrium with its vapor to be 75 mol% propane and 25 mol% butane. This is the initial assumption. If it is correct, the composition of the initial charge can be checked. If it is not correct, the problem must be reworked with a new equilibrium assumption. The composition of the vapor in equilibrium with this liquid is determined from the following equation.

$$Y_e = KX_e$$

where Y_e = mol fraction of one component in the evaporator vapor.
K = an equilibrium constant.
X_e = mol fraction of the same component in the liquid in the evaporator.

(Text continues on page 336)

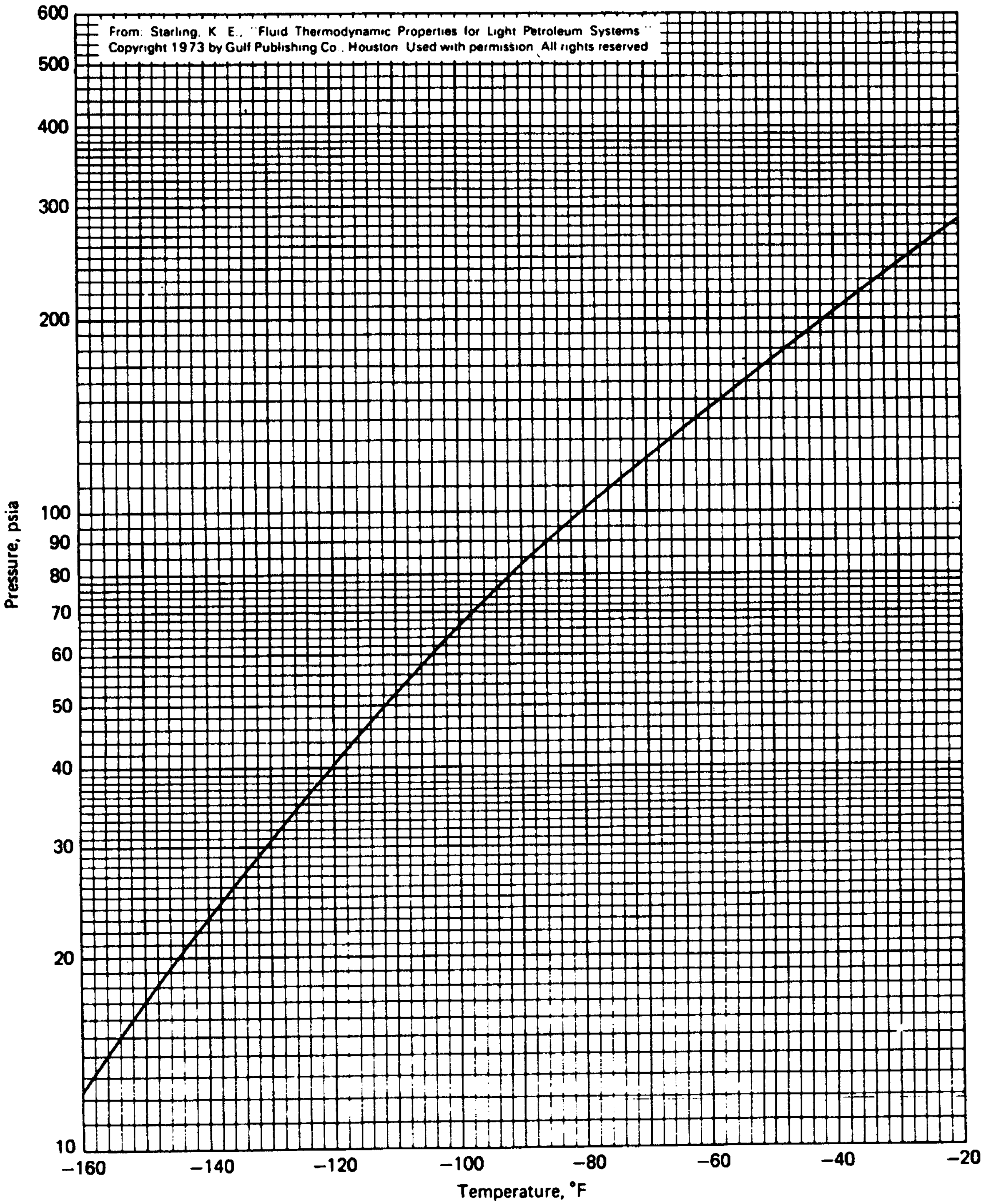

Figure 11-26. Vapor pressure curve for ethylene refrigerant. (Used by permission: Starling, K. E. *Fluid Thermodynamic Properties for Light Petroleum Systems,* ©1973. Gulf Publishing Co., Houston, Texas. All rights reserved.)

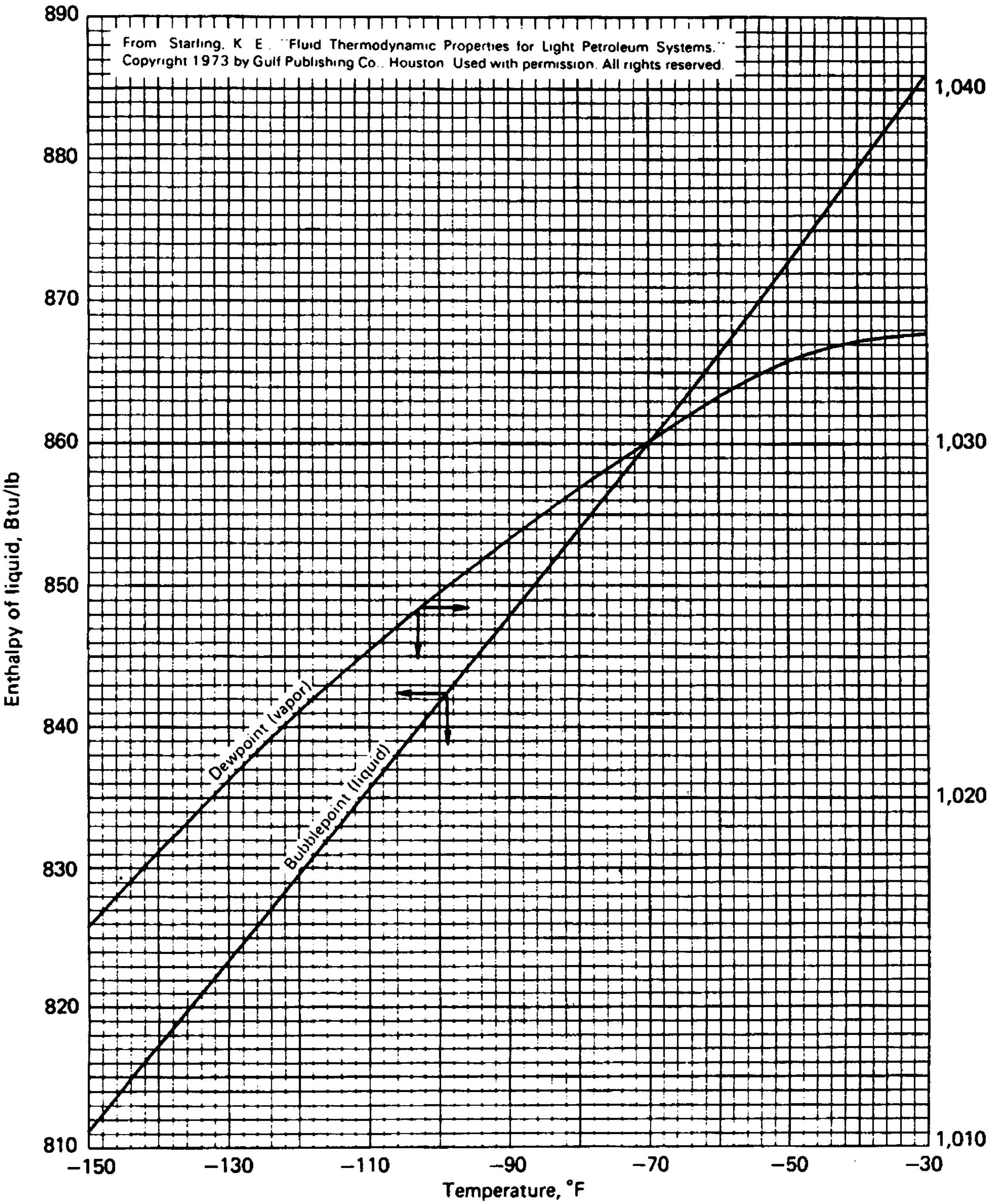

Figure 11-27. Enthalpy of ethylene for liquid and vapor. (Used by permission: Starling, K. E. *Fluid Thermodynamic Properties for Light Petroleum Systems,* ©1973. Gulf Publishing Co., Houston, Texas. All rights reserved.)

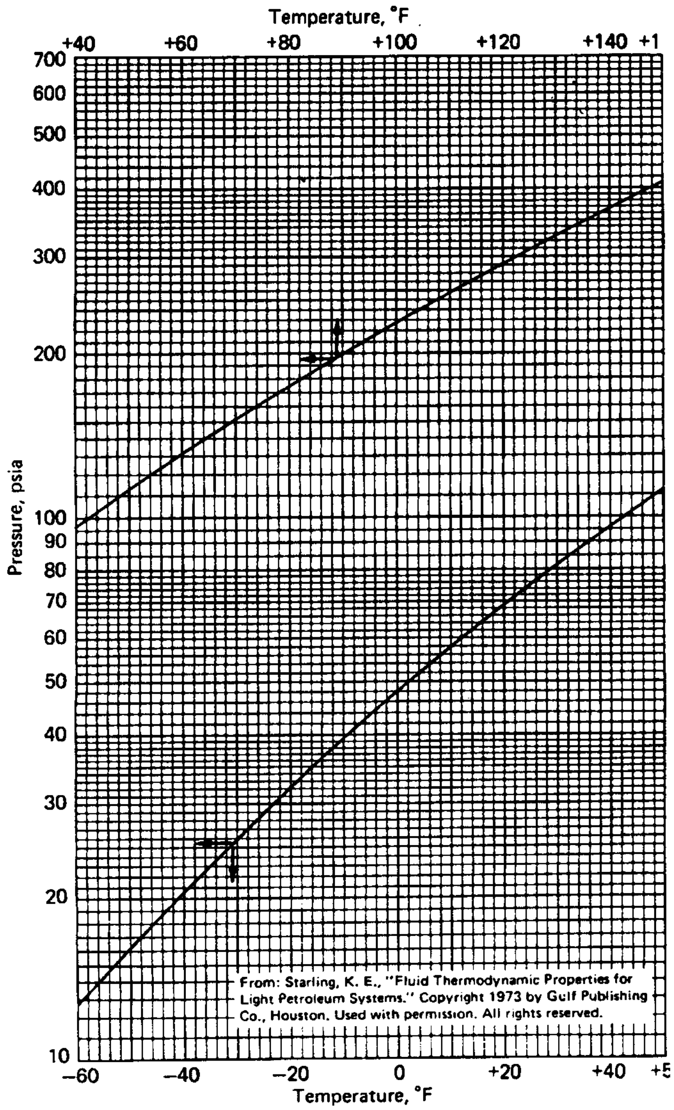

Figure 11-28. Vapor pressure curve for propylene. (Used by permission: Starling, K. E. *Fluid Thermodynamic Properties for Light Petroleum Systems,* ©1973. Gulf Publishing Co., Houston, Texas. All rights reserved.)

(Text continued from page 333)

K versus pressure and temperature for various hydrocarbons can be found in the *Engineering Data Book* of the Natural Gasoline Supply Men's Association, Latest Edition.

Step 2. Determine the vapor pressure in the evaporator. According to the phase rule, for a mixture of two components (propane and butane) it is necessary to establish two variables of the liquid-vapor system in the evaporator to completely define the system and fix the value of all other variables. The assumed liquid mol fraction and a temperature of 0°F is known. The corresponding pressure in the evaporator is therefore fixed. Because the sum of all the Y_e mol fractions must add up to 1,000, assume pressures until the $\Sigma Y_e = 1{,}000$ to determine the correct pressure.

Gas	Constant X_e	Trial 1 (p = 30 psia) K	Trial 1 Y_e	Trial 2 (p = 35 psia) K	Trial 2 Y_e	Trial 3 (p = 32 psia) K	Trial 3 Y_e
Propane	0.75	1.32	0.990	1.15	0.863	1.25	0.938
Butane	0.25	0.263	0.066	0.23	0.0575	0.25	0.062
			1.056		0.9205		1.000

Therefore the pressure in the evaporator is 32 psia, and the vapor composition is 93.8% propane and 6.2% butane. This is the vapor composition that the compressor must handle (if the original assumption holds true).

Step 3a. The calculations to determine condenser pressure are handled somewhat similarly to those of the evaporator except that they must be divided into two parts.

The vapor composition at the top of the condenser (Y_{c1}) is different from that at the bottom (Y_{c2}). The condenser may be compared to a fractional distillation problem in reverse. Butane, having a higher boiling point, will condense out *faster* than the propane, although both are condensing at the same time. Thus, the vapor and liquid mol fractions from the top to the bottom of the condenser tube bundle are always changing. Proceed as follows: The vapor at the top has the same composition as the gas leaving the evaporator. Therefore, $Y_{c1} = Y_e$.

The composition of the liquid in equilibrium with this vapor is calculated from

$$X_{c1} = \frac{Y_{c1}}{K}$$

where X_{c1} is the composition of the liquid being condensed at the top.

The objective now is to assume a condenser pressure that will yield an average condenser temperature of 100°F between the top and bottom sections. Knowing the inlet vapor composition and pressure, the temperature may be determined and the system in the condenser completely defined. ΣX_{c1} must = 1.000 when the correct temperature is assumed. The correct pressure assumption will not be known until the upper and lower temperatures have been averaged.

Assume condensing pressure = 180 psia.

(Text continues on page 54)

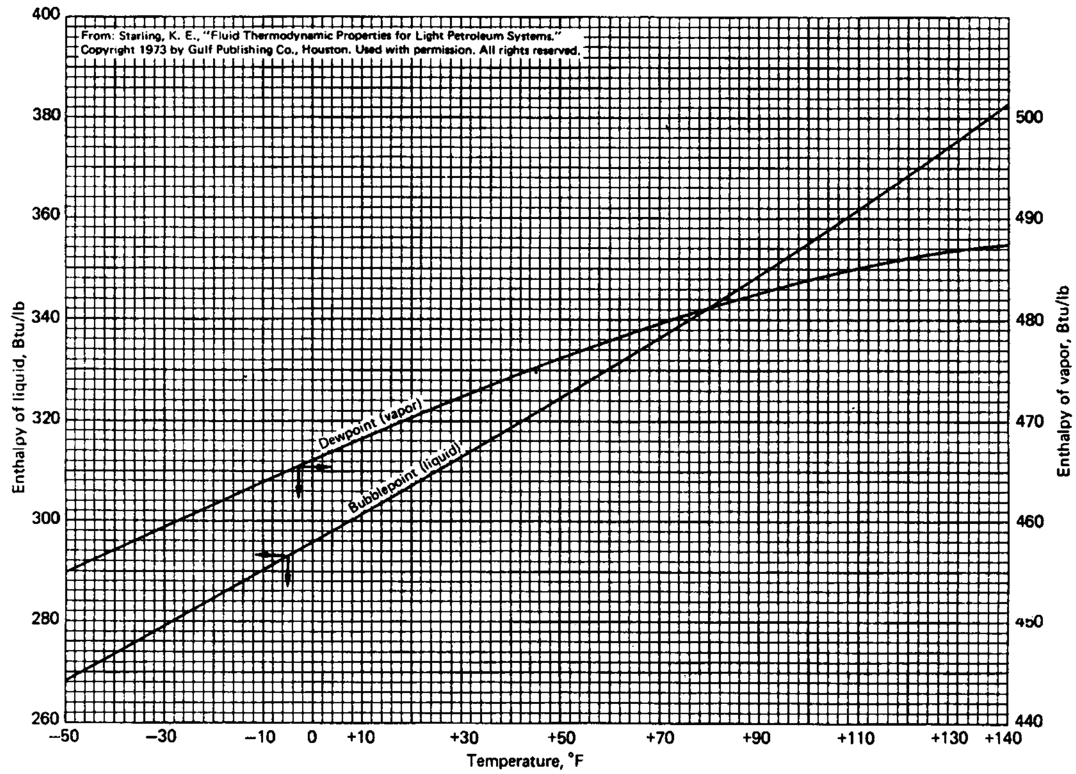

Figure 11-29. Enthalpies of propylene for liquid and vapor. (Used by permission: Starling, K. E. *Fluid Thermodynamic Properties for Light Petroleum Systems*, ©1973. Gulf Publishing Co., Houston, Texas. All rights reserved.)

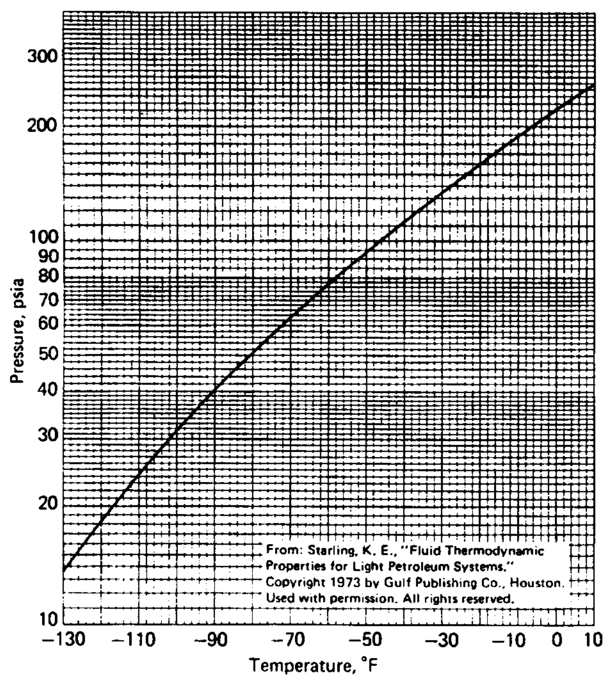

Figure 11-30. Vapor pressure curve for ethane refrigerant. (Used by permission: Starling, K. E. *Fluid Thermodynamic Properties for Light Petroleum Systems*, ©1973. Gulf Publishing Co., Houston, Texas. All rights reserved.)

Gas	Constant Y_{c1}	Trial 1 (T = 100°F) K	Trial 1 X_{c1}	Trial 2 (T = 104°F) K	Trial 2 X_{c1}	Trial 3 (T = 105°F) K	Trial 3 X_{c1}
Propane	0.938	1.06	0.885	1.10	0.853	1.11	0.845
Butane	0.062	0.38	0.163	0.40	.155	0.41	0.151
			1.048		1.008		.996

By interpolating between Trial 2 and 3, the composition of the liquid is found to be 84.8% propane and 15.2% butane, and the corresponding temperature = 104.7°F.

Step 3b. The composition of the liquid leaving the condenser (X_{c2}) is equal to the composition of the vapor leaving the evaporator. Therefore,

$$X_{c2} = Y_e$$

The vapor in equilibrium with this liquid has the following composition:

$$Y_{c2} = KX_{c2}$$

(*Text continues on page 342*)

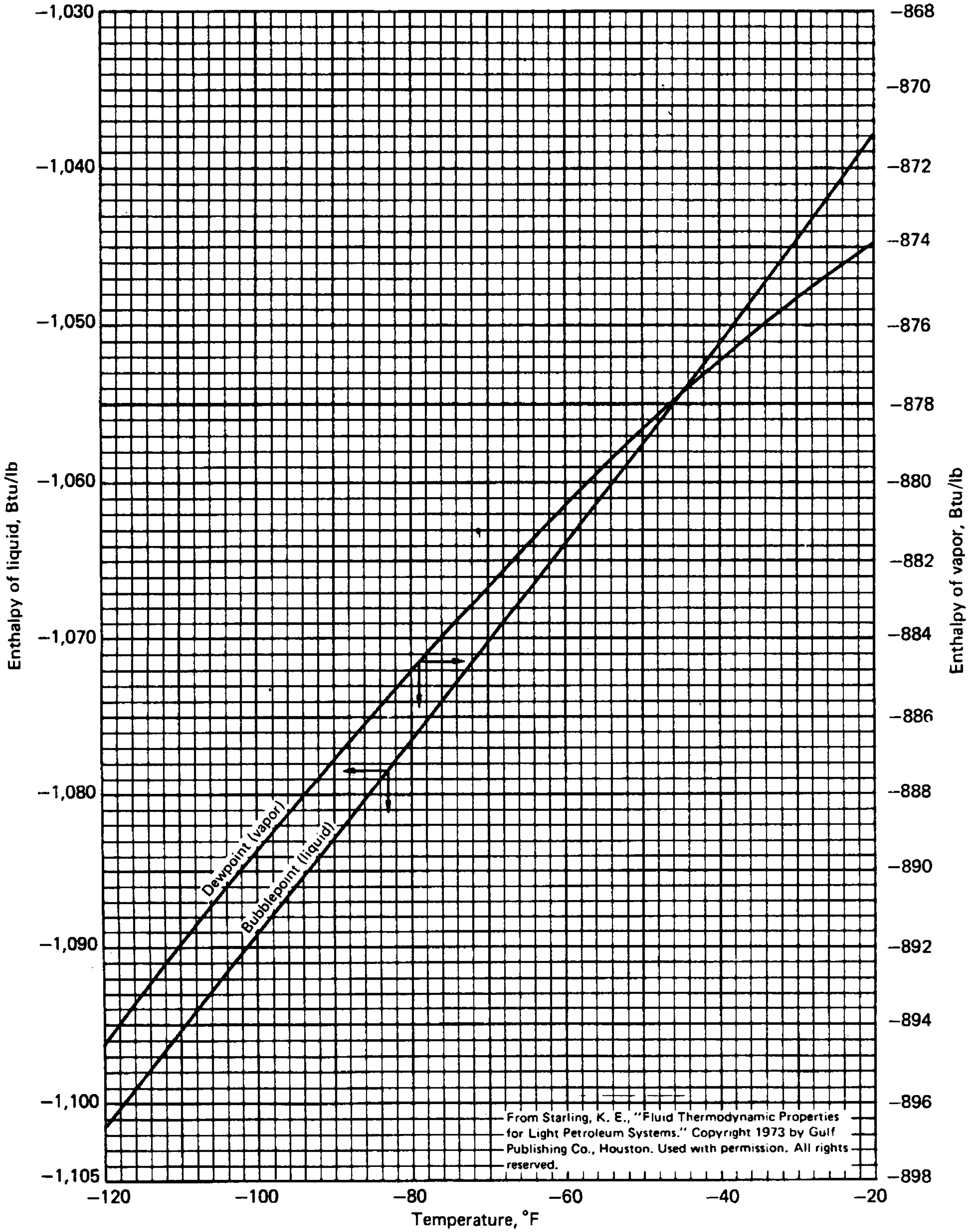

Figure 11-31. Enthalpies of ethane for liquid and vapor. (Used by permission: Starling, K. E. *Fluid Thermodynamic Properties for Light Petroleum Systems,* ©1973. Gulf Publishing Co., Houston, Texas. All rights reserved.)

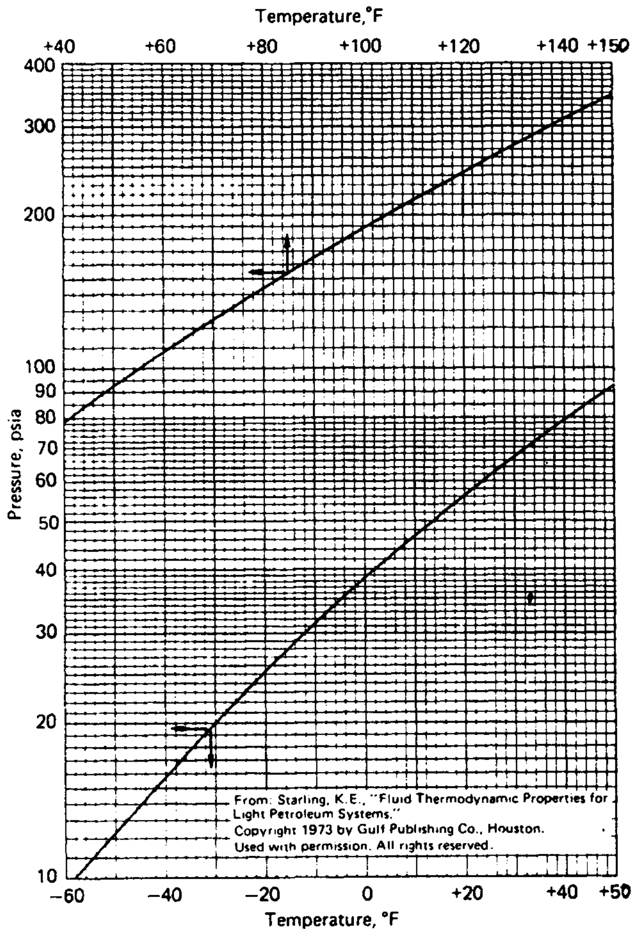

Figure 11-32. Vapor pressure curve for propane. (Used by permission: Starling, K. E. *Fluid Thermodynamic Properties for Light Petroleum Systems,* ©1973. Gulf Publishing Co., Houston, Texas. All rights reserved.)

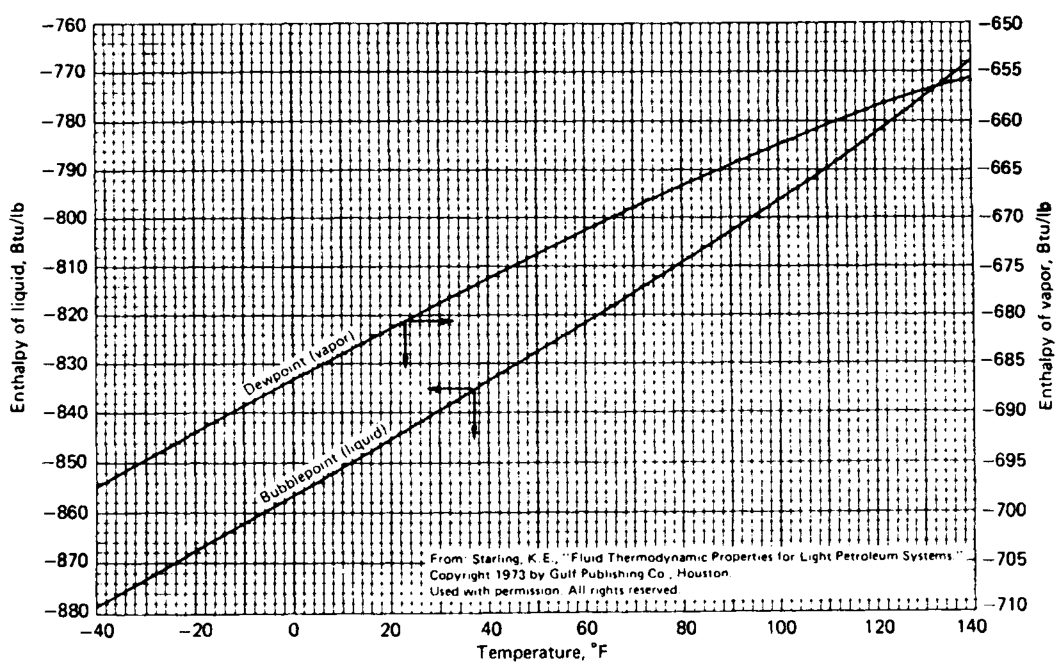

Figure 11-33. Enthalpies of propane for liquid and vapor. (Used by permission: Starling, K. E. *Fluid Thermodynamic Properties for Light Petroleum Systems,* ©1973. Gulf Publishing Co., Houston, Texas. All rights reserved.)

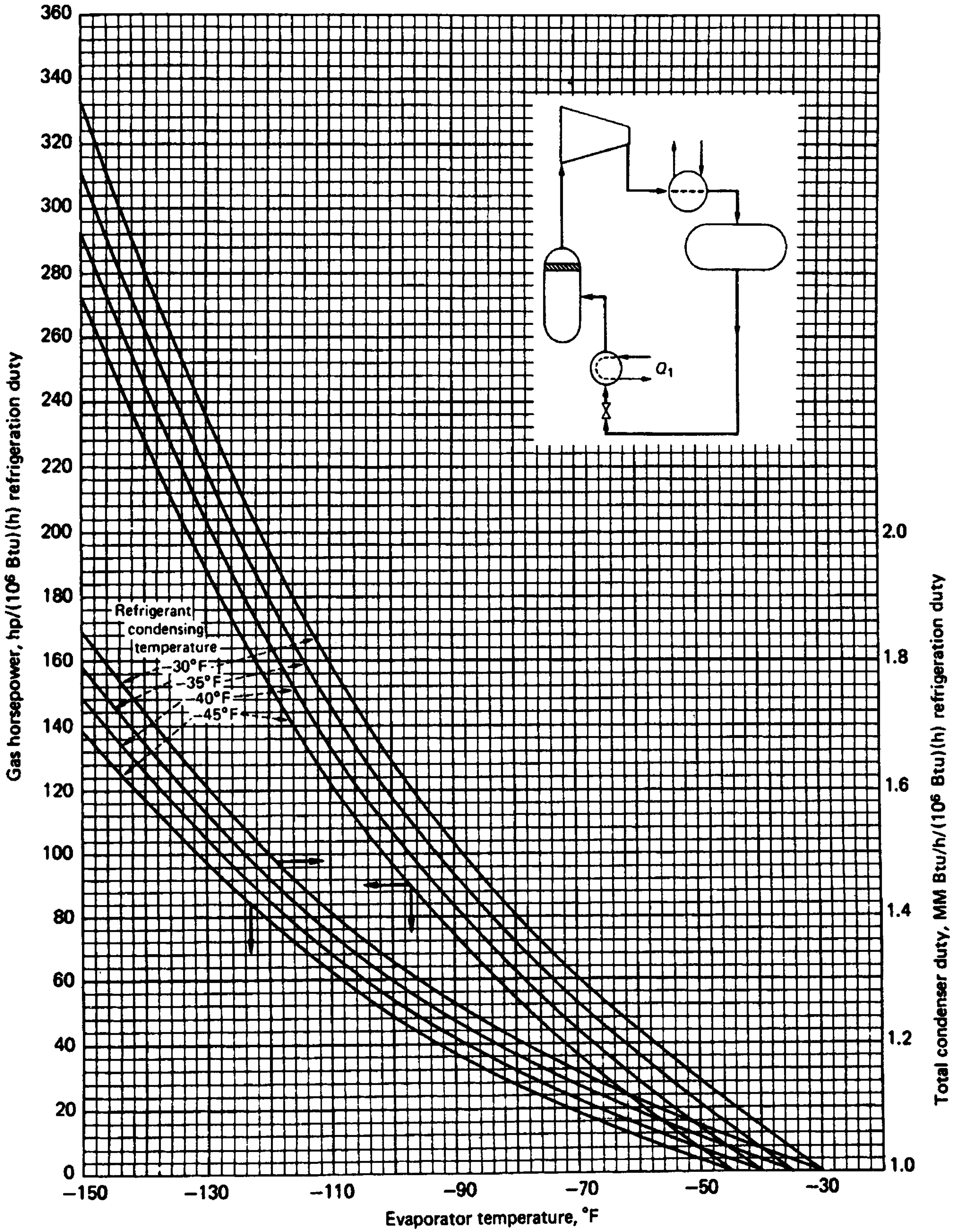

Figure 11-34. Single-stage ethylene refrigeration system. (Excerpted by special permission: Mehra, Y. R. *Chemical Engineering,* Dec. 15, 1978. ©McGraw-Hill, Inc., New York. All rights reserved.)

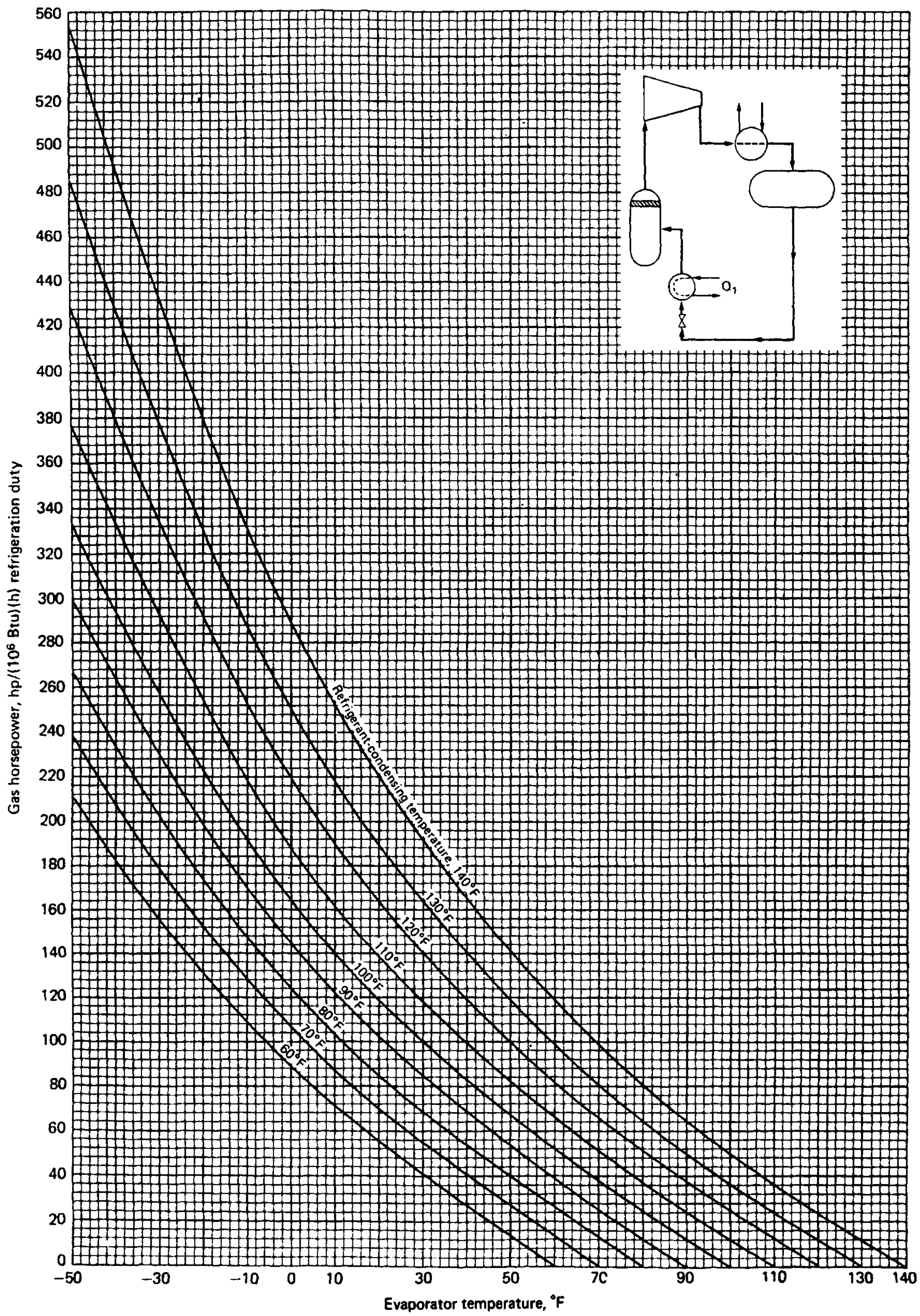

Figure 11-35. Gas horsepower for single-stage propylene refrigeration system. (Excerpted by special permission: Mehra, Y. R. *Chemical Engineering,* Jan. 15, 1979. ©McGraw-Hill, Inc., New York. All rights reserved.)

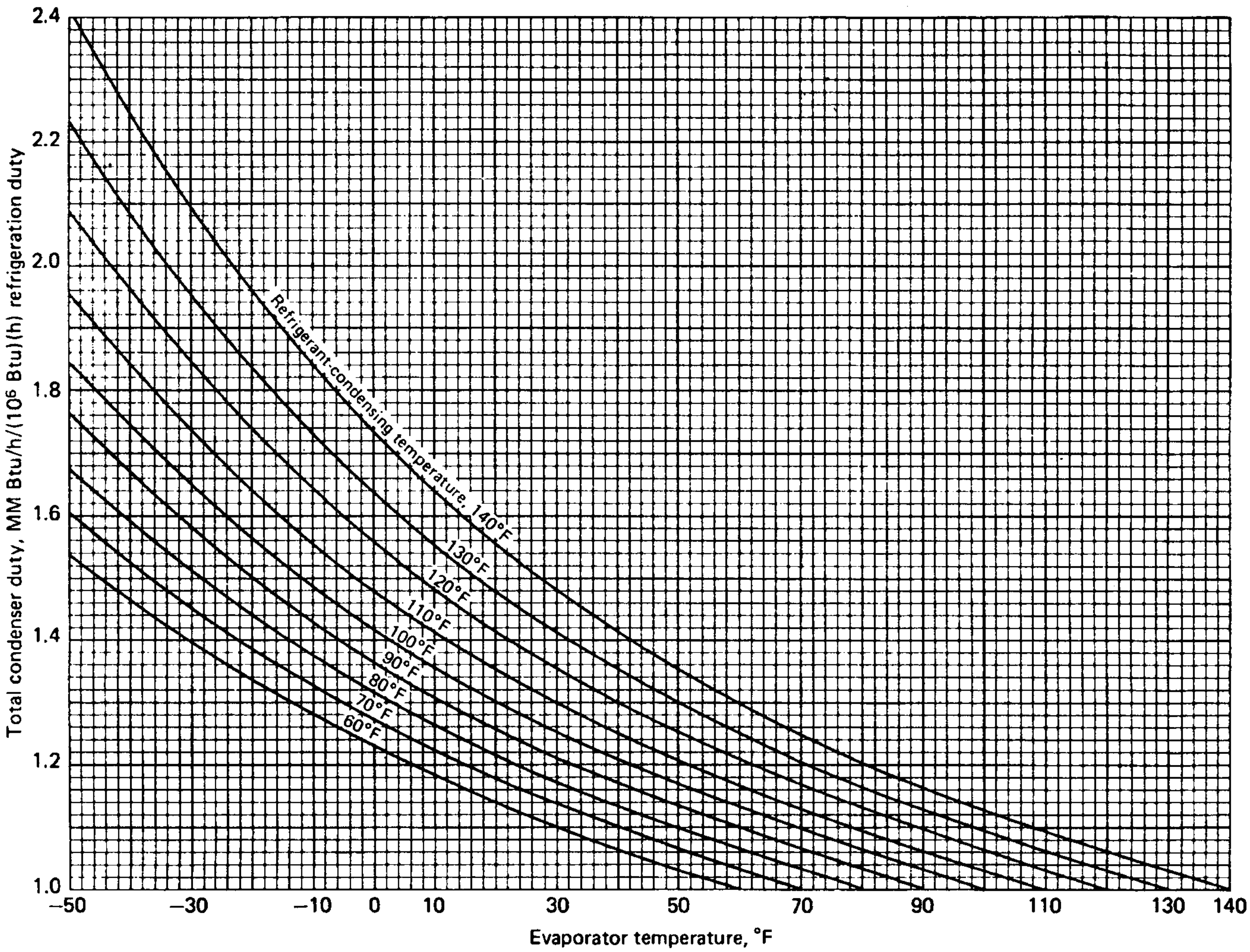

Figure 11-36. Condenser duty for single-stage propylene refrigeration system. (Excerpted by special permission: Mehra, Y. R. *Chemical Engineering,* Jan. 15, 1979. ©McGraw-Hill, Inc., New York. All rights reserved.)

(Text continued from page 337)

Again using trial and error, find the correct temperature for the bottom of the condenser, knowing that $\Sigma Y_{c2} = 1.000$.

	Constant	Trial 1 (T = 100°F)		Trial 2 (T = 98°F)	
Gas	X_{c2}	K	Y_{c2}	K	Y_{c2}
Propane	0.938	1.06	0.995	1.04	0.975
Butane	0.062	0.38	0.0235	0.37	.023
			1.0185		0.998

The composition of the vapor at the bottom of the condenser is, by interpolation, 97.7% propane, 2.3% butane, and the temperature = 98.2°F.

	Temperature, °F	Propane, Mol Frac.	Butane, Mol Frac.
Top	104.7	0.938	0.062
Bottom	98.2	0.977	0.023
Average	101.5	0.957	0.043

The average temperature being 101.5°F (close enough) means that the assumed pressure of 180 psia holds.

Step 4. To check the original assumed equilibrium composition in the evaporator, the total amount of propane and butane in the system must be determined. This total must be equal to the original charge. Therefore, the next step will be to calculate the volume of the total system. These calculations must necessarily be somewhat approximate because the exact installation conditions are not known.

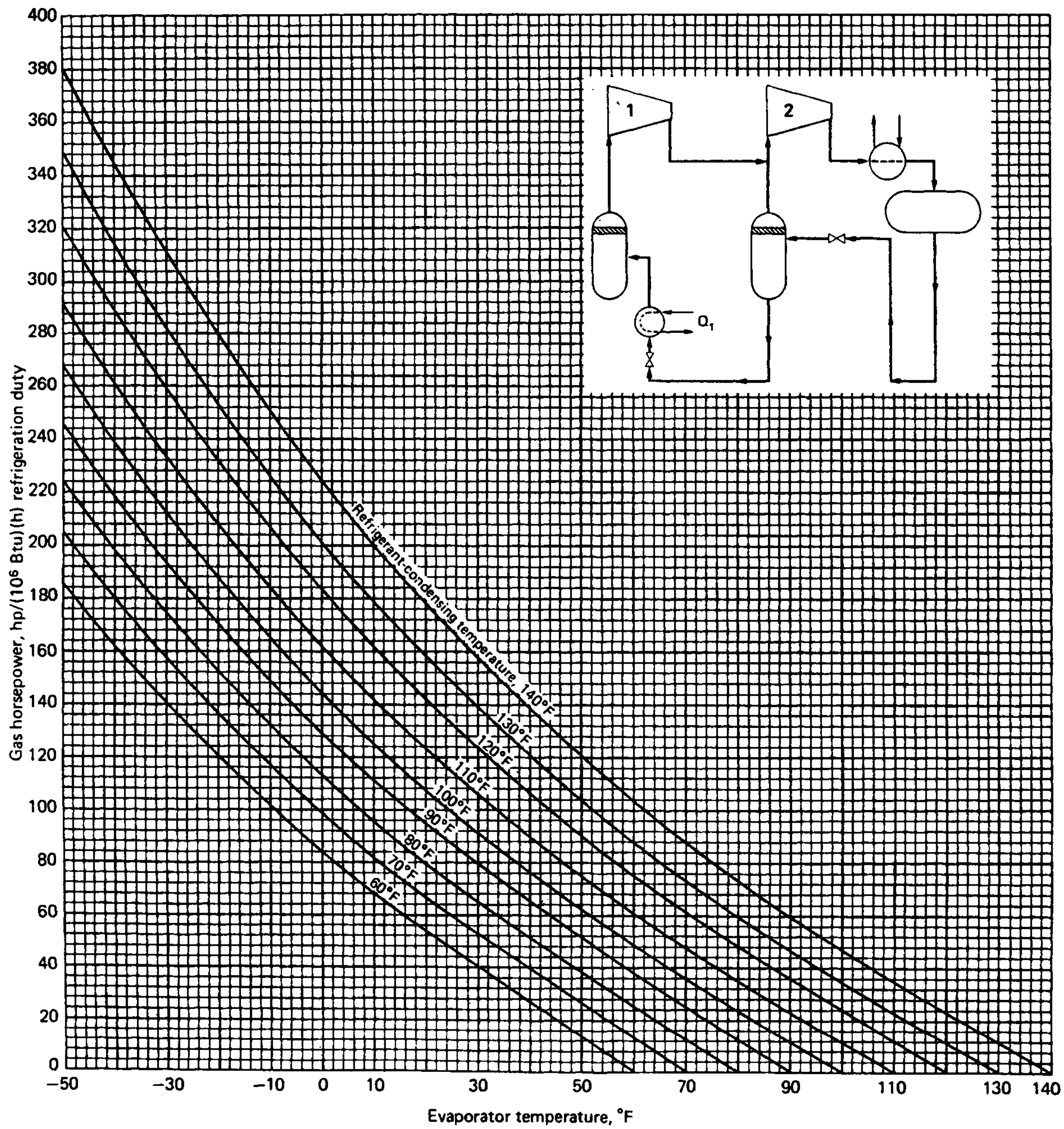

Figure 11-37. Gas horsepower for two-stage propylene refrigeration system. (Excerpted by special permission: Mehra, Y. R. *Chemical Engineering,* Jan. 15, 1979. ©McGraw-Hill, Inc., New York. All rights reserved.)

Step 4a. Evaporator volume:

The load = q = 1,000 tons = 12,000,000 Btu/hr.
Assume an overall heat transfer coefficient.
U = 200 Btu/(hr) (ft^2)(°F) for brine cooling.
Also assume Δt = 15°F.

$$\text{Therefore: } A = \frac{q}{\Delta t\, U} = \frac{12{,}000{,}000}{15 \times 200} = 4{,}000 \text{ ft}^2$$

Now assume $^3/_4$ in. O.D. tubing, 16 ft long.

$$\text{No. of tubes } = \frac{4{,}000}{16 \times 0.196} = 1{,}280$$

The approximate tube bundle diameter for tubes layed out on triangular pitch can be calculated from

(Text continues on page 348)

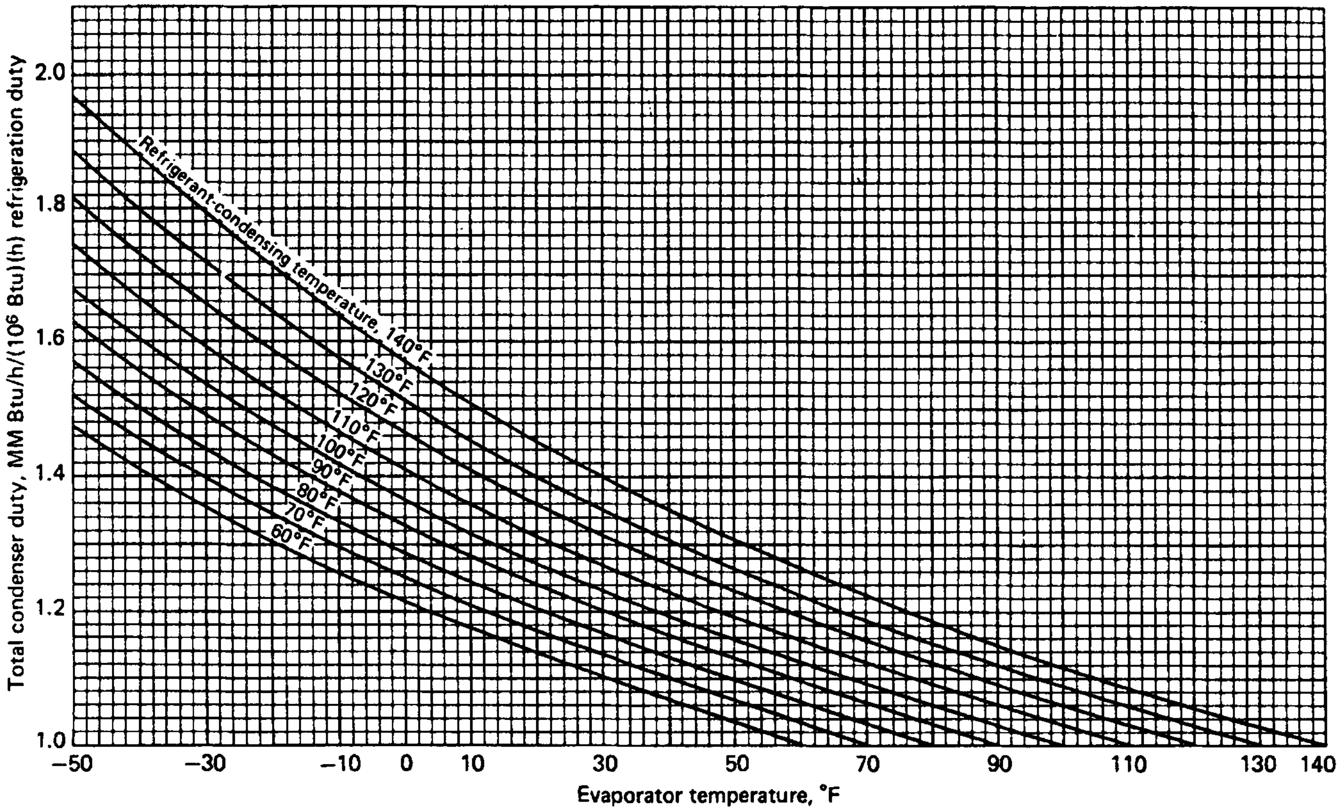

Figure 11-38. Condenser duty for two-stage propylene refrigeration system. (Excerpted by special permission: Mehra, Y. R. *Chemical Engineering,* Jan 15, 1979. ©McGraw-Hill, Inc., New York. All rights reserved.)

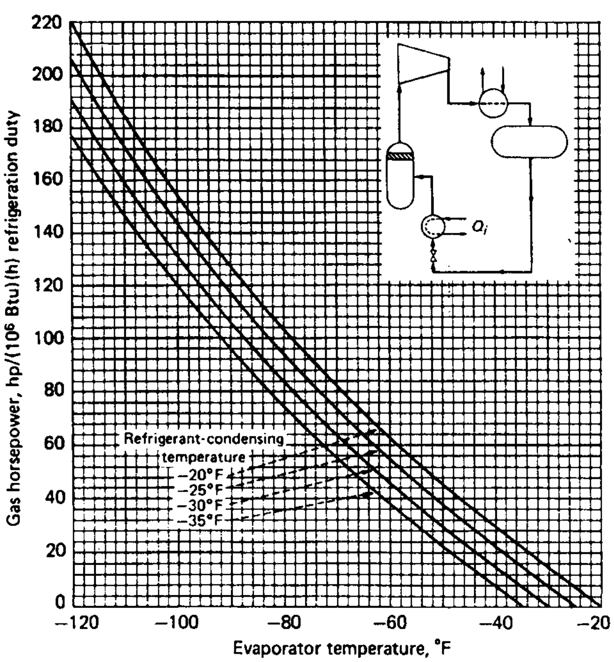

Figure 11-39. Gas horsepower for single-stage ethane refrigeration system. (Excerpted by special permission: Mehra, Y. R. *Chemical Engineering,* Feb. 12, 1979. ©McGraw-Hill, Inc., New York. All rights reserved.)

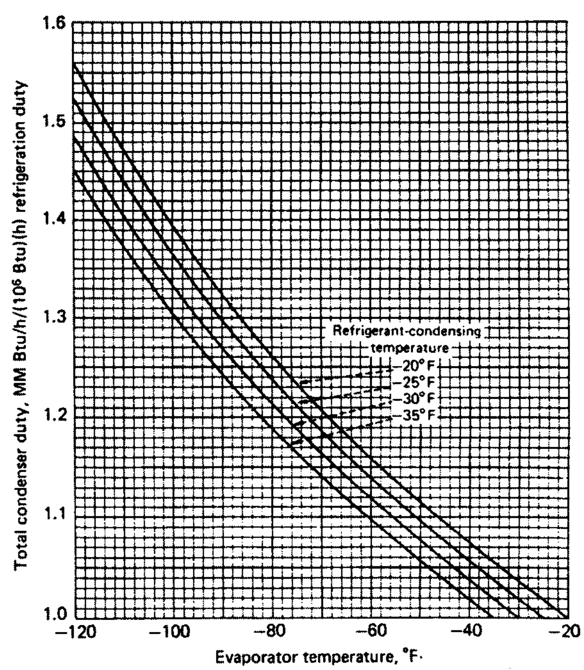

Figure 11-40. Condenser duty for single-stage ethane refrigeration system. (Excerpted by special permission: Mehra, Y. R. *Chemical Engineering,* Feb. 12, 1979. ©McGraw-Hill, Inc., New York. All rights reserved.)

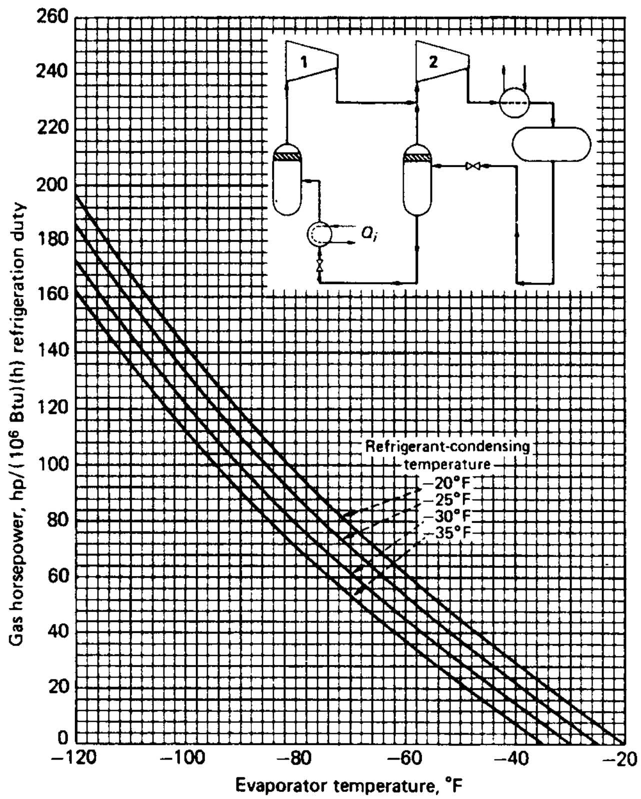

Figure 11-41. Gas horsepower for two-stage ethane refrigeration system. (Excerpted by special permission: Mehra, Y. R. *Chemical Engineering*, Feb. 12, 1979. ©McGraw-Hill, Inc., New York. All rights reserved.)

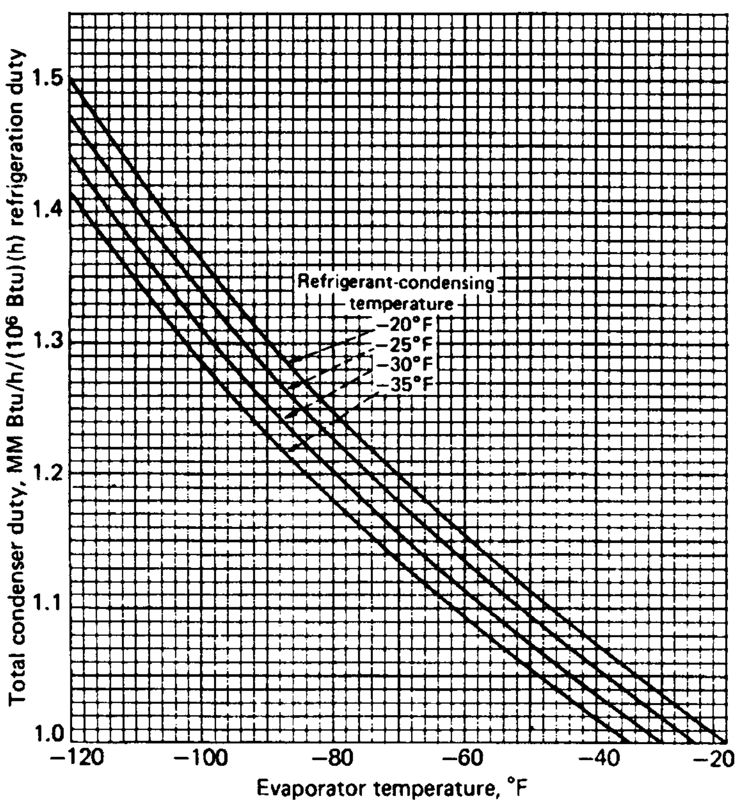

Figure 11-42. Condenser duty for two-stage ethane refrigeration system. (Excerpted by special permission: Mehra, Y. R. *Chemical Engineering*, Feb. 12, 1979. ©McGraw-Hill, Inc., New York. All rights reserved.)

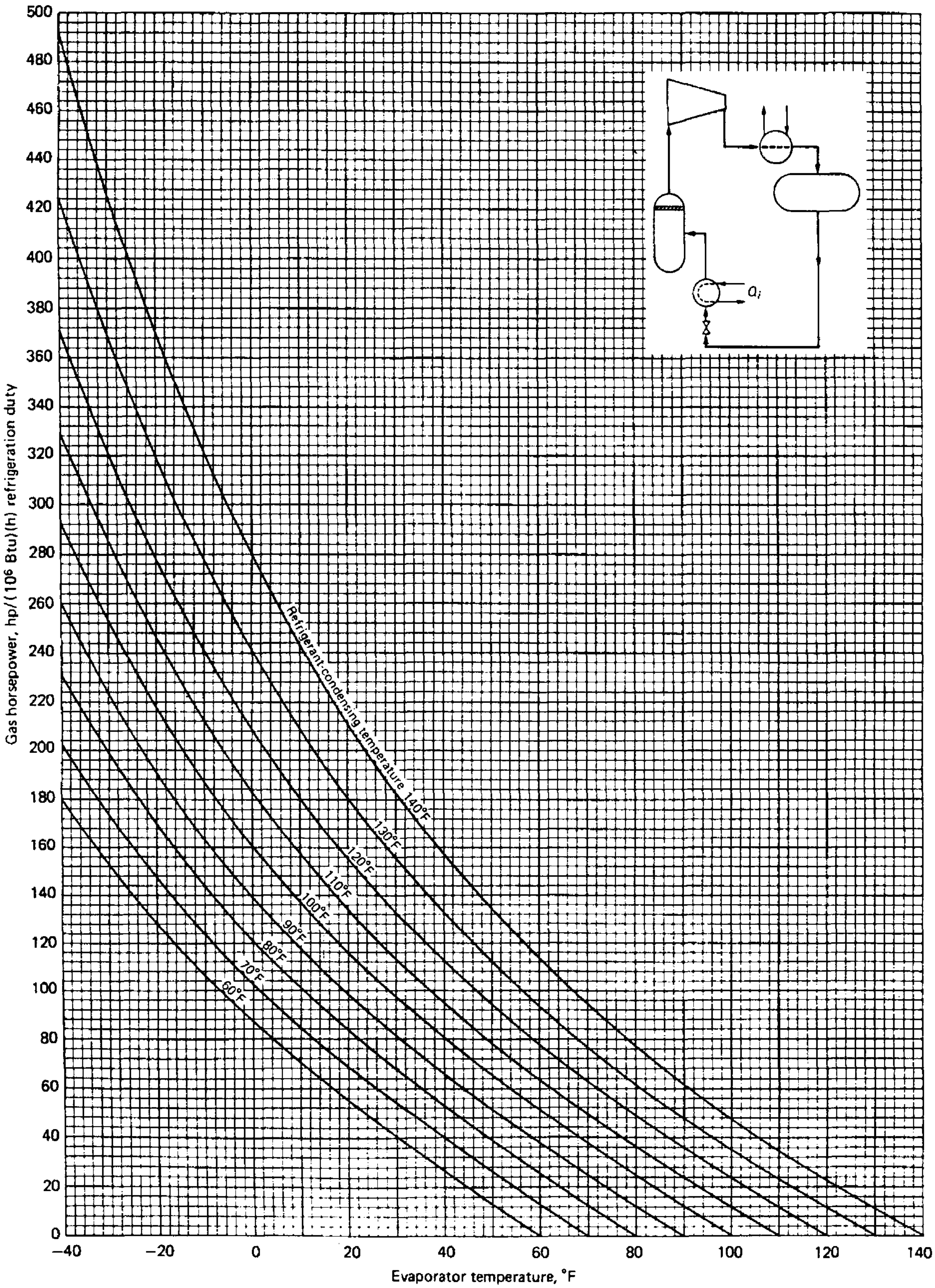

Figure 11-43. Gas horsepower for single-stage propane refrigeration system. (Excerpted by special permission: Mehra, Y. R. *Chemical Engineering*, March 26, 1979. ©McGraw-Hill, Inc., New York. All rights reserved.)

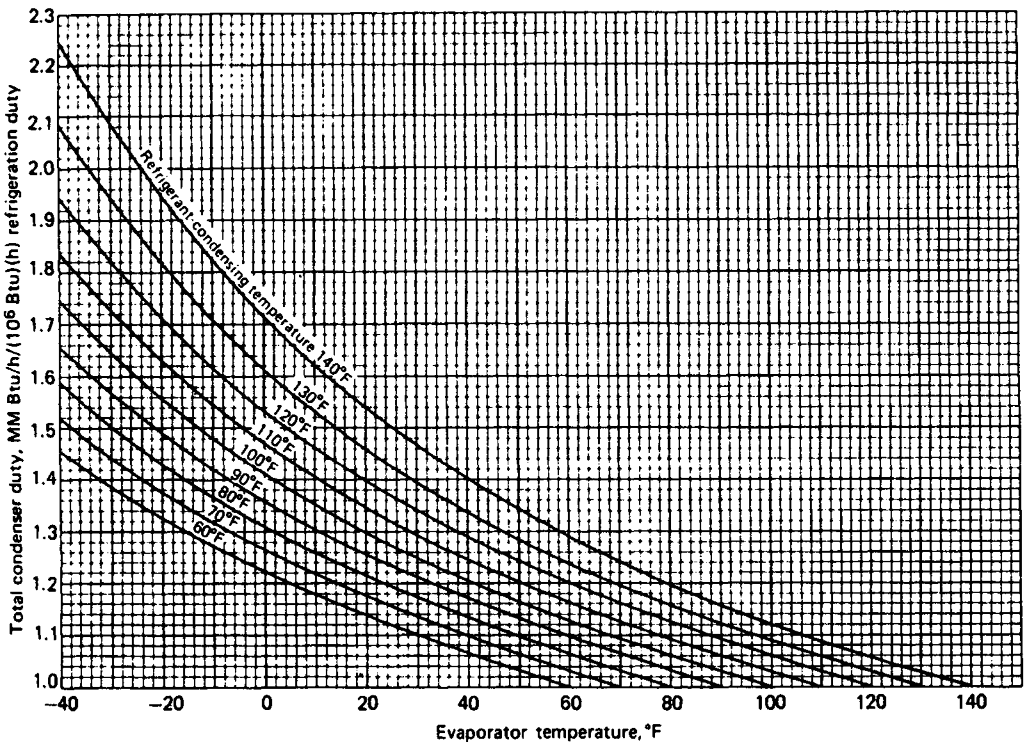

Figure 11-44. Condenser duty for single-stage propane refrigeration system. (Excerpted by special permission: Mehra, Y. R. *Chemical Engineering* March 26, 1979. ©McGraw-Hill, Inc., New York. All rights reserved.)

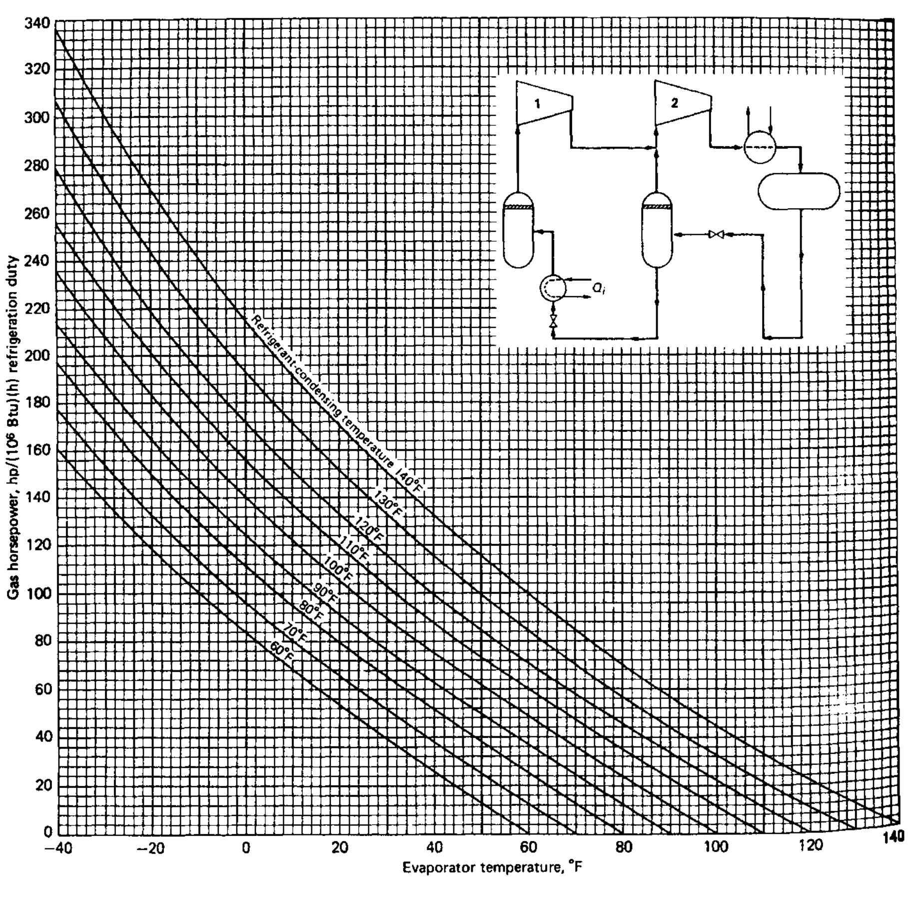

Figure 11-45. Gas horsepower for two-stage propane refrigeration system. (Excerpted by special permission: Mehra, Y. R. *Chemical Engineering,* March 26, 1979. ©McGraw-Hill, Inc., New York. All rights reserved.)

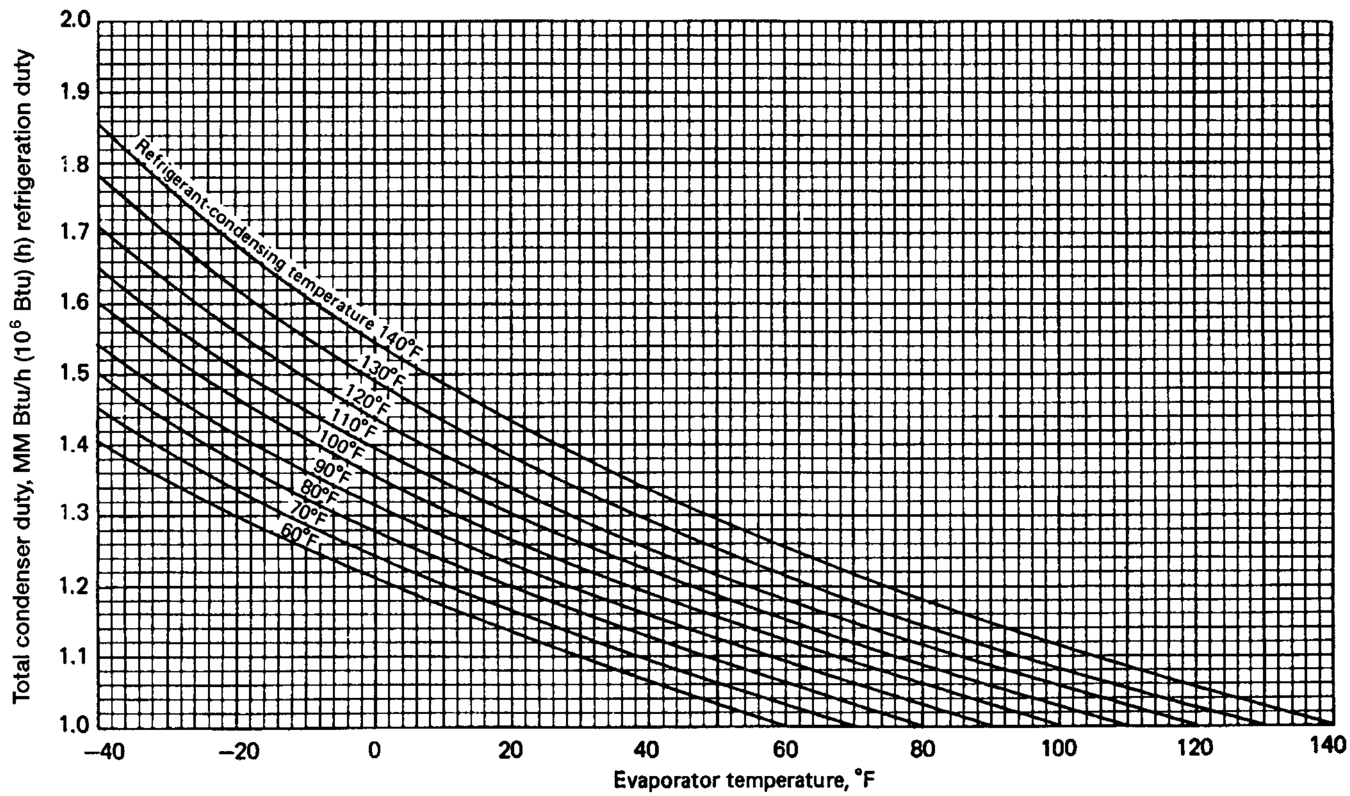

Figure 11-46. Condenser duty for two-stage propane refrigeration system. (Excerpted by special permission: Mehra, Y. R. *Chemical Engineering,* March 26, 1979. ©McGraw-Hill, Inc., New York. All rights reserved.)

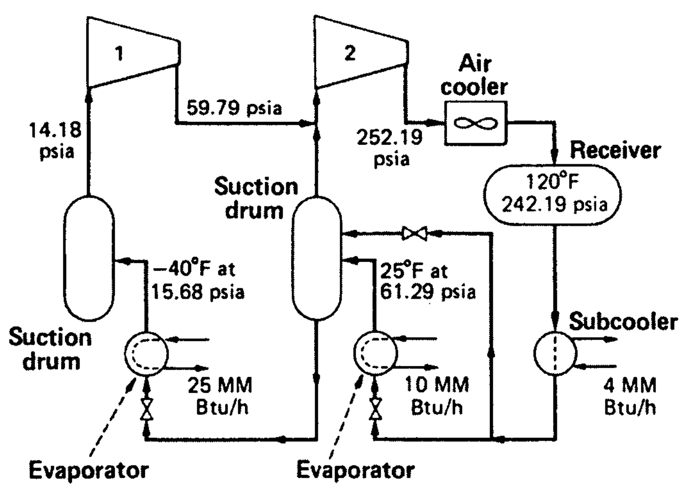

Figure 11-47. Two-stage propane system for Example 11-4. (Excerpted by special permission: Mehra, Y. R. *Chemical Engineering,* March 26, 1979. ©McGraw-Hill, Inc., New York. All rights reserved.)

(Text continued from page 343)

1.15 (tube pitch) $\sqrt{\text{No. tubes}}$

Tube pitch is usually 1.25 × tube diameter.
Therefore, outside tube bundle diameter =

$$1.15 \times 1.25 \times 0.75\sqrt{1{,}280} = 38.5 \text{ in. or } 3.21 \text{ ft}$$

The corresponding area = $^1/_4\,\pi(3.21)^2 = 8.1$ ft^2
Assume 30% extra for vapor zone.
Therefore, $A_E = 8.1 \times 1.3 = 10.5$ ft^2 (say 44 in. I.D.)
Volume of bundle = $8.1 \times 16 = 129.6$ ft^3

Volume of tubes = $1{,}280 \times 16 \times 0.0037$ = $\underline{63.0 \text{ ft}^3}$
Volume of boiling liquid = difference: = 66.6 ft^3
Assume bubbles take up 50% of volume.
Therefore, volume of liquid = 33.3 ft^3
Volume of vapor = $(10.5 - 8.1) \times 16 + 33.3 = 71.7$ ft^3

Step 4b. Condenser volume:

The load = $1.25 \times 12{,}000{,}000 = 15{,}000{,}000$ Btu/hr
Assuming U = 250 Btu/(hr)(ft^2)(°F), and $\Delta t = 15$°F

$$\text{Then } A = \frac{15{,}000{,}000}{250 \times 15} = 4{,}000 \text{ ft}^2$$

Using $^3/_4$ in. tubes, 16 ft long, the volume is the same as the evaporator.
Neglect liquid volume, because it is small in the condenser.
Therefore, total volume = 10.5 × 16 = 168.0 ft^3.

Volume of tubes =	63.0 ft^3
Volume of vapor = difference:	105.0 ft^3

Step 4c. Compressor suction and discharge volume:

Average latent heat of mixture = 170 Btu/lb
and approximate average specific vol. at suction = 3.50 ft^3/lb

$$\text{Therefore, cfm} = \frac{12{,}000{,}000 \times 3.50}{60 \times 170} = 4{,}100$$

This indicates a 350 size compressor with 12 in. diam. suction piping and 8 in. discharge.
The volume of the compressor is assumed to be included in the suction and discharge pipes.
Suction Pipe:
Assumed length = 50 ft
Volume = 50 × 0.78 = 39.0 ft^3
Discharge Pipe:
Assumed length = 75 ft
Volume = 75 × 0.35 = 26.25 ft^3

Step 5. For this step, it is necessary to calculate the specific volume of the vapor and liquid in the system.
Step 5a. Find the specific volume of the vapor in the evaporator and the suction pipe.

Component	(1) Mol%	(2) Mol Wt	(1) × (2)	(3) P_c	(1) × (3)	(4) T_c	(1) × (4)
Propane	93.8	44	41.3	617	580	666	625
Butane	6.2	58	3.6	551	34	766	48
			44.9		614		673

Apparent mol wt = 44.9
P_c (mix) = 614, T_c (mix) = 673

$$P_R = \frac{32}{614} = 0.052\,,\ T_R = \frac{460}{673} = 0.682$$

Therefore Z = 0.920, Chart 2, Chapter 5.

$$v = \frac{ZRT}{P} = \frac{Z \times \dfrac{1{,}544}{\text{mol wt}} \times T}{P} = \frac{0.920 \times \dfrac{1{,}544}{44.9} \times 460}{144 \times 32}$$

$= 3.16\ ft^3/lb$

Step 5b. Find the specific volume of vapor in the condenser and the discharge line.

The average composition in the condenser (vapor)
= 95.7% propane, 4.3% butane,
average temp. = 100°F

Component	(1) Mol%	(2) Mol Wt	(1) × (2)	(3) P_c	(1) × (3)	(4) T_c	(1) × (4)
Propane	95.7	44	42.0	617	590	666	638
Butane	4.3	58	2.5	551	24	766	33
			44.5		614		671

Apparent mol wt = 44.5
P_c (mix) = 614, T_c (mix) = 671

$$P_R = \frac{180}{614} = 0.293\,,\ T_R = \frac{560}{671} = 0.833$$

Z = 0.792,

$$v = \frac{ZRT}{P} = \frac{Z \times \dfrac{1{,}544}{\text{mol wt}} \times T}{P} = \frac{0.792 \times \dfrac{1{,}544}{44.5} \times 560}{144 \times 180}$$

$= 0.595\ ft^3/lb$

Step 5c. Find the specific volume of liquid in the evaporator:

Component	(1) Mol%	(2) Mol Wt	(1) × (2)	(3) Wt %	(4) Sp Vol at 0 F	(3) × (4)
Propane	75	44	33.0	69.5	0.0289	0.0201
Butane	25	58	14.5	30.5	0.0260	0.0079
			47.5	100.0		0.0280 ft^3/lb

Average mol wt = 47.5
Sp vol = 0.0280 ft^3/lb

Step 6. Here the final check is made of the original equilibrium assumption.
Step 6a. First calculate the mol of propane and butane vapor in the low pressure side:

Vapor vol in evap = 71.7 ft^3 (Step 4a).
Vapor vol in suct pipe = 39.0 ft^3 (Step 4c).
Total = <u>110.7 ft^3</u>

$$\text{No. mol vapor} = \frac{\text{volume}}{\text{sp vol} \times \text{average mol wt}}$$

$$= \frac{110.7}{3.6 \times 44.9} = 0.780$$

Mol of propane = 0.938 × 0.780 = 0.732
Mol of butane = 0.062 × 0.780 = 0.048

Step 6b. Next calculate the mols of propane and butane vapor in the high pressure side:

Vapor vol in cond = 105.00 ft^3 (Step 4b)

Vapor vol in disch pipe = 26.25 ft^3 (Step 4c)

Total = 131.25 ft^3

$$\text{No. of mols} = \frac{131.25}{0.595 \times 44.5} = 4.96$$

Mols of propane = 0.957 × 4.96 = 4.747
Mols of butane = 0.043 × 4.96 = 0.213

Step 6c. Finally, calculate the mols of propane and butane in the evaporator liquid.

Liquid volume in evaporator = 33.3 ft^3

$$\text{No. of mixture mols} = \frac{33.3}{0.0280 \times 47.5} = 25.00$$

Mols of propane = 0.75 × 25.00 = 18.75
Mols of butane = 0.25 × 25.00 = 6.25

	Mol of Propane	**Mol of Butane**	**Mol of Mixture**
Evap & Suct Pipe (Vapor)	0.732	0.048	0.780
Cond & Disch Pipe (Vapor)	4.747	0.213	4.960
Evap (Liquid)	18.750	6.250	25.000
Total	24.23	6.51	30.74

Thus the calculated mol% of the initial charge is as follows:

$$\text{Propane: } \frac{24.23}{30.74} = 79.0\%$$

$$\text{Butane: } \frac{6.51}{30.74} = 21.0\%$$

This checks the given composition of initial charge. If it did not, the problem must be reworked.

Finally the weight of the total charge is found:

Wt of propane = 24.229 × 44 = 1,067 lb

Wt of butane = 6.511 × 58 = 378 lb

Wt of total charge = 1,445 lb

The preceding example becomes increasingly complicated when more than two components are involved.

Another item to keep in mind is that if there is any leakage in a system such as this, the leakage will be preferential. For example, a vapor leak in the condenser would leak proportionally more propane than butane. This would change the performance of the cycle considerably.

Example 11-6. Other Factors in Refrigerant Selection

Costs

Refrigerant costs are important when considering the investment in filling and maintaining a full charge in a particular system.

At the time of this chapter's development for this edition, the phase-out of certain refrigerants (discussed earlier in this chapter) has required careful redesign of some existing equipment and/or replacement in order to adapt a suitable fluorocarbon type refrigerant. This has required some re-engineering including instrumentation in order to establish a reliable and workable new or upgraded system. Careful attention should be given to the system performance and even redesign when replacement or upgrading is being considered. See Reference 32.

Refrigerant 22* is seldom used in centrifugal compressors due to the high cost, and 113 and 114 are usually used only in water-chilling applications. Refrigerant 11** is frequently used in the higher temperature ranges. Refrigerant 12*** is popular for centrifugal application due to its low cost and favorable suction conditions. It is limited to relatively large tonnages due to the low cfm per ton. For reciprocating applications, refrigerant 22* is preferred to 12*** due to the low cfm per ton. The cost is not a great factor because in reciprocating applications, the charge of refrigerant is relatively small. Refrigerant 12*** has the advantage of a considerably lower heat of compression, with resulting easier duty on the compressor. Ammonia, propylene, and propane require more stages of compression in a centrifugal machine than the chloro-fluoro-refrigerants, and this increases the compressor costs. The cfm/ton and weight flow rates are low and, thus, give lower piping costs.*

The performance characteristics must be reexamined for the replacement refrigerants, and it cannot be assumed that they will perform as direct replacements. In fact, some hard-

*Soon to be replaced by R-507 (125/143a), R-404A (125/143a/134a), R-407A (32/125/134a), R-407B (32/125/134a), R-402A (22/125/290), R-402B (22/125/290), R-403A (22/218/290), R-408A (125/143a/22), or R-134a, R-410A (32/125), R-410B (32/125) or R-407C (32/125/134a); see Figure 14-24.
**Soon to be replaced by R-123.
***Soon to be replaced by R-134A, R-401B, R-405A, R-406A, R-409A.

ware may require modification or replacement to accommodate the "new" refrigerants.

Flammability and Toxicity (Tables 11-3, 11-4, and 11-5)

Most of the chloro-fluoro-refrigerants are nonflammable and nontoxic. Ammonia does not require explosion-proof equipment, but it will burn and is toxic and somewhat difficult to handle. The hydrocarbons propylene, ethylene, and propane are explosive and somewhat toxic and must receive proper attention to safety, as in the design of a light hydrocarbon plant.

Refer to Tables 11-4 and 11-5 and the ANSI/ASHRAE Standards 15-1994 and ANSI/ASHRAE 34-1992, latest editions. Also refer to the discussion under "Process Performants—Refrigerants," earlier in this chapter.

Action with Oil and Water

When water comes in contact with the chloro-fluoro-refrigerants, an acid condition is established. This moisture may be in the form of water vapor coming in with air and is more likely if the suction side is lower than atmospheric pressure. These systems must be checked for leaks and moisture content. The descending order of reactivity with water is refrigerants 11,** 12,*** 114, 22,* and 113. Water vapor does not affect ammonia, except to modify the pressure-temperature relationship. When this becomes noticeable, the charge must be dried. Water must be purged from hydrocarbon systems, because emulsions or two-phase conditions may develop.

Oil is miscible with all the refrigerants except ammonia. This may create foaming in the crankcase and an unsatisfactory compression condition for reciprocating compressors.

Each of the refrigerant manufacturers has determined the proper lubricant to use in a system. They should be consulted for recommendations. (It is beyond the scope of this chapter to provide all of the detail necessary to utilize each refrigerant.)

Oil is not a real problem in centrifugal machines, except that its carry-through affects condensation in the condenser. In an ammonia system, the oil will settle out and can be purged from low points of the system, as receiver, evaporator, etc.

Generalized Comments Regarding Refrigerants

Each system and its particular requirements must be evaluated from a composite of the conditions affecting the refrigerant. After a refrigerant is selected, the accepted design procedure and materials of construction can be applied.

Where ammonia can be accepted as the refrigerant, it is recommended due to the lower initial equipment and charge costs. Reciprocating compressors are preferred for small tonnages. As a general rule, ammonia is not used in systems handling air conditioning applications.

Refrigerant 12*** is a versatile material for a wide range of applications and will often result in lower first costs due to fewer stages of compression. Refrigerants 114 and 11*** are considered for higher temperature levels and lower tonnage loads than refrigerant 12*** (*** = to be phased out).

Propane, propylene, and ethylene are used in large refrigeration tonnage and very low temperature applications.

Materials of Construction

The chloro-fluoro-refrigerants and hydrocarbons use any reasonable material satisfactory for the pressure-copper (or alloys), galvanized steel, steel, aluminum, tin, etc. Ammonia requires an all steel and/or cast iron system with no copper or its alloys in any part. On ammonia centrifugal compressors, the interstage labyrinths are aluminum, and the associate rotating part is free machining stainless steel. The wheels are steel forging with a lead coating. The shaft seal is mechanical carbon ring.

Standard Ton Conditions. These are taken by industry to represent the refrigeration tonnage of a system when operating with an 86°F condenser temperature and a 5°F evaporator temperature. This is a comparative reference condition and does not need interpolation for effective evaluation of other tonnage requirements and conditions.

Refrigerating Effect. This is the heat absorbed in the evaporator per lb of refrigerant. It is determined by the difference in enthalpy of a lb of refrigerant vapor leaving the evaporator and that of a lb of liquid just upstream (ahead) of the expansion valve at the evaporator. From Figure 11-48A,

$$RE = h_1 - h_3 \tag{11-3}$$

and from Figure 11-48,

$$RE = h_1 - h_6 \tag{11-3A}$$

Coefficient of Performance. COP is the ratio of refrigerating effect to work of compression. The higher the value of COP, the higher the efficiency of the cycle. Referring to Figure 11-48A,

$$COP = \frac{T_4}{T_3 - T_4} = \frac{h_1 - h_3}{h_2 - h_1} \tag{11-4}$$

Work of Compression. This is the enthalpy of a lb of refrigerant at compressor discharge conditions minus the enthalpy of a lb of refrigerant at compressor suction conditions,

$$= h_2 - h_1, \text{Btu/lb, (Figure 11-48A)} \tag{11-5}$$

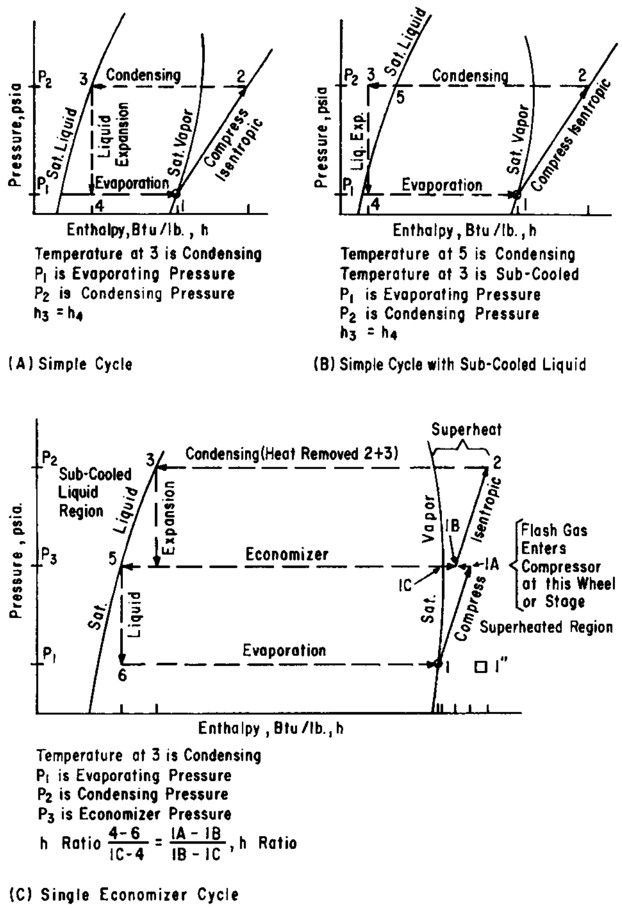

Figure 11-48. Isentropic compression refrigeration cycles.

Horsepower.[1] From Figure 11-48A or 11-48B.

$$\text{Theoretical HP/TR} = \frac{200}{42.42}\left(\frac{h_2 - h_1}{h_1 - h_3}\right) \tag{11-6}$$

$$\text{Theoretical HP/TR} = \frac{0.873\, k\, P_1\, v_1\left[\left(\frac{P_2}{P_1}\right)^{(k-1)/k} - 1\right]}{(k-1)(h_1 - h_3)}, \tag{11-7}$$

for isentropic compression. For polytropic compression, k is replaced by n,

where,

$$\left(\frac{n}{n-1}\right) = (\text{efficiency})\left(\frac{k}{k-1}\right) \tag{11-8}$$

Brake Horsepower. Bhp = theoretical hp/e_o, where e_o is the overall compression efficiency, the ratio of theoretical isentropic to actual bhp.[1] For a large reciprocating compressor system, e_o = 60–70% and 50–65% for small machines. Refer to the section on compressors for detailed data. e_o is not just the mechanical efficiency of the compressor; it is the product of indicated and mechanical compressor efficiencies.[1]

Actual Enthalpy. The actual enthalpy of compressed refrigerant to account for deviation from isentropic compression is referenced to Figure 11-48A, point 2.

Correct enthalpy corresponding to isentropic point 2 is:[1]

$$h_2(\text{corr.}) = h_1 + \left(\frac{h_2 - h_1}{e_o}\right) - \frac{Q}{W} \tag{11-9}$$

For large tonnage requirements the heat loss, Q, is often negligible as far as determining h_2 (corr.) is concerned, but it must be included as far as total tonnage requirements are concerned.

Refrigerant Flow Rate. Refer to Figure 11-48A.

$$\text{lb/min/TR} = \frac{200}{h_1 - h_3} \tag{11-10}$$

For subcooled refrigerant, Figure 11-48B,

$$\text{lb/min/TR} = \frac{200}{h_1 - h_3} \tag{11-11}$$

Note that the temperature at this h_3 is lower than when no subcooling exists.

$$\text{cfm per TR} = \left(\frac{200}{h_1 - h_3}\right)(v_1)\text{, at compressor intake} \tag{11-12}$$

Heat Removed by Condenser.

$$= 200\left(\frac{h_2 - h_3}{h_1 - h_3}\right)\text{, Btu/ton min} \tag{11-13}$$

For Economizer System. See Figure 11-48C. Refrigerant flow in the second (or later) stage is the sum of first-stage weight flow plus the weight flow of the vapor flashed from the economizer.[19]

Lb flashed gas from economizer/lb vapor from evaporator

$$= \frac{h_3 - h_5}{h_{1c} - h_3}$$

When economizer vapors are mixed with the compressor vapors, the temperature from this mixing point on is lower to the next stage of compression than if the economizer had not been used.[19]

h (of mixed vapors corresponding to point 1B)

$$= \frac{w_e h_{1c} + w_1 h_1}{w_e + w_1}, \text{ Btu/lb} \tag{11-14}$$

where w_e = vapor from economizer, lb/min
w_1 = vapor from first-stage of compression, lb/min
h_1 = enthalpy of first-stage compression vapor, Btu/lb
h_e = enthalpy of vapor from economizer, Btu/lb

Assuming a constant specific heat of the vapor, the temperature of the mixture is given by

$$t_m = \frac{w_e t_e + w_1 t_{1A}}{w_e + w_1} \tag{11-15}$$

where t_e is the *saturated* temperature of the economizer vapor (from saturation curve) and t_{1A} is the temperature of the first stage compressor discharge.

System Design and Selection

Basically the system design consists of the selection of component parts to combine and operate in the most economical manner for the specified conditions. Unfortunately, the specific conditions are not only for the evaporator where the refrigerant is actually used but include all or part of the following. These conditions are identified whether the system is a separate component selection or a package furnished assembled by a manufacturer.

1. Evaporator: temperature and refrigerant
2. Compressor: centrifugal, screw or reciprocating; electric motor, steam turbine, or other driver
3. Condenser: horizontal or vertical, temperature of cooling water, water quantity limit
4. Receiver: system refrigerant volume for shut-down refrigerant storage
5. Operation: refrigeration tonnage load changes.

Figures 11-49A–D are convenient to summarize specifications to a manufacturer. They are also used as a condensed summary of a designed system.

For final design horsepower and equipment selection, the usual practice is to submit the refrigeration load and utility conditions/requirements to a reputable refrigerant system designer/manufacturer and obtain a warranted system with equipment and instrumentation design and specifications including the important materials of construction. Always request detailed operating instructions/controls and utility quantity requirements.

Design

A system is designed as follows:

1. Establish total refrigeration tonnage for each evaporator temperature level. When possible, combine these into as few different levels as possible. Do not specify a lower temperature than needed to accomplish the process refrigeration requirements. Allow a minimum of 5°F differential between the lowest required process temperature and the evaporating refrigerant. The larger this Δt, the smaller can be the surface area in the evaporator. The lower the evaporating temperature for any given refrigerant, the higher the required horsepower for the compressor. The compromise suggested must be resolved by comparative cost studies and judgment.
2. Establish a heat balance for the refrigerant throughout the entire system, using thermodynamic property tables or diagrams for the particular refrigerant.[1,2,20]
3. Allowing for pressure drop through piping, equipment, and control valves, establish the expected operating temperatures and pressures.
4. Prepare inquiry specifications for compressors and heat exchange equipment following the forms suggested.

Figure 11-50 illustrates the type of comparisons of performance that may be made to better interpret a given set of design parameters.

Example 11-7. 300-Ton Ammonia Refrigeration System

A process system requires the condensation of a vapor stream at 15°F. The refrigeration load in three parallel evaporators will be equally distributed and totals 3,600,000 Btu/hr, including a 10% factor of safety and 5% system heat loss. Design a mechanical (not absorption) system using ammonia as the refrigerant. Ammonia was selected because (1) the temperature level is good and (2) ammonia is compatible with the process-side fluid in case of a leak. The condenser cooling water is at 90°F for three months during the summer and must be used to ensure continuous operation.

Refer to Figure 11-51A for a diagram of the system. The selected conditions are also presented as a summary of expected operations, Figure 11-51B.

To allow operations at one-half load and flexibility in case of mechanical trouble, use two reciprocating compressors capable of handling 150 tons of refrigeration each.

Pressures Selected

1. Compressor discharge: 214.2 psig (228.9 psia) at a condensing temperature for ammonia of 105°F (see Reference 1). This allows a 105°F − 90°F = 15°F Δt at the cold end of the condenser. This is reasonable.

(Text continues on page 358)

SPEC. DWG. NO.
A-

Job No.

B/M No.

Page	of Pages
Unit Price	
No. Units	
Item No.	

MECHANICAL REFRIGERATION SPECIFICATIONS

SERVICE CONDITIONS

Capacity Tons. Refrigerant . Process Fluid .
Manufacturer . Model . RPM
Compressor Suction PSIG. @ °F. Discharge PSIG. @ °F
Condenser Refeigerant Lb/Hr. °F Exit
Cooling Water Supply (Sea Water) (River Water) °F @ PSIG.
Cooling Water Requirement Gpm @ °F Exit
Evaporator Refrigerant °F @ PSIG, Lb/Hr
Evaporator Process Fluid Lb/Hr °F Inlet, °F Outlet PSIG.
Compressor Driver By (Vendor) (Owner) HP RPM Type

SPECIFICATIONS

Temperature Control (Manual) (Automatic). Instruments & Controls (Weather Protected) (Explosion Proof)
Level Controls - To Be Connections To Be
Pressure Gauges - To Be Safety B.O. Disc., 4½" Dial, ½" Conn.
Condenser Cooling Water: Connection Inlet Outlet Flange Rating Face
Refrigerant to Evaporator: Connection Inlet Outlet Flange Rating Face
Suction Cross Exchanger: Connection Inlet Outlet Flange Rating Face
Compressor Suction: Connection Inlet Flange Rating Face
Compressor Discharge: Connection Outlet Flange Rating Face
Steam Supply: PSIG @ °F
Driver Power Supply: Volt Phase Cycle
Instrument Power Supply: Volt Phase Cycle
Fouling Factors: Cooling Water Process Fluid
Compressor, Driver and Speed Gears: See Specification Sheet
Condenser Tube Water Velocity Limits. Ft/Sec. to Ft/Sec.
Refrigerant Charge Lbs.

MATERIALS OF CONSTRUCTION

Condenser: - Tubes O.D. , BWG , Length Ft. Pitch , No.
Tube Sheet Channel
Evaporator: By (Owner) (Vendor)
Tubes O.D. , BWG , Length Ft. Pitch , No.
Tube Sheet Channel

REMARKS

Vendor is to Specify:
1. Make and Model for Flow, Temperature and Pressure Control Valves, Hand Valves, Thermometers and Steam Traps, Electrical Components and Miscellaneous Equipment.

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Figure 11-49A. Mechanical refrigeration specifications.

SPEC. DWG. NO.
A-

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Unit Price

No. Units

Item No.

Job No. ______________________

____

B/M No. ______________________

CENTRIFUGAL COMPRESSOR SPECIFICATIONS

Service ______ Manufacturer ______
Type ______ Model ______ Size ______
No. Impellers ______ BHP ______ @ ______ RPM ______ Speed Range ______ RPM

OPERATING CONDITIONS PER ______

Gas (Dry) ______ Sat'd. with ______ Avg. Mol. Wt. ______ "K" Value ______
Suction: Pres. PSIA. Normal ______ Guarantee ______ Max. ______
Temp. °F. Normal ______ Guarantee ______ Max. ______
Discharge: Press. PSIA Normal ______ Guarantee ______ Max. ______
Temp. °F. Normal ______ Guarantee ______ Max. ______
Sidestream: Press. PSIA Normal ______ Guarantee ______ Max. ______
Temp. °F. Normal ______ Guarantee ______ Max. ______
C.F.M. @ Suction Conditions ______ Wt. Flow ______ Lbs./Hr.
C.F.M. @ Sidestream Conditions ______ Wt. Flow ______ Lbs./Hr.
Compressibility Factors Suction ______ Discharge ______
Surge ______ % Capacity Surge Pt. ______ First Critical RPM ______
ΔP Intercoolers ______ PSI

COMPRESSOR DETAILS

Type Impeller ______ Overspeed ______ RPM Rated ______ RPM
Intake Flange Size ______ ASA ______ Lbs. Facing ______ Disch. Flg. Size ______ ASA ______ Facing ______
Sidestream Flange Size ______ ASA Rating ______ Lbs. Facing ______ Casing Test Pressure ______ PSIG
Bearings: Journal - Babbited Sleeve ______ Thrust ______ Make ______
Coupling: Make ______ Class ______ Type ______ Intercooler By ______
Water Temp. ______ °F. Type ______ Tubes Mat'l. ______ Size ______

COMPRESSOR DRIVER

Horsepower ______ Type ______ Class ______ Volts ______ Phase ______ Cycles ______
Steam in ______ PSIA ______ Temp. ______ °F. Stm. Nozzle Control ______ °F. No. Hand Valves ______
Exhaust ______ PSIA ______ Temp. ______ °F. Condensing ______
Driver Spec. Sheet No. ______ Gear: Make ______ Rated BHP ______ Model ______ Red'n Ratio ______

COMPRESSOR MATERIAL

Case: ______ Shaft: ______ Sleeves: ______
Impellers: Hub & Cover ______ Blades ______ Diaphragms ______ Labyrinths ______

LUBRICATION SYSTEM

Main Oil Pump Driver ______ With Piping
Aux. Oil Pump Driver ______ Class ______ Volts ______ Phase ______ Cycles ______
Twin Oil Coolers ______ Twin Oil Filters ______ Bearing Temp. Ind. ______ Tube Mat'l. ______
Cooling Water: ______ @ ______ °F. & ______ PSIG. GPM Req'd. ______

SEALING SYSTEM

Type of Seals ______

REMARKS

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Figure 11-49B. Centrifugal compressor specifications.

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A-

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Item No.	

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____

B/M No. ____________________

RECIPROCATING COMPRESSOR SPECIFICATIONS

Service ______ Manufacturer ______ Model ______

Type ______ BHP ______ Fluid ______ Sat'd. with: ______

Avg. Mol. Wt. (Dry) ______ Design Speed ______ RPM. Speed Range ______ RPM

DATA per ______ NORMAL CONDITIONS

Stage					
Class or Type					
Cylinders: Diameter (Bore), Inches					
Stroke, Inches					
Piston Displacement: Cubic Ft. per					
Action of Cyl. Single or Double					
Actual Intake. CFM					
Delivery in Lbs./Hr. (Dry Basis)					
Intake Press. PSIA					
Intake Temp. °F					
Disch. Press. PSIA					
Disch. Temp. °F					
Ratio of Compression					
Compressibility Factor @____ PSIG & ____ °F.					
Ratio of Specific Heats, Cp/Cv					
Specific Volume @ Intake Conditions					
Volumetric Efficiency @ Suct., %					
Normal Clearance, %					
Cyl. Test Press. PSIG					
Press. Drop Allowed Between Stages					
Suction Nozzle Size					
Suction Nozzle Rating & Facing					
Disch. Nozzle Size					
Disch. Nozzle Rating & Facing					
Material: Cylinder					
Liner					
Heads					

Unloading Facilities (Pockets, Valve Lifters) ______

Weights and Unbalanced Forces ______

Driver: Type ______ Mfgr. ______ H.P. ______ RPM ______

Volts ______ Phase ______ Cycle ______ Frame ______

Steam: Inlet ______ PSIA @ ______ °F. Exhaust ______ PSIA @ ______ °F. Steam Rate ______ Lbs./Hr.

Fuel: ______ Heat Value ______ BTU. Full Load Consumption ______ BTU/BHP. Hr.

Heat Rejection: Cooling Water Available @ ______ °F and ______ PSIG. Type ______

Comp. Cyl. Jacket: ______ BTU/Hr. Temp. In ______ °F. Temp. Out ______ °F. Water Req'd. ____ GPM. Δp ____ PSI

Lube Oil Cooler: ______ BTU/Hr. Temp. In ______ °F. Temp. Out ______ °F. Water Req'd. ____ GPM. Δp ____ PSI

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Figure 11-49C. Reciprocating compressor specifications.

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MECHANICAL DRIVE TURBINE SPECIFICATIONS

TURBINE LUBRICATING SYSTEM: FORCED FEED

Independent of (Pumps) (Compressor) ______ Combined ______

Oil Cooler: Tubes ______ Water ______ @ ______ °F.

External Oil Cooler(s) ______ O.D. Tubes. Single ______ Dual ______

External Oil Filter(s): Single ______ Dual ______

Oil Temp. Indicator(s): Dial Type ______ Stem Type ______ (After) (Before) Cooler ______

Bearing Thermometer(s): Dial Type ______ Stem Type ______ For Turbine ______

Oil Pressure Gauge(s): Total ______. Mounted: Piping ______ Panel ______

Sight Flow Indicator in Oil Line ______ Leaving each Bearing ______

Main Oil Pump: Integral, Driven from Turbine Shaft; Separate Turbine Driven; Separate Motor Driven; Motor Class ______

Steam Turbine Driven Auxiliary Oil Pump for ______ PSIG ______ °F. T T Steam Exhaust ______ PSIG, Including Control.

Electric Motor Driven Auxiliary Oil Pump ______ Starter ______

Low Oil Pressure Failure Switch(es): Electric ______ Pneumatic ______

Hydraulic ______ to Actuate Auxiliary Oil Pump Driver ______ Solenoid Trip ______

Pneumatic Dump ______, Classification ______

Low Oil Pressure Alarm Switch ______

Prefabricated Interconnecting Oil Piping ______

Oil Reservoir Located: Separate ______ In Base ______

All Electrical Equipment: Open Construction ______ Explosion Proof. ______

Motors & Starters: ______ Volts ______ Phase ______ Cycle

Protective Devices to be for ______ Volts ______ Phase ______ Cycle

Air Supply Required @ ______ PSIG

HEAT LOADS

Turbine: Oil ______ BTU/Hr. ______ GPM ______ PSIG

Cooling Water ______ °F. ______ GPM ______ PSIG

Compressor: Oil ______ BTU/Hr. ______ GPM ______ PSIG

Cooling Water ______ °F. ______ GPM ______ PSIG

REMARKS

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Figure 11-49D. Mechanical drive steam turbine specifications.

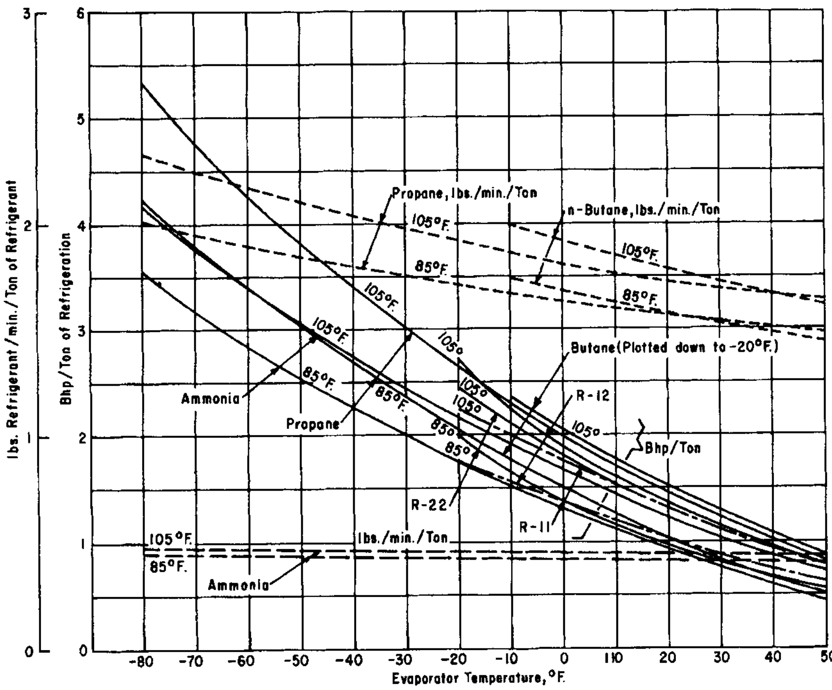

Figure 11-50. Comparison of refrigerants at condensing temperatures of 85°F and 105°F.

Note: Bhp values based on 75% overall efficiency. This is not exact, as efficiency changes with specific system conditions. (Plotted from data of Boteler, H. W. *Natural Gasoline Supply Men's Engr. Data Book*, 7th Ed., p. 40, ©1957; and Huff, R. L. *Petroleum Refiner*, p. 111, Feb. 1959.)

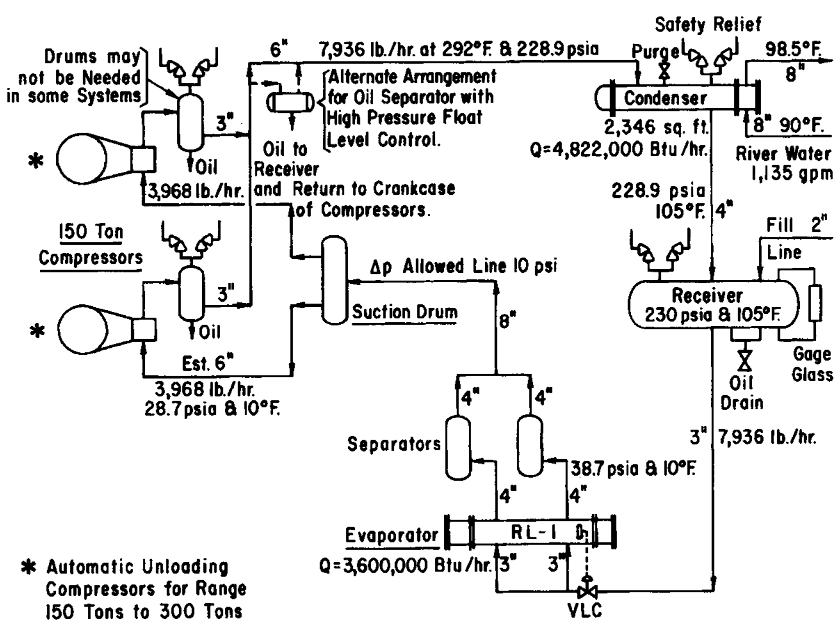

Figure 11-51. Example problem: 300-ton ammonia refrigeration system, Example 11-7.

(Text continued from page 353)

2. Evaporator, refrigerant side: 23.8 psig (38.5 psia). This corresponds to a boiling or evaporating refrigerant temperature of the required 10°F. See Reference 1 or other ammonia tables.
3. Compressor suction: 13.8 psig (28.5 psia). This allows an assumed (must be checked) pressure of 10 psi in the piping from the evaporator to the compressor suction flange. This should be a conservative drop and should be made smaller, if possible, to conserve compressor horsepower.

Ammonia Flow

Heat duty = 3,600,000 Btu/hr.
Tonnage = 3,600,000/12,000 Btu/ton ref. = 300 tons ref.
Latent heat ammonia at 10°F = 561.1 Btu/lb.
Liquid ammonia vaporized = 3,600,000/561.1 = 6,420 lb/hr.

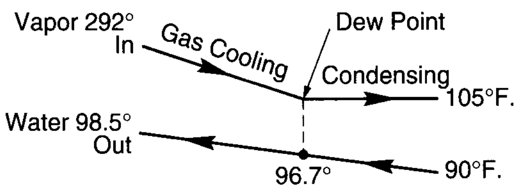

Figure 11-51B Selected conditions for Example 11-7.

The *total flow* required *to* the *evaporator* will be larger due to the flashing of ammonia across the control valve ahead of the evaporator.

Above a −40°F datum:
Enthalpy of liquid at 228.9 psia = 161.1 Btu/lb.
Enthalpy of liquid at 38.5 psia = 53.8 Btu/lb.
Enthalpy of vapor at 58.5 psia = 614.9 Btu/lb.

Let x = fraction (weight) ammonia vapor formed by flashing the 228.9 psia liquid at the control valve down to 38.5 psia. By heat balance per pound of ammonia:

Heat ahead of valve = heat down stream of valve

$$\underset{\text{(liquid)}}{161.1} = \underset{\text{(vapor)}}{(x)(614.9)} + \underset{\text{(liquid)}}{(1 - x)(53.8)}$$

x = 0.191, or 19.1 wt% vapor downstream of control valve.

Note that due to flashing and formation of a vapor-liquid mixture, the control valve is always placed as close to the inlet of the evaporator as possible.

The incoming total liquid ahead of the control valve must be increased to compensate for the loss taken across the control valve; then the required incoming liquid must be

6,420/(1.00 − 0.191) = 7,936 lb/hr

Compressor Suction Flow

= 7,936 lb/hr at 28.5 psia and 10°F
Ratio of specific heats = 1.292 at 150°F
Expected discharge temperature:

$$t = [(t_s + 460)(R_c)^{(k-1)/k}] - 460$$
$$= [(10 + 460)(228.9/28.5)^{(1.292-1)/1.292}] - 460$$
$$t = (470)(1.60) - 460 = 292°F$$

Condenser

Cool 7,936 lb/hr ammonia from 292°F to 105°F and condense at this point. Pressure is 228.9 psia. Reading ammonia superheated vapor tables (or chart):

Enthalpy of vapor at 292°F and 228.9 psia = 758 Btu/lb.
Latent heat of saturated vapor at 105°F = 472.3 Btu/lb.
Enthalpy of saturated vapor at 105°F = 633.4 Btu/lb.
Sensible heat loss of vapor = 758 − 633.4 = 124.6 Btu/lb.
Sensible heat duty of condenser = (124.6)(7,936)
= 988,000 Btu/hr.
Latent heat duty of condenser = (472.3)(7,936)
= 3,740,000 Btu/hr
Total heat duty = 988,000 + 3,740,000 = 4,728,000 Btu/hr

The condenser is designed by usual methods as described in Chapter 10, "Heat Transfer." For this service, a summary design is

Assumed water temperature rise = 8.5°F
Gpm required = 1,135
Designing stepwise:
Gas cooling LMTD = 55.0°F
Condensing LMTD = 11.3°F
Assume a unit:
Duplex tubes: 1-in. O.D. × 16 ft, 0 in. long, 16 BWG steel outside, 16 BWG cupro-nickel inside
No. = 578, 1 $^1/_4$-in. triangular pitch
Shell: 36-in. O.D., 4 tube pass
Film coefficients:
Tube side water film at 6 ft/sec = 1,025 Btu/hr (°F) (ft²)
Shell side gas cooling, U_o = 23.4 Btu/hr (°F) (ft²), with 0.002 fouling and includes tube side film
Shell side condensing, U_o = 246 Btu/hr (°F.) (ft²) with 0.002 fouling and includes tube side film
Areas required:

Gas cooling = 780 ft²
Condensing = 1,370 ft²
Total = 2,150 ft²

Area available in assumed unit = 2,346 ft²
Factor of safety = 1.09 = 9%, this is satisfactory.
Shell side, ΔP calculates 0.140 psi, use 1.0 psi.
Tube side ΔP calculates 10.0 psi, use 12.0 psi.

Baffles on shell side:
6–25% horizontal cut on 12-in centers, for gas cooling area at gas inlet end of exchanger.
2–50% horizontal cut baffles for tube supports, spaced on 3 ft, 0 in. centers in condensing section at liquid outlet end of exchanger.

Provide horizontal cut 1.75 in. deep on *all* lower baffles to allow for condensate drainage. Remove the 9 tubes in this cut area to allow free drainage, see Figures 11-52A and B.

Receiver

Several different sizes can be used. For example,

1. Based on 30-min inventory of flowing ammonia. This is a 5-ft I.D. × 10-ft long steel tank
2. Based on lower holding time, down to a vessel about 12 in. I.D. × 10 ft long.

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	Unit Price
B/M No. **EXCHANGER RATING & SPECIFICATIONS**	No. Units One
	Item No.

Service Ammonia Condenser for Refrigeration System

Min. Req. Eff. Outside Area Exposed In Shell, Sq. Ft. 2346

Number of Units: Operating One Spares No

DESIGN DATA PER Unit

UNIT DATA		SHELL SIDE	TUBE SIDE
Fluid		NH_3	Water
Fluid Flow	Lbs./Hr.	8086	569,000 (1135 gpm)
Temperature In	°F.	232	90
Temperature Out	°F.	105	98.5
Operating Pressure	PSIG	215	60
Density	Lbs./CF	Vap.-0.6 Liq.-35.96	
Specific Heat	Btu/Lb./°F.	Vap.-0.54 Liq.-1.17	
Latent Heat	Btu/Lb.		
Therm. Cond.	Btu/Hr./Sq. Ft./°F./Ft.	Vap.-0.0185 Liq.-0.29	
Viscosity	Centipoise	Vap.-0.0128 Liq.-0.09	
Molecular Weight		17	18
No. of Passes		1	4
Pressure Drop	PSI	Calc: 0.14 Used: 1.0	Calc: 10.0 Used: 12
Fouling Factor		.001	.001

Heat Transferred - BTU/Hr. Cooling - 1,003,000 Condensing - 3,819,000

LMTD 55 & 11.3 °F. Overall U: Calc. 23.4 & 246 Used 201 & 234.4

CONSTRUCTION

	Shell	Tube
Max. Oper. Pressure	300 PSI	150 PSI
Max. Oper. Temperature	450 °F.	450 °F.

Type of Unit Fixed Tube Sheet Approx. Shell Diameter

Tubes: No. (Approx.) 578 O.D. 1" BWG. * Length 16'-0" Pitch 1¼" Δ Joint R & B

Material: Tubes Duplex * Tube Sheet Steel - Monel Clad

Channel Steel Baffle Steel Corr. Allow. 1/16

Shell Steel Baffle Steel Corr. Allow. 1/16

Connections - Shell In: 6" Out 4" Flange 300# RF

Channel In: 8" Out 8" Flange 150# RF

Others: Vent & Drain Size 1" Flange 6000# cplg.

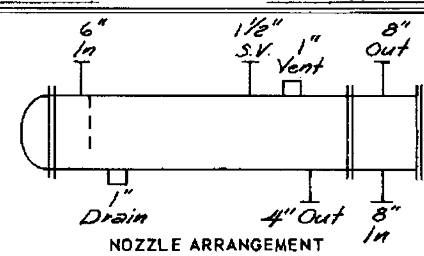
Bolts: Gaskets: JM60

Code ASME Stamp Yes TEMA Class X-Ray No S.R. No

Insulation No Class Cathodic Protection No

See Figure 11-24B

BAFFLE ARRANGEMENT



NOZZLE ARRANGEMENT

Remarks:

* Duplex tubes - 18BWG 70-30 CuNi inside & 16 BWG steel outside

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Figure 11-52A. Ammonia condenser rating and specifications.

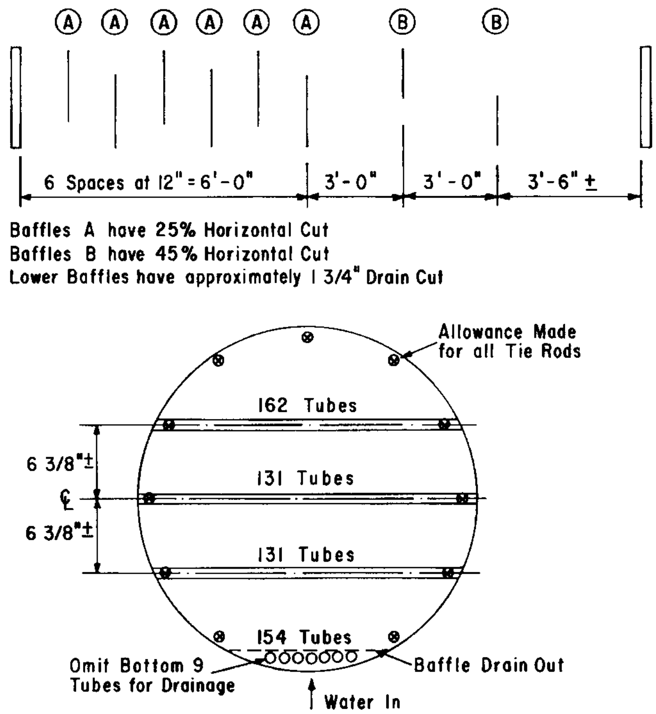

Figure 11-52B. Ammonia condenser for refrigeration system; tubesheet and baffle arrangements.

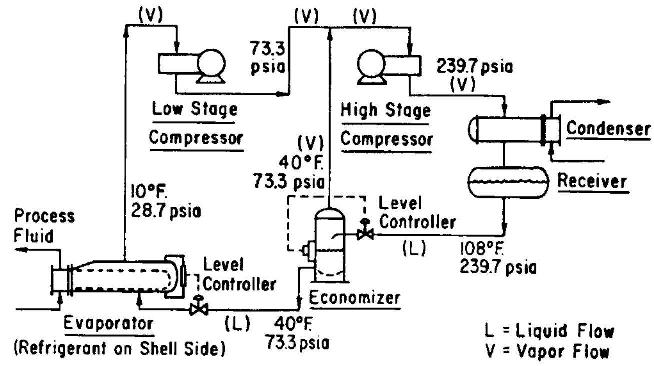

Figure 11-53. Ammonia refrigeration system with economizer.

If complete system drainage and pump-out is to be made into this vessel, then the volume capacity of all of the piping and equipment must be taken into account.

Economizers

The economizer is used in multistage centrifugal compression refrigeration and in two-stage reciprocating applications. It offers operating and performance economics that are specific to the system conditions. The cycle efficiency is improved, and the required horsepower is reduced. Referring to Figure 11-53, the liquid from the condenser expands through the control valve to the pressure maintained in the vessel by its connection to the compressor intermediate stage or wheel. Through the expansion, the liquid is cooled to the temperature corresponding to the pressure in the vessel. Some vapor leaves due to the boiling and flashing action and enters the compressor at the selected tie-in point. The cooled liquid flows to the evaporator for its last expansion (or to another economizer if the system is so designed). This is shown in Figure 11-48C for an isentropic compression. In actuality this produces about the same result as taking the deviations into account.

The use of Mollier refrigerant diagrams is common. The refrigerant is compressed from Point 1 to 2, while at 1A cool flash gas enters from the economizer to cool the gas to 1B, so that the last part of the compression is slightly more efficient than if this were not done. The expansion from condenser pressure at 3 (P_2) to economizer pressure at 4 (P_3) is at constant enthalpy, as is the expansion from economizer pressure to the evaporator pressure at 6 (P_1).

To obtain the most efficient use of an economizer, studies are needed to balance economizer pressure, condenser pressure and temperature, evaporator surface area, and compression horsepower. These are all interrelated, and horsepower can be saved at the expense of surface areas, and vice-versa.

A system using an economizer as in Figure 11-53 can show a thermodynamic improvement in efficiency over that of Figure 11-51 due primarily to the subcooling of the liquid before entering the evaporator. This makes better use of the cool flash gas from the economizer by cooling the interstage compression gas. The saving in compressor horsepower may be 3–15% or more in some special systems. This economizer system has the following advantages over the simple system of Figure 11-51:

1. Lower initial cost for equipment.
2. Lower operating horsepower, and with the lower compression ratios, the maintenance can be expected to be lower.
3. System is more flexible as to changes in required evaporator pressure.
4. Improved control of liquid to evaporators, less flashing at evaporators.
5. Lower compressor foundation costs.

Example 11-8. 200-Ton Chloro-Fluor-Refrigerant-12

Note: This example is included because it represents typical calculations; however, refrigerant R-12 is in process of being phased-out of availability, and the number values of the example showing R-12 cannot be used/substituted (except in concept) for any other newer replacement refrigerant.

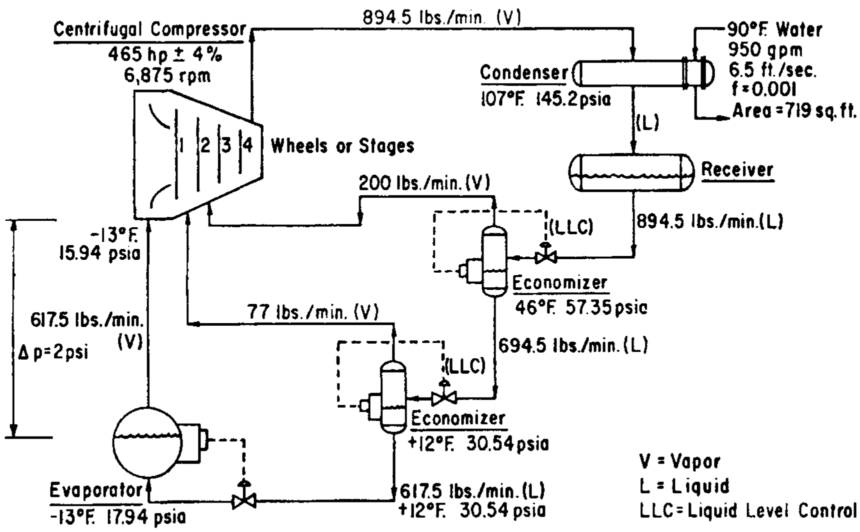

Figure 11-54. 200-ton refrigeration system using R-12 refrigerant. (Used by permission: Carrier Corporation, a United Technologies Company.)

The system shown in Figure 11-54 uses two economizers to achieve a lower liquid temperature entering the evaporator and reduces the total amount of gas being compressed through the entire pressure range from 17.94 psia to 145.2 psia. This requires the entrance of the gas from the economizers at the correct suction pressure to a specific wheel of the centrifugal compressor. The pressures selected for this flashing are not arbitrary, but are coordinated with the expected design of the compressor, usually by trial and error. A wide variety of cycles can be developed, each using certain features of some advantage to the system. The final selection must be one considering horsepower consumption and complexity of operation of the economizers. Usually two economizers are adequate for most industrial requirements, and one is the most common.

Suction Gas Superheat

The vapor entering the suction of the compressor from the evaporator is generally superheated by (1) heat gain in the piping and/or (2) cross-exchange. In general a small amount of superheat is good as it prevents liquid carryover into the compressor. The suction condition for such a vapor can be represented by Point 1 in Figure 11-48C. The compression starts here and is handled in the usual manner. About 5–15°F of superheat is desirable for the average design.

A superheat exchanger is not needed for a centrifugal compressor system as far as gas condition is concerned. It may be a good unit as far as other aspects of the process are involved. The centrifugal compressor requires only a suction knock-out drum to remove entrained liquid and foreign particles.

Superheat is not necessary or desirable for ammonia because the volumetric efficiency is not improved. A few degrees to prevent liquid carry-over are acceptable. The volumetric efficiency can be improved with these typical superheat conditions for R-21:

Sat. suct. temp., °F	− 40	− 30	− 20	− 10	0 and above
Actual suct. temp., °F	+ 35	+ 45	+ 55	+ 65	+ 65

For R-22*, some common maximum superheat conditions are:[3]

Sat. suct. temp., °F	− 40	− 20	− 0	+ 20	+ 40
Actual suct. temp., °F	− 15	+ 5	+ 25	+ 40	+ 55

The compressor discharge temperatures are usually limited to 275–290°F.

Example 11-9. Systems Operating at Different Refrigerant Temperatures

Figures 11-55A and 11-55B illustrate the types of systems that might be developed to temperatures of −25°F, −4°F, and +26°F in evaporating (chilling) equipment. For comparison, the physical process tie-in points using reciprocating and centrifugal compressors are shown. Two stages of reciprocating and four stages of centrifugal are indicated. The centrifugal system has low temperature (and pressure) vapor entering the first stage (wheel); the −4°F vapor entering at the suction to the second wheel; the +26°F vapor entering at the suction of the third wheel; and the flash vapor entering the fourth wheel. These pressures must all be established to balance at the points of tie-in.

*See discussion on replacement refrigerants.

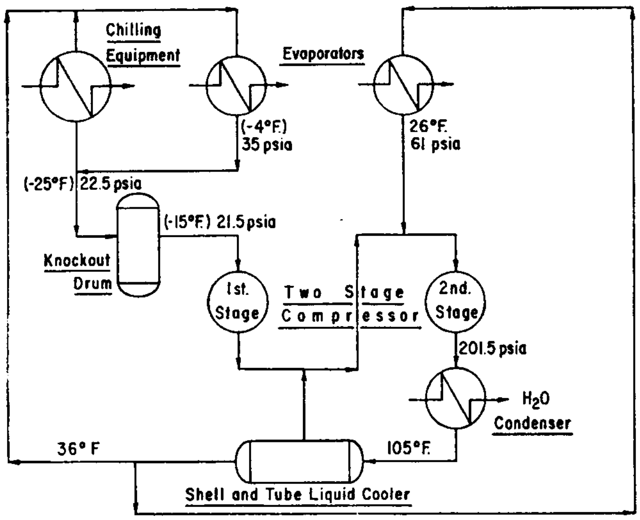

Figure 11-55A. Refrigerating system with propane refrigerant. (Used by permission: Charls, J., Jr. *Tech Paper RP 468,* ©1950. Dresser-Rand Company.)

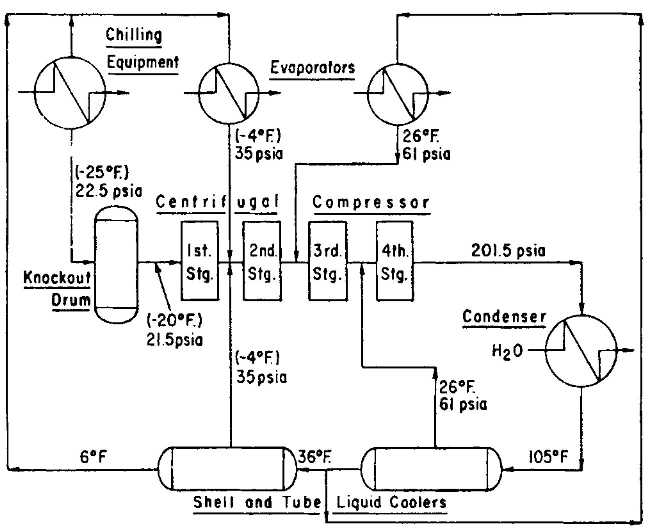

Figure 11-55B. Centrifugal system with propane refrigerant. (Used by permission: Charls, J., Jr. *Tech Paper RP-468,* © 1950. Dresser-Rand Company.)

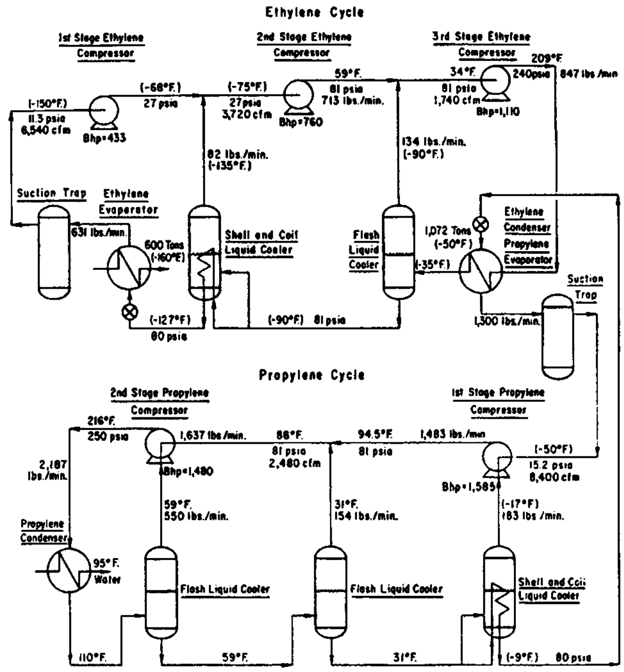

Figure 11-56. Cascade type low-temperature refinery refrigeration cycle. (Used by permission: Charls, J., Jr. *Tech Paper RP-468,* ©1950. Dresser-Rand Company.

Cascade Systems

A cascade system uses one refrigerant to condense the other primary refrigerant that is operating at the desired evaporator temperature. This approach is usually used for temperature levels less than −80°F, when light hydrocarbon gases or other low boiling gases and vapors are being cooled. To obtain the highest overall efficiency for the system, the refrigerants for the two superimposed systems are different.

Figure 11-56 is typical of the type of system that can be arranged to suit a primary refrigeration load of 600 tons at −160°F.[4] Ethylene is the refrigerant being handled in centrifugal compressors (in some designs these three units might be combined in one compressor case), and there is flash intercooling (economizer) and liquid subcooling to the evaporator. The ethylene is condensed by evaporating propylene at −50°F. The propylene comprises a separate system using centrifugal compressors and water condensers.

Compound Compression System

This system uses the same refrigerant throughout the entire compression but includes the advantages of liquid subcooling for close approach to evaporator temperature and direct vapor cooling of interstage compression vapors by a refrigerant operated intercooler. Figure 11-56 includes the shell and coil liquid intercooler in both the ethylene and propylene cycles. It does not include a liquid subcooler-exchanger for suction gas desuperheating. The temperature approach between the liquid passing through the coil and that in the shell of the liquid cooler is usually 10–20°F.[1]

Comparison of Effect of System Cycle and Expansion Valves on Required Horsepower

Table 11-14 compares four basic systems using refrigerant 12, and generally indicates that the total horsepower can be reduced by using a multistage compression arrangement

Table 11-14
Theoretical Compressor Power for a Refrigerant-12 Plant Requiring 10 Tons at 44°F, 30 Tons at 34°F, and 20 Tons at 24°F

Type of System	Hp	% Reduction from Max.
One Compressor:		
All evaporators at same temp. (24°F)	52.7	0
**Individual exp. valves and back-pressure valves	52.7	0
Multiple exp. valves and back-pressure valves	52.7	0
Two compressors *(one dual-effect type):		
Individual expansion valves	47.3	10.2
Multiple expansion valves	45.5	13.7
Three individual compressors:		
Individual expansion valves	45.8	13.1
Multiple expansion valves	44.2	16.1
Compound compressors and intercoolers:		
Individual expansion valves	45.2	14.2
Multiple expansion valves	44.2	16.1

*One compressor has two different suction temperatures (stages) and a second compressor has one. Both compressors discharge to same condenser.

**All other valves at the three needed temperature levels.

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together with multiple expansion valves rather than individual valves directly ahead of each evaporator.[7] The multiple expansion valves successively take refrigerant liquid from the highest to lowest level as the requirement for each evaporator is withdrawn.

Cryogenics

Cryogenics is usually associated with liquefaction of air into its components: nitrogen, oxygen, argon, xenon, etc. This type of refrigeration is beyond the intended scope of this chapter, because it is so specialized that the chemical and petrochemical industry will purchase an air liquefaction plant designed by specialists, for example, to supply pure oxygen and nitrogen for process use and for blanketing or purging.

Low-temperature, high-pressure gases at moderate to low temperatures can be expanded by the use of an expansion turbine (very similar to a steam turbine or even by reciprocating or screw-type compressors) or by the use of a throttling valve. The mechanical expander is theoretically operating at constant entropy with an efficiency of about 80%, and the Joule-Thompson throttling operates at constant enthalpy with no change in heat content.[60] See Figure 11-57, which shows the comparison of the valve-throttling refrigeration process to the expander turbine process. Figure 11-58

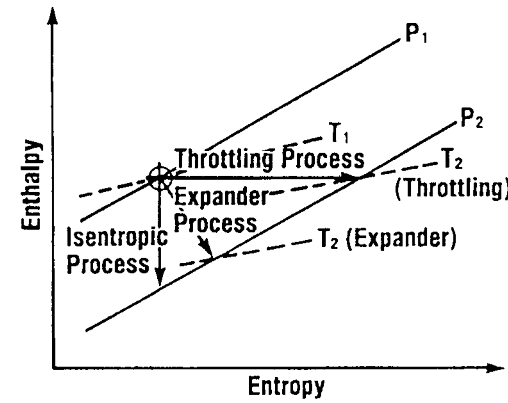

Figure 11-57. Comparison of the energy potential of turboexpanders vs. throttle valves. (Used by permission: Bul. 2781005601. ©Atlas Copco Comptec, Inc.)

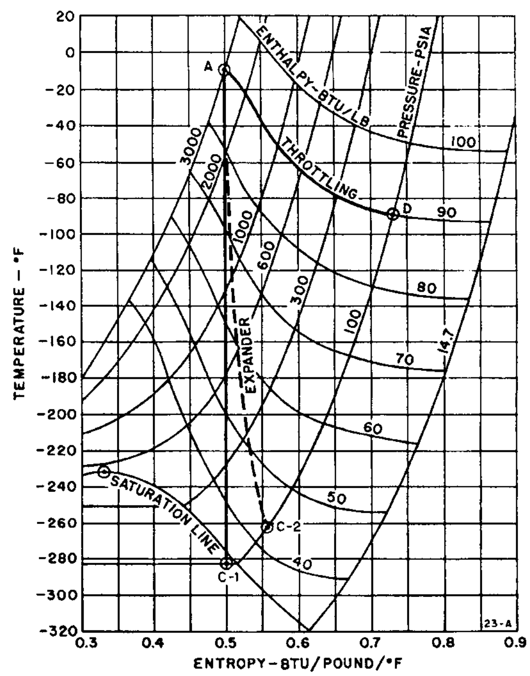

Figure 11-58. Theoretical comparison of Joule-Thompson cooling effect with nitrogen vs. the use of a mechanical expander. (Used by permission: Gibbs, C. W., (Ed.). *Compressed Air and Gas Data*, ©1969. Ingersoll-Rand Co.)

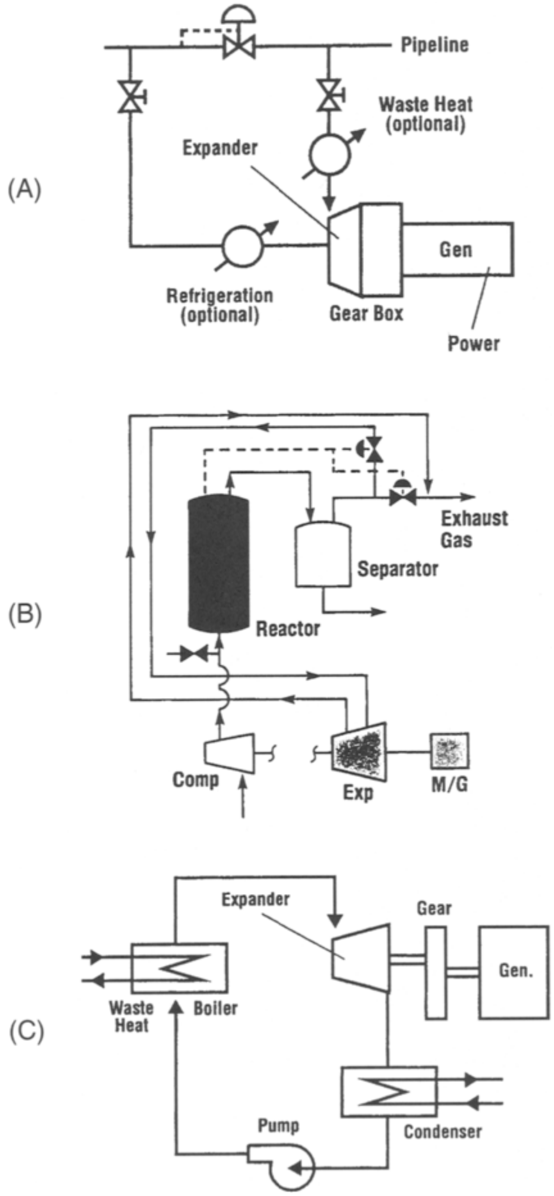

Figure 11-59. Representative expander refrigeration systems for temperature requirements and/or power recovery. (A) Expander coupled to a generator recovers pressure-loss energy for conversion to electric power. (B) System for power recovery from reactors in chemical processing plants. (C) Typical waste-heat recovery system using expansion turbines to generate electrical power. (Used by permission: Bul. 2781005601. ©Atlas Copco Comptec, Inc.)

represents the comparison of throttling (Joule-Thompson Effect) of nitrogen with the use of an expansion turbine starting at the same conditions. Note that the expander as a general rule will produce a lower temperature for most

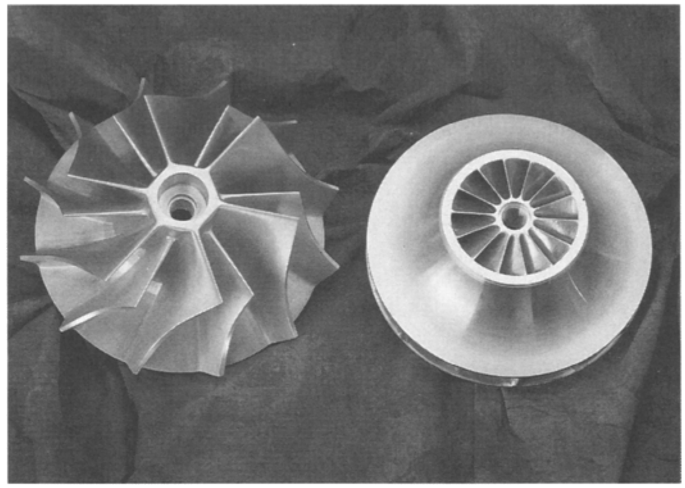

Figure 11-60. Typical closed and open impeller designs for use in a mechanical expansion turbine. (Used by permission: Bul. 2781005601. ©Atlas Copco Comptec, Inc.)

gases than valve throttling when the final pressure is the same. Each gas needs to be examined separately for the specific operating conditions. Pressure reduction refrigeration is used extensively in the natural gas recovery operation for production of low temperatures for vapor condensation and local power recovery. See Figures 11-59A–C. Figure 11-60 illustrates typical impeller designs for mechanical expansion turbines.

The design of low-temperature systems, whether mechanical expansion turbine or throttling valve, is a special technology and cannot be adequately covered in this chapter. References on the subject include 20, 21, 23, 60.

Nomenclature

BHP = Bhp = brake horsepower.
btu = British thermal unit.
e_o = overall compression efficiency, fraction.
gpm = gallons/minute.
hp = horsepower.
h = enthalpy of a vapor or gas at a specific state of temperature and pressure, Btu/lb.
h_2(corr) = enthalpy corrected for overall efficiency and heat loss, Btu/lb.
k = ratio of specific heat, c_p/c_v.
K = equilibrium constant.
n = polytropic compression exponent.
P = absolute pressure, lb/in.^{2} abs, psia.
ΔP = pressure drop, lb/in.^{2}, psi.
Q = system heat loss, Btu/hr.

T = temperature absolute, °R = 460 + °F.
TR = ton of refrigeration.
t = temperature, °F.
t_e = saturation temperature of economizer vapor.
Δt = temperature difference, °F.
U = overall heat transfer coefficient, Btu/hr(ft^2) (°F).
v = specific volume, ft^3/lb.
W = system flow rate, lb/hr.
w = vapor flow, lb/min.
w_e = vapor from economizer, lb/min.
X_{c1} = composition of liquid being condensed at the top.
X_{c2} = composition of liquid being condensed.
X_e = mol fraction of the same component in the liquid in evaporator.
Y_e = mol fraction of one component in the evaporator vapor.

Subscripts

M = mixture.
e = economizer.
1,2,3, etc. = reference points to specific conditions of a system.

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Chapter 12

Compression Equipment (Including Fans)

Compression of gases and vapors is an important operation in chemical and petrochemical plants. It is necessary to be able to specify the proper type of equipment by its characteristic performance. The compression step is conveniently identified for the process design engineer by the principal operation of the equipment:

1. Reciprocating
2. Centrifugal
3. Rotary displacement
4. Axial flow

Compression may be from below atmospheric as in a vacuum pump or above atmospheric as for the majority of process applications. The work done by Scheel[47,77,138] is useful.

The purpose of this chapter is to acquaint the process and mechanical engineer with the basic details of reciprocating, centrifugal, and other major types of process compressors. Determining and specifying the required process performance and mechanical requirements, including the corrosive and hazardous nature and the moisture content of the fluids (gases/vapors) to be compressed, is important.

General Application Guide

Figures 12-1A, 12-1B, 12-1C, and 12-1D present a general view of the usual ranges of capacity and speed operation for the types of compression equipment listed. For pressure conversions of Figure 12-1B, 1000 psi = 6.8947 mpa.

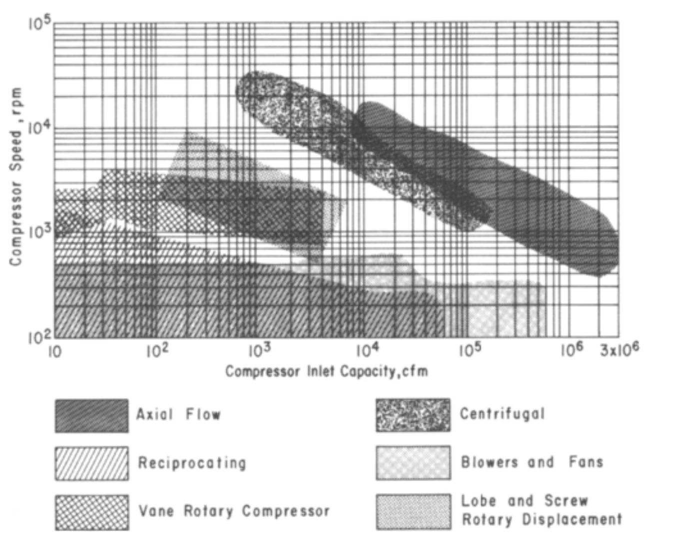

Figure 12-1A. General areas of compressing equipment applications. (Used by permission (with changes and additions by this author): Des Jardins, P. R. *Chemical Engineering,* V. 63, No. 6, ©1956. McGraw-Hill, Inc., New York. All rights reserved.)

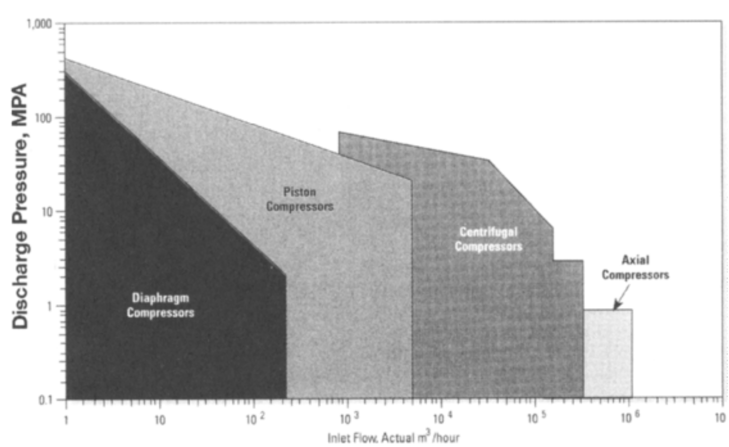

Figure 12-1B. Approximate ranges of application for usual process reciprocating, centrifugal, diaphragm, and axial-flow compressors used in chemical/petrochemical processes. Note that ethylene gas reciprocating compressors in the low-density, high-pressure process can reach 50,000 to 65,000 psi; 1,000 psi = 6.8947 mpa; for example, 500 mpa = 72,519 psi (some additions by this author). (Used by permission: Livingston, E. H. *Chemical Engineering Progress,* V. 89, No. 2, ©1993. American Institute of Chemical Engineers, Inc. All rights reserved.)

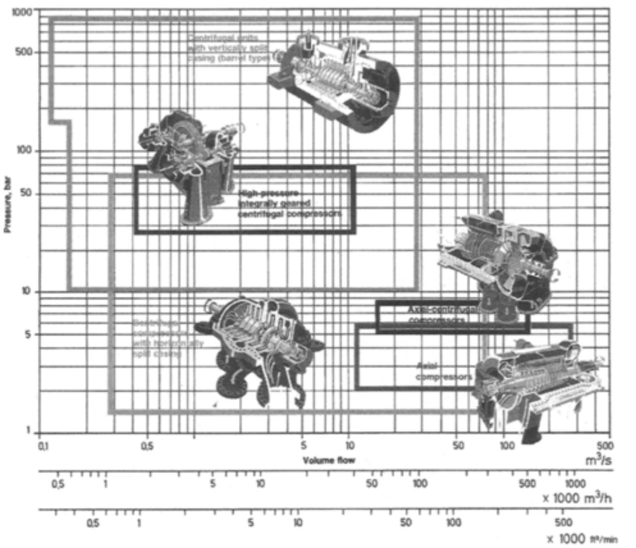

Figure 12-1C. Typical application ranges for turbocompressor capabilities extend over wide ranges of volume flow and pressures. Note: Bar. × 14.50 = psi. (Used by permission: Nissler, K. H. *Chemical Engineering,* V. 98, No. 3, p. 104, ©1991. McGraw-Hill, Inc. All rights reserved.)

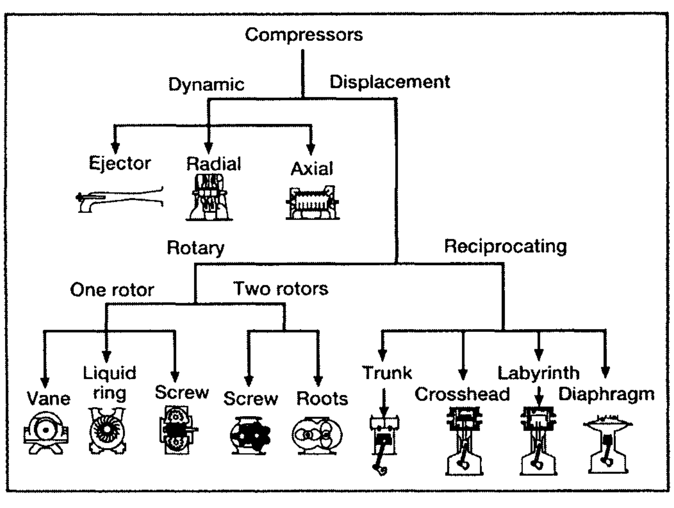

Figure 12-1D. Basic compressor types. (Used by permission: Coker, A. K. *Hydrocarbon Processing,* V. 73, No. 7, p. 39, ©1994. Gulf Publishing Co., Houston, Texas. All rights reserved.)

Table 12-1 outlines the compression limits for this type of equipment. The value of the chart and table is to aid in establishing the probable types of equipment suitable for an operation. However, as in many other process situations, equipment is designed to handle special cases that might not be indicated by the guide. Usually inlet cfm, temperature, and pressure, as well as the outlet conditions and

Table 12-1
General Compression and Vacuum Limits

Compressor Type	Approx. Max. Commercially Used Disch. Press., psia	Approx. Max. Compression Ratio per Stage	Approx. Max. Compression Ratio per Case or Machine
Reciprocating	35,000–50,000	10	as required
Centrifugal	3,000– 5,000	3–4.5	8–10
Rotary displacement	100– 130	4	4
Axial flow	80– 130	1.2–1.5	5–6.5

Vacuum Pump Type	Approx. Suction Pressure Attainable, mm Hg abs
Centrifugal	6
Reciprocating	0.3
Steam jet ejector	0.05
Rotary displacement	10^{-5}
Oil diffusion	10^{-7} (or 10^{-4} micron)
Mercury or oil diffusion plus rotary	less than 10^{-7}

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nature of the fluid, are all involved in identifying the equipment type best suited for the application. See Monroe,[40] Huff,[34] and Patton[43] for comparison. Also see Leonard.[71]

Specification Guides

Compressor cylinders or other pressure-developing mechanisms are never designed by the process companies involved in their operation, except in rare instances in which special know-how is available or secret process information is involved. In the latter case, the process company might have the compressor cylinders (or compression components) built in accordance with special plans, purchase standard frames or housings, and then assemble the driver, cylinder, and packing at the plant site.

Usually the selection of the basic type of compression equipment for the operation can be determined prior to inquiring of the manufacturers. However, when in doubt or where multiple types may be considered, inquiries should be sent to all manufacturers offering the equipment.

Preparation of complete and appropriate specifications is of paramount importance in obtaining the proper performance rating as well as price considerations. Preliminary design rating calculations are usually prepared as guides or checks. The final and firm performance information is obtained from the manufacturer of the specific equipment. No standards of design exist among manufacturers; therefore, the performance will vary according to the details of the specific equipment. All performance will be close to the requirement, but none may be exact. This is the point where knowledge of compressor types and details is important to the engineer. Bid evaluations must include detailed analysis of performance, power drivers, and materials of construction.

General Considerations for Any Type of Compressor Flow Conditions

In establishing specifications, the first important items to identify from the plant process material balance are normal, maximum, and minimum intake or suction flow rates together with corresponding conditions of temperature and pressure. The required discharge pressure must be established. If it is necessary or important to be able to operate at reduced or over-normal flow rates, these should be identified for the manufacturer, together with the length of time of such expected condition; e.g., full time at one-half rate, 20 minutes out of every hour at 10% over normal, etc. These operating requirements may separate the types of equipment. Because it is uneconomical to purchase horsepower that cannot be used by the fluid system, ask that the manufacturer state the maximum load and/or conditions that will fully load the available horsepower of the compressor-driver unit.

Fluid Properties

Fluid properties are important in establishing the performance of compression equipment. Whenever possible, fluid analysis should be given, and where this is not available due to lack of complete information or secrecy, close approximations are necessary. Under these last conditions, actual field performance may not agree with the design data due to the deviation in values of the ratio of specific heats and the average molecular weight. Identify, as to composition and quantity, any entrained liquids or solids in the gas stream. No manufacturer will design for entrained liquids or solids, although some machines will handle "dirty" gases. Solids are always removed ahead of any compression equipment, using suitable wet- or dry-scrubbing equipment, and liquid separators are recommended for any possibilities of liquid carryover.

Compressibility

Gas compressibility has an important bearing on compressor capacity performance. Therefore, it is good practice to state compressibility values at several temperature and pressure points over the compression range under consideration. When possible, a compressibility curve or reference thereto is included in the inquiry. Where specific information is not available, but compressibility is anticipated as being a factor to consider, approximate values should be established and so presented for further study by the manufacturer.

Compressibility is expressed as the multiplier for the perfect gas law to account for deviation from the ideal. At a given set of conditions of temperature and pressure:

$$PV = ZNRT \tag{12-1}$$

where Z = compressibility factor, usually less than 1.0
N = number of lb-mol
R = gas constant, depends on units of pressure, volume, and temperature
T = absolute temperature, °R = °F + 460
P = pressure, absolute, psia
V = volume, ft^3 (see paragraph to follow)

Gas volumes are corrected at the intake conditions on the first and each succeeding stage of the compression step, and compressibility factors are calculated or evaluated at these individual intake conditions. Some manufacturers use the average value between intake and discharge conditions.

Corrosive Nature

Corrosive fluids or contaminants must be identified to the manufacturer. The principle gas stream may or may not be corrosive under some set of circumstances, yet the contaminants might require considerable attention in cylinder design. For example, considerable difference exists between handling "bone-dry" pure chlorine gas and the same material with 5 ppm moisture. The corrosiveness of the gas must be considered when selecting fabrication materials for the compression parts as well as seals, lubricants, etc.

Moisture

Moisture in a gas stream might be water vapor from the air or a water scrubber unit, or it could be some other condensable vapor being carried in the gas stream. It is important in compressor volume calculations to know the moisture (or condensable vapor) condition of the gas.

Special Conditions

Often the process may have conditions that control the flexibility of compression equipment selection. These might include limiting temperatures before polymer formation, chemical reaction, excess heat for lubrication materials, explosive conditions greater than a certain temperature, etc.

Any limiting pressure drops between stages should be specified, in which the gas and vapors are discharged from one stage, pass through piping, cooling equipment and/or condensate knock-out equipment, and are then returned to the next higher stage of the compression process. Usually a reasonable figure of 3–5 psig can be tolerated as pressure drop between stages for most conditions. The larger this drop is, the more horsepower required. Special situations might hold this figure to 0.5–1 psig.

Reciprocating Compression

Mechanical Considerations

Fundamental understanding of the principles involved in reciprocating compression is important for proper application of compressors to plant problems. The reciprocating compressor is a positive displacement unit with the pressure on the fluid developed within a cylindrical chamber by the action of a moving piston. Figures 12-2A–U illustrate the assembly and arrangements of typical cylinders for various pressure ranges and types of services. Figures 12-3A and 12-3B show cross-sections of a cylinder and crankshaft arrangement for two different styles of compressors.

Compressor types, components, and arrangements are designated as:

A. Cylinders

1. *Single Acting:* Compression of gas takes place only in *one end* of the cylinder. This is usually the head end (outboard end), but may be the "crank" end (inboard end or end of cylinder nearest crankshaft of driving mechanism). See Figures 12-2A, 12-3A, and 12-4A.

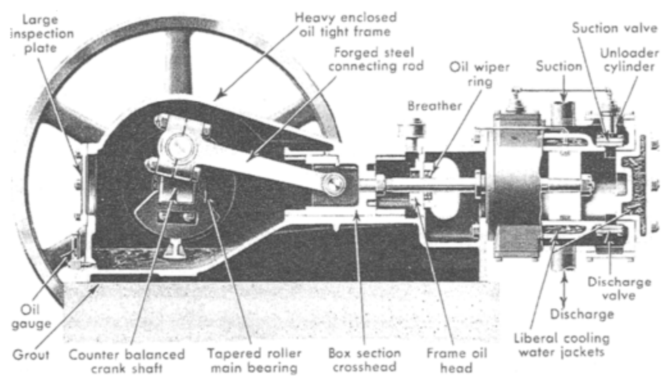


Figure 12-2A. Sectional assembly, Worthington single-stage, belt-driven air compressor showing construction of air cylinder and running gear. (Used by permission: Bul. L-600-B9. Dresser-Rand Company.

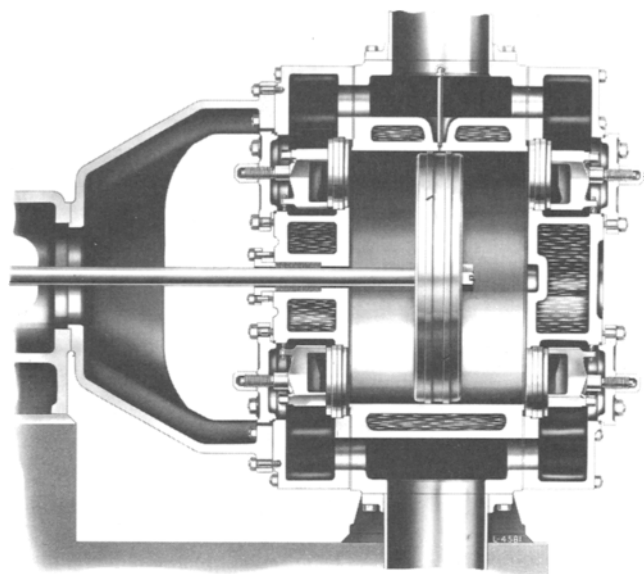


Figure 12-2C. Dry vacuum pump cylinder for very low absolute suction pressures. Valves in heads for low clearance and high volumetric efficiency. (Used by permission: Bul. L-679-BIA, ©1957. Dresser-Rand Company.)

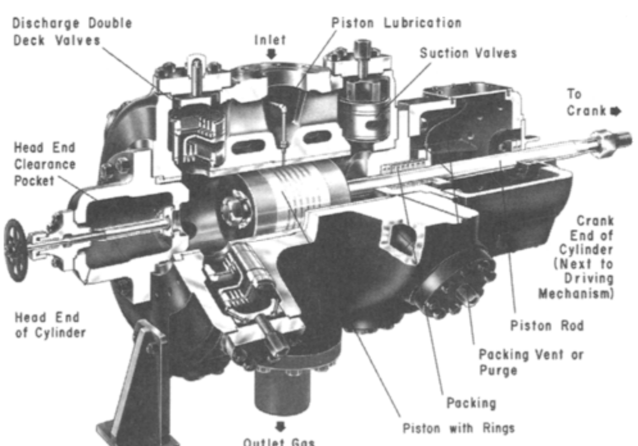


Figure 12-2B. Cut-a-way view of typical high-pressure gas cylinder showing double-deck feather valves in place. (Used by permission: Dresser-Rand Company.)

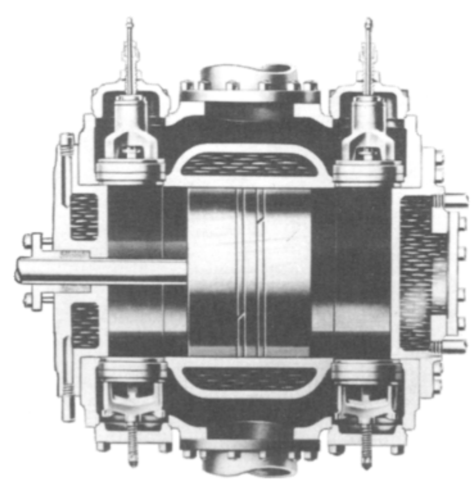


Figure 12-2D. Standard air compressor cylinder for 125 psig discharge pressure. Suction valve unloaders for automatic capacity control. (Used by permission: Bul. L-679-BIA,©1957. Dresser-Rand Company.)

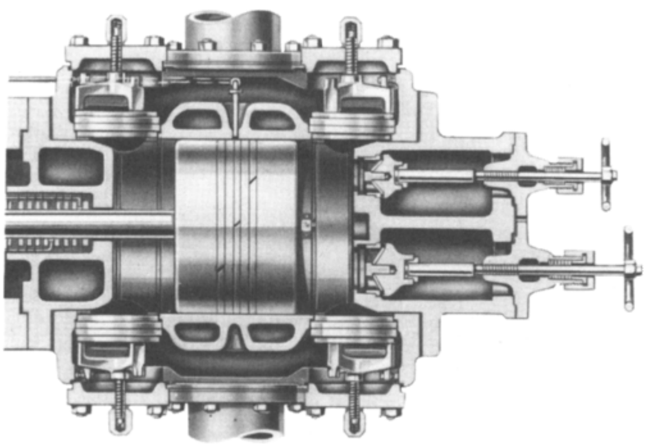


Figure 12-2E. 250 psig working pressure cylinder used in refrigeration service. Auxiliary stuffing box for added sealing on shutdown. Manual fixed volume clearance pockets for capacity control. (Used by permission: Bul. L-679-BIA, ©1957. Dresser-Rand Company.)

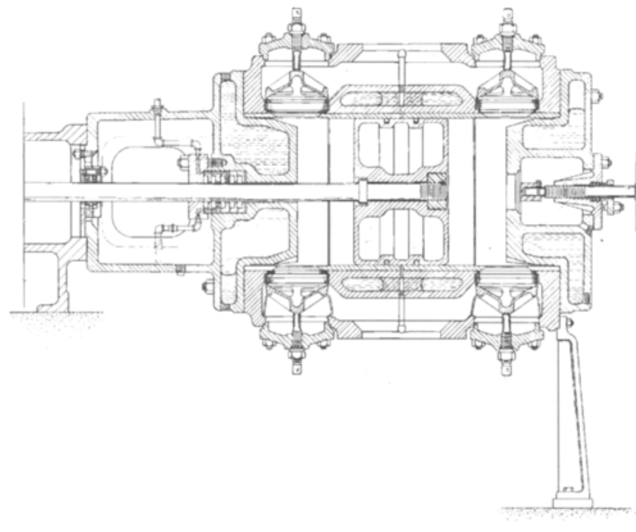


Figure 12-2F. Typical liner-type cast iron cylinder. (Used by permission: Ingersoll-Rand Company.)

TYPICAL LINER-TYPE CAST-IRON CYLINDER

CYLINDER barrel and heads are rugged castings made from an iron composition especially selected for the particular cylinder and service involved. Heads and barrel are thoroughly water-jacketed. Water piping is included from inlet valve to outlet sight-flow indicator.

DRY-TYPE LINER is of cast-iron, shrunk into the barrel, and extending for the full length of the cylinder. It is positively locked by the heads against end movement and by a threaded dowel against rotation.

PISTON is cast-iron, hollow for light weight and well ribbed for strength. The piston is locked on the rod between either a taper or a solid collar and a nut.

PISTON RINGS are normally cast-iron of the single-piece snap-ring type, although other materials may be used when conditions require.

PISTON ROD is carbon-steel, flame-hardened over packing travel area.

The rod is packed with full-floating metallic packing, force-feed lubricated and vented when the gas composition requires. Vented packing is illustrated.

DISTANCE-PIECE OPENING provides free access to packing.

ALL HEAD STUDS AND NUTS are external and accessible without removal of valves to reach internal bolting. There is no possibility of hidden internal leakage.

SUPPORT is provided at outer end (see sketch) to carry weight of cylinder directly to foundation. The piping need not be designed to support the cylinder.

CLEARANCE POCKET shown in the head is the manual fixed-volume type.

Recycle and other services frequently require absolute elimination of lubricating oil contamination of the gas. The cylinder must operate with no oil.

2. *Double Acting:* Gas compression takes place in both ends of the cylinder, head end and crank end, Figures 12-2I, 12-2J, and 12-4B.

Note that the crank end (Figure 12-2F) always has the piston rod running through it, while the head end usually does not, but may, if a tail rod (Figure 12-2L) is used.

B. Frames

The cylinders are arranged on the main frame of the compressor to provide balanced crankshaft power loading (when possible), access for maintenance, piping convenience, and floor space to suit plant layout. Common designations by position of the cylinder are

1. Horizontal.
2. Vertical.
3. 90° angle, cylinders mounted both vertically and horizontally from same crankshaft, Figure 12-5A.
4. V or Y angle, Figure 12-5A.
5. Radial.
6. Duplex, cylinders mounted in parallel on two separate frames from common crankshaft, Figure 12-5B.
7. Balanced Opposed, cylinders mounted opposite (180°) and driven off same crankshaft, Figures 12-5C and 12-5E. Also see Figure 12-5F; the following results are used by permission (Bul. PROM 635/115/95-II, Nuovo Pignone S. P. A.):
   - "Either zero or minimum unbalanced forces and moments on the foundation.
   - Minimum foundation size and expense.
   - Minimum drive-end peak torques, reducing drive train torsional stresses.
   - Reduce motor current pulsations and power costs.
   - Reduce harmonic torques on the foundation.

With a bearing between each crankthrow and an extra main bearing and an outboard bearing for extra support at the HHE's drive end, the result is minimum

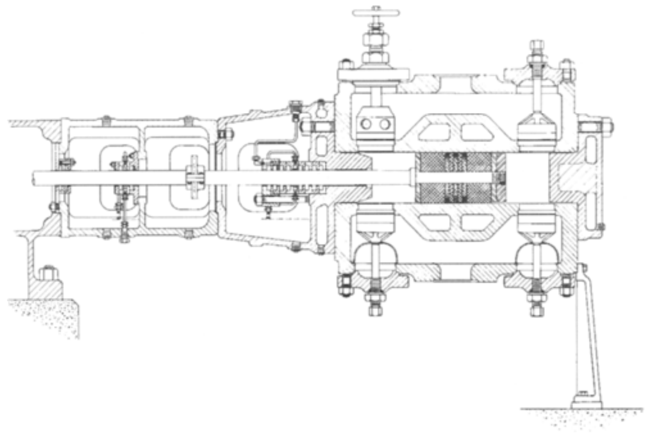


Figure 12-2G. Typical nonlubricated recycle cylinder. (Used by permission: Ingersoll-Rand Company.)

TYPICAL NON-LUBRICATED RECYCLE CYLINDER

CYLINDER barrel and heads are rugged castings and may be either cast-iron, nodular-iron, or cast-steel depending upon pressure and service requirements. In each case, water jackets are supplied, and piping is included from inlet valve to outlet sight-flow indicator.

DRY-TYPE CAST-IRON LINER is furnished in all nodular-iron or cast-steel cylinders for best wearing characteristics. Full cylinder-length liner is shrunk into the barrel and securely locked against any movement.

PISTON is an assembly of carbon rings held between steel end-plates. The piston is locked on the rod between a solid collar and a nut. When the bottom bearing surface of the carbon piston becomes worn, a new surface is easily made available by turning the assembly through an appropriate arc. This will be infrequent; the light solid piston design ensures maximum life.

PISTON RINGS are a special carbon material, of the segmental type, held to the cylinder wall by stainless-steel expanders.

PISTON ROD is carbon-steel, flame-hardened over packing travel area unless otherwise specified.

TWO-COMPARTMENT DISTANCE-PIECE in this illustration is one typical design for maximum protection in two directions. First, to prevent crankcase oil particles from reaching the cylinder, and second,to prevent any contamination of the crankcase oil by gas constituents. The compartment nearest the cylinder is of sufficient length to prevent any part of the rod traveling from one stuffing box into another. Furthermore, a baffle collar stops crankcase oil from creeping along the rod. Normally, in this design, the distance-piece toward the cylinder is enclosed with solid covers over the access openings and has provisions for venting or purging. The section on the crankcase end is open.

FULL-FLOATING PACKING RINGS in the cylinder and middle partition are of a special carbon material. Only the cylinder packing is vented unless a positive pressure is to be held in the cylinder end compartment, in which case the partition packing is also vented. They are vented in both cases in this design.

Considerable heat is generated in the cylinder packing, particularly during break-in periods. It has been found desirable to remove this heat, which might otherwise result in an overheated and warped rod. Water-cooling is used, the water being circulated through a special packing cup.

SUPPORT is provided at outer end (see sketch) to carry weight of cylinder directly to foundation. The piping need not be designed to support the cylinder.

THE CHANNEL VALVE, as designed especially for nonlubricated service.

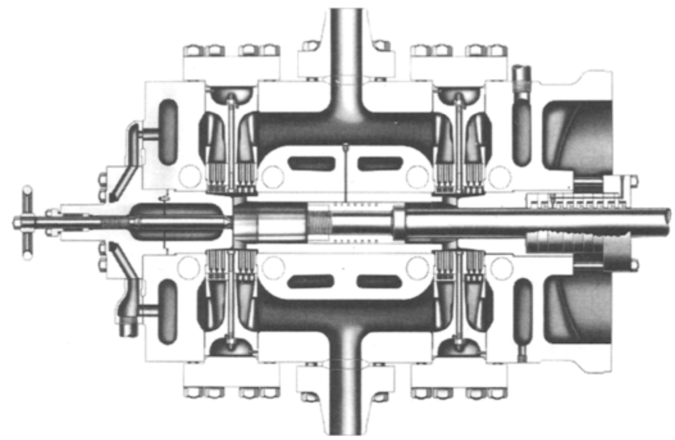


Figure 12-2H. Double-acting cast steel cylinder to 3,500 psi pressure. (Used by permission: Cooper-Cameron Corporation. All rights reserved.)

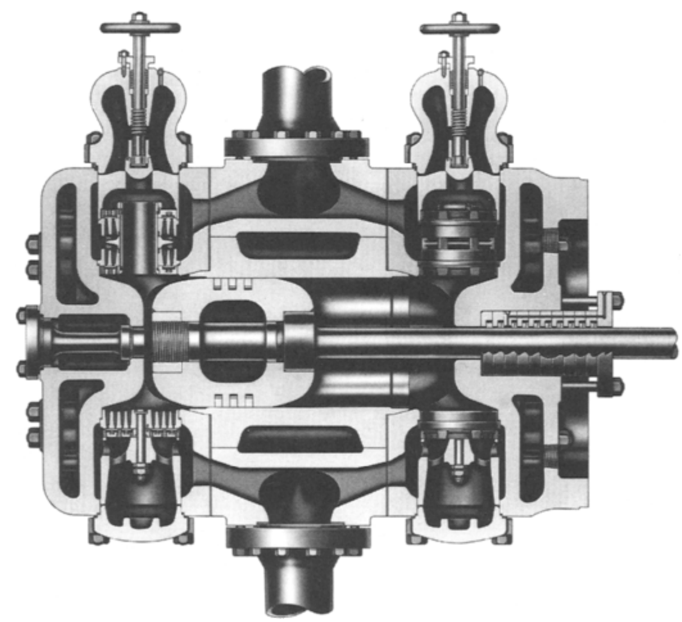


Figure 12-2I. Double-acting cast Meehanite or ductile iron cylinder to 1,250 psi pressure. (Used by permission: Cooper-Cameron Corporation. All rights reserved.)

crankshaft deflections, minimum crankshaft stress, minimum drive-end bearing loads and maximum crankshaft and bearing life.

For example, an HHE for a three cylinder application has three crankthrows set at 120°. "Piston weights may be balanced or balance weights added to the active crossheads to obtain zero unbalanced primary forces. By comparison, a fixed angle crankshaft requires four crankthrows with either an additional compression cylinder or a balance weight dummy crosshead to obtain acceptable unbalanced forces."

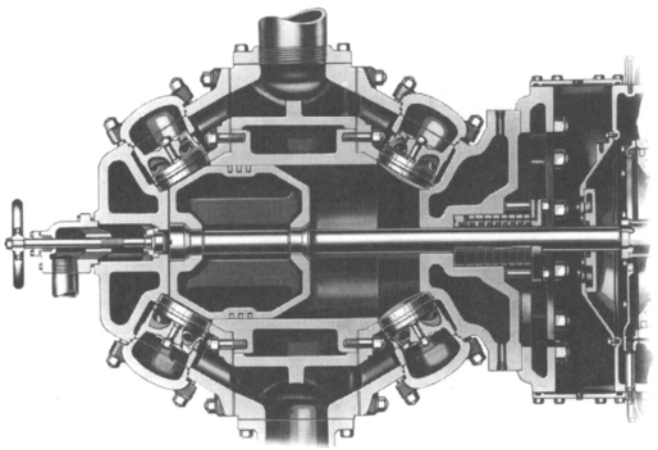


Figure 12-2J. Double-acting Meehanite metal or ductile iron cylinder to 1,000 psi pressure. (Used by permission: Cooper-Cameron Corporation. All rights reserved.)

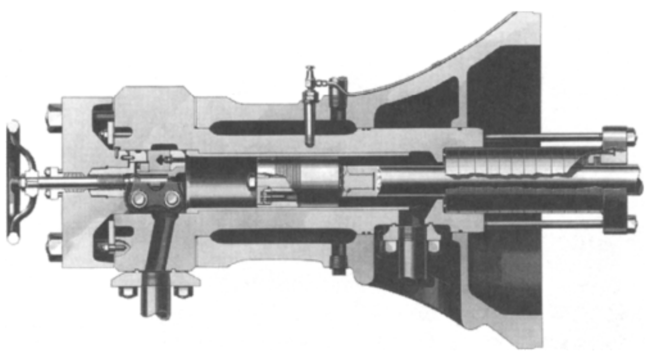


Figure 12-2K. Forged steel single-acting cylinder for 6,000 psi pressure. (Used by permission: Cooper-Cameron Corporation. All rights reserved.)

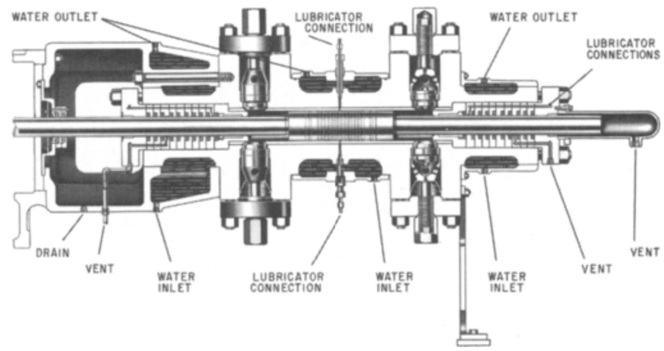


Figure 12-2L. Typical forged steel cylinder with tail-rod. (Used by permission: Ingersoll-Rand Company.)

TYPICAL FORGED-STEEL CYLINDER WITH TAIL-ROD

Construction of forged-steel, tail-rod cylinders (other than circulators) will be substantially as shown here.

CYLINDER barrel with integral packing boxes is a single steel forging of a material especially selected for the design, pressure, and service requirements of each case. There are no heads as such; the forged-steel nose-pieces of the piston and tail-rod packing act as closures. Both cylinder barrel and stuffing boxes are water-jacketed. Water piping is included from inlet valve to outlet sight-flow indicator.

Alignment of packing boxes and cylinder bore is ensured because all three are bored at a single machining setup. This is an important factor in packing life.

DRY-TYPE LINER is a special cast-iron, full length of the cylinder, shrunk in place, and securely locked against any movement.

PISTON is usually cast-iron, locked on the rod between a solid collar and a nut. If a cylinder size is such that sufficient piston wall thickness is not available, the piston and rod are forged integrally as shown, and special inserted rider rings are included to improve wearing qualities.

PISTON RINGS are of the single-piece, snap-ring type.

PISTON-ROD AND TAIL-ROD are one piece of carbon-steel, flame-hardened over packing travel area. The rod is packed with full-floating metallic packing, force-feed lubricated, and vented when the gas composition requires. Vented packing is shown.

DISTANCE-PIECE OPENING provides free access to frame end packing.

PACKING CASES are completely contained and supported in full depth boxes in cylinder. Unequal tightening of flange bolts cannot destroy the alignment.

SUPPORT is provided at outer end (see sketch) to carry weight of cylinder directly to foundation.

VALVES cushioned type valves.

This design is used in 3,000–15,000 psi ammonia, methanol, and hydrogenation plant service.

The minimum number of cylinders are not "locked" into even numbers to handle compression problems. The minimum number required are used; therefore, "balance weight dummy crossheads are not necessary."

8. Four-cornered, opposed, two cylinders mounted opposite (180°) at each end of crankshaft.
9. Tandem, two or more cylinders are on same compressor rod, or one cylinder may be steam operated as drive cylinder with second as compressing cylinder. May also be duplex or multiple tandem, Figure 12-5D. Also see Figure 12-5E.

C. Suction and Discharge Valves

Several types of valves used in compressor cylinders are shown in Figure 12-6. To function properly, a valve must seat uniformly and tightly, yet must not have "snap-action" on opening or closing. Until pressure builds up to the discharge point, the valve must remain closed, open at discharge pressure, and then reseat as the pressure in the cylinder drops below the discharge value. The same type of action is required for the suction valves.

Valves must be made of fatigue-resistant carbon or alloy steel or 18-8 stainless steel, depending upon the service. The 18-8 stainless and 12-14 chrome steel is often used for corrosive and/or high temperature service. Any springs, as in the plate-type valves, are either carbon or nickel steel. Valve passages must be smooth, streamlined, and as large as possi-

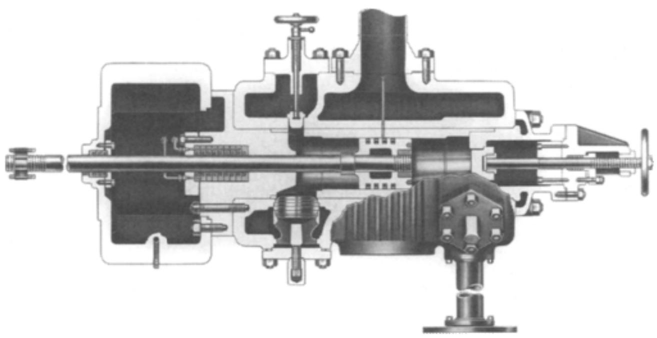


Figure 12-2M. For low compression ratios, designed for 1,000 psi discharge pressure equipped with hand-operated crank and head-end fixed clearance pockets for capacity control. Air-cooled, cast semi-steel, double-acting. (Used by permission: Dresser-Rand Company.)

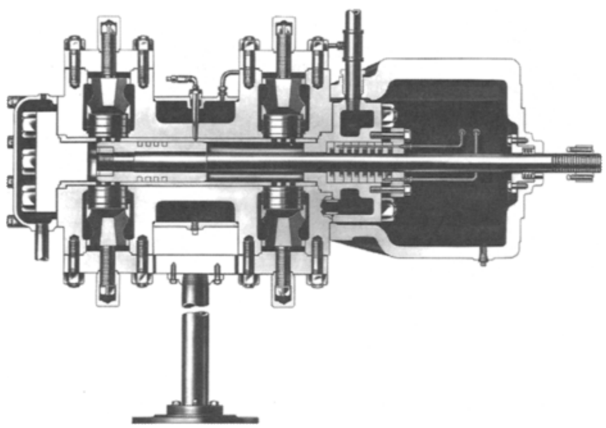


Figure 12-2N. Designed for working pressure up to 6,500 psi. A similar design cylinder is used for pressures in excess of 6,500 psi. Water cooled, forged steel, double-acting. (Used by permission: Dresser-Rand Company)

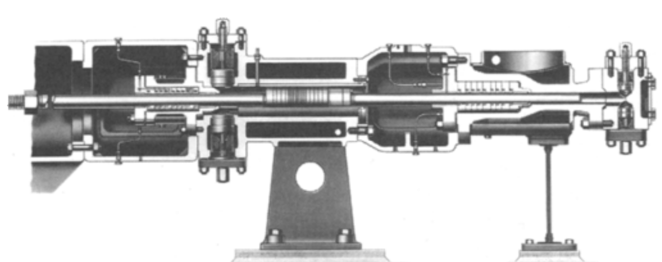


Figure 12-2O. Fourth- and fifth-stage cylinder assembly of 3,500 psig pressure hydrogen compressor. Opposed single-acting cylinders balance frame-bearing loading and minimize torque fluctuations. (Used by permission: Dresser-Rand Company.)

ble. Cylinder efficiency depends to a certain extent upon the proper selection and sizing of the valves. Valves must be adequately cooled, so provision is usually made for water jackets immediately adjacent to the valves, particularly the discharge valves.

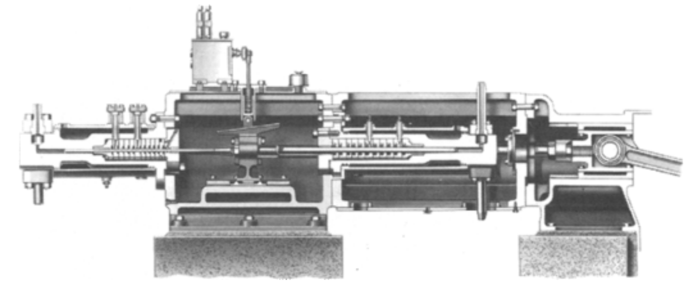


Figure 12-2P. Fifth- and sixth-stage cylinder assembly of 15,000 psi gas compressor. "Bathtub" design of intermediate cross-head permits use of short opposed plungers ensuring operating alignment. A design for pressures as high as 35,000 psi. (Used by permission: Dresser-Rand Company.)

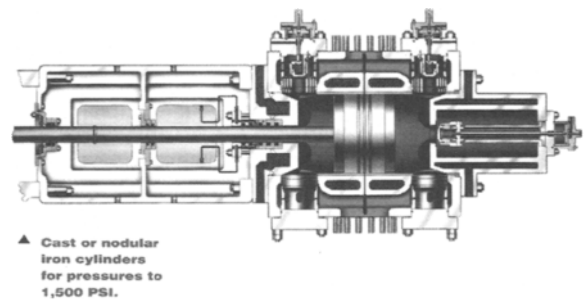


Figure 12-2Q. Cast or nodular iron cylinders for pressures to 1,500 psi. Note double "distance-pieces" (left, vented or purged, to prevent oil and process gas from leaking past the shaft; and right-end fixed clearance pocket). (Used by permission: Bul. 85084, ©1992. Dresser-Rand Company.)

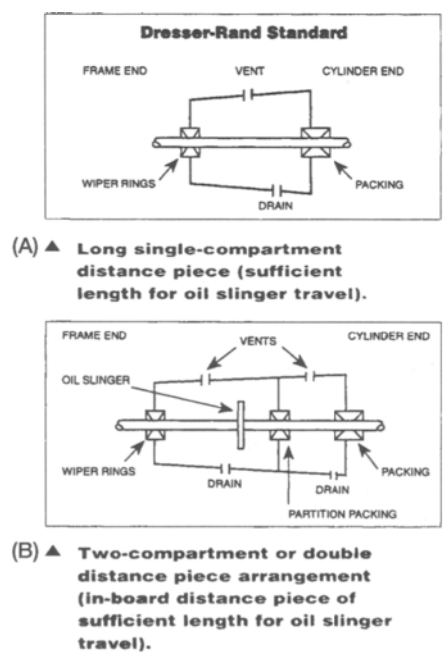


Figure 12-2QA–B. (A) Long, single compartment distance piece (sufficient length for oil slinger travel); (B) Two-compartment or double-distance piece arrangement (inboard distance piece of sufficient length for oil slinger travel). (Used by permission: Bul. 85084 ©1992. Dresser-Rand Company.)

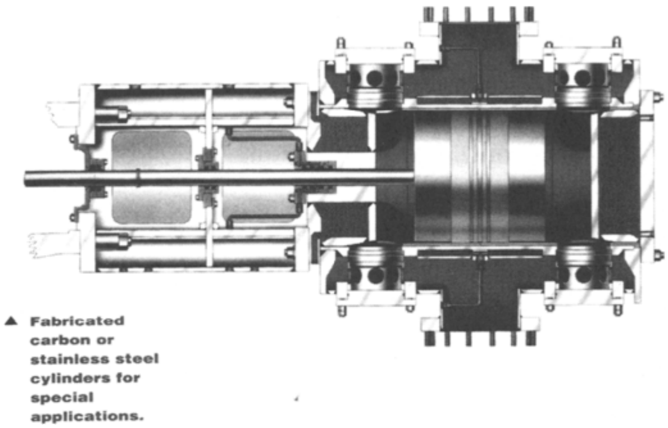


Figure 12-2R. Fabricated carbon or stainless steel cylinders for special applications. Note double-distance piece, no clearance pocket. (Used by permission: Bul. 85084, ©1992. Dresser-Rand Company.)

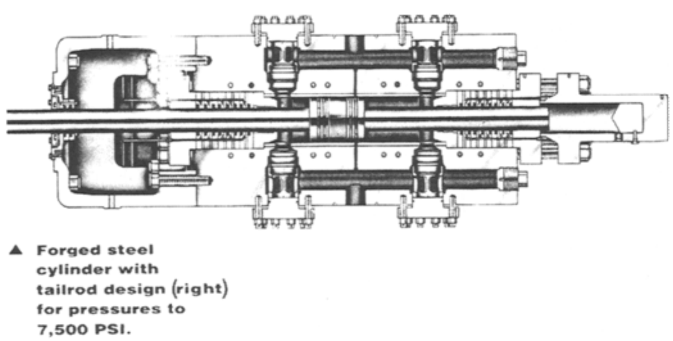


Figure 12-2S. Forged steel cylinder with tail-rod design (right) for pressures to 7,500 psi. (Used by permission: Bul. 85084, ©1992. Dresser-Rand Company.)

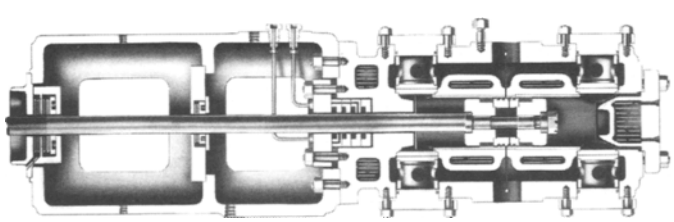


Figure 12-2T. Medium- or high-pressure, double-acting cylinder with flanged liner. The liner is locked in place by a flange between head and cylinder barrel. A step on the liner O.D. permits easy insertion. The cylinder may be made of cast-iron, nodular iron, or cast steel, depending on operating pressure. Note: Optional two-compartment distance piece (type D) designed to contain flammable, hazardous, or toxic gases is illustrated. (Used by permission: Bul. 33640, June 1985. ©Dresser-Rand Company.)

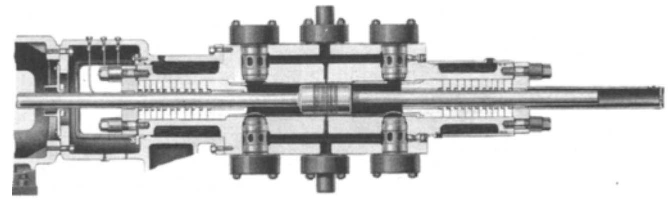


Figure 12-2U. High-pressure, circulator-type cylinder, double-acting. The steel cylinder and packing box are forged in one piece, and the one-piece piston and rod ensures positive alignment. The packing boxes are water cooled, and the packing is additionally cooled by internally circulated oil. Note tail-rod construction. (Used by permission: Bul. 3640, June 1985. ©Dresser-Rand Company.)

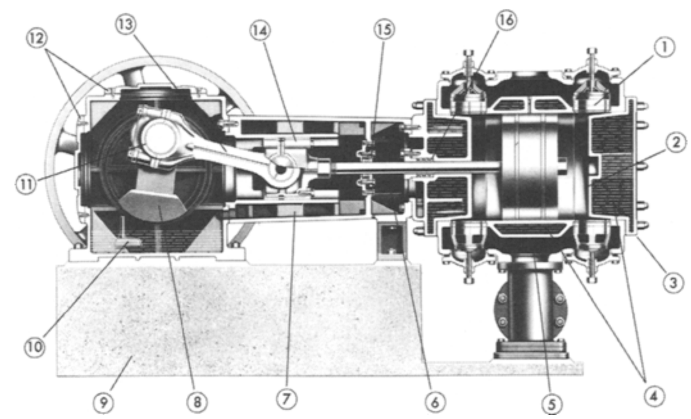


Figure 12-3A. Typical cross-section of motor-driven, single-stage compressor. (Used by permission: Ingersoll-Rand Company. All rights reserved.)

1. Valves.
2. Piston sealed by two single-piece snap rings. Rod threaded and locked into piston.
3. Cylinder head.
4. Cylinder barrel, head, and air passages water-jacketed for cooling.
5. Air passages.
6. Distance piece allows access to packing and oil-wiper rings.
7. Crosshead guide.
8. Counterweights are permanently bolted in place.
9. Foundation.
10. Screened oil suction.
11. Crankpin and main bearings.
12. Frame.
13. Die-forged steel connecting rod has rifle-drilled oil passage.
14. Crosshead pressure-lubricated through drilled passages in crosshead body. Piston rod threaded and locked into crosshead.
15. Wiper rings keep crankcase oil out of cylinder.
16. Full-floating metallic packing is self-adjusting.

Bauer[68] has studied losses in compressor cylinder performance associated with valve losses as they relate to overall efficiency. Bunn[69] examines poppet valves for retrofitting cylinders.

Double-deck valves reduce valve velocities in large diameter cylinders. With these valves, high clearance volumes and

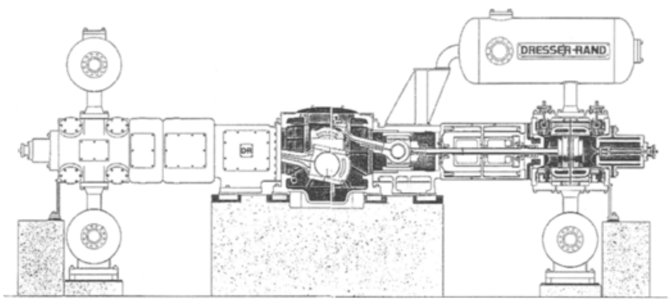


Figure 12-3B. Partial cross-section of balanced-opposed compression cylinders. (Used by permission: Bul. 85084, ©1992. Dresser-Rand Company. All rights reserved.)

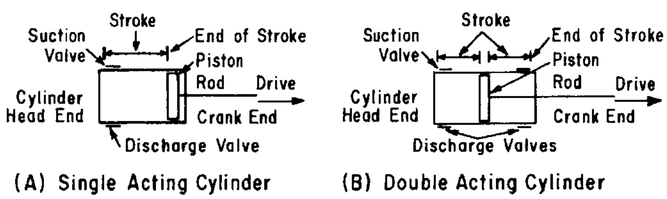


Figure 12-4. Cylinder action.

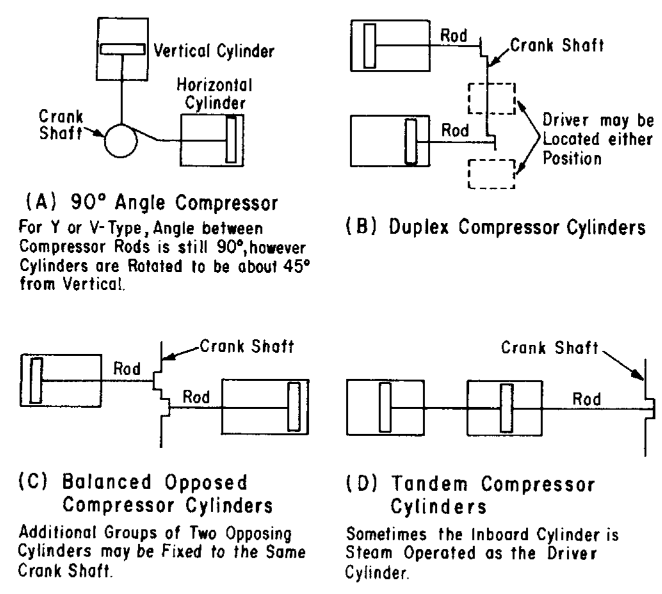


Figure 12-5A–D. Cylinder arrangement.

clearance pockets can be added to give additional unloading of a cylinder as designed to maintain proper loading on the driver. For a typical double-deck valve, the theoretical indicated horsepower loss may be 6% at a ratio of compression, R_c, of 3.0, and 17% at an R_c of 1.5.

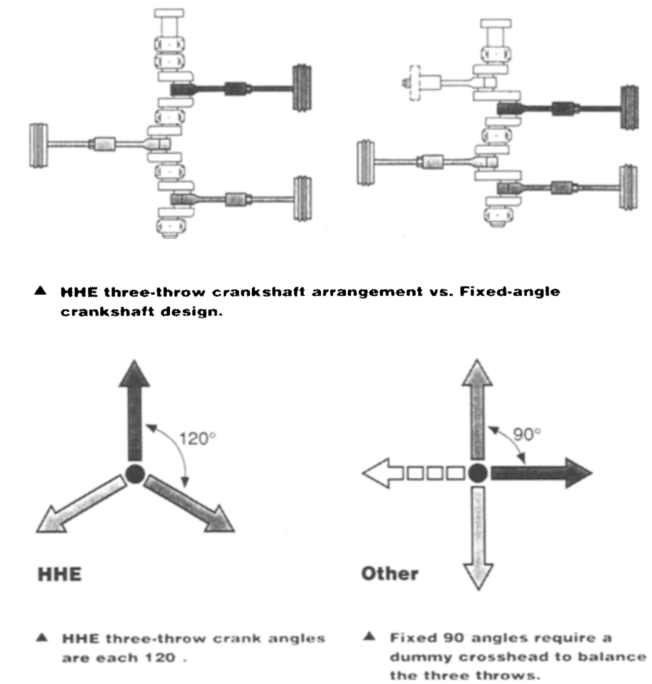


Figure 12-5E. Balanced arrangement for Dresser-Rand shaft system, 1–10 crank throws. (Used by permission: Bul. 85084, ©1992. Dresser-Rand Company.)

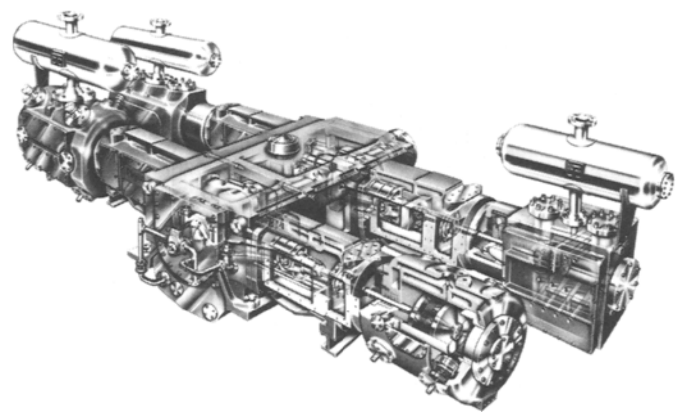


Figure 12-5F. Lubricated and nonlubricated balanced opposed process reciprocating compressors, designed to API 618 code. Fixed- and variable-speed drives using gas or diesel engines, steam or gas turbines, or electric motor. Note power drive to connect to right side of cross-head box in center. (Used by permission: Bul. PROM 635/115/95-II. Nuovo Pignone S. P. A., Florence, Italy; New York; Los Angeles; and Houston, Texas. All rights reserved.)

The valve plates, discs, and springs are mounted in a valve cage, which is inserted in the cylinder. Valve breakage occurs due to fatigue of the metal or improper action. This requires replacement and an evaluation of the materials of construction as well as the basic type of valve. Each manufacturer presents his valve design to match his equipment. Only experience can determine which type of valve works best in a given application

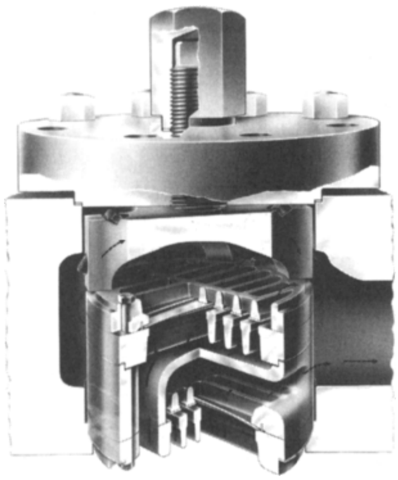


Figure 12-6A. Double-deck feather valve. (Used by permission Dresser-Rand Company. All rights reserved.)

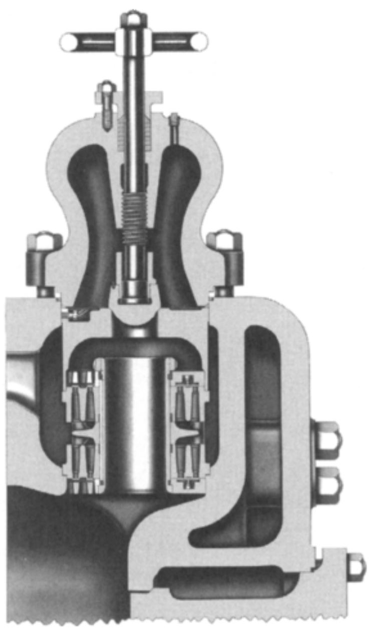


Figure 12-6B. Double-deck valve with valve cap unloader. (Used by permission: Cooper-Cameron Corporation.)

D. Piston Rods

See Figure 12-7. Piston rods are usually forged and hardened steel or alloy.

E. Piston

See Figure 12-8. Pistons may be of aluminum, built-up carbon or graphite, cast iron, cast steel, fabricated and metalized steel, stainless steel, or forged carbon or stainless steel. The selection involves the corrosive nature of the gas plus the weight-balancing problem of the compressor manufacturer.

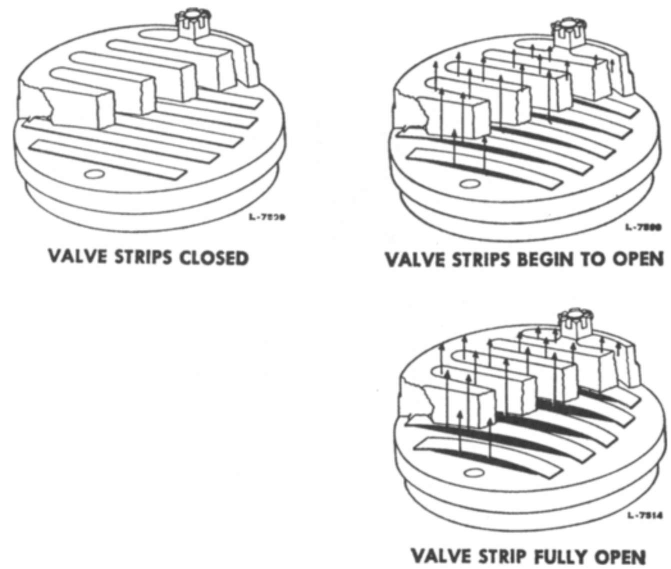


Figure 12-6C. Action of gas flow through strip-type feather valve. (Used by permission: Bul. S-550-B27. Dresser-Rand Comany.)

F. Piston Rings

See Figures 12-8 and 12-8A. Piston rings are rings mounted on the piston that seal against the cylinder wall and allow the piston to develop required pressures. Many types and materials are available. There are usually at least two rings per cylinder for low pressure applications; six or more for high-pressure services. Cast iron, bronze, micarta, aluminum, and carbon (graphite) are common ring materials.

G. Cylinders

Cylinders are made of materials consistent with pressure range and gas service. Sometimes a liner is used to recognize and allow for wear (or corrosion) or possible future changes in capacity. Liners may be graphite, aluminum, cast iron, steel, tungsten carbide, or other suitable materials (see Figure 12-2F). Most cylinders have water jackets to remove some heat of compression and to maintain reasonable cylinder and/or liner temperatures. Any heat removed is reflected in a slight reduction in the compression horsepower. The cooler cylinder walls usually allow more efficient lubrication of the cylinder. When oil cannot be tolerated in the presence of the gas, nonlubricated cylinders are used with graphite (or carbon) liners or piston rings.

H. Piston Rod Packing

See Figure 12-9. The pressure seal between the cylinder pressure and the crank case or atmosphere is maintained by a packing gland.

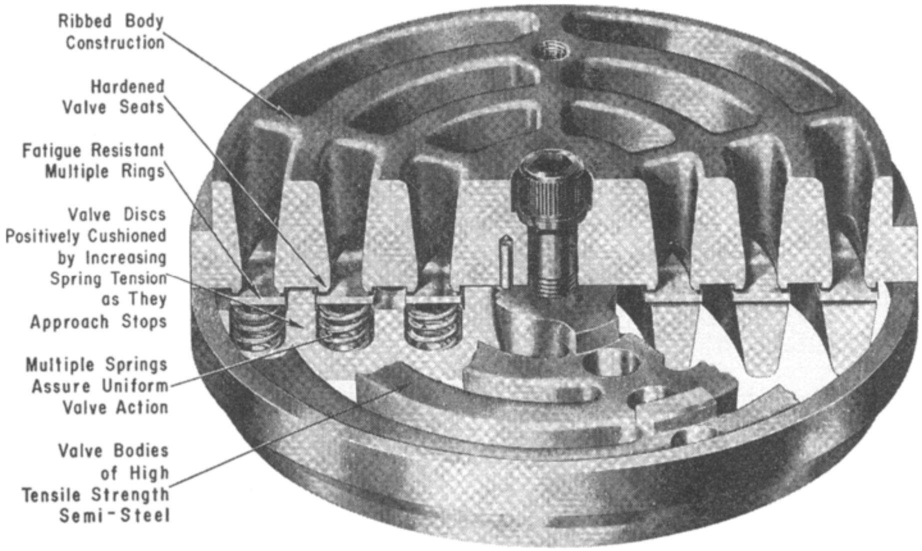


Figure 12-6D. Plate type valves. (Used by permission: Dresser-Rand Company.)

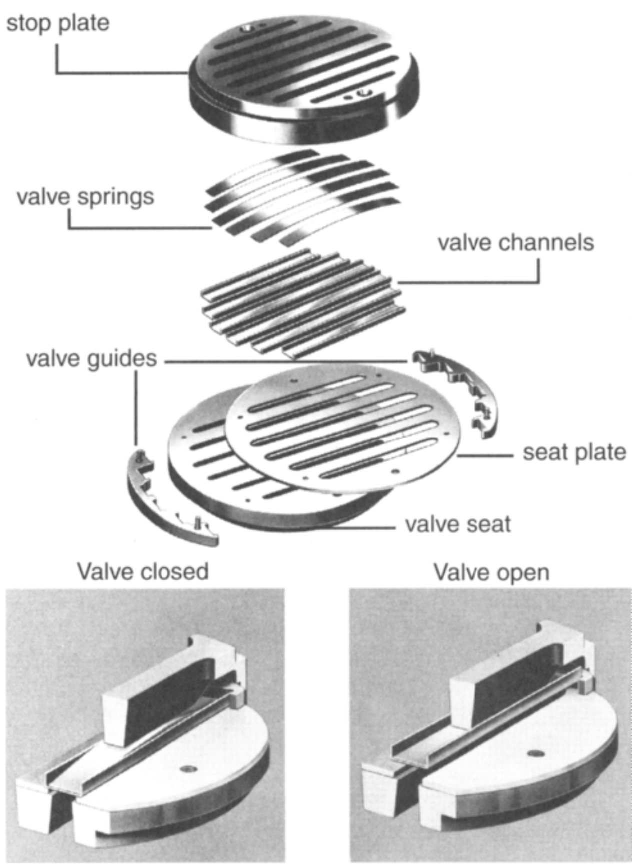


Figure 12-6E. Channel-type valves. (Used by permission: Ingersoll-Rand Company.)

Figure 12-6F. Ring channel valves. (Used by permission: Cooper-Cameron Corporation.)

The motion of the piston rod is reciprocating through this packing as contrasted to the rotating motion of the centrifugal compressor or pump shaft. In many applications, it is important to prevent any part of the shaft that has been inside the cylinder and exposed to the gas from being exposed to outside air. This is particularly important when handling such materials as hydrogen chloride, chlorine, hydrogen flouride, etc. This may be accomplished with a distance piece (Figure 12-2G.) This packing may be arranged for vent or purge in a manner similar to that for reciprocating shaft glands (Figures 12-2A and 12-2Q(A, B).

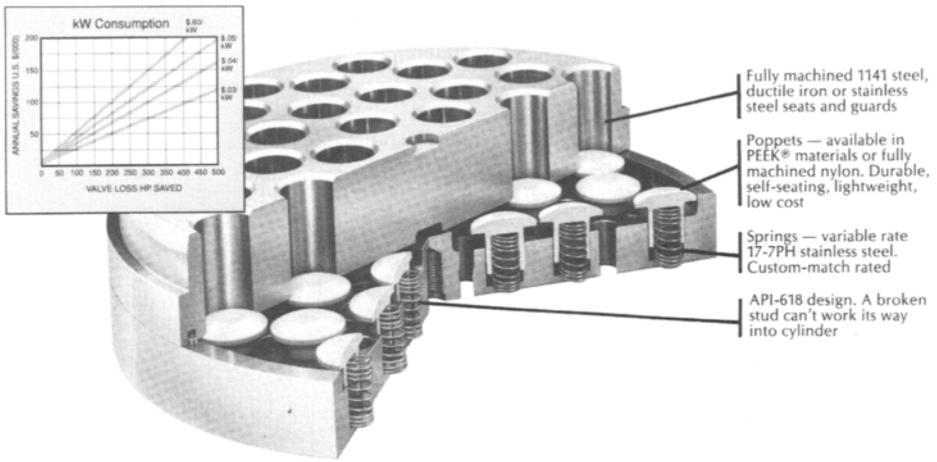


Figure 12-6G. AJAX® APV-1000, high-efficiency compressor valve. Suction and discharge losses are 4–8% compared to conventional valves with losses of 6–20% as a percentage of the total indicated horsepower. (Used by permission: Bul. 2-214. Cooper-Cameron Corporation, Cooper Energy Services, AJAX Superior. All rights reserved.)

A typical Worthington BDC plate valve in closed position. All passageways, lift clearances, springs and plates have been dynamically designed and individually selected for maximum flow efficiencies. The individual spring plate valve offers both efficiency and reliability advantages over valves that have flexing strips or plates.

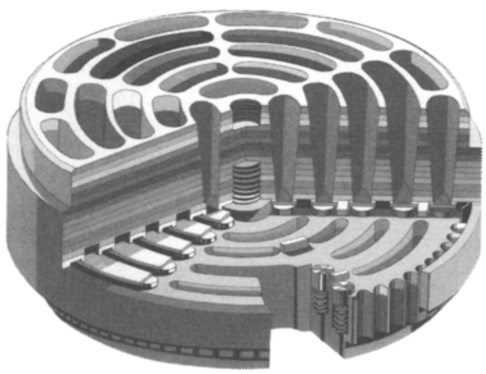


Figure 12-6H. Dresser-Rand specialized valve. (Used by permission: Bul. 3640. Dresser-Rand Company.)

Specification Sheet

Figure 12-10 is convenient for summarizing the main specifications to the manufacturer. Any unusual conditions must be explained, and minimum and maximum ranges must be established. "Normal" conditions *may* be the same as maximum, although this may not always be the case. Detailed driver specifications should be included unless the manufacturer is to make preliminary recommendations before final decisions are reached. Note that some of the data are to be furnished by the manufacturer, and the insistence on receipt of this information facilitates evaluation of competitive bids.

When packings are purged with air or other gas, the manufacturer should specify the quantity passing into the cylinder and out to the air. Accessibility of packing, bearings, and valves should be identified on drawings for review at the time of bid evaluation.

Performance Considerations

Cooling Water to Cylinder Jackets

Most installations use water-cooled compressor cylinder jackets; however, some use air cooling (usually small horsepower units), and a few use no cooling. For water cooling of the cylinder:

Heat Rejected to Water

Btu/Bhp (hr) Small Cyl. < 12 in. dia.	Btu/Bhp (hr) Large Cyl. > 20 in. dia.	Temperature Difference $t_c - t_w$
300	170	20
600	310	60
700	470	100

From reference 39, by permission.

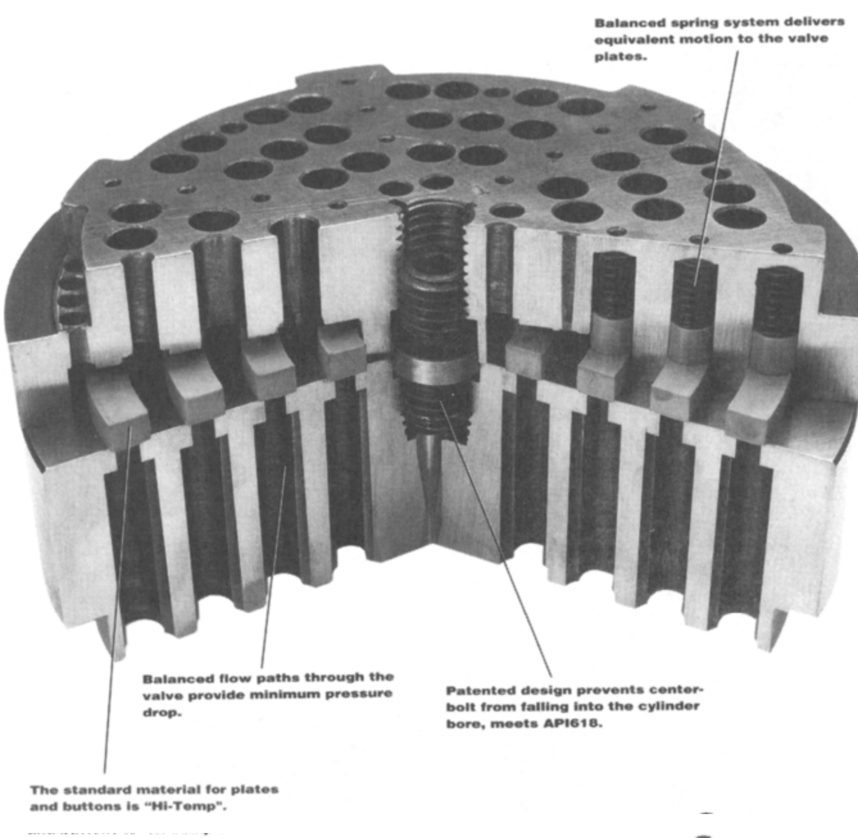
The usual temperature rise of the water is 10–15°F, and its inlet temperature to the cylinder is from 90–140°F, depending upon the manufacturer's design and properties of the gas.

The manufacturer can furnish complete data on temperatures for the particular design together with the quantity of water circulated and the pressure drop through the jackets.

This cooling water is usually arranged in a closed loop with the water being pumped through secondary coolers or over cooling towers and then returned to the jackets for reuse. Water quality must be good, with steam condensate being preferred, properly treated to prevent corrosion, etc.

Drivers

Refer to the chapter on drivers for mechanical equipment to obtain specification details. Reciprocating compressors are driven by the following:



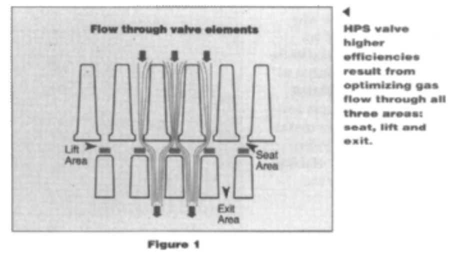


Figure 12-6I. Dresser-Rand HPS proprietary valve design, using proprietary blend of "PEEK," an advanced nonmetallic valve plate material, allowing for temperature ranges greater than previously available nonmetallic plate materials, for lubricated and nonlubricated applications for long life. (Used by permission: Form 85084, ©1992. Dresser-Rand Company.)

(I)

(II)

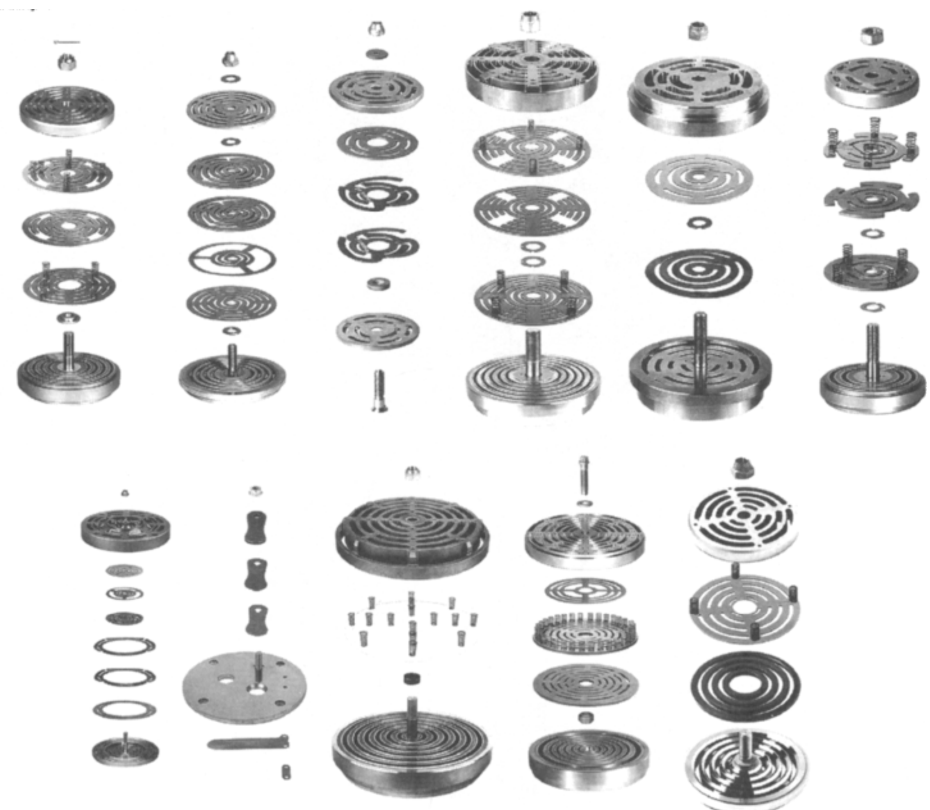


Figure 12-6J. Variety of standard and special valves designed and fabricated by the Hoerbiger Corporation. Many of these designs are used in compressor manufacturer's cylinder designs, and some valves have been designed by the compressor manufacturer. (Used by permission: Bul. V-100A, ©1993. Hoerbiger Corporation of America.)

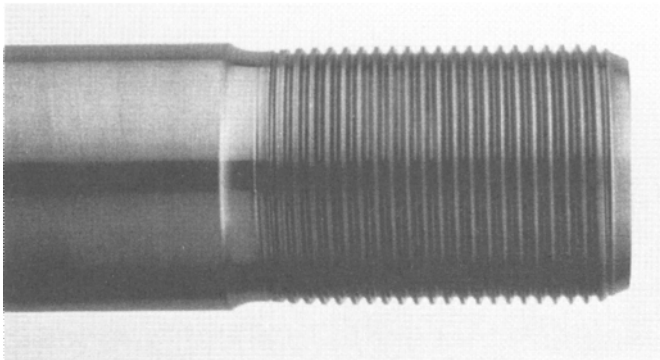


Figure 12-7. Piston rods. Precision-manufactured rolled threads and induction hardening provide high fatigue strength and long life in heavy duty service. Standard rod material is AISI 4142 carbon steel; other materials are available as required. Tungsten carbide coatings are also available for maximum surface hardness. Pistons are locked securely onto the rods. For higher pressure, smaller bore cylinders, the piston may be integral with the rod. (Used by permission: Bul. 85084, ©1992. Dresser-Rand Company.)

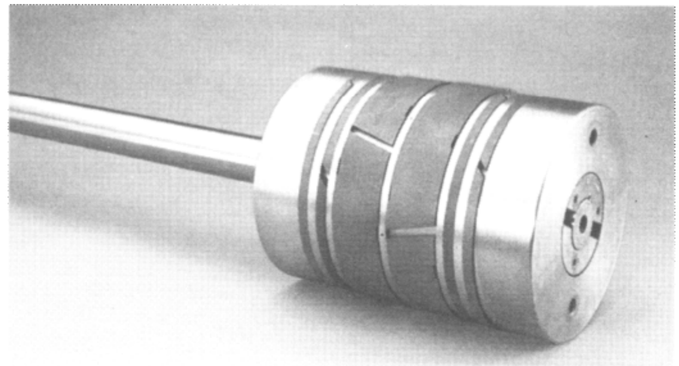


Figure 12-8. Pistons and rings. Lubricated and nonlubricated pistons with PTFE or other composite materials for the piston and rider rings. These designs prevent piston-to-bore contact and provide reliable service life, particularly during possible periods of lubrication interruption. (Used by permission: Bul. 85084, ©1992. Dresser-Rand Company.)

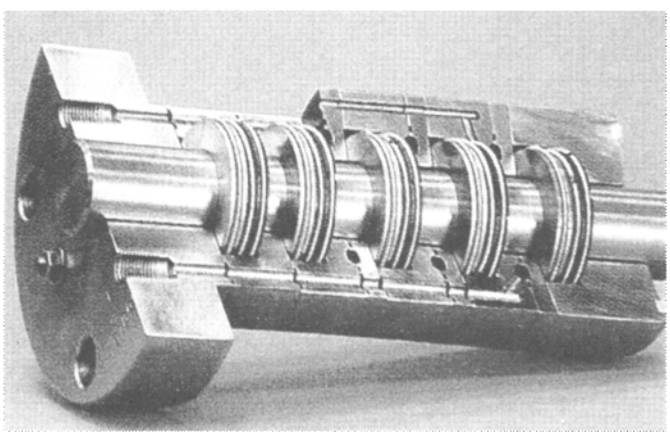


Figure 12-7A. Stuffing box with rod packing-direct and indirect cooling. (Used by permission: Bul. BCNA-3P100. Howden Process Compressors Incorporated.)

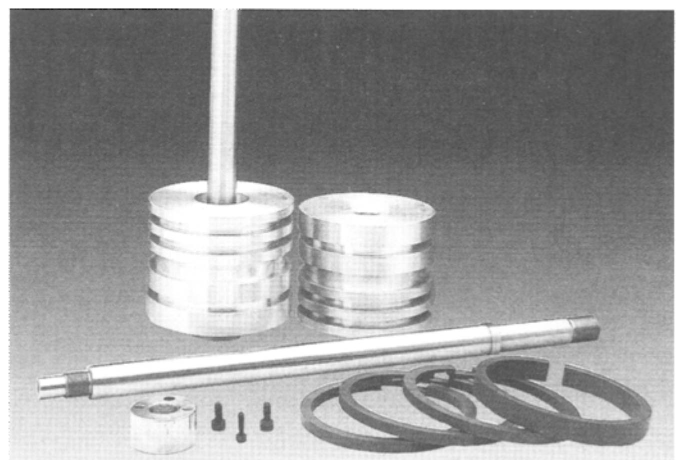


Figure 12-8A. Piston rings. The piston rod is manufactured from heat-treated stainless steel and is coated with wear-resistant overlays, such as ceramic, chromium oxide, and tungsten carbide applied by plasma techniques. Piston rod cross-head attachment has mechanical preloading system for the threads. Rider rings and seal rings are manufactured from PTFE filled resins; fillers are matched to the gas, piston speed, and liner specifications. Typical fillers are glass, carbon, coke, or ceramic. (Used by permission: Bul. BCNA-3P100. Howden Process Compressors Incorporated. All rights reserved.)

Electric motor:	Direct connected, constant or variable speed, belt, gear, fluid coupling.
Gas or diesel engine:	Usually direct connected to crankshaft, jackshaft, belts, or gears.
Steam turbine:	Gear (not a usual application).

Ideal Pressure—Volume Relationship

Although ideal conditions are not encountered in any compression operation, the actual condition is a series of particular deviations from this. Therefore, the theoretical ideal conditions can be practically considered as the build-

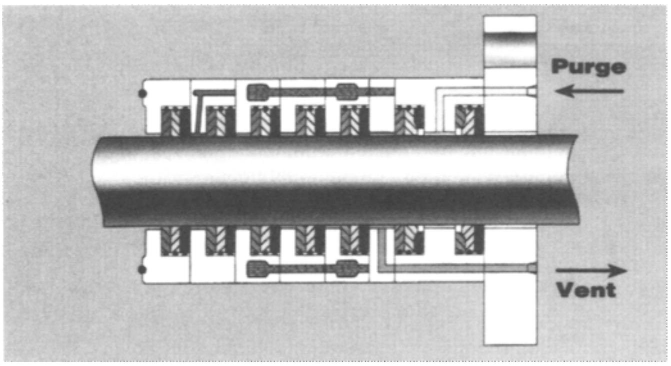


Figure 12-9. Piston Rod Packing. To meet the latest environmental requirements for controlling packing gas leakage, special buffer gas sealing rings are installed in place of conventional. The packing design includes two sets of wedge rings with an inert buffer gas between them. The spring load on the rings forces them to slightly wedge in the cup. Two individual wedge rings seal the buffer gas in both directions. Packing cups with passages for lubrication, coolant, and venting are provided as required by the application and API 618. Packings can be supplied with special gasket designs if stringent emission requirements exist. (Used by permission: Bul. 85084, ©1992. Dresser-Rand Company.)

ing block of the operation (Figure 12-11A and Figure 12-12). The compression operation stepwise is as follows.

Condition 1: Figure 12-12. Start of the compression stroke. The cylinder is full of gas at suction pressure and essentially suction temperature (neglecting valve loss). The piston moves during compression toward condition (2) with suction and discharge valves closed.

Condition 2: Figure 12-12. Start of gas discharge from cylinder. Gas has slightly exceeded the system pressure, and the discharge valve opens releasing gas to the system. The piston begins to sweep the gas from the cylinder as it moves toward condition 3.

Condition 3: Figure 12-12. Completion of gas discharge from cylinder. All the gas to be removed from the cylinder by the piston stroke has passed through the discharge valve. This also is the point where the return stroke of the piston starts, but *not* the beginning of the cylinder suction. As the piston starts its return stroke and the pressure in the cylinder is lowered slightly below discharge pressure, the discharge valve closes. The volume of gas left in the cylinder between the end of the piston and the end of the cylinder (clearance volume) expands from condition (3) to condition (4) as the piston returns.

Condition 4: Figure 12-12. Start of gas suction into cylinder. The cylinder pressure has dropped below the system suction pressure, and the suction valve opens to admit new gas into the cylinder as it returns to condition (1).

Actual Compressor Diagram

The actual compression diagram naturally deviates from the ideal; the extent varying with the physical characteristics of the cylinder and the properties of the gas, Figures 12-12, 12-12A, and 12-12B.

Deviations from Ideal Gas Laws: Compressibility

See Figures 12-13A–D.

The thermodynamic processes that may occur during a compression operation are[61]

Adiabatic—no heat added to or removed from the system
Isothermal—constant temperature
Isometric—constant volume
Isobaric—constant pressure
Isentropic—constant entropy
Isenthalpic—constant enthalpy

For any gas compression,

$$PV^n = \text{constant} = C \tag{12-2}$$

The "ideal" gas law or "perfect" gas law is a combination of Boyle's and Charles' laws for any compressible fluid (gas/vapor).

Based on the perfect gas and the adiabatic equation:

$$P_2/P_1 = (V_1/V_2)^k \tag{12-3}$$
$$T_2/T_1 = (V_1/V_2)^{k-1} \tag{12-4}$$
$$P_2/P_1 = (T_2/T_1)^{k/(k-1)} = r^{(k-1)/k} \tag{12-5}$$
$$PV = WR'T \tag{12-6}$$
$$PV = R_oT \tag{12-7}$$

where R_o = 1,545 ft-lb/lb-°R
P = pressure, lb/ft^2
V = volume, ft^3
T = temperature, abs, °R
R = universal gas constant, 1,545/(mol wt of gas)

Boyle's law: For constant temperature (isothermal, but not realized under actual conditions):

$$P_1V_1 = P_2V_2 = \text{constant} = C_1 \tag{12-8}$$

For the adiabatic condition of no heat lost or gained,

$$P_1V_1^k = P_2V_2^k \tag{12-9}$$
$$k = n$$

A reversible adiabatic process is known as isentropic.[59] Thus, the two conditions are directly related. In actual practice compressors generate friction heat, give off heat, have valve leakage and have piston ring leakage. These deviations

(Text continues on page 385)

SPEC. DWG. NO.
A-
Page ___ of ___ Pages
Unit Price
No. Units
Item No.

Job No. ____________

B/M No. ____________

RECIPROCATING COMPRESSOR SPECIFICATIONS

Service ____ Manufacturer ____ Model ____

Type ____ BHP ____ Fluid ____ Sat'd. with: ____

Avg. Mol. Wt. (Dry) ____ Design Speed ____ RPM. Speed Range ____ RPM

DATA per ____ NORMAL CONDITIONS

Stage					
Class or Type					
Cylinders: Diameter (Bore), Inches					
Stroke, Inches					
Piston Displacement: Cubic Ft. per ____					
Action of Cyl. Single or Double					
Actual Intake. CFM					
Delivery in Lbs./Hr. (Dry Basis)					
Intake Press. PSIA					
Intake Temp. °F					
Disch. Press. PSIA					
Disch. Temp. °F					
Ratio of Compression					
Compressibility Factor @ ____ PSIG & ____ °F					
Ratio of Specific Heats, Cp/Cv					
Specific Volume @ Intake Conditions					
Volumetric Efficiency @ Suct., %					
Normal Clearance, %					
Cyl. Test Press. PSIG					
Press. Drop Allowed Between Stages					
Suction Nozzle Size					
Suction Nozzle Rating & Facing					
Disch. Nozzle Size					
Disch. Nozzle Rating & Facing					
Material: Cylinder					
Liner					
Heads					

Unloading Facilities (Pockets, Valve Lifters) ____

Weights and Unbalanced Forces ____

Driver: Type ____ Mfgr. ____ H.P. ____ RPM ____

Volts ____ Phase ____ Cycle ____ Frame ____

Steam: Inlet ____ PSIA @ ____ °F. Exhaust ____ PSIA @ ____ °F. Steam Rate ____ Lbs./Hr.

Fuel: ____ Heat Value ____ BTU. Full Load Consumption ____ BTU/BHP. Hr.

Heat Rejection: Cooling Water Available @ ____ °F and ____ PSIG. Type ____

Comp. Cyl. Jacket: ____ BTU/Hr. Temp. In ____ °F. Temp. Out ____ °F. Water Req'd. ____ GPM. Δp ____ PSI

Lube Oil Cooler: ____ BTU/Hr. Temp. In ____ °F. Temp. Out ____ °F. Water Req'd. ____ GPM. Δp ____ PSI

REMARKS

By	Chk'd.	App.	Rev.	Rev.	Rev.
Date					

P.O. To: ____

Figure 12-10. Reciprocating compressor specifications.

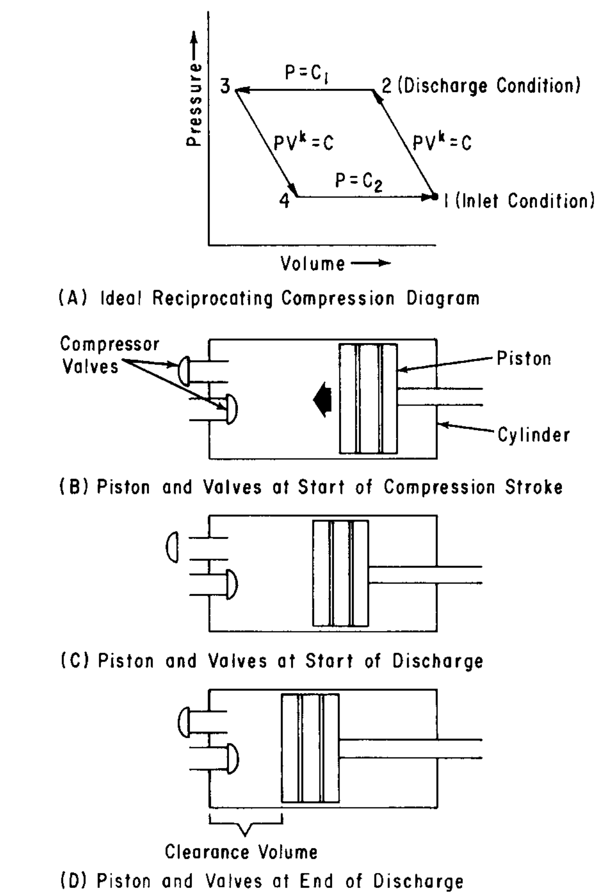


Figure 12-11. Ideal pressure-volume cylinder action for single-acting compressor cylinder with related piston positions. Note: DV = discharge valve; SV = suction valve; k or n may be exponent for gas. (Used and adapted by permission: *Gas Engineers Handbook*, 1st Ed., ©1977. Industrial Press, Inc.,New York. All rights reserved.)

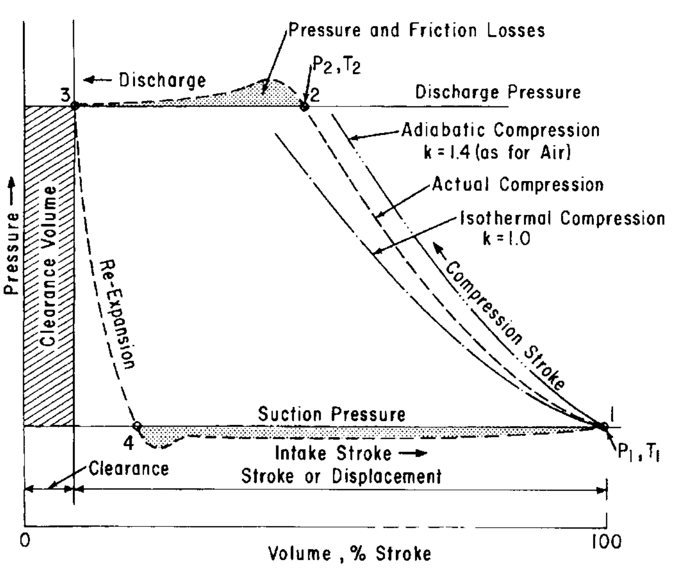


Figure 12-12. Reciprocating compressor compression diagrams. Actual losses and effect of k = c_p/c_v on performance.

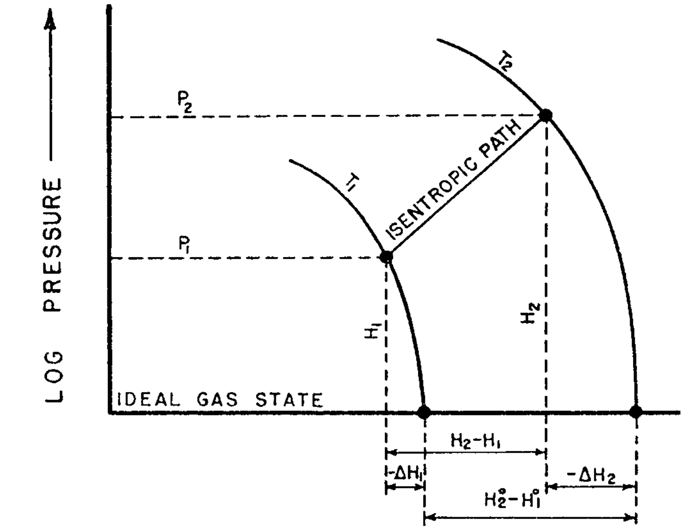
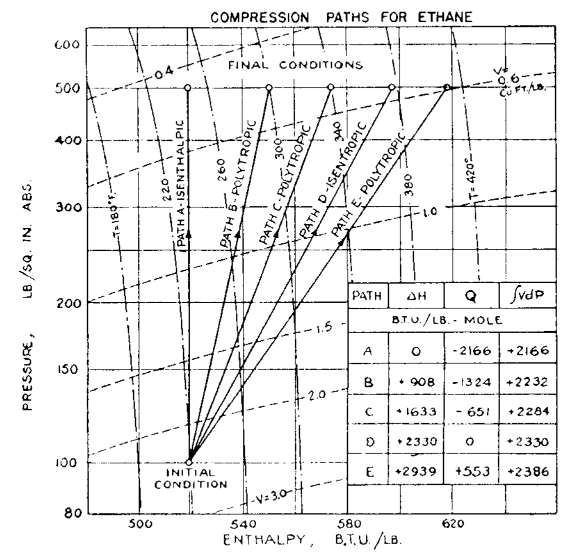


Figure 12-12A. Illustration of isentropic path on log pressure-enthalpy diagram, showing Mollier chart method of finding final temperature and calculation of H for reversible and adiabatic compression. (Used by permission: Edmister, W. C. *Applied Hydrocarbon Thermodynamics*, ©1961. Gulf Publishing Company, Houston, Texas. All rights reserved.)



PATH	ΔH	Q	∫VdP
	B.T.U./LB.-MOLE		
A	0	-2166	+2166
B	+908	-1324	+2232
C	+1633	-651	+2284
D	+2330	0	+2330
E	+2939	+553	+2386

Figure 12-12B. Section of ethane pressure-enthalpy diagram illustrating five compression paths (Used by permission: Edmister, W. C. *Applied Hydrocarbon Thermodynamics*, ©1961. Gulf Publishing Company, Houston, Texas. All rights reserved.)

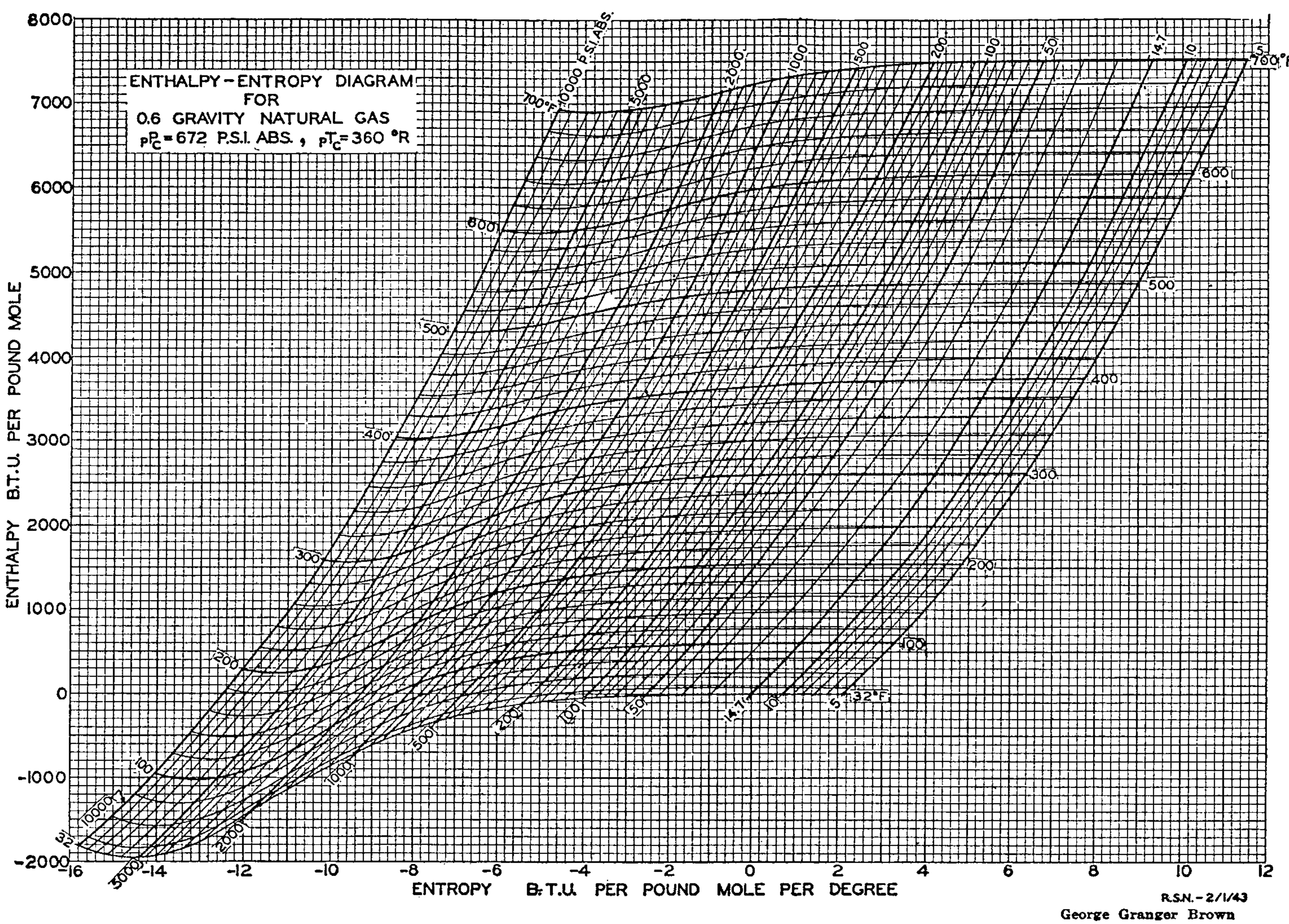
(Text continues from page 383)

from the true adiabatic condition results in the process known as "polytropic." It is defined as an internally reversible change of state[59] where

$$P_1V_1^n = P_2V_2^n = \text{constant (different from preceding)} \quad (12\text{-}10)$$

Series of Enthalpy-Entropy Charts for Natural Gases*

By DR. GEORGE GRANGER BROWN, University of Michigan



EXAMPLE: Cooling or heating a gas at constant pressure.

PROBLEM: What is the amount of heat which must be removed in cooling one pound mole of 0.6 Sp. Gravity gas at 120# Absolute from 300°F to 100°F?

SOLUTION: Read the enthalpy at the interesction of the 300°F line and the 120# Absolute line which is 2,600 BTU. Then follow the 120# Abs. line to its intersection with the 100°F line and read the enthalpy of 550 BTU. Therefore, the heat removed is 2,600—550 = 2,050 BTU per pound mole of gas.

CAUTION: Enthalpy-Entropy charts apply only to the gaseous state, and if a gas is cooled below its dew point, condensation occurs and heat removal cannot be determined directly from the charts.

Figure 12-13A. Enthalpy-entropy chart for Natural Gas, Sp.Gr. = 0.6. (Used by permission: *Engineering Data Book,* 7th Ed., ©1957. Natural Gasoline Supply Men's Association, Inc. All rights reserved.)

EXAMPLE: Adiabatic expansion of a gas.
PROBLEM: If a 0.7 Sp. Gravity gas at 300°F. and 600# Abs. is expanded adiabatically to 160# Abs., what is its temperature?

SOLUTION: Read the entropy at the intersection of the 300°F and 600# Abs. lines as —3. Then follow this entropy line to its intersection with the 160# Abs. line and read the temperature of 140°F.

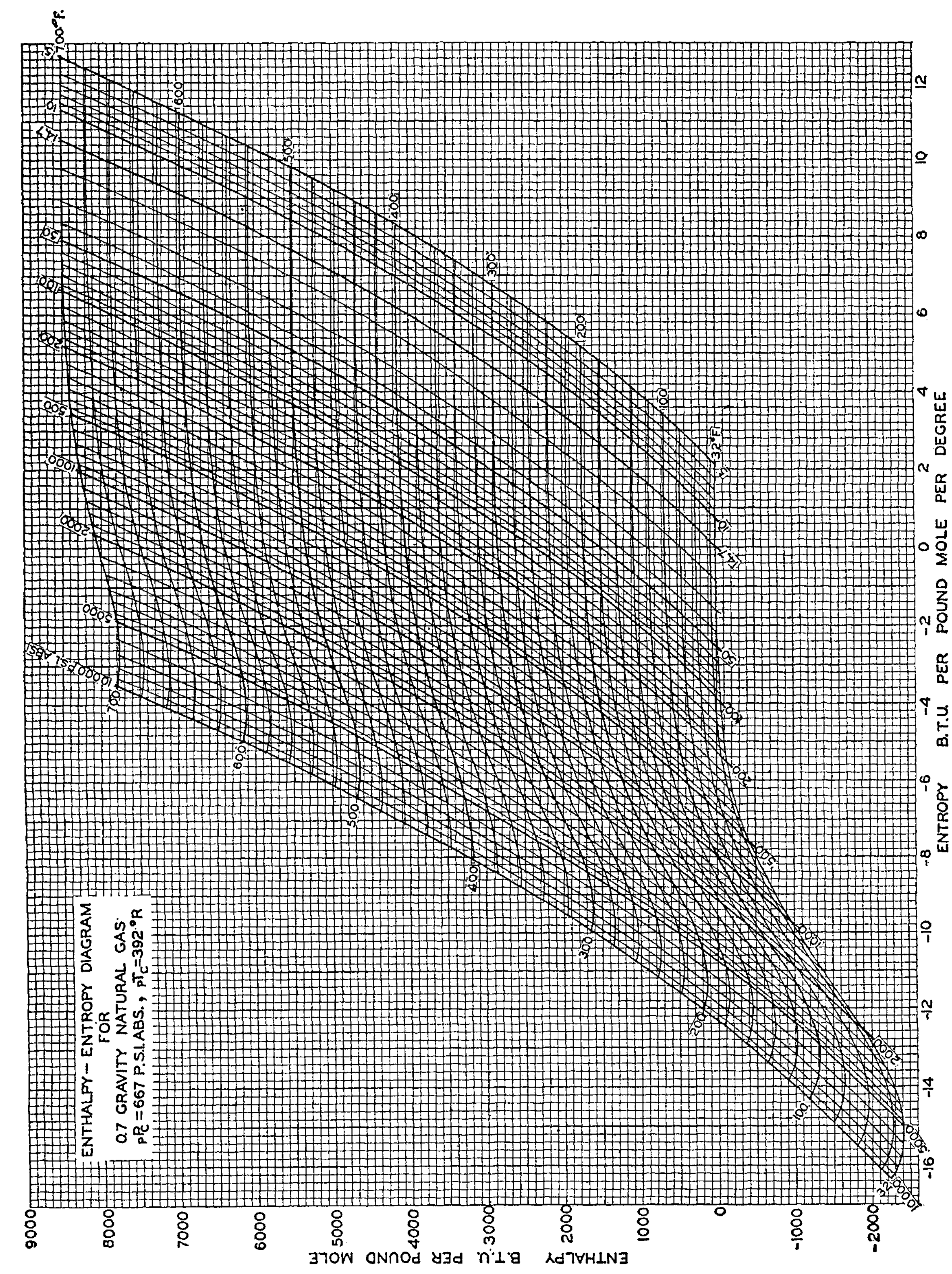


Figure 12-13B. Enthalpy-entropy chart for Natural Gas, Sp.Gr. = 0.7. (Used by permission: *Engineering Data Book*, 7th Ed., ©1957. Natural Gasoline Supply Men's Association, Inc. All rights reserved.)

EXAMPLE: Throttling or Joule Thompson Effect.
PROBLEM: If a gas of 0.8 Sp. Gravity at 2,000# Abs. and 200°F expands through a small orifice without the addition or subtraction of heat and is brought finally to its initial velocity and a pressure of 50# Abs., what is its temperature?

SOLUTION: Read the enthalpy at the intersection of the 2,000# Abs. line and the 200°F line which is 700 BTU. Then follow the 700 BTU enthalpy line to its intersection with the 50# Abs. line and read the temperature which is 100°F.

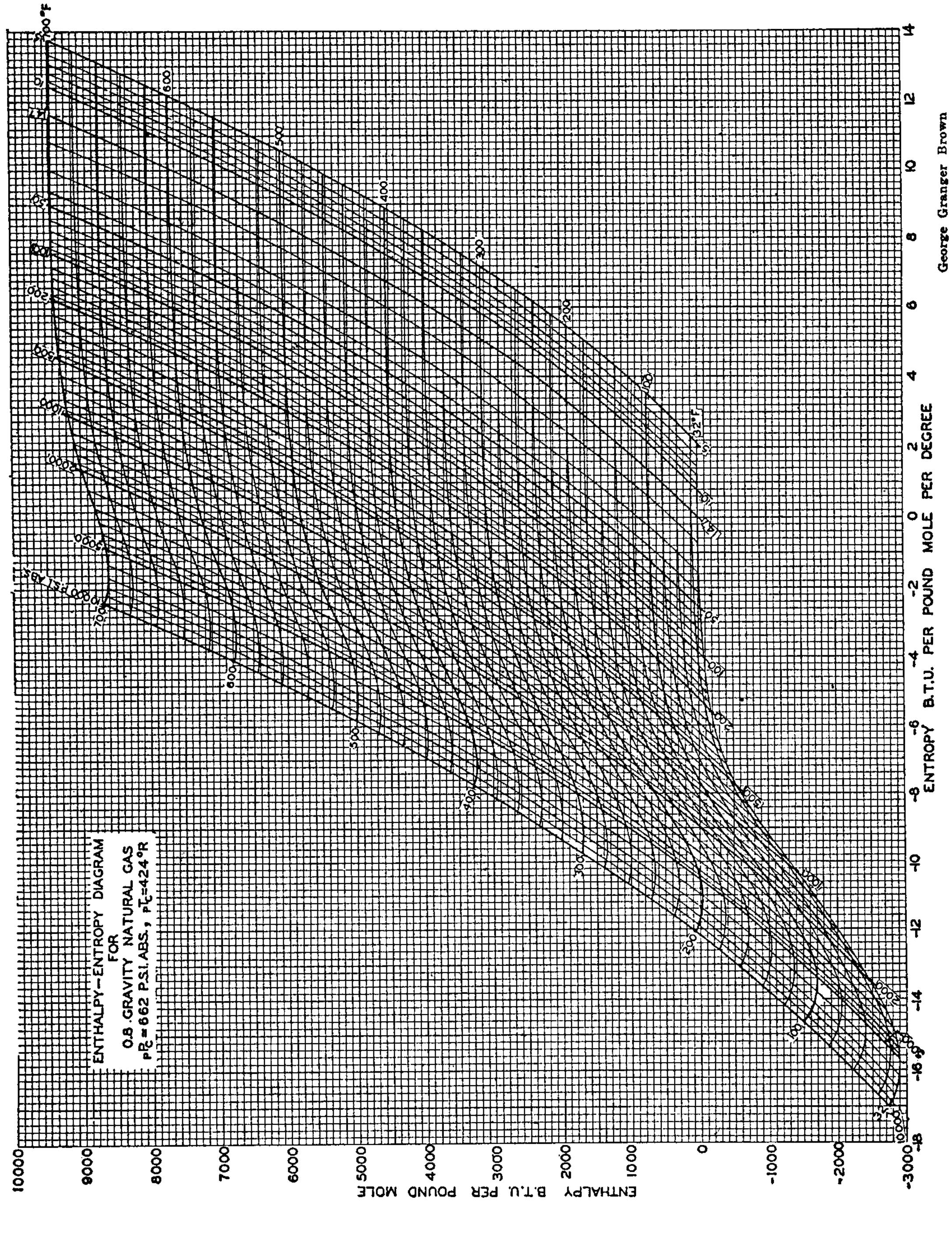


Figure 12-13C. Enthalpy-entropy chart for Natural Gas, Sp.Gr. = 0.8. (Used by permission: *Engineering Data Book*, 7th Ed., ©1957. Natural Gasoline Supply Men's Association, Inc. All rights reserved.)

EXAMPLE: Adiabatic Compression.
PROBLEM: If a 0.9 Sp. Gravity gas at 50# Abs. and 60°F is compressed adiabatically to 250# Abs. what is the Temperature of the compressed gas?

SOLUTION: Read the entropy at the intersection of the 50# Abs. line and the 60°F line which is —2. Then follow the —2 entropy line to its intersection with the 250# Abs. line and read the temperature of 222°F.

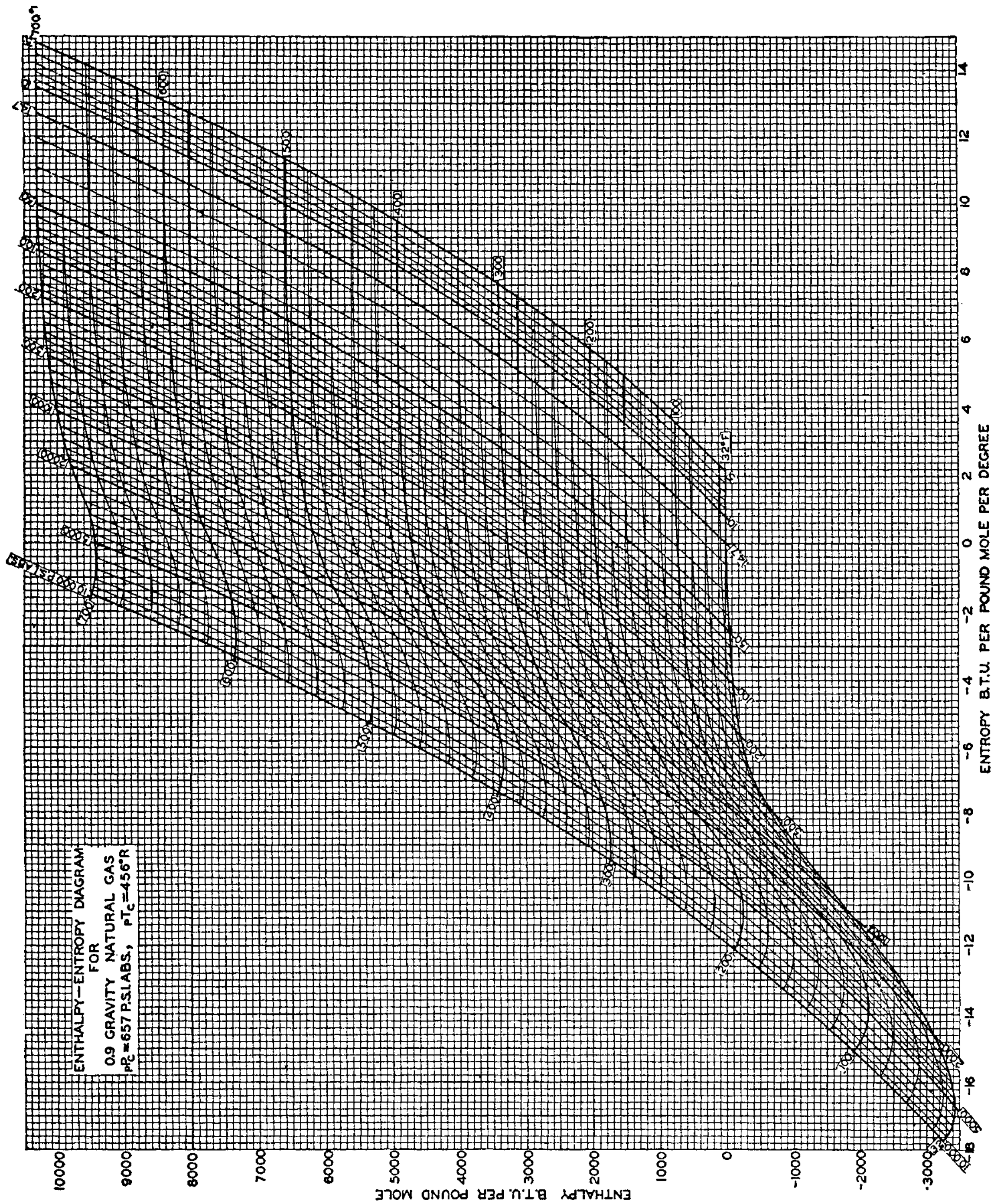


Figure 12-13D. Enthalpy-entropy chart for Natural Gas, Sp.Gr. = 0.9. (Used by permission: *Engineering Data Book*, 7th Ed., ©1957. Natural Gasoline Supply Men's Association, Inc. All rights reserved.)

For a polytropic process the change of state does not take place at constant entropy, but for an adiabatic process, it does. Heat may be added to or rejected from a gas in a polytropic process. For a polytropic process, the correlating exponent for the $P_1V_1^n$ component is the exponent "n," which becomes an important part of the compressor design. "n" values are determined from performance testing.

An adiabatic process is never fully attained but can be closely approached for many processes and is the primary design basis for most positive displacement compression equipment.

where k = ratio of specific heats, c_p/c_v
- n = polytropic coefficient
- P_1 = inlet pressure, abs, $lb_{(force)}/ft^2abs$
- P_2 = discharge pressure, abs, $lb_{(force)}/ft^2abs$
- R = gas constant, $ft\text{-}lb_{(force)}/(lb\text{-}°R)$ = 1,545/(mol wt)
- R′ = specific constant for the gas involved = 1,545/gas mol wt
- R_o = universal gas constant = 1,545 and is same for all gases = 10,729 when P is $lb/in.^2$ abs
- T_1 = inlet gas temperature, °R = °F + 460
- T_2 = discharge gas temperature, °R
- V = volume of gas, ft^3/lb-mole
- W = weight of gas, lb
- 1 = inlet condition (intake)
- 2 = outlet condition (discharge)

Adiabatic Calculations

$$\text{Adiabatic head, } H_{ad} = \frac{(k)}{(k-1)} RT_1 \frac{[(P_2)^{(k-1)/k}]}{(P_1)} - 1 \quad (12\text{-}11)$$

where
- H_{ad} = adiabatic head, ft
- R = gas constant, $ft\text{-}lb_{(force)}/lb°R$
- T_1 = inlet gas temperature, °R
- P_1 = absolute inlet pressure, $lb_{(f)}/in.^2$
- P_2 = absolute discharge pressure, $lb_{(f)}/in.^2$
- W = weight of gas, lb/sec

Work on the gas during compression = H_{ad} (W)

$$HP_{ad} = \frac{WH_{ad}}{550} = [k/(k-1)]\frac{(WRT_1)}{(550)}[(P_2/P_1)^{(k-1)/k} - 1] \quad (12\text{-}12)$$

HP_{ad} = horsepower, hp
Q = volume flow of gas, ft^3/min at inlet conditions

Charles' Law at Constant Pressure[60]

$$V_2/V_1 = T_2/T_1 \text{ or} \quad (12\text{-}13)$$

$$V_2/T_2 = V_1/T_1 \quad (12\text{-}14)$$

Amonton's Law at Constant Volume[60]

$$P_2/P_1 = T_2/T_1 \text{ or} \quad (12\text{-}15)$$

$$P_2/T_2 = P_1/T_1 \quad (12\text{-}16)$$

Combined Boyle's and Charles' Laws

$$\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2} \quad (12\text{-}17)$$

Figures 12-12A and 12-12B illustrate various paths of the compression or expansion of gas that can take place, depending on the properties of the gas/vapor.

a. *Mollier Chart Method*[61]

After identifying the initial temperature (T) and pressure (P) values, the final temperature and both enthalpy values (H) can be read on the same entropy line of the appropriate gas Mollier chart. For the adiabatic process, the work done on the gas is equal to ΔH,[61] see Figures 12-13A–D. The following is reproduced by permission of Edmister, W. C., *Applied Hydrocarbon Thermodynamics,* Gulf Publishing Company.[61]

> "When a Mollier chart is available for the gas involved[62, 64] the first method, which is illustrated by Figure 12-12A is the most convenient. On the abscissa of Figure 12-12A four enthalpy differences are illustrated. $(H_2 - H_1)$ is the enthalpy difference for the isentropic path. $(H_2° - H_1°)$ is the ideal gas state enthalpy difference for the terminal temperatures of the isentropic path. The other ΔH values are the isothermal pressure corrections to the enthalpy at the terminal temperatures. A generalized chart for evaluating these pressure corrections was presented previously.
>
> As Mollier charts are available for only a few pure components and practically no mixtures, this calculation method is very limited. For example, it cannot be used for most process calculations because these gases are usually mixtures. Some of the charts available for mixtures are the H-S charts presented by Brown[62] for natural gases of gravities from 0.6 to 1.0."

b. *Entropy Balance Method*[61]

"Using generalized isothermal effects of pressure and the ideal gas state "S" and "H" values, the final temperature required to satisfy the condition ΔS = 0 is found, and the value of ΔH is determined for the path.

The second method can be applied to mixtures as well as pure components. In this method the procedure is to find the final temperature by trial, assuming a final temperature and checking by entropy balance (correct when $\Delta S_{path} = 0$). As reduced conditions are required for reading the tables or charts of generalized thermodynamic properties, the pseudo critical temperature and pressure are used for the mixture. Entropy is computed by the relation. See reference 61 for details."

$$S = S° - R\ln P + \Delta S' \quad (12\text{-}18)$$

c. Isentropic Exponent Method[61]

"Using exponents defining temperature and volume changes with pressure for the gas, final temperature and work are computed by simple equations."

Many gases deviate from the ideal state when pressures and/or temperatures are greater than 100–500 psia and 100°F. Some deviations yield a compressibility factor, Z, less than 1.0, and others give values greater than 1.0.

$$PV = ZNRT \quad (12\text{-}1)$$

or

$$PV = 10.71ZNT \quad (12\text{-}19)$$

where P = absolute pressure, psia
V = volume of gas, ft^3/lb-mole
T = absolute temperatures, °R (Rankine) = °F + 460
R = universal gas constant = 10.729 for units noted here
Z = compressibility factor
N = number of lb-moles of gas
Values of gas constant, R, for other units:
R = 1,545.3 when P = lb/ft^2, abs
R = 0.7302 when P = atmosphere
R = 10.729 when P = $lb/in.^2$, abs

Generalized compressibility factors for gases are given in Figures 12-14A–E. These charts have been prepared to allow approximately the same accuracy in reading values over the entire range.

Compressibility factors at low pressure for several major hydrocarbons are presented by Pfennig and McKetta.[70] Compressibility charts for specific gases are given in Figures 12-14F–W.

Figure 12-15 is a compressibility chart for natural gas based on pseudo-reduced pressure and temperature. The reduced pressure is the ratio of the absolute operating pressure to the critical pressure, P_r, and the reduced temperature is the ratio of the absolute operating temperature to the critical temperature, T_c, for a pure gas or vapor. The pseudo value is the reduced value for a mixture calculated as the sum of the mol percentages of the reduced values of the pure constituents.

$$\text{pseudo } P_r = y_1P_{r1} + y_2P_{r2} + y_3P_{r3}, \text{ etc.} \quad (12\text{-}20)$$

Similarly, the pseudo-reduced temperature can be determined.

Values of compressibility are available for many pure hydrocarbons and gases.[7,20,22]

Figure 12-16A illustrates a compression path for deviation from the ideal that overestimates the actual power required (area of dotted portion is greater than solid line actual area). Actual volumetric efficiency and inlet volume is less than ideal due to the deviation on the re-expansion path.[29] Table 12-2 compares an example for propane; a compressor with 10% clearance, 1,000 cfm piston displacement, compression from 100 psia and 80°F to 300 psia.

For Figure 12-16B, as illustrated by a 24–76% (volume) mixture of nitrogen-hydrogen at around 5,000 psia, the deviation is opposite to that of Figure 12-16A. The actual power requirements are greater than ideal; volumetric efficiency exceeds ideal gas laws.

Figure 12-16C illustrates ethylene in the extreme high pressure range (30,000-40,000 psi) where the deviation is unpredictable without thermodynamic data.

Figure 12-16D illustrates the type of reciprocating compressor performance problems that can develop from various mechanical details. To maintain peak efficiency in a compressor cylinder, a pressure-time indicator card of the cylinder during operation can be quite helpful in pointing to a problem and its nature.[79]

These figures illustrate what takes place inside the cylinder during the compressor's operation. When specifying performance, the actual capacity at suction and/or discharge conditions must be specified.

Table 12-3 lists the variation of compressibility factor, Z, with pressure as read or computed from accepted charts.

Compressibility must be taken into account along with the adiabatic coefficient, k, (or, if known, the polytropic coefficient, n) and other losses, which will be presented in the following paragraphs.

"k" Value of Gas (Ratio of Specific Heats). The ratio c_p/c_v is known as the "k" value of a gas and is associated with *adiabatic* compression or expansion. The change in temperature during compression (for most average water cooled jackets) is related by

$$P_1V_1^k = P_2V_2^k = P_3V_3^k = \text{constant} \quad (12\text{-}21)$$

for the same weight of gas at three different states or conditions. Most compression and expansion curves are represented by the preceding relationship. The actual value of "n" for a *polytropic* compression is usually 1.0–1.5 and is a function of the gas properties, such as specific heats, degree of cooling during compression (external), and operating features of the cylinder.[26] Figure 12-12 shows the effect of change in "k" on the compression curve. Usual reciprocating compressor performance evaluation uses adiabatic c_p/c_v, and this is the representation here. With the k = 1.0, the compression is isothermal; with "k" = "n" greater than 1.0, the operation is actually polytropic. For air the adiabatic "k" = 1.4.

(Text continues on page 400)

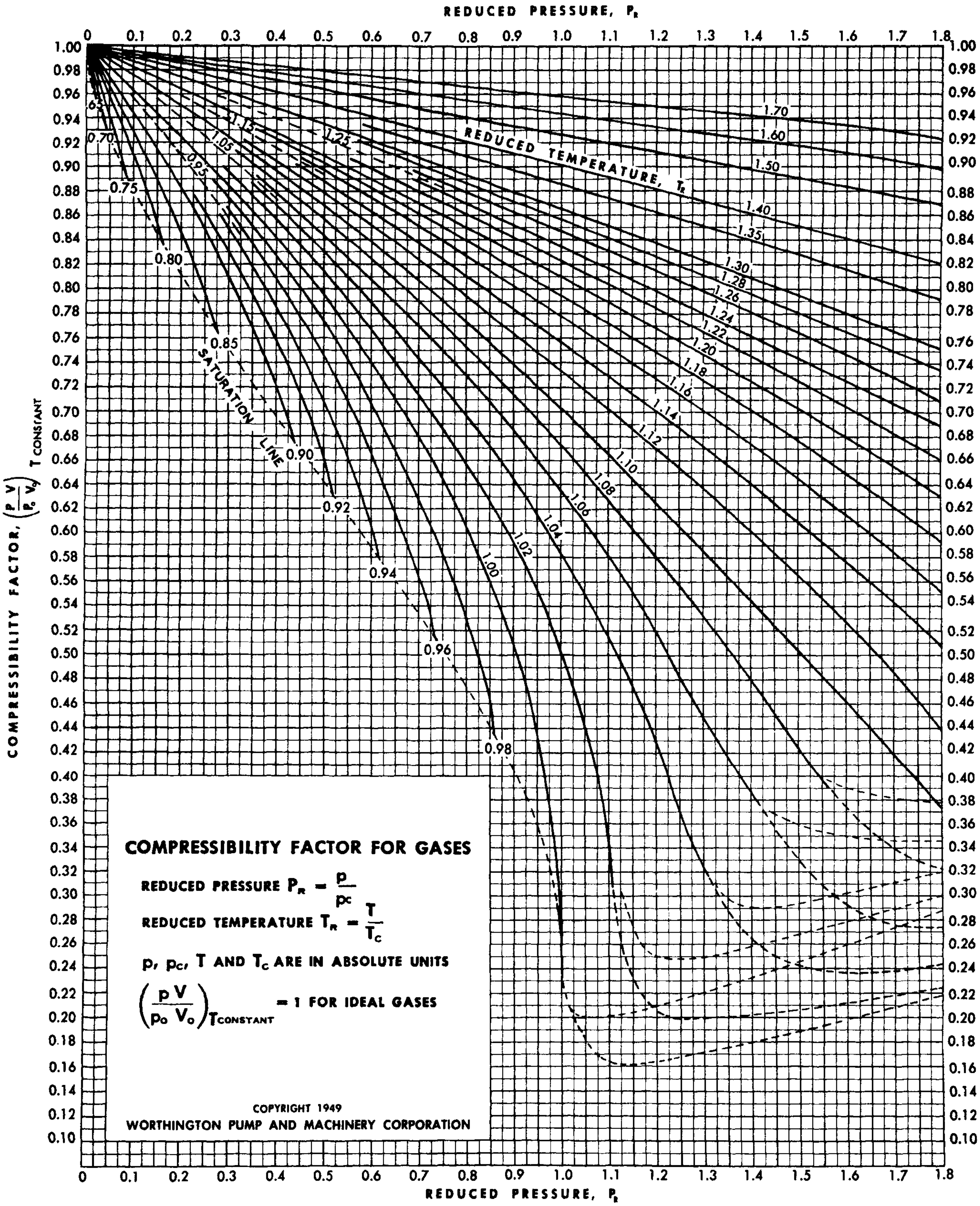


Figure 12-14A. Compressibility factor for gases, Part 1 of 5. (Used by permission: Worthington research bul. P-7637 ©1949. Dresser-Rand Company. All rights reserved.)

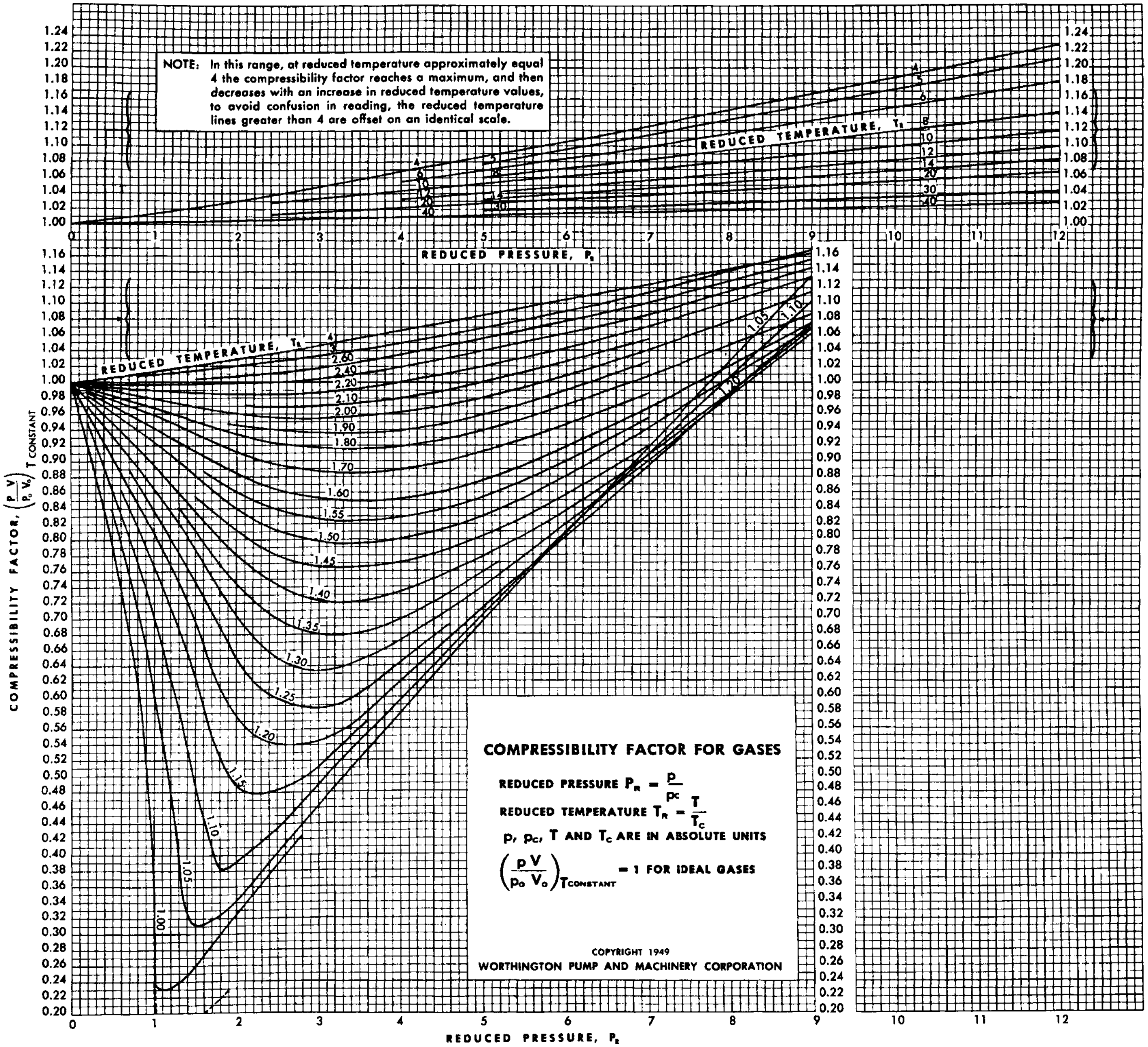


Figure 12-14B. Compressibility factor for gases, Part 2 of 5. (Used by permission: Worthington research bul. P-7637 ©1949. Dresser-Rand Company. All rights reserved.)

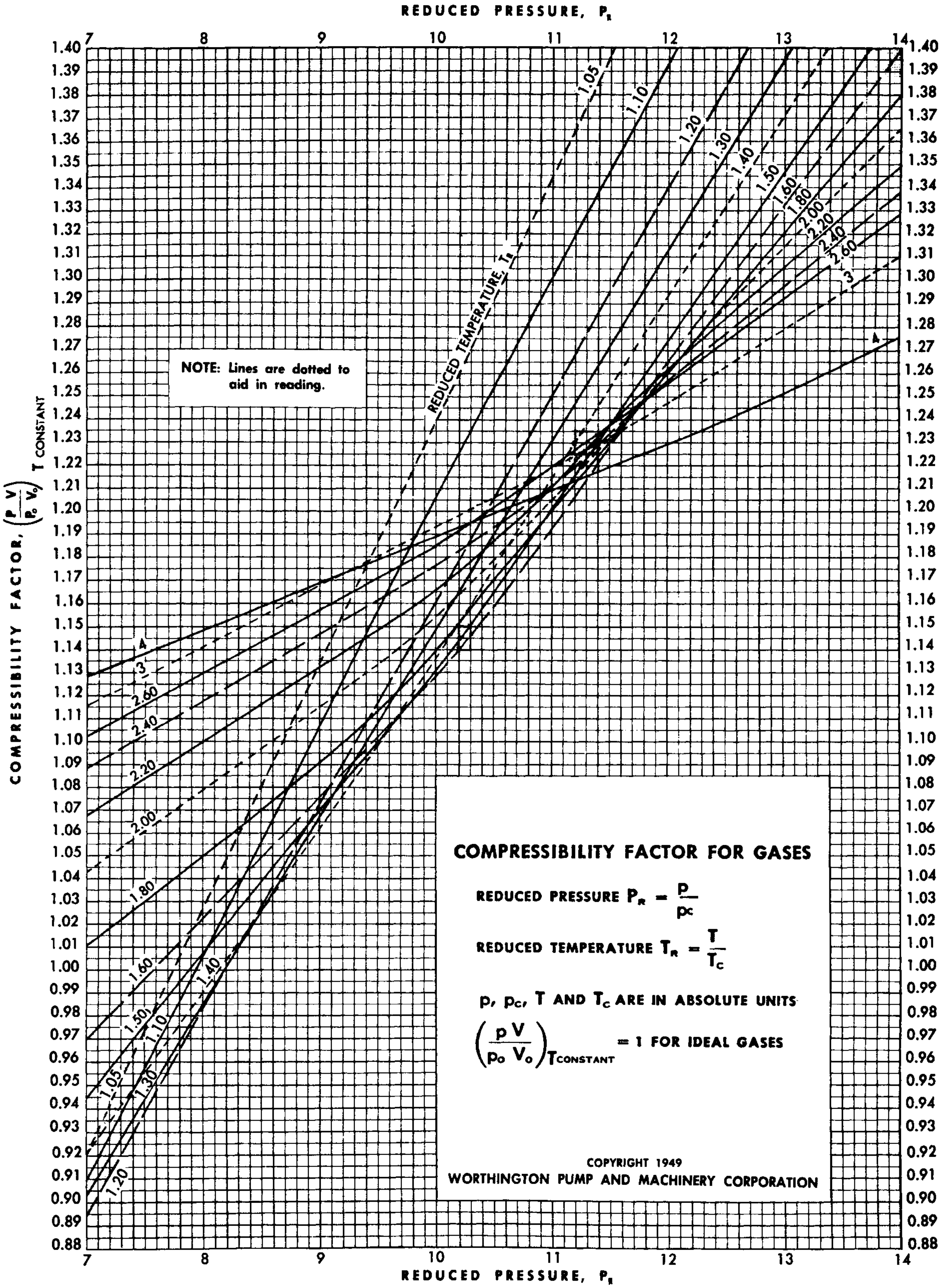


Figure 12-14C. Compressibility factor for gases, Part 3 of 5. (Used by permission: Worthington research bul. P-7637 ©1949. Dresser-Rand Company. All rights reserved.)

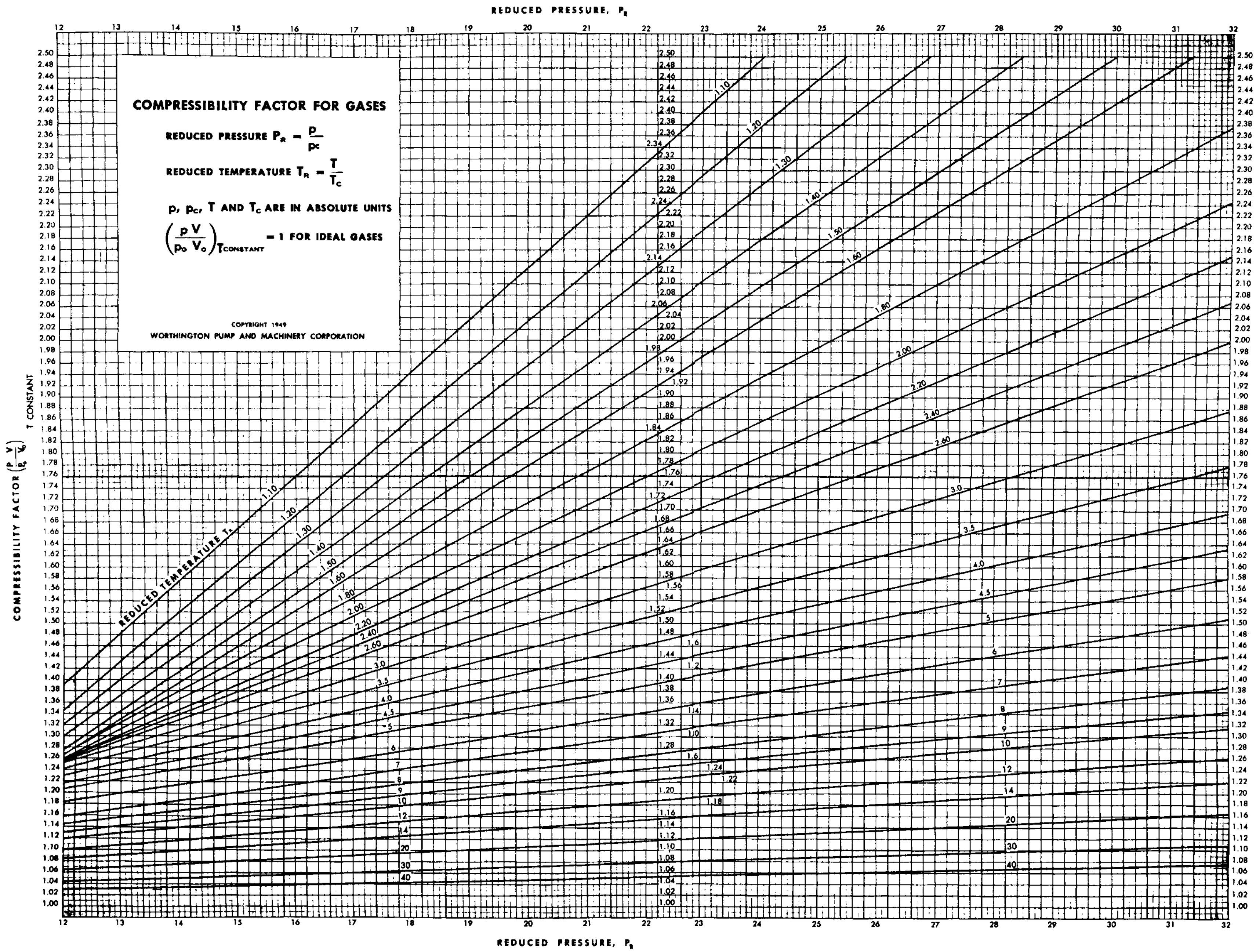


Figure 12-14D. Compressibility factor for gases, Part 4 of 5. (Used by permission: Worthington research bul. P-7637 ©1949. Dresser-Rand Company. All rights reserved.)

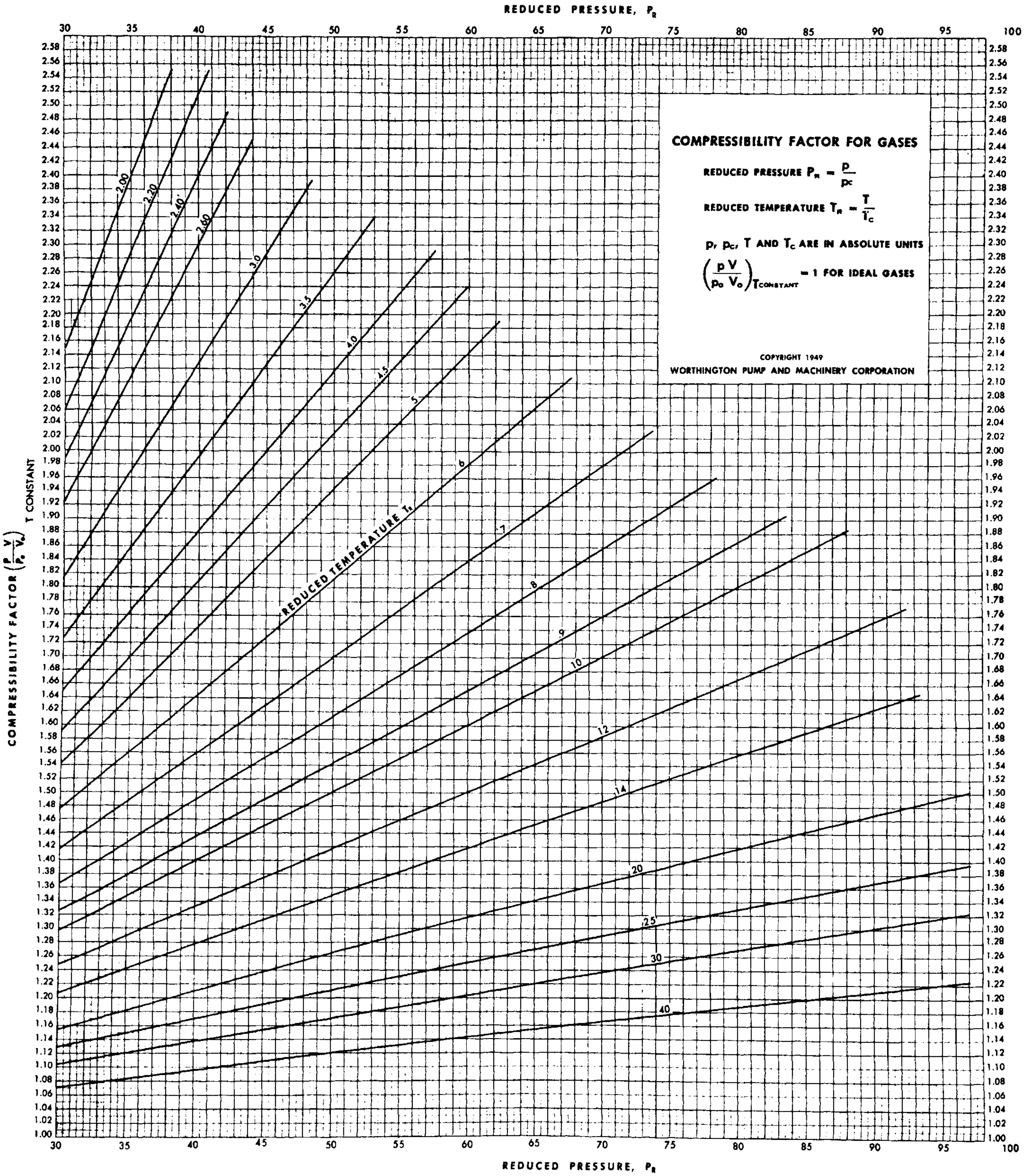


Figure 12-14E. Compressibility factor for gases, Part 5 of 5. (Used by permission: Worthington research bul. P-7637 ©1949. Dresser-Rand Company. All rights reserved.)

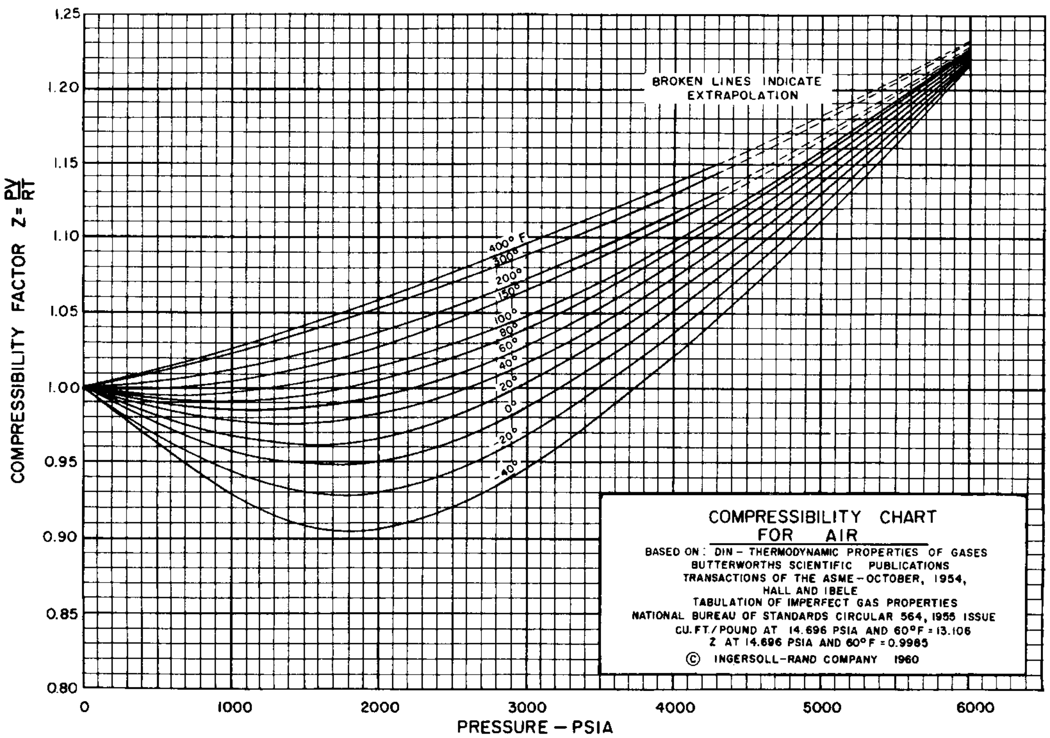


Figure 12-14F. Compressibility chart for air. (Used by permission: Form 3519 D (1981), ©1960. Ingersoll-Rand Company. All rights reserved.)

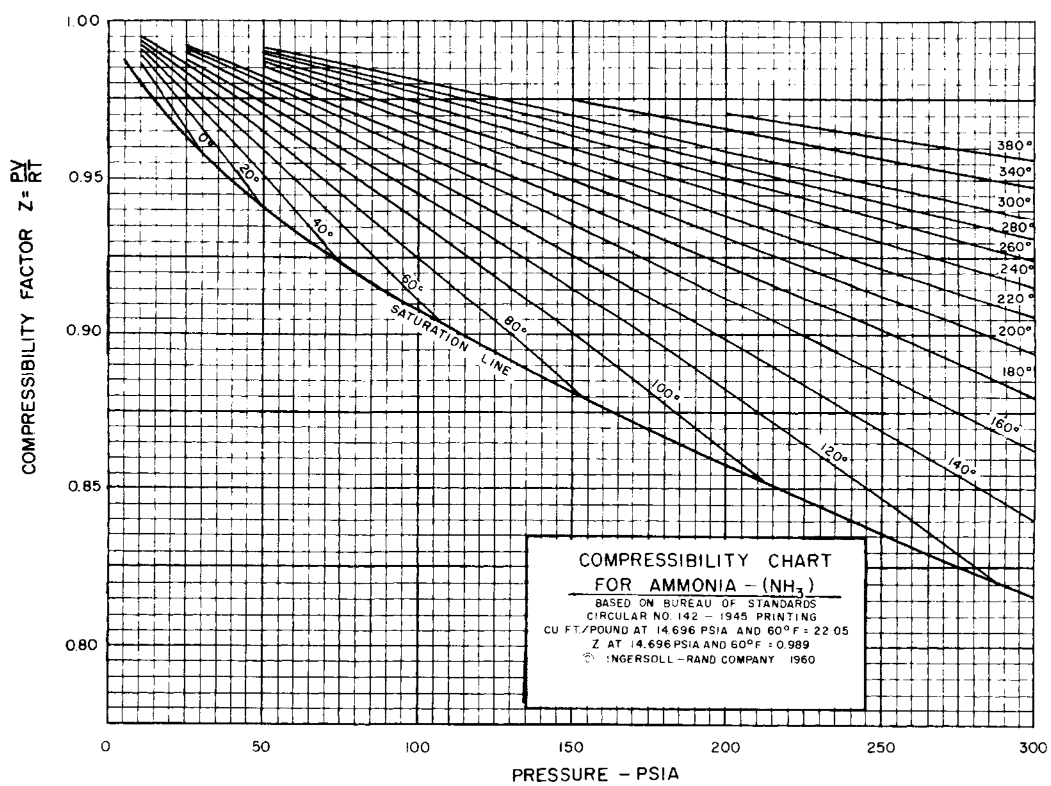


Figure 12-14G. Compressibility chart for ammonia. (Used by permission: Form 3519 D (1981), ©1960. Ingersoll-Rand Company. All rights reserved.)

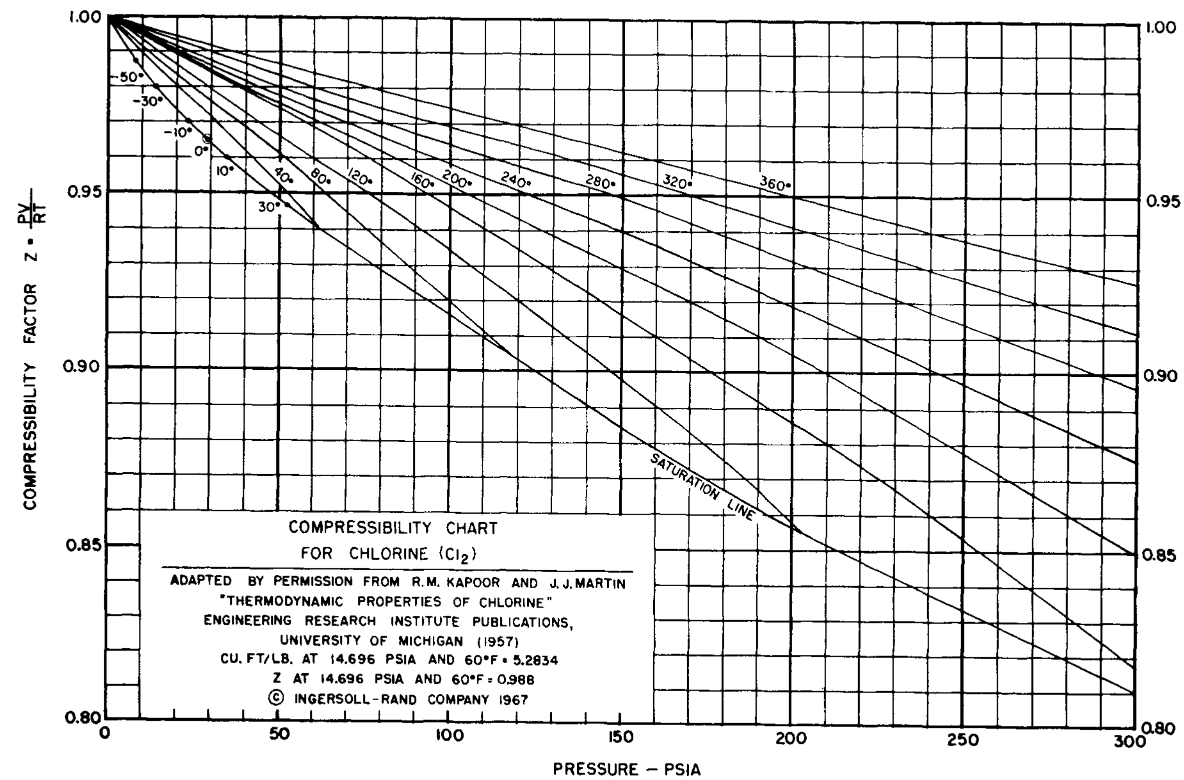


Figure 12-14H. Compressibility chart for chlorine. (Used by permission: Form 3519 D (1981), ©1967. Ingersoll-Rand Company. All rights reserved.)

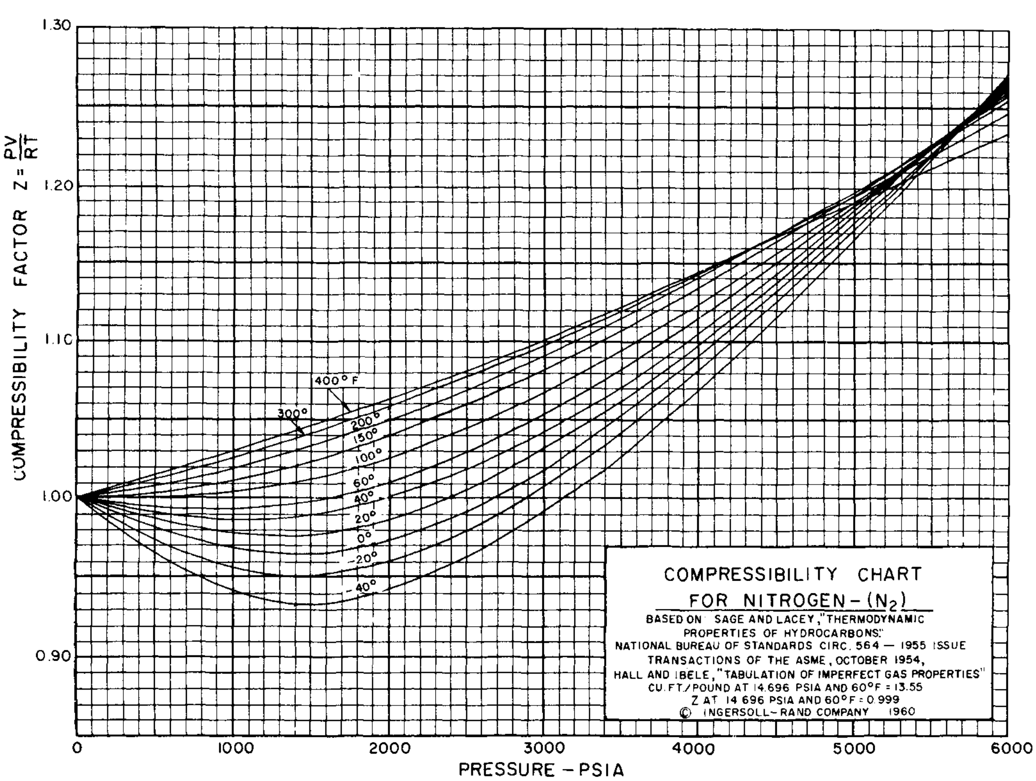


Figure 12-14I. Compressibility chart for nitrogen. (Used by permission: Form 3519 D (1981), ©1960. Ingersoll-Rand Company. All rights reserved.)

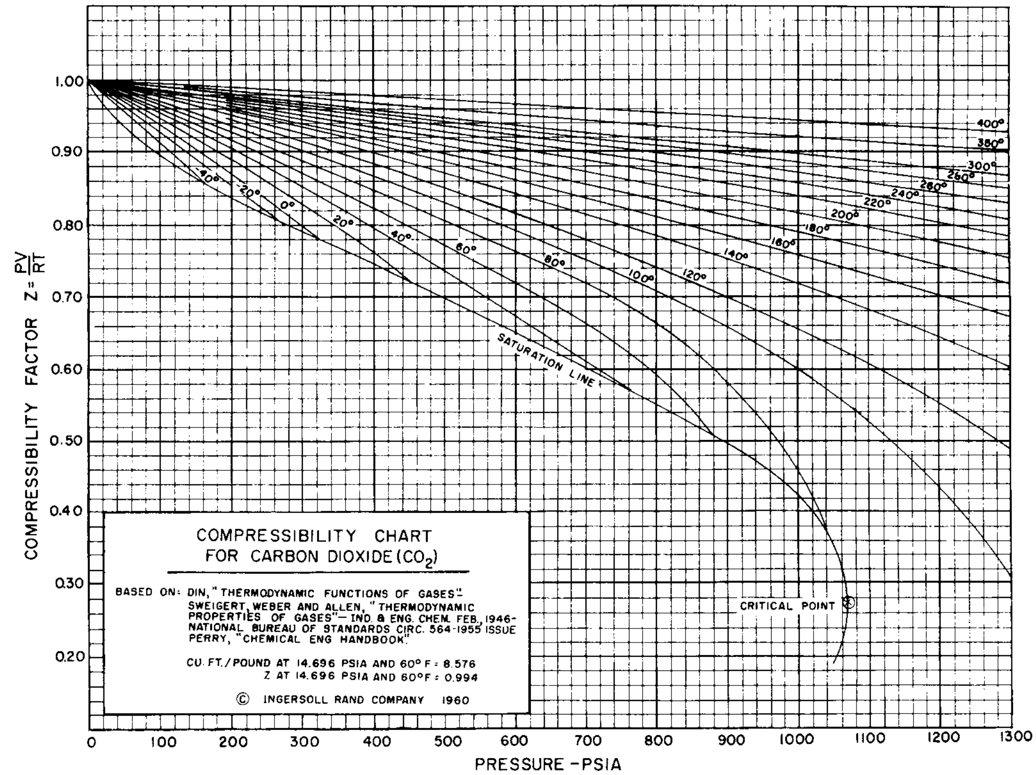


Figure 12-14J. Compressibility chart for low-pressure carbon dioxide. (Used by permission: Form 3519 D (1981), ©1960. Ingersoll-Rand Company. All rights reserved.)

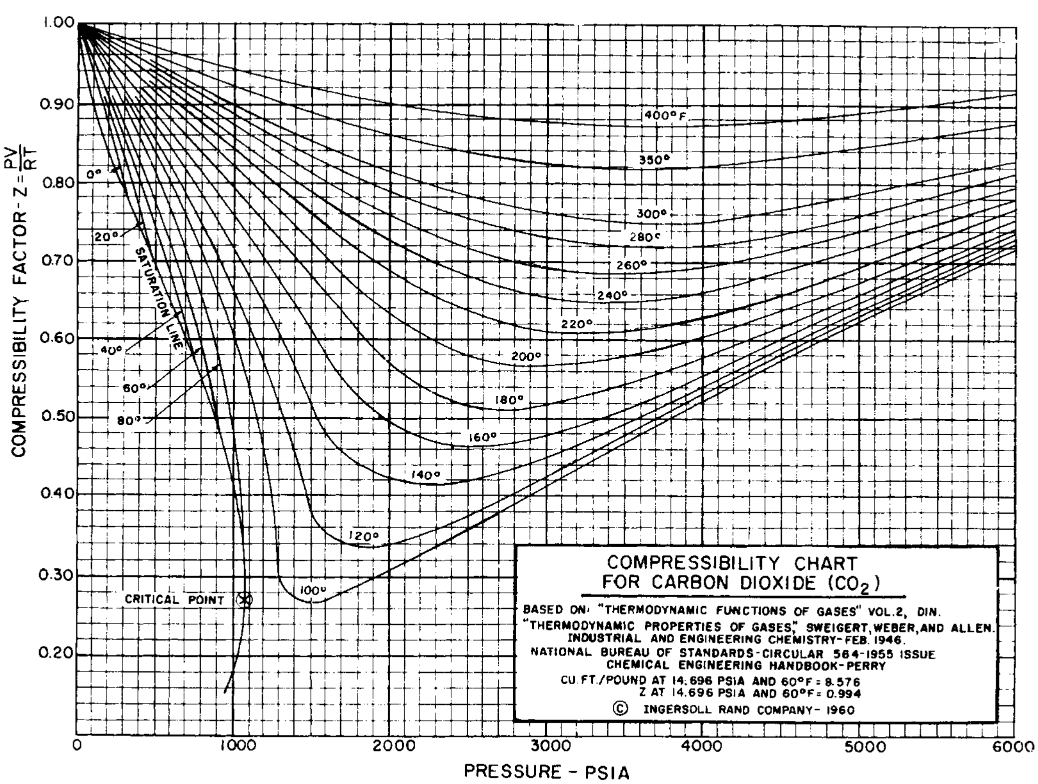


Figure 12-14K. Compressibility chart for high-pressure carbon dioxide. (Used by permission: Form 3519 D (1981), ©1960. Ingersoll-Rand Company. All rights reserved.)

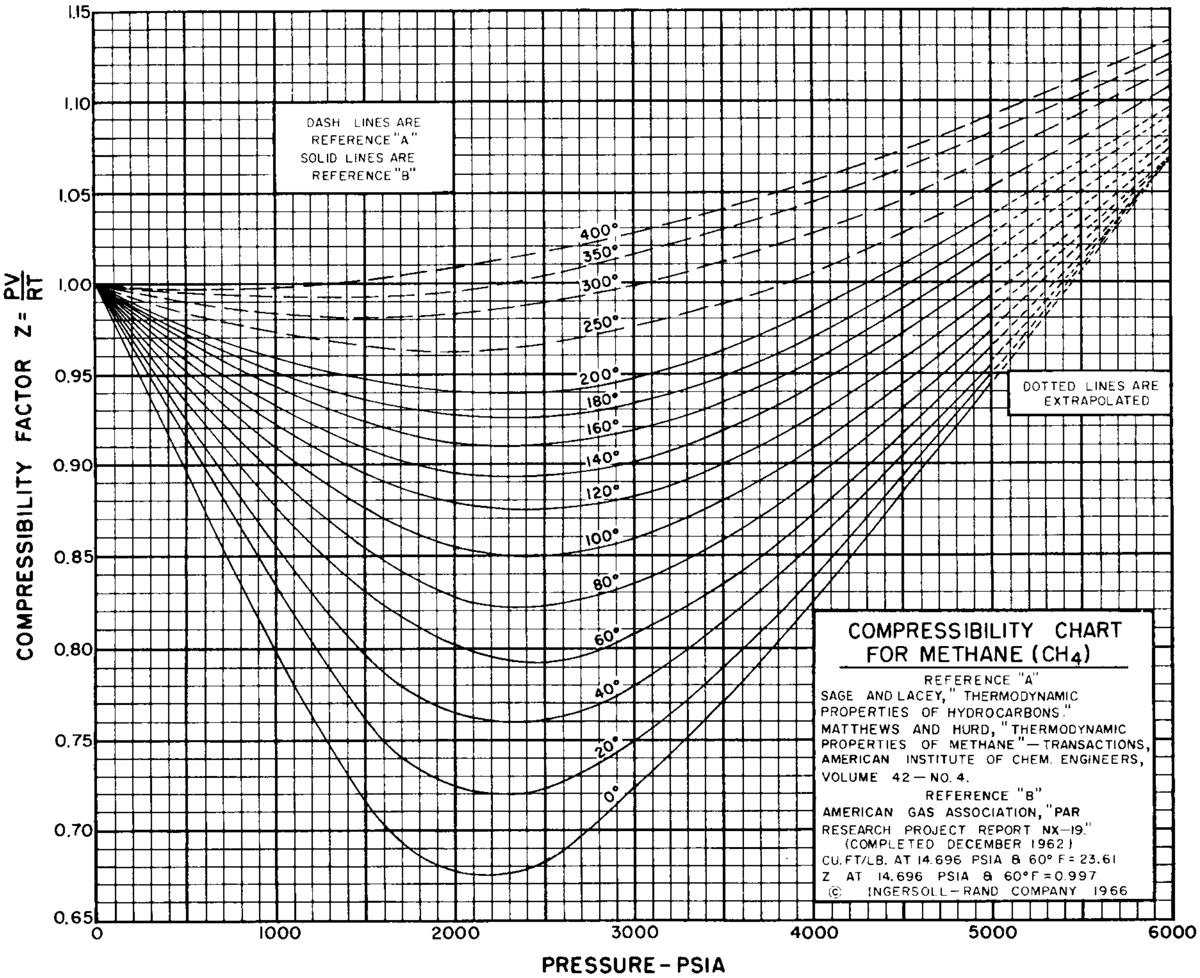


Figure 12-14L. Compressibility chart for methane. (Used by permission: Form 3519 D (1981), ©1966. Ingersoll-Rand Company. All rights reserved.)

(Text continues from page 391)

In adiabatic compression or expansion, no release or gain of heat by the gas occurs, and *no* change occurs in entropy. This condition is also known as isentropic and is typical of most compression steps. Actual conditions often cause a realistic deviation, but usually these are not sufficiently great to make the calculations in error. Table 12-4 gives representative average "k" values for a few common gases and vapors.

The specific heat is the heat required to raise the temperature of a unit mass of material one degree. Specific heat varies with temperature, but essentially no variation occurs with pressure.[74] The ratio, k, is important in most compression-related situations, i.e.,

$$k = c_p/c_v \tag{12-22}$$

For monatomic gases, k is about 1.66; for diatomic gases, k is about 1.40; and for polyatomic gases, k is about 1.30. Details of values for specific gases are available in many engineering tables.

The ratio, k, may be calculated from the ideal gas equation:

$$k = c_p/c_v = \frac{M_{cp}}{M_{cp} - 1.987} \tag{12-23}$$

where M_{cp} = molal heat capacity at constant pressure, Btu/lb-mol (°F)
M = molecular weight

When values of M_{cp} are not available, they may be calculated:

$$M_{cp} = A + BT, \text{BTU/mol/1°R} \tag{12-24}$$

with T in °Rankine at compressor cylinder inlet. The constants A and B may be obtained from Table 12-5.

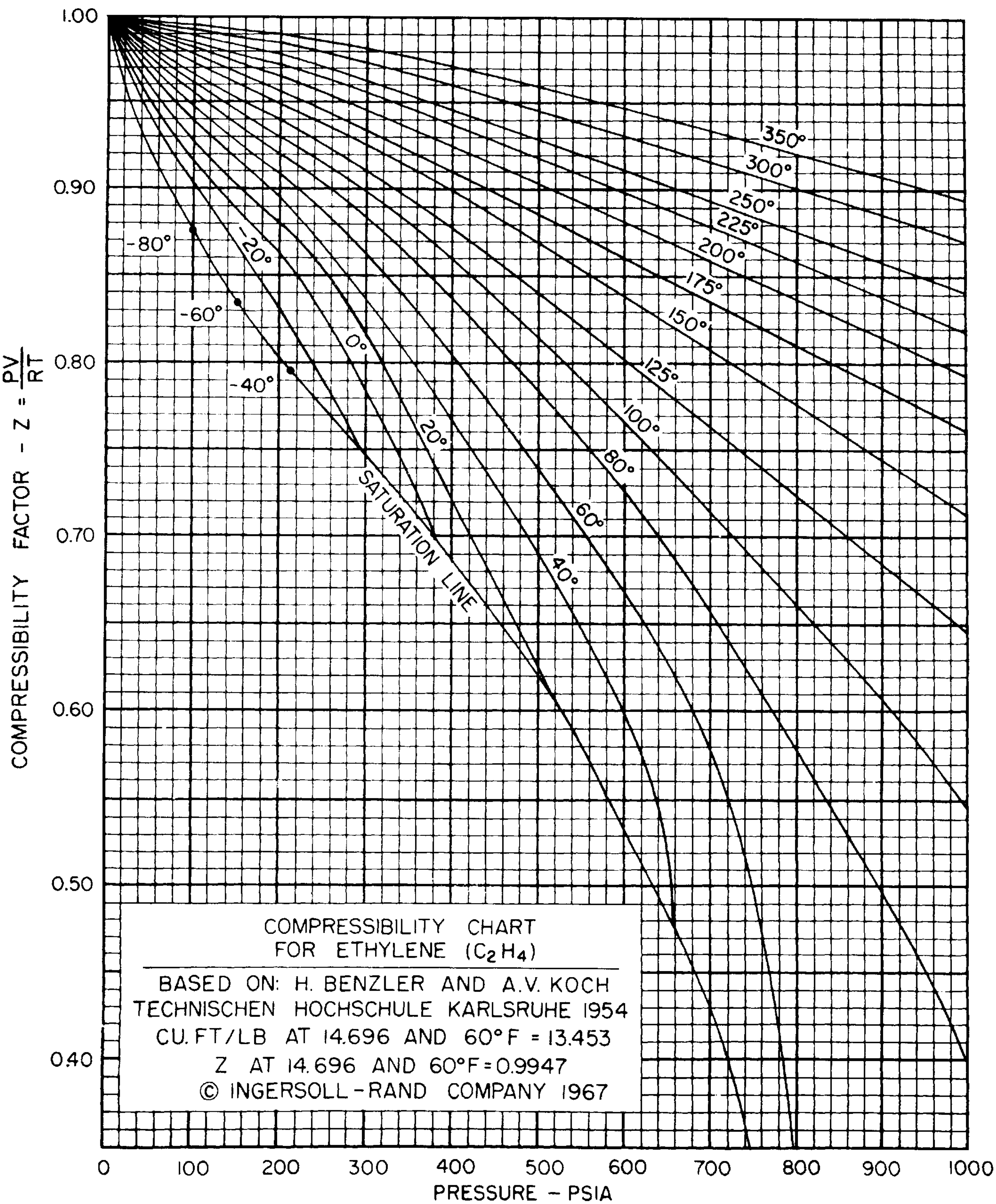


Figure 12-14M. Compressibility chart for low-pressure ethylene. (Used by permission: Form 3519 D (1981), ©1967. Ingersoll-Rand Company. All rights reserved.)

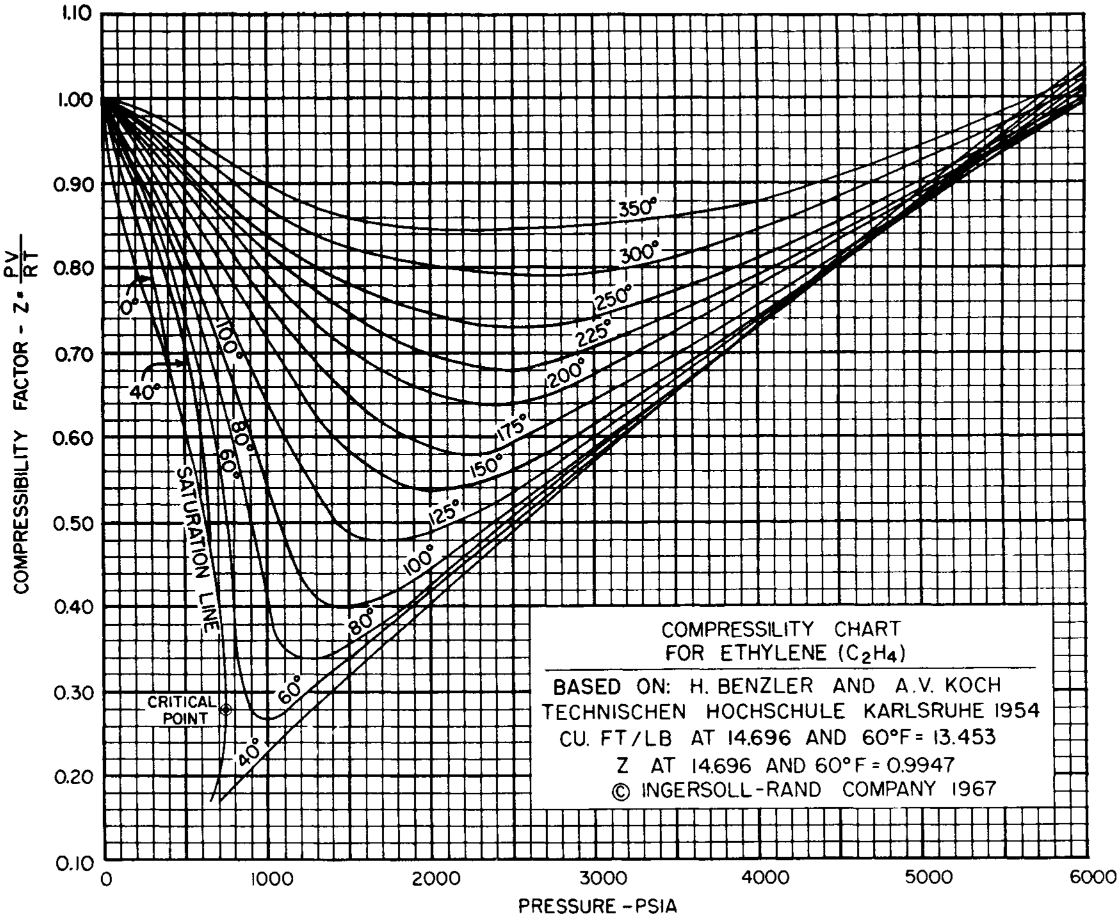


Figure 12-14N. Compressibility chart for high-pressure ethylene. Note: special charts are available for pressures in the range 20,000–75,000 psi. (Used by permission: Form 3519 D (1981), ©1960. Ingersoll-Rand Company. All rights reserved.)

$$k = c_p/c_v = c_p/(c_p - 1.987) \tag{12-25}$$

where c_p and c_v are specific heats at constant pressure and constant volume respectively, Btu/lb-mol-°F.[78]

To obtain the average value of c_p for a gas mixture, use the weighted mole fraction average, evaluating c_p at the average of the suction and discharge temperatures of the compressor cylinder. Depending on the magnitude of the compression ratio, the c_p at suction temperature can be used when the ratio is small.

m = isentropic or adiabatic exponent = $(k - 1)/k$
m′ = n = polytropic exponent = $(k - 1)/kE_p)$
E_p = polytropic efficiency = m/m′
= $[(k - 1)/k]/(n - 1)/n]$

$$m' = \frac{1(k-1)}{E_p(k)} = \frac{m}{E_p} = \frac{(n-1)}{(n)} \tag{12-26}$$

$$Mc_p = Mc_v + 1.987 \tag{12-27}$$

$$\text{Then; } k = \frac{Mc_p}{Mc_v} = \frac{c_p}{c_v} = \frac{Mc_p}{Mc_p - 1.987} \tag{12-28}$$

where M = molecular weight of gas
c = specific heat, Btu/lb-°F temperature rise
Mc_p = molar heat capacity, Btu/mol-°F[60] (see tables this reference), constant pressure; Mc_v = at constant volume
1.989 = constant for all hydrocarbon gases

For mixtures of gases, calculate the average Mc_p by multiplying the individual gas mol % of each component by its respective Mc_p (see reference 60 or other sources for tables) and sum to get the molar average, Mc_p, for the mixture. For the ratio of specific heat, see Equation 12-28.

(Text continues on page 409)

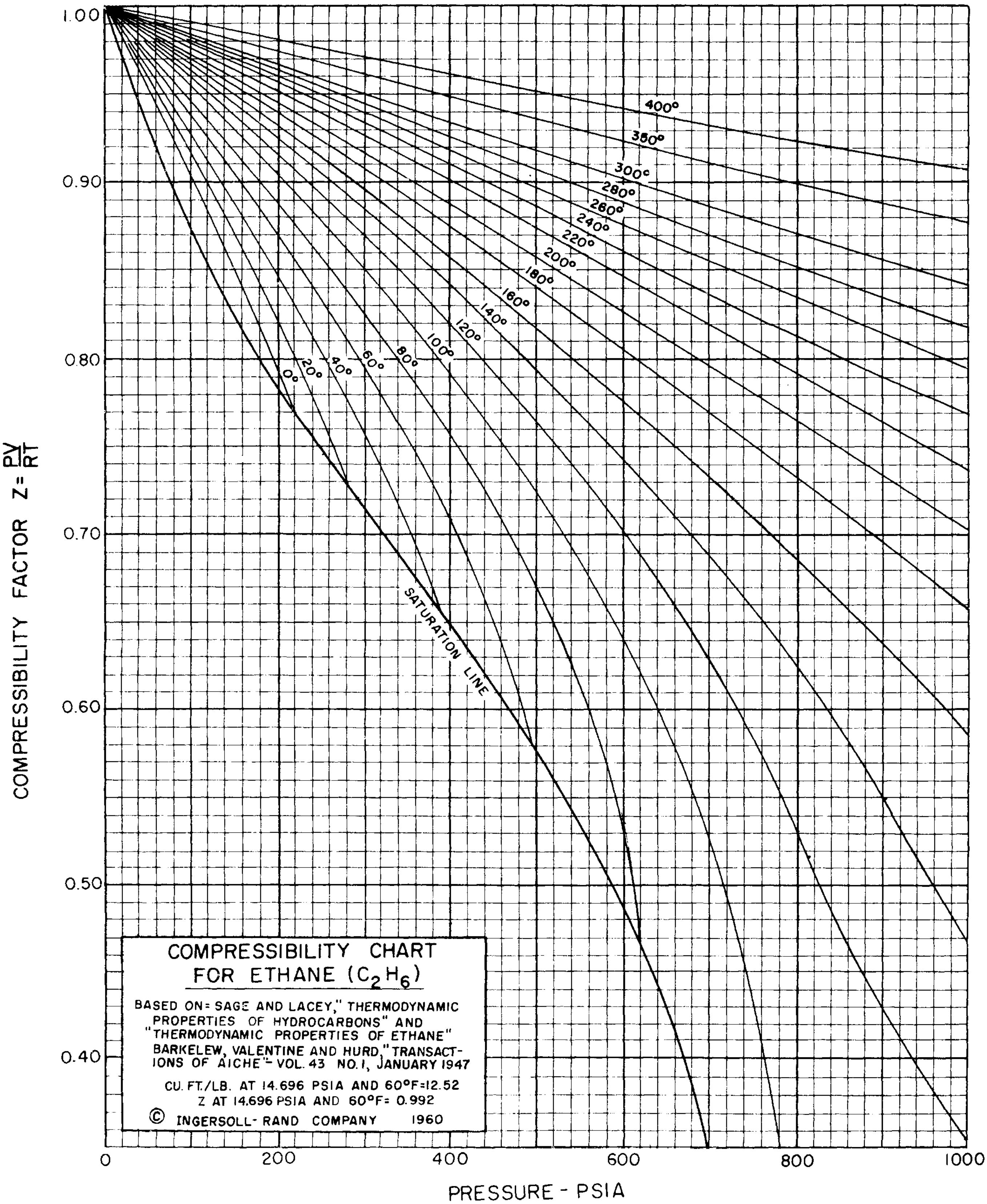


Figure 12-14O. Compressibility chart for low-pressure ethane. (Used by permission: Form 3519 D (1981), ©1960. Ingersoll-Rand Company. All rights reserved.)

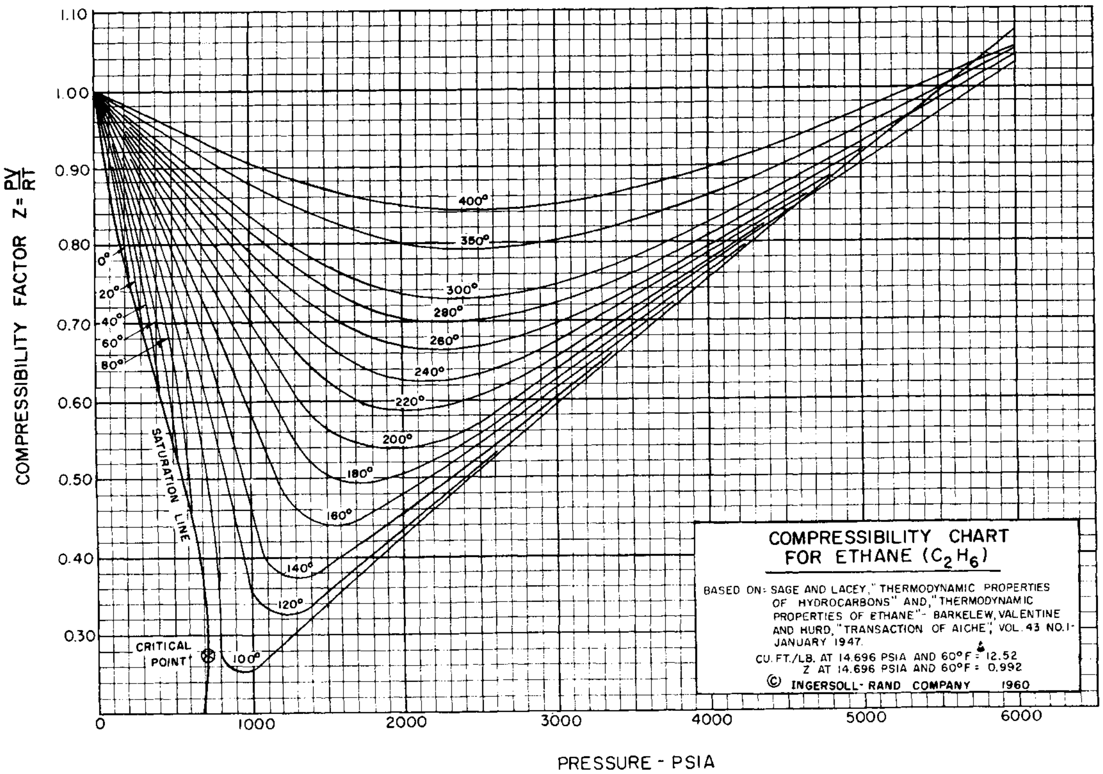


Figure 12-14P. Compressibility chart for high-pressure ethane. (Used by permission: Form 3519 D (1981), ©1960. Ingersoll-Rand Company. All rights reserved.)

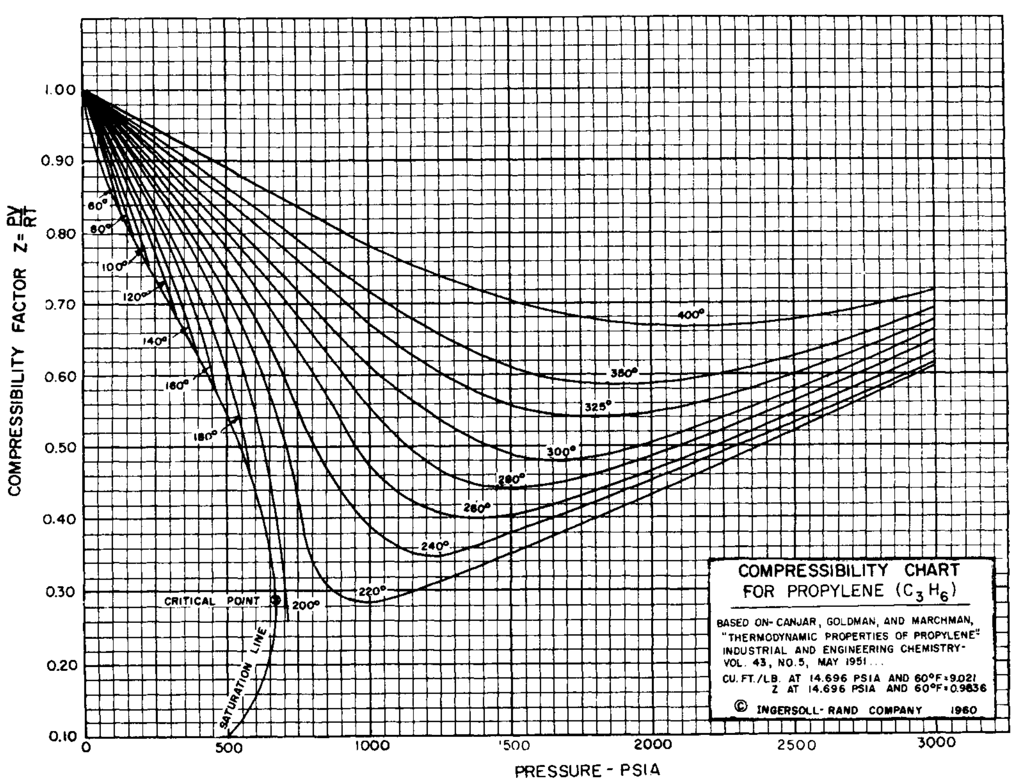


Figure 12-14Q. Compressibility chart for propylene. (Used by permission: Form 3519 D (1981), ©1960. Ingersoll-Rand Company. All rights reserved.)

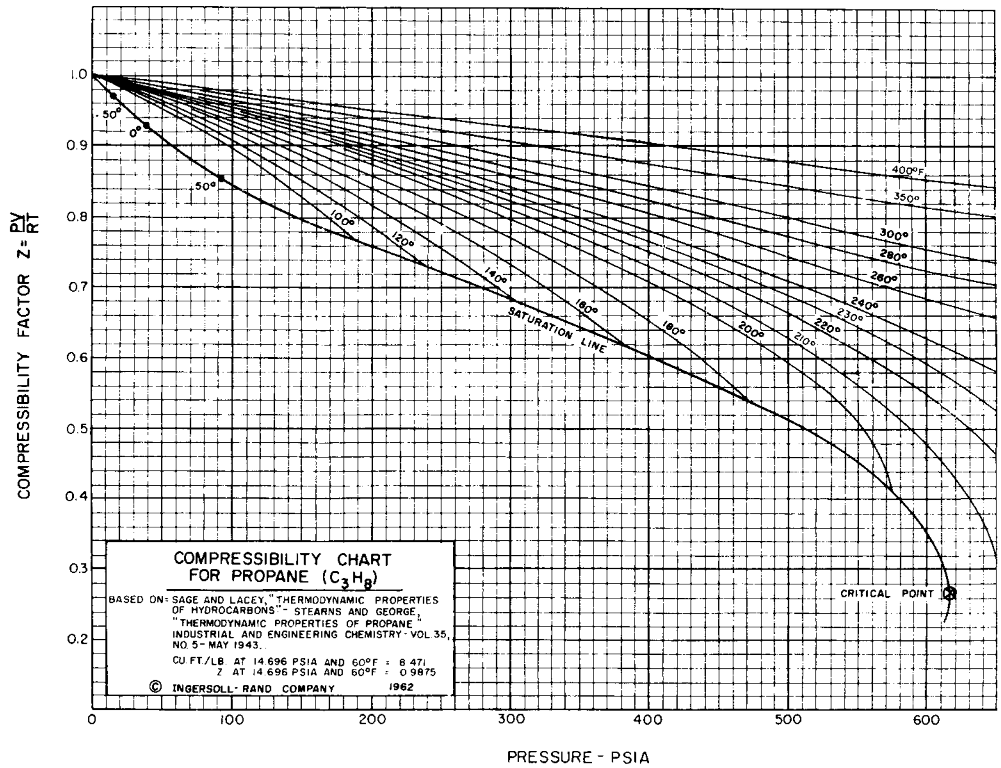


Figure 12-14R. Compressibility chart for low-pressure propane. (Used by permission: Form 3519 D (1981), ©1962. Ingersoll-Rand Company. All rights reserved.)

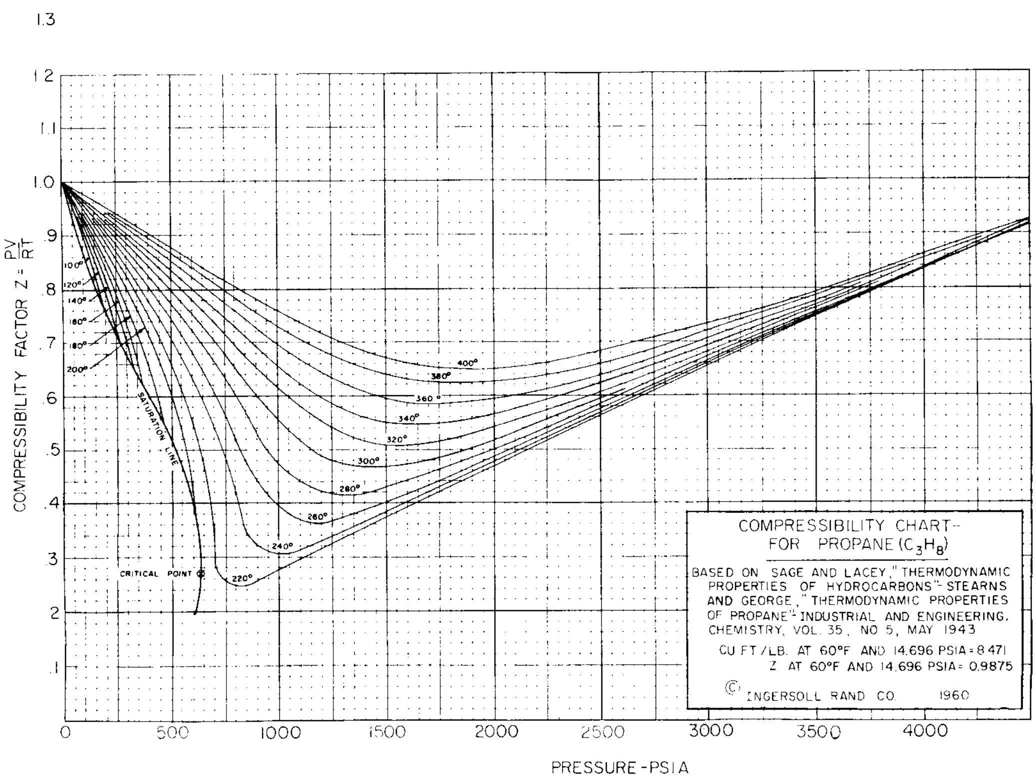


Figure 12-14S. Compressibility chart for high-pressure propane. (Used by permission: Form 3519 D (1981), ©1960. Ingersoll-Rand Company. All rights reserved.)

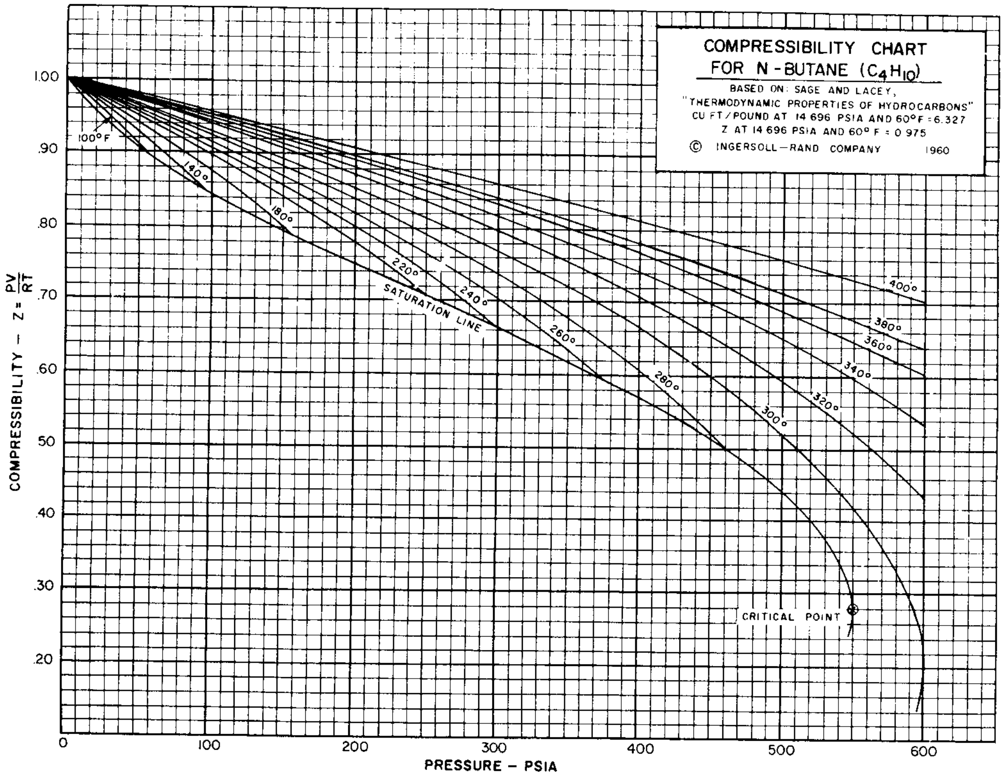


Figure 12-14T. Compressibility chart for low-pressure N-butane. (Used by permission: Form 3519 D (1981), ©1960. Ingersoll-Rand Company. All rights reserved.)

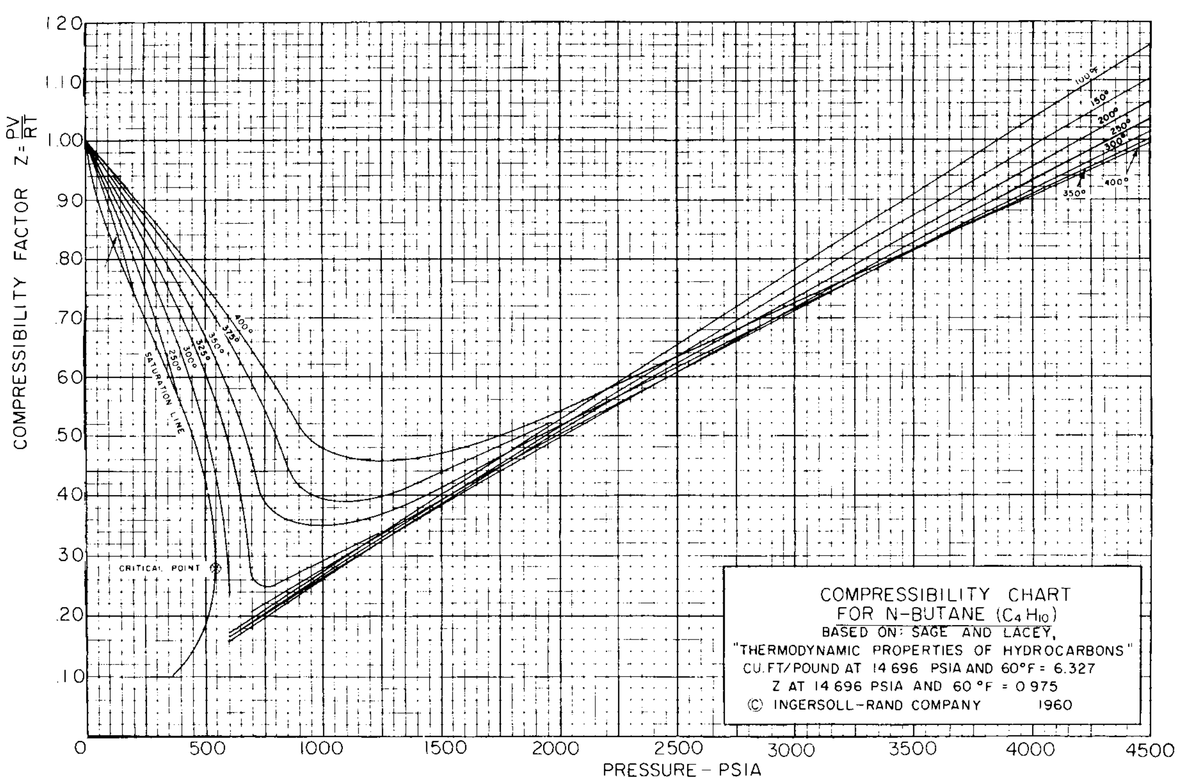


Figure 12-14U. Compressibility chart for high-pressure N-butane. (Used by permission: Form 3519 D (1981), ©1960. Ingersoll-Rand Company. All rights reserved.)

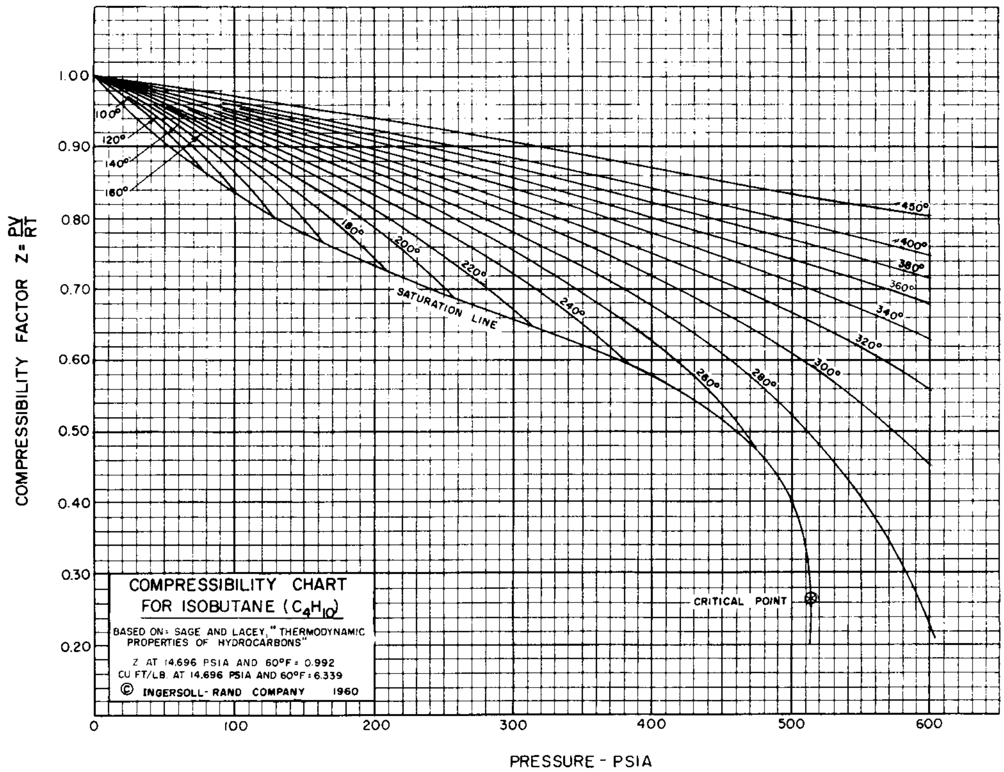


Figure 12-14V. Compressibility chart for low-pressure isobutane. (Used by permission: Form 3519 D (1981), ©1960. Ingersoll-Rand Company. All rights reserved.)

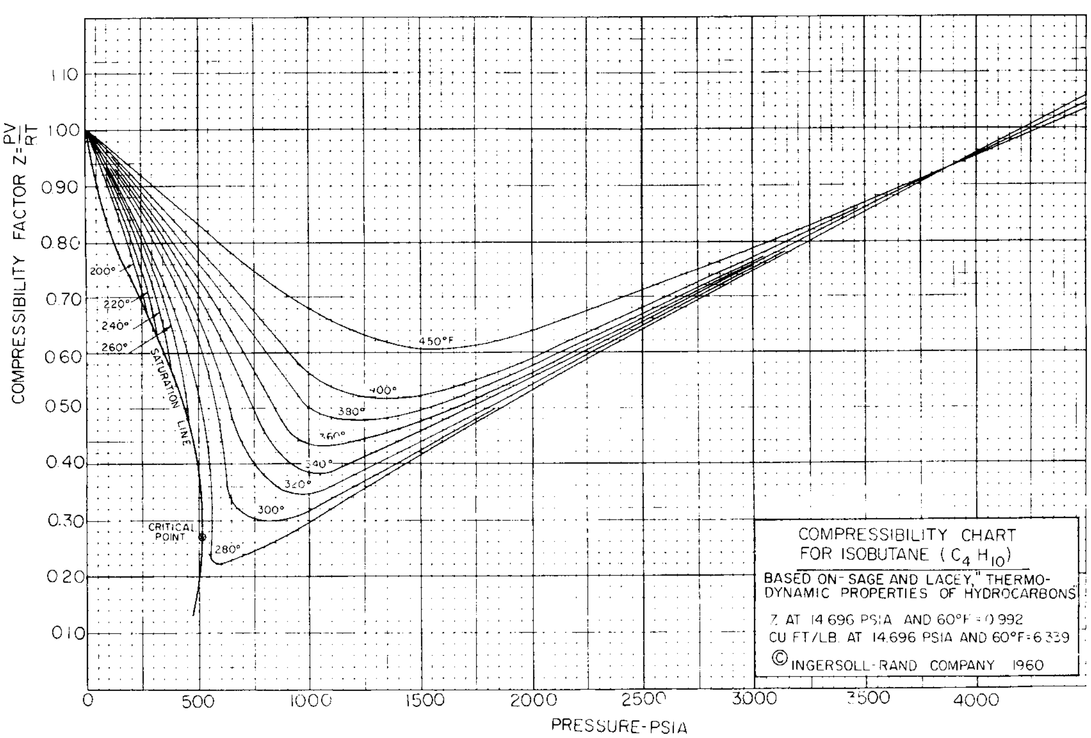
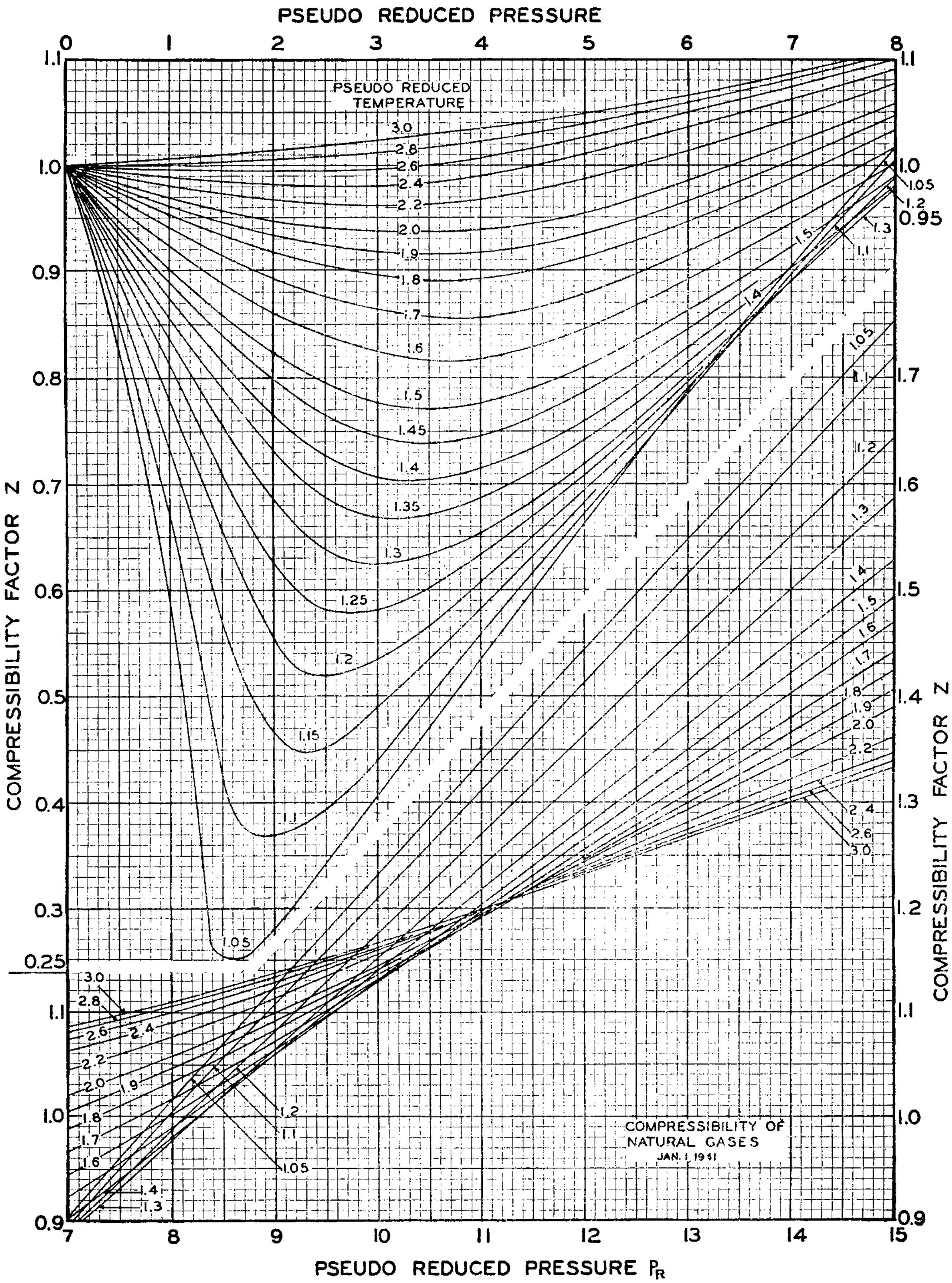


Figure 12-14W Compressibility chart for high-pressure isobutane. (Used by permission: Form 3519 D (1981), ©1960. Ingersoll-Rand Company. All rights reserved.)



Pseudo-reduced temperature $= \dfrac{\text{absolute temperature}}{\text{molecular average critical temperature}}$

Pseudo-reduced pressure $= \dfrac{\text{absolute pressure}}{\text{molecular average critical pressure}}$

Figure 12-15. Compressibility for natural gas. (Used by permission: Brown, G. G., G. G. Oberfell, D. L. Katz, and R. C. Alden. *Natural Gasoline and the Volatile Hydrocarbons,* Section One, ©1948. Natural Gasoline Association of America, Inc. All rights reserved.)

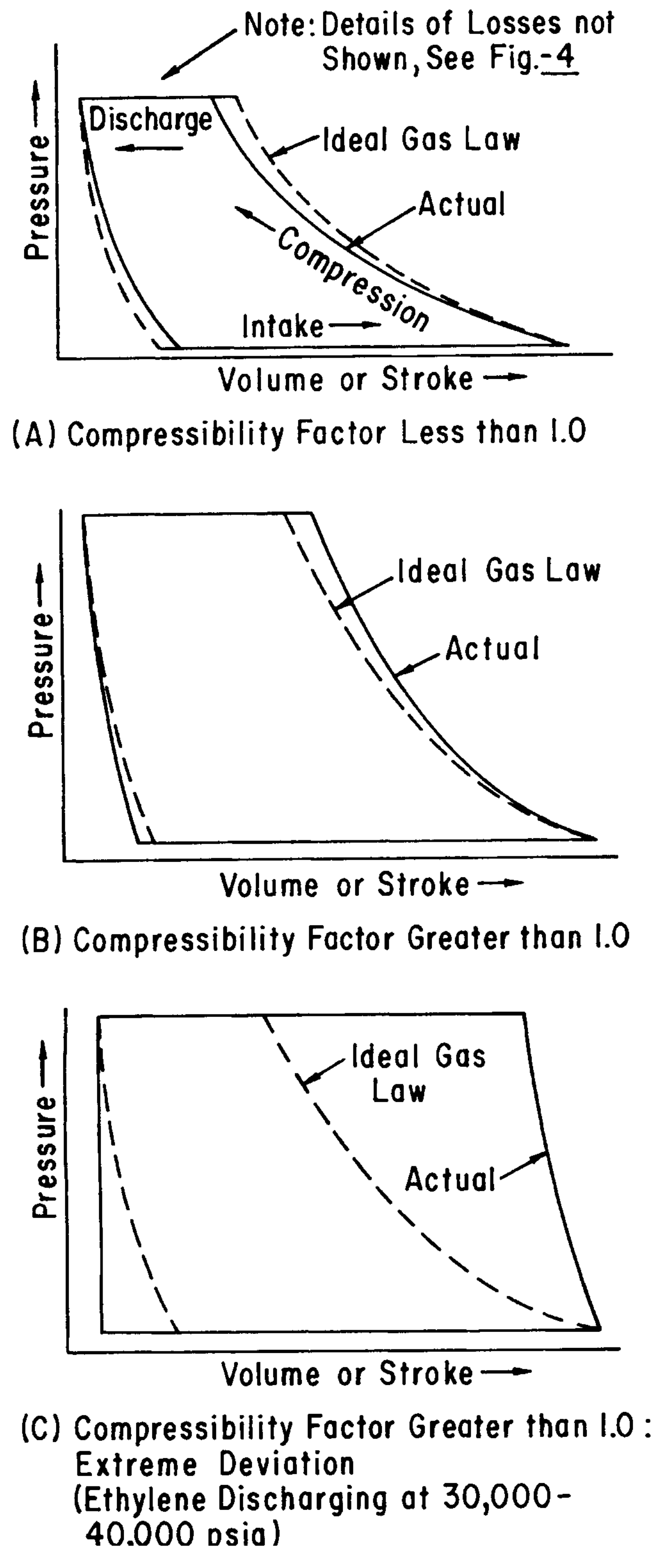


Figure 12-16A–C. Deviations from ideal gas law.

(Text continues from page 402)

Table 12-2
Comparison of Performance for Propane

	Actual	Ideal
Volumetric efficiency	0.802	0.835
Cfm at inlet conditions	802	835
Specific volume at inlet, ft^3/lb	1.160	1.314
Lb handled/min	691	635
Basic horsepower required	388	425
Horsepower/lb	0.561	0.670

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For a perfect gas:

$$c_p - c_v = R \qquad (12\text{-}29)$$

For real gases the relationship applies:[74]

$$c_p - c_v = R/J \qquad (12\text{-}30)$$

$$(c_p/c_v)_{ideal} = c_p/(c_p - R) \qquad (12\text{-}31)$$

where c_p and c_v are specific heats at constant pressure and constant volume respectively, Btu/lb-°R.

R = gas constant, ft-lb/lb-mol-°R (see appendix)
J = Joules' constant = 778 ft-lb/Btu

or, for a real gas:[75]

$$\frac{c_p}{c_v} = \frac{c_p}{c_p - (c_p - c_v)} \qquad (12\text{-}32)$$

From Edmister[76]; $\Delta c_p = 1.44[(c_p - c_v{}^\circ)/k_2]$,

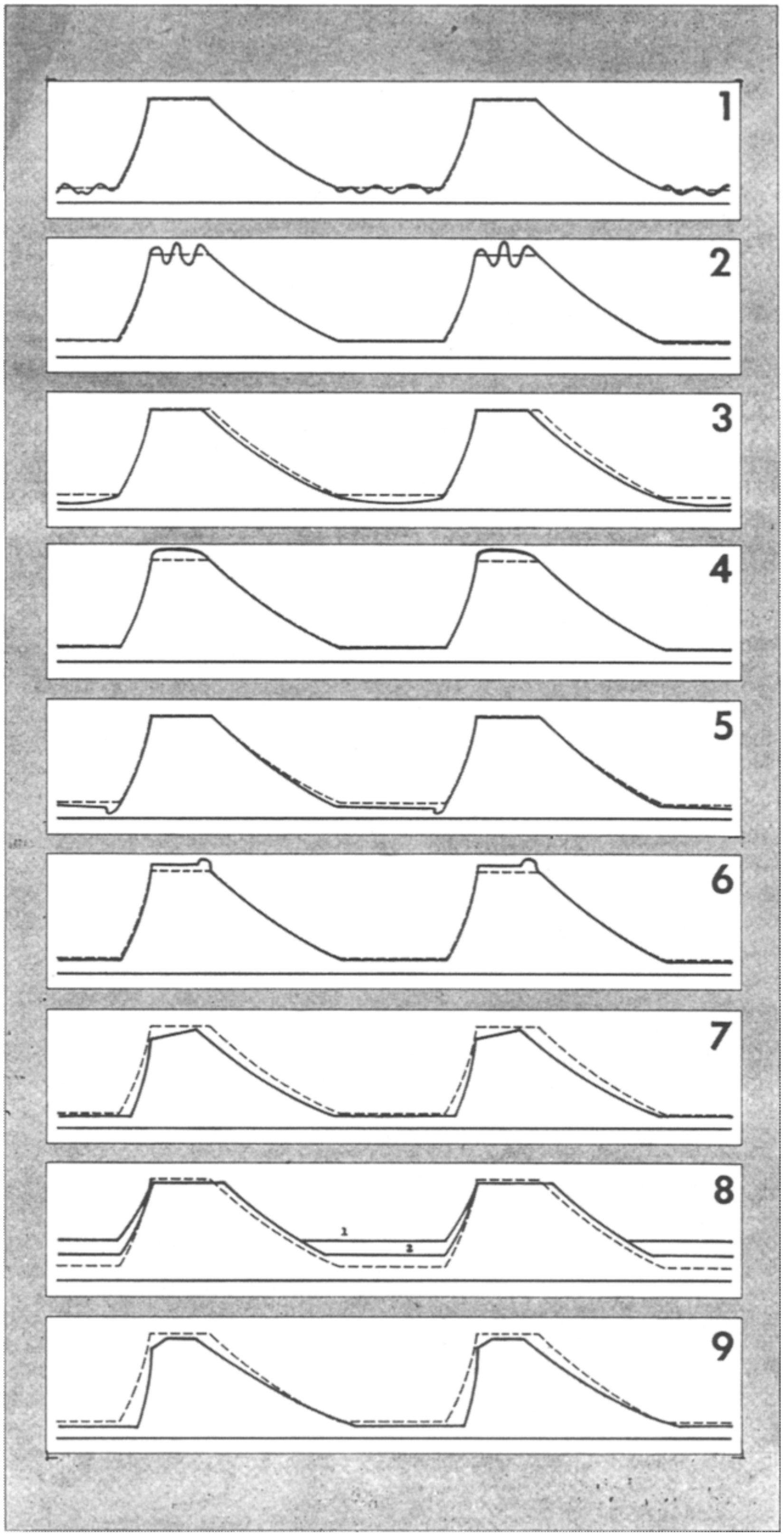
where Δc_p = Btu/(lb-mol)(°R)
$c_p{}^\circ$ = mol heat capacity at ideal gas state
R = universal gas constant = 1,545 ft-lb/lb_m-°R.
For dry air: R = 53.35 ft-lb./lb_m-°R.
R′ = gas constant for a specific gas 1,545/(mol wt)

From combined Boyle's and Charles' Law Equation of State for Perfect Gas:

$$Pv = RT/c_p = RT \qquad (12\text{-}33)$$

v = specific volume, ft^3/lb_m
P = absolute pressure, lb/ft^2 abs
R = gas constant, ft-lb/lb_m-°R
T = absolute temperature, °R (Rankine)
C_p = conversion factor = 1.0

For real gases: Pv = ZRT
z = compressibility factor



1—Suction Valve Chatters

Probably due to weak valve springs, and may result in broken valve plate or a leaky valve.

2—Discharge Valve Chatters

Shows weak springs in the discharge valves, and will result in a broken valve plate or a leaky valve, which in turn will result in cylinder heating and loss of horsepower.

3—Suction Passage Too Small

In addition to too small a suction passage, too small a valve lift could also be indicated.

4—Discharge Passage Too Small

In addition to too small a discharge passage, too small a valve lift could also be indicated.

5—Suction Valve Spring Too Stiff

Too stiff a suction valve means a loss of horsepower. Valve spring of proper tension should be installed.

6—Discharge Valve Spring Too Stiff

Too stiff a discharge valve spring likewise results in loss of horsepower. Valve spring of proper tension should be installed here, also.

7—Suction Valve Leaking

Leak may be in either the valve or the valve gasket.

8—Discharge Valve Leaking

Curve 1 indicates a badly leaking discharge valve; curve 2 a slightly leaking one. Leak may be in the valve or in the valve gasket.

9—Piston Ring Leaking

Leaky piston rings may be due to worn rings, out of round compressor cylinders, or weak expander rings used with plastic-type piston rings.

Figure 12-16D. Typical compressor ailments and how they look on P-T diagrams. (Used by permission: Palmer, E. Y. *Petroleum Processing*, p. 884, June 1954). ©*National Petroleum News,* Adams Business Media.)

Table 12-3
Compressibility Factors, Z

Propane		24% Nitrogen–76% Hydrogen			
Pressure, Psia	Z	Psia	Z	Psia	Z
100	0.884	1,600	1.061	400	0.954
160	0.838	2,400	1.092	500	0.953
220	0.800	3,500	1.129	600	0.955
300	0.765	4,800	1.172	700	0.957

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Table 12-4
Approximate Ratio of Specific Heats ("k" values) for Various Gases

Gas	Symbol	Mol wt	k @ 14.7 psia 60°F	150°F	Density @ 14.7 psi & 60°F lb/ft³
Monatomic	He, Kr, Ne, Hg		1.67		
Most diatomic	O_2, N_2, H_2, etc.		1.4		
Acetylene	C_2H_2	26.03	1.3	1.22	0.0688
Air		28.97	1.406	1.40	0.0765
Ammonia	NH_3	17.03	1.317	1.29	0.0451
Argon	A		1.667		0.1056
Benzene	C_6H_6	78.0	1.08	1.09	0.2064
Butane	C_4H_{10}	58.1	1.11	1.08	0.1535
Isobutane	C_4H_{10}	58.1	1.11	1.08	0.1578
Butylene	C_4H_8	56.1	1.1	1.09	0.1483
Iso-butene	C_4H_8	56.1	1.1	1.09	0.1483
Carbon dioxide	CO_2	44.0	1.3	1.27	0.1164
Carbon monoxide	CO	28.0	1.4	1.4	0.0741
Carbon tetrachloride	$C\,Cl_4$	153.8	1.18		0.406
Chlorine	Cl_2	70.9	1.33		0.1875
Dichlorodifluoromethane	$C\,Cl_2F_2$	120.9	1.13		
Dichloromethane	CH_2Cl_2	84.9	1.18		0.2245
Ethane	C_2H_6	30.0	1.22	1.17	0.0794
Ethylene	C_2H_4	28.1	1.25	1.21	0.0741
Ethyl chloride	C_2H_5Cl	64.5	1.13		0.1705
Flue gas			1.4		
Helium	He	4.0	1.667		0.01058
Hexane	C_6H_{14}	86.1	1.08	1.05	0.2276
Heptane	C_7H_{16}	100.2		1.04	0.264
Hydrogen	H_2	2.01	1.41	1.40	0.0053
Hydrogen chloride	HCl	36.5	1.48		0.09650
Hydrogen sulfide	H_2S	34.1	1.30	1.31	0.0901
Methane	CH_4	16.03	1.316	1.28	0.0423
Methyl chloride	CH_3Cl	50.5	1.20		0.1336
Natural gas (approx.)		19.5	1.27		0.0514
Nitric oxide	NO	30.0	1.40		0.0793
Nitrogen	N_2	28.0	1.41	1.40	0.0743
Nitrous oxide	N_2O	44.0	1.311		0.1163
Oxygen	O_2	32.0	1.4	1.39	0.0846
Pentane	C_5H_{12}	72.1	1.06	1.06	0.1905
Propane	C_3H_8	44.1	1.15	1.11	0.1164
Propylene	C_3H_6	42.0	1.16		0.1112
Sulfur dioxide	SO_2	64.1	1.256		0.1694
Water vapor (steam)	H_2O	18.0	1.33*	1.32	0.04761

*At 212°F

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Compressor Performance Characteristics

1. Piston Displacement

Piston displacement is the actual volume of the cylinder displaced as the piston travels its stroke from the start of the compression (condition (1)) to the end of the stroke (condition (e)) of Figure 12-12 expressed as ft³ of volume displaced per minute. Displacement values for specific cylinder designs are available from the manufacturers, Table 12-6. Neerken[41] is a useful reference. Reciprocating compressors are usually rated in terms of piston displacement, which is the net volume in ft³ per minute displaced by the moving piston.[57] Note that the piston does not move through the clearance volume of Figure 12-12; therefore this volume is not displaced during the stroke.

For single-acting cylinder (Figure 12-4A)

$$PD = A_p s(\text{rpm})/1{,}728 \tag{12-34}$$

where PD = piston displacement, cfm
A_p = cross-sectional net area of piston, in.² If cylinder is head-end, A_p is total area of piston; if cylinder is crank-end, A_p is net area of piston area minus rod cross-section area.
s = stroke length, in.
Rpm = revolutions per minute of crank shaft or number of compression strokes per minute

For double-acting cylinder (Figure 12-4B):
The displacement of the head end and crank end of the cylinder must be added for the total displacement.

The displacement of the crank end is less than that of the head end by the volume equivalent to the piston rod displacement. For a multistage unit, the piston displacement is often only given for the first stage.[16]

$$PD = \frac{A_p s(\text{rpm})}{1{,}728} + \frac{(A_p - A_r)s(\text{rpm})}{1{,}728} \tag{12-35}$$

$$PD = (A_p - A_r/2)2s(\text{rpm})/1{,}728 \tag{12-35A}$$

where A_r = cross-sectional area of piston rod, in.²

2. Compression Ratio

The compression ratio is the ratio, R_c, of the absolute discharge pressure to the absolute suction pressure of the cylinder.

$$P_2/P_1 = R_c \tag{12-36}$$

Table 12-5
Constants for Molal Heat Capacity

Gas	Formula	Molecular Weight	Critical Press, psia	Critical Temp, °R	A	B
Air		28.97	546.7	238.4	6.737	0.000397
Ammonia	NH_3	17.03	1,638	730.1	6.219	0.004342
Carbon dioxide	CO_2	44.01	1,073	547.7	6.075	0.005230
Carbon monoxide	CO	28.01	514.4	241.5	6.780	0.000327
Hydrogen	H_2	2.016	305.7	72.47	6.662	0.000417
Hydrogen sulfide	H_2S	34.07	1,306	672.4	7.197	0.001750
Nitrogen	N_2	28.02	492.3	226.9	6.839	0.000213
Oxygen	O_2	32.00	730.4	277.9	6.459	0.001020
Sulfur dioxide	SO_2	64.06	1,142	774.7		
Water	H_2O	18.02	3,200	1,165	7.521	0.000926
Methane	CH_4	16.04	673.1	343.2	4.877	0.006773
Acetylene	C_2H_2	26.04	911.2	563.2	6.441	0.007583
Ethene	C_2H_4	28.05	748.0	509.5	3.175	0.013500
Ethane	C_2H_6	30.07	717.2	549.5	3.629	0.016767
Propene	C_3H_6	42.08	661.3	656.6	4.234	0.020600
Propane	C_3H_8	44.09	617.4	665.3	3.256	0.026733
1-Butene	C_4H_8	56.11	587.8	752.2	5.375	0.029833
Isobutene	C_4H_8	56.11	580.5	736.7	6.066	0.028400
Butane	C_4H_{10}	58.12	530.7	765.3	6.188	0.032867
Isobutane	C_4H_{10}	58.12	543.8	732.4	4.145	0.035500
Amylene	C_5H_{10}	70.13	593.7	853.9	7.980	0.036333
Isoamylene	C_5H_{10}	70.13	498.2	836.6	7.980	0.036333
Pentane	C_5H_{12}	72.15	485.0	846.7	7.739	0.040433
Isopentane	C_5H_{12}	72.15	483.5	829.7	5.344	0.043933
Neopentane	C_5H_{12}	72.15	485.0	822.9	4.827	0.045300
Benzene	C_6H_6	78.11	703.9	1,011	−0.756	0.038267
Hexane	C_6H_{14}	86.17	433.5	914.3	9.427	0.047967
Heptane	C_7H_{16}	100.2	405.6	976.8	11.276	0.055400

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where P_1 = initial suction pressure, absolute units
P_2 = cylinder discharge pressure at cylinder flange, absolute units

Compression ratios usually vary between 1.05–7 per stage; however, a ratio of 3.5–4.0 per stage is considered maximum for most process operations. Quite often temperature rise of the gas during the compression dictates a limit for the safe or reasonable pressure rise. The maximum temperature rise is governed either by the maximum operating temperature of the compressor cylinder or by the maximum temperature the gas can withstand before decomposition, polymerization, or even auto ignition as for chlorine, acetylene, etc. Because the volumetric efficiency decreases with an increase in compression ratio, this also adds to the selection of a reasonable limiting discharge pressure. With a known maximum temperature, the maximum ratio of compression can be calculated from the adiabatic temperature rise relation.

Optimum minimum horsepower occurs when the ratios of compression are equal in all cylinders for multistage units. With external cooling of the gas between stages, it is necessary to make reasonable allowances for pressure drops through the intercoolers and take this into account when setting the compression ratios:

a. Ideal (no intercooling), for four stages (cylinders)

$$P_2/P_1 = P_3/P_2 = P_4/P_3 \quad (12\text{-}37)$$

b. Actual (with intercooling)

$$P_{i1}/P_1 = P_{i2}/P_{i1}' = P_{i3}/P_{i2}' = \ldots P_{fy}/P_{iy}' \quad (12\text{-}38)$$

where 1, 2, 3, . . . y = conditions of gas across a cylinder represented by (1) for first stage, (2) for second stage, etc.
i = interstage discharge pressure condition, *immediately at cylinder.*
Prime (′) = interstage discharge condition, reduced by the pressure drop through the intercoolers, valves, piping, etc.; therefore, a prime represents *actual* pressure to suction of succeeding cylinder in multistage cylinder system.

Table 12-6
Typical Reciprocating Air Compressor Data

Single-Stage Horizontal

Size, in.	rpm	Max. Press., psi	Piston Displacement, cfm
5×5	550	150	61
6×5		100	88
7×5		60	121
8×5		40	157
10×5		20	248
6×7	450	150	100
7×7		100	138
8×7		60	180
10×7		35	283
12×7		20	410
8×9	360	135	184
9×9		100	234
10×9		75	290
12×9		40	420
15×9		20	658
10×11	327	125	321
12×11		100	465
14×11		60	635
15×11		50	730
17×11		30	940
19×11		20	1,174
20×11		15	1,300
12×13	300	125	502
14×13		100	686
17×13		55	1,016
19×13		40	1,270
20×13		35	1,410
23×13	277	20	1,717
26×13		12	2,202

Two-Stage, Angle Vertical

Size, in.	rpm	Piston Displacement, cfm
$11\frac{1}{4}/7 \times 7$	600	478
$13\frac{1}{2}/8\frac{1}{2} \times 7$	600	690
$14\frac{1}{2}/9\frac{1}{2} \times 7$	600	798
$16/10\frac{1}{2} \times 7$	600	973
$18\frac{1}{2}/11\frac{1}{2} \times 8\frac{1}{2}$	514	1,351
$20\frac{1}{2}/13 \times 8\frac{1}{2}$	514	1,662
$\frac{16}{16}/14\frac{1}{2} \times 9\frac{1}{2}$	450	1,975
$\frac{17\frac{3}{4}}{17\frac{3}{4}}/16 \times 9\frac{1}{2}$	450	2,412

Two-Stage, Horizontal Duplex

Size, in.	rpm	Piston Displacement, cfm
$21/13 \times 14$	277	1,546
$23/14 \times 14$	277	1,858
$24/15 \times 17$	257	2,275
$28/17 \times 19$	225	3,031
$30\frac{1}{2}/18\frac{1}{2} \times 22$	200	3,704
$34\frac{1}{2}/21 \times 25$	180	4,847
$\frac{28}{28}/\frac{17}{17} \times 19$	225	6,065
$\frac{30\frac{1}{2}}{30\frac{1}{2}}/\frac{18\frac{1}{2}}{18\frac{1}{2}} \times 22$	200	7,396
$\frac{34\frac{1}{2}}{34\frac{1}{2}}/\frac{21}{21} \times 25$	180	9,673
$\frac{36\frac{1}{2}}{36\frac{1}{2}}/\frac{22}{22} \times 25$	180	10,808
$\frac{39}{39}/\frac{23}{23} \times 27$	164	12,189

Designation numbers in table for multiple cylinders:

Bore of first stage/bore of second stage × stroke, all in inches.

For example: $\frac{16}{16}/14\frac{1}{2} \times 9\frac{1}{2}$

There are two first-stage cylinders, 16-in. dia. in parallel, one $14\,{}^1\!/_2$-in. second-stage cylinder and all on $9\,{}^1\!/_2$-in. stroke length.

Used by permission: "Feather Valve Compressor Selection Handbook," Worthington bul. L-600-B16. Dresser-Rand Company.

f = final or discharge pressure from multistage unit.

Compression ratios across stages:

$R_1 = P_{i1}/P_1$
$R_2 = P_{i2}/P'_{i1}$
$R_3 = P_{i3}/P'_{i2}$
$\ldots R_f = P_{fv}/P'_{iv}$

$$R_1 = R_2 = R_3 = \ldots R_f = \sqrt[y]{R_t} \quad (12\text{-}39)$$

where R_t = overall compression ratio of unit = P_f/P_1

For two-stage, compression per stage is

$$R_1 = R_2 = \sqrt{P_{f2}/P_1} \quad (12\text{-}39A)$$

For five stages:

$$R_1 = R_2 = R_3 = R_4 = R_5 = \sqrt[5]{P_{f5}/P_1} \quad (12\text{-}39B)$$

It is common practice to use intercoolers on multistage machines. The function of the intercooler is to cool the gas to as near the original suction temperatures as practical with as little pressure drop as possible. With temperature-sensitive material, this is essential. This cooling effects a savings in required brake horsepower as it essentially is cooling at constant pressure and results in less volume of gas to be handled by the next cylinder. To effect the greatest saving, the coldest cooling practically available should be used.

In some cases, it is desirable to use two-stage compression without intercooling. If the composition of the gas must remain constant throughout the compression and the temperature does not limit, intercoolers cannot be used if condensables are present. Sometimes two stages are used on low "k" or "n" value gases to improve the volumetric efficiency. When this is the case and high compression temperatures or economy of operation do not control, it may be advantageous to omit the intercooler.

Note that when intercoolers are not used, the compressor jacket water should be 10–15°F greater than the interstage dew point. This will require warm jacket water through the preceding stage. See the paragraph following.

The intercooler operation does not outwardly affect the theoretical optimum compression ratio per stage. However, it does affect the cumulative horsepower required to do the work of total compression, because all the pressure drop lost must be replaced as horsepower. There is also a gain in performance due to this intercooling as is shown in Figures 12-17A and 12-17B. The allowance for intercooler pressure drop is usually made by increasing the discharge pressure from the cylinder to include one-half of the intercooler pressure drop between stages, and the suction pressure on the following stage is reduced to the other one-half of the pressure drop, when compared to the theoretical pressures with no pressure drop allowance.

Ratio of compression per stage may be calculated:

$$P_f = P_1R^{\gamma} - (\Delta p_1)R^{\gamma-1} - (\Delta p^2)R^{\gamma-2} - (\Delta p_3)R^{\gamma-3} - (\Delta p_4)R^{\gamma-4} \ldots \quad (12\text{-}40)$$

Continue for number of terms on right side of equation equal to number of stages. This is usually best solved by trial and error and can be simplified if most of the ΔP values are assumed equal. It assumes all the intercooler pressure drop is deducted from the suction pressure of the succeeding stage, i.e., first stage intercooler pressure drop is deducted from second stage suction pressure.

P_f = final pressure of multistage set of cylinders
γ = number of compression stages
Δp = pressure drop across interstage coolers, psi
1 = first stage
2 = second stage, etc.

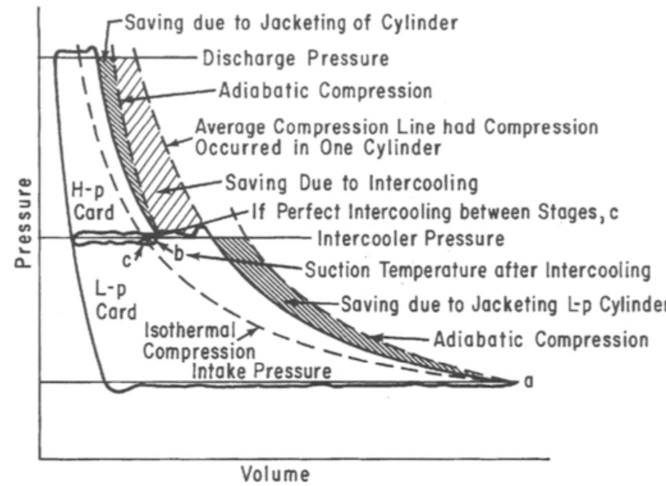


Figure 12-17A. Combined indicator cards from a two-stage compressor showing how cylinder water jackets and intercooler help bring compression line nearer to isothermal. (Used and adapted by permission: Miller, H. H. *Power,* ©1944. McGraw-Hill, Inc., New York. All rights reserved.)

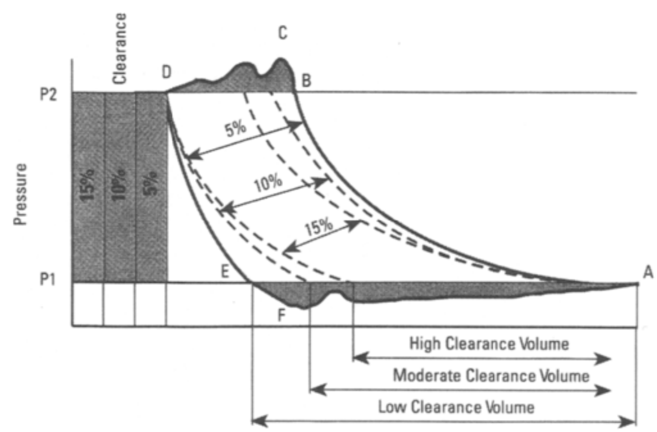


Figure 12-17B. Effects of clearance volume on performance efficiency of reciprocating compressor cylinder (valve design effect). (Used by permission: Livingston, E. H. *Chemical Engineering Progress,* V. 89, No. 2, ©1993. American Institute of Chemical Engineers, Inc. All rights reserved.)

If one half Δp is added to discharge of one stage and one half deducted from suction of next stage:

$$P_f = P_1R^{\gamma} - (^1/_2\,\Delta p_1)R^{\gamma-1} - (^1/_2\,\Delta p_2)R^{\gamma-2} - (^1/_2\,\Delta p_3)R^{\gamma-3} - (^1/_2\,\Delta p_4)R^{\gamma-4} \ldots \quad (12\text{-}41)$$

In practice the ratios for each stage may not work out to be exactly the same. This does not keep the compressor from operating satisfactorily as long as all other factors are handled accordingly.

Compressor Jacket Cooling. The compressor jacket cooling water does not have to be as warm as does the gas engine jacket water. Water 15–20°F warmer than the dewpoint of the gas being compressed will ensure against condensation. A maximum of 15–20°F rise in jacket water temperature is recommended. The flow of water to the jackets should never be throttled in order to maintain this temperature as the lowered velocity tends to facilitate fouling of the jackets.

The amount of heat rejected by compressor jackets varies with the size and type of machine. This heat rejection is usually given as Btu/hr/bhp. Heat rejection to the compressor cylinder will average about 500 Btu/hr/bhp. Some go as low as 130 though, and it is necessary to check with the manufacturer to obtain an accurate figure.

Example 12-1. Interstage Pressure and Ratios of Compression

For two stages of compression, what should be the pressures across the cylinders if the intercooler and piping pressure drop is 3 psi?

Suction to first stage: $P_1 = 0$ psig (14.7 psia)
Discharge from second stage: $P_{f2} = 150$ psig (164.7 psia)

$$\text{Per stage: } R_c = \sqrt{164.7/14.7} = \sqrt{11.2} = 3.34$$

No intercooling:

$$\left.\begin{aligned} P_1 &= 14.7 \text{ psia} \\ P_{i1} &= 3.34(14.7) = 49.2 \text{ psia} \end{aligned}\right\} \quad R_c = 3.34$$

$$\left.\begin{aligned} P_2 &= 49.2 \\ P_{f2} &= 164.7 \end{aligned}\right\} \quad R_c = 3.34$$

With intercooling:

First stage:

$$\left.\begin{aligned} P_1 &= 14.7 \text{ psia} \\ P_{i1} &= 49.2 + (1/2)(3.0) = 50.7 \text{ psia} \end{aligned}\right\} \quad R_c = 3.45$$

Second stage:

$$\left.\begin{aligned} P_{i1}' &= 49.2 - (1/2)(3.0) = 47.7 \text{ psia} \\ P_{f2} &= 164.7 \end{aligned}\right\} \quad R_c = 3.45$$

The example shows that although the ratios per cylinder are balanced, they are each greater than the theoretical. This corresponds to actual operations.

It is important to note that quite often the actual compression ratios for the individual cylinders of a multistage machine will not be balanced exactly. This condition arises as a result of the limiting horsepower absorption for certain cylinder sizes and designs of the manufacturer. In final selection these will be adjusted to give compression ratios to use standard designs as much as possible.

3. Actual Capacity or Actual Delivery, V_a

This is the volume of gas measured at the intake to the first stage of a single or multistage compressor and at stated intake temperature and pressure, ft^3/min. Manufacturer performance guarantees usually state that this capacity is subject to 6% tolerance when intake pressure of first stage is 5 psig or lower and may state a volume tolerance of about 3% for pressures above this 5 psig intake.[44] *It is extremely important to state whether the capacity value has been corrected for compressibility.* At low pressures, compressibility is usually not a factor; however, if conditions are such as to not require the use of compressibility, it is usually omitted and so stated. The actual required capacity may be calculated for process requirements, or if a *known* cylinder is being examined:

$V_a = PD(E_V)$; cfm cylinder will compress at suction pressure and temperature
E_v, = volumetric efficiency, is based on the characteristics of the cylinder. (12-42)

E_v, or sometimes Ev, is the volumetric efficiency of a cylinder and is the ratio of the amount of gas that is actually compressed to the amount of gas that could be compressed if no clearance existed in the cylinder, see Figure 12-12. E_v can be obtained from Figures 12-18A–F.

$$\%E_v = 100 - R_c - (V_{pc})(R_c^{1/k} - 1) \qquad (12\text{-}43)$$

4. Clearance Volume

This is the total volume remaining in the cylinder at the end of the piston stroke. This consists of the volume between the end of the piston and the cylinder head, in the valve ports and the volume in the suction valve guards and the discharge valve seats.[44] See Figures 12-12, 12-17A, and 12-17B.

The effect of clearance volume is shown in Figures 12-17A and 12-17B.[58] The illustrated volumes of 5%, 10%, and 15% usually satisfy a reasonable process compressor range. For example, in the 15% compression slope, ABC will reach pressure shown as P2 sooner than the slope of the 10% curve. Upon re-expansion at the end of the compression stroke DEF, the slope is steeper and allows the gas to enter the cylinder sooner during the suction or intake cycle.[58] Volumetric efficiency increases with a decrease in clearance volume and a decrease in compression ratio.[58] This is the most profound effect, although other design factors do influence the efficiency to a lesser extent. In attempting to balance volumetric and compression efficiency, Livingston[58] points out "for a high compression ratio (6 to 15) clearance volume is the key factor with valve design being secondary. For a compression ratio of 3 to 6, the clearance volume and

(Text continues on page 422)

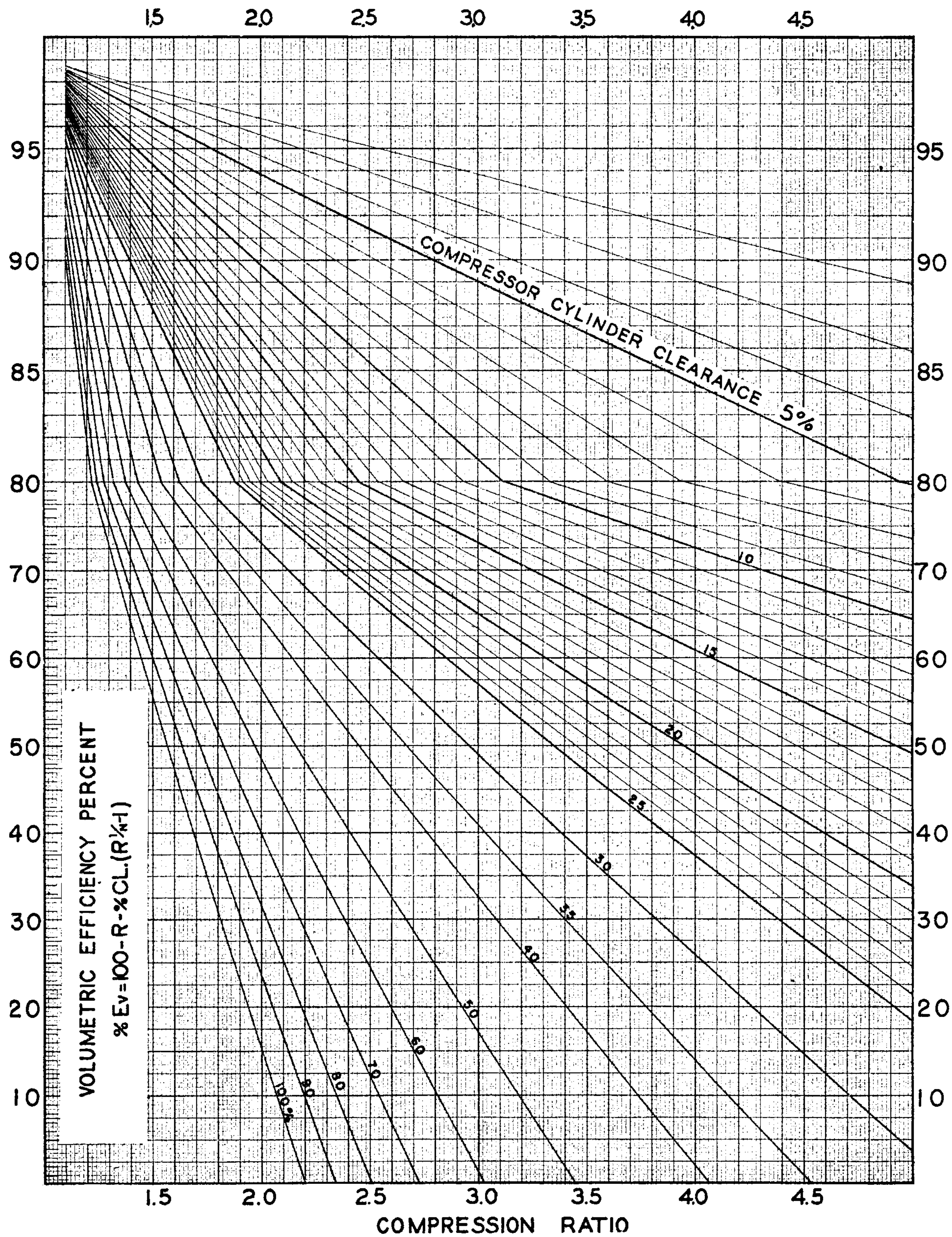


Figure 12-18A. Compressor volumetric efficiency curves for gas with k or n of 1.15. (Used by permission: *Natural Gasoline Supply Men's Association Data Book,* ©1957. Origin Ingersoll-Rand Co. All rights reserved.)

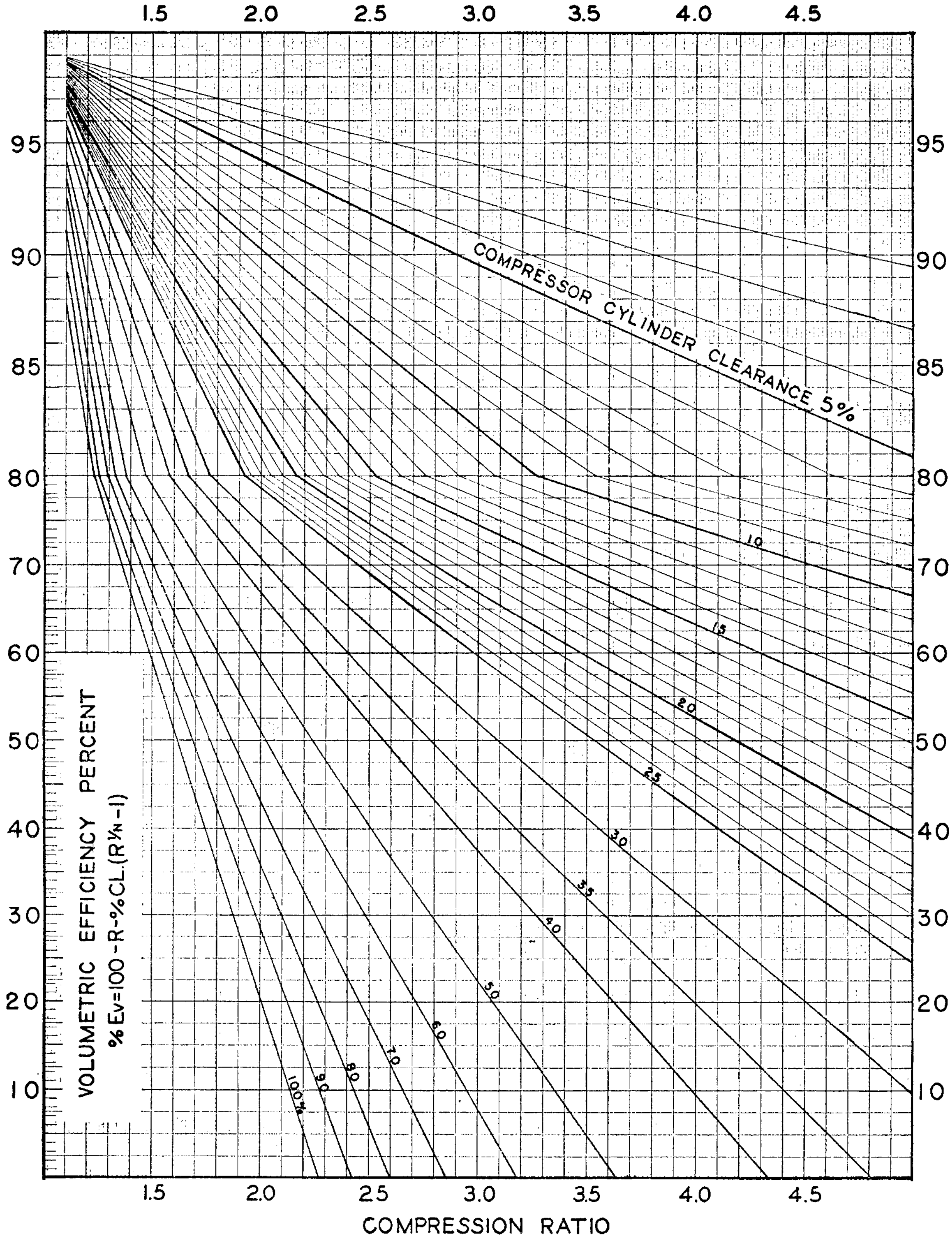


Figure 12-18B. Compressor volumetric efficiency curves for gas with k or n of 1.20. (Used by permission: *Natural Gasoline Supply Men's Association Data Book*, ©1957. Origin Ingersoll-Rand Co. All rights reserved.)

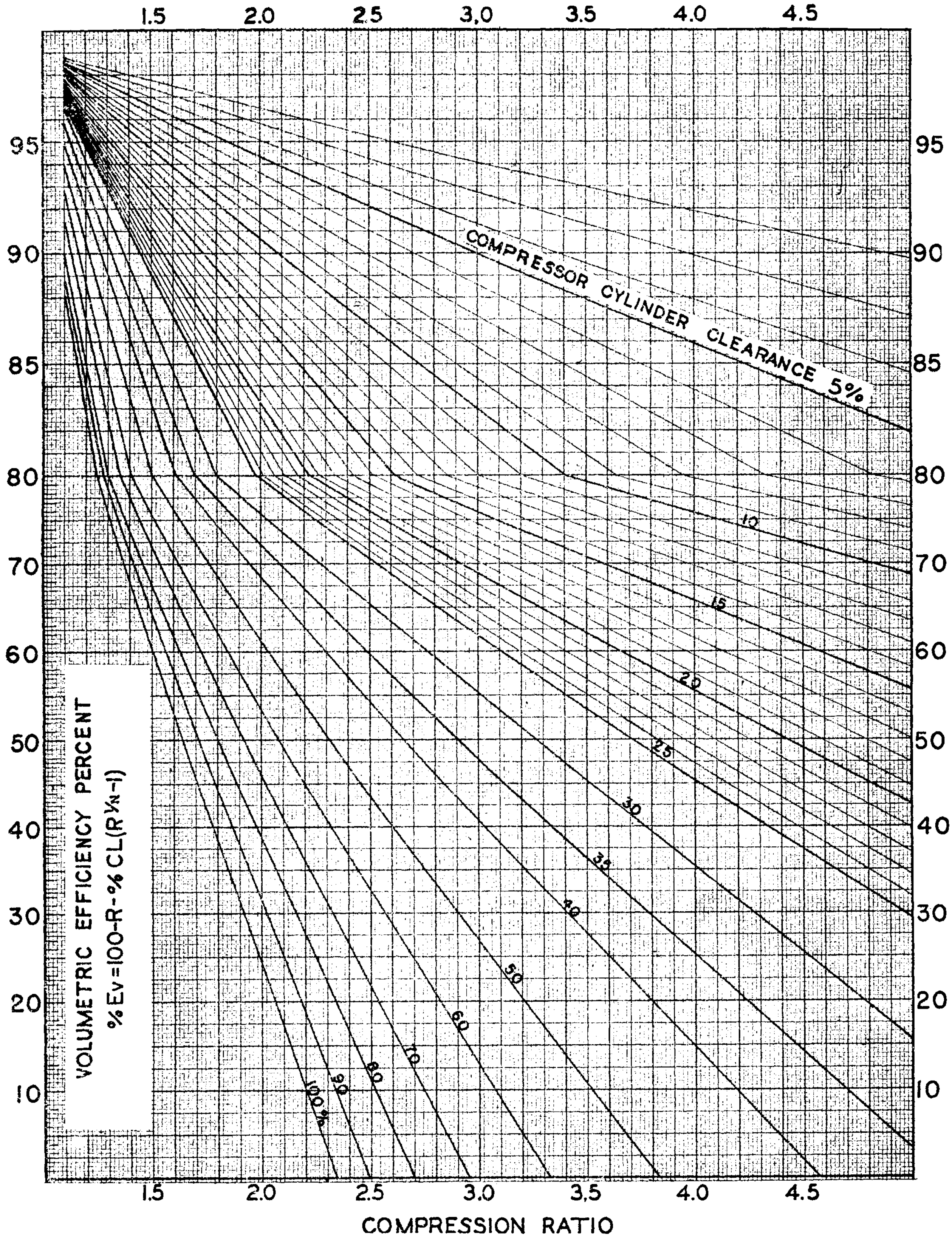


Figure 12-18C. Compressor volumetric efficiency curves for gas with k or n of 1.25. (Used by permission: *Natural Gasoline Supply Men's Association Data Book*, ©1957. Origin Ingersoll-Rand Co. All rights reserved.)

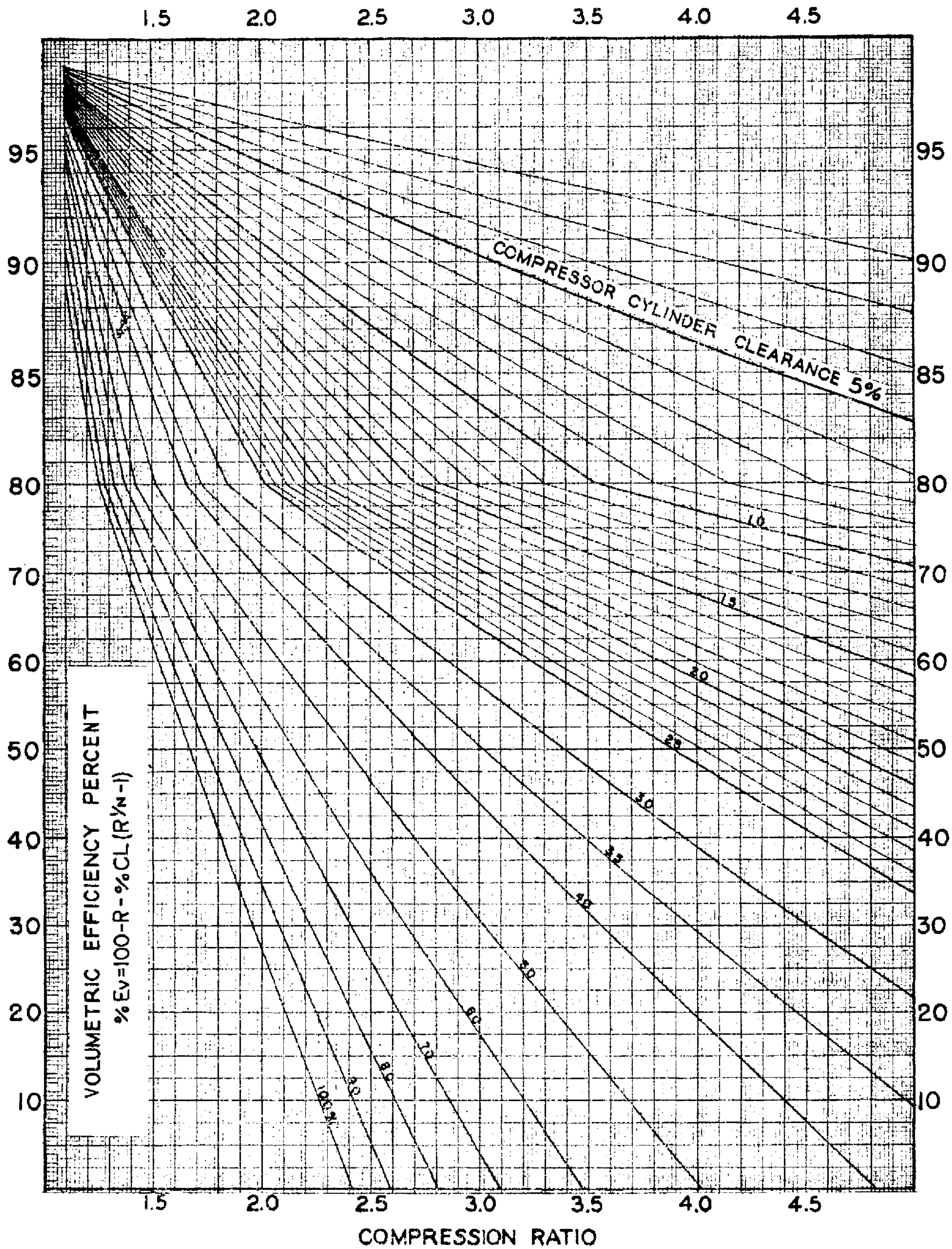


Figure 12-18D. Compressor volumetric efficiency curves for gas with k or n of 1.30. (Used by permission: *Natural Gasoline Supply Men's Association Data Book*, ©1957. Origin Ingersoll-Rand Co. All rights reserved.)

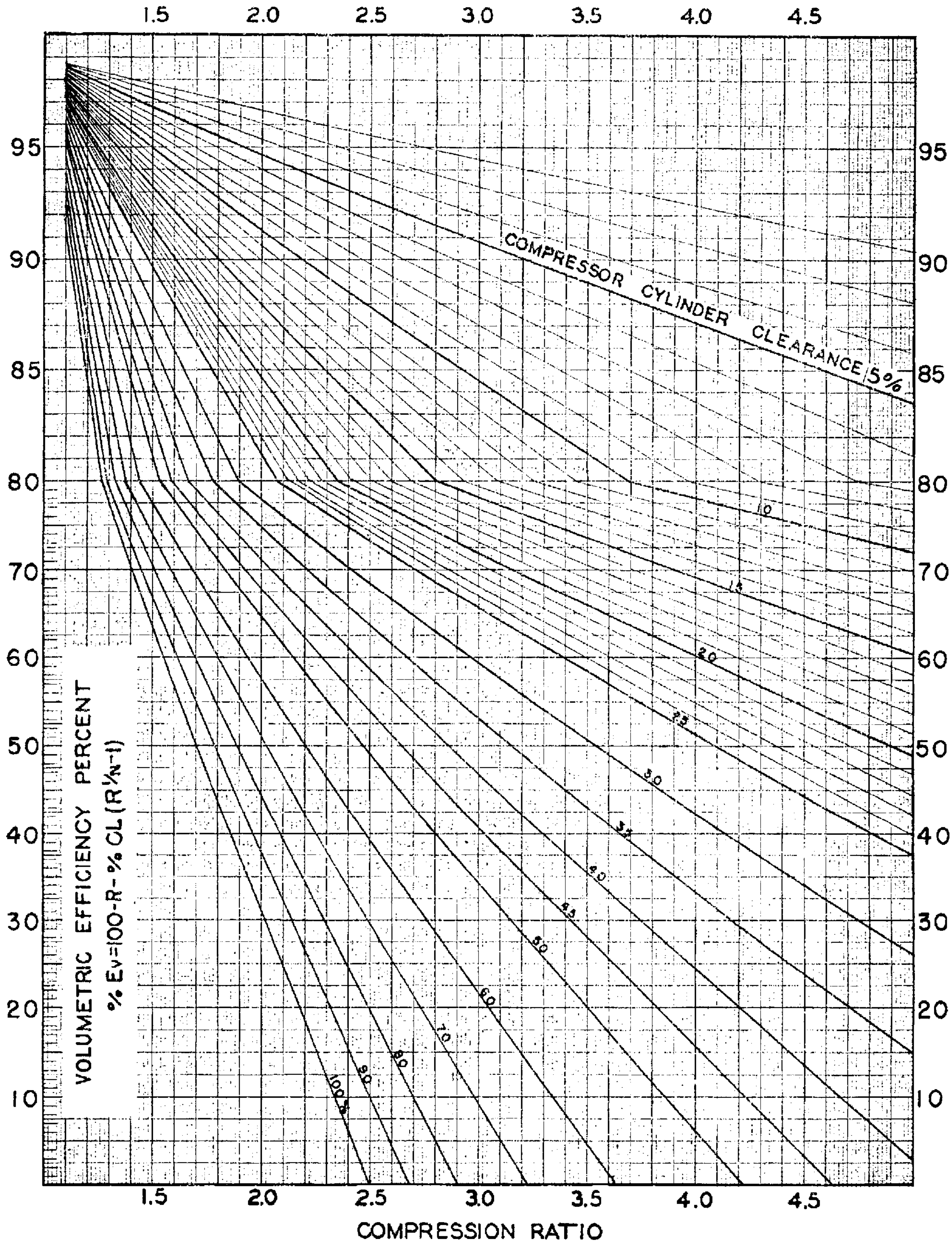


Figure 12-18E. Compressor volumetric efficiency curves for gas with k or n of 1.35. (Used by permission: *Natural Gasoline Supply Men's Association Data Book*, ©1957. Origin Ingersoll-Rand Co. All rights reserved.)

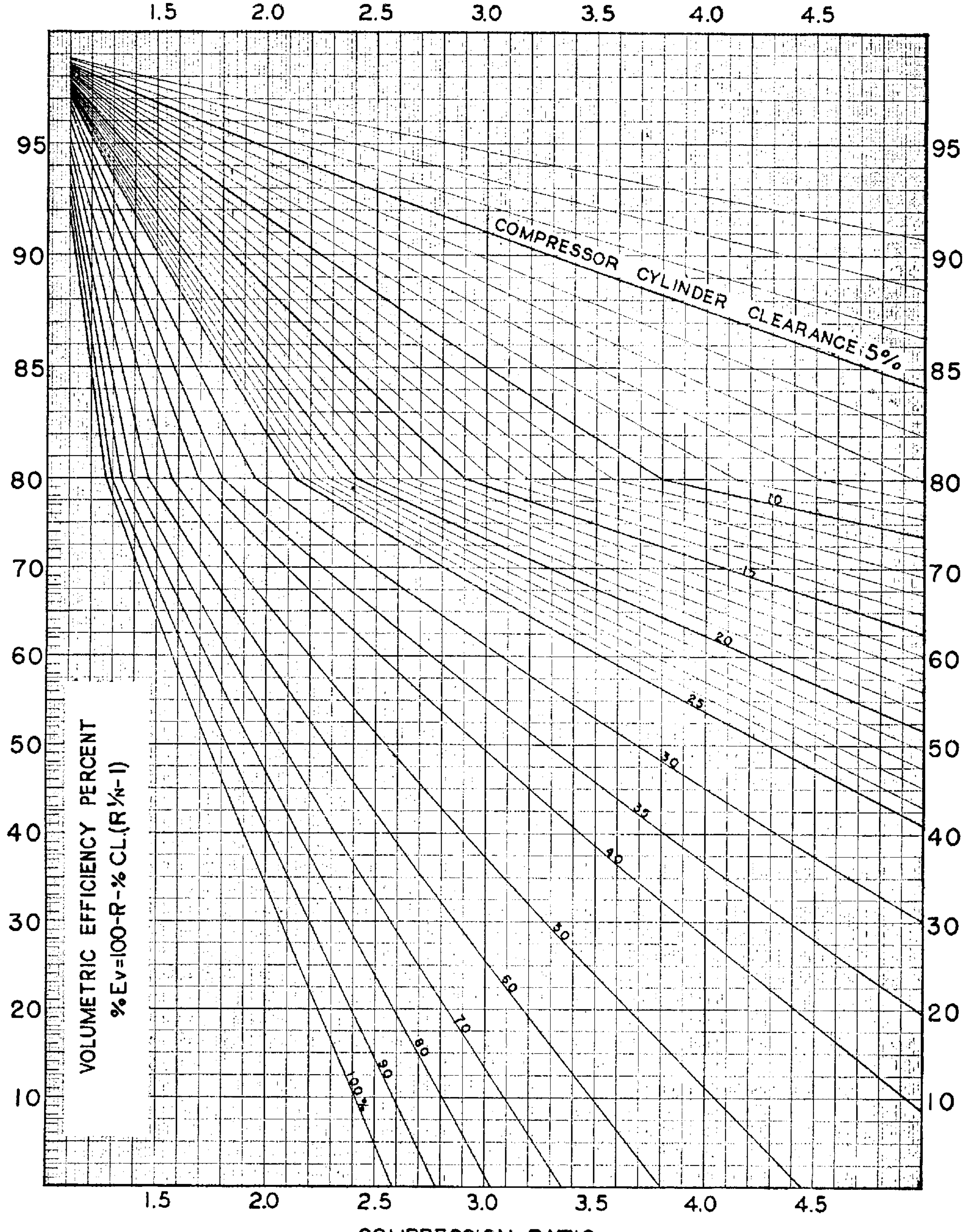


Figure 12-18F. Compressor volumetric efficiency curves for gas with k or n of 1.40. (Used by permission: *Natural Gasoline Supply Men's Association Data Book*, ©1957. Origin Ingersoll-Rand Co. All rights reserved.)

(Text continued from page 415)

valve design should be balanced. For low compression ratio, less than 3, the valve design is the primary factor."

5. Percent Clearance

Percent clearance is the volume % of clearance volume to total actual piston displacement.[44]

$$V_{pc} = \frac{V_c}{PD'}(100) \quad (12\text{-}44)$$

Calculate for each cylinder end.

where V_c = clearance volume, in.^{3}
V_{pc} = percent clearance
PD' = piston displacement, in.^{3}

For double-acting cylinders, the clearance at the head end should be calculated separately from that of the crank end, because for small cylinders, the volume occupied by the piston rod is significant when cylinder unloading is considered.

For double-acting cylinders, % clearance is based on total clearance volume for both the head end and crank end of the cylinder × 100 divided by the *total* net piston displacement. The head and crank end % clearance values will be different due to the presence of the piston rod in the crank end of the cylinder. The % clearance values are available from manufacturers for their cylinders. The values range from about 8% for large 36-in. cylinders to 40% for small 3- and 4-in. cylinders. Each cylinder style is different.

6. Cylinder Unloading and Clearance Pockets

For a discussion of this important performance control topic, refer to "Cylinder Unloading," later in this chapter and Figure 12-27. A reasonable average range is 7–22% with fixed valve pockets.[58]

7. Volumetric Efficiency

Volumetric efficiency is the efficiency of a cylinder performance based on operating experience and actual volume conditions.

$$\%E_v = 100 - R_c - V_{pc}(R_c^{1/k} - 1) \quad (12\text{-}45)$$

where R_c = ratio of compression across an individual cylinder.

Volumetric efficiency may be expressed as the ratio of actual cylinder capacity expressed at actual inlet temperature and pressure conditions, divided by the piston displacement. See Figure 12-17B. Values of E_v may be read from Figures 12-18A–F for values of R_c and V_{pc}.

For example, in a multistage compressor, such as a two-stage compressor, each cylinder does one half of the total work of compression. The low pressure or first stage of the two-stage unit controls the capacity of the overall compressor, i.e., the second stage can handle only the volume of gas passed to it from the discharge of the first-stage cylinder. That is, the gas that passes through the low or first-stage discharge valves must continue on through the compressor's second- (or final in this case) stage cylinder and be discharged at the specified pressure and calculated/actual temperature.

Thus, in all compressors of two or more stages, the volumetric efficiency of the low-pressure cylinder determines the volumetric efficiency of the entire compressor (not recognizing packing leaks).[57]

8. Compression Efficiency (Adiabatic)

Compression efficiency is the ratio of the work required to adiabatically compress a gas to the work actually done within the compressor cylinder as shown by indicator cards, Figures 12-12 and 12-16. The heat generated during compression adds to the work that must be done in the cylinder. Valves may vary from 50–95% efficient depending on cylinder design and the ratio of compression. Compression efficiency (or sometimes termed *volumetric efficiency*) is affected by several details of the systems:

a. Process gas compression ratio across the cylinder.
b. Compressibility of the gas at inlet and discharge conditions, compression ratio.
c. Compression valve action including friction and leakage.
d. Nature of the gas, for example, its ratio of specific heat and compression exponent.
e. Leakage across the piston rings during compression stroke.
f. Type and loss through intake and discharge valves.
g. Moisture or condensibles in the gas being compressed.
h. Clearance volume of cylinder.

The compressor manufacturer can control items a–c, e, f, and h; however, the control of clearance volume at high compression ratios for gases/vapors with low specific heat ratios is of great concern.[58] Compression efficiency is controlled by the clearance volume, valves, and valve pocket design. A decrease in compression efficiency leads to increased power requirements.[58]

9. Mechanical Efficiency

Mechanical efficiency is the ratio of compressor cylinder indicated horsepower to the brake horsepower. Efficiency values range from 90–93% for direct-driven cylinders to 87–90% for steam engine units. The efficiency of the driver is not included.

10. Piston Speed

Piston speed is a useful guide to set relative limits on compressor cylinder selection. It is difficult to establish acceptable and nonacceptable limits because this is best evaluated with operating experience and compressor manufacturer's recommendations.

$$\text{Piston speed} = \frac{(\text{rpm})(\text{s})}{6}, \text{ ft/min} \qquad (12\text{-}46)$$

This is of more significance in corrosive or polymer-forming services than in clean hydrocarbon or air applications. For example in hydrogen chloride and chlorine service using cylinders with either (a) cast iron liners or (b) carbon piston rings, a speed of around 600 ft per min is acceptable.

11. Horsepower

Horsepower is the work done in a cylinder on the gas by the piston connected to the driver during the complete compression cycle. The theoretical horsepower is that required to isentropically (adiabatically) compress a gas through a specified pressure range. The indicated horsepower is the actual work of compression developed in the compressor cylinder(s) as determined from an indicator card.[6] Brake horsepower (bhp) is the actual horsepower input at the crankshaft of the compressor drive. It does not include the losses in the driver itself, but is rather the actual net horsepower that the driver must deliver to the compressor crankshaft.

A. Single Stage.

a. Theoretical Hp

$$= \frac{144}{33{,}000}\left(\frac{k}{k-1}\right)P_1V_1\left[\left(\frac{P_2}{P_1}\right)^{(k-1)/k} - 1\right] \qquad (12\text{-}47)$$

b. Actual Brake Horsepower, Bhp

$$= \frac{144}{33{,}000}\left(\frac{k}{k-1}\right)P_1V_1\left[\left(\frac{P_2}{P_1}\right)^{(k-1)/k} - 1\right](L_o)(F_L)Z_1 \qquad (12\text{-}48)$$

where P_1 = suction pressure, psia
P_2 = discharge pressure, psia
V_1 = suction volume, ft^3/min, at suction conditions
L_o = loss factor, comprised of losses due to pressure drop through friction of piston rings, rod packing, valves, and manifold (see Figure 12-19).
F_L = frame loss for motor-driven compressors only, values range 1.0–1.05 (note, this is *not* a driver efficiency factor).
Z_1 = compressibility factor, based on inlet conditions to cylinder (usually negligible, except at high pressures), see Figures 12-14 and 12-15.

Figure 12-20 is convenient to use in solving the complete theoretical power expression, giving

$$\text{Theoretical hp} = F_wZ_1T_1N_m/2{,}546 \qquad (12\text{-}49a)$$

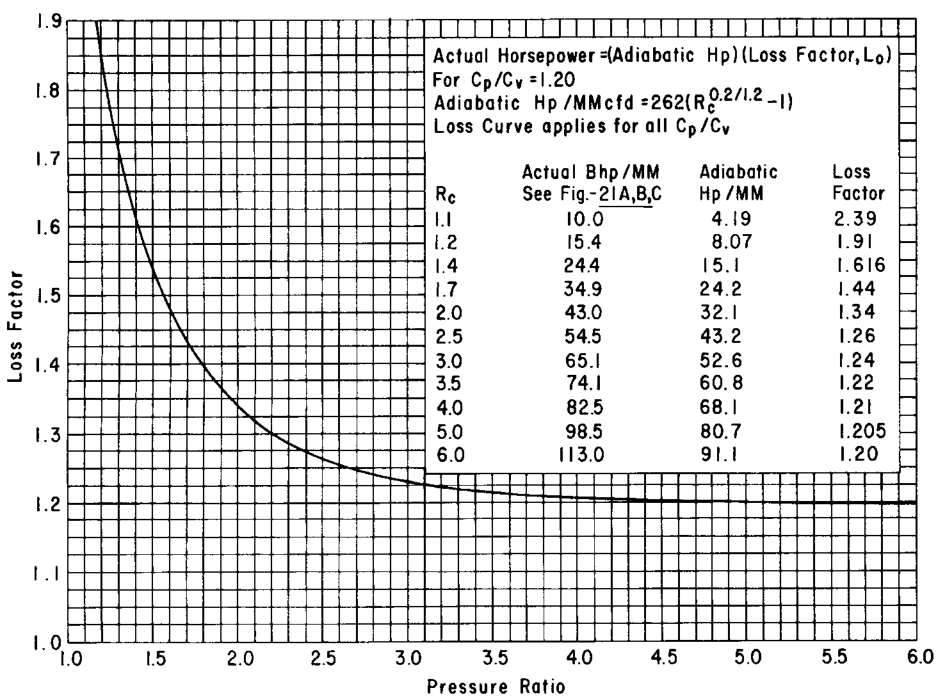


Figure 12-19. Loss factor curve. (Used by permission: Cooper-Cameron Corporation.)

$$\text{where } F_w = R\left(\frac{k}{k-1}\right)\left[\left(\frac{P_2}{P_1}\right)^{(k-1)/k} - 1\right] \quad (12\text{-}49B)$$

R = gas constant = 1.987 Btu/°R
N_m = lb-mol/hr
T_1 = suction or inlet temperature, °R

To obtain actual brake horsepower, bhp, multiply the theoretical hp by L_o and F_L.

The values of n shown on the chart are Edmister's isentropic exponents;[21, 22] however, the chart satisfactorily solves the preceding relation using "k" values.

Loss Factor. The loss factor is a correction factor for standard horsepower curves for high suction pressures at low ratios of compression. The bhp (brake horsepower) is obtained from the curves, Figures 12-21A–C. These curves are a plot of the "n" or "k" value of the gas versus the required brake horsepower (required to compress 1 million ft^3 of gas at 14.4 psia and suction temperatures) for various ratios of compression.

The bhp curves give a value greater than the actual. At high ratios of compression, this deviation is not significant. However, when the compression ratio is less than 2.5 and the suction pressure is initially high, appreciable deviation may be encountered. From a percentage standpoint, this overage is an appreciable part of the total bhp requirements, and a correction should be made by use of the "loss factor" multiplier, Figure 12-19.

When the use of the "loss factor" is indicated and exact thermodynamic data on the gas handled is available, refer to a reliable compressor manufacturer for a more exact correction, which will indicate a lower bhp requirement.

This case is justifiable but must be handled with caution and be accompanied by firm operating conditions. That is, there should be very little change of the suction and discharge temperatures, pressures, or capacity requirements.

For approximate power requirements:[80]

$$\text{Ghp} = \frac{w_1 H}{(33{,}000)(\eta_p)}, \text{ gas horsepower} \quad (12\text{-}50)$$

Adjust for balanced piston leakage, 2%[80]
Add losses for seal hp, (x)

$$\text{shp} = [(\text{ghp})(1.02)] + [\text{losses},(x^*)], \text{ shaft horsepower} \quad (12\text{-}51)$$

* range (60–80) hp depends on number of stages and type of shaft seal

where ghp = gas horsepower
shp = shaft horsepower
H = head, ft
w_1 = weight flow, lb/min
η_p = polytropic efficiency, fraction, for selected units. See Table 12-9B (Centrifugal Compressors)

c. Actual Brake Horsepower, Bhp, (Alternate Correction for Compressibility). The development of Hartwick[29] indicates an approach to correcting the ideal gas horsepower for the effects of compressibility. The results examined with high pressure (a maximum discharge pressure of 15,000 psia) systems give agreement within less than 6% of enthalpy methods.

1. Determine gas specific volume at inlet conditions to cylinder:

$$v = ZRT/(144P), \text{ ft}^3/\text{lb} \quad (12\text{-}52)$$

Obtain Z from compressibility charts, Figure 12-14 (or for specific gas or mixtures, if available).

$$R = 1{,}544/\text{mol wt of gas} \quad (12\text{-}53)$$

2. Determine discharge temperature, T_2, using adiabatic temperature rise Equation 12-62. Use k values for gas or mixture or calculate them by Equation 12-4.
3. Calculate the specific volume at discharge condition, v_2, using Equation 12-52.
4. Determine inlet volume, V_1.

a. Calculate volumetric efficiency from *ideal* equation:

$$E_v' = 1 - V_{pc}'[(P_2/P_1)^{1/k} - 1] = 1 - V_{pc}'(v_1/v_2 - 1), \text{ fraction} \quad (12\text{-}54)$$

Note that this requires or assumes that compressor cylinder clearance V_{pc}' be established. For studies, it may be assumed and the effects calculated. Values range from 5–35% clearance on actual cylinders. Special designs are used for smaller or larger values.

b. Calculate inlet volume.

$$V_1 = (PD)(E_v') \quad (12\text{-}55)$$

5. Determine pseudo compression exponent, k′, to reflect actual shape of compression and re-expansion curves.

$$P_1 v_1^{k'} = P_2 v_2^{k'} \quad (12\text{-}56)$$

6. Calculate horsepower required:

$$\frac{144}{33{,}000}\text{bhp} = \left(\frac{k'}{k'-1}\right) P_1 V_1[(P_2/P_1)^{(k'-1)/k'} - 1](L_o)(F_L) \quad (12\text{-}57)$$

(Text continues on page 429)

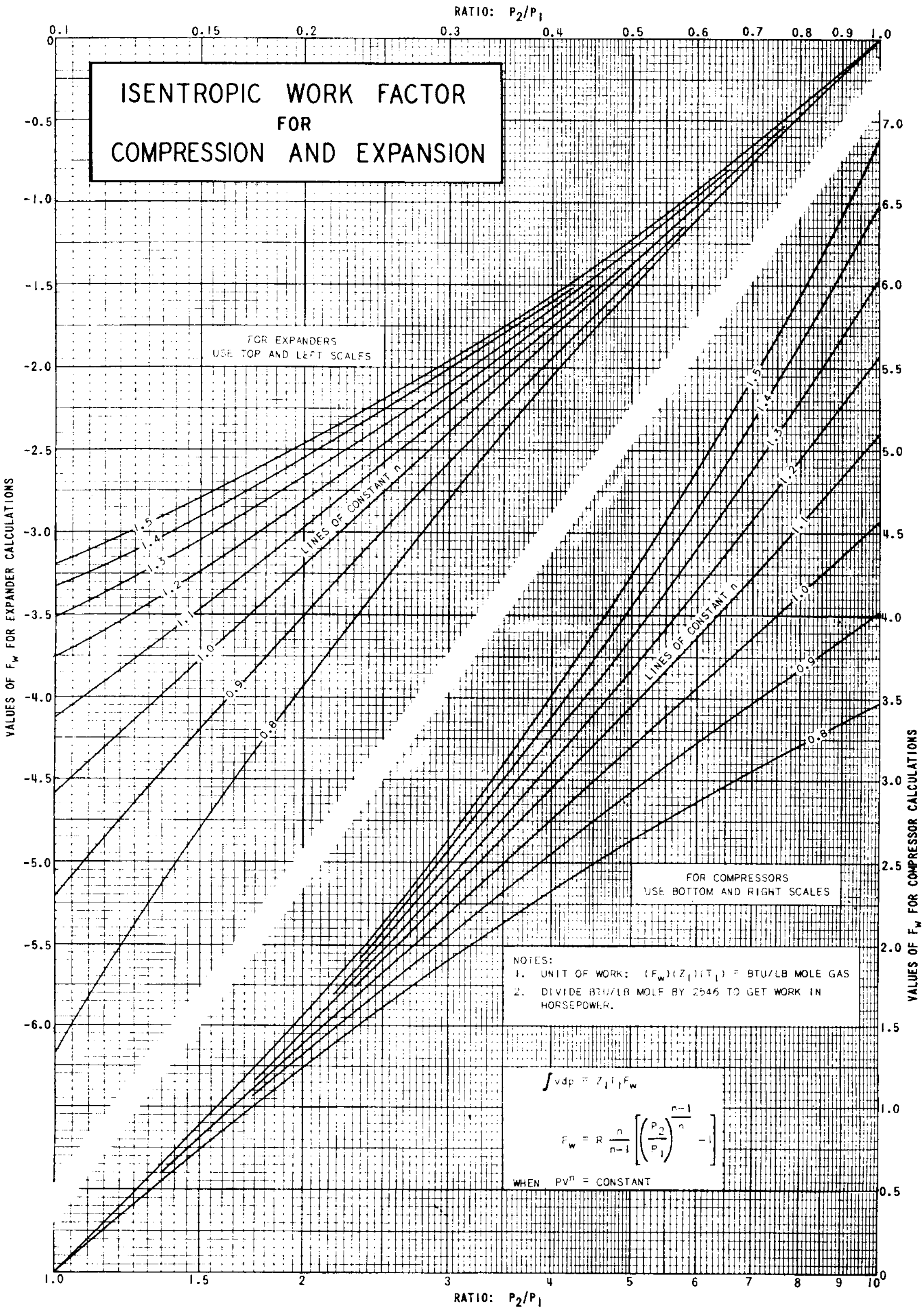


Figure 12-20. Chart for solving theoretical work of compression or expansion. (Used by permission: Edmister, W. C. *Petroleum Refiner,* V. 38, No. 5, ©1959. Gulf Publishing Co. All rights reserved.)

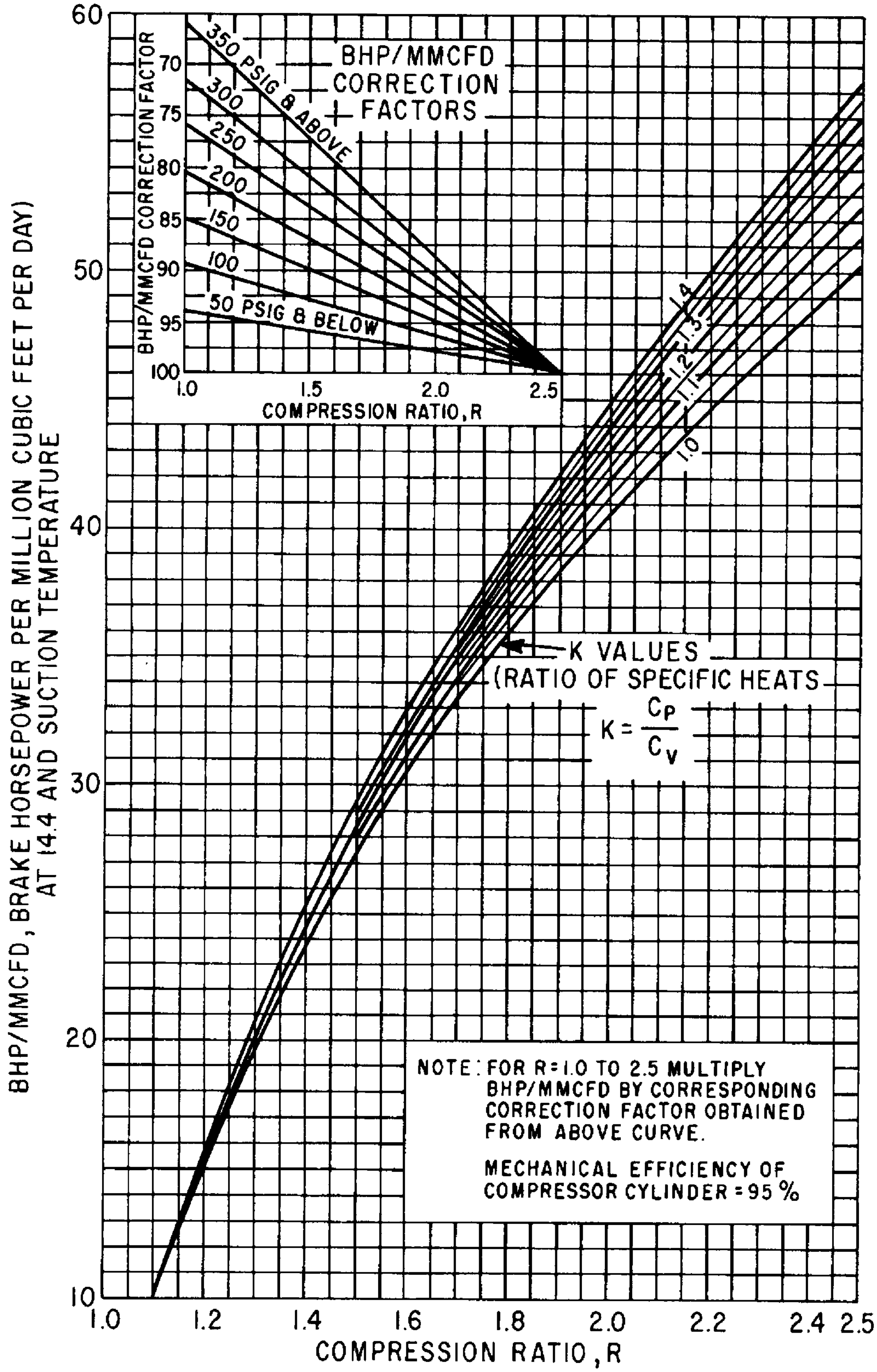


Figure 12-21A. Brake horsepower required to deliver 1 million ft^3 of gas per day, part 1 of 3. (Used by permission: Cooper-Cameron Corporation. All rights reserved.)

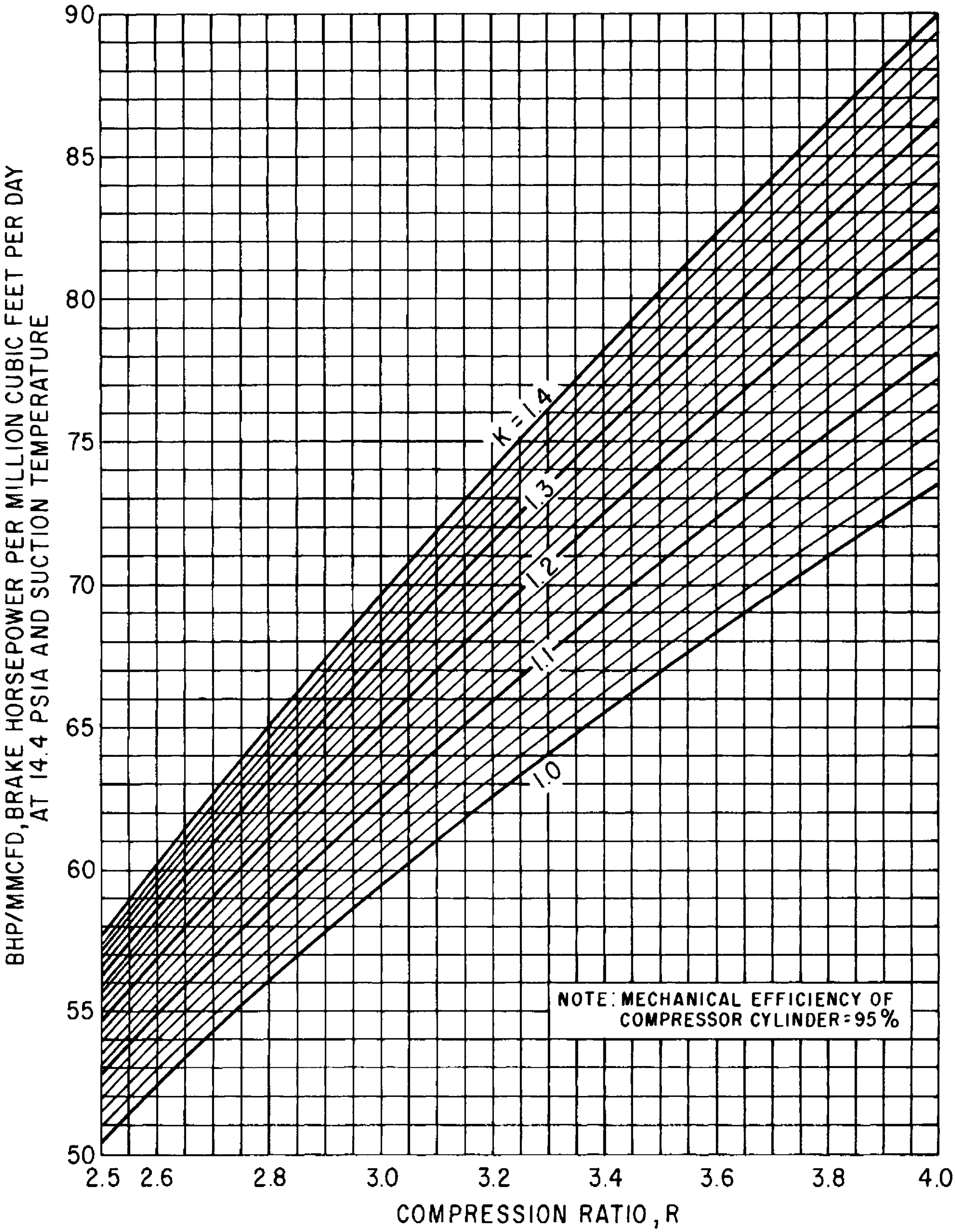


Figure 12-21B. Brake horsepower required to deliver 1 million ft^3 of gas per day, part 2 of 3. (Used by permission: Cooper-Cameron Corporation. All rights reserved.)

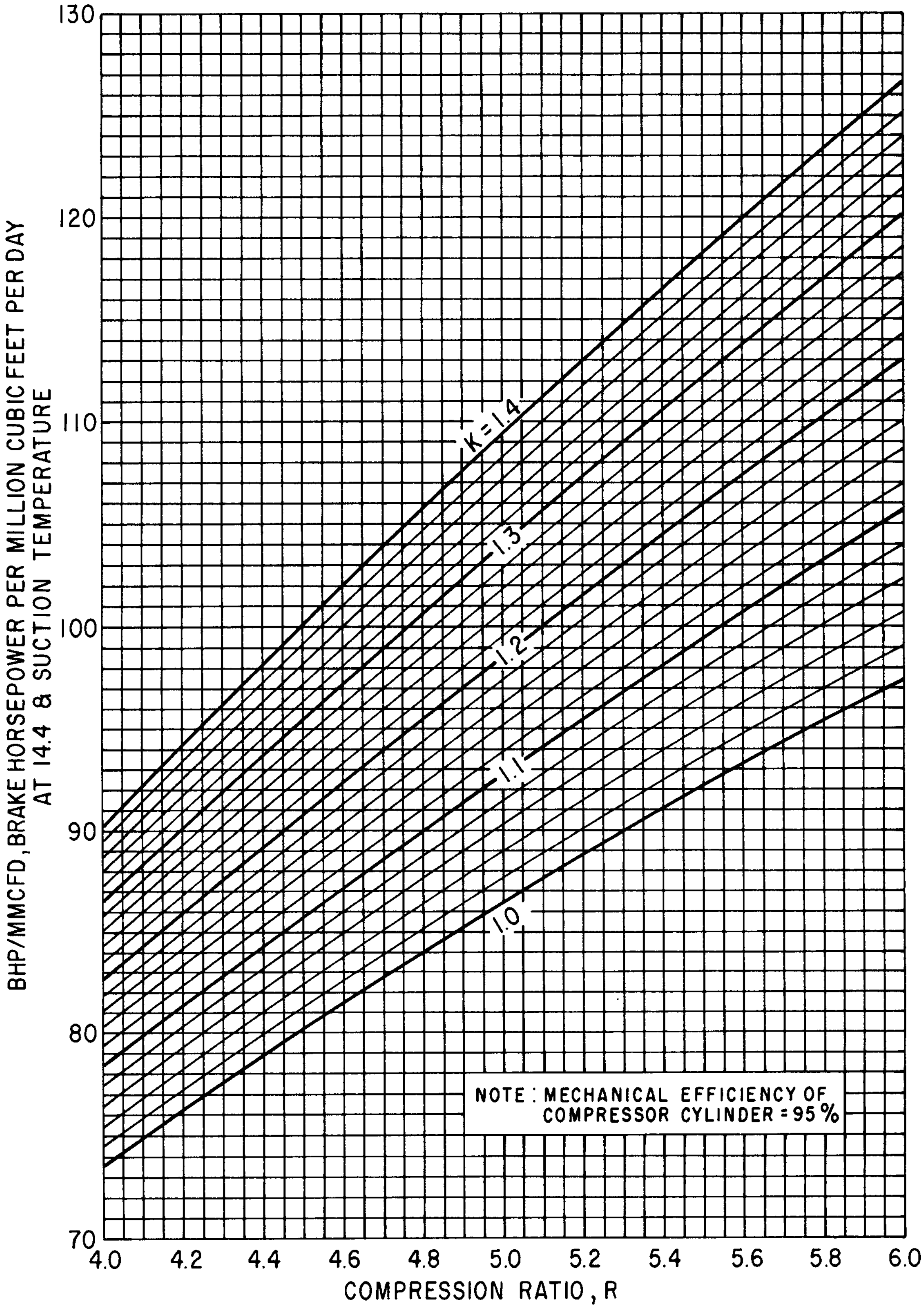


Figure 12-21C. Brake horsepower required to deliver 1 million ft^3 of gas per day, part 3 of 3. (Used by permission: Cooper-Cameron Corporation. All rights reserved.)

(Text continued from page 424)

Another method to account for compressibility is given by Boteler.[6]

d. Approximate Actual Brake Horsepower, Bhp, from Estimating Curves. Many process applications do not require the detailed evaluation of all factors affecting the actual horsepower requirements. A convenient and yet reasonably accurate calculation can be made using Figures 12-21A–C. Note that for low ratios of compression, a correction factor is to be multiplied by the curve reading as shown in Figure 12-21A. A mechanical cylinder efficiency of 95% is included. The curves use the ratio of heat capacities, k. The horsepower *per stage* may be calculated:

$$\text{bhp} = [\text{bhp}/(\text{MMCFD})](C/10^6) \quad (12\text{-}58)$$

(delivered to compressor crank shaft)

$$\text{or, bhp} = (PD)E_vP_1(\text{bhp}/\text{MMCFD})(10^{-4}) \quad (12\text{-}59)$$

where Bhp/MMCFD = brake horsepower required to handle 1×10^6 ft^3 of gas per day, measured at 14.4 psia and suction temperature to cylinder, from Figure 12-21, including correction.

C = capacity of gas to be compressed, ft^3/day, referenced to base conditions of 14.4 psia and suction temperature. Note that corrections for compressibility are not to be included unless the suction temperature at 14.4 psia requires such a correction, assuming most gases require no compressibility correction for pressures as low as 14.4 psia, the reference pressure.

When used to approximate overall bhp for a multistage unit, the calculated results will be high due to the reduction in horsepower obtained from interstage cooling. For best results with these curves, evaluate total compression horsepower as a sum of the individual stages (see Figure 12-17A). The effect of compressibility has been omitted from these curves; however, if it is known to be appreciable, use the more exact calculation methods listed previously. An approximation can be made by multiplying the bhp from the curves by the Z factor at the actual stage inlet conditions. If Z is less than 1.0, it should be neglected for this approximation, and only values greater than 1.0 should be considered.

B. Multistage.

Multistage horsepower is the sum of the horsepower requirements of the individual cylinders on the compressor unit.

Actual bhp:

$$= 0.004364\, F_L(k/k - 1)\{P_1V_1[(P_{i1}/P_1)^{(k-1)/k} - 1]L_{o1} + P_{i1}V_{i1}[(P_{i2}/P_{i1})^{(k-1)/k} - 1]Lo_2 + \ldots P_{ii}V_{ii}\,[(P_f/P_{i1}(^{(k-1)/k} - 1]Lo_f\} \quad (12\text{-}60)$$

where P_i = interstage pressure, psia

P_f = final, or last-stage discharge pressure, psia

1, 2, . . . i = successive interstage designations

$Lo_1, Lo_2 \ldots Lo_f$ = loss factors designated by cylinder stages

V_1 = intake volume including effect of compressibility when applicable

Total bhp/stage and per multistage compressor assembly may be approximated using Figure 12-21.

Corrections for compressibility (Z) may be incorporated as described for the single-stage cylinder, handling this on a per-cylinder basis.

C. Bhp Actually Consumed by Cylinders.

This horsepower is convenient to calculate when a known cylinder(s) exists on a compressor and when its performance is to be studied.

$$\text{bhp} = [(PD)(E_v)](P_1)(\text{bhp}/\text{MMCFD})(10^{-4}) \quad (12\text{-}61)$$

Actual capacity handled = $(PD)(E_v)(100)$, cfm, measured at suction conditions to the cylinder

where $(PD)(E_v)$ = ft^3 per min (cfm) a cylinder must handle, measured at suction pressure and temperature to the cylinder.

(PD) = compressor cylinder piston displacement in ft^3/min(cfm). These values can be calculated from known cylinder data or obtained from the respective compressor manufacturer for the specific cylinder in question, operating at the designated rpm.

E_v = Ev = volumetric efficiency from Figure 12-18A–F.

Note that actual capacity at 14.4 psia and suction temperature = $(PD)(E_v)(P_1)(100)$. Therefore, the bhp value given previously is in the correct units for the curves of Figure 12-21.

12. Temperature Rise—Adiabatic

The relation between the suction and discharge temperatures of a gas during any single compression step is

$$T_2 = T_1(P_2/P_1)^{(k-1)/k} = T_1R_c^{(k-1)/k} \quad (12\text{-}62)$$

where T_1 = initial suction temperature to cylinder, °Rankine = (460 + °F)

T_2 = discharge temperature from cylinder, °Rankine

R_c = ratio of cylinder compression

Figure 12-22 presents a convenient solution to this relation. The value of $R_c^{(k-1)/k}$ read from the chart times the absolute suction temperature gives the discharge temperature T_2. Thus,

$T_2 = T_1$ (value from graph, Figure 12-22)

12A. Temperature Rise—Polytropic

Note that for reciprocating compressor work, values of "n" may be used as "k" up to 1.4. "n" represents the polytropic coefficient that is related to "k" by $(n - 1)/n = (k - 1)/[(k)(e_p)]$, where (e_p) is the polytropic efficiency.

Cylinder temperature rise is an important consideration, not only at the exit or discharge of the gases from the cylinder, but often for temperature-sensitive gas/mixtures. The temperature calculated or projected inside the cylinder as compression proceeds must be calculated or measured to guard against too high a temperature developing during the compression process. Some gases such as chlorine, fluorine, bromine acetylene (and acetylene compounds), ethylene, and others must be carefully evaluated. If temperature rise is too high, conditions can lead to internal fires (actually consuming the metal of the compressor cylinder/liner) and explosions. Also for some gases/mixtures, polymerization can occur in the cylinder and, thereby, change the entire performance of the unit, as the cylinders and valves are not designed to handle polymers that are no longer gases.

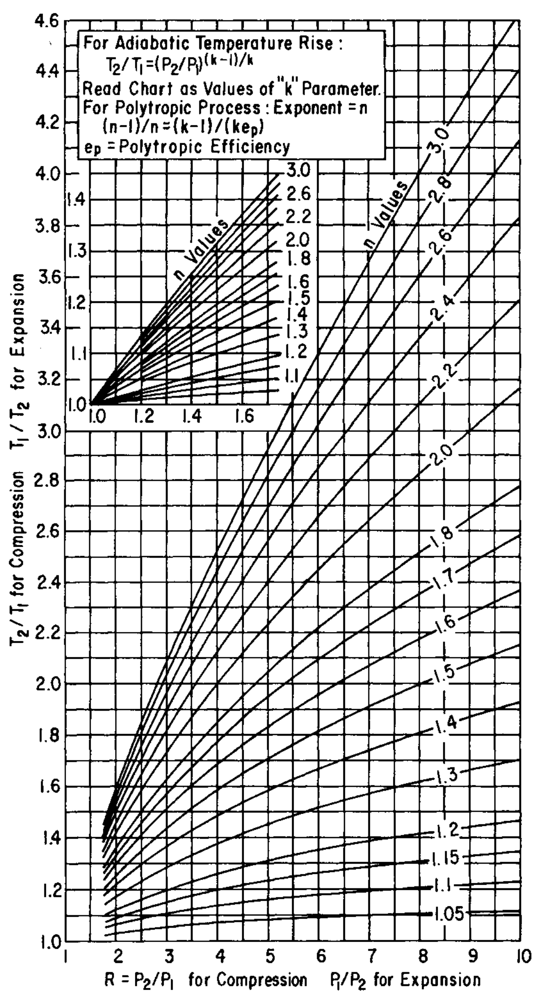


Figure 12-22. Compression temperature rise. (Used by permission: Rice, W. T. *Chemical Engineering,* April 1950. ©McGraw-Hill, Inc. All rights reserved.)

13. Altitude Conversion

Because all compressors do not operate at sea level pressure conditions, it is important to use the proper absolute pressure at the particular locality. Figure 12-23 is useful for converting altitude to pressure (or see Appendix A-6).

Example 12-2. Single-Stage Compression

A compressor is to be installed at a location 2,000 ft above sea level. It will handle a gas mixture with "k" = 1.25 at 5 psig suctions and discharge at 50 psig. Suction temperature is 90° F. The gas capacity is to be 5,250,000 SCFD measured at 14.7 psia and 60°F.

Determine horsepower requirements and discharge temperature.

1. Calculate the altitude conversion, Figure 12-23, or see Appendix A-6.

Atmospheric pressure at 2,000 ft = 13.68 psia

2. Ratio of compression,

$$R_c = \frac{50 + 13.68}{5 + 13.68} = 3.41$$

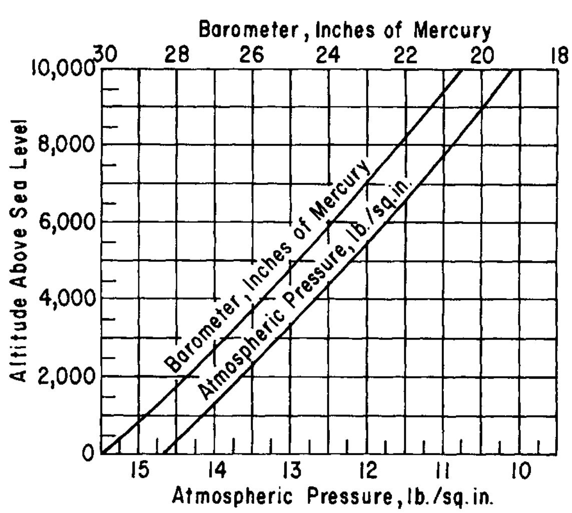


Figure 12-23. Barometric and atmospheric pressure at altitudes. See the appendix for detailed tabular listing.

This is satisfactory for single-stage operation if temperature does not limit.

3. Discharge temperature (adiabatic rise),

$$T_2 = T_1 R_c^{(k-1)/k} = (90 + 460)(3.41)^{(1.25-1)/1.25}$$
$$T_2 = (550)(1.278) = 702°R$$
$$T_2 = 702 - 460 = 242°F.$$

This temperature is safe.

4. Horsepower,

bhp/MMCFD = 74.1 (at $R_c = 3.41$, k = 1.25, from Figure 12-21B)
Note that these curves are for 14.4 psia and suction temperature; therefore, to this basis:

$$\text{Capacity} = 5{,}250{,}000\left(\frac{14.7}{14.4}\right)\left(\frac{460 + 90}{460 + 60}\right)$$

= 5,670,000 CFD at 14.4 psia and 90°F

$$\text{BHP required} = (\text{bhp/MMCFD})\left(\frac{\text{Capacity}}{(10^6)}\right)$$

$$= (74.1)\left(\frac{5{,}670{,}000}{10^6}\right) = 420 \text{ bhp}$$

5. Compressor cylinders,

Cylinder or cylinders volume must provide

$$(PD)(E_v) = \frac{(\text{bhp}) \times 10^4}{(\text{bhp})/\text{MMCFD})(P_1)} \qquad (12\text{-}63)$$

If the suction to a stage does not exceed 10 psig, use:[65]

$$(PD)(E_v) = \frac{(\text{bhp}) \times 10^4}{(\text{bhp/MMCFD})(P_1 - 0.5)} \qquad (12\text{-}64)$$

Note: Use the compression ratio calculated by

$$R_{c2} = \frac{P_2}{(P_1 - 0.5)} \qquad (12\text{-}65)$$

This equation is to be used only for this step in the evaluation of volumetric efficiency and should not be used for any other factor in the performance evaluation. Due to valve and other losses being a significant portion in low suction pressure (less than 10 psig) cylinder performance, this can result in reduced performance if not corrected as noted.

Then, R_{c2} = corrected compression ratio

$$= \frac{50 \text{ psia} + 13.68 \text{ bar}}{(5 - 0.5) + 13.68}$$
$$= 3.5$$

Reading chart Figure 12-18C, E_v = 84.5 at assumed 7% cylinder clearance.

Required Piston displacement (PD)

$$= \frac{\text{bhp} \times 10^4}{(0.845)(\text{bhp/MMCFD})(18.68 - 0.5)}$$

$$\text{Required: } (E_v)(PD) = \frac{(420)(10^4)}{(74.1)(18.18)} = 3{,}117 \text{ cfm (or, cfm)}$$

Because the manufacturer is the only firm qualified to design the actual cylinder, no further details will be presented here.

Note that this is horsepower available to the cylinders. It includes a 95% mechanical efficiency incorporated into the bhp/MMCFD curves. The rating of a gas engine must be such that its delivered horsepower is at least 420 hp, or if an electric motor drive is used, the mechanical losses of the intermediate frame (about 5%) must be added to arrive at the required motor shaft horsepower.

Example 12-3. Two-Stage Compression

A natural gas compressor is required to handle 4 million SCFD (measured at 14.7 psia and 60°F) from a suction condition of 0 psig and 70°F to a discharge of 140 psig. The altitude at the location is 3,000 ft. The cooling water for any interstage cooling is at 80°F for 95% of the peak summer months. Determine the horsepower requirements allowing 5 psi pressure drop through the intercooler.

1. Compression ratio

At 3,000 ft, atmospheric pressure = 13.14 psia (Figure 12-23)

$$P_1 = 0 + 13.14 = 13.14 \text{ psia}$$
$$P_f = 140 + 13.14 = 153.14 \text{ psia}$$
$$R_c = \frac{153.14}{13.14} = 11.66$$

This indicates a two-stage compression because R_c is greater than 5 or 6. Under some designs and for some capacities (not this) it can be satisfactorily handled in a single stage.

$$\text{Approximate } R_c \text{ per stage} = \sqrt{11.66} = 3.42$$

a. First stage: (allowing for one-half pressure drop handled by first stage)

$P_1 = 13.14$ psia

$$P_{i1} = (3.42)(13.14) + \frac{5\text{ psi}}{2} = 44.9 + 2.5 = 47.4$$

$R_c = 3.61$

b. Second stage:

$$P_{i1} = 44.9 - \frac{5\text{ psi}}{2} = 42.4$$

$P_{f2}' = 153.14$ $R_c = 3.61$

2. Discharge temperature first stage

$T_{i1} = T_1 R_c^{(k-1)/k}$
"k" for natural gas = 1.62
$T_{i1} = (70 + 460)(3.61)^{(1.26-1)/1.26} = (530)(1.305)$
$T_{i1} = 691°R$
$T_{i1} = 691° - 460° = 231°F$

Showing the use of Figure 12-22,

$R_c = 3.61$, $k = 1.26$
Read $T_2/T_1 = 1.30$
Then, $T_2 = (1.30)(530) = 689°R$
$T_2 = 689 - 460 = 229°F$

This is usually as close as needed.

3. Discharge temperature second stage
Because the cooling water temperature is low enough to allow good cooling, cool the gas to 95°F. This will be the suction temperature to the second-stage cylinder.

$T_{f2} = T_{i1}' R_c^{(k-1)/k} = (95 + 460)(3.61)^{(1.26-1)/1.26}$
$T_{f2} = (555)(1,305) = 725°R$
$T_{f2} = 265°F$

4. Horsepower
First stage from Figure 12-21B.

Bhp/MMCFD = 78.0 at $R_c = 3.61$ and $k = 1.26$
(Reference to 14.4 psia and suction temperature 70°F.)
Suction volume @ 14.4 psia and 70° F,

$$= 4{,}000{,}000\left(\frac{14.7}{14.4}\right)\left(\frac{460 + 70}{460 + 60}\right) = 4{,}162{,}000 \text{ CFD}$$

Using Figure 12-21B,

$$\text{bhp} = \text{bhp/MMCFD}\left(\frac{\text{suction volume capacity}}{10^6}\right)$$

$$\text{bhp} = (78.0)(4{,}162{,}000/10^6) = 324.6 \text{ horsepower}$$

Second stage,

bhp/MMCFD = 78.0 at $R_c = 3.61$ and $k = 1.26$
(Refer to 14.4 psia and 95°F.)

Suction volume @ 14.4 psia and 95°F,

$$= 4{,}000{,}000\left(\frac{14.7}{14.4}\right)\left(\frac{460 + 95}{460 + 60}\right) = 4{,}358{,}000 \text{ CFD}$$

bhp = $(78.0)(4{,}358{,}000/10^6) = 339.9$ horsepower
Total bhp = $324.6 + 339.9 = 664.5$ hp

This is the horsepower consumed by the cylinders and does not contain any losses in transmitting the power from the driver to the point of use, such as belts or gears. It does contain 95% mechanical efficiency for the cylinder itself.

5. Cylinder Selection
The general steps in cylinder selection will be outlined. However, actual selection can be accomplished only by referring to a specific manufacturer's piston displacement and the volumetric efficiency of a cylinder. The volumetric efficiency is a function of the compression ratio and "k" value of gas (both independent of cylinder) and the % clearance, a function of cylinder design.

$$\text{a. } [(Pd)(E_v)] = \frac{\text{bhp}(10^4)}{(\text{bhp/MMCFD})(P_1 - 0.5)^*}$$

*Use 0.5 *only* when suction pressure less than 10 psig, and the R_c used for E_v selection must be corrected accordingly.[44]

For a solution, use 325 bhp for first stage. However, it is quite likely that either a 660 hp (overloaded) or a 750 hp driver may be available as "standard."
The available hp for the first stage is based on 750 hp.

$$\text{First stage} = \left(\frac{325}{666}\right)(750) = 366 \text{ hp available}$$

$$\text{Second stage} = \left(\frac{340}{666}\right)(750) = \underline{382} \text{ hp available}$$

$$\text{Total} = 748$$

Using these, the first-stage cylinder capacity is

$$\text{Required } [(PD)(E_v)] = \frac{(366)(10^4)}{(78.0)(13.14 - 0.5)} = 3{,}700 \text{ cfm}$$

Usually this would be handled in two parallel cylinders.

$$\text{Each cylinder, } [(PD)(E_v)] = \frac{3{,}700}{2} = 1{,}850 \text{ cfm}$$

Next select a type or class, diameter and PD of a cylinder that will meet the required volume and pressure conditions. This must be done with the manufacturers' tables.

No. Cyl	Diam.	Class or Type	% Clearance	PD	E_v	[(PD) (E_v)]
2	*	*	*	*	*	**

*Manufacturer table values
**Calculated based on Cylinder

The calculated [(PD)(E_v)] should be equal to or greater than the required value of 1,850 cfm (in this example).
Second stage:

$$\text{Required } [(PD)(E_v)] = \frac{(384)(10^4)}{(78.0)(42.2)} = 1{,}160 \text{ cfm}$$

For this stage also select a cylinder and check that its [(PD)(E_v)] is equal to or greater than the 1,160 cfm.

b. If the actual [(PD)(E_v)] is larger than that required, the actual horsepower loading with the cylinders selected must be calculated to be certain that the total cylinder load does not exceed the allowable horsepower operating rating of the driver.

Solution of Compression Problems Using Mollier Diagrams

See Figures 12-24A–H.

The solution of the work compression part of the compressor selection problem is quite accurate and easy when a pressure-enthalpy or Mollier diagram of the gas is available (see Figures 12-24A–H). These charts present the actual relationship of the gas properties under all conditions of the diagram and recognize the deviation from the ideal gas laws. In the range in which compressibility of the gas becomes significant, the use of the charts is most helpful and convenient. Because this information is not available for many gas mixtures, it is limited to those rather common or perhaps extremely important gases (or mixtures) where this information has been prepared in chart form. The procedure is as follows:

Horsepower

$$\text{Work} = h_2 - h_1 \qquad (12\text{-}66)$$

h = enthalpy of gas, Btu/lb
1, 2 = states or conditions of system; 1 = suction, 2 = discharge.

Brake Horsepower, Required

$$\text{bhp} = \frac{778}{33{,}000}(M)(h_2 - h_1)(L_o)(F_L) \qquad (12\text{-}67)$$

where M = gas flow rate, lb/min
L_o = loss factor, Figure 12-19
F_L = frame loss for motor-drive unit only = 1.05, omit if an integral unit as for gas or steam engine drive.

Figure 12-25 represents combined compression and mechanical efficiency of a compression unit. Therefore, for approximation

$$\text{bhp} = (778/33{,}000)(M)(h_2 - h_1)/e_{cm} \qquad (12\text{-}68)$$

Take e_{cm} from Figure 12-25

$$\text{bhp/MMCF/day} = \frac{\text{ihp}}{0.95} \qquad (12\text{-}69)$$

0.95 = average overall compressor mechanical efficiency

Cylinder sizes are determined in the same manner as for the example on two-stage compression. The cfm or [(PD)(E_v)] at suction conditions is determined and sizing continued.

The volumetric efficiency may be expressed:[44]

$$E_v = 100 - R_c - \%Cl\left[\frac{v_s}{v_a} - 1\right] \qquad (12\text{-}70)$$

where v_s = specific volume at suction conditions, ft³/lb
v_a = specific volume at discharge conditions, ft³/lb
%Cl = percent clearance = V_{pc}

Compressor Indicated Horsepower, ihp

$$\text{ihp/MMOF/day} = 0.0432(h_2 - h_1)/e_{ar} \qquad (12\text{-}71)$$

$$\text{ihp} = (\text{MEP})(S)(A_p)(\text{rpm})/33{,}000 \qquad (12\text{-}72)$$

where MEP = mean effective pressure during compression stroke, from indicator card, psi.

$$\text{MEP} = 14.73E_v'\left(\frac{k}{k-1}\right)[R_c^{(k-1)/k} - 1], \text{ Ref. (24)} \qquad (12\text{-}73)$$

E_v' = use Equation 12-54
e_{ar} = compression efficiency, the product of adiabatic and reversible efficiencies, which vary with the cylinder and valve design, piston speed, and fraction; values range from 0.70–0.88 usually.
A_p = cross-sectional area of cylinder, in.²; for double-acting cylinder, use A_p as $(2A_p - A_r)$
S = stroke, ft
MMCF = 1,000,000 ft³ gas at 14.7 psia and 60°F
rpm = revolutions per minute of compressor

Example 12-4. Horsepower Calculation Using Mollier Diagram

An ammonia compressor is required to handle 25,000 lb per hour of gas at a suction condition of 105 psia and 70°F and is to discharge at 250 psia.

1. Ratio of compression,

$R_c = 250/105 = 2.38$

This should be a single-stage compression.

(Text continues on page 442)

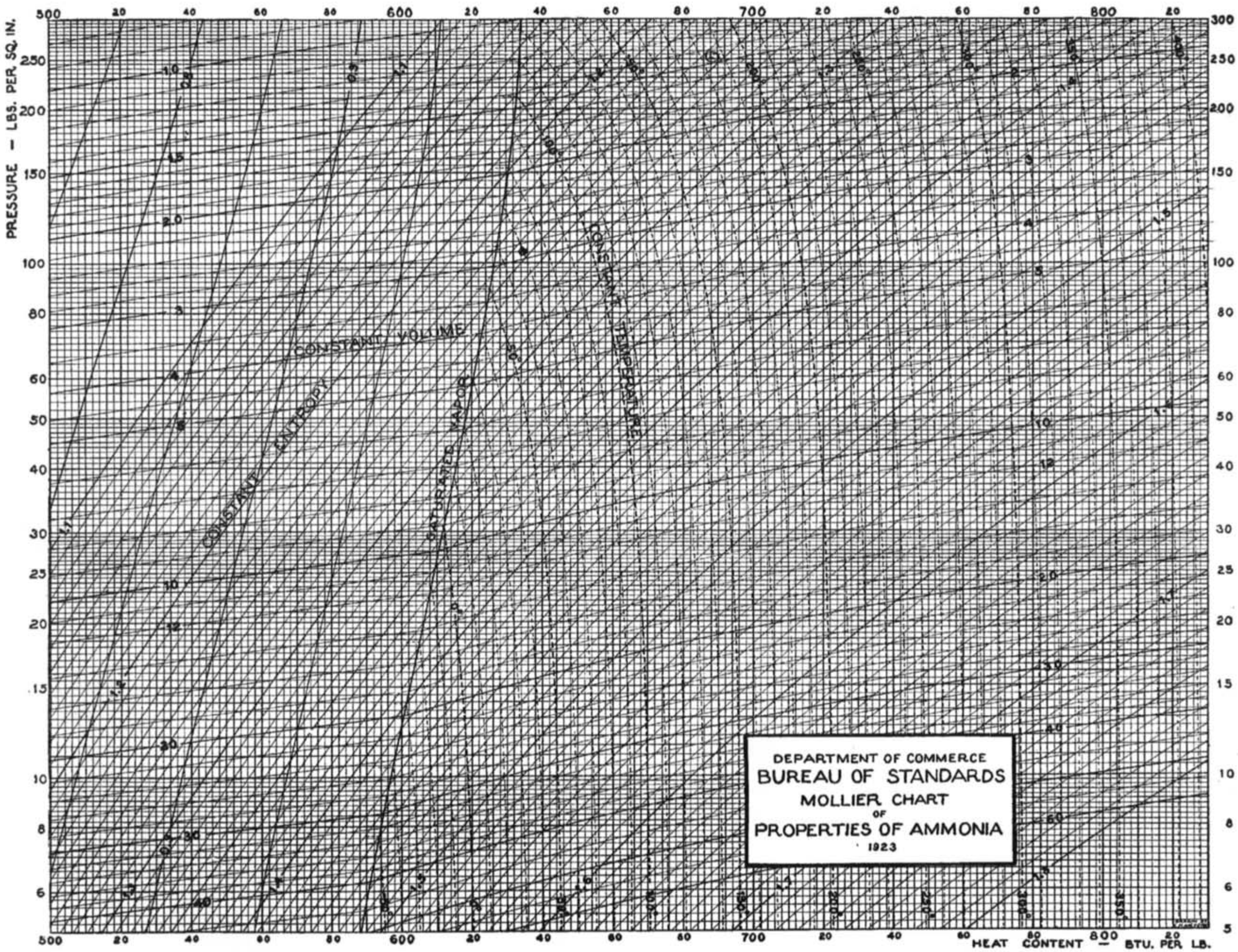


Figure 12-24A. Mollier chart for properties of ammonia. (Used by permission: Dept. of Commerce, U.S. Bureau of Standards.)

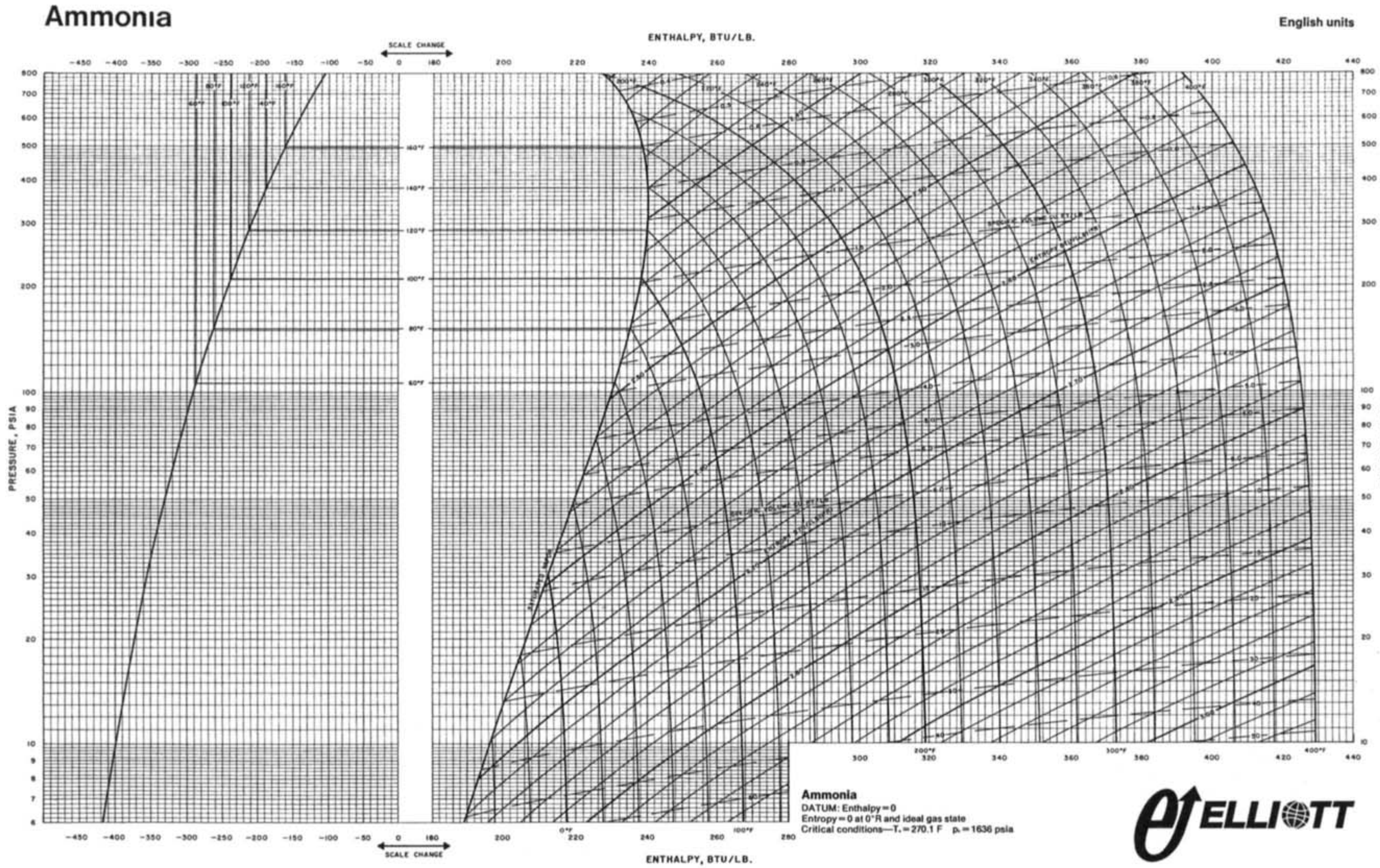


Figure 12-24B. Mollier diagram of properties of ammonia. Note the different construction from Figure 12-24A. (Used by permission: Elliot® Company. All rights reserved.)

Methane

English units

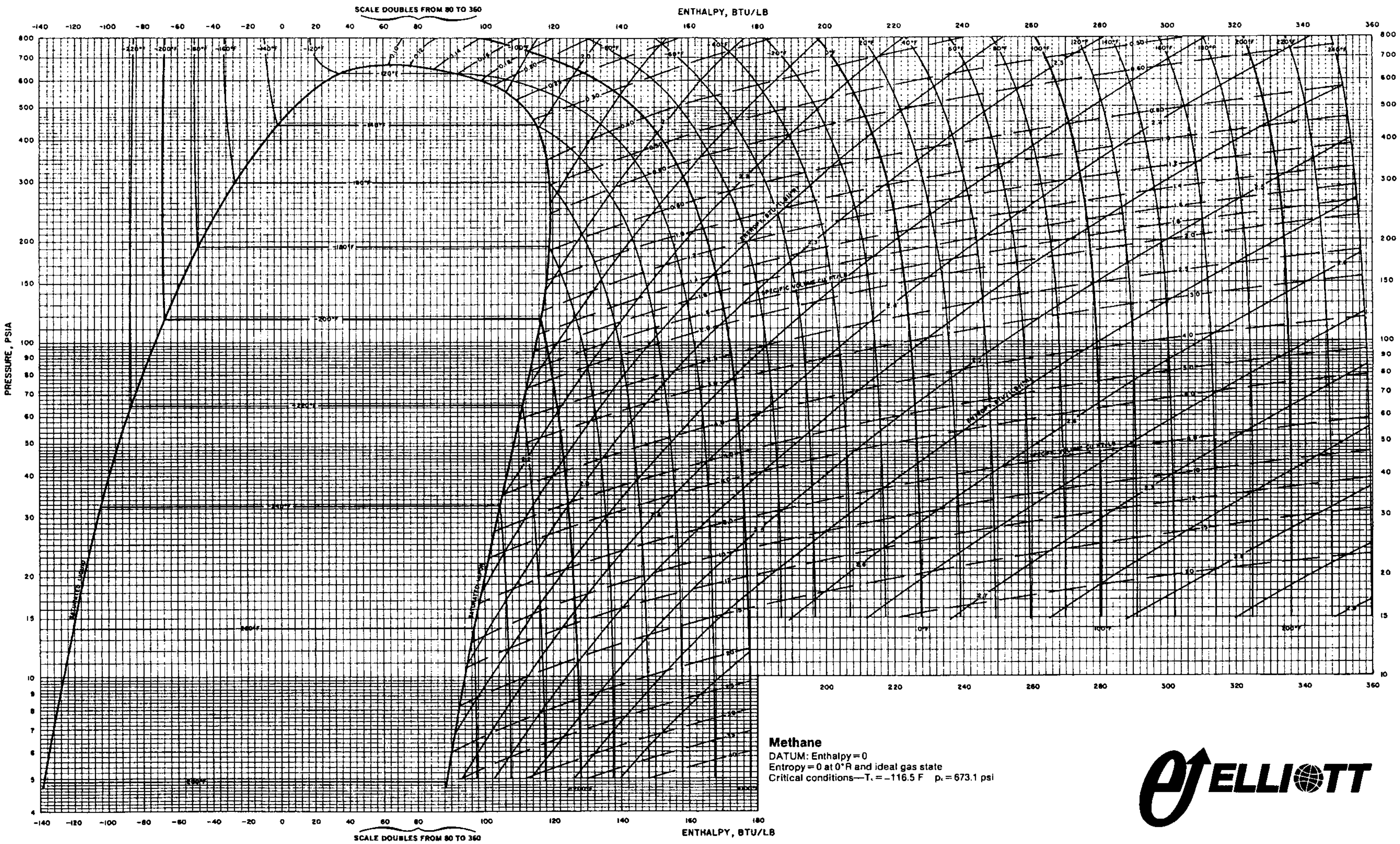


Figure 12-24C. Mollier diagram of properties of methane. (Used by permission: Elliott® Company. All rights reserved.)

Ethane

English units

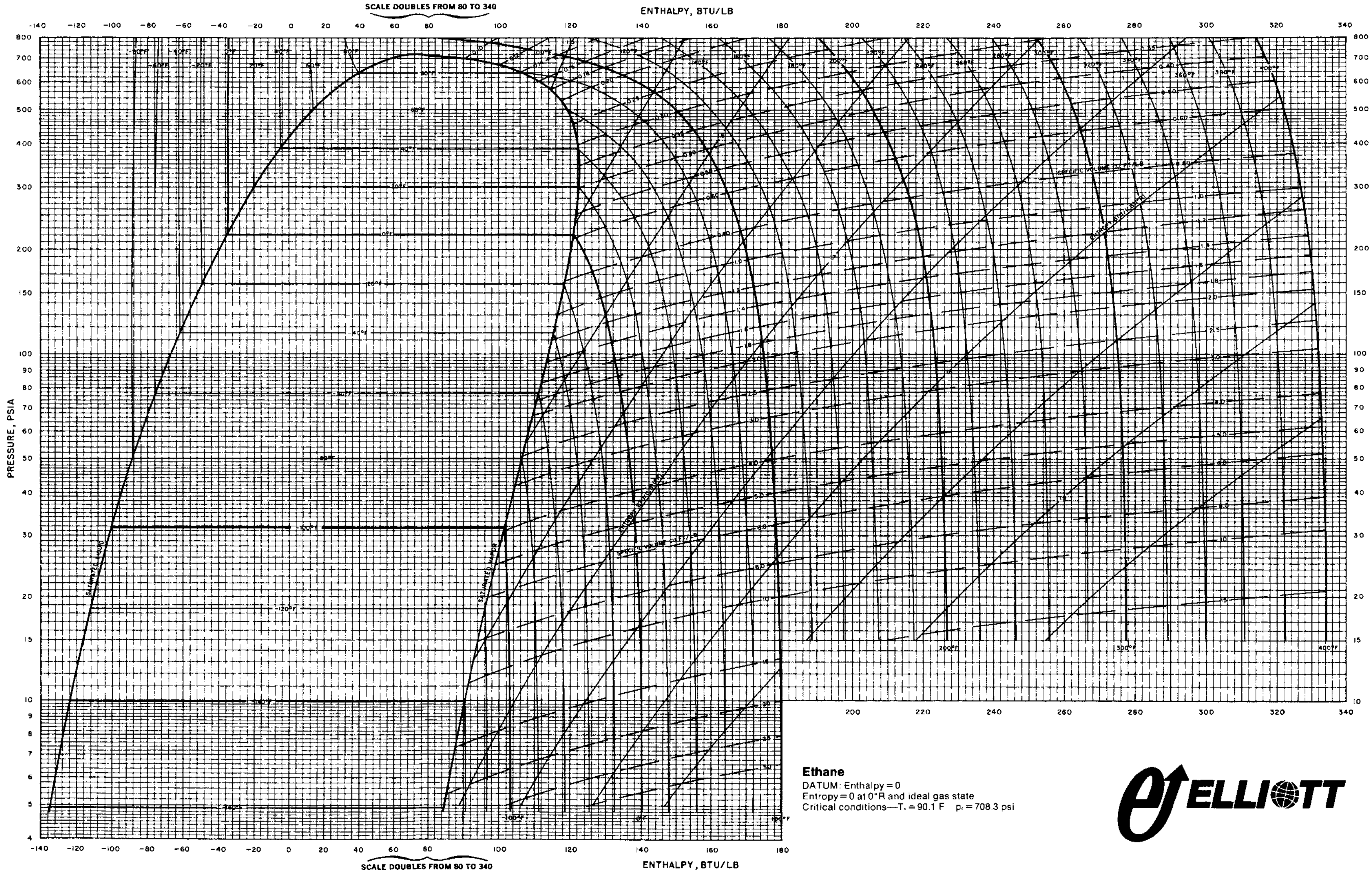


Figure 12-24D. Mollier diagram of properties of ethane. (Used by permission: Elliott® Company. All rights reserved.)

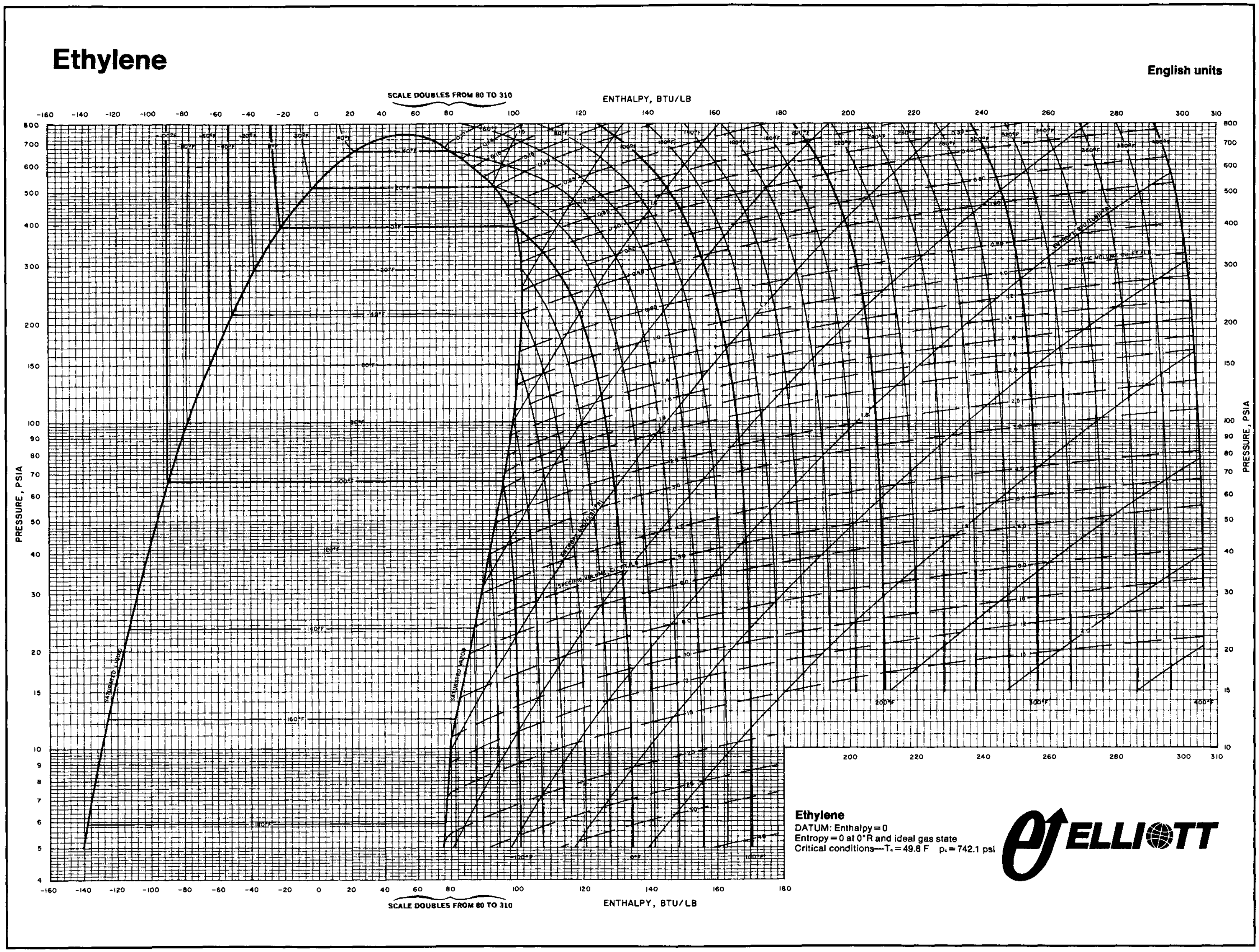


Figure 12-24E. Mollier diagram of properties of ethylene. Note: For ethylene chart of 60,000–100,000 psi, refer to a special chart that is available from some reciprocating compressor manufacturers. (Used by permission: Elliott® Company. All rights reserved.)

Propylene

English units

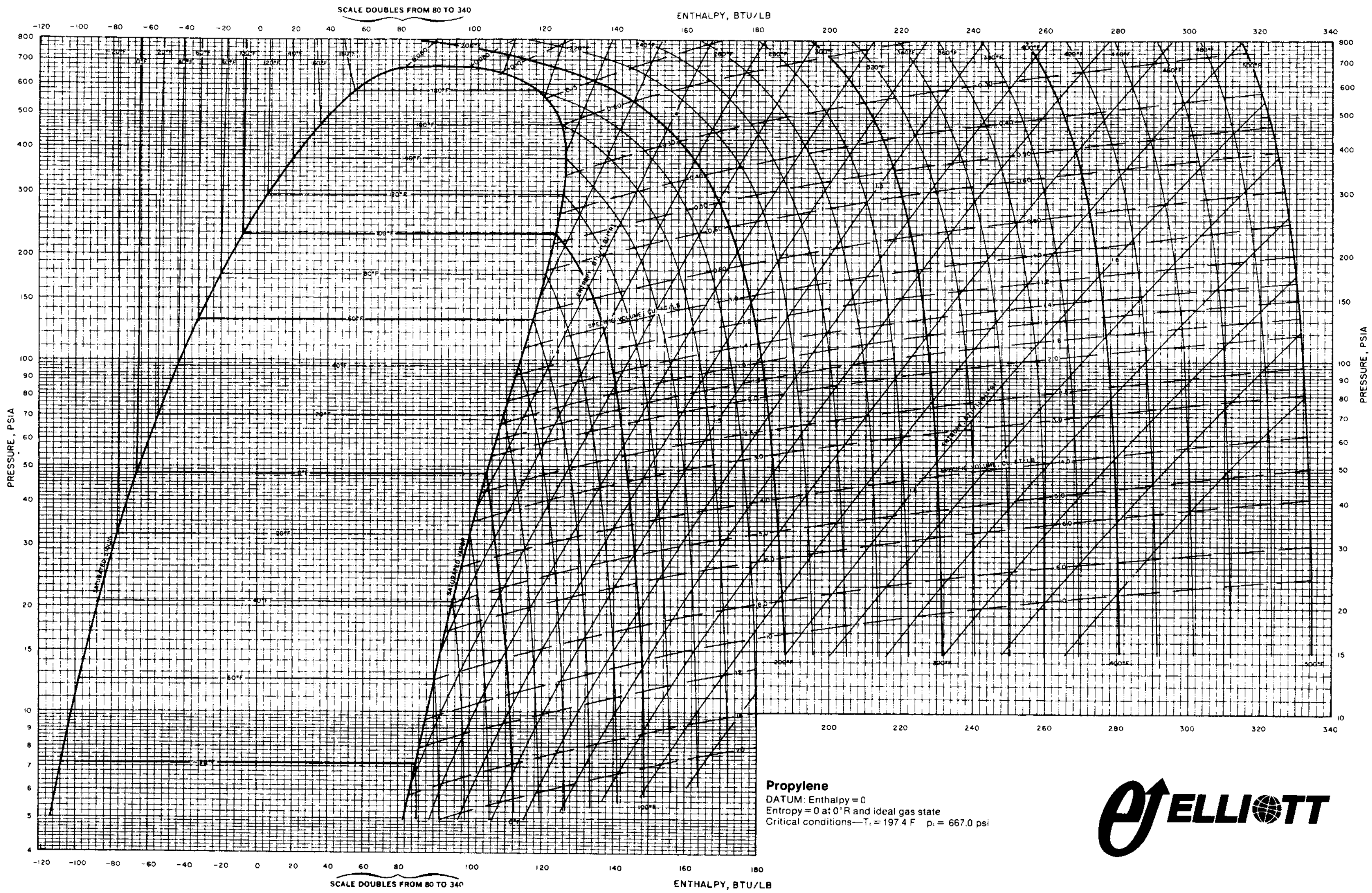


Figure 12-24F. Mollier diagram of properties of propylene. (Used by permission: Elliott® Company. All rights reserved.)

Propane

English units

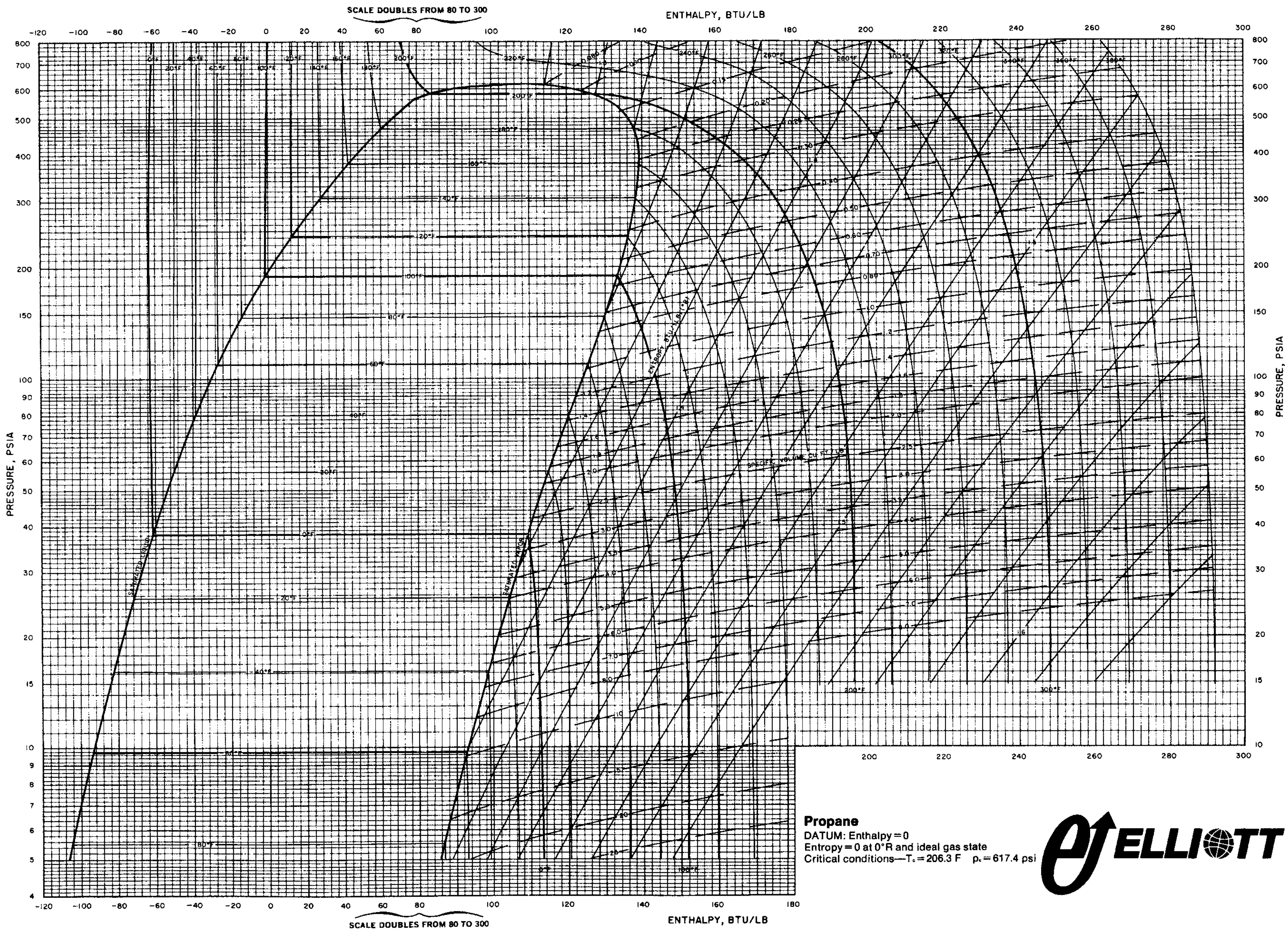


Figure 12-24G. Mollier diagram of properties of propane. (Used by permission: Elliott® Company. All rights reserved.)

Iso-butane

English units

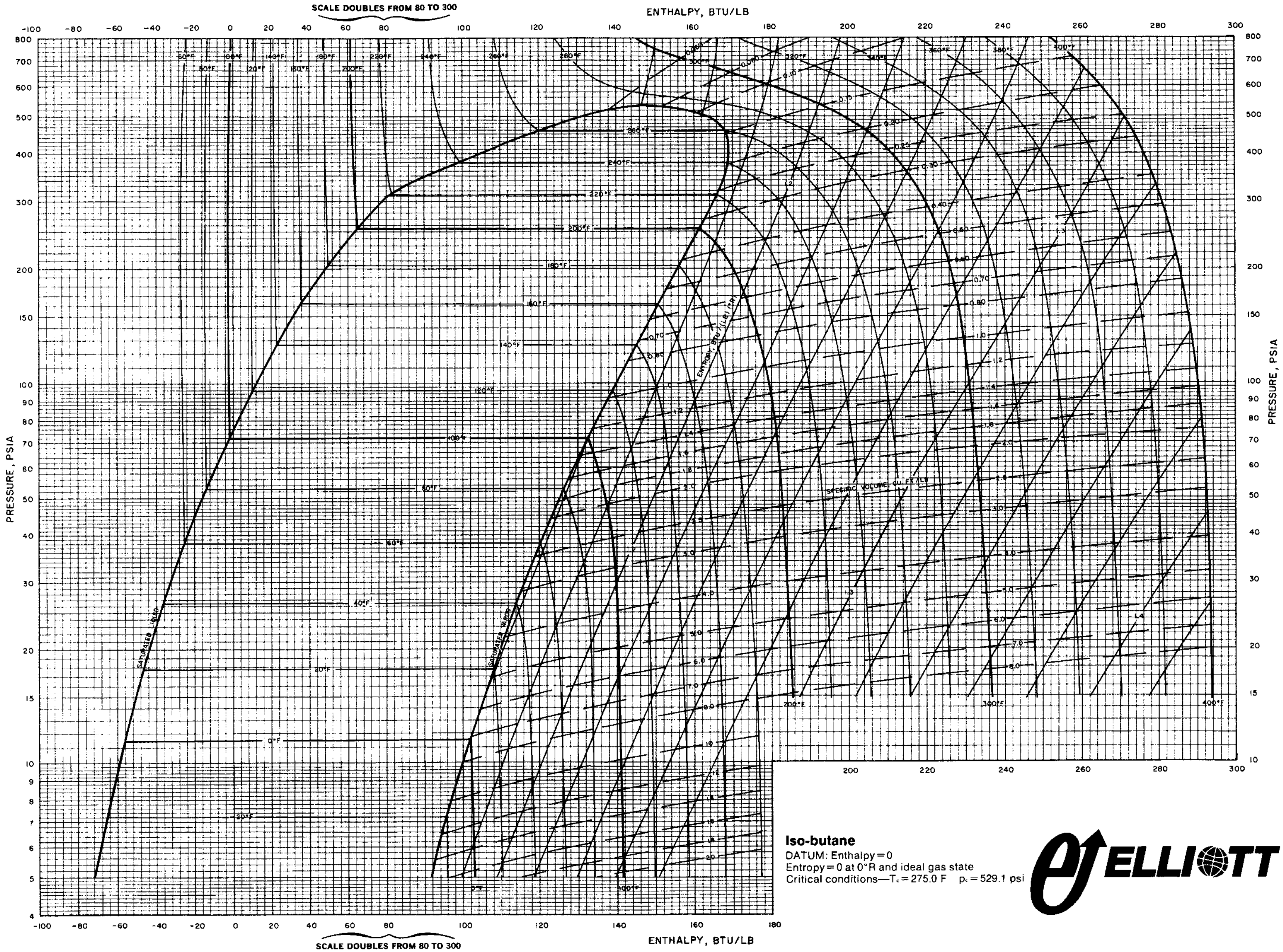


Figure 12-24H. Mollier diagram of properties of iso-butane. (Used by permission: Elliott® Company. All rights reserved.)

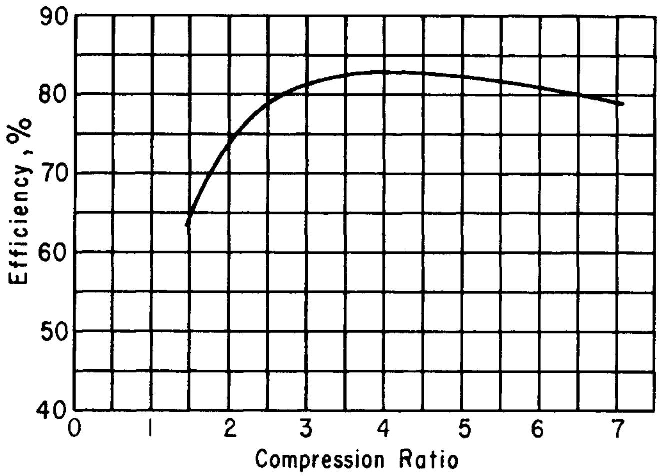


Figure 12-25. Combined compression and mechanical efficiency of reciprocating compressors. (Used by permission: Campbell, J. M. *Oil and Gas Journal;* and Ridgway, R. S. California Natural Gasoline Association Meeting, ©1945. All rights reserved.)

(Text continued from page 433)

2. Ammonia diagram, Figure 12-24A,
   1. Locate the *suction* condition at 105 psia and 70°F.
   2. Read h_1 = 635 Btu/lb.
   3. Read v_1 = 2.9 ft^3/lb.
   4. Follow the constant entropy (isentropic compression) line from the suction point until it intersects the *discharge* pressure line at 250 psia.
   5. Here, read T_2 = 183°F.
   6. Read h_2 = 688 Btu/lb.
   7. Read v_2 = 1.46 ft^3/lb.

3. Horsepower,

Capacity = 25,000 lb/hr = 417 lb/min
Loss factor, Figure 12-19
At R_c = 2.38
Read L_o = 1.275
Using Equation 12-67, omit F_L for an assumed gas engine drive.

$$\text{bhp} = \frac{778}{33{,}000}(417)(688 - 635)(1.275)$$

bhp = 663

4. Cylinder selection

   a. Required [(PD)(E_v)]

= lb/min)(v_1)
= (417)(2.9) = 1,208 cfm

A single cylinder will do this capacity; however, usually it can be handled in two parallel cylinders for better balance.

Then, per cylinder:

[(PD)(E_v)] = 1,208/2 = 604 cfm

   b. Volumetric efficiency

$$\%E_v = 100 - R - V_{pc}(v_1/v_2 - 1)$$
$$= 100 - 2.38 - V_{pc}(2.9/1.46 - 1)$$
$$= 100 - 2.38 - V_{pc}(0.982)$$
$$\%E_v = 97.62 - 0.982V_{pc}$$

The actual value depends on the cylinder chosen, in order to use the proper clearance fraction, V_{pc}.

   c. Select cylinders

From the manufacturer's specific compressor cylinder tables, select cylinders to give the required [(PD) (E_v)]; follow the two-stage compression example here. The final actual capacity depends upon this selection of cylinders.

Just obtaining these cylinders does not settle the design. The manufacturer must verify that no cylinder interferences exist and that the rod loading in tension and compression is satisfactory. This design detail is handled by the manufacturer. The final design agreement should be by the manufacturer, as he should be responsible for the final quoted performance of the unit.

Cylinder Unloading

This section is adapted from reference 44 by permission.

For each compressor unit with its associated individual cylinders, a fixed horsepower characteristic curve exists. The curve rises, peaks, and falls as the range of pressure ratio requirement varies. See Figure 12-17B.

Many compressors are designed and operated at a fixed condition in a process or refrigeration cycle. However, at least an equal number are designed and operated over a varying, or at the initial selection unknown, set of conditions of suction or discharge pressures. This situation is a reality and an economical necessity if the full horsepower of the compressor and driver combination is to be realized.

To understand this, the factors affecting the horsepower characteristic must be evaluated.

$$\text{bhp} = [(PD)(E_v)](P_1)(\text{bhp/MMCFD})/10^4 \quad (12\text{-}61)$$

The variable available for control is the volumetric efficiency, E_v, which is a function of the compression ratio of the process requirement and the % clearance of the cylinder. The % clearance can be varied in the cylinder for capacity control by

1. Head-end unloaders
2. Double-deck valves with valve cap unloaders

3. Adjusting (screwing) the piston rod further into or out of the cross-head for some single-acting units

Figures 12-26 and 12-27 show the relative characteristic horsepower curves for a gas of k = 1.3 when

1. Discharge pressure is constant and suction pressure varies.

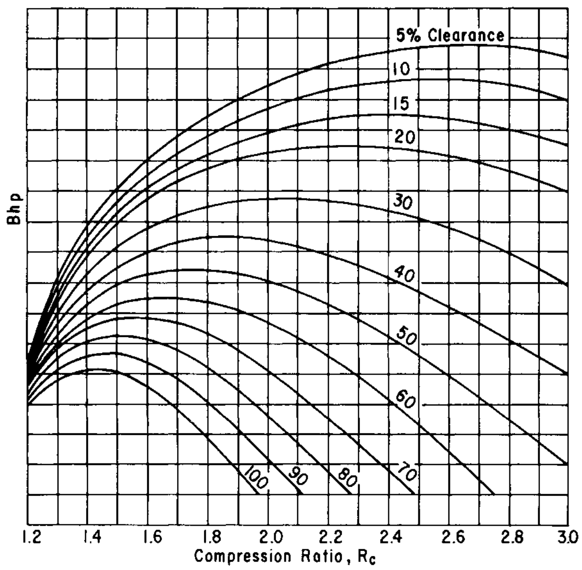


Figure 12-26. Horsepower characteristic curves for constant discharge pressure, k = 1.3. (Used by permission: Cooper-Cameron Corporation.)

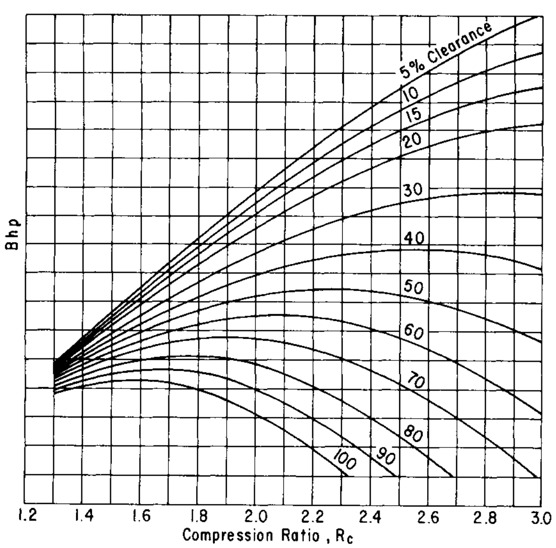


Figure 12-27. Horsepower characteristic curves for constant suction pressure, k = 1.3. (Used by permission: Cooper-Cameron Corporation.)

2. Suction pressure is constant and discharge pressure varies.

Each compressor unit and condition has its own specific horsepower point or requirement for operation. However, the general characteristic shape will be about the same, and for a reasonable range of conditions, the general shape and effect of varying a particular condition can be relatively established even for gases of other k values. Of course, the curves can be recalculated and drawn for the particular gas under consideration. The peaks will be in about the same ratio. Note that Figures 12-26 and 12-27 were established using a bhp/MMCFD correction factor at a mean pressure of 200 psia for the lower compression ratios where this correction is required.[44]

In sizing cylinders with several operating conditions, considering the use of these curves will allow the designer to select conditions that will nearly always keep the cylinder loaded to its peak. After the approximate (or actual) % clearance for the new or existing cylinder is established, reference to the curves will usually indicate the effect of the compression ratio change—that is, whether the horsepower will decrease or increase for a specific change. The curves indicate ranges of % clearance where the horsepower change is small for rather wide changes in R_c. Many problems will fall in this fortunate situation, where a single clearance will satisfy all expected conditions and no cylinder unloading, or where a minimum of unloading will be required.

Unloading of cylinders becomes necessary when the operating conditions vary sufficiently to require changes in % clearance in order to keep the usual 3% overload and 5% underload horsepower condition on the cylinders. This is done by using unloading schemes that change the % clearance within the cylinder. Figure 12-28 illustrates unloading connections mounted on a cylinder. Figures 12-29A and 12-29B show two schemes for unloading double-acting cylinders, one scheme using clearance pockets in the cylinder and the other using pockets plus valve lifters. The limits of the operation will determine whether one or more unloaders are necessary for a particular cylinder.

Five-step unloading of constant speed compressors allows the compressing load to change and match the process demand of full, three-fourths, one-half, one-fourth, and no-load without changing process variables. Three-step unloading provides for full, one-half, and no-load operation of the compressor. No-load operation allows the machine to remain running but not pumping gas into the system. This is particularly useful for air service systems or refrigeration processes. For units using clearance pockets (Figure 12-29A), when one clearance valve is opened the volume of that pocket is added to the normal cylinder clearance volume. Depending upon the pocket volume, this may cut to one-half the amount of gas entering that end of the cylinder.

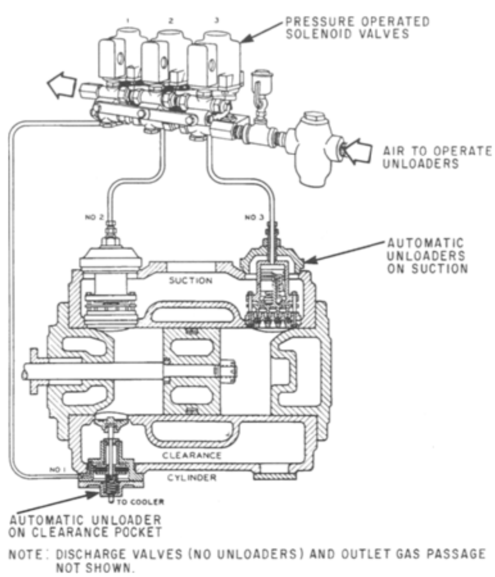


Figure 12-28. Automatic cylinder unloading. (Used by permission: Worthington Bul. L-679-BIA, ©1957. Dresser-Rand Company.)

Several pockets may be in each cylinder, depending upon volume needed and cylinder design. When all pockets (equal to cylinder volume) are open at one end of a cylinder, no gas enters. In the by-pass control scheme, Figure 12-29B, a pressure switch activates a solenoid valve when the system discharge pressure reaches a preset value. Activating air then causes the unloaders to open the suction valve(s), Figure 12-28, allowing suction pressure to pass freely in and out of the cylinder. No compression takes place. The unloaders may be manually operated, although automatic operation usually gives better control.

The clearance pockets may be of many different shapes and arrangements (see Figures 12-6B, 12-30A, and 12-30B). Fixed volume pockets allow for fixed or set volume changes while the variable volume designs allow for changes to suit a particular operating condition or balance and are of value when the cylinder must be used in several different alternating applications.

Capacity can be accomplished in several ways as illustrated in Figures 12-28–12-31A. In Figure 12-31(A) suction valve

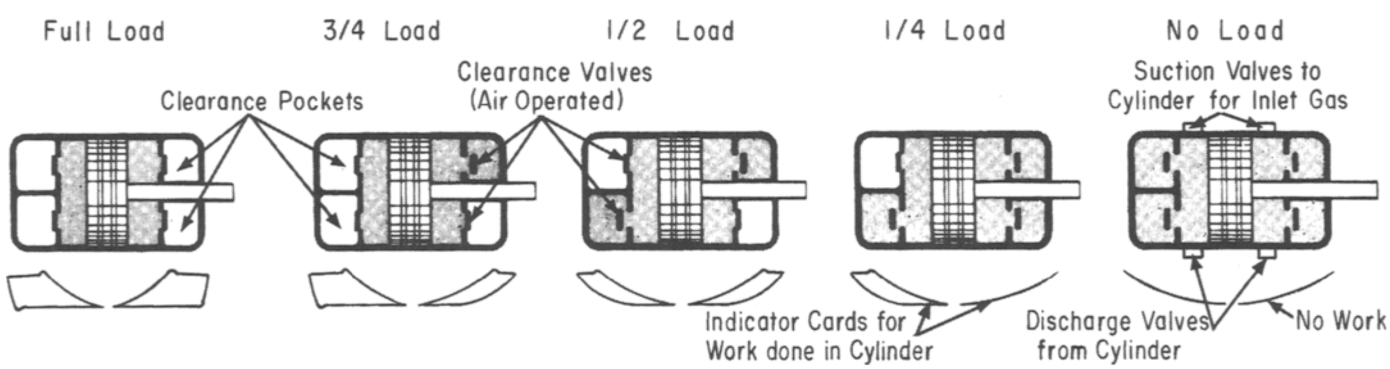


Figure 12-29A. Five-step clearance pocket control for compressor unloading. (Used and adapted by permission: Ingersoll-Rand Company. All rights reserved.)

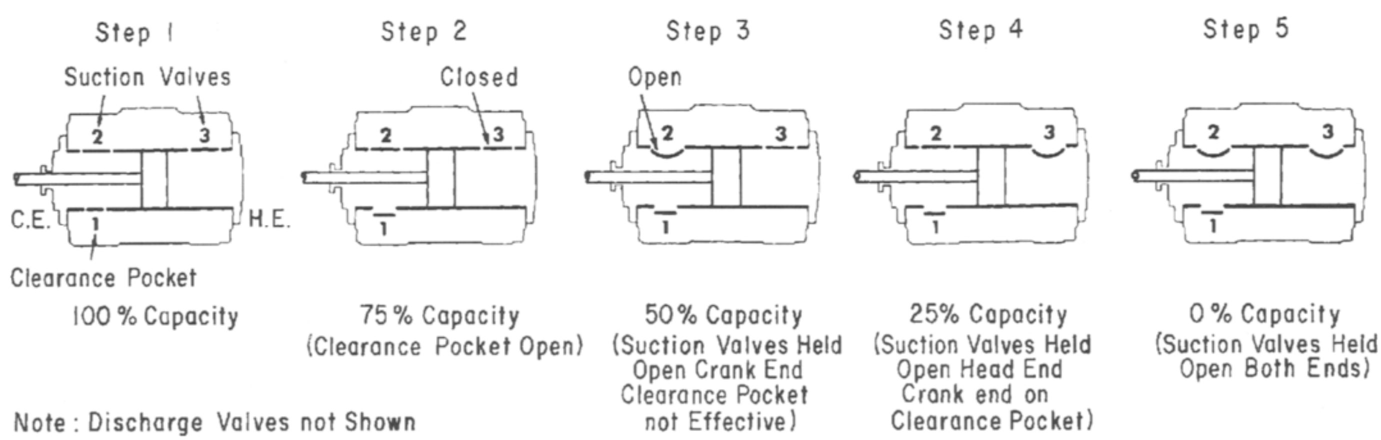


Figure 12-29B. Five-step control for compressor unloading. (Used by permission: Worthington Bul. L-679-BIA, ©1957. Dresser-Rand Company. All rights reserved.)

discs are depressed from their seats to allow the gas to flow freely in and out of the cylinder without compression.[66] Both unloader designs are of the external diaphragm type.

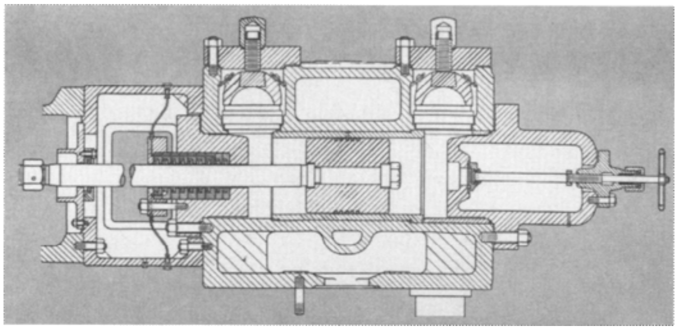


Figure 12-30A. Fixed volume clearance pockets. (Used by permission: Worthington Bul. L-679-BIA, ©1957. Dresser-Rand Co. All rights reserved.)

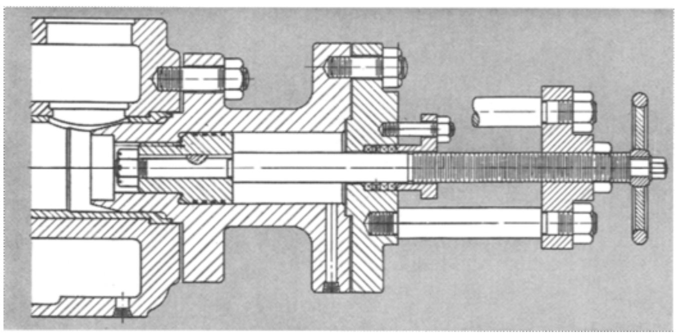


Figure 12-30B. Variable volume clearance pockets. (Used by permission: Worthington Bul. L-679-BIA, ©1957. Dresser-Rand Company. All rights reserved.)

Example 12-5. Compressor Unloading

This section is adapted from reference 44 by permission.

A compressor is required to handle 9,360,000 SCFD at a suction of 75 psig and 100°F with the discharge at 300 psig. It is anticipated that the suction pressure may rise to 100 psig after one year of operation. The k value of the gas is 1.3, and the unit will be installed at a coastal installation.

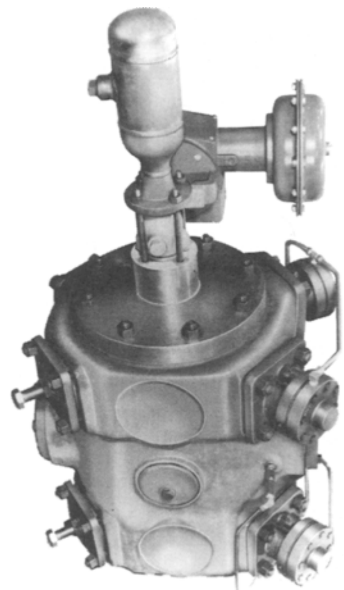


Figure 12-31A. Pneumatically operated clearance bottle. (Used by permission: Bul. 9-201B, ©1991. Cooper-Cameron Corporation. All rights reserved.)

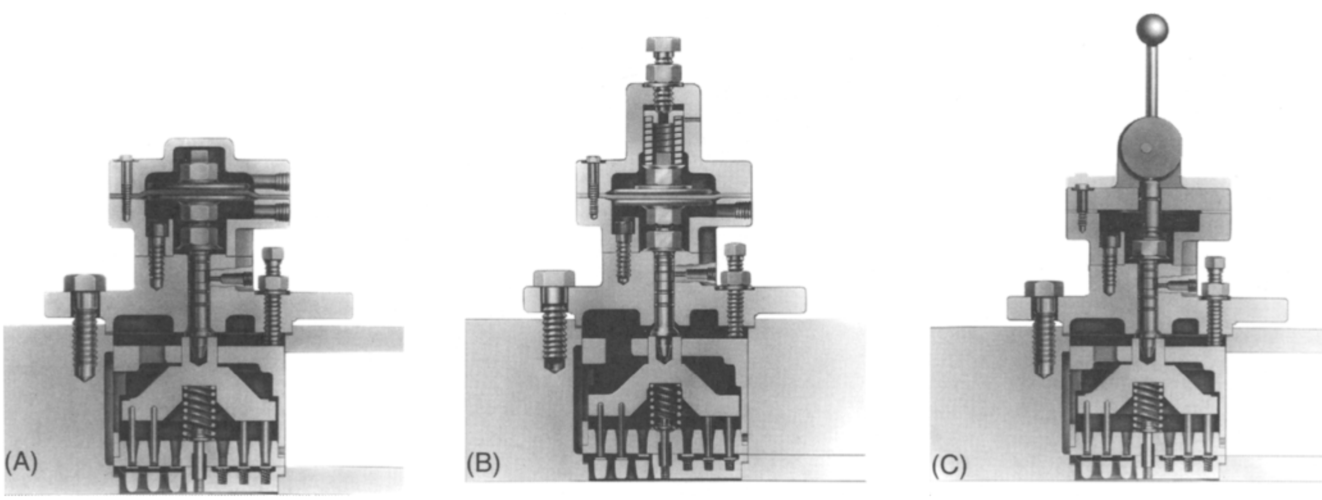


Figure 12-31. Capacity control: suction valve unloaders are available in either of two designs with pneumatic operators (A) direct-acting (air to unload); (B) reverse-acting or fail safe (air to load), which automatically unloads the compressor in the event of control air failure; (C) an innovation in manually operated unloaders. Here, the lever cam arrangement provides positive loading or unloading, eliminating the requirement to turn a handwheel completely in or out. (Used by permission: Bul. 9-201B, ©1991. Cooper-Cameron Corporation.)

Determine the proper unit for this operation.

1. Ratio of compression:

$$\text{Condition 1: } R_c = \frac{300 + 14.7}{75 + 14.7} = 3.51$$

$$\text{Condition 2: } R_c = \frac{300 + 14.7}{100 + 14.7} = 2.74$$

Figure 12-26 shows that at $R_{c1} = 3.51$, the horsepower conditions are lower than at $R_{c2} = 2.74$. Horsepower will have to be provided for R_{c2} condition, and must operate satisfactorily at R_{c1}. The 75 psig suction condition will determine the unloading.

2. Horsepower:

At $R_{c1} = 3.51$
bhp/MMCFD, Figure 12-21B = 77.3
Capacity at 14.4 psia and 100°F = 10,290,000 CFD, converted from given value at 14.7 psia and 60°F.

$$\text{Total bhp} = (77.3)\frac{10{,}290{,}000}{1{,}000{,}000} = 795$$

Use an 800 hp-rated unit.
At $R_{c2} = 2.74$, the required bhp is less than the preceding.

3. Compressor cylinders,

$$[(PD)(E_v)] = \frac{(800)(10^4)}{(77.3)(75 + 14.7)} = 1{,}153 \text{ CFM}$$

For two cylinders in parallel:
$[(PD)(E_v)]$ each = 576.5 cfm

Using Cooper-Cameron cylinder information for this example:

No. cylinders = 2
Diameter = 14 in.
% clearance = 11.1%
PD = 731
E_v (calc.) = 0.784, using Equation 12-45
$[(PD)(E_v)]$ = 573

4. Unloading
From the characteristic horsepower-% clearance curves, Figures 12-26 and 12-27, the maximum amount of unloading will be required when the suction pressure is at Condition 2, 100 psig. From the calculated horsepower for this point, the maximum amount of unloading required can be determined.

A performance curve, Figure 12-32 for this problem, will aid in determining the number of unloading steps.

By examining the curve for the initial compression with no unloaders, it shows that the horsepower requirement crosses the +3% overload line about one-third of the way through the suction pressure range. Figure 12-32 shows the effect of adding first one unloader and then a second one. The simplest way to handle this is a head-end unloader on each of the two parallel cylinders.

Actual size of unloaders:

Piston displacement (double acting cyl.) = 731 cfm each
Piston displacement head end = 374 cfm
Piston displacement crank end = 357 cfm
Clearance, (cylinder data) = 11.1%
Total clearance volume = (0.111)(731) = 81.2 cfm

Assuming equal distribution of clearance:

81.2/2 = 40.6 cfm

$$\text{Head end \% clearance} = \frac{40.6(100)}{374} = 10.85\%$$

$$\text{Crank end \% clearance} = \frac{40.6(100)}{357} = 11.36\%$$

Horsepower at suction pressure of 100 psig with no unloading:

bhp = $[(PD)(E_v)](P_1)(\text{bhp/MMCFD})(10^{-4})$
E_v for R_{c2} and 10.85 % clearance, calculated = 0.845
From Figure 12-21B, bhp/MMCFD for R_{C2} = 62
Head-end bhp = $[(374)(0.845)](114.7)(62)(10^{-4})$
H.E., bhp = 225 hp

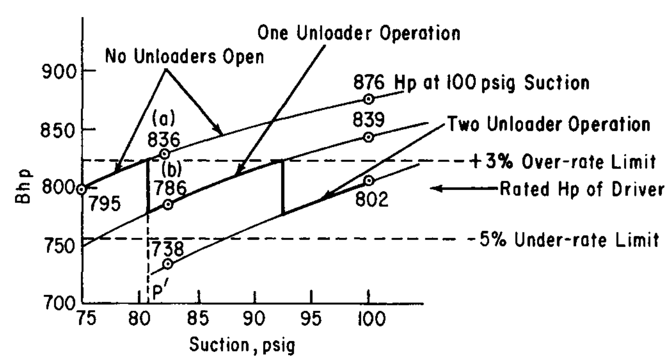


Figure 12-32. Compressor cylinder performance curve for unloading conditions. (Elaborated on by this author based on information used by permission of Cooper-Cameron Corporation. All rights reserved.)

Crank-end bhp = $[(357)(0.839)](114.7)(62)(10^{-4})$
$= 213$ hp
Total bhp/cylinder = 225 + 213 = 438 hp
Total bhp for the two parallel cylinders:
= (2)(438) = 876 hp
bhp available = 800
Excess bhp = 76 hp for cylinders

Using a head-end unloader on each cylinder, the head-end horsepower should be reduced:

76/2 = 38 hp
bhp of one head end = 225
less 38
187 bhp

If each head-end is unloaded to the point of requiring only the 187 hp, the unit will not be overloaded at the maximum point. The head-end E_v would have to be

$$E_v = \frac{187(10^4)}{(374)(62.0)(114.7)} = 0.703$$

Substitute in the E_v relation Equation 12-45,

$70.3 = 100 - 2.75 - \%Cl.(2.75^{1/1.3} - 1)$
$70.3 = 97.3 - \%Cl.(1.18)$
$\%Cl. = 27/1.18 = 22.1\%$, required
Total head end % clearance required = 22.10
Normal fixed head end clearance = 10.85
Additional % clearance required in H.E. = 11.25

In.^{3} required in each head-end unloader valve:

$$\text{unloader volume} = \frac{[(0.1125)(374)](1728)}{300 \text{ rpm}} = 242 \text{ in}^3$$

The performance curve of the unit can be completed as follows:

a. Calculate horsepower at pressure P′ equal to value of the suction pressure where line "a" (the original no unloader curve) crosses the 3% overload. Note that curve "a" is not a straight line.
b. For the new suction condition, calculate the new horsepower (82.4 psig) with one unloader on only one of the cylinders open. At this point,

One cylinder: (no unloader open)

PD = 731 cfm
PD(HE) = 374 cfm
PD(CE) = 357 cfm
HE %Cl. = 40.6/374 = 10.85%
CE %Cl. = 11.36%

$$\text{HE, } E_v = 100 - \left(\frac{300 + 14.7}{82.4 + 14.7}\right) - 10.85\left(3.24^{\frac{1}{1.3}} - 1\right) = 81.1\%$$

CE, E_v = 81.1% (calculated)
HE, bhp = $(374)(.818)(97.1)(72.4)(10^{-4}) = 215$
CE, bhp = $(357)(.811)(97.1)(72.4)(10^{-4}) = 203$
Total = 215 + 203 = 418 bhp (point a, Figure 12-32)

Second cylinder: (one unloader open on head-end)

PD = same
HE %Cl. = 22.1%
CE %Cl. = 11.36% unchanged

$$\text{HE, } E_v = 100 - \left(\frac{300 + 14.7}{82.4 + 14.7}\right) - 22.1(3.24^{1/1.3} - 1)$$
$= 64\%$

CE, E_v = 80.0
HE, bhp = $374(.64)(97.1)(72.4)(10^{-4}) = 168$
CE, bhp = $357(.80)(97.1)(72.4)(10^{-4}) = 200$
Total = 168 + 200 = 368 bhp

At 82.4 psia suction,

Total load = 418 + 368 = 786 hp (point b, Figure 12-32)

Another point on the curve at 100 psig suction,
One cylinder no unloader, second cylinder with unloader:
One cylinder (no unloader open),

$$\text{HE,} E_v = 100 - \left(\frac{314.7}{100 + 14.7}\right) - 10.85(2.75^{1/1.3} - 1)$$

$= 84.7\%$
CE, E_v = 83.8%
HE, bhp = $(374)(.847)(114.7)(62)(10^{-4}) = 225$
CE, bhp = $(357)(.838)(114.7)(62)(10^{-4}) = 213$
Total bhp = 438 hp

Second cylinder (unloader open on head-end),

PD = same
HE: Reading curve for 70.8% E_v at R_c = 2.75, %Cl. = 22.1
CE: %Cl. = 11.36% unchanged

Reading curve:

HE, E_v = 70.8%
CE, E_v = 83.8%
HE, bhp = $(374)(.708)(114.7)(62)(10^{-4}) = 188$
CE, bhp = $(357)(.838)(114.7)(62)(10^{-4}) = 213$
Total = 401 hp
Total for both cylinders = 438 + 401 = 839 hp

Now determine the operating line for the condition of both head end unloaders open, at two suction conditions:

a. Suction = 82.4 psig
The head end % clearance will be 22.1% because this is the condition with the unloader open on each cylinder. Reading curve:

HE, E_v: @R_c = 3.24 and % Cl. = 22.1,
E_v = 64%
CE, E_v: @R_c = 3.24 and % Cl. = 11.36,
E_v = 80%
HE, bhp = (374)(0.64)(97.1)(72.4)(10^{-4}) = 168
CE, bhp = (357)(0.80)(97.1)(72.4)(10^{-4}) = 201
Total bhp = 369 per cylinder
For two cylinders alike, bhp=738 hp (Curve "c")

b. Suction = 100 psig

HE, E_v: @R_c = 2.75 %Cl. = 22.1,
E_v = 71.3%
CE, E_v: @R_c = 2.75, %Cl. = 11.36,
E_v = 83.8%
HE, bhp = (374)(.708)(114.7)(62)(10^{-4}) = 188
CE, bhp = (357)(.838)(114.7)(62)(10^{-4}) = 213
Total per cylinder = 401
Total for two cylinders = 802 hp (Curve "c")

Example 12-6. Effect of Compressibility at High Pressure

A mixture of 3,000 scfm, dry basis, (14.7 psia and 60°F), 60% methane and 40% nitrogen is to be compressed from 16 psig to 3500 psig. Suction temperature is 90°F. Intercoolers will use 85°F water cooling gas to 90°F, and the installation is essentially at sea level. The gas is saturated with water vapor. Five lb pressure drop is to be allowed for the interstage coolers.

This problem involves the compressibility of the gas and its moisture content. These will be taken into account in the following design.

1. Ratio of compression:

$$R_c = \frac{3500 + 14.7}{16 + 14.7} = 114.48$$

This must be broken down into stages:

For three-stage $R_c = \sqrt[3]{114.48} = 4.85$ uncorrected

For four-stage $R_c = \sqrt[4]{114.48} = 3.26$ uncorrected

Although the 4.85 could be used, usually it will be preferable to go to the extra stage and have the lower ratio.
For this solution, use four stages.
The interstage pressures will be by trial balancing and assuming that one half the intercooler ΔP of 5 psi is carried by each cylinder:

First-stage suction:

16 + 14.7 = 30.7 psia

First-stage discharge:

(30.7)(3.26) = 100.08 psia + 5/2 = 102.58
R_{c1} = 3.34

Second-stage suction:

102.58 − 5.0 = 97.58 psia

Second-stage discharge:

(97.58)(3.26) = 318.11 + 5/2 = 320.6
R_{c2} = 3.28

Third-stage suction:

(320.6) − 5 = 315.6 psia

Third-stage discharge:

(315.6)(3.26) = 1,028.8 + 5/2 = 1,031.3
R_{c3} = 3.26

Fourth-stage suction:

1,031.3 − 5 = 1,026.3 psia

Fourth-stage discharge:

(1,026.3)(3.26) = 3,345.7 psia
R_{c4} = 3.259

Note that the first approximation for "R_c" was obtained from Equation 12-41. This figure is not the exact compression ratio as it is difficult to calculate an exact ratio over such a wide range. The final R_c values calculated are close enough for process design calculation.

2. "k" value for the gas mixture:

	Fraction, y	Mcp*	(y) (Mcp)
Methane	0.6	9.15	5.48
Nitrogen	0.4	7.035	2.81
			8.29

*at 150°F from average data tables

$$k = c_p/c_v = \frac{8.29}{8.29 - 1.99} = 1.315$$

3. Moisture
Vapor pressure of water at cylinder suction temperature of 90°F is 0.6982 psia.

Total pressure at suction = 30.7 psia for 1st stage

$$\text{Mol \% of water in gas} = \frac{(0.6982)(100)}{30.7} = 2.275\%$$

Average molecular weight (dry basis),
= (0.60)(16) + (0.40)(28)
= 9.6 + 11.2 = 20.8
Total mol of gas on dry basis (60°F and 14.7 psia)
= 3,000/370 = 7.91 mol/min
Mol of water vapor

$$= \frac{(7.91)(0.02275)}{(1 - 0.02275)} = 0.1841$$

Total mol to first stage = 7.91 + 0.1841 = 8.0941 mol/min.

Second-stage suction:

$$\text{Mol \% water in gas} = \frac{0.6982(100)}{97.58} = 0.717\%$$

$$\text{Mol water vapor} = \frac{(7.91)(0.00717)}{1 - 0.00717} = 0.0571$$

Total mol to second stage = 7.91 + 0.0571 = 7.9671 mol/min.

Third-stage suction:

$$\text{Mol \% water in gas} = \frac{0.6982(100)}{315.6} = 0.221$$

$$\text{Mol water vapor} = \frac{(7.91)(0.00221)}{(1 - 0.00216)} = 0.01751$$

Total mol to third stage = 7.91 + 0.1751 = 8.085 mol/min.

Fourth stage:
Neglect effect of water vapor, as it will be considerably less than for the third stage.
Compressibility:
T_c-methane = 343°R, N_2 = 227°R,
P_c-CH_4 = 673 psia, P_c-N_2 = 492 psia
Pseudo-critical temperature:
T_c = (0.60)(343) + (0.40)(227) = 296.6°R
Pseudo-critical pressure:
P_c = (0.60)(673) + (0.40)(492) = 600.6 psia

$$\text{Reduced temperature at suction: } \frac{460 + 90}{296.8} = 1.85$$

Reduced Pressure:	*Compressibility, Z
1st stage: 30.7/600.6 = 0.5111	0.998 ≅ 1.00
2nd stage: 97.58/600.6 = 0.1624	0.992
3rd stage: 315.6/600.6 = 0.5254	0.976
4th stage: 1,026.3/600.6 = 1.708	0.925

*From compressibility charts, Figure 12-14.

4. Brake horsepower (Figure 12-21):

$$\text{bhp} = (\text{bhp/MMCFD})\left(\frac{\text{capacity}}{10^6}\right)(Z)$$

First stage, R_c = 3.34
Volume/capacity = 8.0941 mol/min (gas + water vapor, see previous paragraph 3)

$$\text{bhp} = (74.5)[(8.0941)(359\ \text{ft}^3/\text{mol @ 14.7 psia and 32°F}) \times (60\ \text{min/hr} \times 24\ \text{hr/day}) \times (14.7/14.14)\frac{(460 + 90)}{(460 + 32)}(1.00)/10^6]$$

$$= 74.5\left[\left(\frac{8.0941}{10^6}\right)(1440)(359)\left(\frac{14.7}{14.4}\right)\left(\frac{460 + 90}{460 + 32}\right)\right] \times (1.00)$$

$$= 355.7\ \text{bhp}$$

Second stage, R_c = 3.28

$$\text{bhp} = 73.3\left[\left(\frac{7.9671}{10^6}\right)(1440)(359)\left(\frac{14.7}{14.4}\right)\left(\frac{460 + 90}{460 + 32}\right)\right] \times (0.992) = 341.7$$

Third stage, R_c = 3.26

$$\text{bhp} = 72.8\left[\left(\frac{8.085}{10^6}\right)(1440)(359)\left(\frac{14.7}{14.4}\right)\left(\frac{460 + 90}{460 + 35}\right)\right] \times (0.976) = 338.8$$

Fourth stage, R_c = 3.259 (See note paragraph 3 previous)

$$\text{bhp} = 72.8\left[\left(\frac{7.91}{10^6}\right)(1440)(359)\left(\frac{14.7}{14.4}\right)\left(\frac{460 + 90}{460 + 32}\right)\right] \times (0.925) = 314.2$$

Total bhp = 355.7 + 341.7 + 338.8 + 314.2 = 1350.4, use 1,350
*Some manufacturers will not use a Z less than 1.0 in horsepower calculations.

Cylinders can be selected in the same manner as previously presented, remembering that it is the volume *at* the suction to the cylinder that is important. Therefore, the effect of the compressibility and moisture content must be reflected in the suction volume being considered. Be careful not to apply these factors twice, when calculating actual [(PD) (E_v)] from the horsepower equation.

If driven by an electric motor through a crankshaft frame, the horsepower at the output of the motor would have to be

(1,358)(1.05) = 1,428 hp

If driven by a gas engine, the rated hp of the engine must be at least 1,358 hp.

Air Compressor Selection

Air compressors are required for many services. Figures 12-33A–C and 12-34 Table 12-6, and performance curves are presented to indicated the type of units that are suitable. Other styles, sizes, and types are available, and their omission does not indicate a lack of suitability. Also see Schaefer[72] and Skrotzki.[73]

When establishing intake capacity for air compressors, the moisture content of the air must be taken into account. It is not always necessary or desirable to assume that the air is saturated; however, this is the maximum condition with respect to water content. Intake air temperature must be selected with some recognition of maximum-minimum-normal for summer conditions. Figure 12-35 is convenient for reading air conditions.

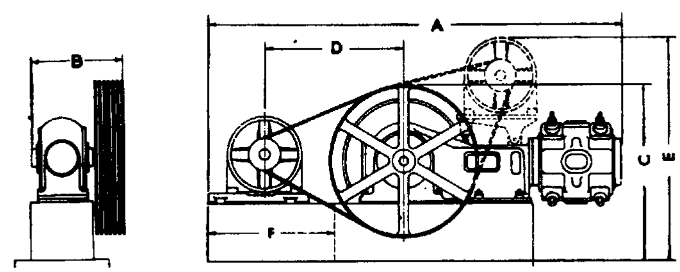


Figure 12-33A. Single-stage horizontal compressor. (Used by permission: Dresser-Rand Company.)

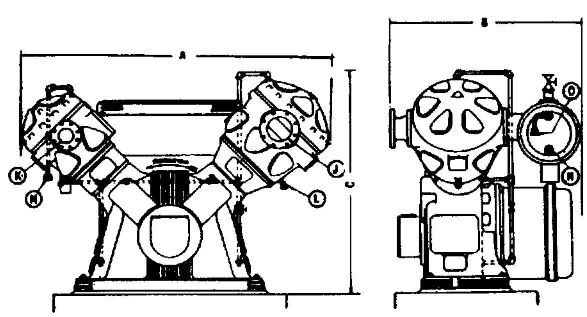


Figure 12-33B. Two-stage angle-type vertical compressor. (Used by permission: Dresser-Rand Company.)

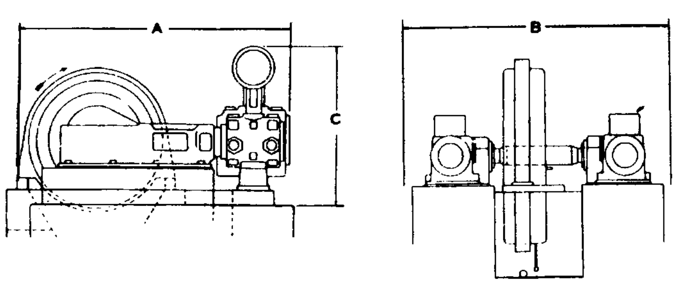


Figure 12-33C. Two-stage horizontal duplex compressor. (Used by permission: Dresser-Rand Company.)

Figure 12-36(1), part 1a illustrates[73] the state variation of the two-stage intercooled compressor on the T-S diagram (reproduced from Skrotzki[73] by permission):

"For points 2–3, there is constant entropy (S) compression for a one pound of air from P_2 to P_3. From points 3–5 the air cools at constant pressure, and gives up heat, Q, to the intercooler. From points 5–6 the air is compressed at constant S to the final pressure P_6. Note that point T_5 = point T_2 for constant temperature. For minimum work $T_6 = T_3$. Then the heat, Q, equals the Work, W_L of Figure 12-36B. Figure 12-38 is convenient for estimating the moisture condensed from an airstream, as well as establishing the remaining water vapor in the gas-air.

For three stages with two intercooling stages, Figure 12-36(2), the work of compression would be reduced more than for single or two stages. Thus the work can be deceased by increasing the number of stages and the intercooling between them. As discussed earlier the minimum total work to the gases can be achieved by the cube root of the total overall ratio of compression. Referring to Figure 12-36(2)b, $W_H = W_I = W_L$. Then for the optimum conditions, heat, Q, given up in the low pressure intercooler equals W_L work input

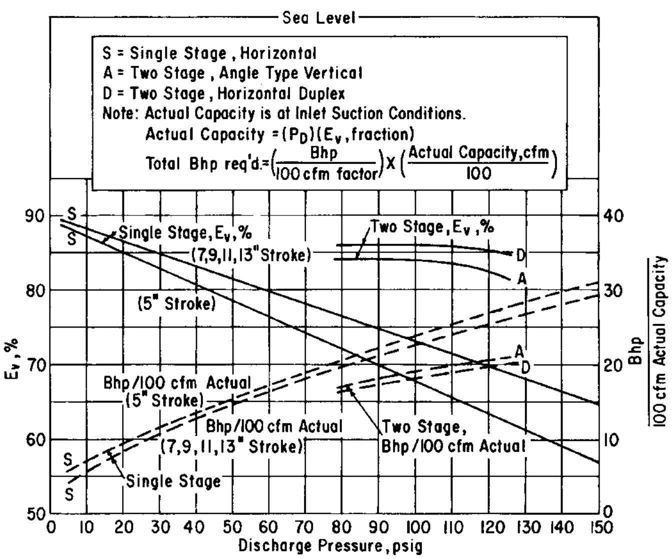


Figure 12-34. Typical air compressor performance, single- and two-stage. (Adapted and used by permission: Dresser-Rand Company)

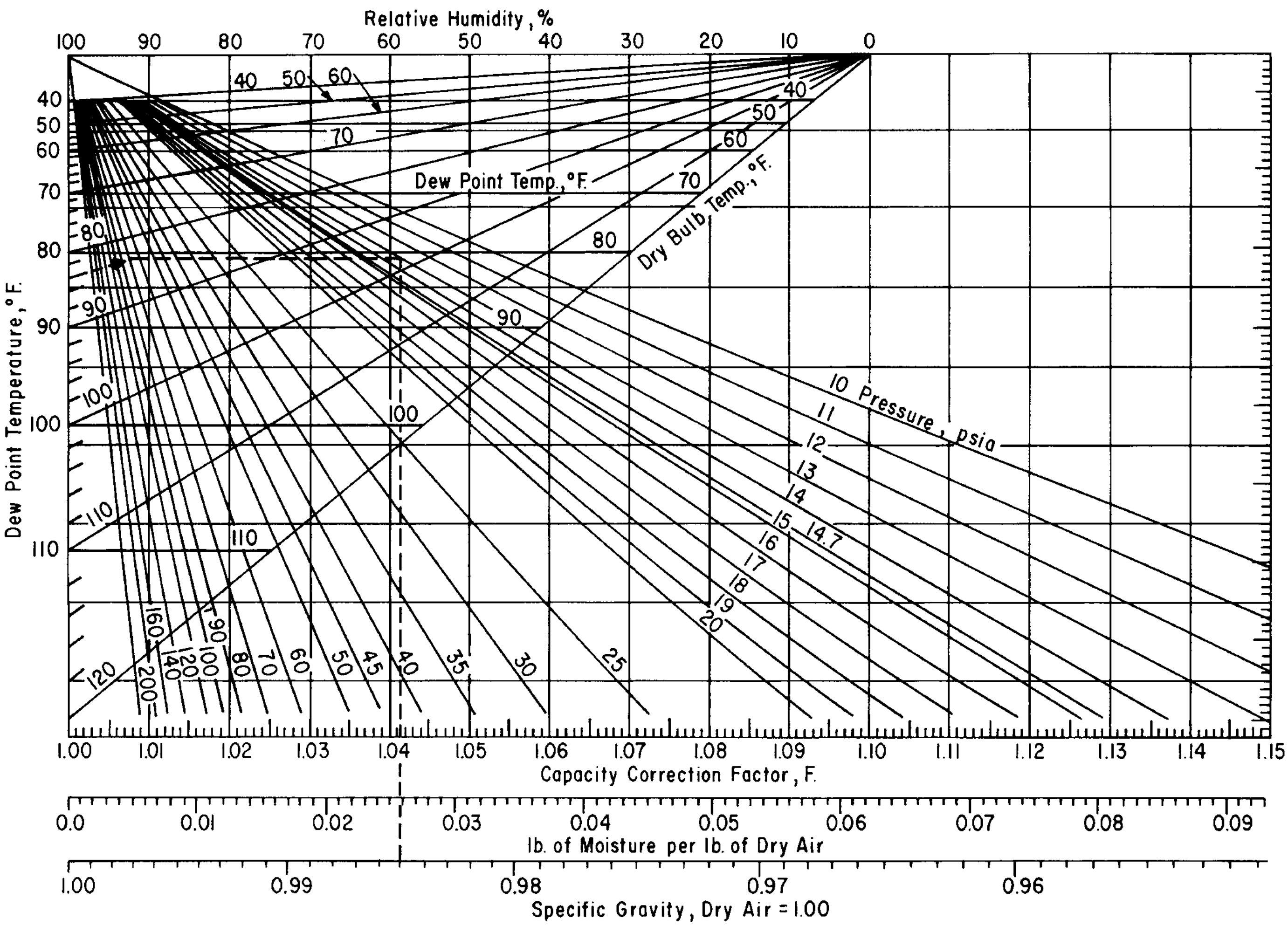


Figure 12-35. Air properties compression chart. (Used by permission: Rice, W. T. Report RP 383. Dresser-Rand Company; and *Chemical Engineering.* McGraw-Hill, Inc. All rights reserved.)

to the L-P (low-pressure) cylinder, and Q_2 equals W_I work input to I-P (intermediate pressure) cylinder. If there were an aftercooler following the H-P cylinder to cool the air to B in Figure (a), the area under the curve 9-B would equal the work of the H-P (high pressure) cylinder W_H."

Figure 12-37 illustrates work associated with three types of compression: (1) constant temperature, (2) polytropic, and (3) constant entropy. Figure 12-36(1)(a) shows the arrangement of a single-stage compressor with aftercooler and receiver. Note that the receiver acts as a reservoir for high pressure air to operate the engine, which may have a varying demand. The compressor runs at a steady rate.

Figure 12-38 provides a quick method of determining the change in moisture content of air.

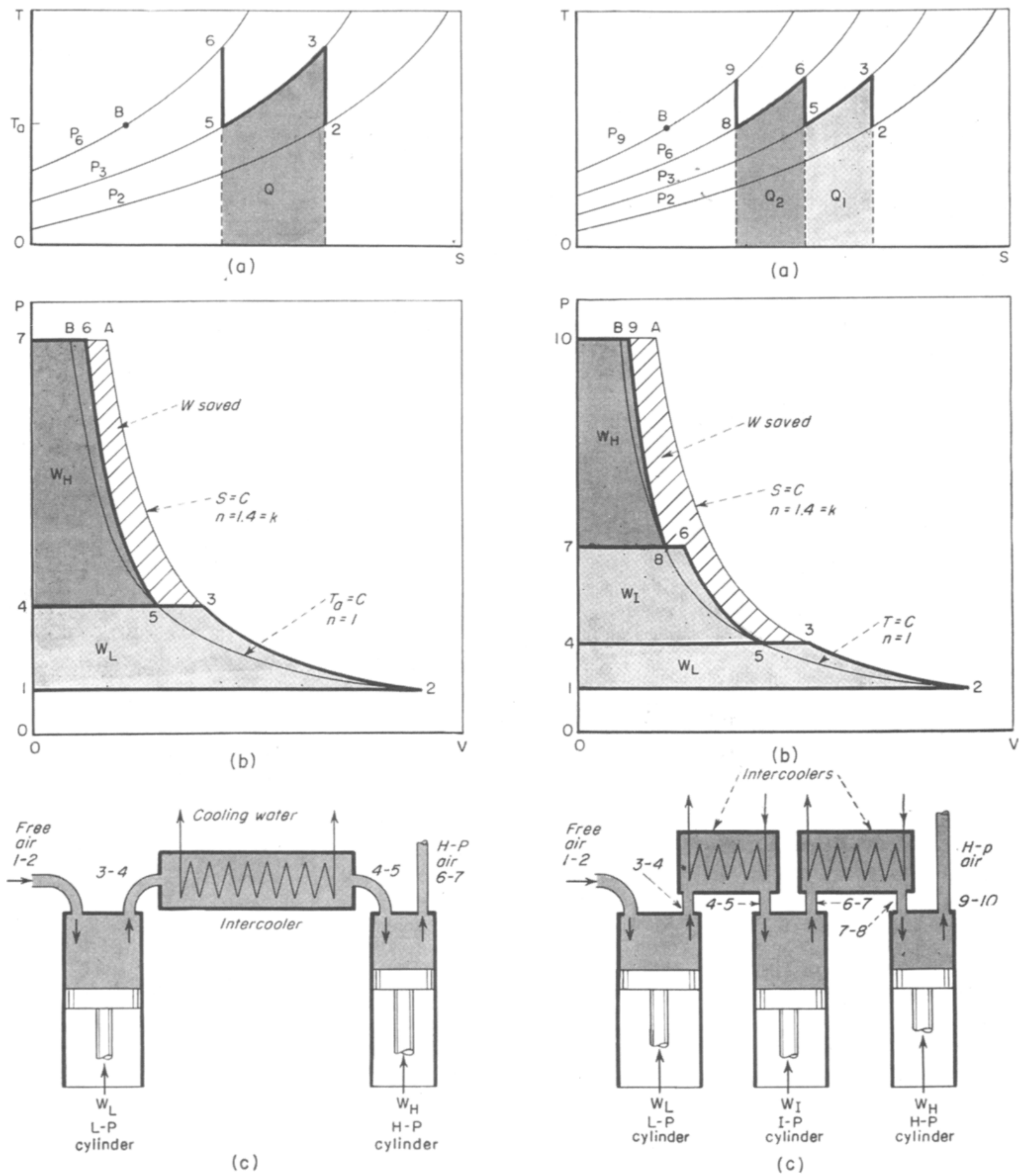
W_{in} is the power into the compressor shaft. Heat is rejected at Q_r and Q_{ri}. W_{out} is the work output of the system. Q_a is heat introduced into the system. Then for the energy balance (reproduced by permission from Skrotzki, B. G. A., *Power*, p. 92, Jan. 1958):

$$\text{"}W_{in} + Q_a = W_{out} + Q_r \tag{12-74}$$

The receiver acts as a storage reservoir for the h-p air; it keeps a reserve of air for the engine whose demand may vary widely. This allows the compressor to run at a steadier pace.

Energy flow

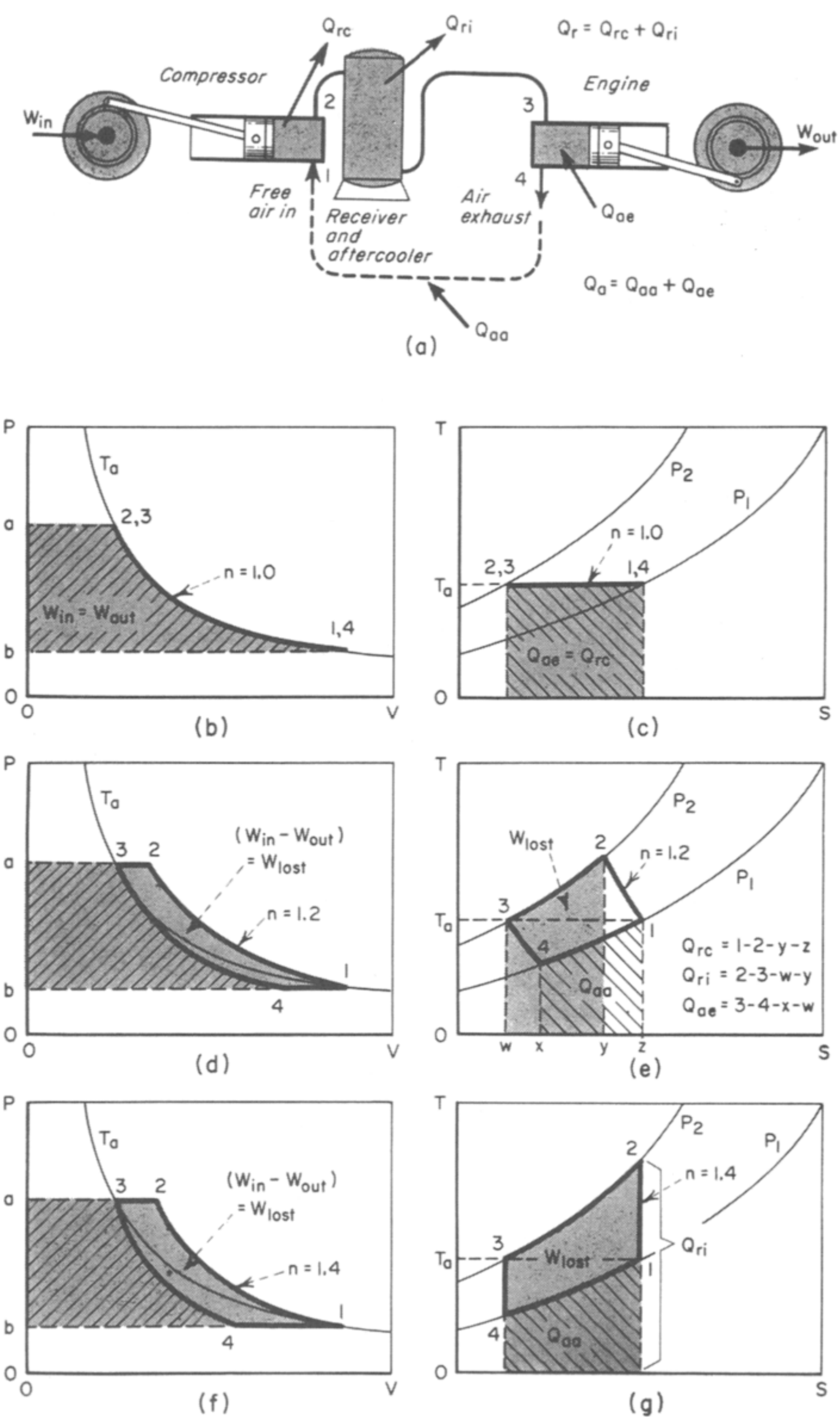
Energy flow into and out of the system takes two forms, work and heat. Prime energy source, Figure 12-37A, for the



1 Two-stage compression with intercooling cuts shaft work input, avoids too high a temperature of discharged air

2 Three-stage compression with two intercooling stages reduces work input more—used on higher-pressure air

Figure 12-36. Benefits of interstage cooling for air compression system. (Used by permission: Skrotzki, B. G. A. *Power,* p. 72, Jan 1958. ©McGraw-Hill, Inc. All rights reserved.)



Simple compressed-air system could work on constant-temperature processes, *b* and *c*; or on polytropic processes, *d* and *e*; or on constant-entropy processes, *f* and *g*. As *n* rises above 1.0, work lost keeps growing quite rapidly

Figure 12-37. Work associated with three types of air compression. (Used by permission: Skrotzki, B. G. A. *Power,* p. 72, Jan. 1952. ©McGraw-Hill, Inc. All rights reserved.)

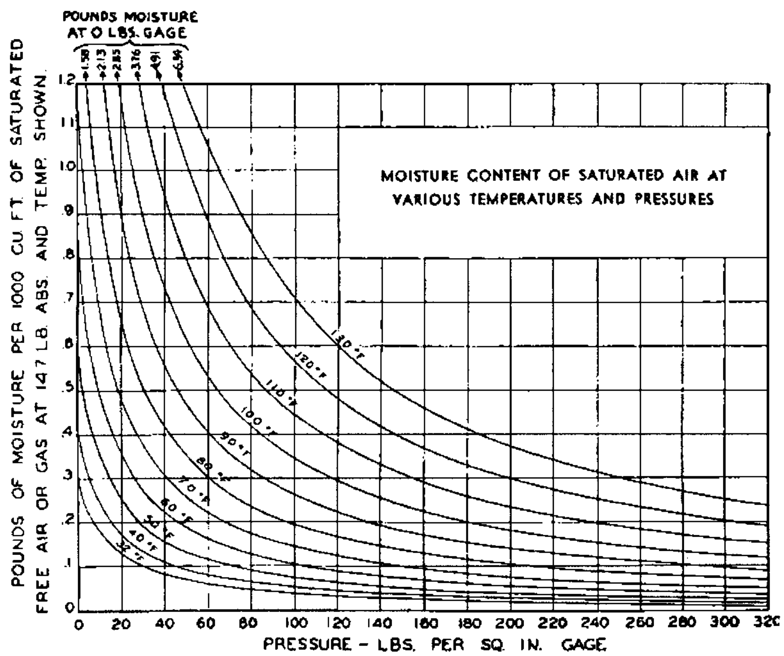


Figure Example: Saturated air at 80 at compressor intake (0 lbs.) contains 1.57 lbs. of moisture per 1000 cu. ft. Compressed to 100 lbs. and cooled to 80° with 65° water the air contains only 0.20 lbs. per 1000 cu. ft. or 13% of the moisture originally taken into the compressor. The rest has been condensed in the intercooler (if used) and the aftercooler.

Moisture as precipitated by aftercoolers.

Figure 12-38. Moisture as precipitated by aftercoolers. (Used by permission: Bul. L-600-B9-4, No. 4 in a series. Dresser-Rand Company.)

system is W_{in} to the compressor shaft. The compressor cylinder and receiver-aftercooler usually reject heat to the atmosphere, Q_r. The engine produces the major product of the system, $W_{out.}$ Usually the engine and atmosphere introduce heat into the "closed" system as Q_a.

According to the first law of thermodynamics, these four energy flows must always be in balance. For steady air flow through the system:

Energy in = energy out
$W_{in} + Q_a = W_{out} + Q_r$

Constant-T system

Figure 12-37B and 12-37C show the events for an ideal constant-T compressor and engine with an atomospheric temperature, T_a. W_{in}, the gray area, raises air pressure from 1 to 2 while Q_{re}, shaded area, flows through the compressor-cylinder walls to atmosphere. This system does not need an aftercooler. Air expands in the engine from 3 to 4 while it absorbs Q_{ae} from the atmosphere, hatched area, through the engine cylinder walls and produces shaft work W_{out}, hatched area. For reversible processes:

$$W_{in} = W_{out} = Q_{ae} = Q_{re}$$

This assumes, of course, that no pressure drops are in the system.

Polytropic System

Polytropic system, Figures 12-37D and 12-37E, gets nearer to the conditions of a practical system, with polytropic process having n = 1.2. The gray area W_{in} compresses air from 1 to 2 while the air rejects heat Q_{rc} to atmosphere through compressor cylinder walls. Q_{rc} equals area 1-2-y-z in 4e. In the aftercooler-receiver, the air rejects heat Q_{ri} equal to gray area 2-3-w-y.

In the engine, the air expands from 3 to 4 while absorbing heat Q_{ae} equal to area 3-4-x-w and produces work W_{out}, hatched area. Because the pressurized air draws on its internal energy to produce the work, its temperature drops below T_a, even though Q_{ae} enters the air during expansion. (In the constant-T cycle, Q_{ae} prevents the drop in temperature.) As the exhaust air returns through the atmosphere from engine to compressor, it absorbs heat Q_{aa}, hatched area.

PV graph shows that workflow into the system W_{in} is larger than work output W_{out}. The net cycle area for the system 1-2-3-4 measures the work lost by the system—external temperature irreversibilities cause this. All the processes, however, have been considered as internally reversible.

This contrasts with the engine cycles studied; for these, net area measured shaft work output, but for compressed-air systems, net area measures work lost. Remember, completely available energy, shaft work, runs compressed-air systems; higher-temperature heat runs engine cycles.

To continue the analysis, from the circuit flow diagram,:

$$W_{in} + Q_{ae} + Q_{aa} = W_{out} + Q_{rc} + Q_{ri} \quad (12\text{-}75)$$

$$W_{in} - W_{out} = Q_{rc} + Q_{ri} - Q_{ae} - Q_{aa} \quad (12\text{-}76)$$

From the PV graph,

$$W_{lost} = W_{in} - W_{out} \quad (12\text{-}77)$$

Substituting from the energy flow balance:

$$W_{lost} = Q_{rc} + Q_{ri} - Q_{ae} - Q_{aa} \quad (12\text{-}78)$$

The last equation means that the net area 1-2-3-4 on the TS graph also measures the net work lost, even though this is in Btu rather than ft-lb as on the PV graph.

Constant-S System

Figures 12-37F and 12-37G use adiabatic compression and expansion (zero heat transfer). All heat added to the cycle comes from heating the engine exhaust by Q_{aa}. Heat rejected from the cycle, Q_{ri}, leaves through the aftercooler.

Adiabatic expansion of the air in the engine causes a maximum temperature drop of the exhaust. Adiabatic compression causes a maximum temperature rise of the compressed air. These effects combine to cause the greatest work loss of any compressed-air system, when pressurized air must be cooled back to atmospheric temperature. The energy analysis parallels the one just made for the polytropic system. This shows that net areas on both PV and TS graphs measure the work lost.

If the pressure parts of an adiabatic system can be thoroughly insulated to prevent loss of Q_{ri} and the aftercooler can be dispensed with, this would be an efficient energy transmitting system, with no work lost. The compressor would work along 1-2 in Figures 12-37F and 12-37G and the engine along 2-1. There would be no areas on the TS graph, and the two areas for compressor and engine would be equal on the PV graph."

Example 12-7. Use of Figure 12-35 Air Chart[45] (©W. T. Rice)

Assume that a plant air system requires 10,000 ft^3/min of dry air measured at 14.7 psia and 60°F. The air is required at 100 psia. Intake conditions are

Atmospheric pressure at location = 13.0 psia
Temperature = 84°F
Humidity = 90°F

Aftercooler: Water used can cool air to 75°F. Pressure at this point = 100 psia. Referring to Figure 12-35, follow the dashed line, starting at the left scale at a dry bulb temperature of 84°F. Follow up and to the right to the intersection with a vertical relative humidity of 90%; follow across to the intersection with an inlet pressure of 13.0 psia, and read vertically down:

Moisture capacity correction, F = 1.0412
Lb water vapor/lb dry air = 0.026
Specific gravity of air-water vapor mixture = 0.985

At aftercooler conditions, the dry bulb equals the wet bulb temperature (air is saturated):

Wet bulb = dry bulb temperature = 75°F
Pressure = 100 psia
Lb water/lb dry air = 0.003
Dry air capacity at inlet = 10,000(14.7/13)(544/520)
= 11,820 cfm
Atmospheric air required = (1.0412)(11,820) = 12,310 cfm
Specific volume of moist air, v_{in}
= $(0.37)(T_1)/(\text{Sp. Gr.})(P_1)$
= 0.37(544)/0.985(13)
= 15.72 ft^3 moist air/lb
Weight of moist air = 12,310/15.72 = 783 lb/min
Weight of dry air = 783/(1.0412) (0.985) = 763 lb/min
Weight of water vapor entering compressor = 783 − 763
= 20 lb/min

Leaving:

Weight of water leaving *in air* from aftercooler = 763(0.003)
= 2.289 lb/min
Weight of water condensed = 20.0 − 2.289 = 17.711 lb/min

Centrifugal Compressors

The centrifugal compressor is well established for the compression of gases and vapors. It has proven its economy and uniqueness in many applications, particularly in which large volumes are handled at medium pressures. This compressor is particularly adaptable to steam turbine or other continuous speed change drives, as the two principles of operation and control are quite compatible. It is also adaptable to the electric motor, gas engine, and gas turbine with each installation being specific to a particular problem or process. Installation as well as operating costs can be quite reasonable.

Nissler[106] presents a rather complete review of centrifugal and axial compressors.

Mechanical Considerations

A centrifugal compressor, Figures 12-39A–D, raises gas pressure by accelerating the gas as it flows radially out through the impeller and converting this velocity energy to pressure by passage through a diffuser section. The casing is stationary, and the wheels or impellers mounted on the shaft are rotated by the driver. The units are usually mounted horizontally with horizontal split case for low pressures and vertical split case (barrel design) for high pressures about 800 psi.

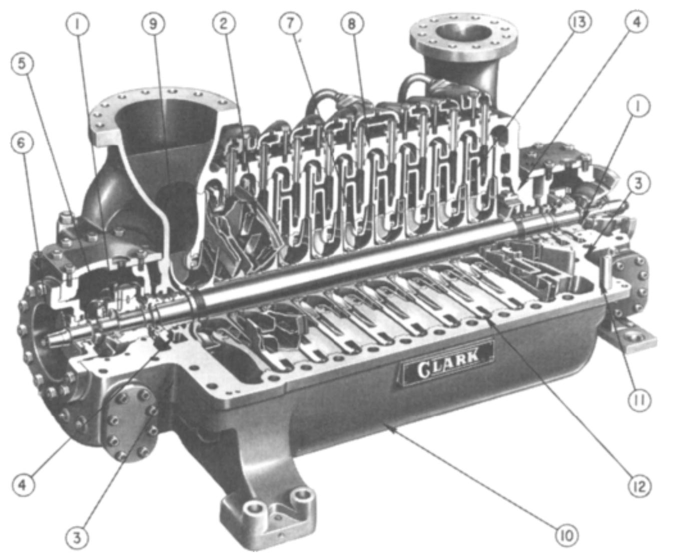
In general configuration the centrifugal compressor resembles a centrifugal pump. However, the significant difference in performance requirements lies in the compressibility of the gas. A dynamical analogy between these two items could be used to simplify the fundamental principles involved. Both receive mechanical energy from an outside source, and by rotating impellers they transform this into pressure energy in the fluid pumped. The centrifugal force depends on the peripheral speed of the impeller and the density of the fluid. The functioning of a centrifugal compressor depends more on the density and fluid characteristics of the material handled than does a reciprocating compressor. The peripheral speed and, hence, the head developed per stage is limited by the acoustic velocity, as it is believed that the peripheral speed should not exceed the speed of sound in the fluid being handled.

Significant features of centrifugal compressors are as follows.

Case

Centrifugal compressor cases are classified as *horizontally split* or *vertically split* (Figures 12-40A–C). The 5th Edition, April, 1988, API Standard 617 refers to the horizontally split style as "Axial Split" and vertically split units as "Radial Split." The first terminology is more standard than that of the API-617.[82] Figure 12-40D illustrates one flow arrangement for back-to-back internal flow, with the following advantages:

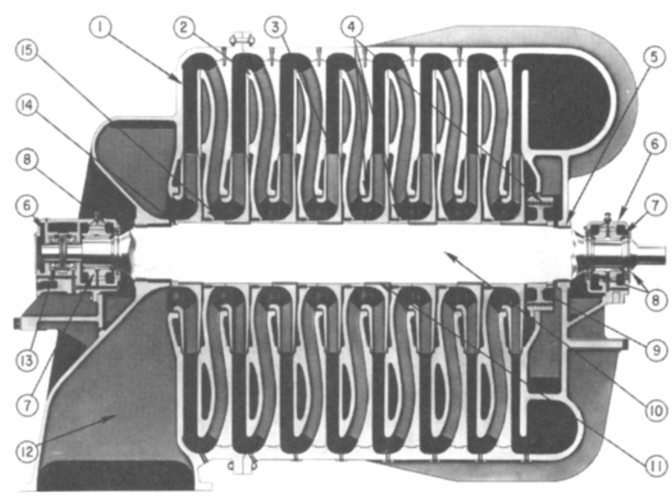
- Elimination of potential thrust bearing failure due to failure of the large diameter balance piston labyrinth.



The horizontally split units are accessible for inspection and cleaning. Removal of the upper half of the case permits access to the rotor, diaphragms, guide vanes, and other internal parts. Journal and thrust bearing and seals, however, can be examined or removed without disturbing the upper half of the case.

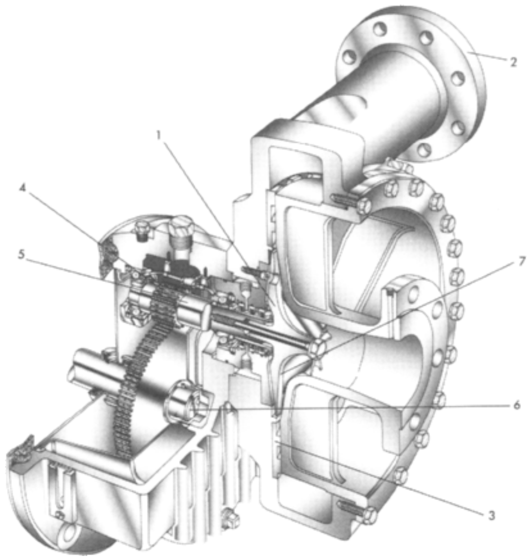
1. Integral bearing construction.
2. Double-wall cooled diaphragms reduce horsepower, allow higher compression ratios, maximum safety, avoid gas contamination, and ensure accurate process temperature control. No water connections are between halves. These are not used for all applications.
3. Bearing chambers are cast integral with case for freedom from alignment problems.
4. Isolation chamber eliminates external gas leakage to atmosphere.
5. Single- or double-thrust bearings accurately locate shaft. Balancing drum removes all but residual thrust.
6. Shaft extension permits tandem drive of five or more cases by a single driver.
7. Labyrinth seals are replaceable.
8. Return bends. Good hydraulic efficiency.
9. Fixed or adjustable inlet guide vanes for economy.
10. Interstage drains permit removal of condensate during operation or draining of "washdown" fluid.
11. Integral studs for maximum tightness.
12. Horizontally split design
13. Maximum compression ratio, maximum stages per unit.

Figure 12-39A. Internal construction features of multistage centrifugal compressor. (Used by permission: Dresser-Rand Company.)



1. Casing
2. Diaphragm
3. Impeller
4. Interstage seals
5. Shaft end seals
6. Bearing housing
7. Load bearing
8. Oil baffles
9. Balance piston
10. Shaft
11. Keyway
12. Inlet nozzle
13. Double-acting thrust bearing
14. Retaining nuts
15. Shaft sleeves

Figure 12-39B. Construction features of multistage centrifugal compressor. (Used by permission: ©A C Compressor Corporation. All rights reserved.)



1. Radial vaned impellers are investment cast to ensure precise contours and dynamically balanced for a smooth operation.
2. Discharge may be purchased in horizontal or vertical position for installation ease.
3. Choice of vaned or vaneless diffusers, providing optimum curve shape and high efficiencies.
4. Rugged integral speed increasing gearbox with precision ground gearing ensures smooth operation, with high mechanical efficiency and low noise levels.
5. Removable high-speed shaft assembly contains the bearings and seals and is easily removed to facilitate servicing.
6. Splash and pressurized lubrication systems are available. They are completely self-contained and require no external connections. Cooling is available. (Pressure lubrication is required on some models.)
7. Generous clearances of .035 in. (.889 mm) elminate performance deterioration caused by wear and the need for mechanical adjustment.

Figure 12-39C. Single-stage gas compressor with integrally geared drive shaft. (Used by permission: Bul. 1.10, Jan 1994. ©Sunstrand Compressors, Sunstrand Fluid Handling.)

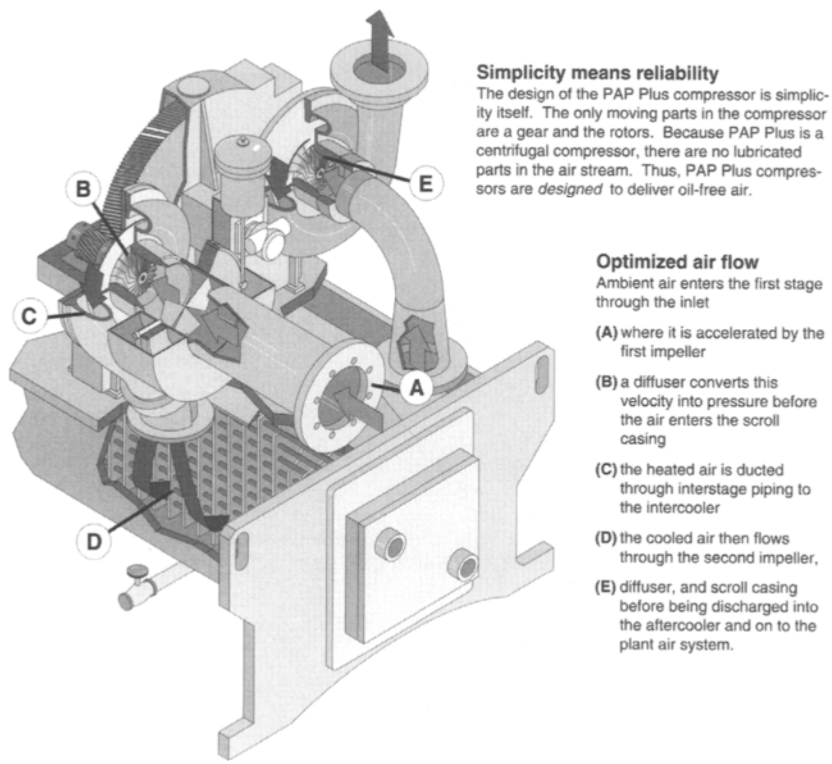


Figure 12-39D. Oil-free air compressor with two impellers, Elliott Company. "Plant Air Package®, Plus," gear driven. (Used by permission: Bul. P-51B. Elliott® Company.)

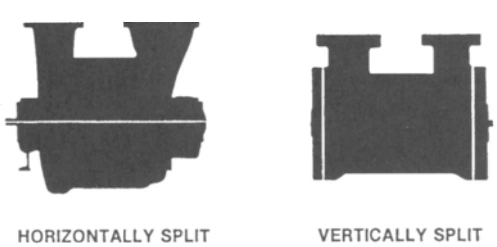
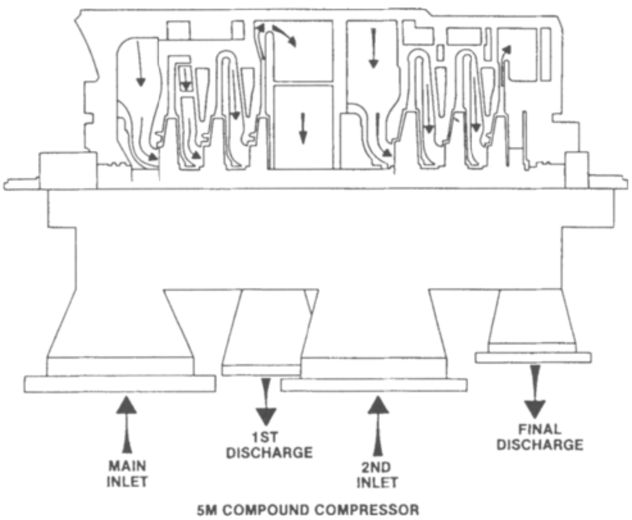


Figure 12-40A. Centrifugal compressor case types. These usually apply to multistage units rather than single-stage units. See Figures 12-39A and 12-39B. [Note API standards 617, April 1988, Par. 2.2.8.] (Used by permission: Bul. 423, ©1992. Dresser-Rand Company.)

(Balance piston labyrinths in straight-through designs are required to withstand differential pressures as high as 5,000 psi).

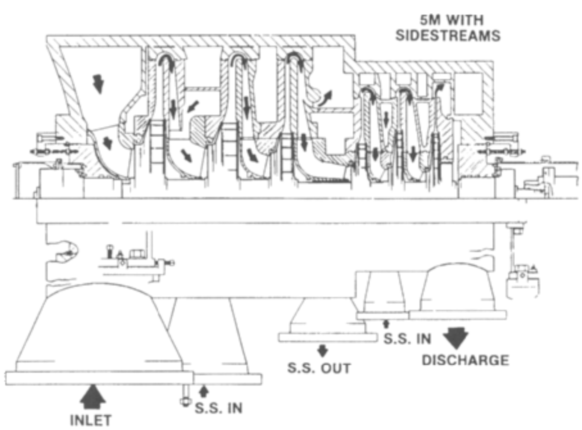
- Reduction of recirculation losses in the compressor since (1) the pressure exposed to the seals balanced to the compressor suction is an intermediate pressure as opposed to full discharge pressure on straight-through flow designs; (2) the seals that are balanced to suction are a much smaller diameter (and, therefore, have much smaller flow clearance area) than balance piston labyrinths on straight-through flow designs.[82]



Flow Path Arrangements: COMPOUND FLOW

For many high ratio applications, the capability to extract the total gas flow for intercooling is desirable to minimize gas temperature and power requirements. In many applications, compounding can reduce the number of compressor casings required. The flow path is the same as two "straight-through" compressors in series. that is, the total flow enters at the main inlet of the compressor and is totally discharged at the first discharge connection, is cooled or otherwise reconditioned, and re-enters the compressor at the second inlet connection and is totally discharged at the final discharge nozzle.

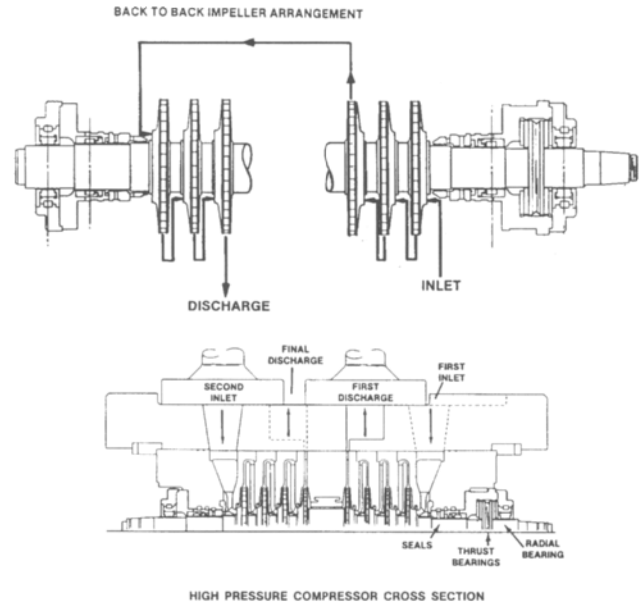
Figure 12-40B. Flow path arrangements: Compound flow. (Used by permission: Bul. 423, ©1992. Dresser-Rand Company.)



Flow Path Arrangements SIDESTREAM FLOW

For refrigeration cycles and other process requirements, the capability to admit or discharge gas at intermediate pressure levels is required. Compressors can be provided with sidestreams with minimum flow disturbance and provide effective mixing of the main sidestream gas flows. In the cross-section shown, examples of both incoming" and "outgoing" sidestreams are shown. Flow enters the main inlet and is compressed through one impeller to an intermediate pressure level at which point an incoming sidestream flow is mixed with the main inlet flow in the diaphragm area ahead of the next impeller. The total mixed flow is compressed to a higher pressure level through an outgoing sidestream to satisfy a process requirement. The remainder of the flow is compressed thrcugh one impeller, is mixed with an incoming sidestream, compressed through two stages, and exits through the final discharge.

Figure 12-40C. Flow path arrangements: Sidestream flow. (Used by permission: Bul, 423, ©1992. Dresser-Rand Company.)



This arrangement internally utilizes back-to-back impeller arrangements providing for thrust balance without the use of a balance piston. This type of configuration has added advantages in high pressure, high case lift (pressure rise across the case) applications.

Figure 12-40D. Flow path arrangement: Back-to-back flow. (Used by permission: Bul. 423, ©1992. Dresser-Rand Company.)

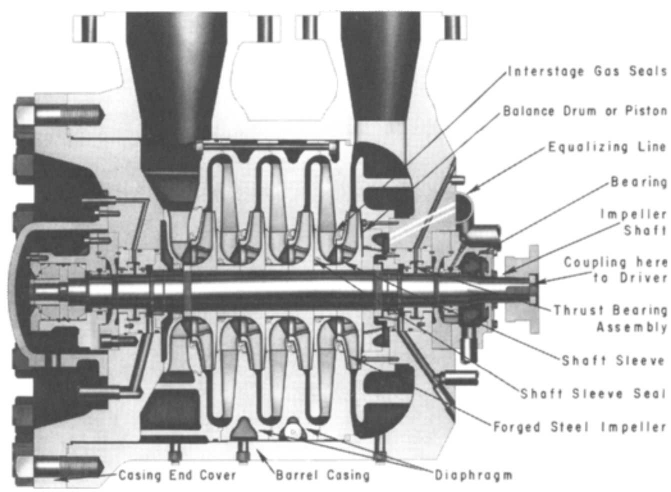
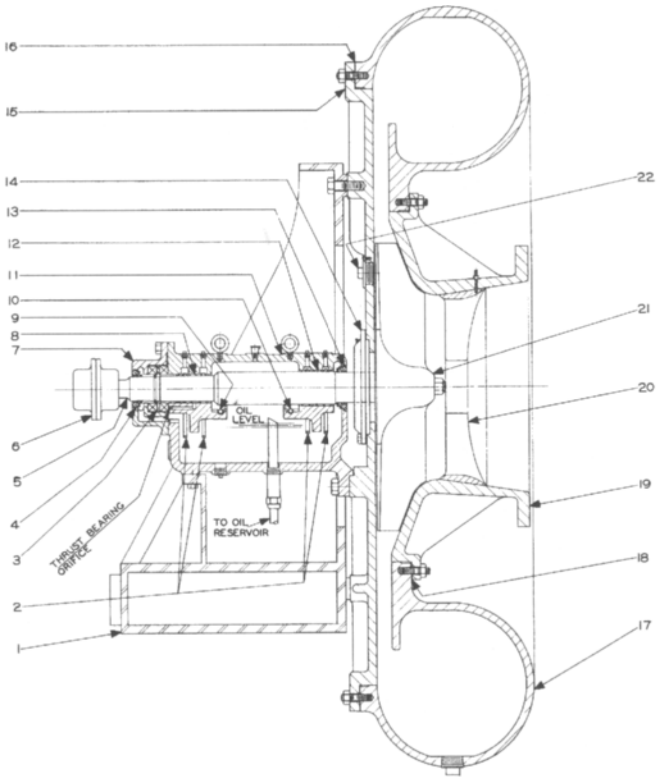


Figure 12-40E. Barrel type compressor section, horizontal mounting. (Used by permission: Cooper-Cameron Corporation.)

The outer shell or case is usually split for assembly along a horizontal centerline, Figures 12-39A–D, for units operating with pressures up to 800 psi. The single-stage centrifugal blower operates as a centrifugal compressor but is limited to a case pressure of about 275–375 psig and 2 to 3.5 ratios of compression, Figure 12-41. Above this pressure ratio, the multistage split case units, Figure 12-40E, become more economical and have better mechanical design features for applications as high as 5,000 psi.



1 Base
2 Oil rings (if required for start-up)
3 Thrust bearing
4 Oil guard—coupling end
5 Shaft
6 Coupling
7 Thrust bearing housing
8 Bearing sleeve—coupling end
9 Thermometer—coupling end
10 Thermometer—impeller end
11 Bearing housing
12 Bearing sleeve—impeller end
13 Oil guard—impeller end
14 Packing box
15 Backplate
16 Gasket
17 Casing
18 Gasket
19 Inlet connection
20 Inlet guide vanes
21 Impeller
22 Inspection hole plug

Figure 12-41. Single impeller centrifugal blower. (Used by permission: Elliott® Company.)

For the usual situation the gas inlet and outlet connections can be arranged at either the top or bottom and sometimes in horizontal locations. This should be arranged to suit the piping of the installation and also to avoid liquid holdup or drainage into the unit. The manufacturer's mounting instructions regarding expansion, etc., should be carefully followed, as the nozzle connections on the case are not designed to carry piping loads.

Materials of construction depend on the metallurgical requirements and pressure of the gas being compressed but usually the more "popular" materials include cast iron, cast steel, alloy steel, or forged steel (high pressure). Figure 12-42 illustrates the extent of damage internally when materials of construction are not resistant to possible effects of internal corrosion.

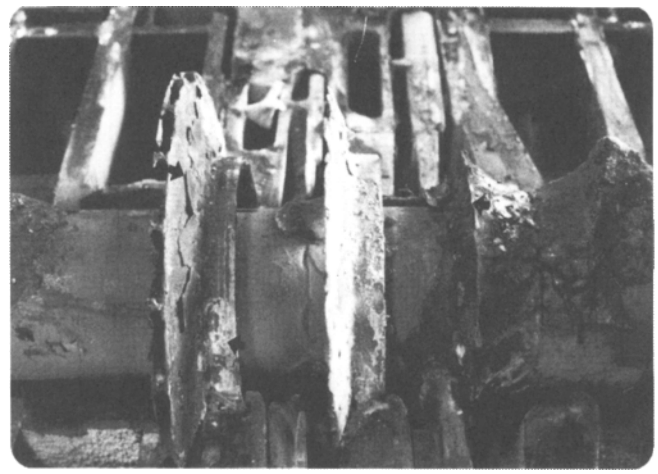


Figure 12-42. Results of *internal* chlorine gas fire and extensive corrosion in centrifugal compressor. Middle section of shaft with extensive damage to loss of center compression wheels. Note that ferric chloride is present.

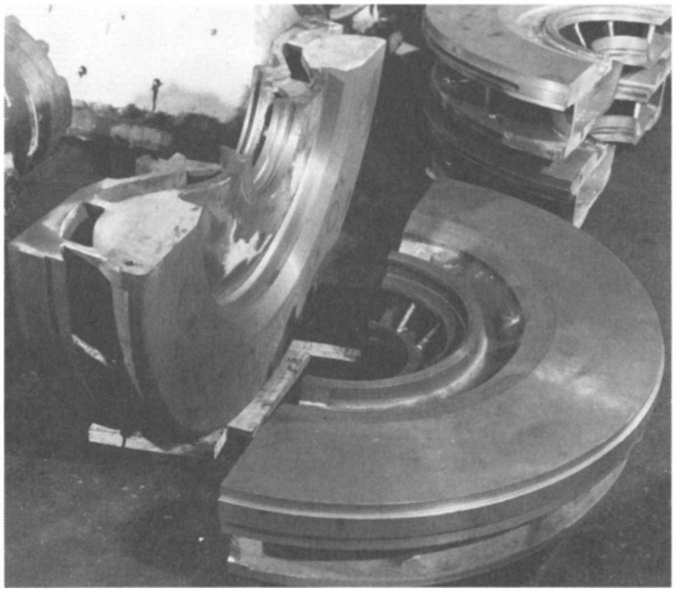
Diaphragms and Diffusers

The diaphragms shown in the assembly, Figures 12-39A and 12-39B, may be uncooled or liquid cooled. Figures 12-43A and 12-43B represent the two halves of uncooled diaphragms for a horizontally split unit. These are inserted in matching casing locating grooves and locked in each casing half. The diaphragms are the separation walls between the successive impeller stages. The diaphragms form the diffuser walls and the return passages for guiding the gas to the inlet of the next higher stage impeller. The design of these diaphragms has an important bearing in the operational characteristics of the machine. The spaces between faces are carefully proportioned to match the impeller and to minimize pressure losses. See Figure 12-44A.

Diffuser vanes are used to decelerate a high velocity flow to create a pressure rise. They are usually at the periphery of each impeller. The variable diffuser vane system may be controlled manually by a handwheel or automatically by a hydraulic or air-operated positioner.[88] See Figure 12-44B.

The return channels (passages) contain vanes that direct and evenly distribute the flow of gas into the guide vanes of the next impeller. Most diaphragms for horizontally split compressors are of the uncooled open diffuser type. Water-cooled diaphragms are used to cool the discharge temperature of the gas stream, but this cooling may be limited in degree and not provide a total cooling in place of a final external cooler.[87]

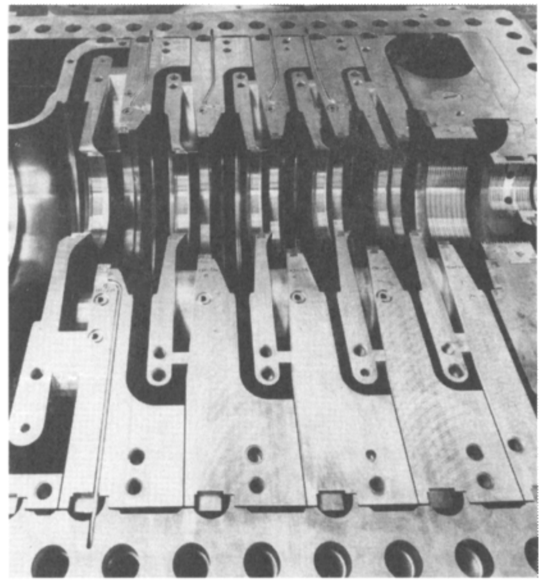
Water-cooled diaphragms, Figures 12-44A and 12-39A, are used to cool the surfaces of the metal passages and, thereby, reduce the temperature of the gas as it passes through the machine. This will often allow for higher ratios of compression in the same machine case. For some designs, these are



Interstage diaphragms, usually made of cast iron are horizontally split and inserted in matching casing locating grooves and locked in each casing half. The diaphragms form the diffuser walls and the return passages for guiding the air or gas to the inlet of the next impeller.

Diffusion action is accomplished by radially increasing the flow area of the diffuser and thus converting velocity head into pressure head. The diffuser passages are parallel-sided and do not include vanes. However, return passages do contain vanes to redirect the flow to the next impeller. Diaphragms are designed to accommodate the interstage labyrinth seals at the shaft and impeller cover disc. The seals prevent interstage recirculation.

Figure 12-43A. Multistage centrifugal compressor uncooled diaphragms for horizontally split casings. (Used by permission: A C Compressor Corporation.)



diaphragms

Suction, intermediate and discharge diaphragms create the gas flow path within the stationary components.
The suction diaphragm conveys the gas into the eye of the first impeller and can be fitted with adjustable guide vanes to alter the inlet flow angle. Intermediate diaphragms perform the dual function of forming the diffuser passage (where gas velocity is transformed into pressure) and the return passage to channel gas to the eye of the next impeller.
The discharge diaphragm forms the diffuser for the last impeller as well as the discharge volute.
The diaphragms are usually horizontally-split. In the MCL series the upper halves are fixed to the casing to facilitate rotor inspection.
The diaphragms are made of cast-iron, special cast iron, steel or stainless steel.
Easily removable labyrinth seals are installed on the diaphragms at impeller shrouds, to prevent return flow from discharge to suction and on the shaft sleeves to eliminate interstage leakage.

Figure 12-43B. Diaphragms. (Used by permission: Bul. PROM 526/I5/95 -II, © S.p.A., Nuovo Pignone, Florence, Italy; New York; Los Angeles; and Houston. All rights reserved.)

used on high-pressure or high compression-ratio applications, for hazardous gases, or temperature-sensitive materials. The careful installation of the water-cooled diaphragms prevents contamination of the process gas by the cooling fluid (usually water). Despite the care and attention to proper sealing, assembly, etc., the chance of a coolant leak into the gas or vice versa always exists. Extreme attention to mechanical details at this point are important. Some gaseous dry-chlorine compressors use a noncorrosive chlorinated solvent as the diaphragm coolant to guarantee no water in the system.

Internal liquid injection into the diffuser passage is used in a few applications for direct contact cooling. The quantity and quality of the liquid must be carefully controlled.

Materials of fabrication again vary with the nature of the gas being compressed but are usually low alloy steel, such as AISI 4140 or 4340, heat treated at 1,100°F to Rockwell hardness C-26 to 30, AISI Type 410 stainless steel, precipitation-hardening stainless such as Armco 17-4PH or 15-5 PH, Type

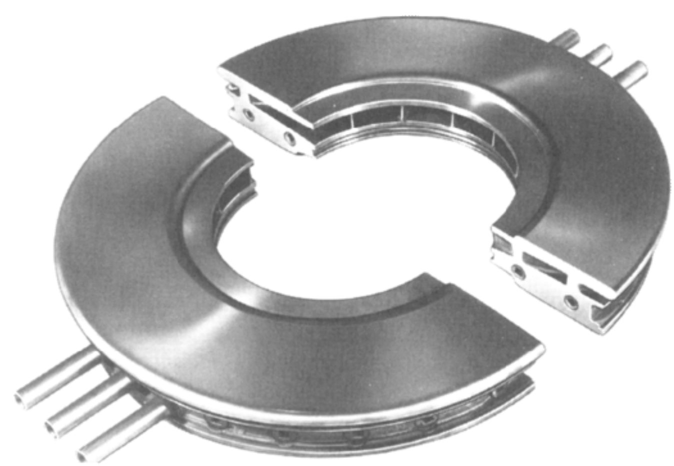


Figure 12-44A. Water-cooled diaphragm. (Used by permission: Dresser-Rand Company.)

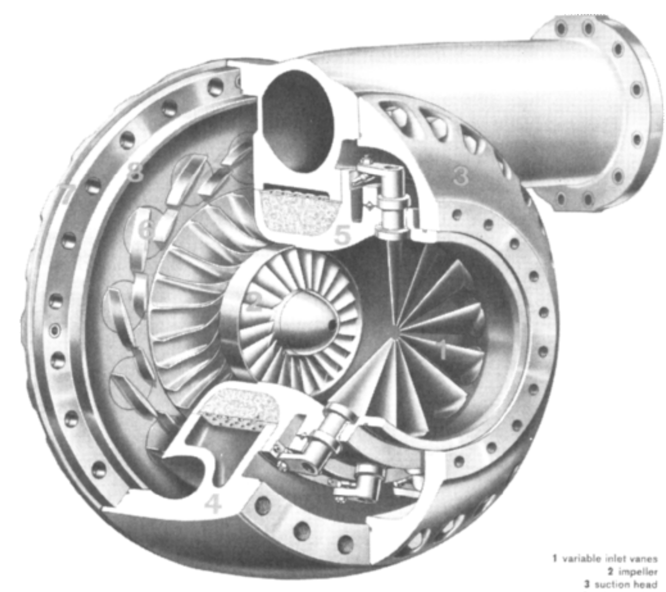


Figure 12-44B. Single-stage centrifugal compressor with variable inlet vanes, diaphragm, and diffuser vanes illustrated. (Used by permission: *Worthington Power and Fluids,* V. 8., No. 2, Winter, 1965. ©Dresser-Rand Company. All rights reserved.)

304 or 316 austenitic stainless for very low tip speed impellers, or Monel® K-500 for halogen gases. Aluminum is used for certain applications.[99] Hydrogen embrittlement can cause severe failure problems with many metals when hydrogen is present in the gas mixture.

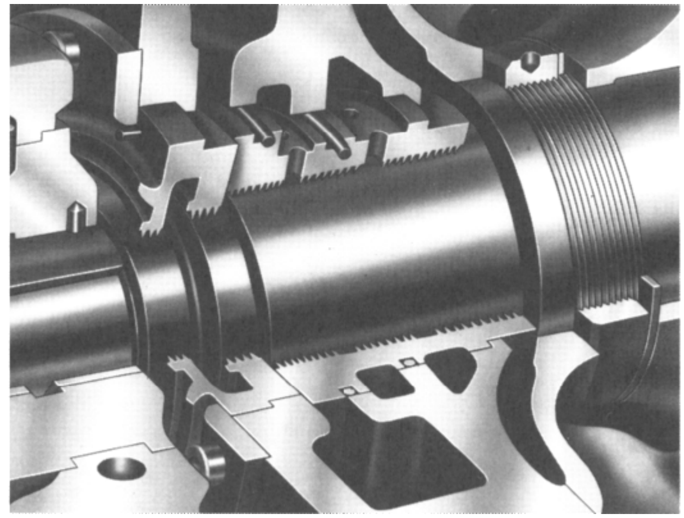
Labyrinth Seals

The seal of the rotating shaft to the diaphragm is usually a labyrinth, Figure 12-45. With proper design, these can effectively seal the pressure of one stage or wheel from the other in the case. Extreme care is needed in selecting the labyrinth material, as it must be soft enough not to score the shaft, yet maintain its shape and be resistant to any corrosive materials. The seal is effected when the knife edges of the labyrinth shape themselves on the shaft. Care in the initial shaft rotation and the cleanliness of the parts are important for an effective seal.

Impeller Wheels

The common wheel types are (1) backward and (2) radial. The types of construction of wheels are milled, fabricated, welded, and cast. See Figures 12-46A–H.

The fabricated or built-up wheels are machined from high-strength alloy steel forgings. These may be riveted or welded for assembly. The one-piece wheels are usually smaller than the built-up wheels and are milled from a solid forging. The cast and welded wheels are usually only made above a certain diameter due to fabrication problems in smaller diameters. Each manufacturer has his own recommendations for each application.



The simplest and most commonly used seal is the labyrinth. It is composed of a series of restrictive rings or a grooved sleeve with the edges of the rings or grooves machined to knife edges. The single labyrinth is used when there is a low sealing pressure differential. Interlocking labyrinths are used for higher differentials. Labyrinth seals operate by breaking down the pressure differential across the labyrinth.

Figure 12-45. Labyrinth seals. (Used by permission: Dresser-Rand Company.)

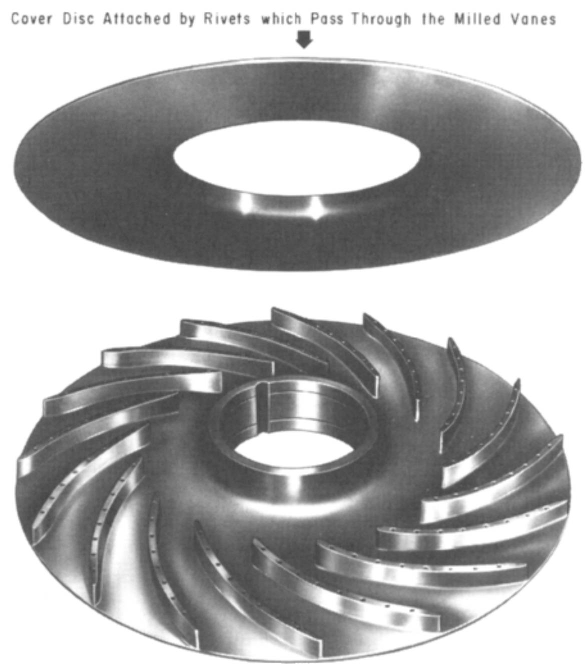


Figure 12-46A. Impeller with milled vanes on solid disc forging. (Used by permission: Dresser-Rand Company.)

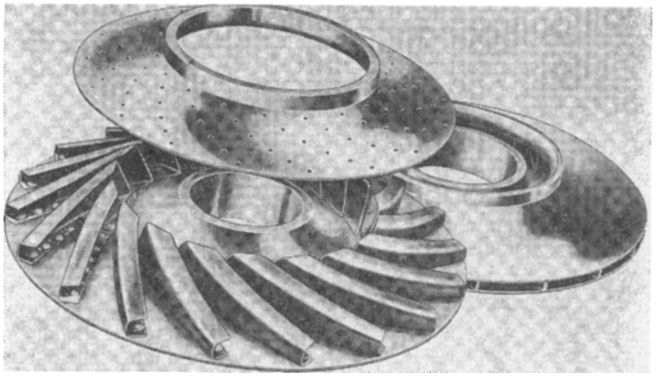
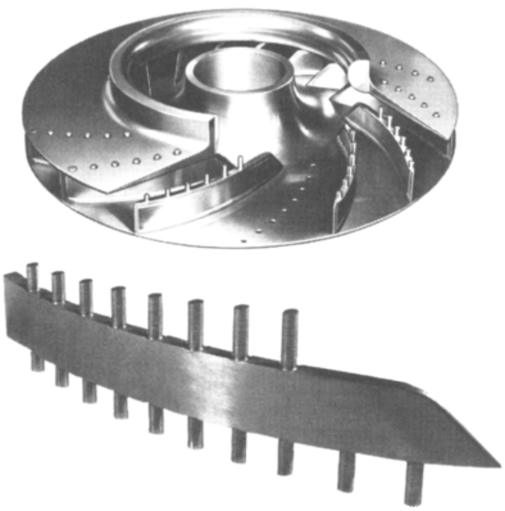


Figure 12-46B. A special design riveted wheel, used only on small wheels where welding is impractical. (Used by permission: Elliott® Company.)



The impellers for centrifugal compressors are assemblies consisting of three parts: the hub disc, the blades, and the cover disc. The hub and cover disc are machined from single-piece forgings of an alloy steel suitable for the application of the compressor. Blades are machined from forged steel plates of identical material. Each forging is checked with either sonic or X-ray machines to detect flaws or inclusions.

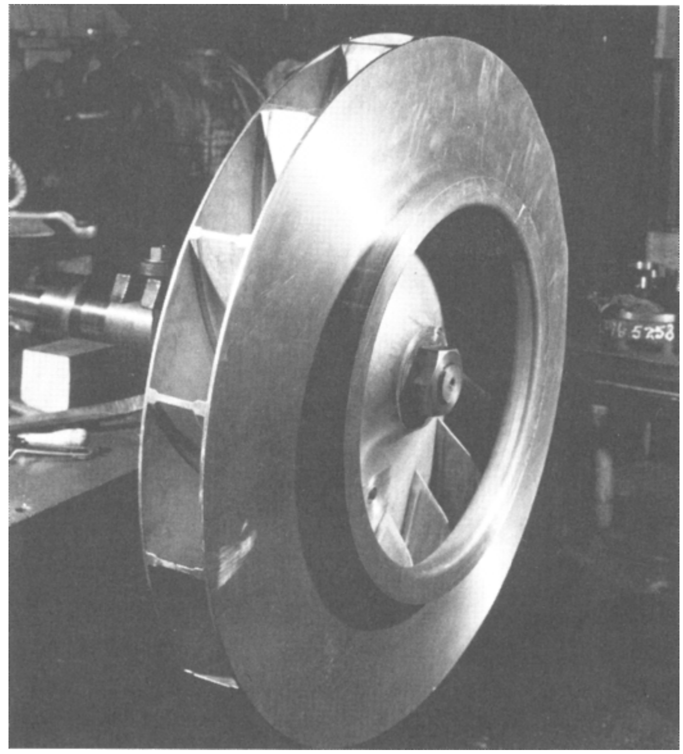
The hub disc forms the back portion of the impeller. The hub disc has the proper design to form one-half of the inlet nozzle of the impeller and carries the blade and cover disc on the shaft.

The blades are completely machined and formed to shape, leaving materials that form the rivet integral with the blade.

The cover disc forms the inlet of the impeller and is accurately machined to allow smooth flow. A sealing surface is provided on the cover disc to prevent recirculation of the gas.

The component parts are placed in the fixture, and the drilled holes for the rivets on both the hub and cover disc are countersunk and polished. The entire assembled impeller is statically balanced. The impeller is mounted on an overspeed mandrel and run at 25% overspeed.

Figure 12-46C. Cutaway of riveted wheel. Blades are riveted to hub disc, and the cover disc is drilled, ready for securing blades. (Used by permission: A C Compressor Corporation.)



The component parts are heated. The blade shape is laid out on the hub disc and tack welded, which ensures that the blade cannot be misaligned. The hub disc with the tack welded blades are placed on the automatic impeller welding machine and heated for welding. After the blades are completely welded to the hub disc, the cover disc is welded on by the same means. The completely welded impeller is then ready for annealing and finish machining.

Figure 12-46D, Part 1. View toward inlet of 4 $^1/_2$-in. diameter brazed aluminum impeller. Note: Regardless of the metal of manufacture, enclosed impellers with back-leaning blades are extremely useful in applications requiring a steep head-volume characteristic and the highest attainable efficiency. Applications include parallel operation with other compressors or boosting of another compressor's output.[83] The power-volume curve will show a self-limiting feature at higher volumes. This feature is very beneficial when the driver has limited power available but operation throughout the full capacity range is required. (Used by permission: A C Compressor Corporation.)

Figure 12-46D, Part 2. Close-up welding of section of welded impeller. (Used by permission: Elliott® Company.)

Figures 12-46 illustrates the key styles of wheels for use on the rotating shafts of multistage centrifugal, single-stage compressors and fans, and axial flow compressors (to be discussed later in this chapter). The flow-handling capability of a specific impeller geometry is described by the "flow coefficient,"

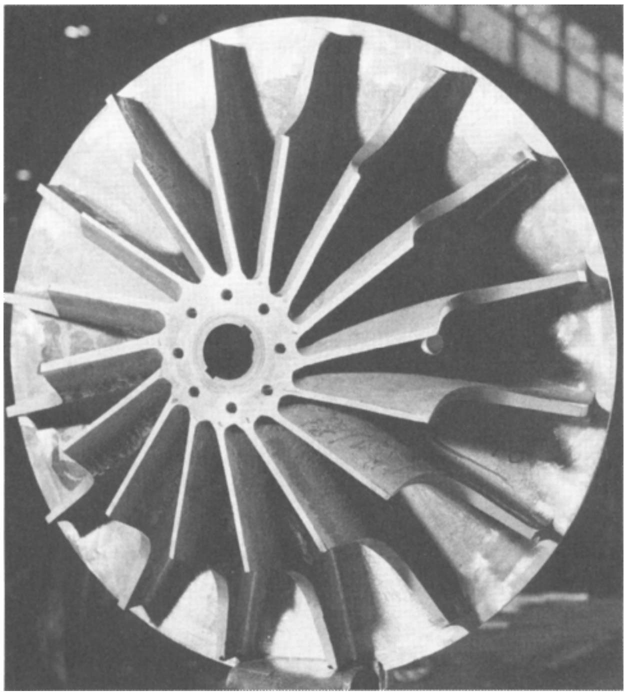


Figure 12-46E. Radial impeller for single-stage compressor. (Used by permission: A C Compressor Corporation.)

$$\Phi = Q/(\pi/4)(D^2\upsilon) \qquad (12\text{-}79)$$

where Q = gas flow, ft^3/sec
D = impeller diameter, ft
υ = tip speed of impeller, ft/sec, expressed by:
$\upsilon = (\pi/720)D'N$, ft/sec
N = shaft speed, rpm (revolutions per minute)
D′ = impeller diameter, in.

$$\text{or, } \Phi = 3.056\, Q/(D^2)(\upsilon) \qquad (12\text{-}80)$$

$$\text{or, } \Phi = 700\, Q/D'^3N \qquad (12\text{-}81)$$

$$\text{Then: } Q = \frac{(Q)}{(3.056)}(D^2\upsilon) \qquad (12\text{-}82)$$

This represents the linear effect of tip speed and the square of impeller size on the flow of a specific impeller. Referring to Figure 12-46, very low flow coefficients for a specific type of centrifugal or axial flow machine cause excessive wall friction or leakage losses, and very high-flow coefficients tend to be subject to turbulence losses due to insufficient flow guiding.[85]

The values of Φ for high-pressure barrel compressors at the inlet are not to exceed 0.07 for high tensile strength steel impellers.[86] For multistage (not barrel) compressors with flow coefficients

- at high Φ values, gas velocities in the impeller flow channels increase and lead to a fall in efficiency with increasing Φ.
- at low Φ values, efficiency falls rapidly due to the increasing influence of the shroud leakage loss and of

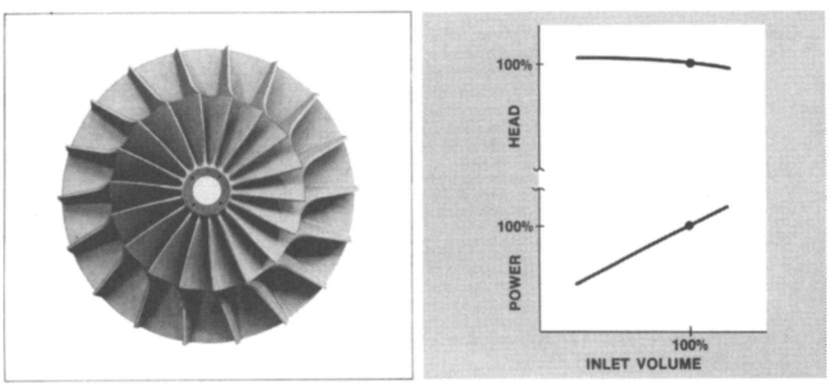


Figure 12-46F. Open radial blade impeller, Type "R." (Used by permission: Bul. "Centrifugal Compressors Single Stage." A C Compressor Corporation.)

Ideally suited for dirty gas, corrosive or high head applications. Their inherent self-cleaning design permits longer operation on dirt-laden or corrosive gas service without shutdown. At a given tip speed, higher heads are possible than with other impeller designs. A wide variety of materials are available to meet special application needs. Radial bladed impellers have a relatively flat head-volume characteristic, thus permitting a wide variation in volume with little change in pressure. The efficiency is almost constant over the normal range of operation. As a result, the power requirements are proportional to flow.

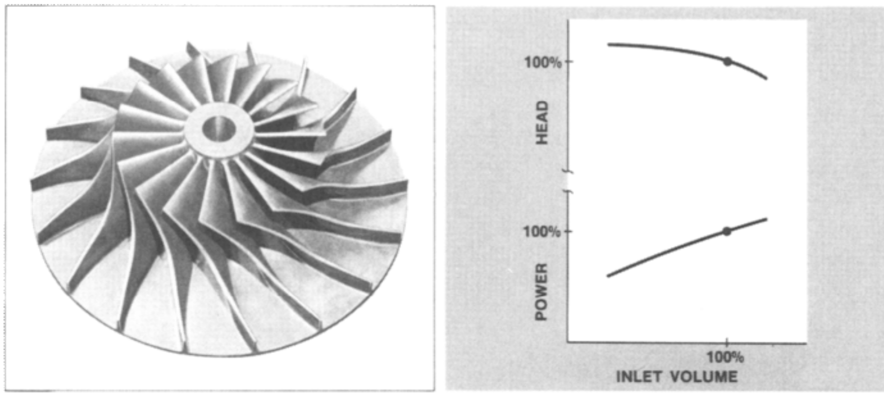


Figure 12-46G. Open impeller with back-leaning blades, Type "SR." (Used by permission: Bul. "Centrifugal Compressors Single Stage." A C Compressor Corporation.)

Open impellers with back-leaning blades combine the best features of both the radial bladed type and enclosed type with back-leaning blades. They have moderately rising head-volume characteristics, which not only permit a wide range of flows in a given impeller but also allow a moderate variation in pressure. Efficiency is greater than that of a radial bladed type. The power-volume characteristic rises with flow at a rate slightly less than that of the radial bladed type.

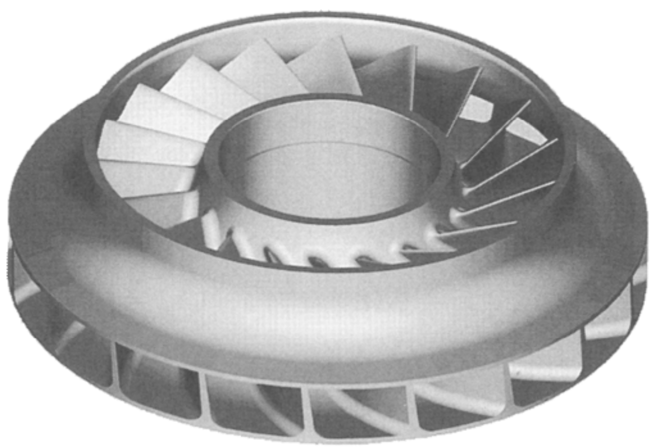


Figure 12-46H. Advanced datum impeller design reduces operating stresses and improves flow velocity. (Used by permission: Dresser-Rand Company. All rights reserved.)

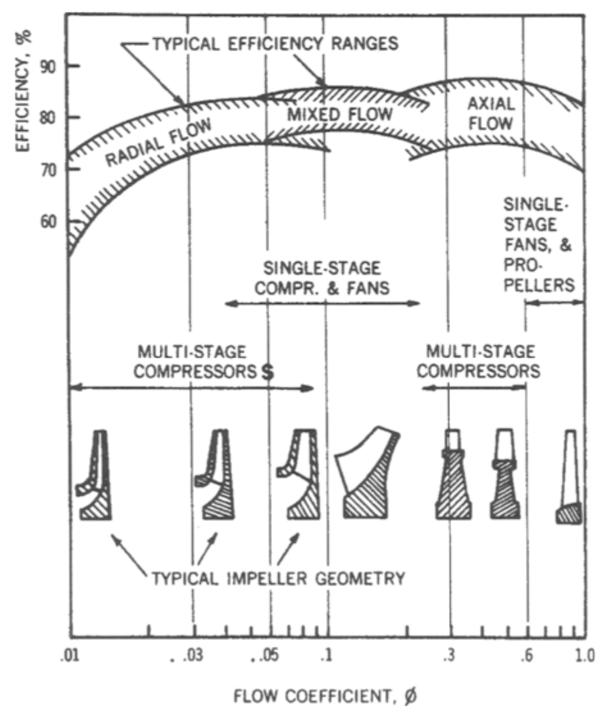


Figure 12-46I, Part 1. Impeller geometry versus flow coefficient, dimensionless to define the flow-handling capability of a specific impeller (wheel) geometry, ($\Phi = Q/(\pi/4)(D^2\upsilon)$). (Used by permission: Fullemann, J., Technical Report "Centrifugal Compressors," ©1963. Cooper Energy Services, Cooper-Cameron Corporation. All rights reserved.)

unavoidable friction loss in the boundary layer of the impeller disks and blades.

- where the Φ values of the impeller are kept in the 0.03–0.09 range, the efficiency of a multistage compressor can be improved.

Another design parameter is impeller tip speed as this relates to pressure differences by changing fluid velocities. Tip speed is defined previously.[85]

The head output of a compressor is a square function of the tip speed alone[85] and independent of the size of the machine itself. The volume flow is determined by both the rotor/wheel diameter and the tip speed for a given blade

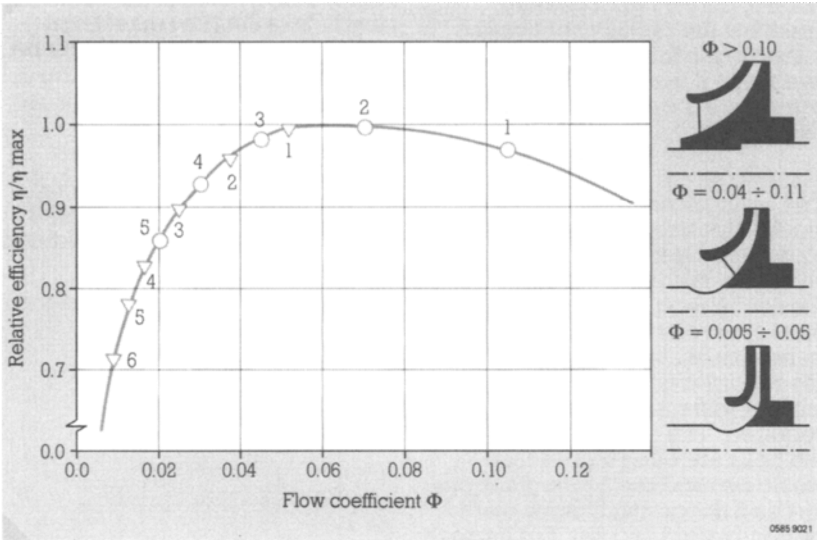


Figure 12-46I, Part 2. The graph shows the influence of the flow coefficient, Φ, on the efficiency. 1,2,3 . . . are the individual impellers or stages and their efficiencies in a multistage compressor designed for optimum speed and overall efficiency (O,O,O) and alternatively, for reduced speed (∇, ∇, ∇). (Used by permission: Bul. 27.24.10.40-Bhi50. Sulzer Turbo Ltd.)

- Φ Dimensionless flow coefficient
- V Volume flow at inlet (m^3/s, acfm)
- u Circumferential or tip speed at impeller outlet (m/s, fps)
- H_p Polytropic head (J/kg, Btu/lb)
- H_e Effective head (J/kg, Btu/lb)
- μ Impeller polytropic head coefficient
- n Mechanical speed (rpm)
- η_p Polytropic efficiency
- D Impeller outlet diameter (m, ft)
- z Number of impellers

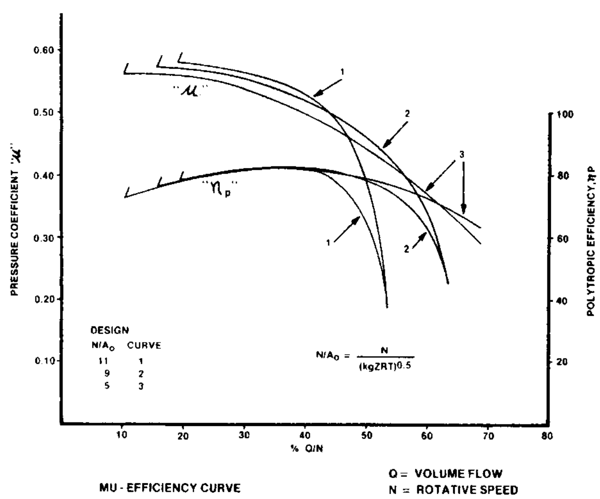


Figure 12-46I, Part 3. Stage performance of a compressor is usually represented in a pressure coefficient, μ or Mμ, and efficiency, η, versus Q/N (capacity vs. speed). A given impeller stage design will have a different characteristic depending on the relationship of its operating speed to the inlet sonic velocity of the gas. For higher ratios of speed to sonic velocity, N/A_o, the head or pressure coefficient curve will be steeper at flows higher than the design. (Used by permission: Bul. 423, ©1992. Dresser-Rand Company.)

Stage performance of a compressor is usually represented in a pressure coefficient (Mu) and efficiency (Eta) vs. Q/N (Capacity vs. Speed)

A given impeller stage design will have a different characteristic depending on the relationship of its operating speed to the inlet sonic velocity of the gas. For higher ratios of speed to sonic velocity (N/A_O), the head or pressure coefficient curve will be steeper at flows higher than design.

and flow geometry. For example, if a compressor is scaled to twice its size and the tip speed remains constant, it will produce the same head but will handle four times the flow of the original size compressor.[85] It can be seen that wheel/impeller/rotor tip speed rather than rotating shaft speed (rpm) is a most useful design tool.[85]

The "pressure coefficient" or "head coefficient" is dimensionless and is expressed:

$$\Psi = H_{ad}\, g/v^2, \text{ or } = \mu \qquad (12\text{-}83)$$

where $\Psi = \mu$ = pressure or head coefficient
g = gravity acceleration, ft/sec/sec
υ = rotor/wheel/impeller tip speed, ft/sec

Head for a single-stage, independent of machine size or rpm is

$$H_{ad} = \Psi \upsilon^2/g \quad (12\text{-}84)$$

The values for Ψ for best design for centrifugal compressors should be in the range of 0.50 ($\pm$ 0.04) for industrial compressor stages to 0.63 ($\pm$ 0.04) for radial and forward-swept blading.[85]

H_{ad} = adiabatic, ft-lb/lb = ft

Wheels are statically balanced individually; then the total assembly of wheels on the shaft is also statically and dynamically balanced. An internal balancing drum (see Figure 12-40E) is usually placed on the discharge end of the shaft to balance thrust loads.

Guide or Prerotation Vanes

Vanes are formed on the diaphragm assembly to direct the flow of gas into the eye of each succeeding impeller. The vanes are arranged to suit the wheel design and flow rates of the gas through the compressor. Figure 12-47 shows these interwheel vanes. Special designs allow for these to have the pitch manually adjusted during operation to properly balance the compressor operation.

When the vanes are placed on the inlet to the first stage wheel, they are usually of a variable pitch design. This allows for variation in compressor performance similar to that shown in Figure 12-62. The adjustable vanes vary the angle of the gas flow as it enters the first wheel, thereby changing the effective inlet gas pressure and/or volume without throttling losses (Figure 12-47). Figure 12-48A shows the inner side of the prerotation vanes as they would be directly adjacent to the first wheel. The control linkage is shown. Figure 12-48B illustrates the exterior view with control air motor in place. All compressors do not have nor need inlet guide vanes; however, it is advisable to discuss this with the manufacturer as his designs may perform better with the vanes. They are usually an extra cost item.

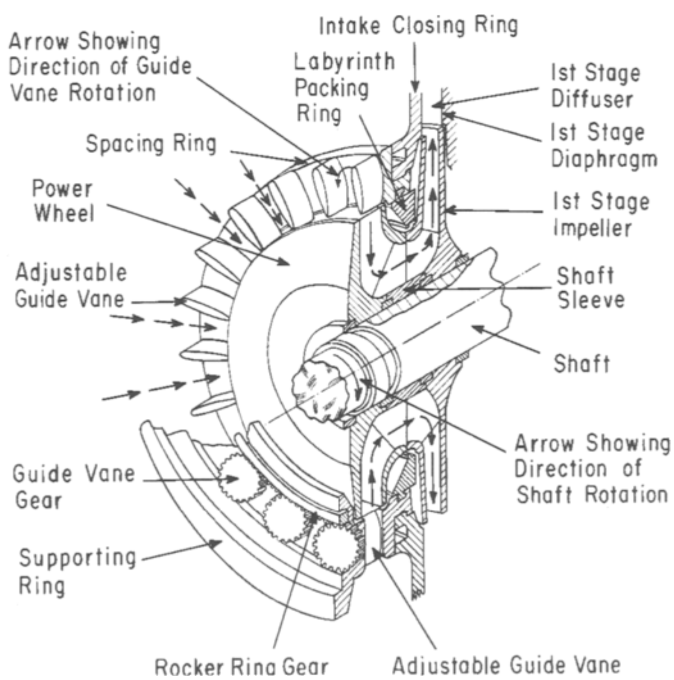


Figure 12-47. Gas flow through inlet guide vanes; power wheel shown ahead of first stage impeller. (Used by permission: Ingersoll-Rand Company. All rights reserved.)

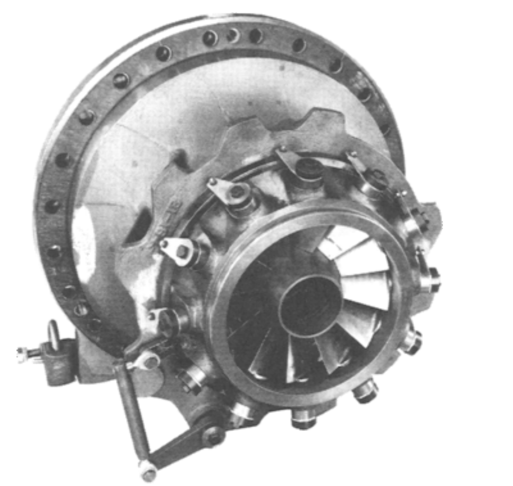


Figure 12-48A. Inlet prerotation vanes. (Used by permission: York International.)

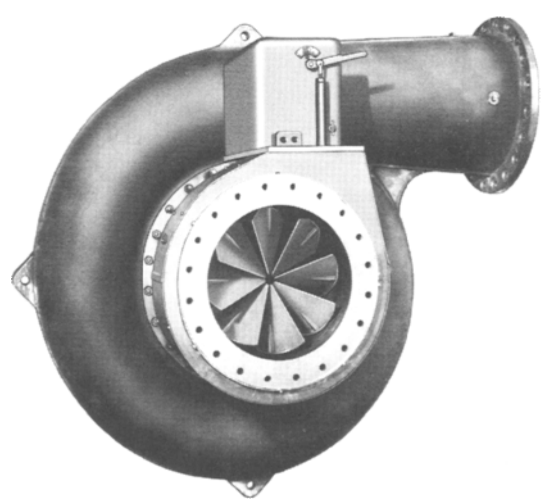


Figure 12-48B. Single-stage blower with automatically controlled inlet vanes. (Used by permission: A C Compressor Corporation.)

Power savings of 15% or more may result from the use of such vanes,[83] depending on the design and final operating point on the compressor's performance curves. The inlet guide vanes increase the turndown and, thus, increase the operating range.

Shaft

The shaft is a forging and may be designed as a stiff shaft or flexible shaft. A stiff shaft design means that the shaft will operate below any of its critical speeds. Usual practice limits design operation to 60% of the first critical speed. This requires a heavier shaft than the flexible design that allows the shaft to pass through its first critical speed at 40–60% of normal and maximum operating speeds.

Bearings

Shaft journal bearings for compressors operating in most process services and some air applications are located "outside" the case, rather than being an inside bearing. This is important for maintenance, as well as for reducing the problems in keeping oil out of the gas stream; although this problem still exists, but not to such a great extent. Thrust bearings of the Kingsbury type are one example of a good bearing for this equipment (Figures 12-49A and 12-49B). The double-acting bearing can absorb thrust loads in either direction.

Accessories

Oil coolers for bearing oil (and often sized for turbine oil requirements when turbine is used) must be a part of the system and are usually mounted on or in the compressor-driver base plate(s). This makes a compact arrangement but is sometimes so congested as to give poor maintenance conditions. The cooler, oil filter, and regulator can also be mounted adjacent to the unit. For some sizes, the compressor and driver can be mounted on a single base plate; however, this arrangement becomes bulky to ship and handle for large sizes. For these, as well as the smaller units, separate base plates can be used, but here extreme care in alignment and in foundation design is necessary to avoid trouble.

Usually, dual oil pumps are included, so that one pump failure will not shut down the compressor-driver unit. The first or main pump may be driven by electric motor, and the standby steam or gas may be driven by turbine. Any combination is acceptable as long as the selection takes into account the specific local conditions and service reliability. Figures 12-50A–C show an overall assembly, including accessories.

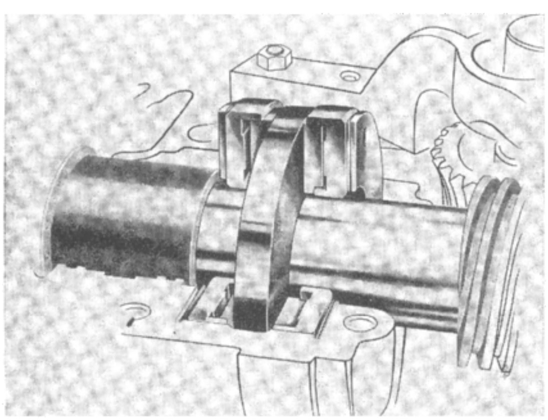


Figure 12-49A. Kingsbury-type thrust bearing. (Used by permission: Elliott® Company.)

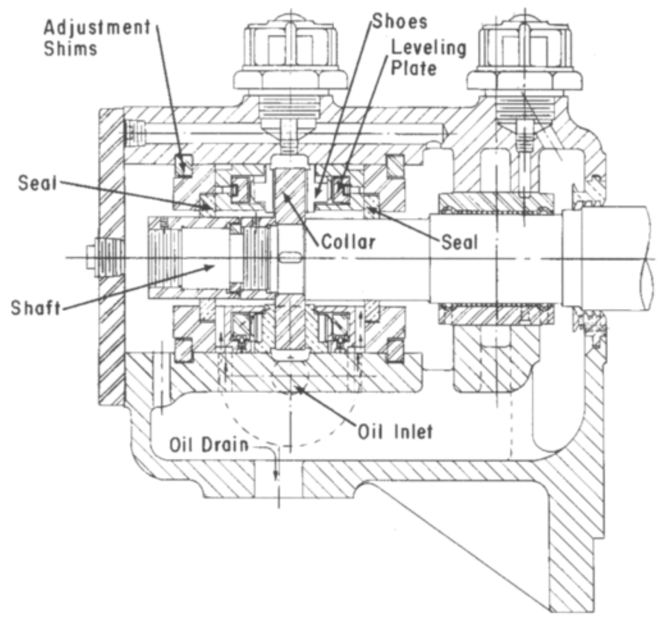


Figure 12-49B. Kingsbury-type thrust bearing for centrifugal compressor. (Used by permission: Kingsbury Machine Works, Inc.)

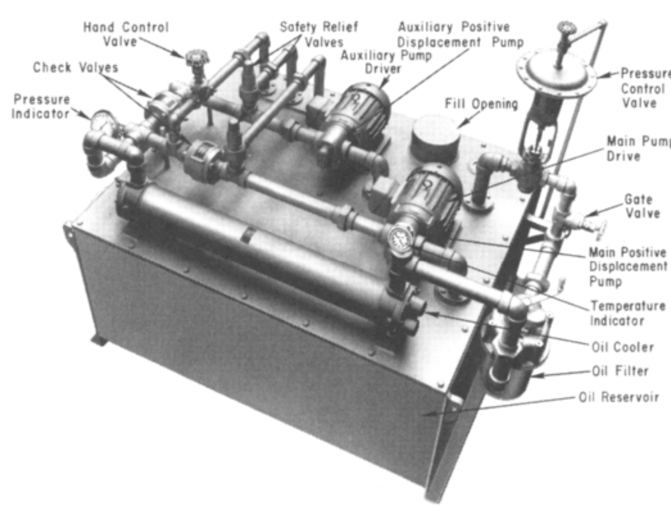
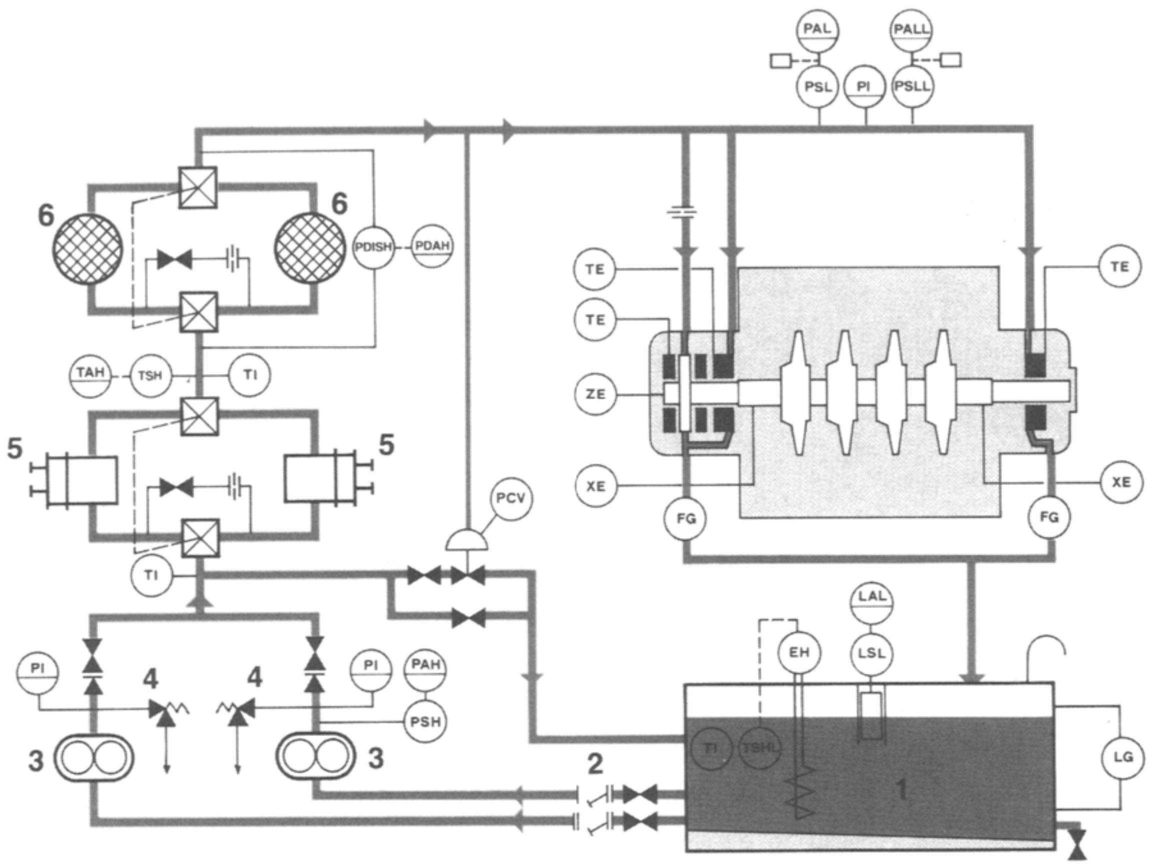


Figure 12-50A. Centrifugal compressor auxilliaries—forced feed lubrication system. (Used by permission: A C Compressor Corporation.)



A force feed lube oil system ensures that journal and thrust bearings are properly lubricated. The system is basically designed in accordance with API 614 and is suitable for continuous compressor operation. It generally comprises:

a) oil reservoir;
b) steam turbine or electric motor driven screw or centrifugal pumps;
c) full-flow oil coolers;
d) twin filter (5/10 micron) allowing cartridges to be changed during operation;
e) automatic by-pass valve to control oil pressure at the journal bearing manifold;
f) monitoring and safety instrumentation.

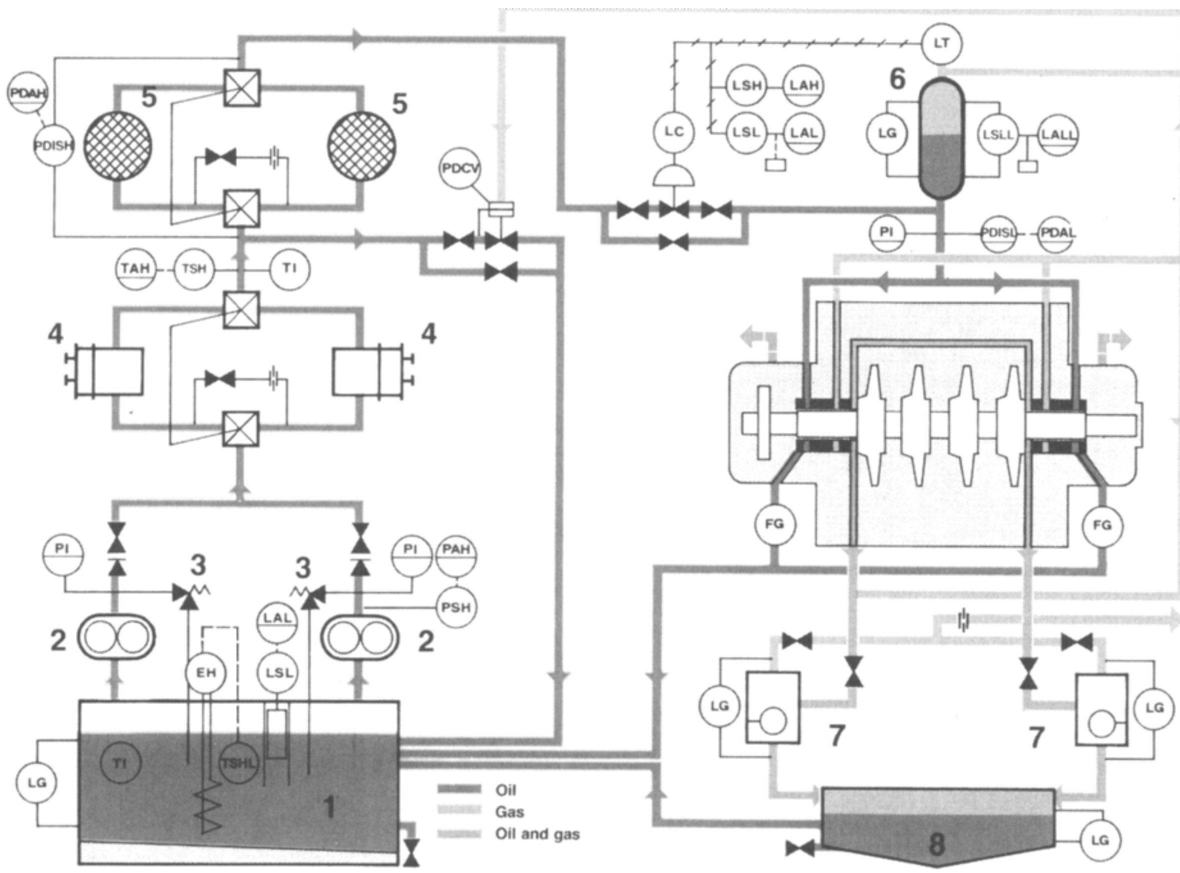
LEGENDA

1. Oil reservoir
2. Suction strainer
3. Oil pump
4. Relief valve
5. Oil cooler
6. Oil filter

○ - Locally mounted instrument
⊖ - Panel mounted instrument
LG - Level glass
LSL - Low level switch
LAL - Low level alarm
PI - Pressure indicator
PCV - Pressure control valve
PSL - Low pressure switch
PAH - Low pressure alarm and stand-by pump start-up

PSLL - Minimum pressure switch
PALL - Minimum pressure shut-down
PSH - High pressure switch
PAH - High pressure alarm
PDISH - High diff. pressure indicator and switch
PDAH - High diff. pressure alarm
TI - Temperature indicator
TE - Thermoelement
TSH - High temperature switch
TAH - High temperature alarm
TSHL - High/low temperature switch
ZE - Axial displacement probe
XE - Radial probe
FG - Flow glass

Figure 12-50B. A forced-feed lube system ensures that journal and thrust bearings are properly lubricated. The system is basically designed in accordance with API 614 and is suitable for continuous compressor operation. (Used by permission: Bul. PROM 526/I-5/95-II. ©Nuovo Pignone, S.p.A., Florence, Italy; New York; Los Angeles; and Houston, Texas. All rights reserved.)



The seal oil system supplies filtered oil to the liquid film rings or to mechanical type seals at the correct pressure and temperature. The system is basically designed in accordance with API 614 and is suitable for continuous compressor operation. It generally comprises:

a) oil reservoir;
b) steam turbine or motor driven screw or positive displacement pumps;
c) full-flow oil cooler;
d) twin filters (5-10 micron);
e) overhead tank with level or pressure control;
f) automatic seal oil traps;
g) monitoring and safety instrumentation.

LEGENDA

1. Oil reservoir
2. Oil pump
3. Relief valve
4. Oil cooler
5. Oil filter
6. Overhead seal oil tank
7. Seal oil trap
8. Drain pot

○ - Locally mounted instrument
⊖ - Panel mounted instrument
LG - Level glass
LSL - Low level switch
LAL - Low level alarm
LT - Level transmitter
LC - Level controller
LSH - High level switch
LAH - High level alarm
LSLL - Minimum level switch
LALL - Minimum level shut-down
PI - Pressure indicator
PSH - High pressure switch
PAH - High pressure alarm
PDISL - Diff. pressure indicator and switch
PDAL - Low diff. pressure alarm
PDISH - High diff. pressure indicator and switch
PDAH - High diff. pressure alarm
TI - Temperature indicator
TSH - High temperature switch
TAH - High temperature alarm
TSHL - High/low temperature switch
FG - Flow glass

Figure 12-50C. The seal oil system supplies filtered oil to the liquid film rings or to mechanical-type seals at the correct pressure and temperature. The system is basically designed in accordance with API 614 and is suitable for continuous compressor operation. (Used by permission: Bul. PROM 526/I-5/95-II. Nuovo Pignone, Florence, Italy; New York; Los Angeles; and Houston, Texas. All rights reserved.)

Shaft End Seals

The sealing of process gas along the rotating shaft is a delicate and important problem. Many factors enter into the selection of the type of mechanical seal best for the service.[35, 97]

1. Properties and the nature of gas in the case—corrosiveness, viscosity, abrasiveness, explosiveness, lubricity, temperature, and pressure.
2. Rotational speed of shaft and peripheral speeds of seal.
3. Mechanical limitations—dimensions of space required versus space available, shaft deflection and whip, shaft end play, shaft diameter, and maintenance.
4. Miscellaneous factors—cost, allowable by-pass or out-leakage, allowable contamination of gas with air, inert gas, oil, and other fluid.

These factors are not necessarily all the guide-points in final seal selection, as individual conditions may be so special or unusual as to justify a special design or a compounding of the features of several "standard" designs. Only rarely will any seal be "perfect" for the service or job application. Many types and designs of seals are available, each to fill a certain need. Table 12-7 lists a few for general applications.

Figures 12-51–12-55 are summaries of the common types of shaft seals. For some seals it is preferable to connect the vent opening between the seals to an area of low pressure, such as an ejector system, in order to draw the leaking gases out of the seal, in preference to pressurizing the seal with contaminating oil, air, or other gas.

Figure 12-56 can be used to estimate process gas leakage across a single or tandem gas seal arrangement.

Materials of Construction

Tables 12-8 through 12-8F summarize the usual materials for the components of the compressor. In general these tables apply to medium pressure (125–400 psi) machines in noncorrosive hydrocarbon service (Table 12-8A). Other fluids may require the use of special steel, nonferrous parts, or nickel alloys.[97, 105, 109, 110, 111, 112, 113, 114]

Note that many of the materials are limited by temperature (high or low), and some processes cannot afford for high temperatures to develop inside the compressor due to the risk of explosion, auto ignition, surface reaction with metals, and polymer formation, which can restrict impeller and other flow passages. For a chlorine dry gas compressor, for example, Figure 12-42 shows the results of extensive high temperature formation of ferric chloride and the resulting internal fire that melted some metal parts. This was SAE 4140 steel and indicates the importance of temperature control. This unit operated at 7,050 rpm, was electric motor-gear driven, had an inlet dry gas of 12.5 psia at 41°F, discharged at 165 psia with two sets of interstage cooling (for example, see Figure 12-40C), and at time of failure the discharge temperature was 275–320°F. The temperature then jumped to more than 320°F (off the chart and beyond). Rehrig[105] discusses the selection of materials; also see Tables 12-8A–12-8F. Several manufacturers have developed corrosion resistant coatings to apply to driven compressor components and industrial gas turbine parts to provide protection against corrosion and fouling. One example is low pH wet chloride environments as seen in steam turbines.[121]

Specifications

Centrifugal compressors are not items that the process company or its engineers should attempt to design in great detail. Rather, it is more important that they be in a strong position to (a) specify what is needed for the process, (b) understand the manufacturer's recommendations, and (c) evaluate the recommended design and performance in the process situation. Also see reference 117.

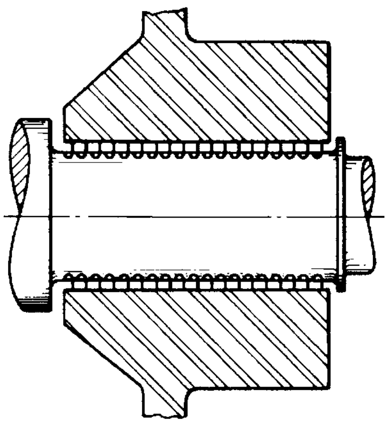
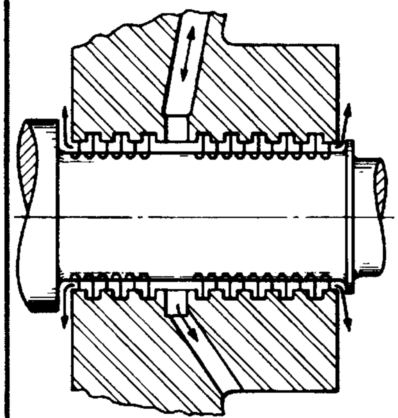
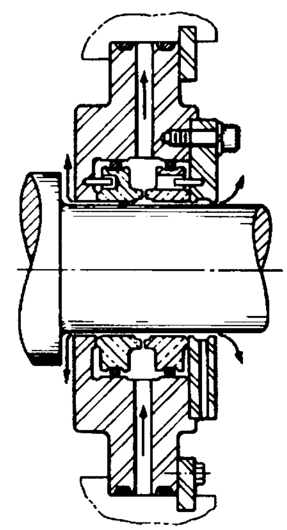
(Text continues on page 473)

Table 12-7
Index of Relative Leakages for Single-Sealing Elements

High Rating Indicates High Relative Leakage Past Seal

Seal	Out-Leakage Index**	Application	Figure No.
Straight-pass labyrinth	100	Low pressure, moderate temp.	12-51A, 12-51B
Staggered labyrinth, also stepped	56	Medium pressure, moderate temp.	12-52B, 12-52C
Restrictive ring, carbon ring	20	High temp. (700°F), medium pressure	12-51, 12-53
Mechanical wet-contact seal	0	High pressure, high speeds	12-54
Liquid film seal	0	High pressure, high speeds	12-55
Mechanical contact (running dry)*	2		

**Used by permission: Koch, D. T. "Centrifugal Compressor Shaft Seals," ref. 35, Jan. 3, 1958. ©Cooper-Cameron Corp. *Reprinted with permission: Nelson, W. E. *Hydrocarbon Processing*, V. 5, No. 2, p. 91[97], ©1977. Gulf Publishing Co., Houston, Texas. All rights reserved.

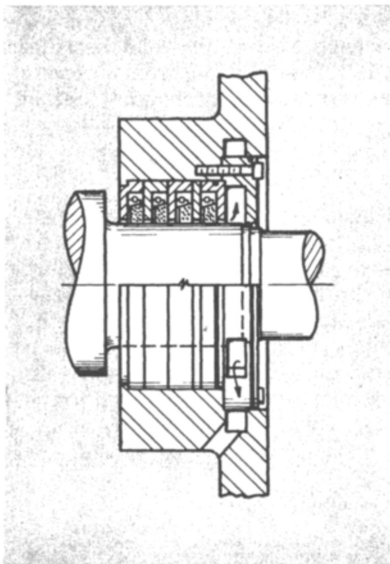
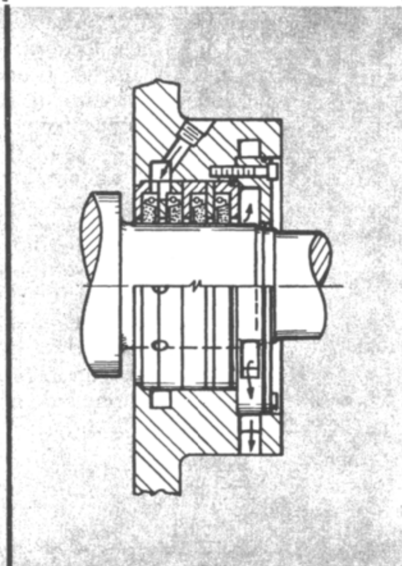
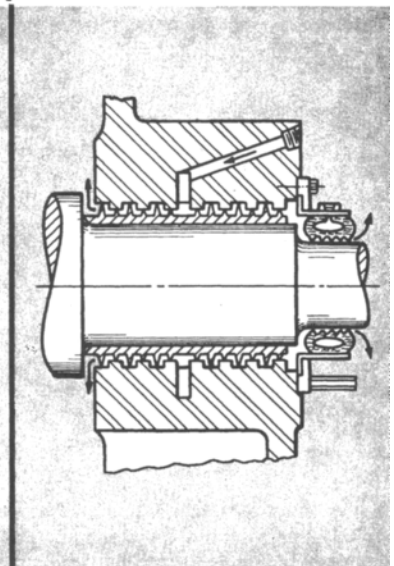
	LABYRINTH	BUFFERED LABYRINTH	LIQUID BUFFERED BUSHING
			
Suitable For Adaptation to:	All multi-stage compressors	All multi-stage compressors	All multi-stage compressors
Buffering Media	None	Clean, dry, air or inert gas.②	Water or oil.③
Material①	Stainless steel	Stainless steel.	Bronze in stainless steel housing.
Sealing Ability	Compressor duty—Moderate leakage of process gas. Exhauster duty—moderate in-leakage of ambient air.	No leakage of process gas. Moderate leakage of buffer gas to process and/or ambient.	No leakage of process gas to ambient. Leakage toward gas stream is trapped in baffle to prevent contamination of gas in compressor casing. This liquid may be recycled if not contaminated. Out-leakage of liquid is recycled.
Buffering System	None	Buffering media supplied directly from customer's source. Filter or pressure regulator may be required. In cases where inlet pressure is below atmosphere and in-leakage of ambient is undesirable, buffering media may be process gas from compressor discharge.	Separate pressure system similar to lubrication system including pumps, cooler, filters, etc., or if buffering media is oil, sealing system can be combined with bearing lubrication system.
Applications and Limitations	Generally used in air applications or applications where moderate leakage is tolerable. Normally not used in high pressure applications.	Applications where positive sealing is desirable and an inexpensive buffering media is available. Normally not used in high pressure applications. Only gases may be used for buffering.	Applications where positive sealing is necessary. No in-leakage of ambient air and no out-leakage of process gas. Specially suited for higher pressure applications.

Notes:

① Other materials available for special applications—consult factory.

② Process gas may be used as buffering media, if clean and dry, to avoid in-leakage of ambient for exhauster duty.

Figure 12-51A. Multistage centrifugal compressors shaft seal arrangements. (Used by permission: A C Compressor Corporation.)

CARBON RING	BUFFERED CARBON RING	COKE OVEN SEAL
		
All multi-stage compressors	All multi-stage compressors	Coke oven exhausters and boosters
None	Clean, dry, air or inert gas.②	If buffering is required, use steam process gas, water or inert gas. (Usually not necessary.)
Carbon rings in stainless steel housing.	Carbon rings in stainless steel housing.	Stainless steel.
Compressor duty — small leakage of process gas. Exhauster duty — small in-leakage of ambient air.	No leakage of process gas. Small leakage of buffer gas to process and/or ambient.	Compressor duty — small leakage of process gas. Exhauster duty — small in-leakage of ambient air.
None	Buffering media supplied directly from customer's source. Filter, pressure regulator, etc., may be required. In cases where inlet pressure is below atmosphere and in-leakage of ambient is undesirable, buffering media may be process gas from compressor discharge.	Buffering media supplied from customer's source.
Applications where buffering media is not readily available and only small leakage is tolerable. Not suitable for very high speed or very high pressure applications.	Applications where positive sealing is desirable and only small amount of buffering gas leakage tolerable. Not suitable for very high speed or very high pressure application. Only gases may be used for buffering.	Applications where positive sealing is desirable and an inexpensive buffering media is available. Normally not used in high pressure applications. Only gases may be used for buffering.

Notes:

① Other materials available for special applications—consult factory.

② Process gas may be used as buffering media, if clean and dry, to avoid in-leakage of ambient for exhauster duty.

③ Other liquids may be used if suitable—consult factory.

Figure 12-51A. (Continued)

Shaft Seal Options. The dry face seal for optimum sealing. *The Atlas Copco overhung impeller design is ideally suited to the use of a dynamic dry face seal — a proven technology which eliminates the need for an oil-film seal and expensive support systems. The dry face seal operates on the principle of a hydrostatic and hydrodynamic force balance. Spiral grooves on a rotating ring force process gas inward between the seal faces. The gas flow is restricted by a sealing dam, increasing the gas pressure on the outer portion of the faces. The pressure provides the opening force which allows the seal to work at minimum clearance and with minimum leakage. The dry face seal is recommended when leakage can be hazardous and/or costly.*

Labyrinth seals *are available for low-pressure air applications or when leakage of process gas into the atmosphere can be tolerated.*

Buffered labyrinth seals *permit injection of buffer gas between the labyrinths to help prevent leakage of process gas.*

Buffered carbon ring seals *can be used in moderate pressure applications. They can be operated dry, buffered with gas, or buffered with liquid.*

Oil bushing seals or wet mechanical seals *are also available.*

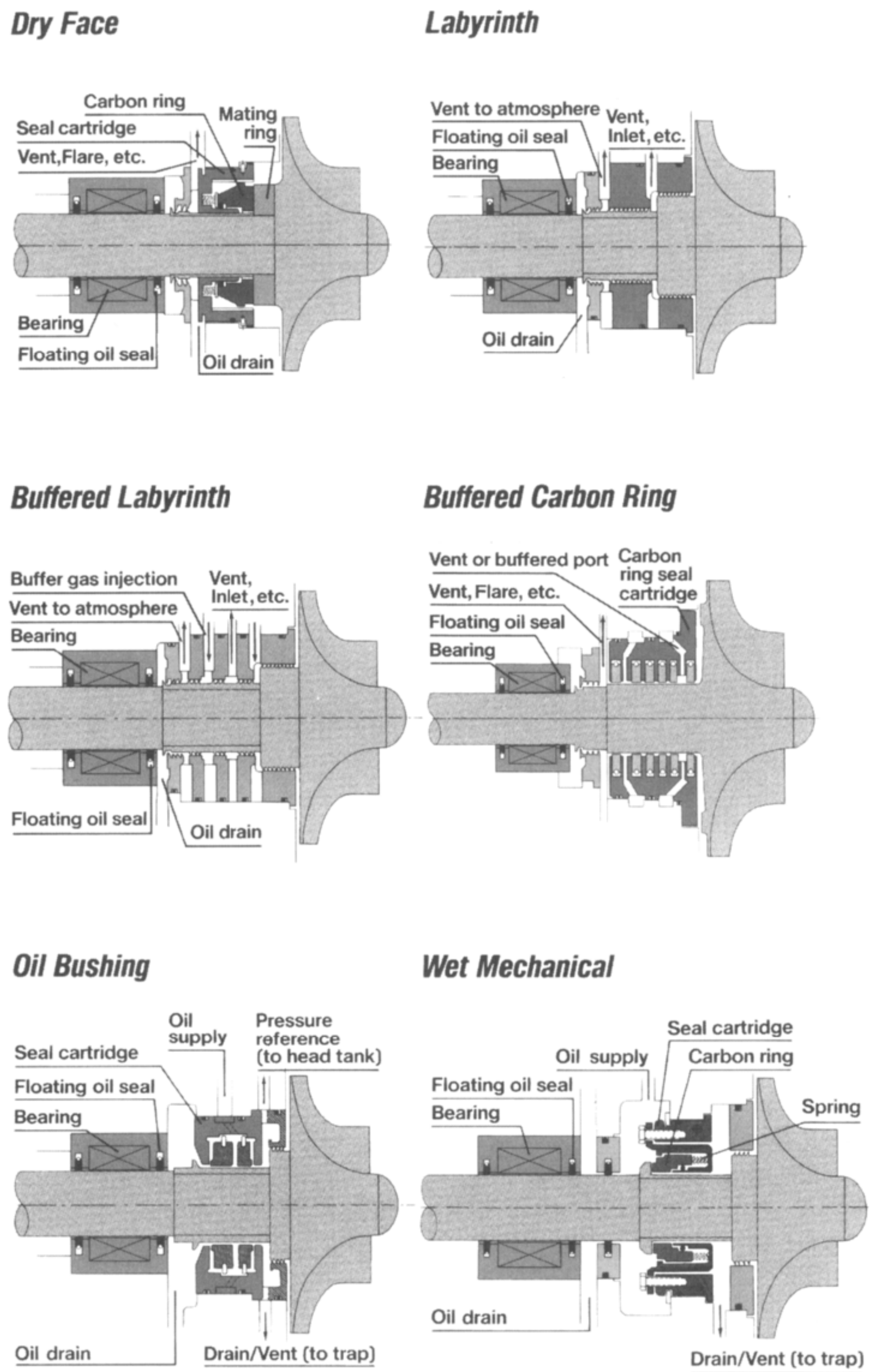


Figure 12-51B. Shaft seal options. (Used by permission: Bul. 2781005301, ©1988. Atlas Copco ACT.)

(Text continued from page 470)

To do these things, the process engineer must establish the function of the compressor; its capacities under conditions of normal, maximum, and minimum load; the acceptable materials of construction for the parts exposed to process fluids; and the importance and effect on performance of various fluid seals. In addition to the important process-centered specifications, the layout and general

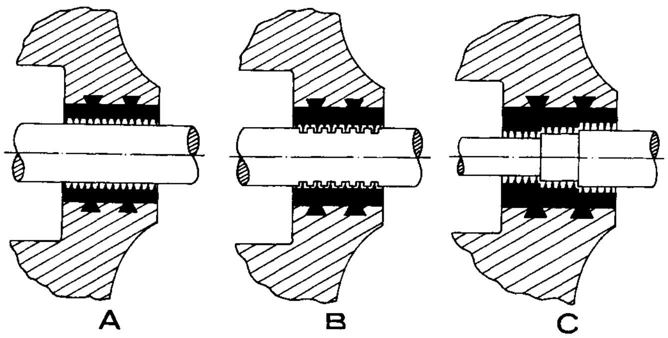


Figure 12-52A, B, C. A) Straight-pass labyrinth seal; B) staggered labyrinth seal; C) stepped labyrinth seal. Note: Materials usually are aluminum, bronze, babbit, other soft material, or steel. (Used by permission: Cooper-Cameron Corporation.)

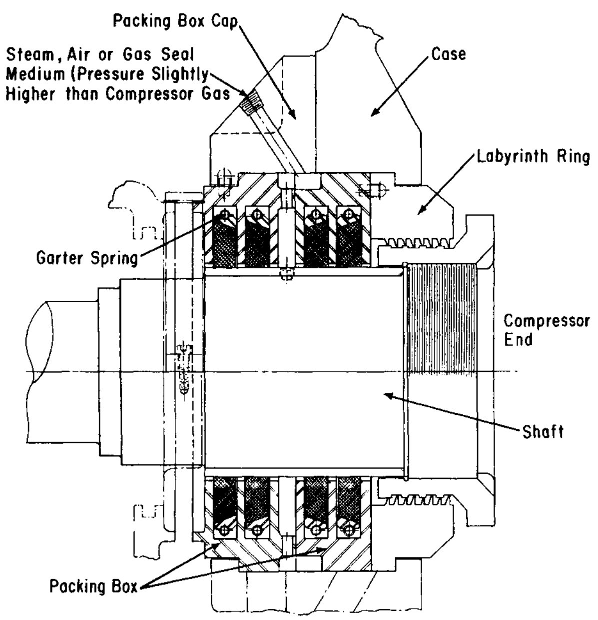
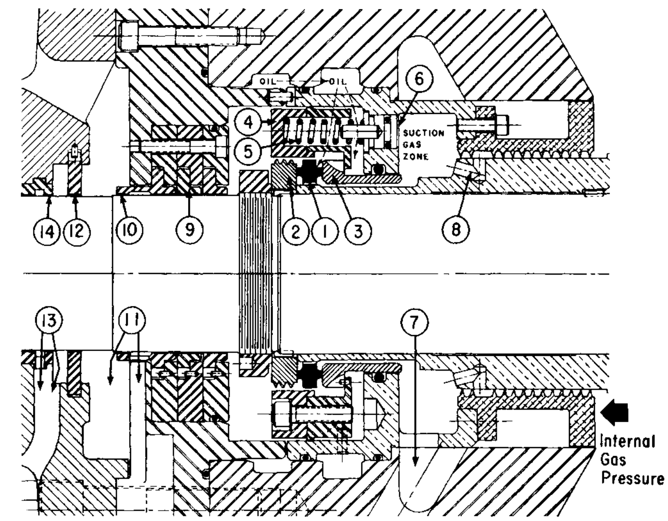


Figure 12-53. Packing box arrangement. (Used and adapted by permission: Demag Delaval Turbomachinery Corporation.)

service conditions should be established for evaluation purposes.

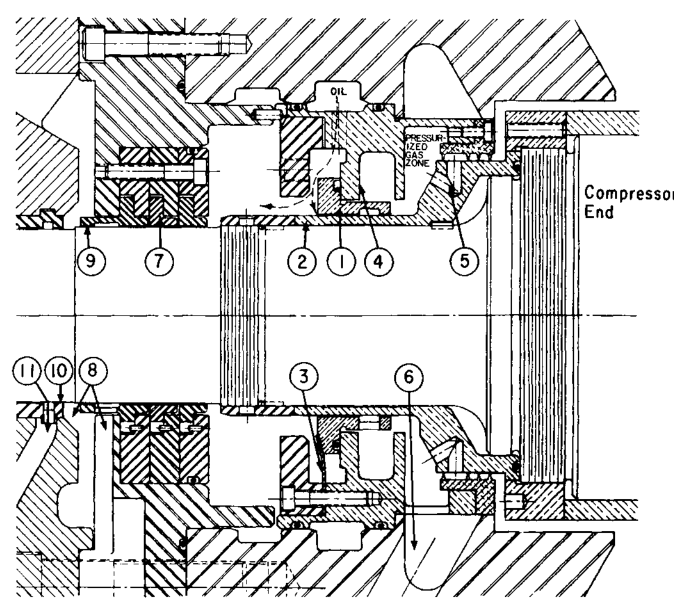
In order to establish the initial inquiry, a specification sheet similar to Figure 12-57 can be used. API Standard 617 also has a comprehensive data form and excellent performance and mechanical standards.[4] In general all of the information on these sheets need not be given to the manufacturer, just the basic performance data and known specifications. The manufacturer is expected to complete the information as it applies to his equipment. A summary of the essential information is as follows:

1. Flow rate and suction conditions.
   a. Pressure psia } at inlet suction flange
   b. Temperature, °F }



1. Rotating carbon ring
2. Rotating seal ring
3. Stationary sleeve
4. Spring retainer
5. Spring
6. Shutdown seal piston
7. Gas and contaminated oil drain
8. Centrifugal oil separator
9. Floating babbit-faced steel ring
10. Seal wiper ring
11. Seal oil drain line
12. Secondary wiper ring
13. Bearing oil drain line
14. Bearing wiper ring

Figure 12-54. Mechanical wet contact type seal. (Used by permission: Elliott® Company.)



1. Stationary seal sleeve
2. Rotating shaft sleeve
3. Spring
4. Seal casing partition
5. Centrifugal oil separator
6. Gas and contaminated oil drain
7. Floating babbitt-faced steel ring
8. Seal oil drain line
9. Wiper ring
10. Bearing wiper ring
11. Bearing oil drain line

Figure 12-55. Liquid film seal. (Used by permission: Elliott® Company.)

This chart, based on nitrogen gas, is used to estimate process gas leakage across a single or tandem gas seal arrangement. For a single or tandem seal, the pressure drop on the abscissa represents the difference between suction pressure and atmospheric pressure. The chart can also be used to estimate buffer gas leakage across the faces of a double gas seal arrangement. Since the top seal is exposed to the full pressure drop from buffer gas pressure to vent pressure, the single seal curve can be used to estimate leakage across the top seal. The lower seal is exposed to a pressure differential represented by the difference between buffer gas pressure and suction pressure. The single seal curve can also be used to determine the buffer leakage rate across the lower seal into the process gas.

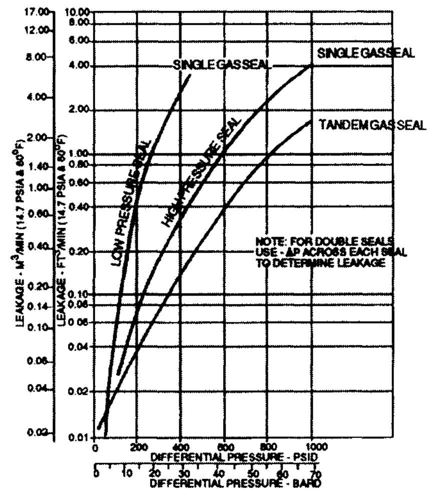


Figure 12-56. Estimates of process gas seal leakage using nitrogen as sealing gas for single- and tandem-gas seal arrangements. (Used by permission: Sundstrand Compressors Bul. 450 April 1995. ©Sundstrand Fluid Handling Corporation.)

Table 12-8A
General Material Specifications for Noncorrosive Applications

Also See Tables 12-8B–F

Part	Material
Casing (low pressure)	Cast semi-steel or cast steel
(high pressure)	Cast steel or forged steel
Shaft	Carbon steel (AISI-C1045), 18-8 stainless, or alloy steel forging AISI 4340.
Impeller[a] (discs, covers, blades)	Forging: SAE 1040, 1045, ASTM A-294 B-4, 18-8 stainless or AISI 4130, 316 SS.
Rivets	Forged AISI Type 410, or as previously listed.
Diaphragms (uncooled)	Cast iron, ASTM-A48-Cl 30
(cooled)	Cast iron, ASTM-A48-Cl 30
Inlet guide vanes	Cast iron, ASTM-A48-Cl 30
Shaft sleeves	Steel AISI-1010, or alloy steel, 316 SS
Labyrinths (internal)	Aluminum, lead (ASTM-B-23 gr. 8 high lead), bronze
(shaft)	Aluminum, lead (ASTM-B-23 gr. 8 high lead), bronze
Seals,[b] rotating face	Bronze, carbon as required, tungsten carbide
Mechanical seals	316-Carbon
Bearings (journal, precision faced thrust)	Steel-backed, babbitt-faced, ASTM B-23 gr. 3 high tin as recommended by manufacturer.
Thrust balancing disc	Steel, AISI-1023, or ASTM-A-294 gr. B forging.

[a]For tip speed of 1,100 fps — Titanium

[b]For high pressure — 17-4 Ph SS

Table 12-8B.
Cast Casing Materials for Low-Temperature Applications

Cast Casing Material	Commercial Designation	Minimum Temperature Limits—°F (°C)
Steel	ASTM A352 gr. LCB (0% nickel)	− 50 (− 46)
Steel	ASTM A352 gr. LC2 (2–3% nickel)	−100 (− 73)
Steel	ASTM A352 gr. LC3 (3–4% nickel)	−150 (−101)
Steel	ASTM A352 gr. LC4 (4–5% nickel	−175 (−115)
Stainless steel	ASTM A743 gr. CF3, CF8, CF3M, CF8M	−320 (−196)
Stainless steel	ASTM A351 gr. CF3, CF8, CF3M, CF8M	−320 (−196)

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Table 12-8C
Welded Casing Material for Low-Temperature Applications

Welded Casing Material	Commercial Designation	Minimum Temperature Limits—°F (°C)
Steel	ASTM A516 gr. 55	− 50 (− 46)
Steel	ASTM A537	− 75 (− 59)
Steel	ASTM A203 gr. A, B	− 75 (− 59)
Steel	ASTM A203 gr. D, E	−160 (−107)
Steel	ASTM A553 Types I, II	−275 (−171)
Stainless steel	ASTM A240 Types 304, 304L, 316, 316L, 321	−320 (−196)
Steel	ASTM A353	−320 (−196)

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Table 12-8D
Impeller Material for Low-Temperature Applications

Impeller Material	Commercial Designation	Minimum Temperature Limits—°F (°C)
Titanium	ASTM B367 gr. C3, C4	− 50 (− 46)
Steel	AISI 3140	− 50 (− 46)
Stainless steel	ASTM A744/351 gr. CA6NM	− 50 (− 46)
Stainless steel	ASTM A747 gr. CB7CU-1, CB7CU-2	−150 (−101)
Steel	AISI 4320-4345	−175 (−115)
Steel	ASTM A543	−175 (−115)
Monel K500	AMS-4676	−175 (−115)
8% Nickel steel	ASTM A522 Type II	−275 (−171)
Stainless steel	ASTM A743/351 gr. CF3, CF3M, CF8, CF8M	−320 (−196)
Stainless steel	ASTM A473 Type 304, 304L, 316, 316L	−320 (−196)
9% Nickel steel	ASTM A522 Type I	−320 (−196)

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Table 12-8E
Materials for Usual Construction of Components for Process Type Gas Applications

	Material Recommendation				
Gas	**Casing**	**Impeller**	**Diaphragm**	**Shaft**	**Notes**
Acetic acid	Titanium	Titanium	Titanium	C.S.	
Ammonia	316 S.S.	Inconel	316 S.S.	C.S.	
Wet carbon dioxide	316 S.S.	17-4PH	316 S.S.	C.S.	
Wet chlorine	Titanium	Titanium	Titanium	C.S.	
Cyanogen chloride	Hastelloy	Hastelloy	Hastelloy	C.S.	
Dry hydrogen chloride	C.S.	Inconel	C.S.	C.S.	Temp. ≤ 400°F
Wet hydrogen chloride	Hastelloy	Hastelloy	Hastelloy	C.S.	
Hydrogen fluoride	316 S.S.	316 S.S.	316 S.S.	C.S.	Temp. ≤ 480°F
Dry hydrogen sulfide	C.S.	17-4PH	316 S.S.	C.S.	
Wet hydrogen sulfide	316 S.S.	Titanium	316 S.S.	C.S.	
Nitric acid	316 S.S.	316 S.S.	316 S.S.	C.S.	
Dry phosgene	C.S.	Inconel	C.S.	C.S.	
Wet phosgene	Hastelloy	Hastelloy	Hastelloy	C.S.	
Polyvinyl chloride monomer	C.S.	Titanium	C.S.	C.S.	
Sulfuric acid	316 S.S.	316 S.S.	316 S.S.	C.S.	
Wet sulfur dioxide	316 S.S.	316 S.S.	316 S.S.	C.S.	

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Table 12-8F
Typical Materials of Construction for Selected Centrifugal Applications

Service	Air	Refrigeration at −175°F	Hydrogen Reforming	Wet Gas	Toxic and Corrosive Gases	Coke Oven
*CASING	Cast iron	Cast Ni-steel	Forged steel	Cast iron or cast steel	Cast iron, cast steel, or cast stainless steel	Cast iron
IMPELLERS	Alloy steel	Alloy steel or nickel steel	Alloy steel	Alloy steel or stainless steel	Alloy steel or stainless steel	Alloy steel or stainless steel
SHAFT	Alloy steel	Nickel steel	Alloy steel	Alloy steel	Alloy steel or stainless steel	Carbon steel
IMPELLER SPACERS	Carbon steel	Carbon steel or nickel steel	Carbon steel	Carbon steel or stainless steel	Carbon steel or stainless steel	Carbon steel or stainless steel
BALANCING DRUM	Carbon steel	Carbon steel or nickel steel	Carbon steel	Carbon steel	Carbon steel or stainless steel	Alloy steel or stainless steel
INTERSTAGE LABYRINTHS	Aluminum	Aluminum	Aluminum	Aluminum	Aluminum, bronze, teflon or stainless steel	Aluminum or stainless steel
DIAPHRAGMS AND GUIDE VANES	Cast iron	Cast iron or cast Ni-iron	Cast iron	Cast iron	Cast iron	Cast iron
SHAFT SEAL TYPE	Labyrinth with bleed-down for higher pressures	Mechanical oil with automatic positive shut-off seal	Mechanical oil or liquid film	Mechanical oil with sweet gas injection or labyrinth with gas ejectors	Mechanical oil or labyrinth with sweet gas injection or dry carbon rings	Labyrinth with steam injection and automatic shut-off device

*Nodular ASTM A-338 casings may be furnished where service permits.

Used by permission: Bul. P-11A, ©1966. Elliott® Company.

c. Lb per hour, or volume, cfm (state whether dry or conditions relative to liquid saturation).
d. Gas composition at suction conditions.
   (1) Molecular weight.
   (2) Isentropic exponent (if known).
   (3) Compressibility factor (if known, or necessary).
e. Barometric pressure, psia or mm Hg (at installation site).
f. Request performance curves for design, 90%, 80%, 70%, 60%, and 50% of rated speed.

2. Discharge conditions.

a. Pressure, psia.
b. Temperature (if limitation exists; otherwise manufacturer will be controlled by his experience and the effect of temperature on materials of construction. Request temperature at rated pressure but 90–50% of rated volume).

3. Cooling water

a. Temperature in summer, winter, and *design* average.
b. Type (cooling tower, fresh, salt, etc.).

4. Compressor construction details.

a. Type and location of inlet and outlet connections on case.
b. Type case (horizontally split, vertically split).
c. Cooling—diaphragm cooled, external intercoolers, liquid injection in case (this should be selected only on recommendation of manufacturer).
d. Minimum case design pressure. Request case test pressure.
e. Materials of construction:
   (1) Case
   (2) Shaft
   (3) Diaphragms
   (4) Impeller or rotor
   (5) Labyrinth seals
f. Overspeed rpm for each wheel, operating tip speed.
g. Critical shaft speed, rpm.
h. Noise level in decibels.

5. Shaft seals and packing. Request detailed drawings of oil seals, shaft seals, and any purge gas or oil details. Request guaranteed seal air, gas, or oil leakage rates at a given buffering media pressure.
6. Driver (See the appropriate specifications in the chapter on drivers.)

a. Steam turbine (give steam conditions). Request performance curves at varying speeds.
b. Electric motor (give power conditions).
c. Gas engine (give gas conditions).
d. Others (belt, gears, etc.).
e. Controls for speed.

SPEC. DWG. NO.
A-

Page ___ of ___ Pages	
Unit Price	
No. Units	
Item No.	

Job No. ____________________

____

B/M No. ____________________

CENTRIFUGAL COMPRESSOR SPECIFICATIONS

Service ____________ Manufacturer ____________

Type ____________ Model ____________ Size ____________

No. Impellers ________ BHP ________ @ ________ RPM ________ Speed Range ________ RPM

OPERATING CONDITIONS PER ________

Gas (Dry) ________ Sat'd. with ________ Avg. Mol. Wt. ________ "K" Value ________

Suction: Pres. PSIA. Normal ________ Guarantee ________ Max. ________

Temp. °F. Normal ________ Guarantee ________ Max. ________

Discharge: Press. PSIA Normal ________ Guarantee ________ Max. ________

Temp. °F. Normal ________ Guarantee ________ Max. ________

Sidestream: Press. PSIA Normal ________ Guarantee ________ Max. ________

Temp. °F. Normal ________ Guarantee ________ Max. ________

C.F.M. @ Suction Conditions ________ Wt. Flow ________ Lbs./Hr.

C.F.M. @ Sidestream Conditions ________ Wt. Flow ________ Lbs./Hr.

Compressibility Factors Suction ________ Discharge ________

Surge ________ % Capacity Surge Pt. ________ First Critical RPM ________

ΔP Intercoolers ________ PSI

COMPRESSOR DETAILS

Type Impeller ________ Overspeed ________ RPM Rated ________ RPM

Intake Flange Size ________ ASA ________ Lbs. Facing ________ Disch. Flg. Size ________ ASA ________ Facing ________

Sidestream Flange Size ________ ASA Rating ________ Lbs. Facing ________ Casing Test Pressure ________ PSIG

Bearings: Journal - Babbited Sleeve ________ Thrust ________ Make ________

Coupling: Make ________ Class ________ Type ________ Intercooler By ________

Water Temp. ________ °F. Type ________ Tubes Mat'l. ________ Size ________

COMPRESSOR DRIVER

Horsepower ________ Type ________ Class ________ Volts ________ Phase ________ Cycles ________

Steam in ________ PSIA ________ Temp. ________ °F. Stm. Nozzle Control ________ °F. No. Hand Valves ________

Exhaust ________ PSIA ________ Temp. ________ °F. Condensing ________

Driver Spec. Sheet No. ________ Gear: Make ________ Rated BHP ________ Model ________ Red'n Ratio ________

COMPRESSOR MATERIAL

Case: ________ Shaft: ________ Sleeves: ________

Impellers: Hub & Cover ________ Blades ________ Diaphragms ________ Labyrinths ________

LUBRICATION SYSTEM

Main Oil Pump Driver ________ With Piping

Aux. Oil Pump Driver ________ Class ________ Volts ________ Phase ________ Cycles ________

Twin Oil Coolers ________ Twin Oil Filters ________ Bearing Temp. Ind. ________ Tube Mat'l. ________

Cooling Water: ________ @ ________ °F. & ________ PSIG. GPM Req'd. ________

SEALING SYSTEM

Type of Seals ________

REMARKS

By	Chk'd.	App.	Rev.	Rev.	Rev.
Date					

P.O. To: ________

Figure 12-57. Centrifugal compressor specifications.

Note: If driver economics are to be evaluated by the manufacturer, utility costs must be furnished.

7. Controls. Request diagram of shutdown and alarm for over- or under-pressure, over-speed, high bearing temperature, lube oil system.
8. Pressure lubricating system for compressor and driver bearings.

   a. Oil pumps driven from governor end of turbine shaft.
   b. Emergency oil pump (separate motor or turbine drive).
   c. Oil reservoir with connection for air purge.
   d. Oil cooler using water coolant at a specific temperature. Specify tube and shell materials if manufacturer's standard of admiralty, muntz metal, and steel (shell) is not acceptable. Specify water pressure.

9. Accessories usually include

   a. Main oil pressure gage.
   b. Bearing oil pressure gage.
   c. Temperature indicator at each bearing.
   d. Shaft coupling, flexible, spacer type.
   e. Coupling guard.
   f. Tachometer, vibrating reed, or electric.
   g. Baseplate, single or common for compressor and driver.
   h. Steam, oil and water piping directly associated with connecting the compressor-driver unit.
   i. One set special tools and wrenches for dismantling compressor and driver.
   j. Operating instructions.
   k. Dimensional drawings, including dimensions and details of all accessory equipment.

10. Additional details useful in most applications.

    a. Impeller: mach number at eye and at periphery.
    b. Maximum possible speed of compressor, also of driver.
    c. Maximum horsepower possible for driver to develop, with any changes necessary to bring up to this maximum (such as changing nozzles, nozzle ring of steam turbine, changing blades or buckets).
    d. Paint specifications for exterior of unit.
    e. Suitability of entire unit for outdoor erection (if required).

11. Statement of performance guarantee. Compressor performance at the rated or design point is normally guaranteed with an allowable variation of ± 4% on speed and horsepower.

12. Parts warranty.

Performance Characteristics

Figure 12-58 illustrates a pressure-capacity relationship for horizontally split centrifugal compressors (described earlier). Although specific ranges and pressure breaks vary between manufacturers, the chart shows a fairly representative range for multistage, horizontally split centrifugal applications[82] in standard case sizes or frames, either cast or welded (fabricated) case construction. From Figure 12-58 the capacity range extends through 360,000 actual cfm (suction conditions) for double-flow configurations, and single-flow cases would extend to 180,000 actual cfm. Figure 12-59 illustrates a special type of single-stage vertical shaft/horizontal impeller, high-speed centrifugal compressor used at speeds to 33,900 rpm and with case pressures of 1,000–2,160 psia working pressure. The usual motor drive uses a high-speed gear to drive the impeller. These are used in many light hydrocarbon and light gas applications.[84]

The fundamental characteristics of compression are the same for centrifugal and reciprocating compressors. The manner in which these fundamentals are interpreted must be adapted to the particular machine type and operating characteristics, and this accounts for the difference in design procedures.

The general operation of a centrifugal compressor is like a centrifugal pump, except that the fluid is compressible. Theoretically, the head developed by a centrifugal impeller or wheel is the same regardless of the characteristics of the

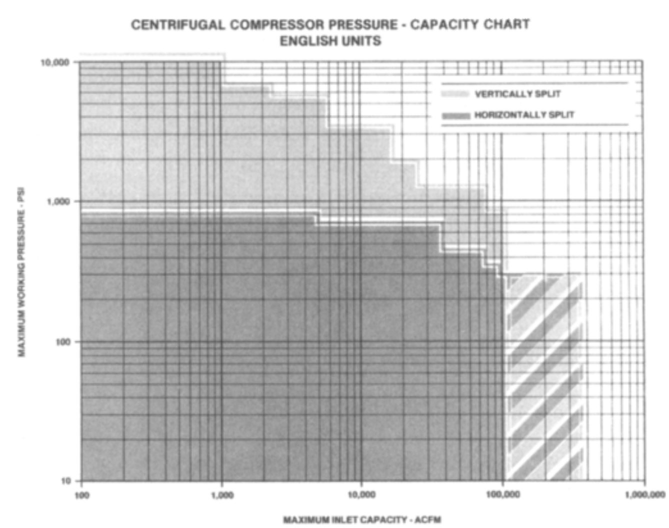

Figure 12-58. Generalized centrifugal compressor pressure-capacity chart for vertically and horizontally split cases. This chart is representative of the general ranges of most major manufcturers. As shown, the flow range extends through 360,000 cfm. This flow is generally achieved in centrifugal compressors using a double-flow configuration. A single-flow configuration would extend to 180,000 cfm. (Used by permission: Bul. 423. Dresser-Rand Company.)

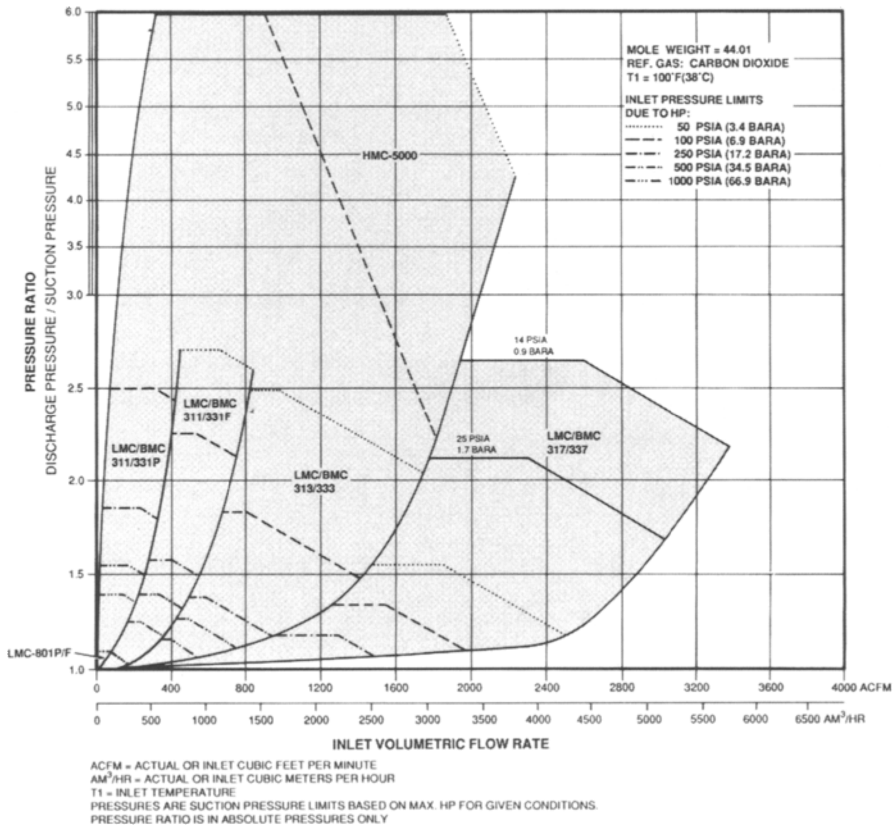

Figure 12-59. Performance ranges for special high-speed/high-pressure, single-stage Sundstrand Compressors (except model HMC 5000) centrifugal compressors. Special impellers are available for performance outside the envelopes shown. Performance varies with the gas involved. (Used by permission: Sundstrand Compressors Bul. 450, ©April, 1995. Sundstrand Fluid Handling Corporation.)

gas involved. This is more strictly true for single-stage wheels (units) than for the multistage machines. It is a satisfactory approximation for this latter condition when the changes in performance (from design) are not greater than about 20%. It is *important* to remember that the impeller wheel recognizes and acts only in terms of the number of *actual* ft^3 *per min* (or unit of time) going through, and not the number of lb of gas or mol of gas, or even standard ft^3 per min.

The impeller (wheel) of the centrifugal compressor imparts kinetic energy to the gas by increasing the gas velocity through the rotation of the impeller. A static pressure rise in the impeller comes from part of this energy, and the balance is converted to velocity head, which in turn converts to additional pressure rise in the compressor wheel assembly. (See Figures 12-43, 12-44B, 12-46I, and 12-47.)

The kinetic energy is a function of the square of the velocity; thus, the head produced by the rotating impeller is directly proportional to the square of the tip speed of the impeller. Then,[96]

$H_p \propto \upsilon/g_c$

where H_p = polytropic head, ft-lb/lb = ft

υ = impeller tip speed, ft/sec

g_c = gravitational constant, 32.2 ft-lb/lb-sec-sec = $(ft/sec)^2$

and, $H_p = \Psi \upsilon^2/32.2$ (12-85)

where Ψ = head or pressure coefficient, see earlier discussion.

Thus, Ψ is a function of any specific impeller. The variation of Ψ with flow defines the characteristic curve of the impeller.

n = k (ratio of specific heats, c_p/c_v) for an adiabatic, isentropic process

n = 0 for isobaric (constant pressure) processes

n = ∞ for isometric (constant volume) process

The terminology will not be repeated here unless the interpretation must be supplemented or modified. The design details will serve as an aid for estimating operating characteristics and not as a final basis for design. Neerken,[41] Lapina,[36] and Fullemann[85] are useful references.

Inlet Volume

The inlet or suction volume to a compressor can be determined in several ways, depending on the data available and the processing conditions. A common relationship is

Inlet or suction volume, $V_i = (w_1)(v_1)$ (12-86)

w_1 = flow, lb/min,

v_1 = specific volume at suction conditions, ft^3/lb

$$v_1 = \frac{ZRT_1}{144P_1} = \frac{Zr_iT_1}{144P_1} = \frac{Z(1545)(T_1)}{144(MW)(P_1)} \tag{12-87}$$

Z = compressibility factor, dimensionless
R = universal gas constant = 1,545 (ft)(lb force)/(lb-mol)(°R)
R_i = individual gas constant = R/mol wt of gas
MW = mol wt of gas
T_1 = inlet gas temperature, °R
P_1 = inlet suction gas pressure, psia

Compressor Piping

It is extremely important to have a careful (usually computer) analysis of the piping stresses and strains where the piping is attached to the compressor flanges. The API and ASME, as well as the compressor manufacturers, have some established limits as to what forces the flange-nozzle connections on a compressor can withstand. Excellent piping stress analysis computer programs exist to evaluate these connections and the stresses transmitted back to the compressor from other parts of the immediate piping system. See Reference 107 for one illustration, and for a more detailed discussion, see Reference 111. Large horsepower reciprocating compressors usually require the installation of pulsation surge drums or pulsation bottles and/or specially designed piping systems associated with the compressor; see Chapter 13 of this volume.

The compression of a gas as it passes through a single or multistage centrifugal machine is shown in Figure 12-60. The diagram may look similar to a reciprocating compression card; however, the return or expansion stroke never takes place in a centrifugal machine. As gas enters the machine at P_1V_0, the speed and corresponding volume increase until the steady state condition of P_1V_1 is reached. Compression, which has been taking place as the speed and volume increase, finally follows the "k" or polytropic "n" value line for the gas from P_1V_1 to P_2V_2. Gas is discharged continuously at P_2 as long as the steady-state suction condition is maintained. The compressor is now operating, and each increment of suction gas enters at the rate of V_1, keeping the machine in balance. The effect of changing operating conditions will be discussed later.

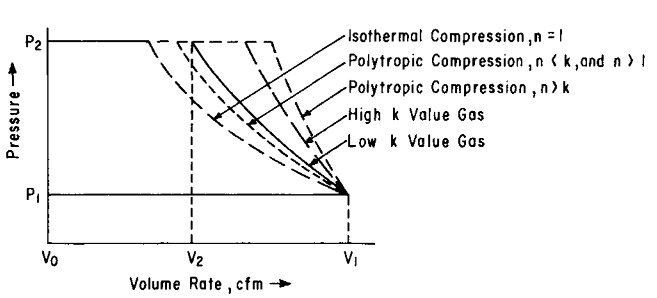

Figure 12-60. Compression in a centrifugal machine.

The actual performance curves of Figures 12-61 and 12-61A–E represent a typical presentation from a manufacturer. The rated design point is established at 100% of the rated speed; then other expected performances at lower or

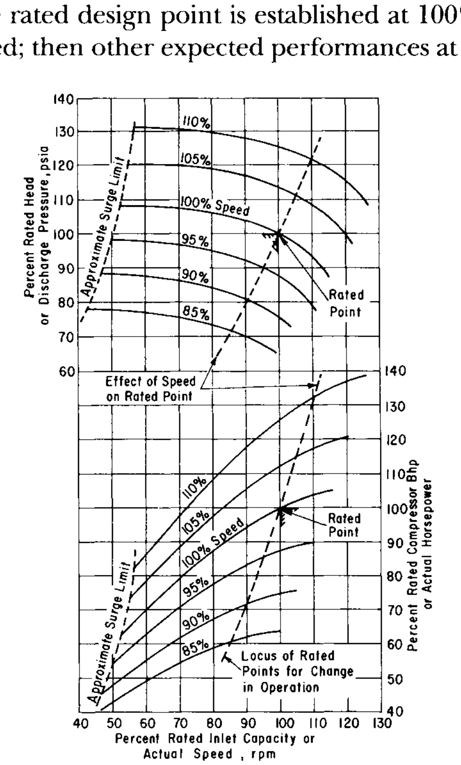

Figure 12-61. Manufacturer's typical centrifugal compressor characteristic curve.

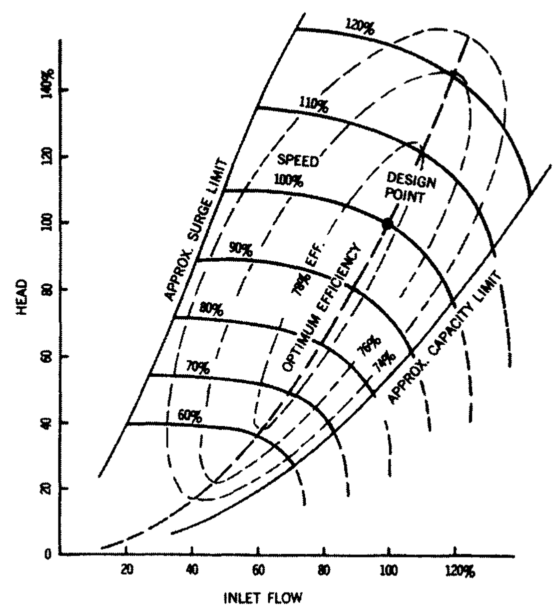

Figure 12-61A. Typical performance map of centrifugal compressor. (Used by permission: Fullermann, J. Report, "Centrifugal Compressors," ©1963. Cooper Energy Services, Cooper-Bessemer Rotating Products Div., Cooper-Cameron Corporation. Originally printed: *Advances in Petroleum Chemistry and Refining,* Interscience Publishers Div., John Wiley and Sons. No longer in publication. All rights reserved.)

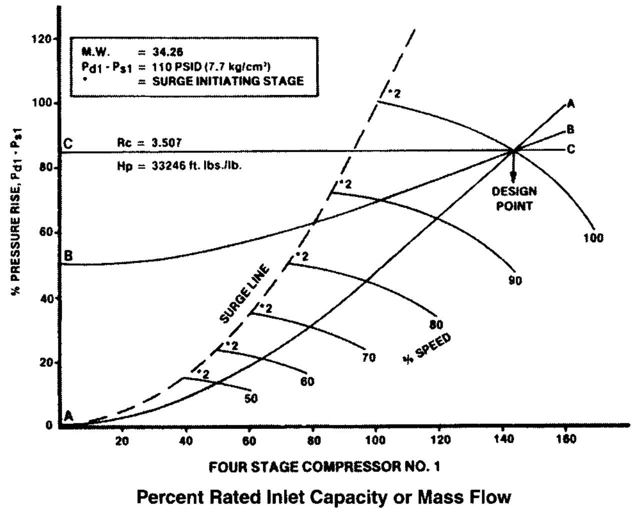

Figure 12-61B. Performance examination of one set of centrifugal compressor conditions. The pressure rise required by most applications exceeds the capability of a single stage; therefore, multistage compressors are used far more frequently. Multistage compression ratios within the range of 3:1 to 15:1 are typical, but the ratio of a single stage usually is less than 2.5:1. The surge line shape of a single stage can be predicted adequately using the fan laws, but this is usually not true for a multistage machine. (Used by permission: Bul. 423, ©1992. Dresser-Rand Company.)

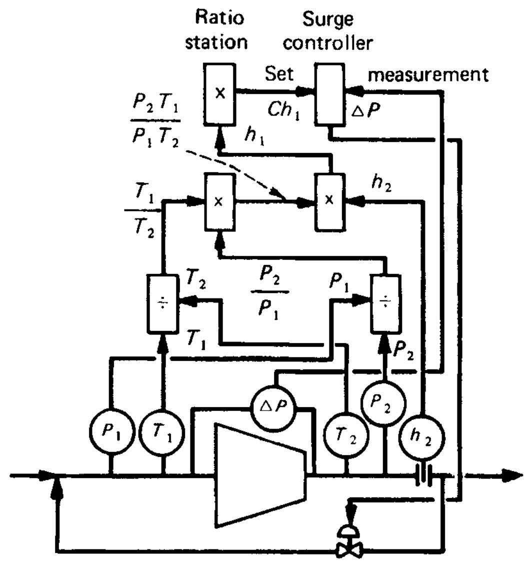

Figure 12-61D. Centrifugal compressor surge control schematic diagram shows instrumentation required when primary flow-measuring device is located in centrifugal compressor discharge line. Symbols: T = temperature; P = pressure; Δ = differential across compressor outlet to inlet. See Reference 89 for a detailed discussion. (Used by permission: White, M. H. *Chemical Engineering*, p. 54, Dec. 25, 1972. ©McGraw-Hill, Inc. All rights reserved.)

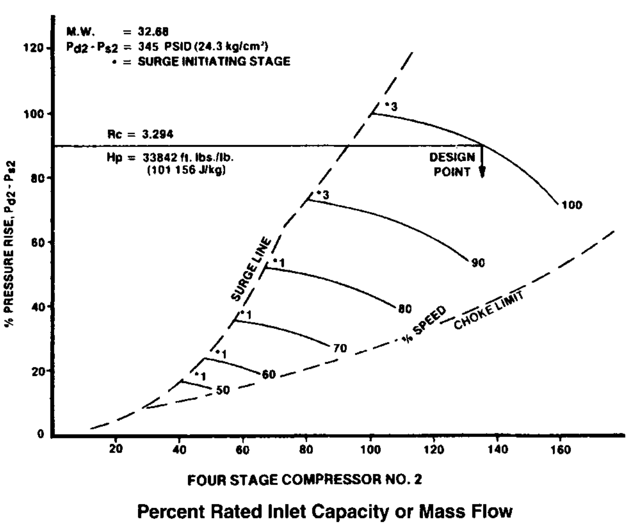

Figure 12-61C. Performance examination of a second set of centrifugal compressor conditions. The volume reduction per stage is less when operating at speeds lower than design; therefore, stages near the discharge are forced to handle more than their rated volume, and those nearer the inlet handle less than normal. The accumulated effect is that the surge line shape tends to depart from the fan law predictions as more stages are added. It is common for surge to be initiated by one of the latter stages when the compressor is operating at design speed but by an earlier stage at lower speeds. A distinct change in the surge line is evident at the point where this occurs. (Used by permission: Bul. 423, ©1992. Dresser-Rand Company.)

higher speeds will follow the locus of rated points indicated by the dotted line. The 100% rated speed line is the single line that defines the performance of the machine with changes in volume at the fixed speed. When inlet guide vanes are used at a fixed operating speed, the performance curve at that speed takes the form of Figure 12-62, being dependent upon the position of the vanes.

The discharge pressure developed by the compressor must be equal to the process gas's total system resistance, of control valves, hand valves, orifices, heat exchangers, and any other process-related devices through which the discharge gas from the compressor must flow. As this resistance changes, the gas flow through the compressor will automatically adjust itself to equal the new resistance.[82]

Referring to Figures 12-61 and 12-61A–C, the performance characteristic curve at any % speed tends to become vertical (right end of curve), which relates to the maximum volume that any compressor can deliver. At this point the flow cannot increase despite a significant reduction in the system resistance. As the flow decreases, the minimum point is called "surge," below which the operation becomes very unstable. Vibration of the compressor most probably will result, even to the point of cracking anchor flanges or possibly compressor cases. On Figure 12-61, the region to the left of the "Approximate Surge Limit" is the unstable condition. All centrifugal compressors have a flow at which maximum pressure is possible, and further reduction in flow beyond this point results in decreasing discharge pressure.

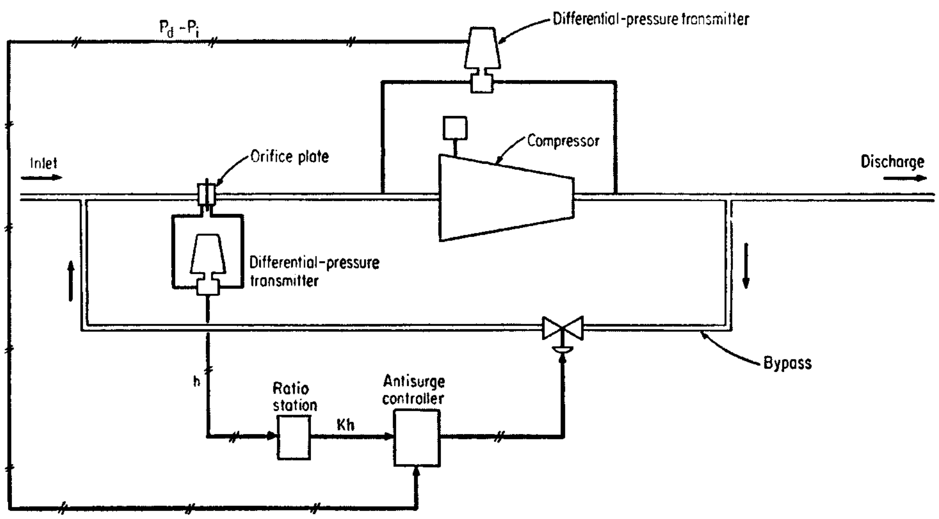

Figure 12-61E. Control system monitors discharge pressure and flow to prevent surging in gas compressor. (Used by permission: Magliozzi, T. L. *Chemical Engineering*, p. 139, May 8, 1967. ©McGraw-Hill, Inc. All rights reserved.)

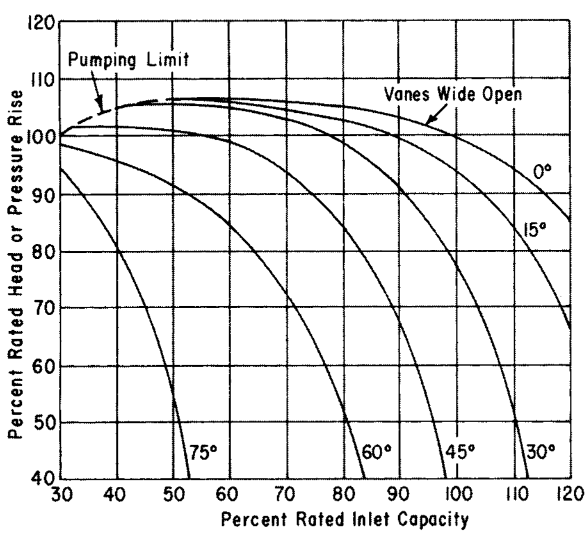

Figure 12-62. Representative centrifugal compressor performance with inlet guide vanes. (Used by permission: A C Compressor Corporation.)

"Pressure is built up again by the compressor. The flow proceeds in the normal direction. Surge, or back-and-forth oscillation, continues until the discharge pressure (system resistance) is decreased. The compressor should be run in a stable region and any excess flow should be blown off or sent through a manual or automatic surge control valve if the process flow is less than the compressor surge. In air applications, blow this surplus off to the atmosphere; in applications where the gas cannot be lost, recirculate the excess through a cooler to the inlet of the compressor." (Paraphrased from reference unknown.)

When a variable-speed driver, such as a steam turbine, is used for the compressor, the compressor performance can be varied to meet various operating flows and pressures by will be discussed later, for a fixed system resistance, the flow varies directly with the speed; head varies as the square of the speed; and horsepower varies as the cube of the speed. This latter condition of horsepower requirements is one that can control the flexibility of the system, as the possibility of exceeding available shaft horsepower can be an important factor.

The "surge limit" shown in Figures 12-61A–C is an important operating limit, as it is the minimum flow point below which the centrifugal or axial compressor becomes unstable through pressure and flow pulsations and results in a loud "roaring" noise. The forces involved can be great enough to rip compressor cases loose on their foundations. An antisurge control system is necessary to limit the capacity to a point in a safe region away from the surge limit.[82]

Surge Control

Refer to Figures 12-61C, 12-61D, or 12-61E.

Assume that the compressor is operating at the "Design Point" designated on the figures on the 100% speed curve, with the inlet flow as V_1 and the head as H_1. For example, if the external system resistance (friction, ΔP, etc.) increases while the speed remains constant, the flow decreases, and the operating line point will move along the 100% speed line to the left until it reaches the "Surge Limit" line. The flow has reached the appropriate value of V_2; the head has increased to H_2, which is the maximum head the compressor can produce at this speed. Here, the characteristic operating line is practically flat, and the compressor operation becomes unstable. This is called "surging," which produces rapid high frequency reversals in the axial thrust on the compressor shaft.[89] See Figure 12-61D for one instrument control diagram; this presents a simpler diagram but similar in concept to Figure 12-61C.[90] References 91, 92, 93, and 104 present useful analyses of the problem, as well as other

The lower right end of the performance curve on the figures terminates before reaching a limiting condition known as the "choke limit." Extension of the performance curve would be vertical at the choke limit. Controls are usually not required for this condition but should be considered in overall system control design.[82]

A "choke" or "stonewalling" condition can develop as indicated in Figure 12-61C. This occurs when the flow velocity in the compressor approaches the velocity of sound in the gas (sonic velocity or Mach 1) at the specific point shown on the operating characteristic curve. Usually velocities are well below sonic, but for heavy gases, such as some refrigerants, this situation must be recognized, and the performance of the unit designed to avoid the condition.[60] Note that the right side of the characteristic curves begin to turn down vertically as the "choke" or "stonewall" condition actually develops in the unit.

Tables 12-9A and 12-9B give a typical summary of multistage compressor selection. The efficiency, head, and speed data are orders-of-magnitude for several manufacturers; however, some designs normally are rated at values below or above those listed.

The gas compression in practically all commercial machines is polytropic. That is, it is not adiabatic or isothermal, but some form peculiar to the gas properties and the hydraulic design of the compressor. Actual machines may be rated on adiabatic performance and then related to polytropic conditions by the polytropic efficiency. Other performance rating procedures handle the calculations as polytropic. For reference, both methods are presented.

Compression Process

If the gas to be compressed contains water vapor (saturated or only partially saturated), this water content must be determined by (1) test of the mixture or (2) calculation. Then the properties of the gas-water vapor mixture must be determined by the usual gas calculations for weighted average molecular weight, average lb/min (or hr), "k" value or "n" value, and others as necessary.

$$y = P_w/P_t \tag{12-88}$$

where P_w = vapor pressure of water at temperature, psia
P_t = total system pressure, psia
y = mol fraction water vapor

This value of "y" can be used to calculate the weighted gas properties noted previously.

Adiabatic

Adiabatic compression (termed adiabatic isentropic or constant entropy) of a gas in a centrifugal machine has the same characteristics as in any other compressor. That is, no heat is transferred to or from the gas during the compression operation. The characteristic equation

$$PV^k = C' \tag{12-89}$$

applies, where "k" is the ratio of specific heats, c_p/c_v, for the gas. An increase in gas temperature accompanies the compression. The use of the adiabatic process in calculations enables the designer to work from Mollier diagrams. This is closer to being correct for an internally cooled compressor than an uncooled machine. It is still a theoretical operation.[46]

C′, C″, and C‴ are constants

Isothermal

Isothermal compression takes place when the heat of compression is removed during compression and when the temperature of the gas stays constant. The characteristic equation is

$$PV = C'' \tag{12-90}$$

Table 12-9A
Summary of Typical Multistage Compressor Data

Machine Series	No. of Stages per Case	Nominal Overall Efficiency %	Cfm Intake Volume Range (Approx.)	Nominal Head per Stage-ft	Nominal Speed rpm[a]	Max. Case Working Pressure Cast Iron	Max. Case Working Pressure Cast Steel
A	up to 7	78	18,000 to 40,000	9,000	4,700	125	Special order
B	3	75	20,000 to 28,000	9,000	5,000	60	Not available
C	up to 7	78	12,000 to 22,000	9,000	6,200	125	250
D	up to 7	77	3,500 to 12,000	8,500	8,100	250	400
E	up to 8	73	1,500 to 4,500	8,000	9,800	250	500
F	up to 8	73	1,000 to 3,500	8,000	9,800	not available	1,200[b]

[a]Maximum allowable continuous operating speed is 105% of values given.

[b]Forged steel.

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Table 12-9B
Summary of Typical Multistage Centrifugal Compressor Data

2 M-Line & MB-Line Frame Data

Frame	Nominal Flow Range (cfm)	Nominal Max No. of Casing Stages	Max Casing Pressure (psig)	Nominal Speed (r/min)	Nominal Polytropic Efficiency	Nominal H/N^2 (per stage)	Maximum Q/N
29M	750– 9,500	10	750	11,500	0.78	7.5×10^{-5}	0.83
38M	6,000– 22,000	9	625	7,725	0.79	1.52×10^{-4}	2.85
46M	16,000– 34,000	9	625	6,300	0.80	2.28×10^{-4}	5.40
60M	25,000– 58,000	8	325	4,700	0.81	3.85×10^{-4}	12.34
70M	50,000– 84,000	8	325	4,200	0.81	5.67×10^{-4}	20.
88M	70,000–135,000	8	325	3,160	0.81	9.1×10^{-4}	42.7
103M	110,000–160,000	8	45	2,800	0.82	11.6×10^{-4}	57.1
110M	140,000–190,000	8	45	2,600	0.82	13.4×10^{-4}	73.1
10MB	90– 1,600	12	10,000	18,900	0.77	2.6×10^{-5}	0.085
15MB	200– 2,350	12	10,000	15,300	0.77	3.6×10^{-5}	0.153
20MB	325– 3,600	12	10,000	12,400	0.77	6.2×10^{-5}	0.29
25MB	500– 5,500	12	10,000	10,000	0.78	9.5×10^{-5}	0.55
32MB	2,000– 8,000	10	10,000	8,300	0.78	1.39×10^{-4}	0.96
38MB	6,000– 22,000	9	1,500	7,725	0.79	1.52×10^{-4}	2.85
46MB	16,000–34,000	9	1,200	6,300	0.79	2.28×10^{-4}	5.40
60MB	25,000–58,000	8	800	4,700	0.80	3.85×10^{-4}	12.34
70MB	50,000–84,000	8	800	4,200	0.80	5.67×10^{-4}	20.

[1]Number of casing stages is determined by critical speed margins. These numbers are a general guideline only.

[2]These values are typical. Flexibility in types of available staging can allow final computer selections to have significant variations in head and efficiency.
Used by permission: Elliott Co., © Bul. P-26.

This process is not achieved in commercial units.

Polytropic

The polytropic process is mathematically easier to handle than the adiabatic approach for the following: (1) determination of the discharge temperature (see later discussion under "Temperature Rise During Compression") and (2) advantage of the polytropic efficiency:

$$e_p = [n/(n - 1)]/[k/(k - 1)] \quad (12\text{-}91)$$

which describes the relationship between "n" and "k". Thus, the polytropic efficiency is independent of the thermodynamic state of the gas during compression.[96]

Polytropic compression is characterized by being neither adiabatic nor isothermal but is a variable entropy process. Its relation is expressed

$$PV^n = C''' \quad (12\text{-}92)$$

where "n" is the characteristic of the gas that determines its compression performance. The exponent "n" is determined by the actual conditions of the gas at inlet to and discharge from the compressor, or to and from a specific cylinder.

When n = 1, the compression is isothermal; when n=k, it is adiabatic. The slope of the compression curve is a function of the exponent "n." Figure 12-60 illustrates the effect of the "n" and "k" values on the gas compression and the work associated with this compression.

The usual centrifugal compressor is uncooled internally, and hence, operates with polytropic characteristics having "n" greater than "k"; however, if the unit is internally cooled, then "n" will be greater than 1.0 but may be less than "k." The inefficiencies caused by internal losses (friction, etc.) keep the operation from being truly adiabatic; however, some compressions are close to this condition and may be used for approximations.

Woodhouse[54] presents the following relation for the polytropic exponent, "n," based on actual inlet and discharge specific volumes of the gas being compressed:

$$n = \frac{\log_{10}(P_2/P_1)}{\log_{10}(v_1/v_2)} \quad (12\text{-}93)$$

This applies with good accuracy for single wheels and for the overall multistage compressor. This is extremely useful in determining polytropic efficiency.

Values of "n" may be read from Figure 12-63 for approximate *actual* inlet flow capacity, cfm, to the suction of the

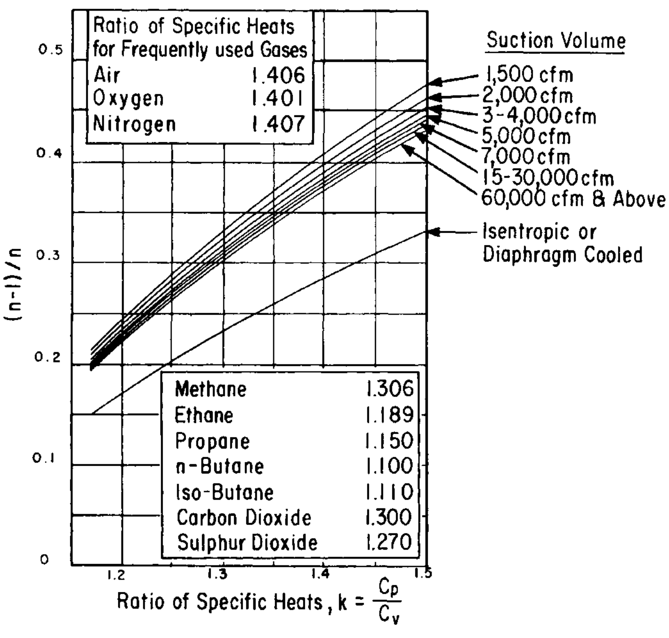

Figure 12-63. Ratio of specific heats (n − 1)/n. (Used by permission: Dresser-Rand Company.)

compressor, or to individual wheels of a multistage unit, if the individual wheels are being evaluated. Usually the process engineer is concerned with estimating the overall performance and not the individual wheels.

Specific volumes may be conveniently read from Figure 12-64.

Efficiency

Adiabatic Efficiency. The ratio of theoretical adiabatic horsepower to actual brake horsepower required at the compressor shaft is adiabatic efficiency. It is equal to compression efficiency × mechanical efficiency.[24, 54]

$$e_a = \frac{\text{adiabatic work}}{\text{polytropic work}} = \frac{[(P_2/P_1)^{(k-1)/k} - 1]}{[(P_2/P_1)^{(n-1)/n} - 1]} \tag{12-94}$$

$$e_a = \frac{\text{theoretical adiabatic temperature rise}}{\text{actual temperature rise}}$$

$$= \frac{T_1[(P_2/P_1)^{(k-1)/k} - 1]}{T_2(\text{actual}) - T_1} \tag{12-95}$$

Adiabatic Shaft Efficiency. The adiabatic and polytropic efficiencies do not include the losses of packing glands, oil pump, journal bearings, thrust bearings, etc.[48] To account for these losses in horsepower and to relate these losses to the actual brake horsepower, an overall efficiency can be used and expressed as the adiabatic shaft efficiency, e_{as}. The manufacturers usually know these losses in horsepower, or the values may be summarized as approximately 1–3% for units of 500–1,500 hp (approximately) and larger for smaller horsepowers and about 1–1.5% for horsepowers greater than 1,500.

Adiabatic efficiency is expressed as follows:

$$e_{ad} = \frac{H_{ad}}{H_p}(e_p) \tag{12-96}$$

$$= \frac{ZRT_1(k/(k-1)[(P_2/P_1)^{(k-1)/k} - 1]}{ZRT_1(n/(n-1)[(P_2/P_1)^{(n-1)/n} - 1]}(e_p) \tag{12-97}$$

The adiabatic efficiency is a function of the pressure ratio, and thus, dependent on the thermodynamic state of the gas undergoing compression.[96]

Polytropic Efficiency. This is the ratio of theoretical polytropic horsepower to actual brake horsepower at the compressor shaft. The polytropic efficiency does not include packing, bearing, or other losses. This efficiency is a measure of the hydraulic perfection of the compressor, and the value remains the same for any gas and for any speed (within reasonable limits).[37] For an uncooled compressor, the polytropic, hydraulic, and temperature rise efficiencies are the same.[54]

$$e_p = \frac{n(k-1)}{k(n-1)} \tag{12-98}$$

Values of e_p usually average between 0.77 and 0.82. For estimates, a value of 0.72–0.75 is reasonable.

Polytropic efficiency can also be defined as follows:[54]

$$e_p = \frac{\log_e[(P_2/P_1)^{(k-1)/k} - 1]}{\log_e(T_2(\text{actual})/T_1)} \tag{12-99}$$

Figures 12-65 and 12-65A give the relationship between polytropic and adiabatic efficiencies. The adiabatic efficiency can be calculated from operating data, and the polytropic efficiency can be read from the curves. For other cases, e_p may be calculated from the preceding relation and the adiabatic efficiency may be determined from the curves. Figure 12-66 illustrates the relationships that may exist while evaluating a particular compressor design.

Head

An important point to remember when calculating is that when you are using adiabatic head, use adiabatic efficiency; and when using polytropic head, use polytropic efficiency.[96]

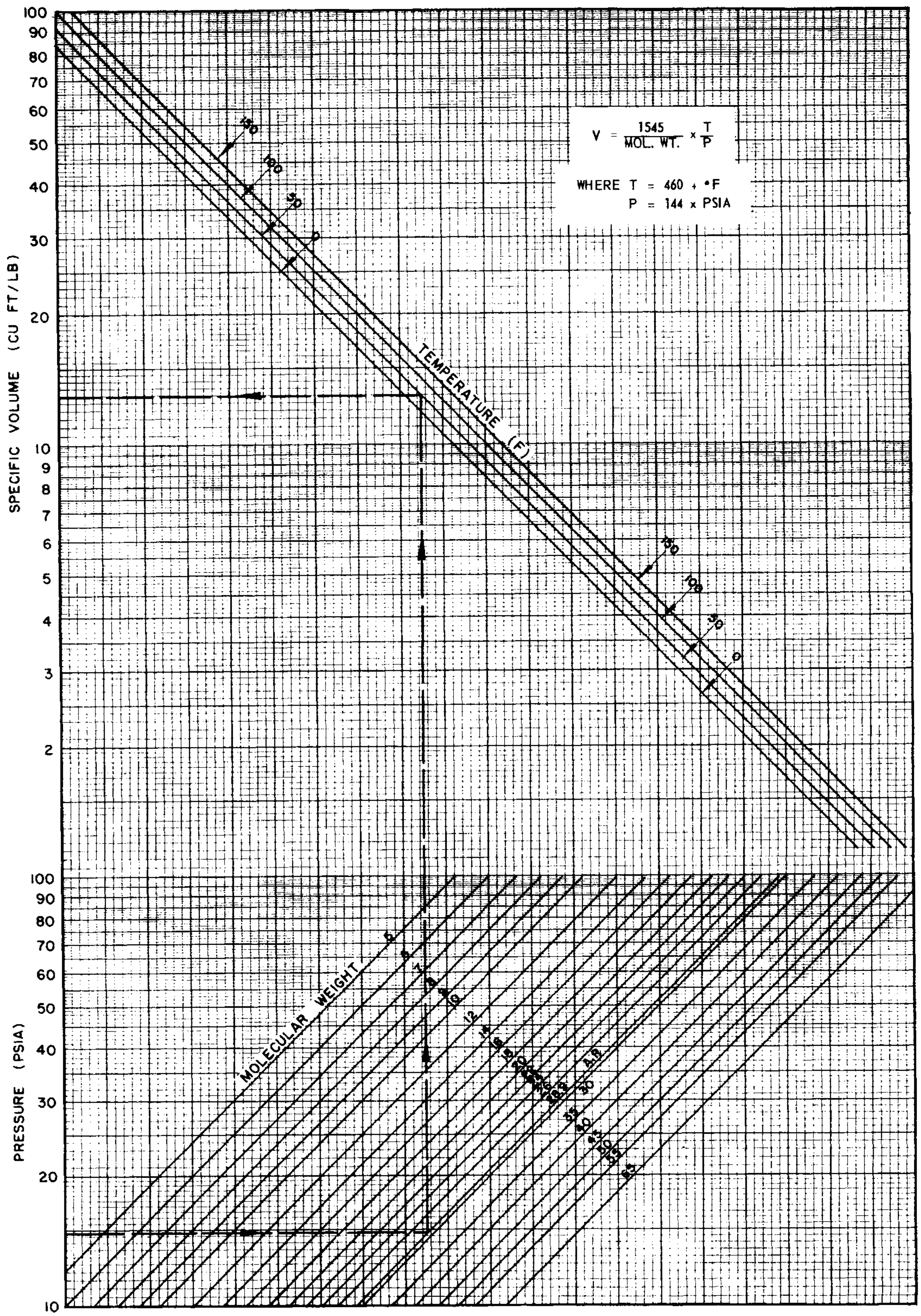

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Figure 12-64. Specific volume chart. (Used by permission: © Elliott Co.)

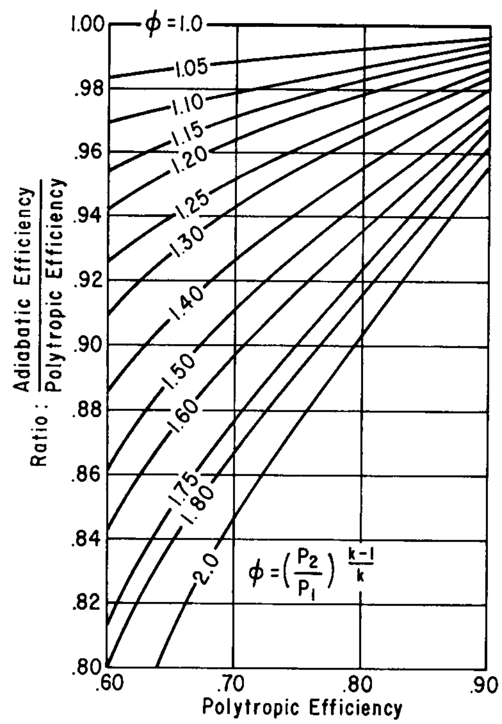


Figure 12-65. Relationship between adiabatic and polytropic efficiencies. (Used by permission: Woodhouse, H. *Petroleum Engineer,* Oct. 1953. ©Hart Publications, Inc. All rights reserved.)

Adiabatic Head. The height in ft of gas "supported" at the compressor discharge as the gas discharges into a system at the desired pressure level is the adiabatic head. The compression of the gas column is adiabatic. The temperature and pressure of the compression column will be related by the adiabatic expression.

Adiabatic head is expressed:[95]

$$H_a = 144\, P_1 v_s \left(\frac{k}{k-1}\right)\left[\left(\frac{P_2}{P_1}\right)^{(k-1)/k} - 1\right]\left(\frac{Z_s + Z_d}{2}\right),$$

ft-lb/lb or ft (12-100)

or,

$$H_a = RT_1\left(\frac{k}{k-1}\right)\left[\left(\frac{P_2}{P_1}\right)^{(k-1)/k} - 1\right]\left(\frac{Z_s + Z_d}{2}\right),$$

ft-lb/lb or ft (12-101)

$$H_a = \frac{1{,}545}{(\text{mol wt})} T_2 \left(\frac{k}{k-1}\right)\left[\left(\frac{P_2}{P_1}\right)^{(k-1)/k} - 1\right] Z_1,$$

ft-lb/lb or ft (12-101A)

where H_a = total head in ft, equal to work of compression in ft-lb/lb

R = gas constant = 1,545/mol wt

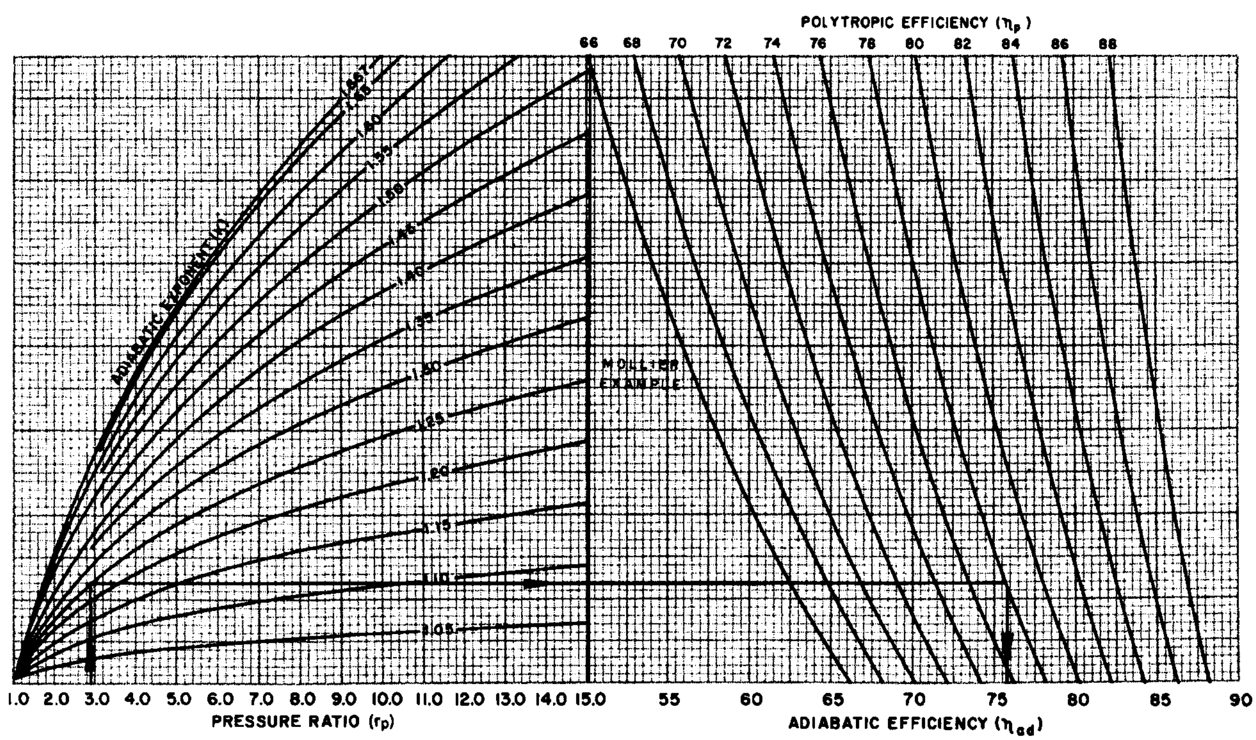


Figure 12-65A. Polytropic to adiabatic efficiency conversion. (Used by permission: Bul. P-26 and P-11A, ©1966. Elliott Co.)

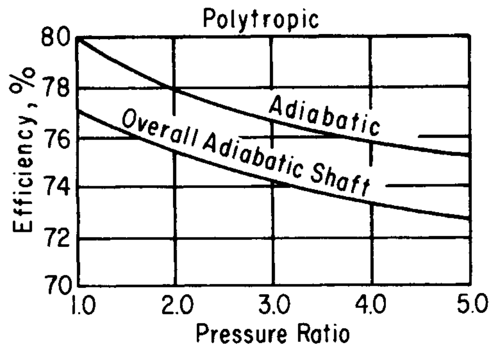


Figure 12-66. Comparative efficiencies of a 1,550 bhp centrifugal compressor based on 80% polytropic efficiency. (Used by permission: Woodhouse, H. *Petroleum Engineer,* Oct. 1953. ©Hart Publications, Inc. All rights reserved.)

T = temperature, °R
k = c_p/c_v, for gas
Z_1 = compressibility factor, dimensionless
P_1 = inlet pressure, psia
P_2 = discharge pressure, psia
v_s = specific volume of gas at suction conditions, ft^3/lb
1 = inlet or suction

Adiabatic Head Developed per Single-Stage Wheel. The head developed by a single stage of compression, consisting of an impeller and diffuser, depends upon the design, efficiency, and capacity and is related to its speed.[13]

$$(H_{ad})_s = H_a = \mu u^2/g, \text{ feet} \tag{12-102}$$

where μ = pressure coefficient, values range 0.50–0.65 for radial and forward-swept blading
u = rotor peripheral velocity, ft/sec, values range 600–900 ft/sec
g = gravitational constant, 32.2 ft/sec^2
υ = $N\pi d/720$ = impeller tip speed, ft/sec
N = rotational speed, rpm
π = 3.141
d = impeller tip diameter, in.
$(H_{ad})_s$ = adiabatic head developed by a single centrifugal stage, ft-lb/lb, or ft

An average value is about 0.55 for the coefficient, μ. Peripheral velocities will usually vary between 600–900 ft/sec; however, this varies with the gas being compressed and may run up to 1,100 ft/sec. The results of this head calculation will give values of 8,000–12,000 ft for a single stage. From this value, the total number of stages in the compressor can be approximated.

Polytropic Head. The polytropic head more closely approaches the conditions of an actual compressor and is the actual height of a gas column that can be maintained at the compressor discharge flange in order to support a particular pressure. The compression of the gas follows a polytropic path.[60]

$$H_p = 144\, P_1 v_s \left(\frac{n}{n-1}\right)\left[\left(\frac{P_d}{P_1}\right)^{(n-1)/n} - 1\right]\left(\frac{Z_s + Z_d}{2}\right),$$

ft-lb/lb or ft (12-103)

or:

$$H_p = Z_1 RT_1 \left(\frac{n}{n-1}\right)\left[\left(\frac{P_d}{P_1}\right)^{(n-1)/n} - 1\right], \text{ ft}$$

$$= \frac{1{,}545\, Z_1 T_1}{(\text{mol wt})}\left(\frac{n}{n-1}\right)\left[\left(\frac{P_2}{P_1}\right)^{(n-1)/n} - 1\right], \text{ ft} \tag{12-104}$$

where Z_m = compressibility factor of gas, expressing deviation from perfect gas law; at suction conditions to each wheel, if multistage unit, $(Z_s + Z_d)/2$
Polytropic head = adiabatic head/e_a
v_s = specific volume of gas at suction conditions, ft^3/lb
n = polytropic exponent
H_p = polytropic head, ft-lb/lb = ft
P_d = discharge pressure, psia
T_1 = inlet temperature, °R

Figure 12-67 is convenient to use in approximating the polytropic head.

Brake Horsepower

The power requirement for compression (only) of an ideal gas:[94]

$$hp_{id} = \frac{144\, P_1 V_1}{33{,}000}\frac{(n)}{(n-1)}\left[R_c^{n/(n-1)} - 1\right] \tag{12-105}$$

where hp_{id} = ideal compression horsepower
P = pressure, psia
V = inlet volume, ft^3/min
R_c = compression ratio, P_2/P_1
n = polytropic exponent of compression
1 = inlet condition
2 = discharge or exit condition

For several stages of compression the differences between the polytropic and adiabatic efficiency can become significant, due to the variation from the ideal compression, which gives an increase in the temperature of compression that results in an increase in the volume of gas at the exit from each stage. This requires the following stages to do more work when compared to all stages being ideal.

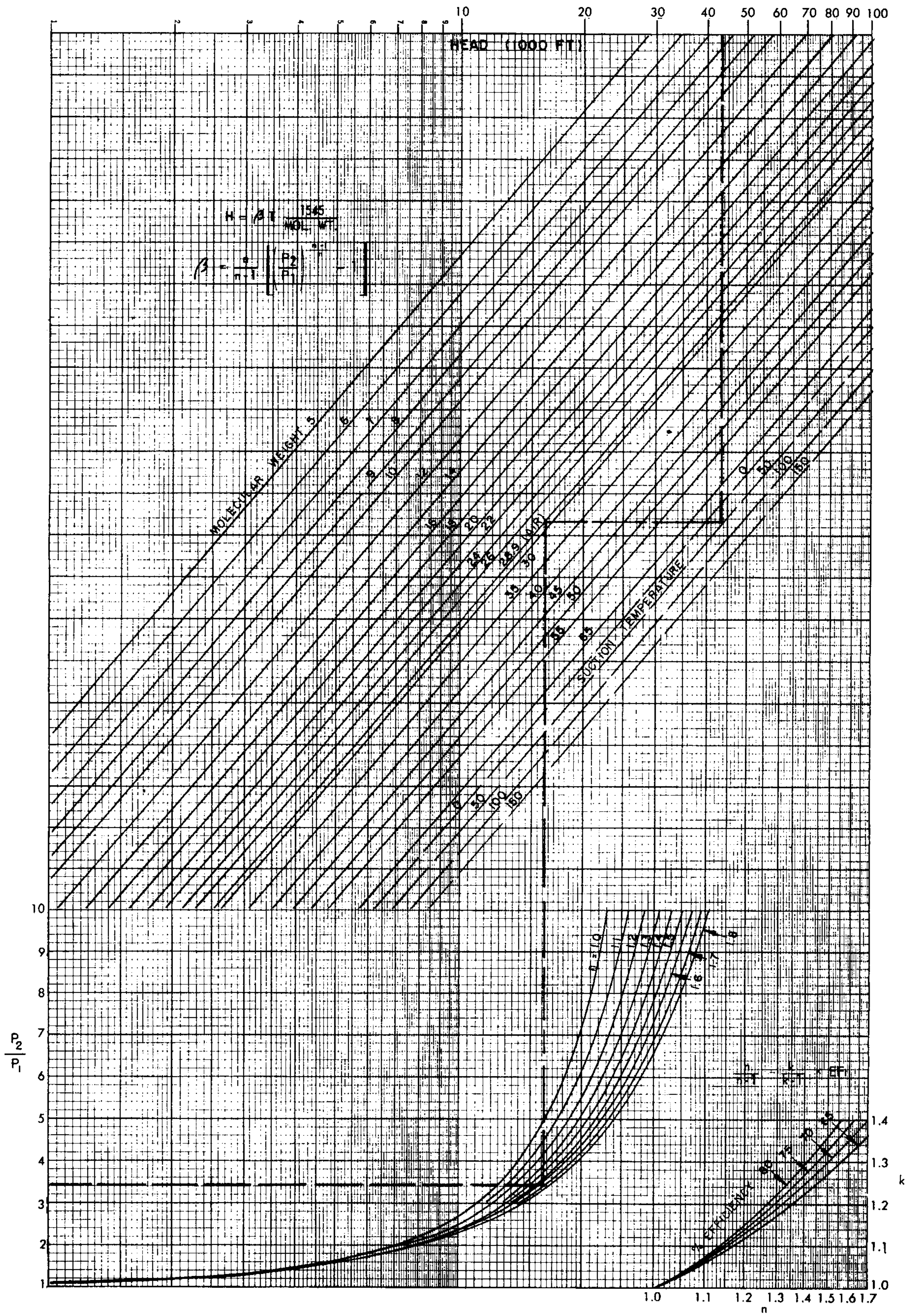


Figure 12-67. Approximate head selection. (Used by permission: Elliott® Co.)

For an ideal compression using enthalpy change:[94]

$$hp_{ad} = \frac{V_1(\Delta h)}{42.42\ v_2\ e_p} \qquad (12\text{-}106)$$

where Δh = enthalpy change, Btu/lb, from condition 1 to condition 2
v_2 = specific volume, ft^3/lb
v_1 = volume flow, ft^3/min
e_p = polytropic efficiency, fraction
hp_{ad} = adiabatic horsepower

The actual horsepower input to the compressor shaft is the sum of the gas compression horsepower plus losses from the compressor wheel friction, fluid friction, gas turbulence, gas by-passing internally, and seal and bearing friction.

$$\text{Gas hp} = (hp_g) = \frac{778(h_2 - h_1)W}{33{,}000} = \Delta h\ W/42.8 \qquad (12\text{-}107)$$

$$\text{Gas hp} = (hp_g) = (W)(H_p)/(33{,}000)(e_p) \qquad (12\text{-}108)$$

$$\text{Actual shaft bhp} = hp_g/(0.99 \text{ to } 097) \qquad (12\text{-}109)$$

ghp = gas horsepower *net* to shaft from driver,

$$\text{ghp} = \frac{(W)(\Delta H)}{(33{,}000)(e_p)} \qquad (12\text{-}110)$$

e_p = polytropic efficiency
Actual horsepower required by rating of driver (minimum),

$$= \frac{\text{Gas Horsepower}}{\text{Mechanical Eff.}} = \frac{\text{ghp}}{0.99 \text{ to } 0.97 \text{ est.}} \qquad (12\text{-}111)$$

H_p = polytropic head
ME = mechanical efficiency

This is approximately correct because the mechanical losses in the compressor are only about 1–3%.[13, 37] The head determined from Figure 12-67 can be used for the polytropic gas horsepower relation given previously.

If the polytropic head and efficiency are known, these values can be substituted in[85]

$$hp_g = H_{ad}W/(33{,}000)(e_a) \qquad (12\text{-}112)$$

for the adiabatic values:

$$H_{ideal} = H_{ad}/e_{ad} = H_{poly}/e_{poly} = \text{work input/lb fluid} \qquad (12\text{-}113)$$

Because the H/e versus Q relationship is nearly linear, the HP_g versus Q is essentially a straight line through the origin for a compressor with radially bladed impellers. For back-swept impeller blades, the hp_g versus Q curve is a flat parabola, peaking near or above the maximum flow[85].

Shaft horsepower is greater than gas or hydraulic (hp_g) horsepower due to the effects of bearings, seals, and windage (or wheel friction losses). Although these losses can be determined individually, the sum of all amount to 1–3% of rated gas (hp_g) horsepower on large- to medium-size compressors.[85]

$$\text{bhp} = \frac{P_1V_1[n/(n-1)]\left[\left(\frac{P_2}{P_1}\right)^{(n-1)/n} - 1\right]\left[\frac{(Z_1 + Z_2)/2}{Z_1}\right]}{229(e_a)} \qquad (12\text{-}114)$$

where W = flow of gas, lb/min
h_1, h_2 = enthalpy, discharge and inlet, Btu/lb
Δh = enthalpy change, Btu/lb
hp_g = gas horsepower = hydraulic horsepower
H_p = polytropic head, ft of fluid, Equation 103
e_p = Hydraulic or polytropic efficiency, usually 0.70–0.80
bhp = brake horsepower at compressor shaft

Brake horsepower per 1 million ft^3/day measured at 14.7 psia and suction temperature using 75% overall compressor efficiency is given in Figure 12-68, and a volume correction factor is shown in Figure 12-69.

Centrifugal Compressor Approximate Rating by the "N" Method

Outline adapted by permission of Elliott Co.[118]

1. Determine mixture properties of gas or gas mixtures.
2. Determine inlet flow as Q, taking into account the compressibility factor, Z.
3. Select a compressor frame as in Table 12-9B and note the average polytropic efficiency listed, speed, and head/stage.
4. Calculate average gas compressibility (inlet + outlet)/2.
5. Calculate polytropic head, H_p, using Equation 12-103.

Calculate discharge temperature to be certain that no internal or separate outside cooling is required, such as in Figure 12-40C and 12-40D:

$$T_2/T_1 = (P_2/P_1)^{(n-1)/n} \qquad (12\text{-}115)$$

T = °R, and P = psi, abs

6. Determine the number of casing stages required. Using Table 12-9B, determine nominal speed. Calculate Q/N. Then, pick H/N^2 from table.

$$H/\text{stage} = (H/N^2)(N^2), \text{ ft-lb/lb, or ft‡} \qquad (12\text{-}116)$$

Approximate number of stages = H_p calculated in (5) ; divide by H/stage calculated previously = No. stages

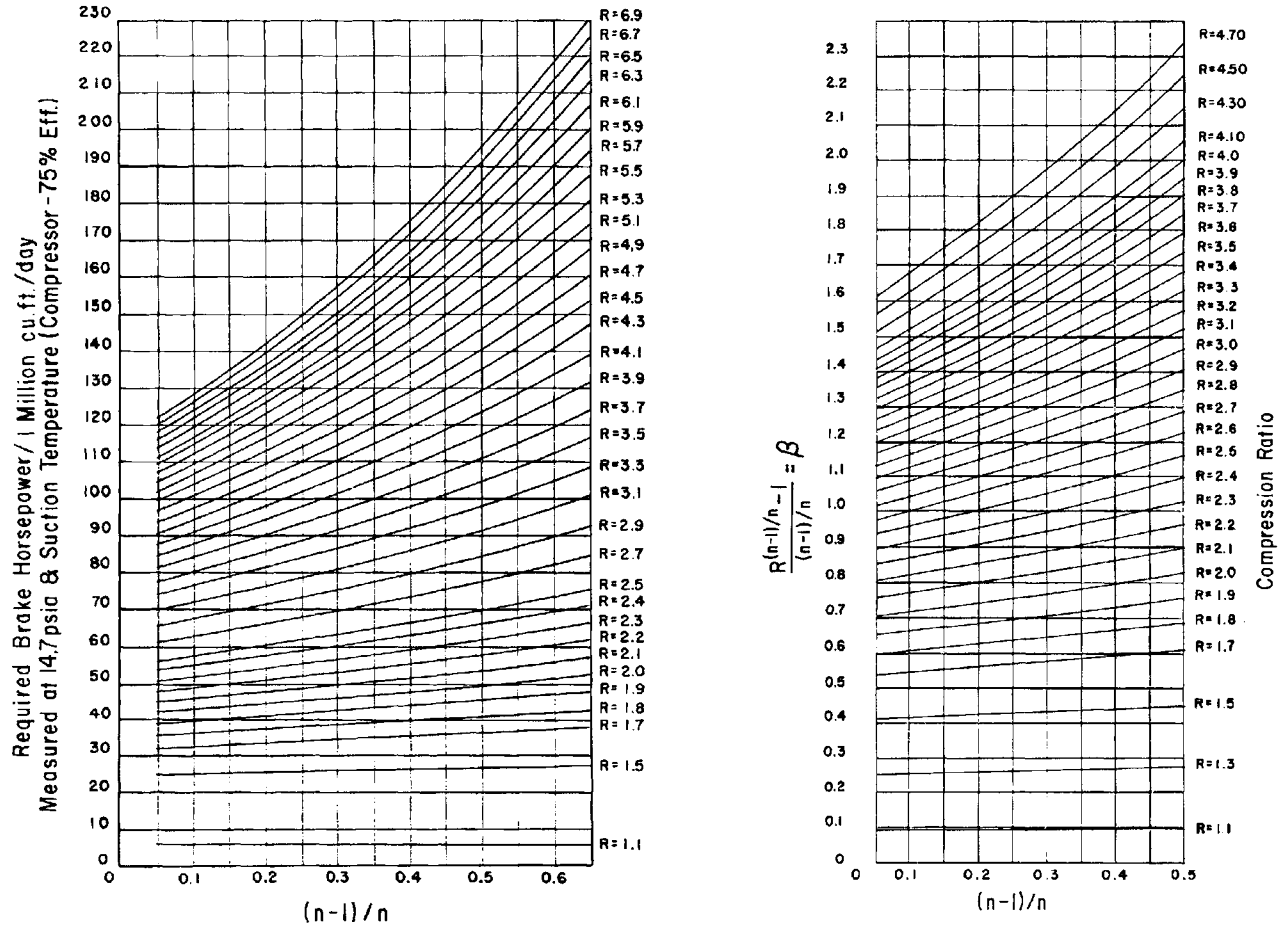


Figure 12-68. Brake horsepower per million ft^3 per day for compressors as a function of (n − 1)/n value. (Used by permission: Dresser-Rand Company.)

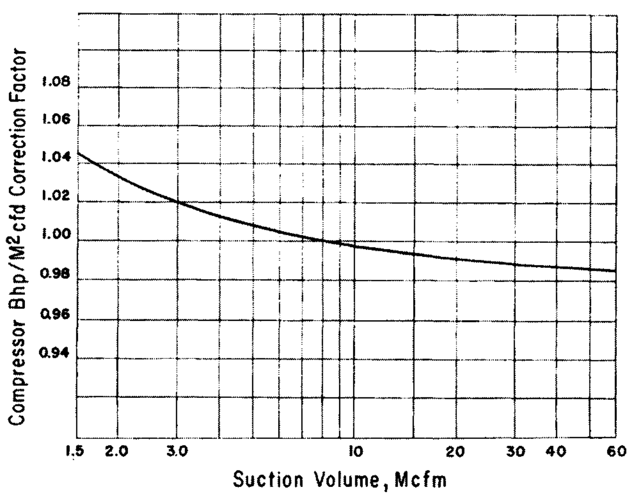


Figure 12-69. Correction factor for compressor bhp/million ft^3 per day at 14.7 psia and suction temperature versus suction volume. (Used by permission: Dresser-Rand Company.)

7. Adjust speed based on casing stages.
   No. stages from (6) must develop head from (5) or determine average head/stage = H_p/no. stages = head/stage, ft-lb/lb or ft, per stage
   From the Fan Laws, $H \propto N^2$.
   $N = N_{nominal}$ [H(avg/stage)/H‡], rpm
8. Approximate power required.

$$\text{Gas horsepower} = \frac{(w_1)(H_p)\,(\text{from } (5))}{(33{,}000\ e_p)} \qquad (12\text{-}117)$$

9. Adjust for balance piston leakage.

$$= (\text{ghp})(1.02) = \text{corrected ghp} = (a) \qquad (12\text{-}118)$$

10. Add losses from Figure 12-70B = (b).
11. Total shaft or brake horsepower:

$$= (\text{Corrected ghp, (a)}) + (\text{Losses, (b)}) \qquad (12\text{-}119)$$

The use of an enthalpy diagram or Mollier chart is perhaps the most accurate and is an easy method for determining horsepower. Figure 12-70 illustrates the compression paths on an ammonia diagram.

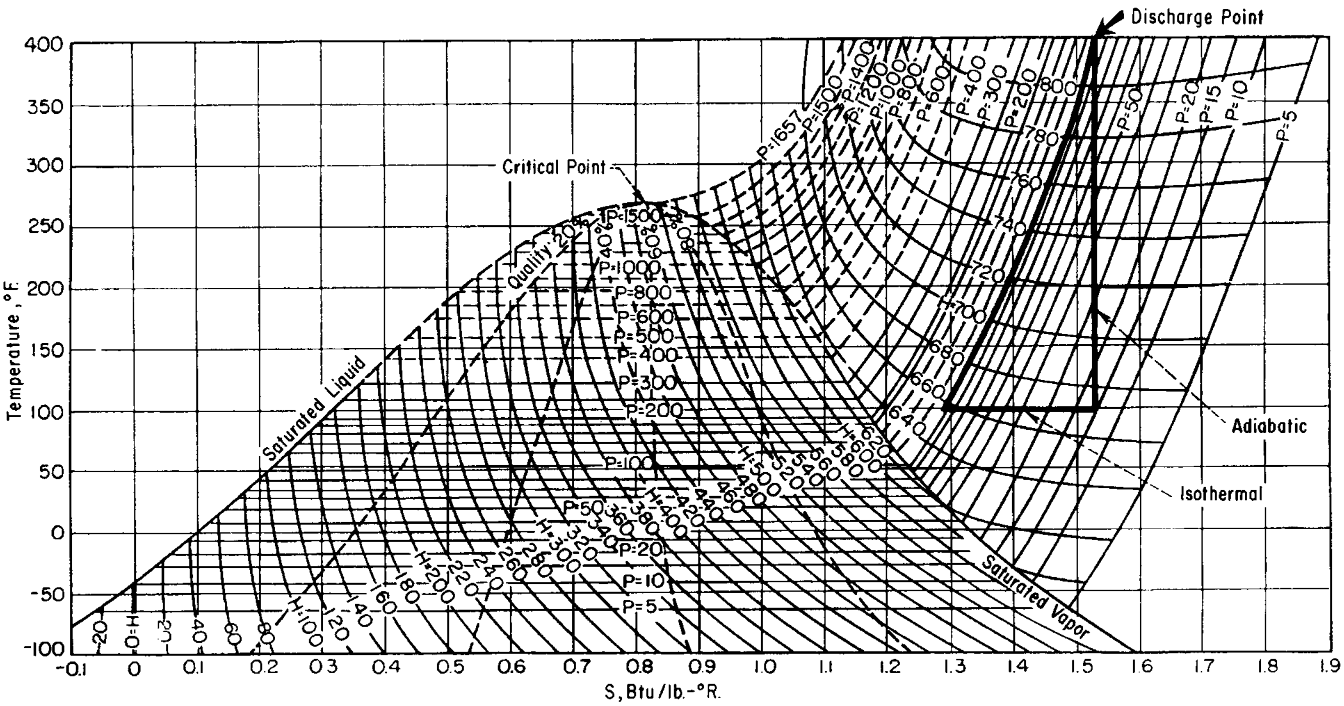


Figure 12-70. Entropy-temperature diagrams help to solve compression work problems. Data for ammonia provided. (Used by permission: Corrigan, T. E. and A. F. Johnson. *Chemical Engineering,* V. 61, No. 1, ©1954. McGraw-Hill, Inc. All rights reserved.)

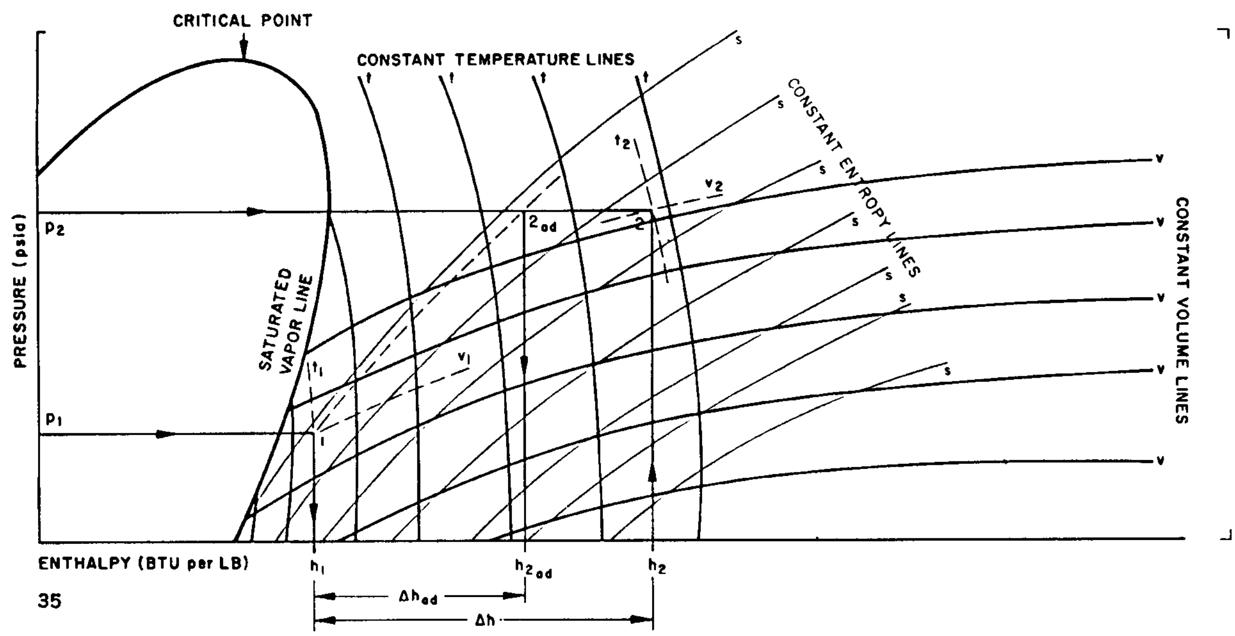


Figure 12-70A. Basic concepts for solutions using the Mollier Diagram of a specific process gas (or gas mixture when diagram is available). (Used by permission: Bul. P-11A, ©1966. Elliott® Co.)

Compressor Calculations by the Mollier Diagram Method

Figure 12-70A illustrates the basic concepts of using the Mollier diagram to solve centrifugal compressor problems. The steps involved using this method are[118] as follows (adapted from Elliott® Co. Reference 118 by permission):

1. Determine inlet flow, Q.

   $Q_1 = v_1(w)$, volume flow, ft^3/min

   w = weight flow, lb/min

   v_1 = inlet specific volume, ft^3/lb

On Mollier Diagram, Figure 12-70A, locate inlet state point (1) at intersection of p_1 (psia) and t_1 (°F).

2. Select the compressor frame from Table 12-9B for a typical example, not attempting to promote any particular manufacturer (each manufacturer has its own tables). Based on the inlet volume and the required discharge pressure, select frame size.
3. Calculate adiabatic head (H_{ad}).
Read inlet enthalpy, h, directly below point (1). See Figures 12-70A and 12-65A.
Follow the line of constant entropy (s) to discharge pressure, p_2, locating adiabatic discharge state point (2_{ad}). Read adiabatic enthalpy (h_{2ad}) directly below point (2_{ad}).

Then, $\Delta h_{ad} = h_{2ad} - h_1$, Btu/lb
Conversion: 778 ft-lb/Btu

$$\text{Then, } H_{ad} = (\Delta h_{ad})(778), \text{ ft, adiabatic} \tag{12-120}$$

At polytropic efficiency from Table 12-9B, using Figure 12-65A, read adiabatic efficiency, e_{ad}.

4. Polytropic head (H_p)

$$H_p = \frac{H_{ad}(e_p)}{e_{ad}}, \text{ see ++ in 7} \tag{12-121}$$

Determine k for the specific gas or gas/vapor mixture from Table 12-4 or other sources.
Calculate the compression ratio, $P_d/P_s = P_2/P_1 = R_c$.
From Table 12-9B for the compressor frame selected, select polytropic efficiency, e_p, and using Figure 12-65A, determine adiabatic efficiency, e_{ad}.

5. Number of Casing Stages

From Table 12-9B select the nominal speed for the size compressor casing (frame) established in Step 2.

$$\text{No. Stages} = \frac{H_p(\text{from Step 4})}{[\text{maximum head per stage (Table 12-9B)}]} \tag{12-122}$$

From Table 12-9B, $H/N^2 = x$

$$H_p/\text{stage} = \text{head/stage} = (H/N^2)(N^2) = (x)(N^2), \text{ ft-lb/lb**} \tag{12-123}$$

N from Table 12-9B
No. casing stages = H_p/(head/stage), stages rounded up to nearest whole number

6. Adjust speed.

(No. stages from (5) must develop, H_p, ft (from Step 4) = ft-lb/lb
Average head/stage = H_p/(no. stages) = ft-lb/lb per stage*
From Fan Law, $H \propto N^2$, then
required speed, $N = N_{nominal}$ (from Table 12-9B)

$$\frac{[H_{req'd\ avg./stg.*,actual\ above}]^{1/2}}{[H_{p,(calc.\ Step\ 5,**\ above}]} \tag{12-124}$$

7. Gas horsepower

$$ghp = \frac{w_1(H_p)}{33{,}000(e_p)} = \text{horsepower, ghp, ++ see Step 4} \tag{12-125}$$

8. Shaft horsepower

shp = total horsepower required to compressor shaft
shp = gas hp + bearing + oil seal losses + balance piston leakage
Balance piston leakage = 1.02 (ghp) (balanced piston hp)
Add losses from Figure 12-70B = y, then,
Total shaft horsepower = (balanced piston leakage, Bal P_{HP}) + y (12-126)

9. Actual discharge enthalpy, h_2

$$h_2 = \frac{(\Delta h_{ad})}{e_{ad}} + h_1 \tag{12-127}$$

Δh_{ad} = from Step 3.
e_{ad} = from Step 4.

10. Discharge yemperature, t_2
In Figure 12-70A (Mollier Diagram), plot vertically from h_2 (Step 9) to discharge pressure, p_2. At this point, read discharge temperature, t_2, following temperature lines.
11. Discharge specific volume
From Step 10 at point (2), in Figure 12-70A, read discharge specific volume, v_2.
12. Discharge flow, $Q_2 = (w)(v_2)$, ft³/min at discharge conditions (12-128)

where Q = capacity flow, ft³/min
w = weight flow, lb/min
v_1 = specific volume, ft³/lb, inlet
v_2 = specific volume, ft³/lb, outlet
P = p = inlet pressure, psia
H = head, ft-lb/lb = ft
h = enthalpy, Btu/lb
e = efficiency, fraction
R_c = ratio of compression = $P_d/P_s = P_2/P_1$
t = temperature, °F
T = temperature, °R
N = speed, rpm, (revolutions/min)
ghp = gas horsepower
shp = shaft horsepower

Subscripts:
ad = adiabatic
p = polytropic
1 = inlet
2 = outlet
d = discharge
s = suction

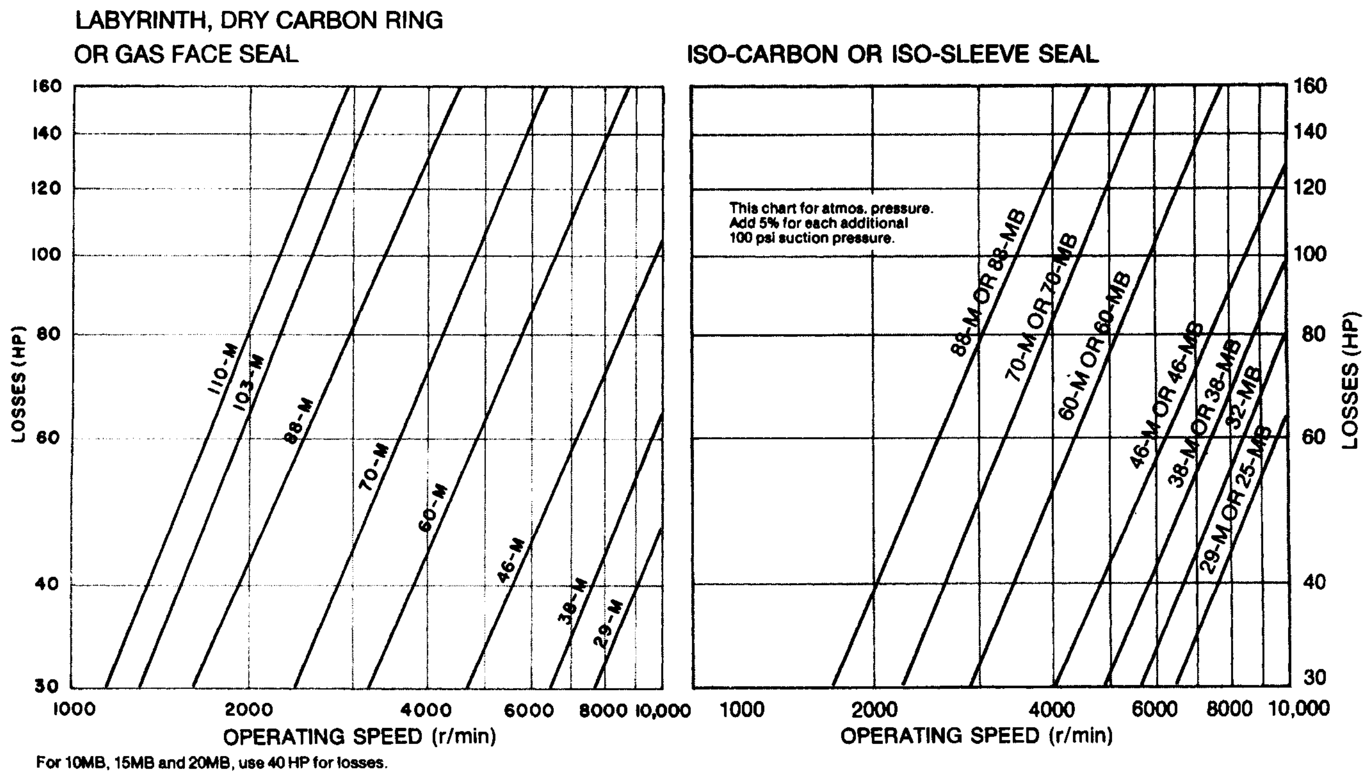


Figure 12-70B. Typical mechanical losses for seals on shafts of centrifugal compressors. (Used by permission: Bul. P-26. Elliott® Co.)

The initial and final enthalpy values can be located, and the Δh can be calculated.

Example 12-8. Use of Mollier Diagram

(Reproduced by permission of Elliott® Co., Bul. P-26, 15-194-FL., with clarifying notes by this author)

An ethylene gas centrifugal compressor required the following operating conditions:

Flow: w_1 = 1,769 lb/min
Inlet pressure: P_1 = 80 psia
Inlet temperature: T_1 = 90°F (550°R)
Discharge pressure: P_2 = 225 psia

1. Calculate inlet volume.

v_1 = 2.6 (from Mollier Diagram)
$Q = w_1 \times v_1 = 1{,}769 \times 2.6 = 4600$ icfm (inlet ft³/min, which is *actual* ft³/min (acfm):
acfm = [scfm @ 60°F, 14.7 psia and dry] $\times P_s/P_1 \times T_1/T_s \times Z_1/Z_s$

where P_s = standard pressure, usually 14.7 psi absolute
P_1 = inlet pressure, psi absolute
T_s = standard temperature, usually 520°R
T_1 = inlet temperature, °R
Z = compressibility; 1 = inlet, s = standard, usually 1.0

2. Select compressor frame size.
   Based on an inlet volume of 4,600 icfm and knowing the required discharge pressure is 225 psia, select a 29M frame size from Table 12-9B.
3. Calculate the required head.
   At given inlet conditions, determine inlet entropy (s) and enthalpy (h) from Mollier Diagram:
   P_1 = 80
   T_1 = 90°F
   s_1 = 1.75
   h_1 = 163
   At required discharge pressure and constant entropy ($s_1 = s_2$), determine h_2 from chart.
   P_2 = 225
   T_{2i} = N/A
   s_2 = 1.75
   h_{2i} = 205
   H_{ad} = head required = 778 ($h_{2i} - h_1$)
   H_{ad} = 778 (205 − 163) = 32,676 ft-lb$_f$/lb$_m$ (adiabatic)

Check the discharge temperature for a need to intercool. (Cool if $T_2 > 400$°F.)

Step 1. Determine adiabatic efficiency.
R_c = 225/80
k = 1.24
η_p = 0.78 (from Table 12-9B)
η_{ad} = 0.76 from Figure 12-65A

Step 2. Determine actual (not isentropic) Δh.

$$\Delta h = (h_{2i} - h_i)/\eta_{ad} = (205 - 163)/0.76 = 55.3$$

Step 3. Determine h_2 and read T_2 from Mollier Diagram.

$$h_2 = h_1 + \Delta h = 163 + 55.3 = 218 \text{ Btu/lb}$$

Plot vertically from h_2 to P_2 (225 psia) and read T_2 along temperature lines (not on vertical or horizontal scales).

$T_2 = 232°F$ (from Mollier Diagram)
No isocooling is therefore required.

5. Determine the number of casing stages.
From Table 12-9B, the nominal speed for a 29M is 11,500 rpm. Convert adiabatic head to polytropic head by the ratio of efficiencies.

$H_p = 32{,}676\ (0.78/0.76) = 33{,}536$ ft
From Table 12-9B, $H/N^2 = 7.5 \times 10^{-5}$
Therefore,
$H/_{stage} = H/N^2 \times N^2 = (7.5 \times 10^{-5})(11{,}500)^2 = 9.919$ ft-lb$_f$/lb$_m$
Determine approximate number of casing stages.
Number of stages = 33,536/9,919 = 3.38 ≅ 4 stages

6. Adjust speed.
Adjust the nominal speed according to the casing stages.
4 stages must develop 33,536 ft-lb$_f$/lb$_m$ or an average of 33,536/4 = 8,384 ft-lb$_f$/lb$_m$ per stage
Using Fan Law relationships, adjust speed:

$H \alpha N^2$

$$N = N_{norm}\,[H_{req'd}/H]^{1/2} = 11{,}500[8{,}384/9{,}919\,]^{1/2}$$

$$= 10{,}573 \text{ rpm}$$

7. Calculate the approximate gas horsepower.

$$ghp = \frac{w_1 \times H}{33{,}000 \times \eta_p} = \frac{1{,}769 \times 33{,}536}{33{,}000 \times 0.78} = 2{,}305 \text{ hp}$$

8. Shaft horsepower

Total shaft or brake horsepower = (adjust. leakage plus losses)
shp = total shaft horsepower = balanced piston leakage (Bal. P_{HP}) + y = 2,351 + 70 = 2,421 shp
Adjust for balanced piston leakage.

$$2{,}305 \times 1.02 = 2{,}351 \text{ hp} = (a)$$

Adjust losses from Figure 12-70B for assumed isocarbon seal.
y = 70

9. Actual discharge enthalpy, h_2; see Paragraph 3, Step 3.
10. Adiabatic head, see Paragraph 3.
11. Polytropic head, see Paragraph 5.
12. Discharge temperature, T_2.
Plot h_2 (Step 3) at discharge pressure of 225 psia on ethylene Mollier Diagram; read $T_2 = 235°F$.
13. Discharge specific volume.
Read $v_2 = 1.15$ ft^3/lb at point of Paragraph 12.
14. Discharge volume.

$$Q_2 = (w)(v_2) = (1{,}769)(1.15)$$

$$= 2{,}034 \text{ ft}^3\text{/min at discharge conditions}$$

Example 12-9. Comparison of Polytropic Head and Efficiency with Adiabatic Head and Efficiency

A process system is planned to operate as follows:

Inlet: $P_1 = 14.5$ psia; $v_1 = 15$ ft^3/lb
Discharge: $P_d = 42.0$ psia; k = 1.41
Polytropic efficiency = 75%
Specific volume: = 15.8 ft^3/lb

Determine: Polytropic head, adiabatic head, and adiabatic efficiency.

Compression ratio: = 42/14.5 = 2.896
$n/(n-1) = k(e_p)/(k-1)$
$n/(n-1) = 0.75(1.41)/(1.41-1) = 2.579$
$(n-1)/n = 1/(2.579) = 0.387$

Polytropic head:

$= 144(p_1v_1)(n/(n-1))[p_d/p_1]^{(n-1)/n} - 1]$; (omit "Z" for low pressure)
$= 144(14.5)(15)(2.579)[2.896^{0.387} - 1]$
$= 144(14)(15)(2.579)(1.513 - 1)$
= 41,475 ft

Assuming one pound basis:

$[hp_g/e_p]$(No. lb, flowing/min)
Shaft work = 1(41,475)/0.75
= 55,300 ft-lb/min

$$\text{shaft horsepower} = \frac{55{,}300}{33{,}000} \text{ (Assum mech./hydraulic eff. = 0.98)}$$
$$= 1.71 \text{ hp}$$

Adiabatic head:

Determine adiabatic k.
$k/(k-1) = 1.41/(1.41-1) = 3.439$
$(k-1)/k = 1/3.439 = 0.291$

Adiabatic head = $144\ k/(k - 1)(p_1v_1)[(p_d/p_1)^{(k-1)/k} - 1]$
= $144(3.439)(14.5)(15)(2.896^{0.291} - 1)$
= 38,990 ft
Adiabatic efficiency: = (38,990/55,300)(100)
= 70.5%
Adiabatic shaft work = 38,990/0.705 = 55,300 ft = lb/min
Shaft horsepower 55,300/(0.98) (33,000) = 1.71 hp

Speed of Rotation

The maximum speed of a compressor is fixed by mechanical or structural limiting of the peripheral velocity of the impeller wheels. The required velocity is established by the head to be developed. The capacity of the machine at suction conditions is a function of the individual wheel designs and the diameter.[13,37]

Peripheral velocity (or tip speed):
$\upsilon = \pi D\ (\text{rpm})/720$, ft/sec
$\pi = 3.1416$

The tip speed of an impeller is a crude guide as to the relative conservatism in its rating in the compressor case when reviewed on a competitive basis between different makes or designs. The feeling exists that the lower the tip speed, the better the design, and the longer the unit will run with trouble-free service. This is only partially true, if at all, as the real factors lie in the structural design together with materials of construction. The rotor assembly life is a function of a bearing size and design.

$$\text{rpm} = \frac{1300}{D}\sqrt{\frac{H'}{\mu}} = 229.3\ \upsilon/D \qquad (12\text{-}129)$$

where
rpm = rotative speed, revolutions per minute
υ = peripheral velocity, ft/sec
D = impeller diameter, in.
H′ = head per stage, ft of liquid
μ = pressure coefficient, average value 0.55
T = absolute temp., °R, t = F

Temperature Rise During Compression

Adiabatic discharge temperature, T_2:

$$T_2 = T_1\frac{\left(\frac{P_2}{P_1}\right)^{(k-1)/k} - 1}{e_{ad}} + T_1 \qquad (12\text{-}130)$$

Polytropic:

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(n-1)/n} \qquad (12\text{-}131)$$

where T = absolute temp., °R
Sub-1 = suction or inlet
Sub-2 = discharge
R = 1544/MW

The values for polytropic conditions represent an uncooled compressor, that is, no internal diaphragm cooling, no liquid injection, and no external coolers for the pressure range being considered.

Adiabatic.[96]

$$T_2 = T_1\frac{[(P_2/P_1)^{k-1/k} - 1]}{e_{ad}} + T_1 \qquad (12\text{-}132)$$

The discharge must be the same regardless of whether the process is considered adiabatic or polytropic. Thus,[96]

$$T_2 = T_1(P_2/P_1)^{n-1/n} = \frac{T_1[(P_2/P_1)^{k-1/k} - 1]}{e_{ad}} + T_1 \qquad (12\text{-}133)$$

Relationship between adiabatic compression and polytropic compression:[80]

$$\frac{n}{n-1} = \frac{(k)e_p}{(k-1)} \qquad (12\text{-}134)$$

where n = polytropic efficiency coefficient for compression
k = adiabatic efficiency coefficient for compression
= specific heat ratio, c_p/c_v
e_p = polytropic efficiency, fraction
e_{ad} = adiabatic efficiency, fraction

Polytropic efficiency may range from 77–82%, from Elliott.[80]

Figure 12-22 is convenient for solving for the temperature rise factor for either polytropic or adiabatic conditions, depending upon whether k or n is used.

Figure 12-71 can be used to solve for polytropic discharge temperature directly. Note the temperature limit line of 450°F, applicable to most mechanical designs.

Sonic or Acoustic Velocity

The velocity of sound, V_s, in any gas may be calculated from

$$V_s = [k(32.2)(R)(T)(Z)]^{1/2}, \text{ feet/second} \qquad (12\text{-}135)$$

where k = ratio of specific heats, c_p/c_v
R = gas constant = 1,545/mol wt
T = average absolute temperature of gas, °R, or may be calculated at suction temperature
Z = compressibility factor for gas at temperature, T
p′ = absolute pressure, lb/ft² abs
γ = specific weight of gas, lb/ft³
g = acceleration due to gravity, = 32.2 ft/sec/sec
or, $V_s = [kgp'/\gamma]^{1/2}$, ft/sec

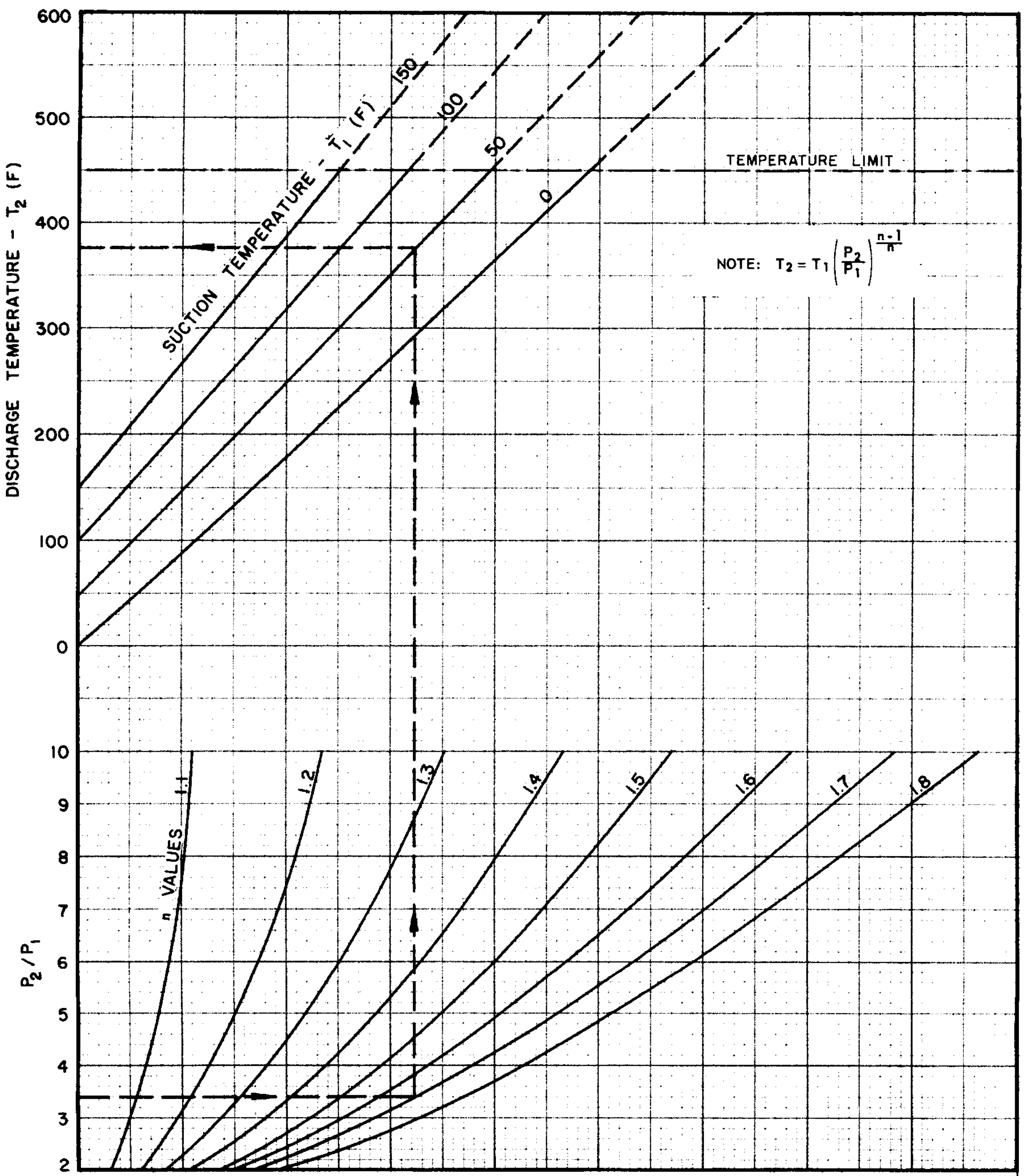


Figure 12-71. Polytropic compressor discharge temperature. (Used by permission: Elliott® Co.)

General design practice avoids using gas velocities near or greater than the sonic velocity. Figure 12-72 indicates the effect of temperature on the sonic velocity.

Mach Number[50]

The ratio of the gas velocity at any point to the velocity of sound in the gas is known as the Mach number, M′.

$$M' = \upsilon/V_s \qquad (12\text{-}136)$$

υ = gas velocity at any point
V_s = speed of sound in gas

Usual practice uses the peripheral velocity, υ, of the impeller as a criterion for establishing an approach to this number. This may not be the point of maximum velocity in the unit, and if it is not, the effects of the Mach number will

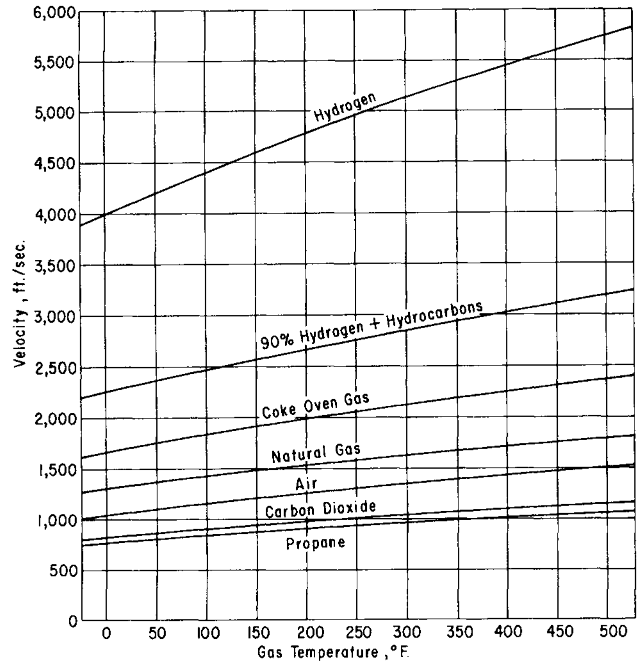


Figure 12-72. Sonic velocity of common gases. (Used by permission: Koenig, C. F. III. *Refining Engineer*, Aug. 1958. ©Hart Publications, Inc. All rights reserved.)

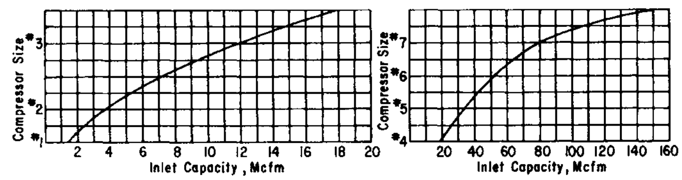


Figure 12-73. Centrifugal compressor size versus capacity. (Used by permission: Dresser-Rand Company.)

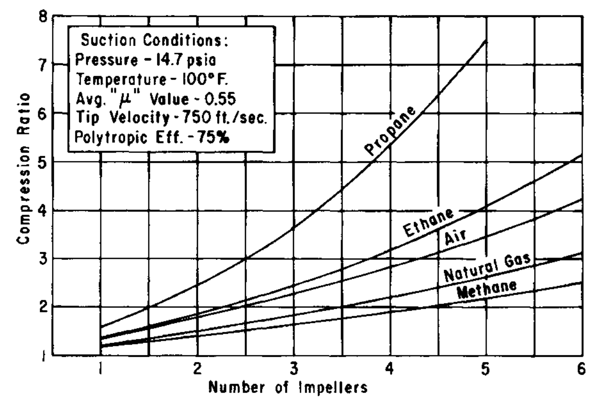


Figure 12-74. Compression ratio versus number of impellers; uncooled compression. (Used by permission: Dresser-Rand Company.)

show up with ratios less than 1.0. Values of 0.5–0.75 of M′ are usually used in design as efficiency falls off near M′ = 1.0. At M′ values of 0.9–1.0 and above, the compressor wheel ceases to produce additional pressure and flow. The flow has reached its maximum.

If the velocity of the gas/fluid equals or exceeds the speed of sound, shock waves are set up, and vibrations and other mechanically related problems may result, compared to the conditions when velocities are below the speed of sound.[81] For a Mach of 1.0, the gas velocity equals the velocity of sound in the fluid.

Specific Speed

Speed for centrifugal compressors as well as blowers, fans, pumps, and similar rotating and "pumping" equipment is a useful relative correlating factor. It is more important to detailed designers, although the concept is valuable in the performance evaluation of such equipment. At a given point, the *specific speed* correlates the important performance factors of adiabatic head, capacity, and rpm for geometrically similar wheels. The specific speed of all *geometrically similar wheels* is the same and does not change when the speed of the wheel size is changed. At the peak efficiency point, the wheels can be classified as to type and performance characteristics:[50, 47]

$$N_s = \frac{(\text{rpm})\sqrt{V_1}}{(H_a)^{0.75}} \tag{12-137}$$

where V_1 = actual flow rate, cfm at suction conditions
N_s = specific speed, dimensionless
H_a = Total polytropic head of wheel, ft-lb/lb (adiabatic or polytropic)
rpm = actual speed of rotation revolutions/min

Specific speed is defined as the speed in revolutions per minute at which an impeller would rotate if reduced proportionately in size so as to deliver 1 ft^3 of gas per minute against a total head of 1 ft of fluid.[81]

Centrifugal compressors usually have specific speeds of 1,500–3,000 rpm at the high efficiency point. The axial flow fans and blowers are high specific-speed wheels; mixed flow units are lower; and the centrifugal compressor wheels are the lowest in specific speed range because they have narrow impellers.

Compressor Case and Impellers

Tables 12-9A and 12-9B are a guide to a specific compressor case's capabilities. This is not a standard for each manufacturer. On the contrary, each is considerably different. Figure 12-73 is also useful as a guide to inlet suction condition capacities for various case sizes. These case sizes have no relation to the cases in Table 12-9.

Figure 12-74 is an approximate guide for the number of impeller wheels that will be required to develop the

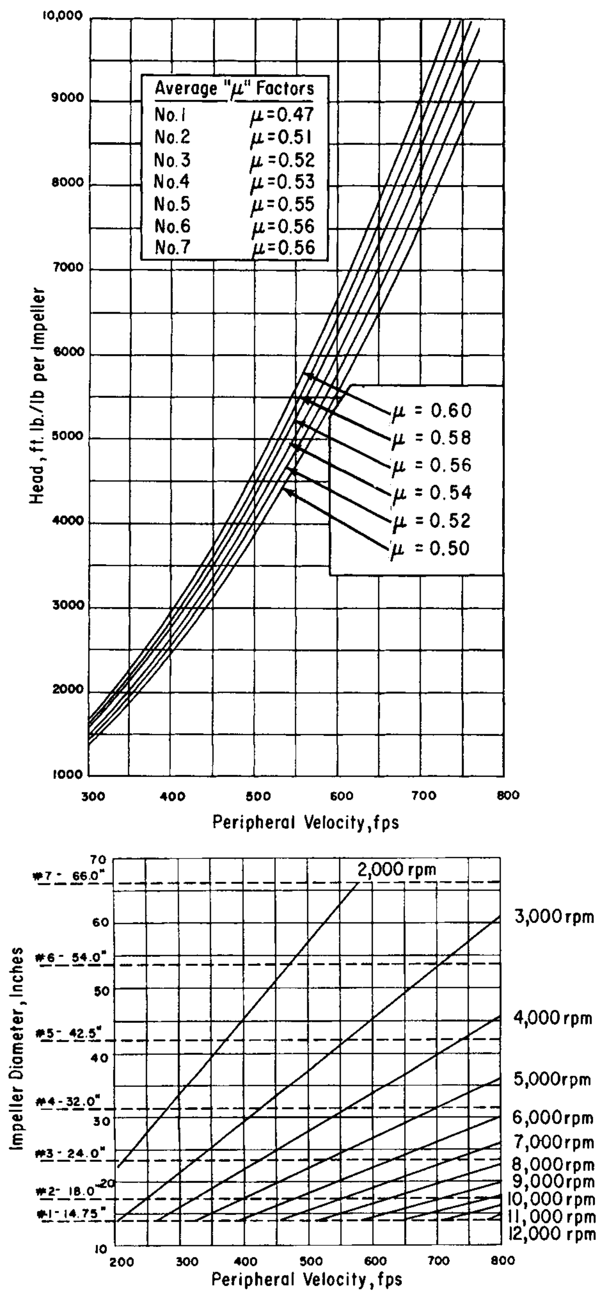


Figure 12-75. Peripheral velocity or impellar tip speed versus head per impeller. (Used by permission: Dresser-Rand Company.)

compression ratio, P_2/P_1, for a selected group of gases using uncooled compressors. That is, no diaphragm cooling or liquid injection and no exterior cooling between wheels. This is the most common condition. Figure 12-75 is also useful in estimating wheel selection and head/stage or wheel, and the case numbers here agree with those in Figure 12-73.

The need for and use of multiple inlets/outlets for centrifugal compressors becomes apparent when balancing process flow and pressure requirements. See the later discussion, and also refer to Figures 12-40B and 12-40D.

Example 12-10. Approximate Compressor Selection

An air compressor is required to raise 4,600 scfm of atmospheric air to 100 psig. The ambient summer temperature is 95°F dry bulb for two months and lower for the balance of the operating time. The air usually has a relative humidity of 65%, but during the "wet" season, the humidity may be 100% while the temperature is 95°F. The elevation is sea level; the barometer 14.7 psia. The continuity of air supply is very critical.

Approximate Selection for Preliminary Studies (Prior to Formal Inquiry to Manufacturers)

Basis. 100% relative humidity at 95°F due to critical service. (For other applications an R.H. of 80% might be quite satisfactory.)

1. Suction volume

   This volume, 4,600 scfm (14.7 psia and 32°F), is dry and must be increased by the water vapor that will accompany it into the compressor suction.[53]

$$V_w = V_d\left(\frac{P_1}{P_1 - P_v'}\right) \quad (12\text{-}138)$$

where P_1 = total pressure of system, psia
V_w = volume of gas containing condensable vapor (water), ft^3/min
V_d = volume of dry gas (no moisture), cfm
P_v' = $(P_v)(RH)$, psia (12-139)
P_v = vapor pressure of water vapor in the saturated gas at specified temperature, use steam tables, psia
RH = relative humidity, fraction
P_v = 0.8153 psia at 95°F
P'_v = (0.8153)(100) = 0.8153 (for 100% RH)

$$V_w = \left[(4{,}600)\left(\frac{14.7}{14.7}\right)\left(\frac{460 + 95}{460 + 32}\right)\right]\left(\frac{14.7}{14.7 - 0.8153}\right)$$

= 5,499 sfm at 14.7 psia and 95°F

Suction volume also

$$V_1 = Q_s = Wv = W(ZRT_1/144P_1) \quad (12\text{-}140)$$

where W = weight flow, lb/min
v = specific volume, ft^3/lb
R = gas constant, (ft-lb)/(lb)(°R)
T_1 = inlet temperature, °R
P_1 = inlet pressure, psia

2. Compression ratio

$$R_c = \frac{14.7 + 100}{14.7} = 7.8$$

This is too large for one wheel and indicates that an intercooler must be used between cases to cool the gas back to a reasonable temperature.
Assume a 3% pressure loss between cases due to intercooling. The actual overall compression ratio for each of two cases will be

$$R_{c1} = \sqrt{\frac{7.8}{0.97}} = 2.84$$

$$\text{or overall: } R_c = \frac{7.8}{0.97} = 8.04$$

3. Average molecular weight

$$\text{Percent water vapor} = \frac{(0.815)(100)}{14.7} = 5.55\% \text{ (volume)}$$

$$\begin{aligned} (0.0555)(18) &= 1.0 \\ (0.9445)(28.9) &= \underline{27.3} \\ \text{avg mol wt} &= 28.3 \quad \text{(use this on chart)} \end{aligned}$$

4. Polytropic head, (use chart in Figure 12-67)

For air, k = 1.40
Assume polytropic or hydraulic efficiency = 0.73
Reading chart, Figure 12-67, n = 1.65

From Figure 12-67, solving polytropic head equation:

At $R_c = 2.84$ and $t_1 = 95°F$
H = 40,000 ft per case (2 cases)

5. Discharge pressure from first case to intercooler

= (14.7)(2.84) = 41.7 psia

Pressure entering second cse:
Assume the 3% pressure loss (1% due to entrance and exit losses plus 2% due to intercooler and piping losses).
Suction pressure at second case:

= (0.97)(41.7) = 40.5 psia

6. Compression ratio across second case:

= 114.7/40.5 = 2.83

7. Required polytropic head from second case:

H_p = 40,000 ft

This neglects the effect of moisture removal on molecular weight, assumes constant "k" and "n" values, and assumes the gas is cooled back to 95°F as it enters the second case.

8. Suction volume to second case:

Assume intercooling of air down to 100°F.

$$\text{Volume} = \left[(4600)\left(\frac{14.7}{40.5}\right)\left(\frac{460 + 100}{460 + 32}\right)\right]\left(\frac{40.5}{40.5 - 0.9492}\right)$$

V = 1,945 at 100°F and 40.5 psia saturated.

Reading Figure 12-67, H_p = 40,500 ft when the suction temperature is 100°F.

9. Wheel selection
In actual design, the manufacturer uses wheel capacity data to properly select the sequence of wheels required to develop the head in each compressor case. Each wheel has its own efficiency at the rated speed (usually 70–75%).
From Table 12-9, for an intake volume of 5,499 cfm, Case No. D looks appropriate for the first case. Summarizing Case No. D:

No. stages per case max.: 7
Nominal overall eff, %: 77
Intake volume range: 3,500–12,000
Nominal head per stage, ft: 8,500
Nominal speed, rpm: 8,100
Max. case pressure: 250 psi cast iron
Number of wheels required per case before intercooling
= 40,000/8,500 = 4.7. Use 5 wheels in this case.

This requires a slight speed decrease or the selection of special impellers (at the rated speed) to ensure proper capacity and head.

Uncorrected approximate speed

$$= \text{nominal rpm} \sqrt{\text{required head/rated head}}$$

10. Uncorrected approximate speed

$$= (8100)\sqrt{\frac{40,000}{(5)(8,500)}} = 7,860 \text{ rpm}$$

This is acceptable.
By returning the air after intercooling to the sixth wheel in the same case (when the case is so designed), *or* into the first wheel of a new second case, the following conditions exist.
From Table 12-9A, the second case might be a No. E, based on inlet volume. Summarizing Case No. E:

No. stages per case max.: 8
Nominal overall eff. %: 73
Intake volume range: 1,500–4,500
Nominal head per stage, ft: 8,000
Nominal speed, rpm: 9,800
Max. case pressure: 250 psi
Number of wheels required per case = 40,500/8,000 = 5.06.
Use 5 wheels; the manufacturer can usually furnish wheels of sufficient capacity to make up the 1.2% increase in the five wheels.

11. Uncorrected approximate speed

$$= (800)\sqrt{\frac{40{,}500}{(5)(8{,}000)}} = 8{,}050 \text{ rpm}$$

If the case were running below 100% speed, experience shows general multiplying correction factors must be applied to

	Factor
Head	0.98
Efficiency	0.99

Corrections are necessary for the first case because its speed is below nominal.
Therefore *total* available head for Case 1:

= (0.98) (head summation of individual wheels)
= (0.98) [(5) (8,500)] = 41,700 ft

$$\text{Approximate speed case 1} = (8{,}100)\sqrt{\left(\frac{40{,}000}{41{,}700}\right)} = 7{,}920 \text{ rpm}$$

This is satisfactory.

12. For specific volume at suction conditions, use Figure 12-64.

Mol wt at suction = 28.3
Temp. = 95°F
Suction pressure = 14.7 psia
Reading chart, specific volume = 14.3 ft^3/lb

13. Weight flow

= (5,499)/(14.3) = 385 lb/min, entering suction of first wheel of the first case.

14. Brake horsepower

$$\text{bhp} = \frac{(W)(H)}{(33{,}000)(e_p)} + \text{Mechanical hp loss (from manufacturer's data for a specific case size)}$$

Case 1:

$$\text{bhp} = \frac{(385)(40{,}000)}{(33{,}000)(0.77)} + 25 \text{ (assumed as a reasonable value)}$$

$$= 630 \text{ hp}$$

If mechanical losses are assumed at 2%:

bhp = (630 − 25)/0.98 = 618

This is about as close as approximate methods will check.
Case 2:

Weight flow = 385 lb/min

$$\text{bhp} = \frac{(385)(40{,}500)}{(33{,}000)(0.73)} + 25 \text{ (assumed)} = 665 \text{ hp}$$

Total horsepower = 630 + 665 = 1,295 bhp.

15. Alternate brake horsepower calculation:
Using Figure 12-63 for air.

At k = 1.40
(n − 1)/n = 0.378

$$\text{Solving: } \frac{n-1}{n} = 0.378 = \frac{k-1}{ke_p} = \frac{1.4-1}{1.4e_p}$$

$e_p = 0.755$
n = 1.61

Case 1, using Figure 12-68:

bhp/MMCFD = 76 at R_c = 2.83, 14.7 psia and 95°F.
Flow at suction conditions:

$$\text{MMCFD} = \frac{(5{,}499)(60)(24)}{10^6} = 7.91$$

Uncorrected bhp = (7.91) (76) = 602 hp (assumes 75% eff.)
Correction factor from Figure 12-69.
Suction volume = 5,499 cfm
Mcfm = 5.499

Reading curve:

Factor = 1.008
Corrected bhp = (1.008) (602) = 607 hp

Case 2:

bhp/MMCFD = 76
Suction to second case (actual conditions are 40.5 psia and 100°F):

$$\text{MMCFD} = \left[\frac{(1{,}945)(60)(24)}{10^6}\right]\left(\frac{40.5}{14.7}\right) = 7.72 \text{ at } 100°\text{F and } 14.7 \text{ psia}$$

Uncorrected bhp = (7.72)(76) = 587

From Figure 12-69, correction factor at 1.945 Mcfm

Factor = 1.033
Corrected bhp = (1.033)(587) = 607 hp
Total horsepower = 607 + 607 = 1,214 hp

Actually, because the alternate scheme is based on 75% efficiency, if the values are corrected for the differences in efficiency of the two individual compressor cases used, the results will be close enough for engineering application. As they stand, the bhp values cannot be resolved to a more accurate basis without specific data on case selection, efficiency, and losses.

16. Discharge temperatures
From Case 1 with 95°F suction temperature, $R_c = 2.84$

$$\text{Exit temperature} = T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(n-1)/n}$$

Use Figure 12-22 because this is an uncooled case, and compression is closer to polytropic.

At n = 1.61
$R_c = 2.84$
Temperature rise factor = $R_c^{(n-1)/n} = 1.48$
Exit temperature = (460 + 95)(1.48) = 822°R = 362°F
From Case 2, with 100°F suction temperature, $R_c = 2.83$
$T_2 = (100 + 460)(2.83)^{(n-1)/n} = 560(2.83^{0.378})$
$T_2 = (560)(1.48) = 830°R = 370°F$

If the case had been cooled internally, then the isentropic "k" value can be used to calculate the temperature rise.

17. Estimating the number of wheels using Figure 12-73

First Case:

The polytropic head previously calculated = 40,000 ft
At an inlet capacity to the case of 5,499 cfm:
Figure 12-73 reads greater than size No. 2, so use size No. 3.
From Figure 12-75, it appears that about 9,000 ft/stage is a reasonable maximum. Number of impellers based on this 9,000 ft head per impeller = 40,000/9,000 = 4.45, use 5. This is the same number reached by the other approach.
Head per stage = 40,000/5 = 8,000 ft
From Figure 12-75 the approximate wheel diameter for (first compressor) case size no. 3 is 24 in. dia.

Peripheral Velocity. Use Figure 12-75.

Because individual wheel information is not usually known, use μ (pressure coefficient) = 0.55; however, the data for case size No. 3 gives $\mu = 0.52$.

Case 1: at 8,000 ft/wheel and $\mu = 0.52$
peripheral velocity = 710 ft/sec
Case 2: at 8,100 ft/wheel, using case size No. 2 (Figure 12-73 for the second case).
Peripheral velocity = 720 ft/sec at $\mu = 0.51$ for case No. 2.

Speed. From Figure 12-75, the rpm is approximately 6,800 rpm for a (first compressor) size No. 3 case. This is not unusual for different manufacturers to operate at significantly different speeds, as this is a function of wheel design.

Quick Approximation Method for Centrifugal Compressor Performance

Cole[15] has summarized a basic approximation for evaluating compressor selection. Although certain details of design selection are not included, the results are quite good (within 2–5%) for the average application and will establish the required size unit within engineering estimating accuracy. It should be used for compression ratio greater than about 6 because this would not normally be designed in equipment used to establish the curves. This can be controlled by not allowing a discharge temperature greater than 450°F.

A. For Horsepower.

1. Use Figure 12-76.
Intake volume must be expressed at cfm/1,000, with cfm at *actual* suction pressure and temperature.
Read "basic horsepower" at compression ratio, R_c.

2. Read "k" value correction multiplier from Figure 12-77.

3. Required bhp at coupling,

$$\text{bhp} = (\text{basic horsepower})(\text{"k" multiplier})(\text{actual intake pressure, psia}/14.5)[(Z_1 + Z_2)/2] \quad (12\text{-}141)$$

Note that compressibility Z may be neglected for many conditions, but if used Z_1 is at intake condition and Z_2 is at discharge.

B. Number of Stages.

1. Use Figure 12-78.
Read "basic head" in thousands of ft using R_c and intake temperature.
2. Read "k" value correction multiplier from Figure 12-77.
3. Polytropic head

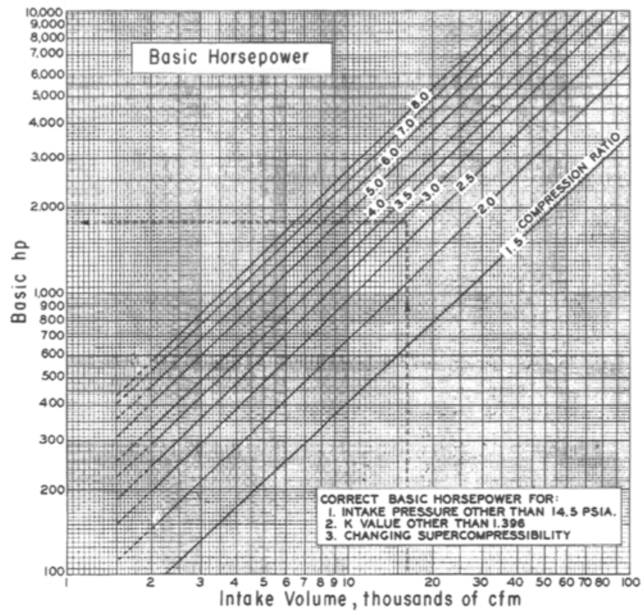


Figure 12-76. Basic horsepower for a machine with intake of 14.5 psia, with "k" value for air (1.396) and supercompressibility factors neglected. (Used by permission: Cole, S. L. *Oil and Gas Journal,* V. 58, No. 6, ©1960. PennWell Publishing Company. All rights reserved.)

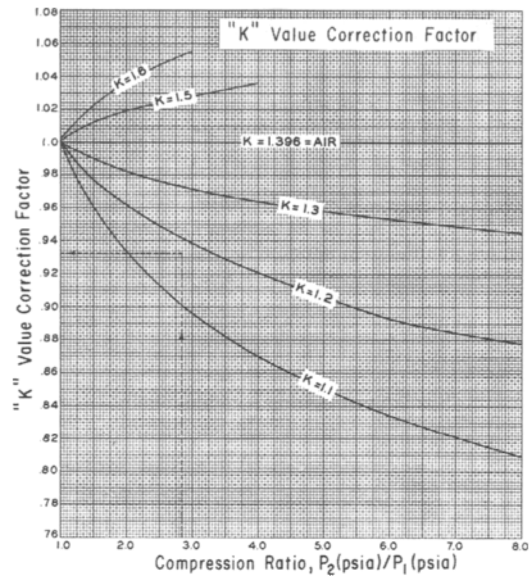


Figure 12-77. "k" value correction factor. (Used by permission: Cole, S. L. *Oil and Gas Journal,* V. 58, No. 6, ©1960. PennWell Publishing Company. All rights reserved.)

$$H_p = (\text{basic head})(\text{“k” multiplier})\ (28.95/\text{mol wt})[(Z_1 + Z_2)/2] \quad (12\text{-}142)$$

4. No. of stages = polytropic head/9,500 (12-143)

C. Discharge Temperature.

1. Use Figure 12-79.

Read the "temperature rise multiplier." Use "k" values.

2. Temperature rise = (intake temperature, °R)(multiplier)
3. Discharge temperature,

°F = (temp. rise) + intake temp., °F (12-144)

D. Compressor Speed (Use Figure 12-80).

Some experience indicates that the speed may be 5–10% lower than the lower curve, particularly in the low intake volume region, less than 10,000 cfm.

Operating Characteristics

Probably the most important of the fundamentals concerning centrifugal compression equipment is an understanding of the basic operating characteristics. Although some basic mathematical relations should be kept in mind, the graphical representation makes these points easier to understand. Figure 12-61 is a representation of typical oper-

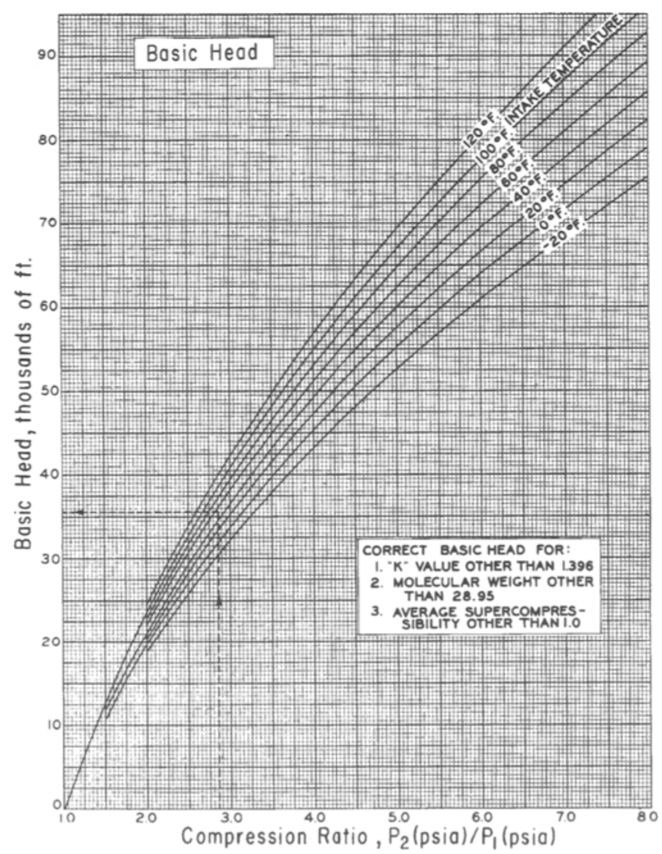


Figure 12-78. Basic head for machine with "k" value for air (1.396), molecular weight for air (28.95), and supercompressibility factors neglected. (Used by permission: Cole, E. L. *Oil and Gas Journal,* V. 58, No. 6, ©1960. PennWell Publishing Company. All rights reserved.)

ating curves for centrifugal compressor performance when the unit is driven by a steam turbine. Usually when a machine is purchased, only one curve representing the 100% speed line is furnished to the operator. The manufacturer will, on request, furnish the other informational curves to give a better picture of practical performance.[115] These points can be calculated by the engineer from the 100% curve and the design point. The fundamental impeller operating efficiencies and performance characteristics should be known for a very exact representation. These can be obtained from the manufacturer.[98] References 115 and 116 present discussion of the same topic, but with varying detail.

Because the general characteristic performance of a steam turbine and a centrifugal compressor are quite similar, they pair up excellently for many process applications. The effect of variation of speed on capacity and brake horsepower is shown. At the surge point or limit, the operation of the machine is unstable. The machine vibrates and heats up if run at or near this point. Usually the surge point is designated to be $^1/_3$ to $^1/_2$ of the normal operating capacity of the unit. This will vary with some designs and may be $^1/_4$ on the lower extreme to $^2/_3$ on the upper extreme. It is important to know where this surge point (or region) is on any centrifugal machine. During start up, the machine must pass through this region, but it is important for the preceding reasons not to stop and allow the machine to run here, but rather to bring the speed and capacity on through.

Usually a constant speed, motor-driven compressor will be equipped with inlet guide vanes to the first wheel to allow suction volume control. If the guide vanes are not used, some other throttling device should be available. Figure 12-62 presents the constant speed performance of a centrifugal machine with inlet guide vanes.

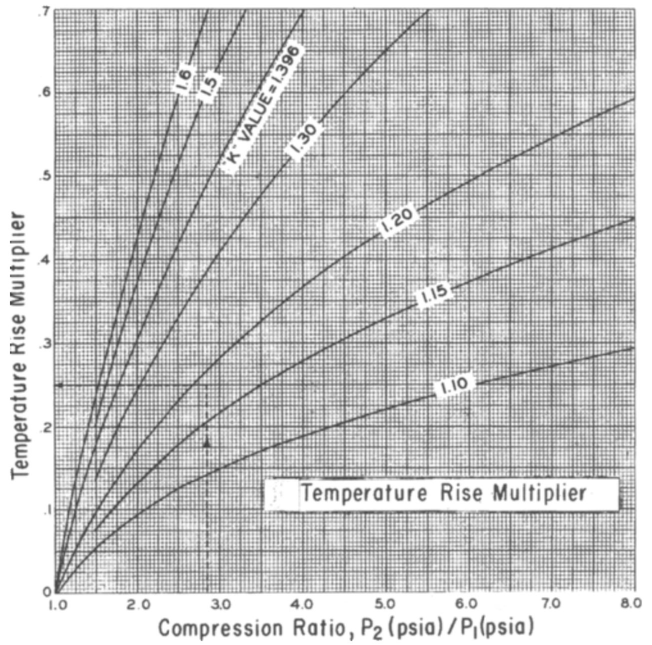


Figure 12-79. Temperature rise multiplier. (Used by permission: Cole, S. L. *Oil and Gas Journal,* V. 58, No. 6, ©1960. PennWell Publishing Company. All rights reserved.)

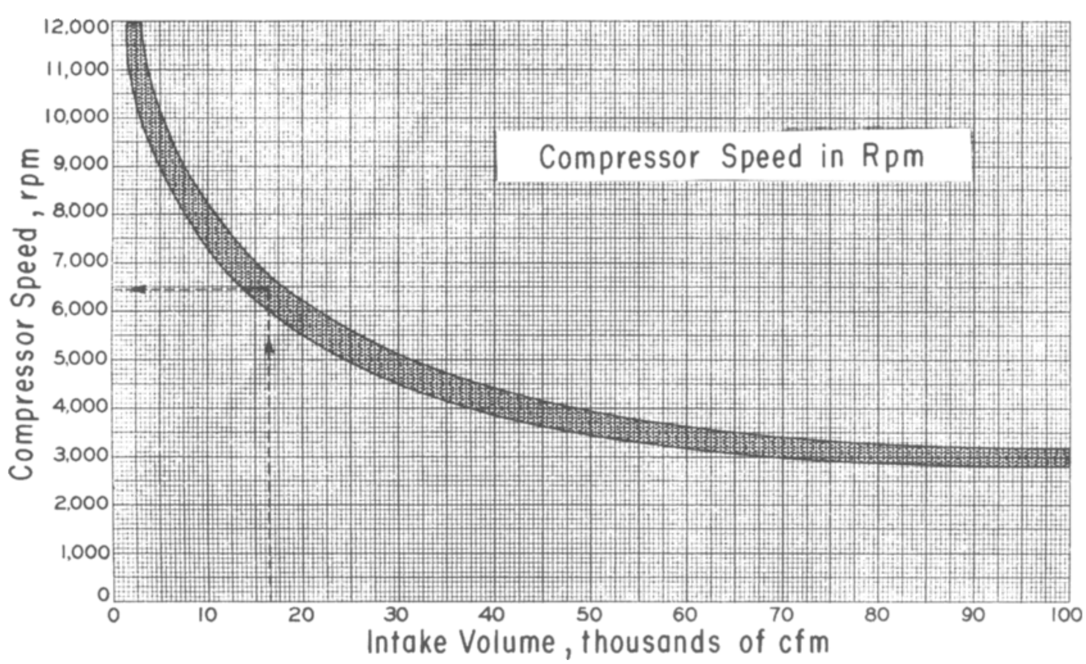


Figure 12-80. Compressor speed in rpm. (Used by permission: Cole, S. L. *Oil and Gas Journal,* V. 58, No. 6, ©1960. PennWell Publishing Company. All rights reserved.)

Affinity Laws or Fan Laws

The affinity laws express the relationship between the head, capacity, speed, and size of centrifugal blowers and compressors. In general these relations can be applied to inlet volume conditions for good preliminary designs, but all final designs apply these laws to the actual discharge volumes from the impeller.[50, 115]

It is *essential* to remember that these "laws" do not act independently of one another, but a correction/change in one variable requires an evaluation of the other factors to properly present the new/corrected/revised performance of the compressor wheels.

A. Speed.

1. The capacity varies as the speed for fixed diameter:

$$V_2 = V_1\left(\frac{(\text{rpm})_2}{(\text{rpm})_1}\right) \tag{12-145}$$

Where sub-1 represents the first, and sub-2 the second condition of operation.

Speed cannot be increased indefinitely due to mechanical stresses developed in the rotating impeller and due to the limit of Mach 1 for the tip speed and gas velocity. The limitations will vary according to the impeller designs of the various manufacturers.

Table 12-16 (presented in the fan section of this chapter) indicates a more complete examination of the variables and their effects on performance of the rotating devices, such as centrifugal compressors and fans.

V = capacity, ft^3/min

2. The polytropic head varies as the speed squared

$$H_{p2} = H_{p1}\left(\frac{\text{rpm}_2}{\text{rpm}_1}\right)^2 \tag{12-146}$$

3. The theoretical horsepower varies as the speed cubed

$$\text{bhp}_2 = \text{bhp}_1\left(\frac{\text{rpm}_2}{\text{rpm}_1}\right)^3 \tag{12-147}$$

A "new" or calculated point of performance is not defined until all three factors have been determined. It is not sufficient to determine one point and draw conclusions, but rather the total effect of the change must be considered.

B. Impeller Diameters (Similar).

For two geometrically similar, same family, impeller wheels with the same specific speed and operated at the same rpm,

1. The head varies as the impeller diameters squared.

$$H_2 = H_1\left(\frac{D_2}{D_1}\right)^2, \text{ft} \tag{12-148}$$

2. The capacity cfm varies as the impeller diameters cubed.

$$V_2 = V_1\left(\frac{D_2}{D_1}\right)^3 \tag{12-149}$$

3. The brake horsepower varies as the impeller diameter to the fifth power.

$$\text{bhp}_2 = \text{bhp}_1\left(\frac{D_2}{D_1}\right)^5 \tag{12-150}$$

C. Impeller Diameter (Changed).

When an impeller diameter is *reduced*, but the speed is held constant,

1. The head decreases as the impeller diameter squared.

$$H_2 = H_1\left(\frac{D_2}{D_1}\right)^2 \tag{12-151}$$

Note that diameter, D_2, is always smaller in this series of evaluations than D_1, the original size of the impeller before cutting or reduction in diameter.

2. The ft^3 per min decreases as the ratio of impeller diameters cubed.

$$\text{cfm}_2 = \text{cfm}_1\left(\frac{D_2}{D_1}\right)^3 \tag{12-152}$$

3. The brake horsepower decreases as the impeller diameters to the fifth power.

$$\text{bhp}_2 = \text{bhp}_1\left(\frac{D_2}{D_1}\right)^5 \tag{12-153}$$

These relations do not hold closely for large impeller cuts, as the head and capacity drop a little faster than the relations indicate. Allowance should be made by a "trial-and-error" approach when actually reducing an impeller size. Efficiency will remain nearly constant during all of the changes discussed.

D. Effect of Temperature.

For constant intake volume, compressor speed, efficiency, and no throttling, but with discharge pressure changing to reflect the effect of temperature:

$$\text{bhp}_2 = \text{bhp}_1\left(\frac{T_1}{T_2}\right) \tag{12-154}$$

Affinity Law Performance

For single-stage or single wheel compressors (sometimes referred to as blowers) the effect of changing inlet or discharge conditions can be rather closely predicted. However, in the case of the multistage machines, the variations in performance as well as the effects of preceding wheel conditions can have a marked effect on the final wheel performance. The last stage wheel gives the performance curve for the machine, but it is valid only when operating in conjunction with the other wheels. The first wheel discharges into the suction of the second; the second discharges into the suction of the third; etc., until the last stage is reached, and it represents the output of the machine. As the gas passes through the machine, its specific volume decreases, and each impeller, therefore, is usually smaller in gas passageway than the preceding one.

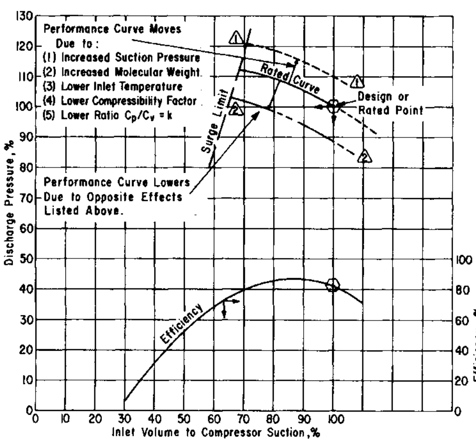


Figure 12-81. Effect of changing inlet conditions on performance curves. Fixed operating speed. (Adapted by permission: Hancock, R. *Chemical Engineering,* V. 63, No. 6, ©1956. McGraw-Hill, Inc. All rights reserved.)

Head-Capacity Changes. For a given constant speed the characteristic operating curve is fixed. The pressure differential between discharge and suction will change with varying suction or system conditions.

The pressure differential will increase for any condition that causes *increased* suction inlet gas density:[28]

1. Increased molecular weight.
2. Increased inlet pressure.
3. Lower or decreased inlet temperature.
4. Lower or decreased gas compressibility factor.
5. Lower or decreased gas "k" value (results in increased gas density entering the diffuser).

Curve 1-1 of Figure 12-81 represents the effect of the *increased* density inlet gas for a fixed operating speed.

Curve 2-2 results from a decrease in the gas density as might be represented by the factors listed. Note that these Curves 1-1 and 2-2 might represent the new rated curve for a particular set of inlet operating conditions. Because most processes cannot fix the gas analysis and system conditions exactly, it is important to recognize the possible implications of changes in the suction conditions on the compressor performance.

Figures 12-82 and 12-83 illustrate the effects of temperature and gas density changes on the pressure rise for a constant-speed operation.

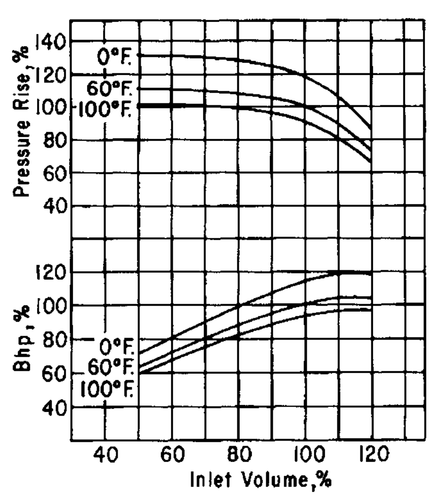


Figure 12-82. Relative effects of inlet temperature change on head and horsepower. (Used by permission: A C Compressor Corporation.)

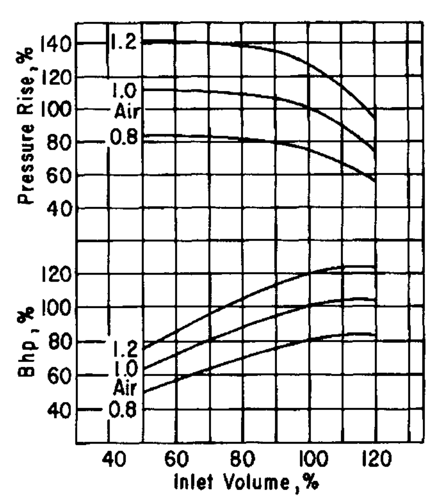


Figure 12-83. Relative effects of inlet gas specific gravity change on head and horsepower. (Used by permission: A C Compressor Corporation.)

Constant-speed operation will be encountered,

1. With a motor drive, with or without a gear box.
2. With another drive (such as a turbine), with control set for constant-speed operations.

Variable-Speed Operation. Each of the performance curves shown in Figure 12-61 is for a different speed. This illustrates that the characteristics retain their "family" pattern with speed changes.

The affinity laws will define all the calculations of performance for the different conditions. Thus, every point on the 100% speed capacity-head curve can be adjusted to the 105% or the 95% speed curves by these laws.

The Operating System. Regardless of calculated centrifugal compressor performance, the machine will operate only on or along its operating curve to fit the system of which it is a part. This is quite similar to the system performance of a centrifugal pump. Friction, other pressure drops of the system, and how friction varies with operating conditions determine machine performance.

Figure 12-84A[28] illustrates a system with all line friction. Here, the operations would follow the system curve and operate at the intersection with the speed curve. For example, if the speed is cut 10%, the flow decreases 8%. Figure 12-84B shows a system with essentially constant back pressure. Following the operating system curve shows that a small speed cut back of say 10% results in a flow drop of 40%.

Figure 12-84C represents a system comprised of both friction and back pressure resistance. In this case a speed drop of 10% creates a flow drop of 15%. These examples are for constant suction conditions.

For a variable-speed machine and a resistance comprised of friction and back-pressure, Figure 12-85 should be typical.[53]

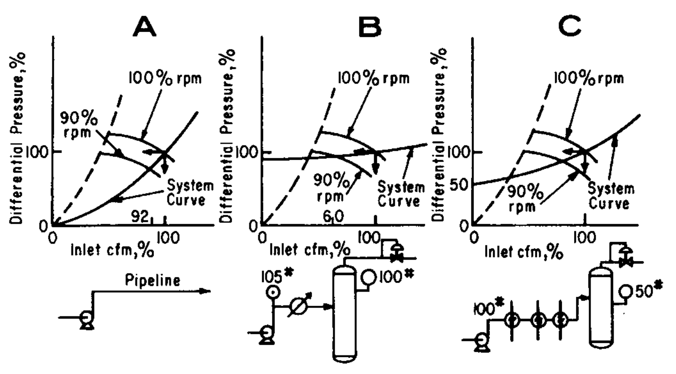


Figure 12-84. Effect of gas system flow resistance on comparative performance in same centrifugal compressor. (Used by permission: Hancock, R. *Chemical Engineering,* V. 63, No. 6, ©1956. McGraw-Hill, Inc. All rights reserved.)

Here again it is important to remember that the compressor will operate only at the points of intersection of the system friction curve and the operating curves for the various speeds. The friction curve is obtained by calculating a few points for given flows using reliable methods for pipeline and equipment pressure drop calculations. This then is what the machine must operate against. At 100% speed the machine operates at Point 1. If the flow is to be reduced to 95% of Point 1's capacity, by the affinity laws, Point 5 would be calculated as x% rpm. The compressor cannot operate at Point 5 because the system head loss curve does not go through Point 5. The curve goes through Point 2, but Point 2 is less than 95% of capacity. Therefore, the machine must operate at about y% of rated speed, Point 3, in order to give 95% of rated capacity.

At any constant or steady speed of operation of a compressor, the head-capacity and efficiency curves are characteristic of the impeller and casing design only. These curves that are determined by test can be translated to other reasonable speeds and conditions of operation of the wheel-casing combination of the affinity laws. The operation of the compressor must meet or establish the desired point on the head-capacity-system curve, which requires a combination of controls.

Control of Performance.

1. Speed control
   Figure 12-86 shows the effect of changing speed on the compressor characteristic curve. The speed can be adjusted to meet a desired point on the system curve. This is the most popular form of control.

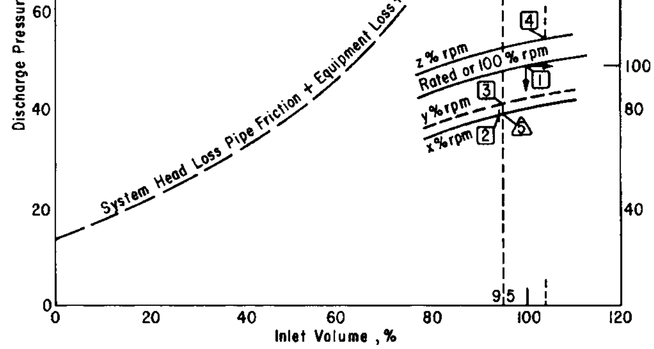
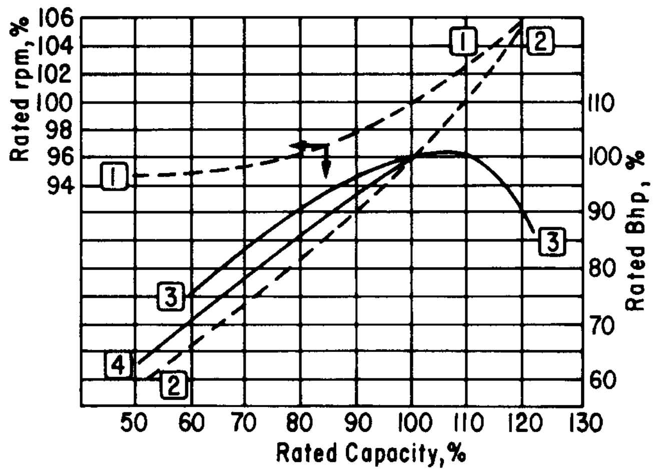


Figure 12-85. System operation of variable-speed centrifugal compressor.



1 - 1 Capacity versus Speed Change
2 - 2 Horsepower Change as Speed 1 - 1 Changes
3 - 3 Horsepower Change using Discharge Throttling at a Constant Speed
4 - 4 Horsepower Change using Suction Throttling at a Constant Speed
All Curves for Compression Ratio of 8 : 1

Figure 12-86. Typical effects of capacity control on horsepower for centrifugal compressors. (Used by permission: Stepanoff, A. J. *Turbo-blowers,* ©1955. John Wiley & Sons Inc. All rights reserved.)

2. Throttling inlet
This is also a very common and simple way to vary the capacity, particularly when using a constant-speed drive. The gas density at the inlet is reduced by the throttling action, but the important point of the system characteristic after the discharge remains unchanged. Throttling takes place at constant enthalpy. An energy loss occurs with this operation; however, it is much less than the loss associated with throttling on the discharge side of the machine. "The pumping capacity in terms of inlet cfm (or weight of flow) is reduced in proportion to the density decrease, whereas the pumping capacity based on the impeller discharge volume remains the same."[50]
3. Throttling on discharge
The power consumption remains constant for this type of operation; thus, no savings are effected by a reduced flow. The pumping point remains unchanged as contrasted to a reduced point for inlet throttling.
4. Other schemes
Stepanoff[50] discusses (a) operation at capacities below the pumping point, (b) recovery gas turbine, (c) cut-off capacity control, (d) double-flow blower, (e) power wheels, (f) adjustable diffuser vanes, (g) adjustable impeller vanes, and (h) two-speed gear increasers.

Example 12-11. Changing Characteristics at Constant Speed

A constant speed air compressor has been designed for the following conditions:

P_1 = 14.7 psia inlet
t_1 = 90°F inlet
V_1 = 12,000 cfm inlet (at actual temperature and pressure)
P_2 = 38 psia discharge
e_a = 70%
Rel. hum. = dry = 0%

After operating at essentially design conditions for one year, the process has been changed to have the inlet temperature drop to 50°F. What will be the new operating conditions: (1) discharge pressure or (2) bhp? The suction volume remains the same at 12,000 cfm.

1. Ratio of compression,

$$R_c = \frac{38}{14.7} = 2.58$$

2. Adiabatic head for initial operations,

$$H_a = \frac{53.5}{\text{sp. gr.}}(T_1)\left(\frac{k}{k-1}\right)\left[\left(\frac{P_2}{P_1}\right)^{(k-1)/k} - 1\right] Z_1$$

$$= \frac{53.3}{1.0}(460) + 90)\left(\frac{1.396}{1.396-1}\right)[(2.58)^{(0.396)/1.396} - 1]$$

$$= 32{,}000 \text{ ft (adiabatic)}$$

3. Brake horsepower for initial operations

$$\text{bhp} = \frac{(W)(H)}{(33{,}000)e_p}$$

From Figure 12-65, by trial-and-error interpolation, when $e_a = 0.7$, $e_p = 0.735$.
Using Figure 12-67 to obtain polytropic head,

H_p = 34,000 ft
Sp. vol. = 13.85 ft^3/lb at suction conditions

$$\text{lb/min} = \frac{12{,}000}{13.85} = 867$$

Gas horsepower,

$$hp_g = \frac{(867)(34{,}000)}{(33{,}000)(0.735)} = 1215$$

Shaft bhp = $hp_g/0.99$
= 1,215/0.99
= 1,230

Alternate bhp by adiabatic calculation:

$$bhp = \frac{P_1V_1\,(k/k-1)[(P_2/P_1)^{(k-1/k)}-1}{229e_a}$$

$$= \frac{14.7(12{,}000)(1.396/1.396-1)[(2.58)^{(1.396-1)/1.396}-1]}{229\,(0.70)}$$

$$= 1201$$

4. Discharge pressure for new conditions
Speed remains constant, and head remains the same.

$$32{,}000 = \frac{53.1}{1.0}(460+50)$$

$$\left(\frac{1.396}{1.396-1}\right)\left[\left(\frac{P_2}{P_1}\right)^{(0.369)/1.396}-1\right]$$

$$\left(\frac{P_2}{P_1}\right)^{0.284} - 1 = 0.335$$

$$\left(\frac{P_2}{P_1}\right)^{0.284} = 1.335$$

$$\frac{P_2}{P_1} = (1.335)^{1/0.284} = 2.74$$

Discharge pressure:

$P_2 = (2.74)(14.7) = 40.3$ psia, with 50°F suction temperature

5. Brake horsepower at 50°F, assuming constant polytropic efficiency.

Specific volume = 12.7 ft^3/lb at 50°F

$$lb/min = \frac{12{,}000}{12.7} = 946$$

From Figure 12-65, by trial-and-error interpolation, when $e_a = 0.7$, $e_p = 0.735$. Using e_p in the horsepower equation, convert adiabatic head to polytropic head using Figure 12-67.

H = 34,000 ft. (polytropic)

$$hp_g = \frac{(946)(34{,}000)}{33{,}000(0.735)} = 1{,}326$$

Shaft bhp = 1,326/0.99 = 1,339

For comparison, at 45°F, constant intake volume, efficiency, speed, and no discharge throttling:

$$bhp = 1{,}230\left(\frac{550}{510}\right) = 1{,}328$$

This compares with 1,339 bhp calculated by the gas horsepower relation. The values should agree exactly, and the difference may lie in the reading of charts to determine the polytropic head. Using the bhp calculated with the adiabatic efficiency, e_a,

$$bhp = 1{,}202\left(\frac{550}{510}\right) = 1{,}298$$

for the new condition at 50°F suction.

Example 12-12. Changing Characteristics at Variable Speed

The original design for a centrifugal compressor was as follows.

Gas: process gas, chlorine mixture
Condition: bone dry
P_1 = 10.98 psia inlet
t_1 = 100°F inlet
V_1 = 9,600 cfm at inlet
P_2 = 20.7 psia discharge
rpm = 7,840
bhp = 466
k = 1.33
Gas mol wt: 69.8
Sp. gr. at inlet: 2.4

If process conditions change, and the gas temperature drops to 80°F, what will be the new speed to keep a constant discharge pressure at an intake volume of 8,000 cfm? Refer to Figure 12-87.

1. Ratio of compression

$R_c = 20.7/10.98 = 1.888$

2. Adiabatic head at initial design conditions

$$H_{ad} = \frac{53.3}{2.4}(460+100)\left(\frac{1.33}{1.33-1}\right)[(1.888)^{(1.33-1)/1.33}-1]$$

$$= 8{,}560 \text{ ft}$$

3. New discharge pressure
Assuming a constant head from the impeller at 80°F.

$$8{,}560 = \frac{53.3}{2.4}(460+80)\left(\frac{1.33}{1.33-1}\right)[(P_2/P_1)^{0.248}-1]$$

$$(P_2/P_1)^{0.248} - 1 = 0.1775$$

$$(P_2/P_1)^{0.248} = 1.1775$$

$$(P_2/P_1) = (1.1775)^{1/0.248} = 1.932$$

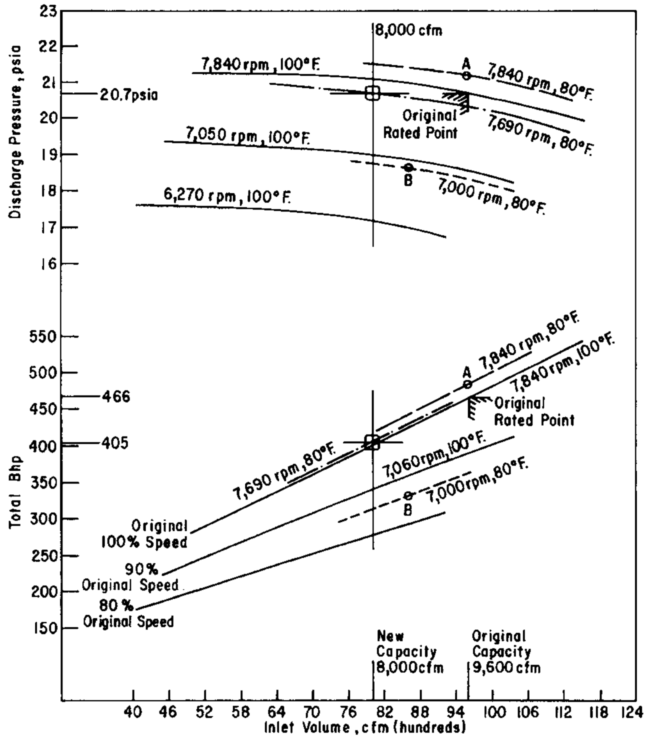


Figure 12-87. Changing characteristics with variable speed.

$P_2 = (1.932)(10.98) =$ 21.1 psia with 80°F inlet temperature and a speed of 7,840 rpm. Plotted as Point A on Figure 12-87.

4. Brake horsepower

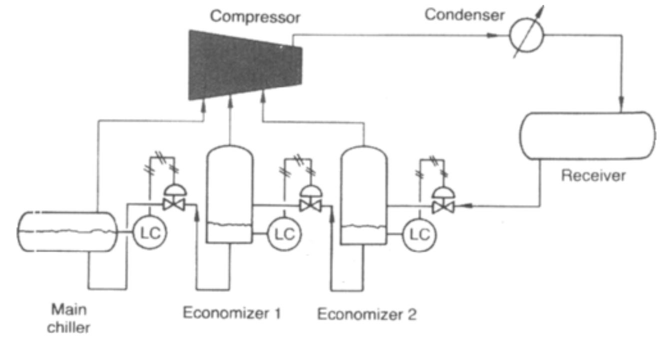
$\text{bhp} = 466\left(\dfrac{560}{540}\right) =$ 483 for 80°F inlet temperature and a speed of 7,840 rpm.

5. To establish new bhp and performance curves, do the following:

Try 7,000 rpm and relate to the 7,840 rpm and 80°F curve
Vol at 7,000 rpm = 9,600(7,000/7,840) = 8,580 cfm
H_{ad} at 7,000 rpm = $8{,}560(7{,}000/7{,}840)^2$ = 6,800 ft
New discharge pressure:
$6{,}800 = (53.3/2\,.4)(460 + 80)\,(1.33/1\,.33 - 1)[(P_2/P_1)^{0.248} - 1$

$(P_2/P_1)^{0.248} - 1 = 0.1408$
$P_2 = (1.7)(10.98) = 18.68$ psia
bhp at 7,000 rpm and 80°F = $483(7{,}000/7{,}840)^3$ = 331 hp
7,000 rpm is 89.5% of rated speed.
Plot point B at 8,580 cfm, 331 bhp, 8,580 cfm, and 18.68 psia.
Draw the new lines parallel to existing curves.

By continuing to calculate new values for pressures and horsepowers for several points of cfm from the 7,840 rpm, 80°F curve, the new curve for the 7,000 rpm, 80°F may be completed.



6. Obtain the desired rpm for 80°F and 8,000 cfm by close calculation of many points or by interpolation from the new curves as established.
   By cross-plotting and interpolation, for 8,000 cfm at 20.7 psia discharge, and 80°F inlet temperature:

Approximate speed = 7,690 rpm
Approximate bhp = 405 hp

This can now be verified by calculation at the estimated point.

Side Load Compressors. The design of centrifugal compressors with more than one process stream entering and leaving the unit is common and is used extensively for units handling gaseous refrigerants from ethylene through propane and many other specialty applications. Peters[102] presents a discussion of the performance expected when various process gas streams enter a compressor with multiple wheels on a single shaft. See Figures 12-40B–D for physical illustrations. Some of the streams may be just an exit of the gas for cooling between wheels with the same process gas re-entering the unit and flowing on to the exit, and some streams may in effect be mixing with the initial gas entering the unit as shown in Figure 12-88.[102]

All operating parameters must be considered to make certain that an acceptable operating compressor and process "fit" exists. The key parameter must be evaluated as part of an overall operating analysis and considered independently. Peters[102] points out that the API Specification 617, the ASME codes, and other applicable codes may require some modification when applied to side load compressors.

The following list of parameters may have a direct effect on compressor selection:

- Head rise to surge, surge margin, and overload margin. See Figure 12-89.

- Head per compression section.
- Compressor parasitic flows, (i.e. balance piston leakage, etc.).
- Excess margins on other process equipment.

Referring to the first bullet item, depending on other parameters specified, this addition of information may have no effect or little effect or may result in nonoptimum compressor selection. Referring to Figure 12-89, a 5% head rise to surge will result in nonoptimum efficiency and overload,[102] but the 2% level will yield the best efficiency and overload selection.

A refrigeration system as shown in Figure 12-88 is one of the most used applications of side-load compressors and requires almost constant discharge pressure. Figure 12-90 indicates that with a required 5% HRTS, the compressor would trip off stream at 97% of design flow. Even 2% would make the surge margin immaterial. For some process applications, the overall flow ranges 40–50%, and the range on a refrigeration system may be closer to 20–30%. Thus, imposing an excessive HRTS and/or surge margin criteria may result in minimal overload capacity as noted in Figure 12-90.[102]

Head per section is another important parameter, as the smaller the number of impellers per section, the higher the head required per compressor stage. This leads to higher rotative speeds and operation at higher Mach numbers, which in turn, tends to limit the operating range, flatten the head rise characteristic, and reduce efficiency.[102]

Compressor parasitic flow involves the re-entry of the seal equalization line flows into the main gas stream. If such flows are significant, re-entry at a point other than the main flow may be appropriate for better performance. The true operating point on the compressor section (group of wheels) characteristic may be considerably different when the side (or parasitic flow) is introduced. This situation can be modified by introducing the flow into the first side load.[102]

When too many factors of safety are added to the true required design flows, excess margins may affect compressor stability. When the compressor manufacturer begins the design, therefore, unjustified excess capacity flows exist. Be realistic in regard to building in flexibility or capacity.

Davis[108] discusses the evaluation of multistage compressors for conversion to new or different process applications.

Troubleshooting. Gresh[108] presents useful mechanical troubleshooting factors to consider in setting up a centrifugal compressor. An important item is the coupling connecting the compressor to the driver (electric motor, steam turbine, gas turbine, gear train, etc.). Several good quality couplings carry the horsepower load between the driven compressor shaft and the driver or driver's gear unit. These should be carefully evaluated for each specific application.

Expansion Turbines

Expansion turbines are related in many design features to the centrifugal compressor. The key exception being that the turbine receives a high pressure gas for expansion and power recovery to a lower pressure and is usually accompanied by the recovery of the energy from the expansion. For example, applications can be (1) air separation plants; (2) natural gas expansion and liquefaction (for gas let-down in pipeline transmission to replace throttle valves where no

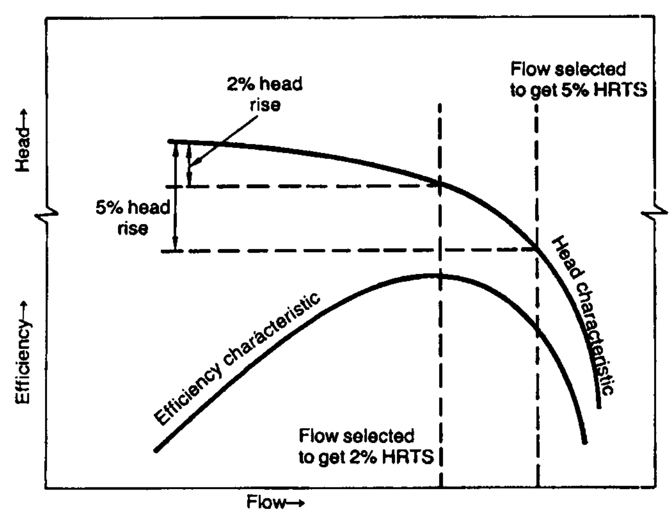


Figure 12-89. Effects of 2% and 5% "Head Rise To Surge" (HRTS). (Used by permission: Peters, K. L. *Hydrocarbon Processing,* V. 60, No. 5, p. 171, ©1981. Gulf Publishing Company. All rights reserved.)

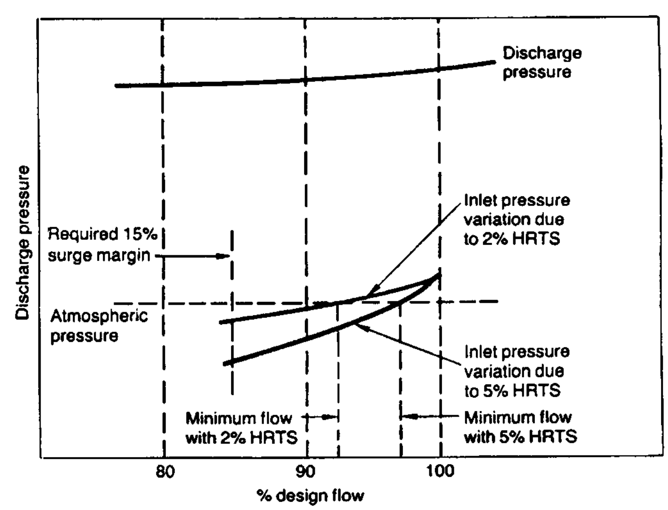


Figure 12-90. Effects on inlet pressure for HRTS variations (constant speed drives). (Used by permission: Peters, K. L. *Hydrocarbon Processing,* V. 60, No. 5, p. 171, ©1981. Gulf Publishing Company. All rights reserved.)

energy is recovered, see Figure 12-91); (3) generator applications to generate electricity; and (4) waste heat recovery applications that convert the waste to electricity or another useful energy.[119, 120]

A Mollier Diagram is useful for the expansion of a specific gas/vapor or multicomponent vapor fluid. See Figure 12-91 for comparison of (1) constant enthalpy (Joule-Thompson effect), isenthalpic, and (2) isentropic (constant entropy), which provides the colder temperature.[120] Note that the expander indicated on the figure is somewhere between isenthalpic and isentropic or polytropic. See Figure 12-92.[21, 61]

For best performance, contact the respective manufacturers to present the full energy recovery program for their design and evaluations. The choices of alternate power recovery approaches can be valuable in the final evaluation and performance of a process.

Axial Compressor

The axial compressor is usually a single inlet, uncooled machine consisting essentially of blades mounted on a rotor turning between rows of stationary blades mounted on the horizontally split casing. A typical unit is shown in Figures 12-93, 12-94, and 12-95. The stationary blades can be either fixed or movable, Figure 12-96. The movable blades allow for better control of and increased flexibility in operations. Figure 12-97 shows a rotor assembly. Most units will have inlet guide vanes for at least the first row of blades and may have exit vanes. The general size of these machines is often much larger than the centrifugal compressor, although this is not necessarily a firm condition. The casings require extremely good casting to obtain the shapes usually associated with the arrangements of these machines.

The sealing systems for the shaft are quite similar to those for the centrifugal; however, internal shaft seals are not necessary between stages. The materials of construction are similar to those for the centrifugal compressor.

Operating Characteristics

The typical operating characteristic of the axial machine is shown in Figure 12-98, for a fixed stator blade unit capable of operating at variable speed.

Figures 12-99, 12-100, and 12-101 show approximate speed-capacity comparisons with the centrifugal machine. Each stage consists of one stationary plus one rotating blade row. Figure 12-101 shows an axial compressor performance handling 100 psia air.

The operation of an axial compressor accomplishes approximately one half its pressure rise as the gas passes through the stationary blades and the other half as it goes through the rotating blades. The static pressure and kinetic energy increase as the gas goes through the machine. The analogous comparison between an axial and a centrifugal machine is stationary blades to diffuser and diaphragm and rotor blades to impeller wheel.

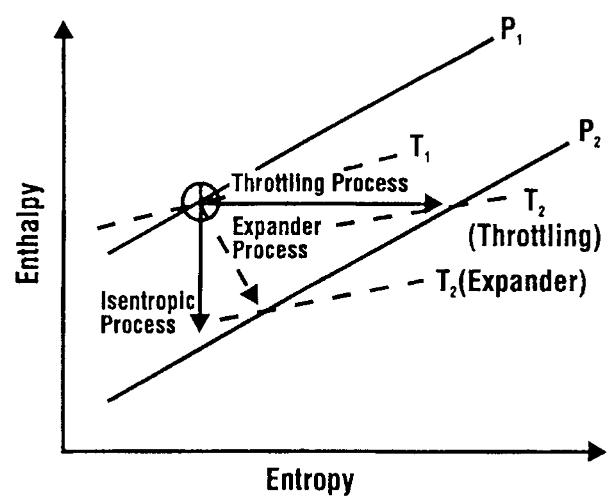


Figure 12-91. Comparison of the energy potential of turboexpanders versus throttle valves. (Used by permission: Bul. 2781005601. ©Atlas Copco Comtec, Inc.)

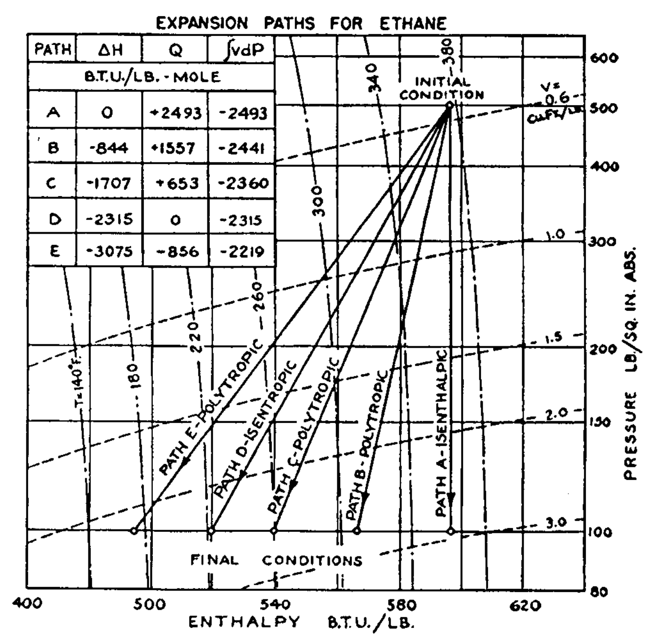


Figure 12-92. Section of ethane pressure-enthalpy diagram illustrating five expansion paths. (Reprinted by permission: Edmister, W. C. *Applied Hydrocarbon Thermodynamics*, p.66, ©1961. Gulf Publishing Company. All rights reserved.)

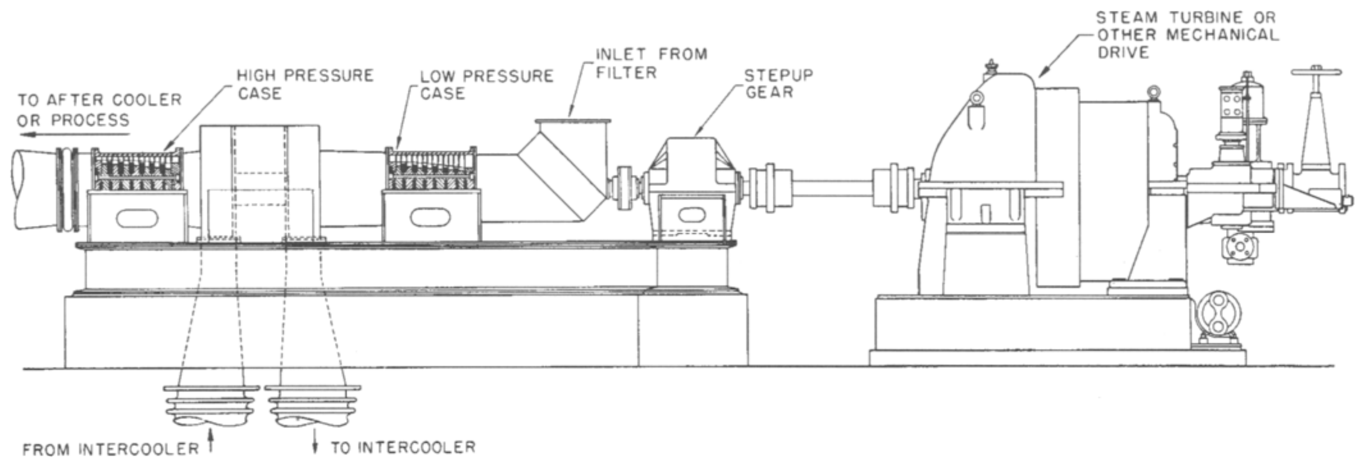


Figure 12-93. Typical geared turbine-driven axial flow compressor unit. (Used by permission: Dresser-Rand Company.)

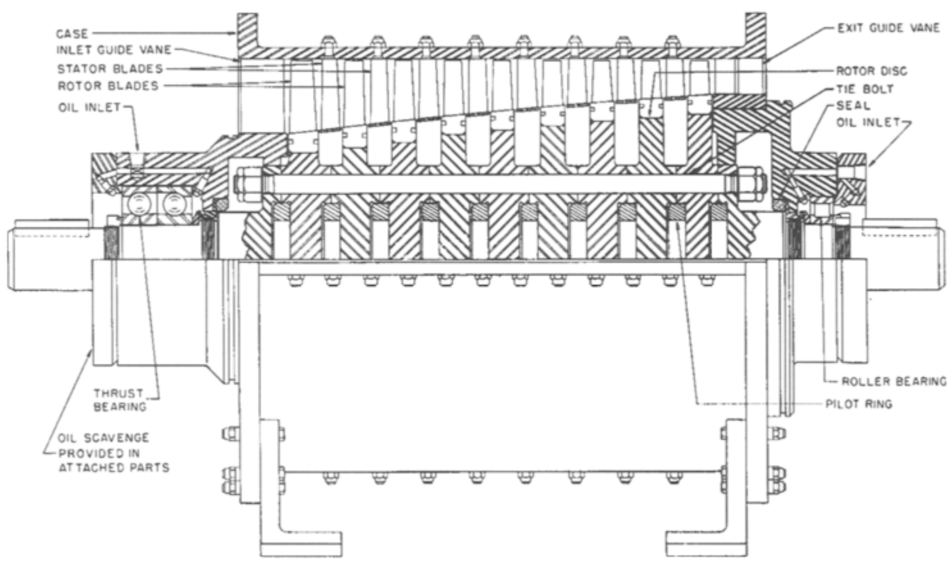


Figure 12-94. Cross section of typical nine-stage axial flow compressor. (Used by permission: Dresser-Rand Company.)

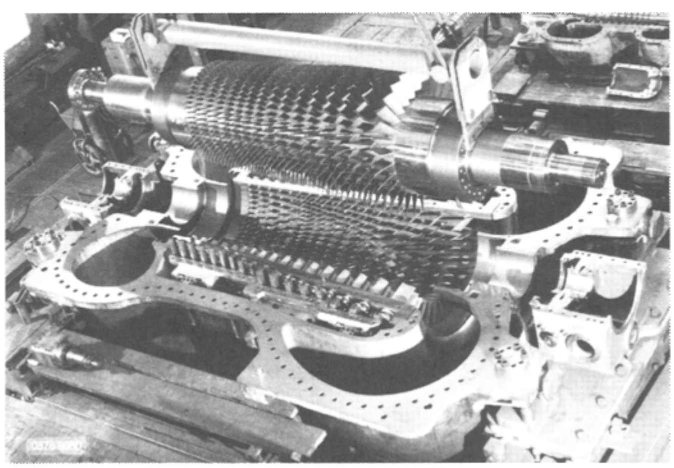


Figure 12-95. Axial compressor Type AV 100-16, during erection. Note stationary and rotating blades. Two identical steam turbine-driven machines supply air to blast furnace at steel works. Suction volume = 560,000 Nm^3/h; discharge pressure = 6.2 bar; power input = 52,000 kW each. (Used by permission: Bul. 26.13.10.40-Bh$_i$. ©Sulzer Turbo Ltd.)

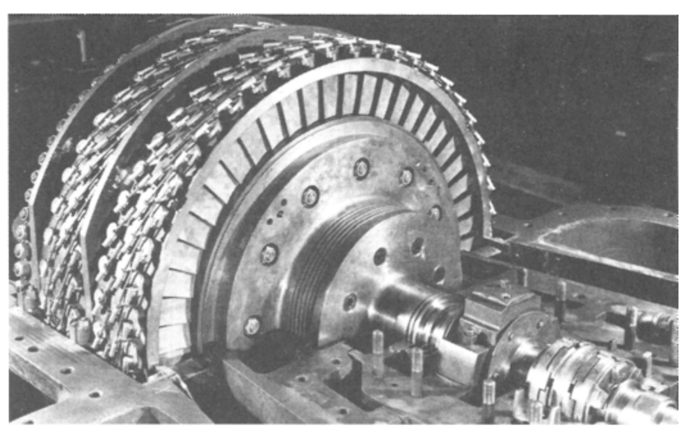


Figure 12-96. Stator blade control mechanism on four-stage axial compressor. (Used by permission: A C Compressor Corporation.)

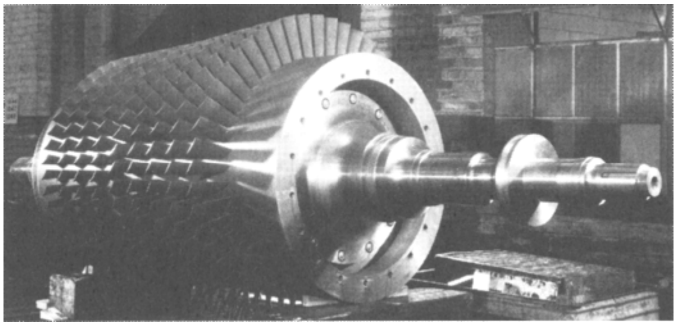


Figure 12-97. Axial rotor assembly. (Used by permission: A C Compressor Corporation.)

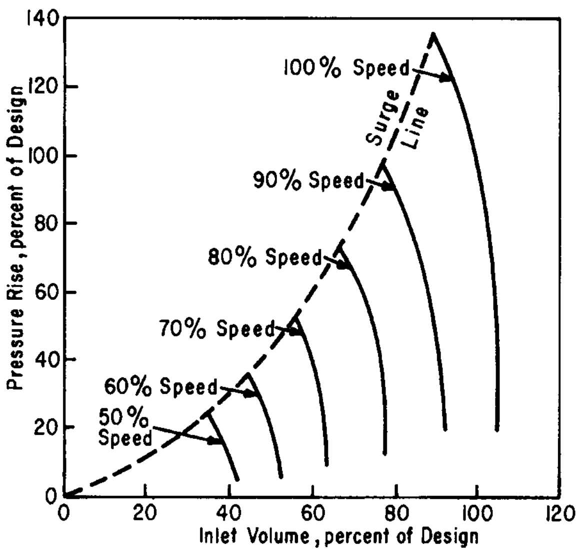


Figure 12-98. Performance of axial compressor at various speeds. (Used by permission: Claude, R. E. *Chemical Engineering,* V. 63, No. 6, ©1956. McGraw-Hill, Inc. All rights reserved.)

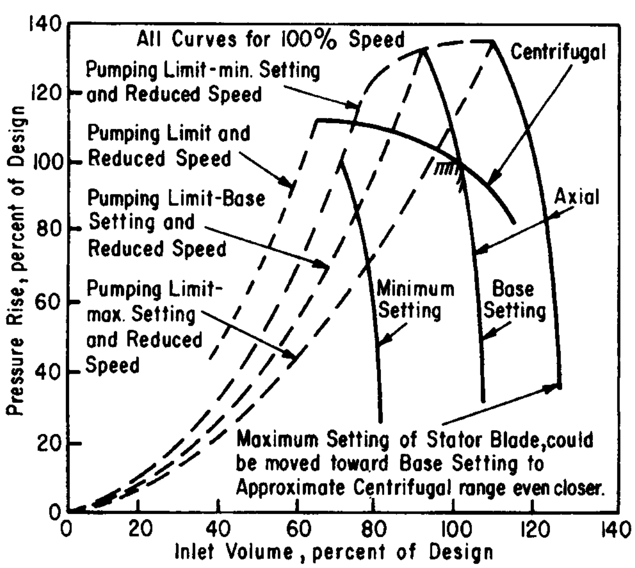


Figure 12-99. Stable volume range extended by stator blade control. (Used by permission: Claude, R. E. *Chemical Engineering,* V. 63, No. 6, ©1956. McGraw-Hill, Inc. All rights reserved.)

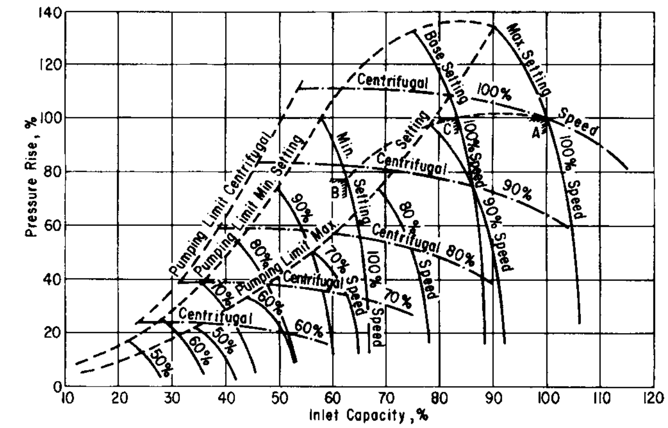


Figure 12-100. Pressure-capacity characteristic comparison of an axial compressor with adjustable stator blades with a centrifugal compressor. (Reprinted with permission: Lowell, W. O. *Petroleum Refiner,* V. 34, No. 1, ©1955. Gulf Publishing Company. All rights reserved.)

Gas Velocities

General guide lines for good design practice[14] indicate an axial velocity for air of 300–450 ft/sec. For other gases, the axial velocity range is in direct proportion to the speed of sound of the gas compared to air. The internal shape of the machine is usually arranged to give constant gas velocity as the gas travels through.

Stages

As a general rule-of-thumb, the axial compressor will require about twice as many stages for a given requirement as the centrifugal compressor.[14] The maximum number of axial stages is approximately 16. The temperature rise limitations as well as structural problems also limit the maximum stages for a given application.

Volume

The size is determined by the inlet volume. The unit will generally be smaller than the equivalent-rated centrifugal compressor.[14] The larger the inlet volume, the more advantage develops for the axial machine.

The lower volume limit is approximately 5000 cfm, but the upper limit practically does not exist. That is, units have been built to handle well above a million cfm.

Horsepower

The horsepower characteristic is shown in Figure 12-102.

Efficiency

The efficiency of the axial is about 8–10% higher than the comparable centrifugal. Thus, the driver power requirements are lower for this type of unit.[14]

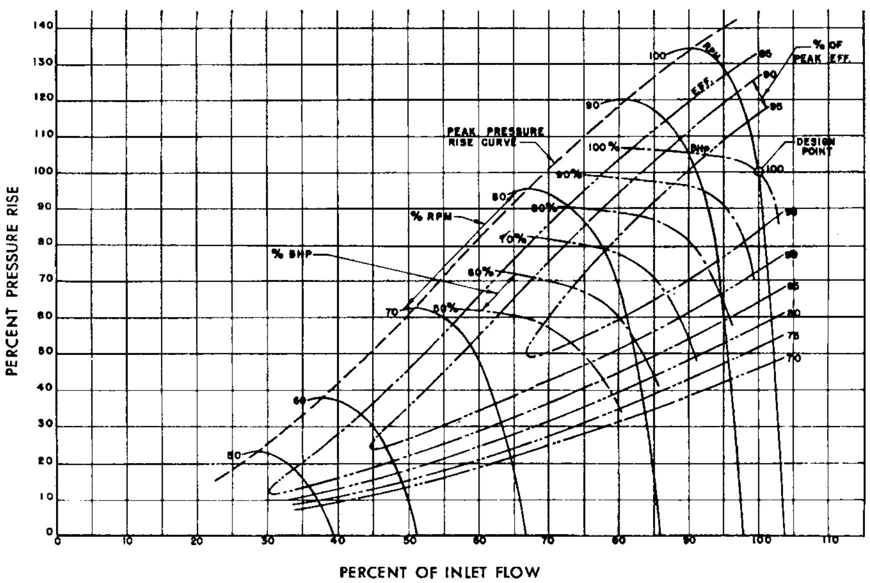


Figure 12-101. General axial-flow compressor performance for typical 100 psia air compressor. (Used by permission: Dresser-Rand Company.)

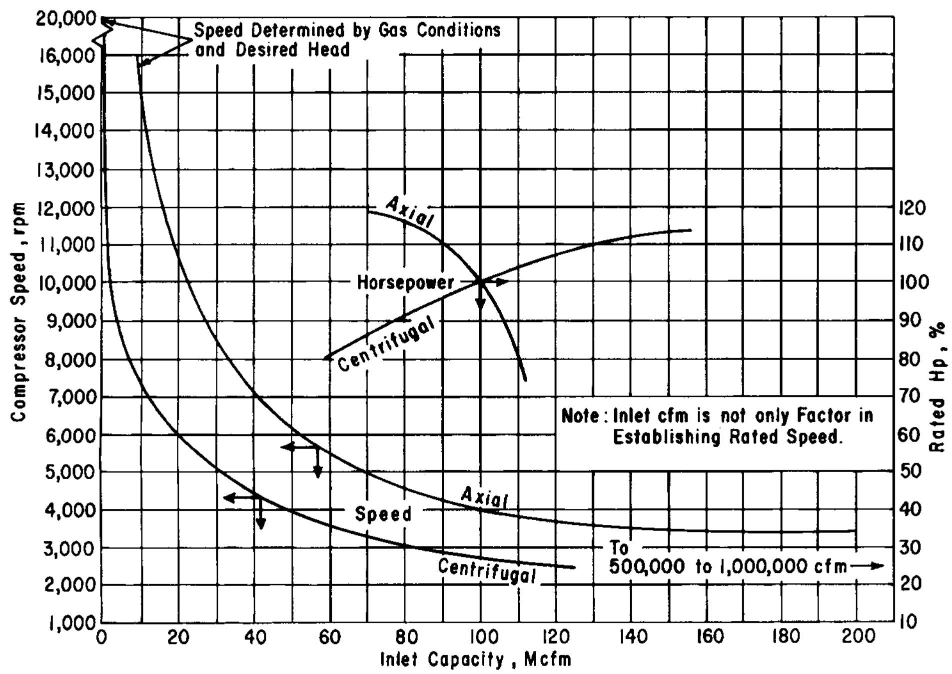


Figure 12-102. Speed comparison between axial and centrifugal compressors. (Used by permission: A C Compressor Corporation.)

Liquid Ring Compressors

This type of compressor is unique in that a centrifugal action on the sealing liquid creates a reciprocating type of action on the gas or vapor being handled. Figures 12-103A–C and 12-104 show the multibladed rotor rotating in a ring of liquid in the elliptical casing. "The eccentric path of the revolving liquid ring produces an out and in, or reciprocating radial motion to each liquid piston relative to its rotor cylinder, which revolves about the fixed center."[2] As the unit operates, the liquid discharges with the gas or vapor through the discharge ports, and at the same time, make-up or seal liquid is admitted to keep the pump complete with the proper amount of liquid.

Note in Figure 12-103A that the various operational sections of the functioning unit are indicated; however, these sections may vary mechanically between the competing manufacturers, but the general concept is the same.

Generally this type of unit serves to compress gases from vacuum up to atmospheric pressure. Often after initial startup, the unit must *pump-down* the system (such as a closed process system of vessels) by gradually bringing down the

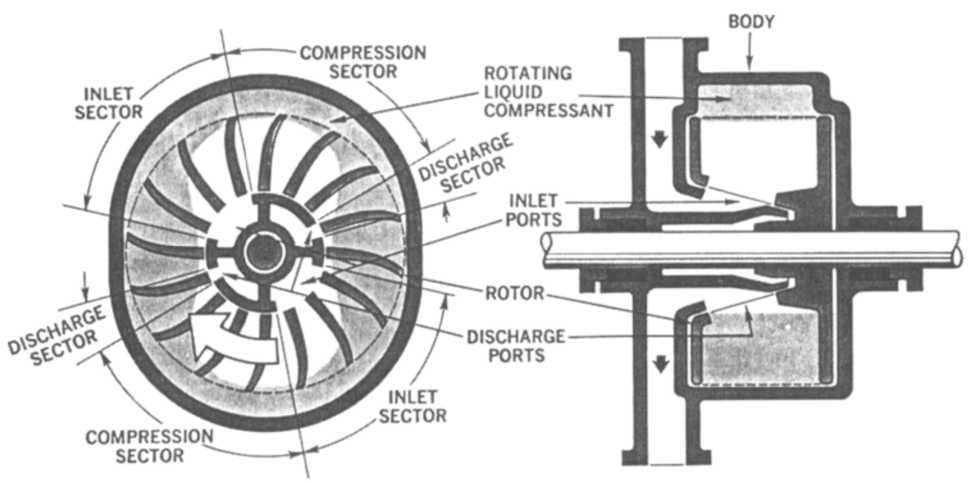


Figure 12-103A. Functional operational schematic of Nash liquid ring compressor. (Used by permission: Bul. 474-D, p. 3. ©Nash Engineering Co.)

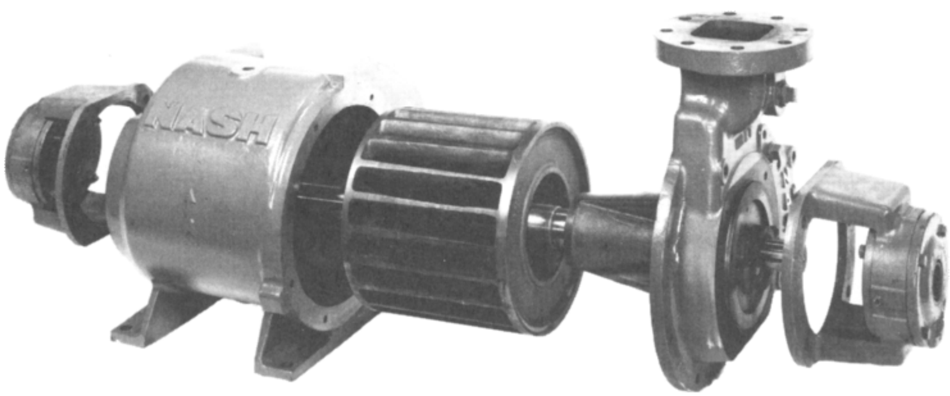


Figure 12-103B. A partly disassembled Nash liquid ring compressor view shows the pump body, the rotor, the cone, and the separate bearing blocks. (Used by permission: Bul. 778-A2/89, p.5. ©Nash Engineering Co.)

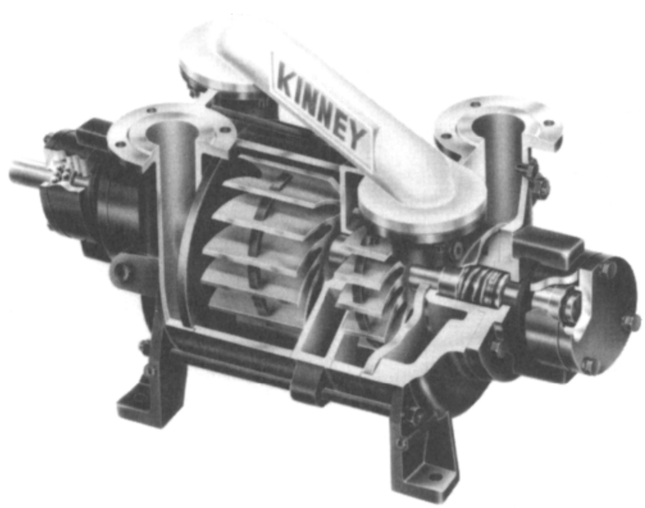


Figure 12-103C. In a compound pump, the gas is compressed in two stages. First, the gas is compressed in the larger low pressure section and then transferred through the cross-over and internal passageways to be compressed in the smaller, high pressure stage. (Used by permission: Form 4114. ©Kinney Vacuum Division, Tuthill Corporation.)

absolute pressure to the desired operating level and then maintaining the status of the system by discharging to atmospheric pressure for air. In the case of the process gas or vapor compressor design, the maximum operating pressure may reach 130 psig.

For a more complete discussion of this and other systems using this style of vapor compression, see Chapter 6, Volume 1 of this series. Useful references are Huff[34] and Patton.[43]

Operating Characteristics

Capacity

Compressors are available for vacuum as well as positive pressure service. Vacuums are obtained to 29 in. Hg abs and pressures up to 125 psig, with volumes of 2–25,000 cfm.

Temperature Rise

The temperature rise during compression when using water as the sealing liquid is approximately 4°F for vacuum

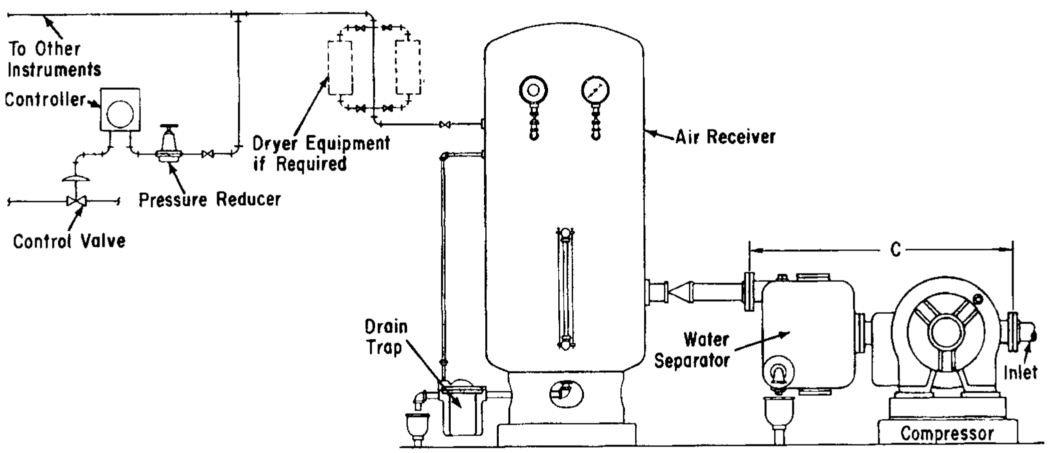


Figure 12-104. Liquid ring compressor as gas compressor. (Used by permission: Nash Engineering Company.)

service to 21 in. mercury vacuum and entering air not greater than 59% saturated.[2] At vacuums above 21 in., the temperature rise is approximately 4°F plus 2°F for each in. of vacuum greater than 21 in.

For a compression operation handling some condensable vapors, the temperature rise will be approximately 13°F. These values are considerably lower than for conventional polytropic compression. They are almost isothermal and demonstrate the contact cooling that takes place during compression.

Seal Liquid

The seal liquid may be water or any other liquid suitable for cooling, sealing, or even neutralizing the vapors being compressed.

Horsepower

The horsepower is established by the manufacturer by testing the various types and models. In general, the horsepower requirements will be a combination of the power to pump the liquid inside the compressor casing plus the power to compress the gas or vapor. If a recirculating seal liquid system is used, the recirculating pump horsepower is not reported as a part of the compressor requirements.

Applications

The outstanding applications include:

1. Oil-free process gas, such as air for sanitary uses, etc.
2. Hazardous and toxic gases.
3. Hot gases and vapors.
4. Jet and surface condenser.
5. High vacuum, single-stage to 27 in. Hg vacuum; two-stage to 29 in. Hg vacuum.
6. Solvent recovery from seal liquid.
7. Non-pulsating flow.

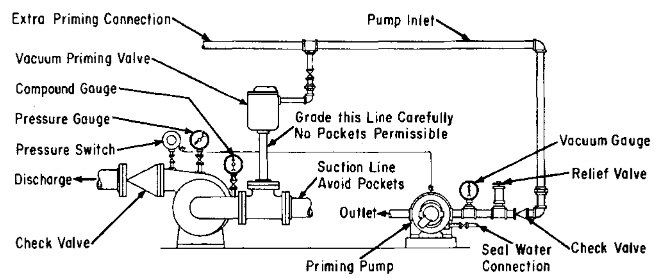


Figure 12-105. Automatic operation of primer by pressure control. (Used by permission: Nash Engineering Company.)

Figures 12-104 and 12-105 illustrate system connections for gas compression and vacuum service.

Rotary Two-Impeller (Lobe) Blowers and Vacuum Pumps

For a more detailed discussion, see Chapter 6, V. 1 of this series.

Figures 12-106, 12-107, and 12-107A show an exploded view of a typical positive displacement blower. The impeller lobes rotate in opposite directions on parallel mounted shafts, Figure 12-107. One shaft serves as the drive shaft and drives the other through the gears. A timing hub allows for adjusting the timing angle of the lobes. The rotation may be either for upward, downward, or side gas flow. When liquid is entrained in the gas, the downward flow is preferred to assist in case drainage.

The lobes do not touch each other or the casing during operation. The separation is a few thousandths of a inch. There is no internal lubrication; thus, the units can operate in a dry gas service.

The bearings and gears are mounted externally to the gas chamber and are separated by stuffing box seals to prevent gas leakage along the shaft. The packing is usually vented to the suction, so that it is necessary to pack only against suction pressure.

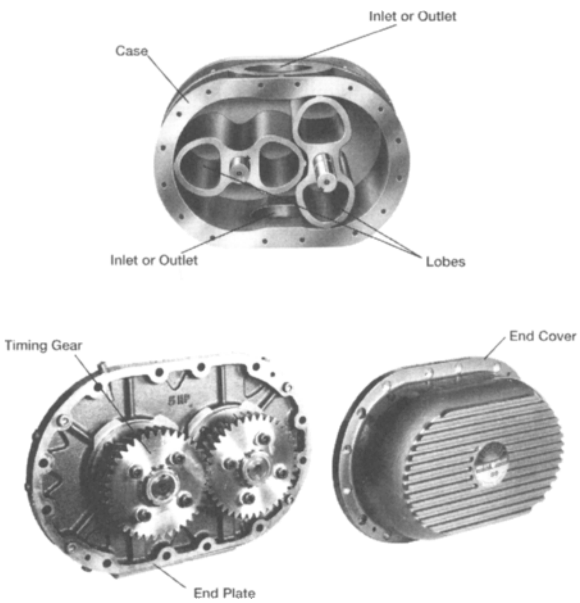


Figure 12-106. Lobe-type blower construction. (Used by permission: Sutorbilt Corp., Div. Gardner Denver Corporation.)

For atmospheric air intake, an air filter and silencer are usually required. The air filter is to keep out dirt in the air; the silencer is to reduce the local noise level.

For additional details, see Chapter 6, V. 1, of this series.

Construction Materials

Most units are good-grade cast iron for the casing and lobes. Shafts are high-grade carbon, alloy steel, or stainless steel. Conventional packing boxes are usually satisfactory Because operating pressures are not extremely high.

Performance

The discharge pressure on these constant-volume machines (at constant speed) is determined by the system pressure. No suction or discharge valves are on this type of machine, as it is not designed for a specific pressure. Being positive displacement in principle, it discharges to match the controlled discharge side pressure. They may be operated as vacuum pumps or compressors. The rotating lobes push the constant volume of gas trapped between the lobes and the casing out the discharge opening. With each revolution, the blower delivers a fixed amount of gas measured at the inlet conditions. The pumping or compression cycle repeats four times for every revolution of the drive shaft.

Due to the operating clearances, some gas "slip" occurs. That is, some gas slips by the clearances back to suction as the lobes rotate, creating a loss in pumping efficiency. This slip is larger for high discharge pressure and low gas densities. The rate of gas delivery is

$$\text{cfm} = (\text{blower rpm} - \text{slip in rpm})(D)', \text{ft}^3/\text{min} \qquad (12\text{-}155)$$

where D' = blower displacement, ft^3/rev

The slip is constant at constant pressure, and at high pressures and low speeds, the slip becomes a considerable percentage of the total capacity. Lower volumetric efficiencies result under these conditions.

The discharge flow may have some pulsing characteristics depending upon the blower speed; the lower speed exhibits more pulsing.

Typical performance curves for this type of machine are given in Figures 12-108 and 12-109 for constant-speed operation. The capacity and bhp for other constant speeds are also represented. Note that the capacity increases as the speed, and the bhp varies directly with the speed and pressure. Increasing the speed against a constant pressure increases the volume pumped by an amount directly related to equation 12-155.

As a guide, the temperature rise for rotary lobe units is 13°F per psi pressure. The maximum discharge pressure usually ranges from 14–18 psig and depends entirely on the back pressure into which the unit is discharging to develop its required head. The discharge pressure is limited by the design strength of the casing and other parts and by the available horsepower.

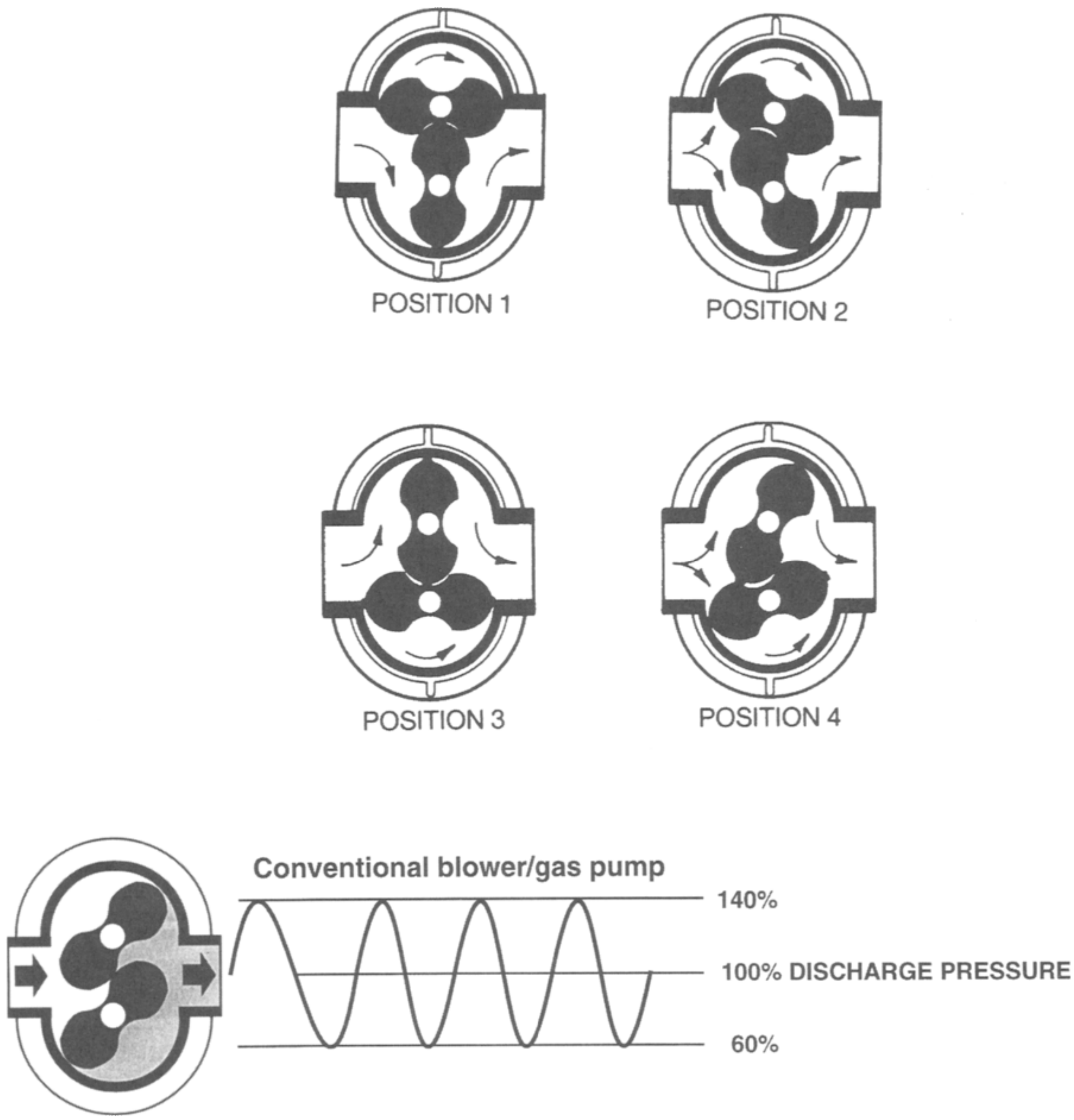
Capacity

Units of this type are available for capacities of a few cfm to approximately 50,000 cfm.

The capacity rating of blowers manufactured by the several companies are similar; however, the basis of an inlet ft^3 per min volume flow can vary depending on the design ratings published in the respective literature. Therefore, it is important to carefully examine the particular reference standard. Most units are rated for air and must be corrected by the factory representative for conditions of other process gases.

a. *Usual pressure practice* is a standard based on Compressed Air and Gas Institute and the American Society of Mechanical Engineers: 14.7 psia, 68°F and 36% relative humidity (RH), or the equivalent air density of 0.075 lb/ft^3
b. Vacuum ratings are based on 68°F and a discharge pressure of 30 in. Hg, sp. gr. for air = 1.0
c. For conditions of the actual inlet flow to blower, covert to actual ft^3/min (acfm):

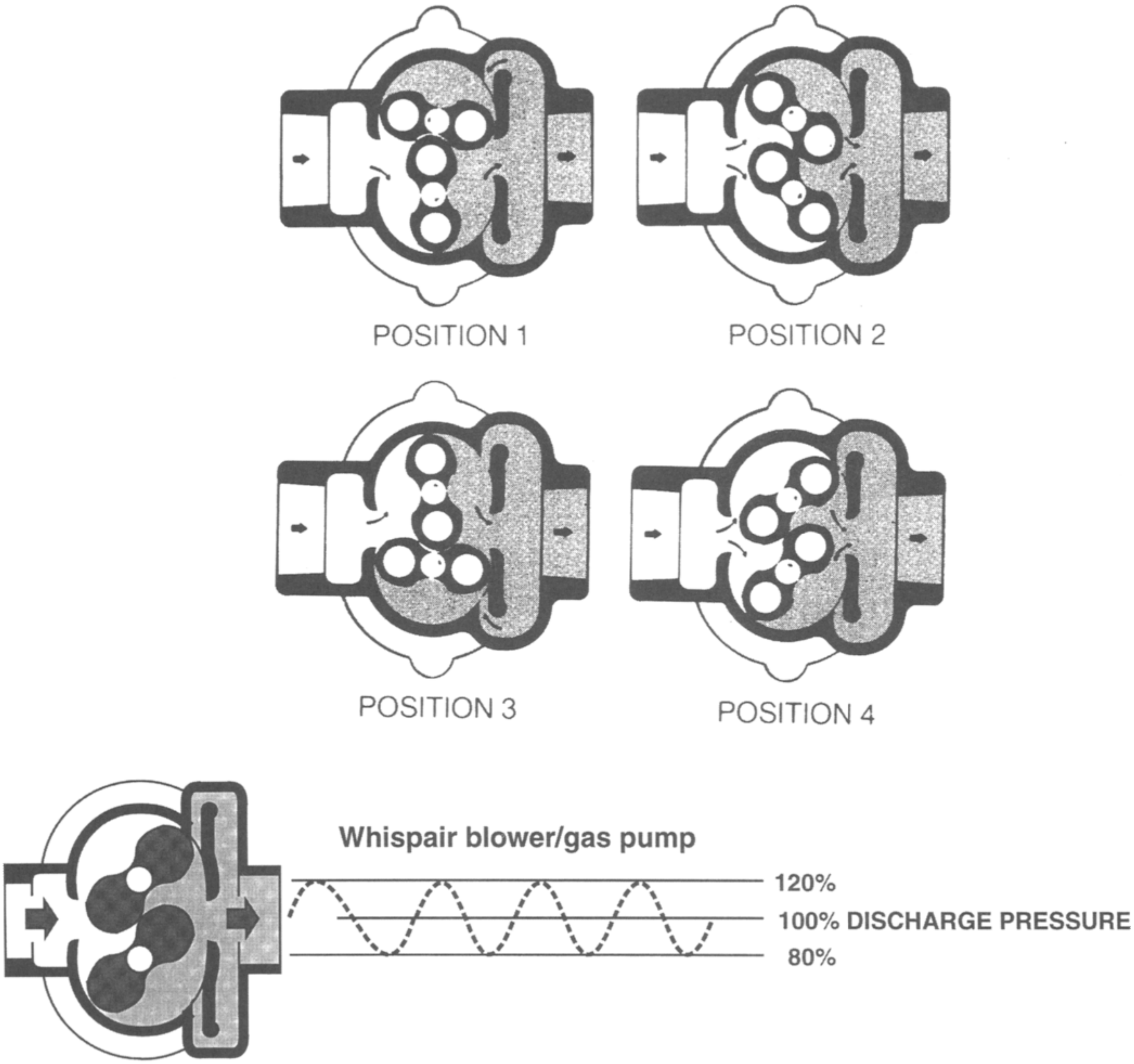
$$\text{acfm} = \text{scfm}\frac{(P_s - (RH_s \times PV_s)(T_a)(P_b)}{(P_b - (RH_a \times PV_a)(T_s)(P_a)} \qquad (12\text{-}156)$$



Two "Figure 8" lobe impellers, mounted on parallel shafts, rotate in opposite directions. As each impeller passes the blower inlet, it traps a definite volume of air and carries it around the case to the blower outlet, where the air is discharged. With constant speed operation, the displaced volume is essentially the same regardless of pressure, temperature or barometric pressure.

Timing gears control the relative position of the impellers to each other and maintain small but definite clearances. This allows operation without lubrication being required inside the air casing.

Figure 12-107. Operating principle for two-lobe blowers. (Used by permission: Bul. 12x95 Rev. 8/97 and Bul. B-5219 Rev. 1/97. ©Roots Div. Dresser Industries, Inc.)



The Whispair® blower operates on the same basic principle as all other rotary positive displacement blowers with one important advantage – units with the Whispair design offer reduced pulsation, operating noise and power loss by utilizing an exclusive wrap-around plenum to control pressure equalization. Whispair blowers have a proprietary jet to feed backflow in the direction of impeller movement, aiding rotation and lowering power requirements.

Incoming air is trapped by the impellers and moved through the machine as in the basic rotary positive displacement principle. As pressure builds against the wrap-around plenum due to system resistance, the Whispair blower jet equalizes the pressure between the trapped air and the discharge area. This action reduces shock and feeds the backflow in the direction of rotation.

As the impeller completes its cycle, it discharges the trapped air, which now has the same pressure as the discharge line. Backflow is controlled, resulting in reduced pulsation compared to the conventional blower. This improves efficiency, reduces noise level, and increases bearing and gear life.

Figure 12-107A. Roots Whispair® Blower principle. (Used by permission: Bul. B-05x93, Rev. 7/97 and Bul. B5219, Rev. 1/97. ©Roots Div., Dresser Industries, Inc.)

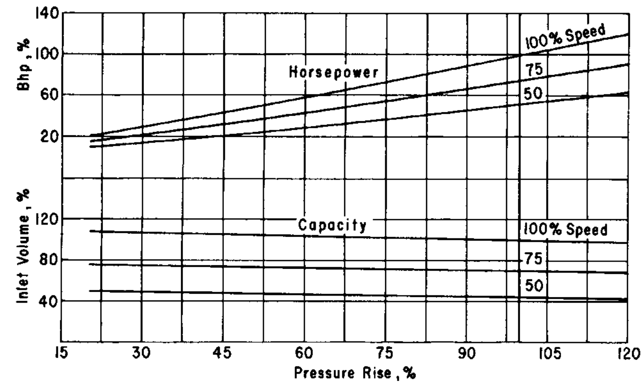


Figure 12-108. Typical performance curve for lobe-type blower. (Used by permission: ©1961. Roots Division Dresser Industries, Inc.)

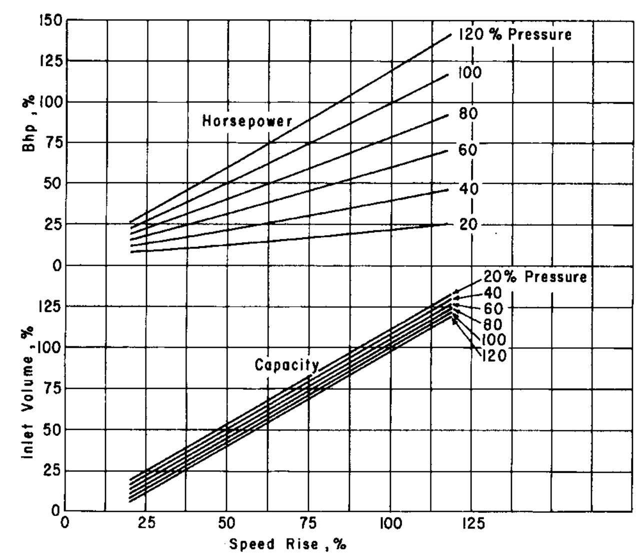


Figure 12-109. Typical performance curve for variable-speed operation for lobe-type blower. (Used by permission: ©1961. Roots Division, Dresser Industries, Inc.)

where P_s = standard pressure, psia, 14.7
P_b = atmospheric pressure, barometer (psia)
P_a = actual pressure (psia), suction
RH_s = standard relative humidity, 36%, fraction
RH_a = actual relative humidity, %, fraction
PV_s = saturated vapor pressure of water at standard temperature (psi)*
PV_a = saturated vapor pressure of water at actual temperature (psi)*
T_s = standard temperature (°R); °R = °F + 460; °F = 68°
T_a = actual temperature (°R)
scfm = standard conditions of 14.7 psia, 68° F and 36% relative humidity
acfm = actual flow conditions, ft^3/min
*From water vapor or steam tables
Note: Relative humidity values apply only for water vapor.

Saturated process vapor (with water or process vapor) is to be omitted for most process gases. However, it is important to determine the correct total volume of vapor that the compressor is to handle by discussing the requirement with a qualified equipment manufacturer.

Pressures in single-stage machines are limited to about 15 psig with a few models good for 20 psig. In a multistage arrangement, the pressures can go to 30 psig. In vacuum service, the inlet pressure can be down to 8 in. Hg abs. In some applications the clearances wear or change, and the lobes must be built up by metalizing or baking on coatings. In water seal units, the clearances may be maintained by allowing the seal water to deposit carbonate scale on the casing and lobes, under controlled conditions.

Efficiency

The volumetric efficiency varies with the speed of operation, being lower at lower speeds and higher for low discharge pressures. As a general guide, the orders of magnitude of efficiencies are as follows:

Speed, rpm	Volumetric Eff.	Pressure Range psig
360	80–95	14 to 1.0
588	70–82	14 to 1.0
720	90–97	14 to 1.0

The maximum speed for these machines ranges from 500–4,000 rpm, depending upon bearing design and unit size. For comparison selections refer to Huff[34] and Patton.[43]

Rotary Axial Screw Blower and Vacuum Pumps

These units are rotary positive displacement compressors that operate somewhat like the lobe-type blowers; however, in the case of the axial screw units, the gas passing through receives compression from the internal meshing of the screw-like rotors, Figure 12-110A. The resulting compression is a modified adiabatic process.[10] Several different units use screw designs as shown in Figures 12-110B and 12-110E. The screw surfaces do not touch each other or the casing wall as they rotate.

The gas being handled flows axially through the unit, with most of the compression taking place just before discharge. The amount of this compression is determined by the arrangement of the discharge opening. By changing discharge valve positions, the internal compression can be changed to give improved performance or different discharge pressure.

The spiral-lobe and helical-screw compressors are rotary positive-displacement machines and quite adaptable to a wide assortment of process and refrigeration gases. This class of equipment is usually built to comply with the American Petroleum Institute Standard #619. These units operate at higher "tip" speeds than the straight "lobe" type units and are capable of higher compression ratios.

The general construction features and materials are quite similar to the lobe type units.[1] For comparison selections, refer to Huff,[34] Patton,[43] Price,[124], Abraham, [125] and Van Ormer.[126]

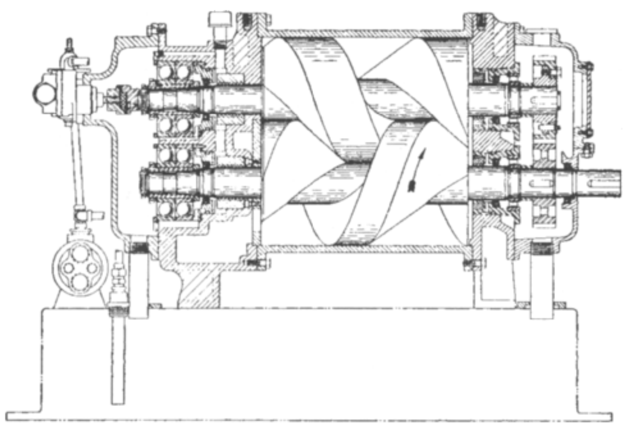


Figure 12-110A. A typical cross-section showing the spiral screw rotors, lubrication system, and other details of internal construction. (Used by permission: ©1961. Roots Division, Dresser Industries, Inc.)

Performance

The general performance of these units is shown by the curves of Figures 12-111, 12-112, and 12-113.

The range of suction volumes is approximately 175–35,300 scfm per minute, and the units are capable of discharging up to 580 psi. In vacuum application, the units can draw down to 1.3 psi abs.[123]

Two types of these units are

1. Dry machines use shaft-mounted gears for proper meshing of the rotors and can compress gases free of entrained water vapor and other contaminants.[123]
2. Liquid injected units do not usually require gearing for proper meshing of the counter-rotating screws. The injected liquid, Figures 12-110A–F, can be clean demineralized water, oil, or other fluid that separates the two screws. Advantages exist to the liquid injection: (a) internal cooling that reduces potential explosion hazards or polymerization and (b) higher compression ratios (often one stage with liquid can do job of two stages dry).[123]

The rotary screw compressor is designed for a specific compression ratio, (i.e., discharge pressure divided by intake pressure). If the pressure rises on the system into

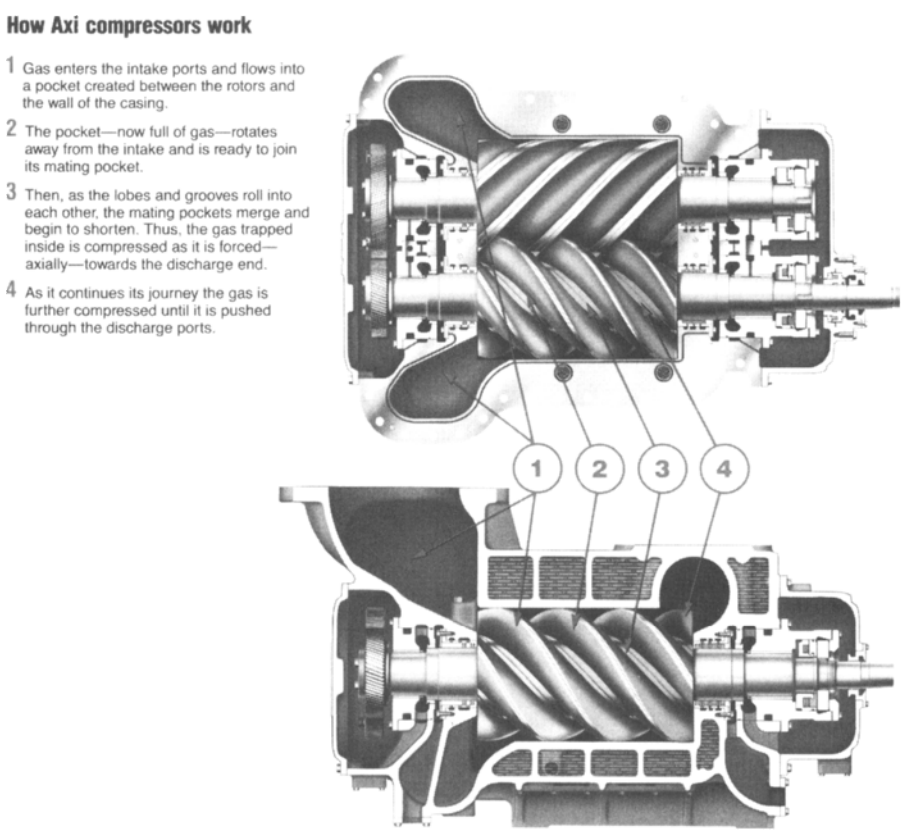


Figure 12-110B. Type H Axi® Helical Rotor positive displacement compressor. Applications include gases with entrained liquid, oil-free, low-mol wt gases and a wide range of operating conditions. (Used by permission: Form 11232-A, ©1987. Dresser-Rand Company.)

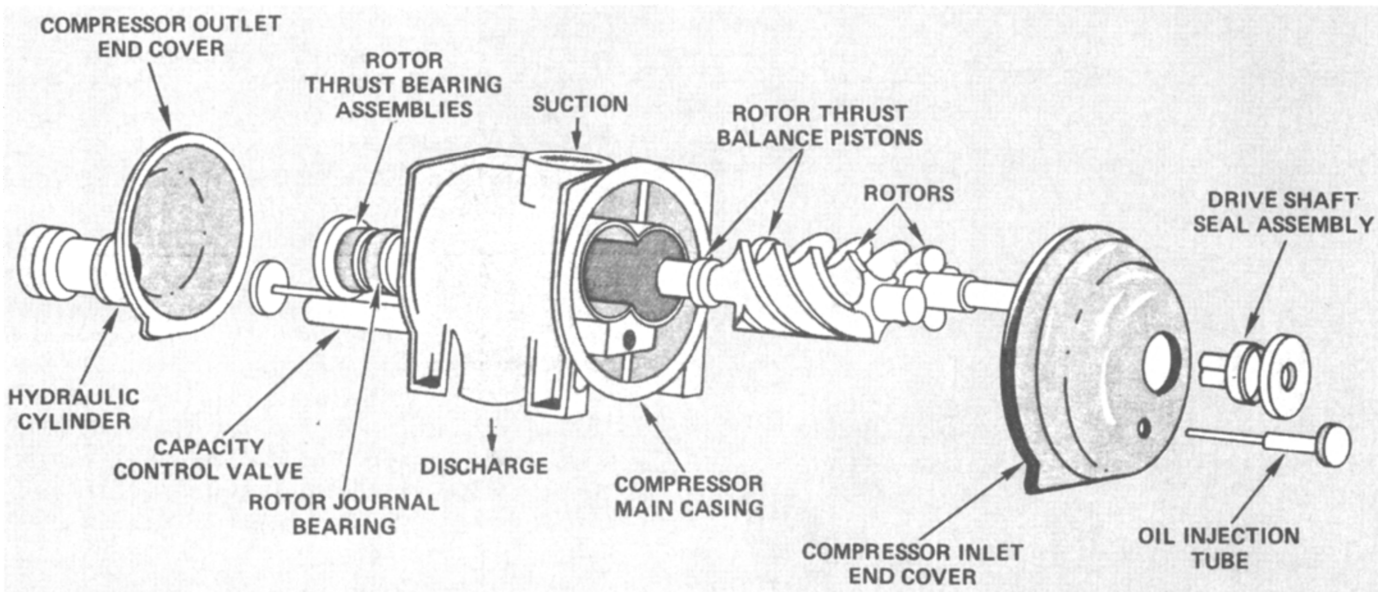


Figure 12-110C. Exploded view of a screw compressor. (Used by permission: Price, B. C. *Chemical Engineering Progress,* V. 87, No. 2, p. 51, ©1991. American Institute of Chemical Engineers. All rights reserved.)

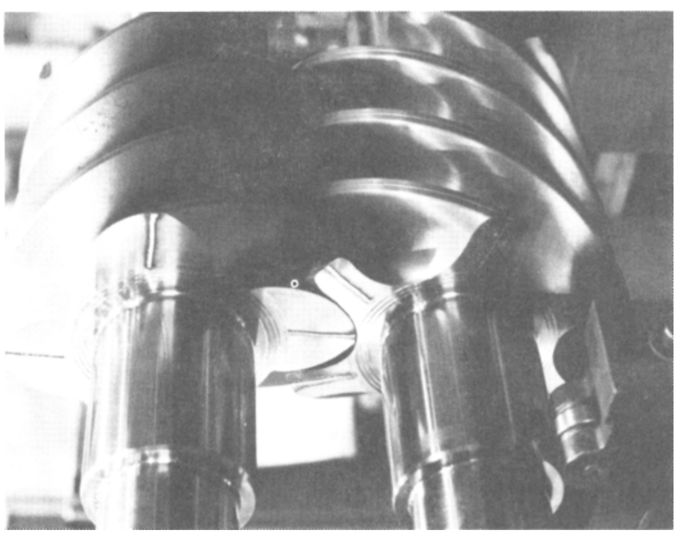


Figure 12-110D. Rotor set for oil-free rotary screw compressor. (Used by permission: Bul. CCB-0057-3-295. A C Compressor Corporation.)

which the rotary screw is discharging, the unit will perform up to the physical limits of strength of the casing and the available input power to the shaft. As the discharge pressure falls, the compression ratio falls for a fixed inlet condition, and the efficiency of the unit will fall off.

The total power input to the shaft is composed of the following:[123]

a. Energy of adiabatic compression of the gas.
b. Dynamic-flow power loss (typically 10–15% of the actual power). This is a function of the built-in compression ratio and the Mach number at the compressor inlet conditions.
c. Mechanical losses (typically 8–12% of actual power). These losses include viscous or frictional losses due to bearing, timing, and step-up gears.

Referencing to the comprehensive article[123] on this class of equipment by Bloch and Noack, an abbreviated listing of the advantages and disadvantages of this equipment is as follows:

Advantages

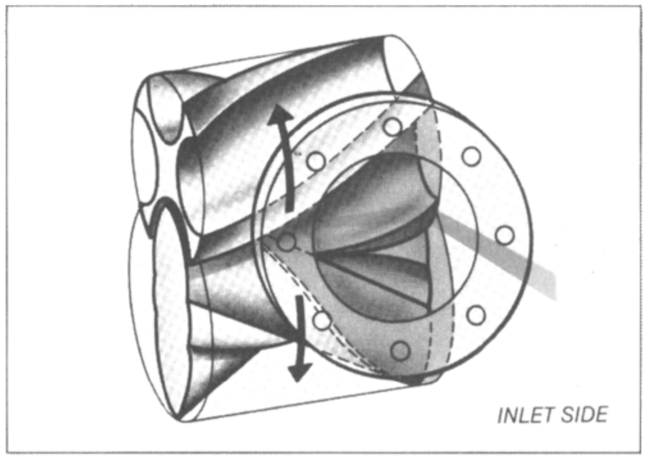
1. Greatly reduced sensitivity to molecular weight change.
2. Greater tolerance for polymerizing conditions.
3. Ability to accept liquid and fine solids entrainment.
4. Higher efficiency and less maintenance than the liquid rings units.
5. Estimated availability > 99.5%.
6. Smaller size and lower capital cost than same capacity range of reciprocating compressors.
7. Higher pressure capabilities (compared to other types of rotary positive-displacement units).
8. Oil-flooded units operate over wider range of compression ratios (compared to dry units), due to better temperature control.

Disadvantages

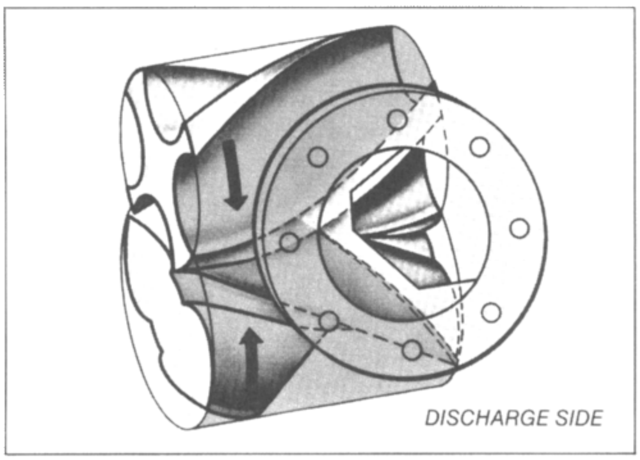
1. Sensitivity of close tolerances to discharge temperature, affecting operability. This problem can be solved by proper cooling and temperature control.

The Type L Axi compressor consists essentially of two mating helical rotors inside a casing. As the rotors turn away from each other on the inlet side, air or gas is drawn into pockets formed between them and the casing wall.

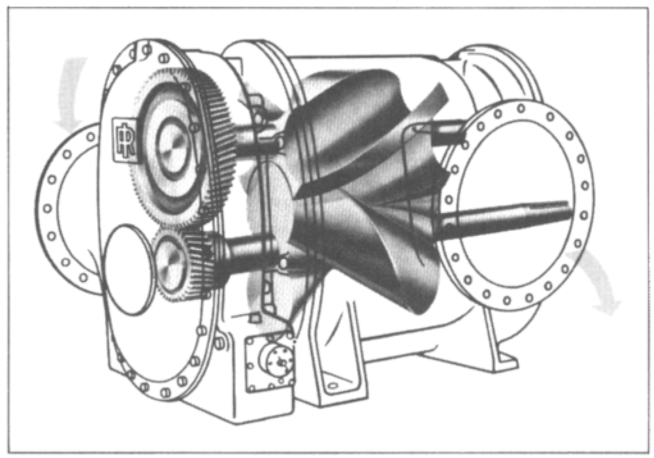
The pockets move helically along and around the axes, then join and diminish in size; thus the Axi provides **internal** compression, unlike the straight-lobe blower types. The result is significantly higher efficiency and less power cost.



At the inlet, pockets form between the rotors and the casing wall, and draw in the air.



As the rotors turn, mating pockets join and reduce in volume, compressing the air.



The pockets move along and around the axes while diminishing in size.

Figure 12-110E. Type L Axi® Helical rotor assembly and rotation details. (Used by permission: Form 11193-F, ©1987. Dresser-Rand Company.)

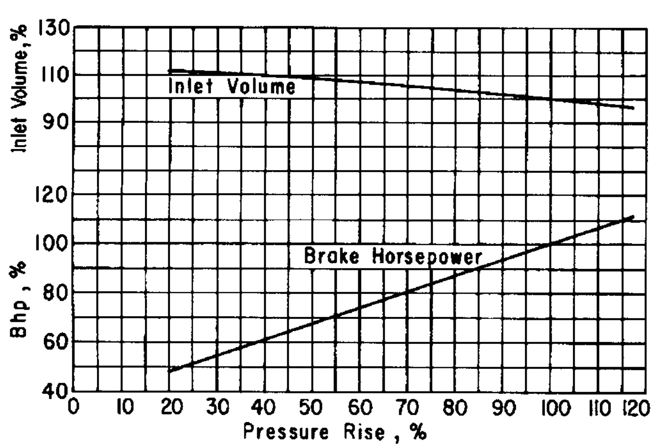


Figure 12-111. Typical constant-speed performance of spiral screw rotor compressor. (Used by permission: ©1961. Roots Division, Dresser-Rand Industries, Inc.)

2. Affected by corrosion and erosion of the rotor and casing, increasing the gas-slip internal recycle. This problem is not serious for water- or oil-injected screws.
3. Requires pulsation suppression.
4. Selection of construction materials for rotors and casings is more limited than for centrifugal compressors.
5. Maintenance cost and length of down time are higher than for centrifugal units.
6. Flexibility in flow control is not as good as centrifugal or axial compressors.
7. High noise level, but this can be reduced.

Efficiency

The overall compression and mechanical loss efficiency of these units averages between 70–75%. Peak values will reach 78%, and on the extremes of the performance curves, the values reach 60–65%. The efficiency increases with the larger units and at higher speed operations.

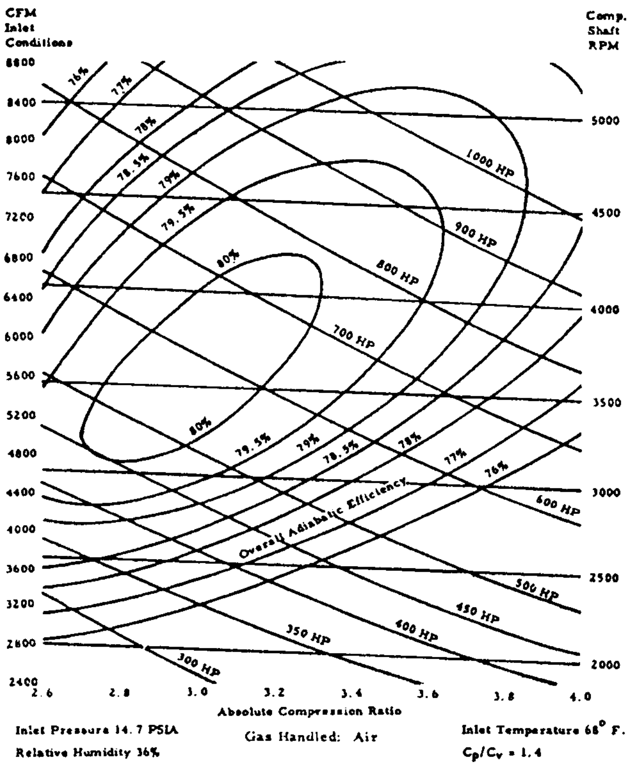


Figure 12-112. Performance characteristics for a typical single-stage rotary helical rotor compressor, using matching helical rotors. (Used by permission: Fairbanks, Morse, & Co. for earlier editions. [Company no longer exists producing compressors, 1998, per research information.])

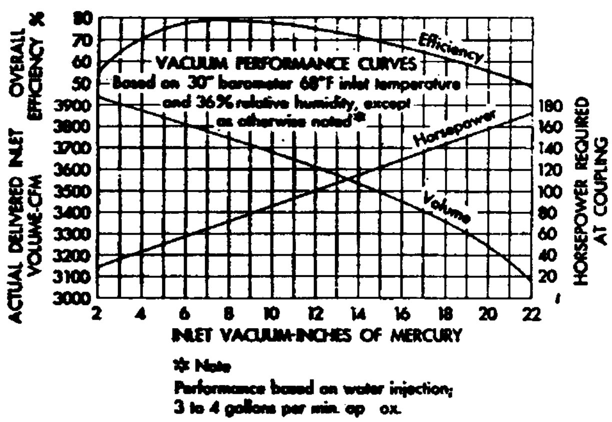


Figure 12-113. Typical test results of a medium-capacity spiral lobe compressor vacuum pump using intermeshing lobes. (Used by permission: Ingersoll-Rand Co.)

Speed

Usually these units are run at 1,750–3,600 rpm; however, they may be belt, gear, or steam turbine driven at any reasonable speed consistent with the rating of the compressor gears.

Capacity

The capacity ranges up to 12,000 (and sometimes greater) cfm inlet volume with a discharge ranging from 3–20 psig in single case units. Special units reach 60–100 psig. Multiple cases can carry the pressures to higher values. The units also handle vacuum service of 500–10,000 cfm from 5 in. Hg to 25 in. Hg vacuum (25–5 in. Hg abs). Water may be sprayed into the unit to help maintain the higher vacuums by keeping the temperature below 125°F.

Slip

The slip for this type of compressor is similar to that of the lobe units; however, the passages are basically different, and this changes the approach to slip correction. The manufacturer should be consulted for data specific to a particular unit. The slip is dependent on the pressure differential across the unit and the gas density. It does not vary with speed or length of the rotor.

Total Capacity

$$V_T = V_s' + V_1 \tag{12-157}$$

where V_T = total internal capacity pumped, cfm
V_s' = slip cfm
V_1 = intake volume, cfm

Temperature Rise

Estimate as for usual adiabatic calculation.

Rotary Sliding Vane Compressor

The sliding vane compressors and vacuum pumps have internal sliding vanes mounted longitudinally on an eccentric rotor in the body casing, Figures 12-114 and 12-115.

The body cylinder is usually cast iron with integral internal water jackets for cylinder wall cooling. These water jackets are tested for tightness.

The rotor (forged steel) may or may not be an integral part of the shaft. In any case the material for both rotor and shaft is usually a high-strength alloy cast iron. The rotor has radial slots machined along its entire length. The blades or vanes are of heat-treated phenolic resin, metal, or suitable material to withstand the gas and the pressure fit in these slots.

Because these machines require internal lubrication of the sliding-vane surface, no extreme effort is made to arrange special bearing and shaft seals, except that a gas seal of the conventional mechanical design is usually used to prevent gas from escaping to the outside or to prevent air inleakage in the case of vacuum service. For hazardous or

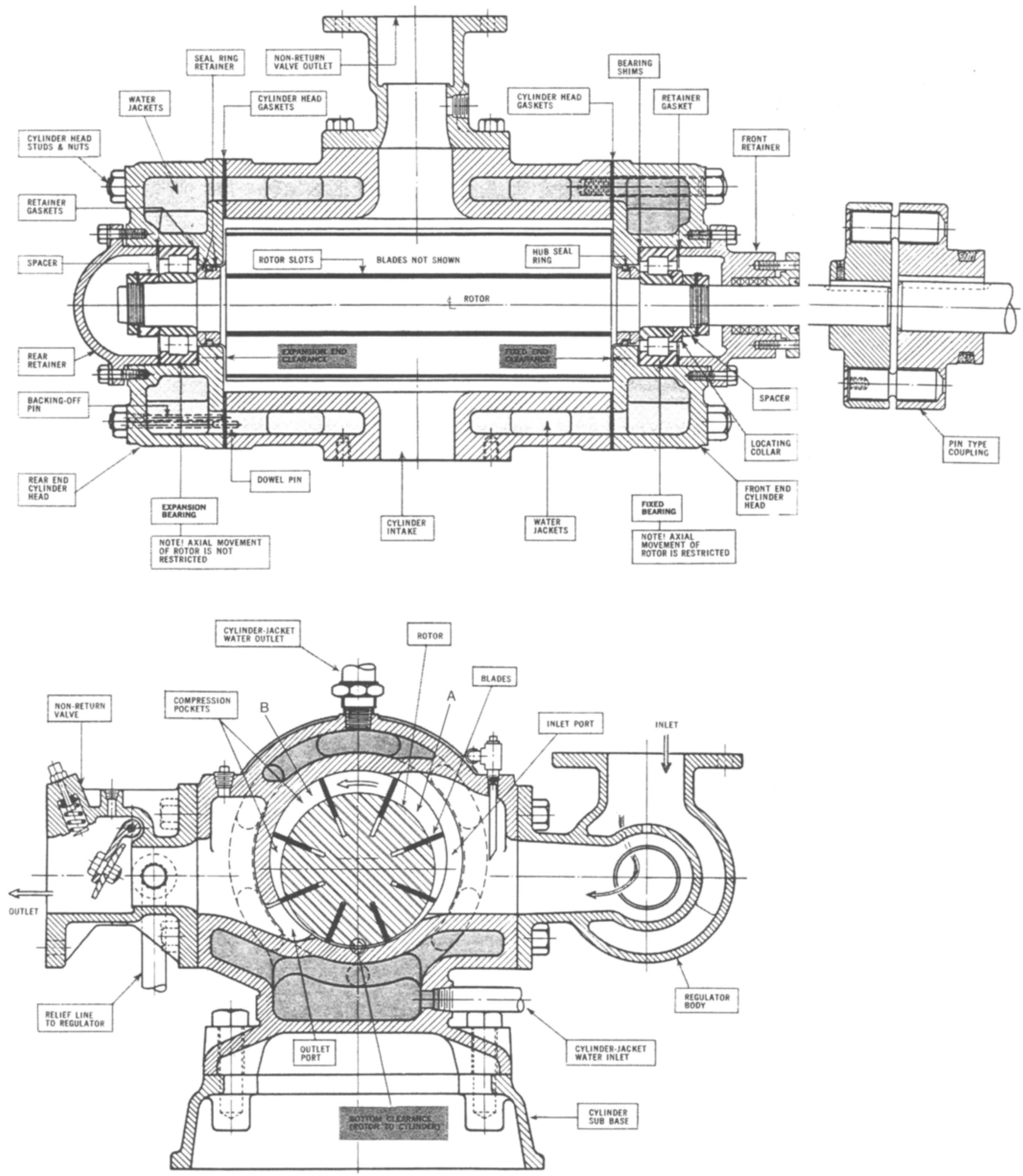


Figure 12-114. Cross-section of sliding-vane rotary compressor. (Used by permission: Fuller Bulk Handling.).

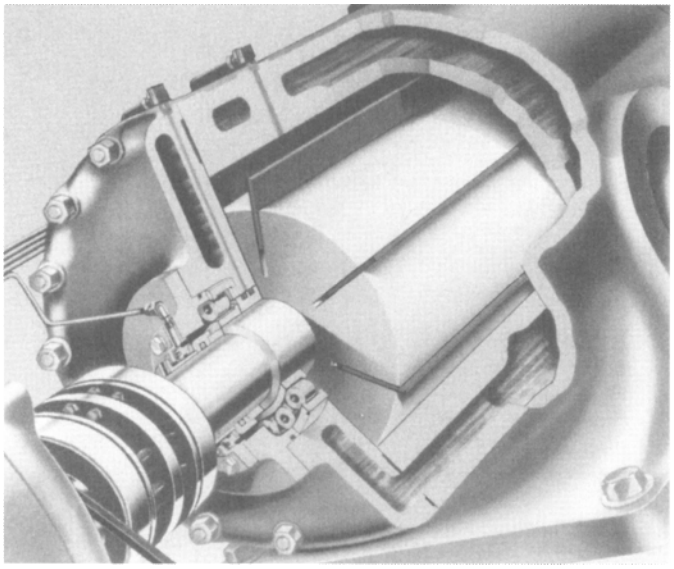


Figure 12-115. Coupling drive end of Ro-Flo Sliding-Vane compressor showing vanes in rotor or shaft slots and bearing and shaft seal. (Used by permission: Bul. CCB-0072-2. ©A C Compressor Corporation.)

corrosive gases, extra care is used in the shaft seals. Lubrication is of the forced-fluid type for both the inside of the cylinder and the bearings. The oil or other lubricant is usually injected into the entering nozzle on the machine to ensure a running surface between the vanes and the cylinder wall. This also effectually seals against gas slippage between the compartments.

Performance

These machines are positive pressure in operation because no internal means exists for the gas to bypass from discharge to suction.

The units may be belt, gear, or direct driven. Single operating units generally can develop differential pressures to 60 psig maximum while being limited to a discharge temperature of 350°F. When applied as a booster compressor, the units can develop discharge pressures of 250 psig. These booster units are designed for the higher rated service. When two units are operated in series as a two-stage assembly, usually required intercooling of the gas occurs from the discharge of the first to the inlet of the second unit. Generally, the two-stage units can operate with larger throughputs and discharge pressure from 100–125 psig.

When used as a vacuum pump, these single-stage units can pull vacuum down to 27 in. of mercury vacuum (not absolute) and handle approximately 1,800 ft^3/min. As two-stage vacuum pumps they can pull vacuum to 29.97 in. mercury (see Chapter 6, V. 1 of this series) with about 2,000 cfm at high vacuum.

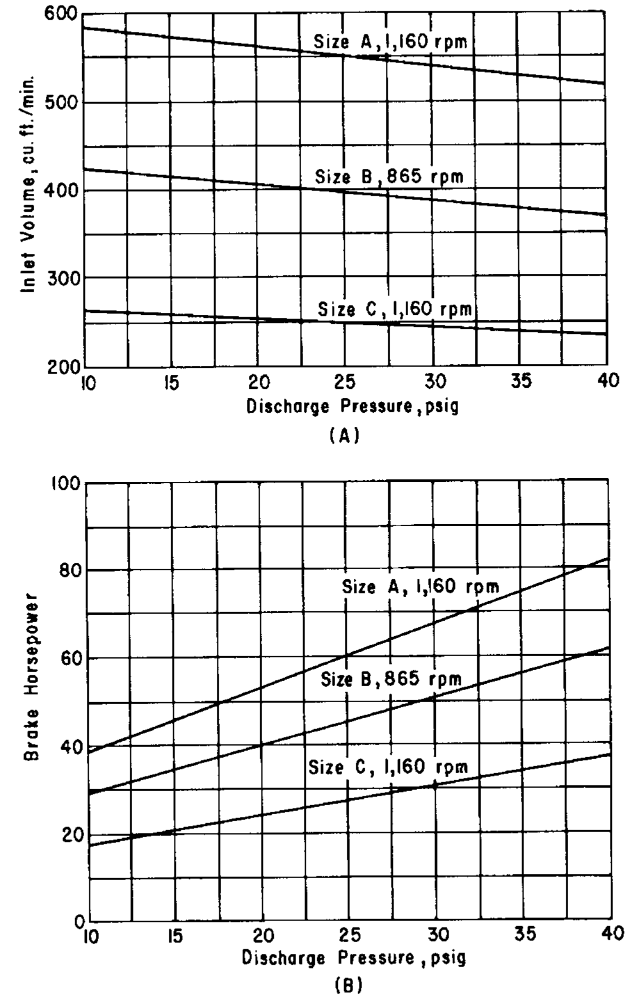


Figure 12-115A. Performance curves for rotary-vane compressor. (Used by permission: A C Compressor Corporation.)

Speed ranges for single-stage and two-stage units are from 500–1,180 rpm. Consult manufacturers for design-capacity ratings.

As the cycle starts, the vane passes over the intake port, and the space (A) in Figures 12-114 and 12-115 fills with gas at suction conditions. The completion of the filling takes place near the point of maximum volume between the vanes. Then as the enclosed chamber (B) rotates toward the discharge, its volume becomes smaller as the vanes are forced to recede due to the eccentric position of the shaft. At the point of minimum volume and maximum compression, the gas discharges from the machine. These machines will not compress until the speed is sufficient to throw the vanes against the cylinder walls; thus, the machine always starts unloaded. These compressors have no inlet or outlet valves because they discharge against the system pressure. The operation is free from pulsing flow. Typical performance curves are shown in Figure 12-115A. For comparison selections, refer to Huff[34] and Patton.[43]

Speed

The units operate at an electric motor and/or internal combustion engine speeds of 450–3,600 rpm but can be adapted to V-belt or gear for any driver speed.

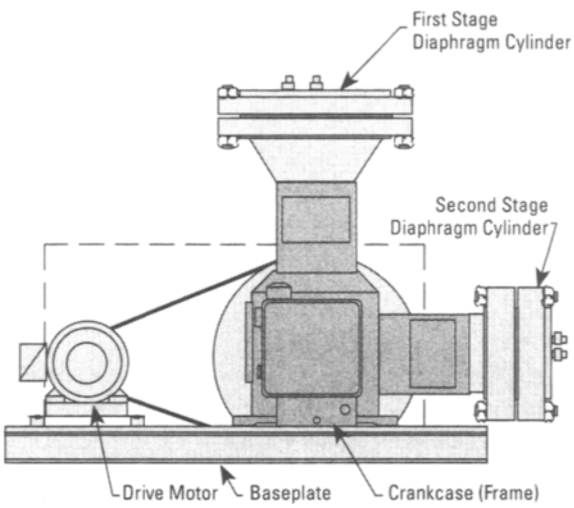


Figure 12-116. Diaphragm gas compressor. Gas remains oil-free, capable of handling all clean gases. (Used by permission: Livingston, E. H. *Chemical Engineering Progress,* V. 89, No. 2, ©1993. American Institute of Chemical Engineers, Inc. All rights reserved.)

Applications

Typical process-related applications have included ammonia in refrigeration, pneumatic conveying, gas gathering and boosting, vapor recovery, flow gas recovery, and others in petroleum refining; air supply and methane recovery in the mining industry; aeration and methane boosting in sewage/biogas operation.

Other Process-Related Compressors

Several other types of compressors are used in the process industries for a wide variety of special applications that may not fit the larger reciprocating or centrifugal compressor selections. One of the smaller compressors is a diaphragm compressor, Figures 12-116 and 12-117 and Table 12-10.

These units are usually oil-free, noncontaminating, and leak-proof and are usually belt- or gear-driven. The ranges of capacities are 0.5–72 cfm, and the gas is compressed by the action of a metal diaphragm that completely isolates the process gas from the displacing element driving the diaphragm. The motion of the displacing element is transmitted to a hydraulic fluid, and this fluid transmits its motion to one or more thin, flexible metal diaphragms or discs. As the diaphragm moves in the compression chamber,

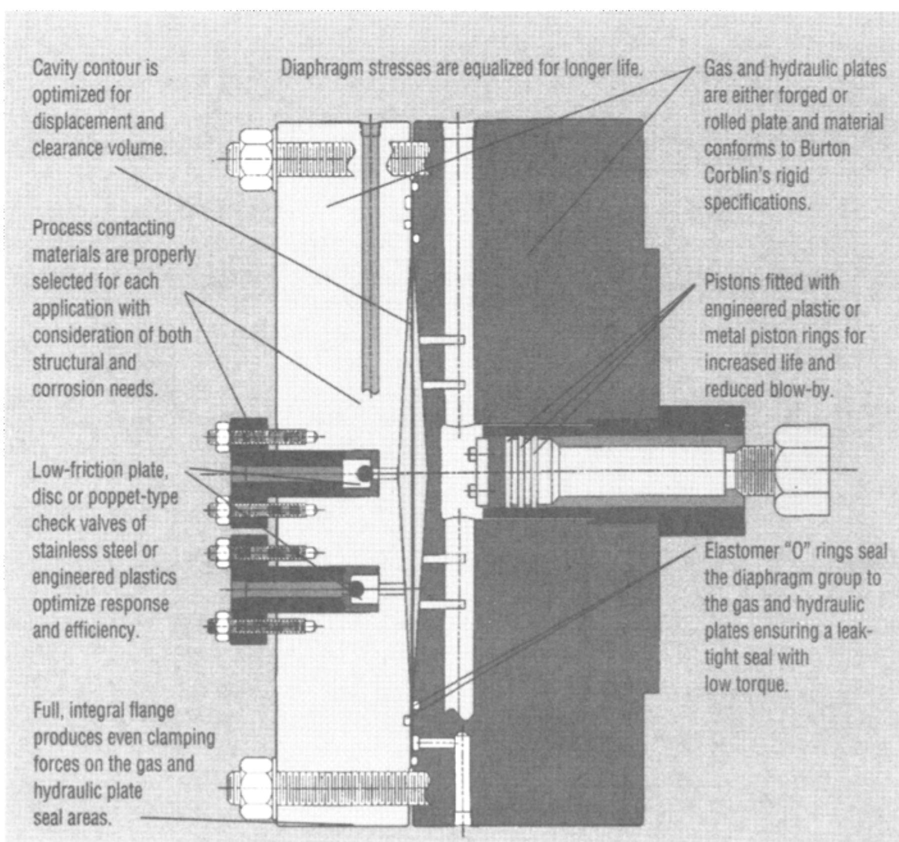


Figure 12-117. Motion of the displacing element causes the diaphragm to move into the compression chamber to reduce the volume and, thereby, increase gas pressure. (Used by permission: Bul. BCHB-2D101. ©Howden Compressor, Inc.)

the gas pressure is increased by reducing the volume of the fluid (see Figure 12-117).

Clearance volumes are approximately 4–7% depending on the size of the diaphragm cylinder. With pressure applications to 300 mpa (43,500 psi), the clearance volumes may be as high as 10–12% due to manufacturing tolerance limits in the valve pocket area.[58] Leakage from standard construction "O"-rings is approximately 1×10^{-7} std cc/sec. For extremely low leakage, metallic "O" or seal welding can yield 1×10^{-8} std cc/sec.[58] Detection devices exist in case of a significant leakage or failure of the diaphragm to prevent oil loss to the process.

Note: 1 Pa = 0.0001450 psi
1 mpa = 145.0 psi
63 mpa = 63(145) = 9135 psi

Table 12-10
Typical Diaphragm Compressor Process Construction Materials

Component	Material	Remarks
Gas plate	Carbon steel	Pressure to 63 mpa
	Low alloy steel	Pressure to 200 mpa
	304 SS, 316 SS	Pressure to 63 mpa
	17-4 PH, A286	Pressure to 200 mpa
	High nickel alloys	Pressure to 63 mpa
	20 Cb-3	Pressure to 63 mpa
Diaphragms	301 SS, 316 SS	Standard for most service
	Ni-cu alloy	Oxidizer service

Used by permission: Livingston, E. H. *Chemical Engineering Progress*, V. 89, No. 2, ©1993. American Institute of Chemical Engineers. All rights reserved.

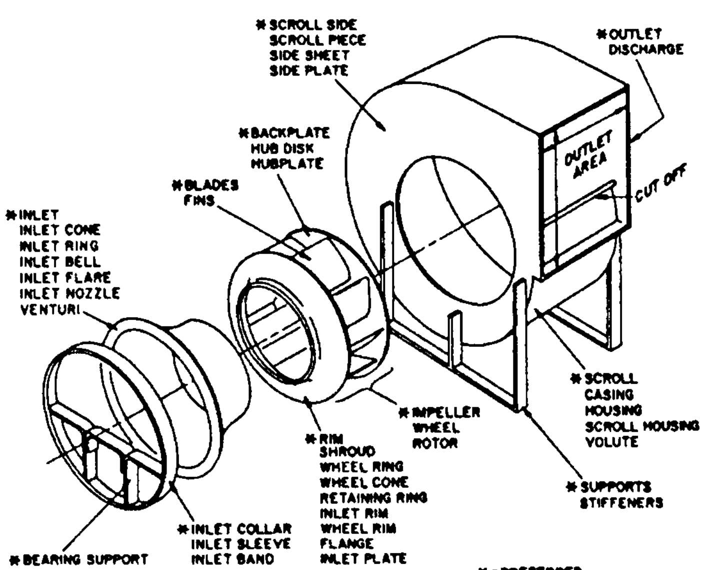


Figure 12-118A. Common names associated with centrifugal fan components. (Used by permission: *ASHRAE Handbook and Product Directory*, ©1979, pp. 3.1. American Society of Heating Refrigerating and Air-Conditioning Engineers, Inc. All rights reserved.) Also see 1996 Edition, *HVAC Systems and Equipment*.

Fans

Fans (or blowers) of large volumes for industrial applications are usually applied to air service, and essentially all manufacturer's performance data/charts/tables are so referenced to standard air; however, they can be readily adapted to chemical/petrochemical process applications in which relatively large volumes of clean gas mixtures are processed at low pressures.[127]

Fans are rather generally identified as machines with relatively low pressure rises that move gases or vapors by means of rotating blades or impellers and that change the rotating mechanical energy into pressure or work on the gas or vapor, Figures 12-118A–D. The result of this work on the fluid will be in the form of pressure energy or velocity energy or some combination of both.[5, 11, 38, 49, 129] The fan wheel is a constant volume device.

Regardless of type, overall fan action must depend on a rate of change of gas momentum in a tangential direction. Without this change in momentum, no resisting torque can exist, and no fan power input is required or absorbed.[11]

As the air or gas flows through the blower system (piping/ducts, filters, etc.), the movement causes friction between the flowing air/gas. This friction translates into resistance to flow, whether on the inlet (suction side) or outlet (discharge side) of the system in which the blower is a part and that creates the pressure drop (see Chapter 2, V. 1, 3rd Ed., of this series) which the blower must overcome in order for the air/gas to move or flow. This resistance to flow becomes greater as the velocity of flow increases, and more energy or power is required to perform the required flow movement at the required pressures.

The usual maximum pressure rise from inlet to discharge through a fan is around 1 psi. However, some heavy-duty fans are available for 2 and even 3 psi rise. Fans are normally used in applications taking suction at atmospheric pressure or at only a few inches negative or positive pressure because of the general constructional features of fans, which are basically sheet metal and not suitable for high inlet pressures.

Types of Fans

Ventilating and industrial fans are classified by the Air Movement and Control Association, Inc. (AMCA) as shown in Table 12-11 and Tables 12-12A and 12-12B with more detail.

To standardize the many fan arrangements, the AMCA has established the most probable standard configurations in Figures 12-119A–C. If these designations are used when inquiring and specifying fans, the manufacturer can immediately interpret the situation.

Figure 12-119E is one of several standards that the AMCA developed to establish strength classes for specific centrifugal fan designs. Note that Class I from the figure requires that a fan of this class must physically be capable of performing in the range of inlet (2 $^1/_2$ in. WG @ 3,200 cfm) through discharge of (5 in. WG @ 2,300 cfm). Similar requirements apply to Classes II and III as noted. Discuss the details with the manufacturers.

Fans are classified according to the discharge pressure.[49] Reprinted per written permission from the Air Movement and Control Association International, Inc., the AMCA Standard 99-1401-66 from *Standards Handbook 99–86* ©1986, the total static pressure classification for operating limits for central station units is as follows:

Class	*Total Static Pressure, Max. In. WG
A	0–3
B	3–5.5
C	more than 5.5

*Total static pressure includes the internal static pressure losses.

Class	Max. Wheel** Tip Speed ft/min	Maximum Total Pressure in. Water
I	7,—10,000	3 3/4
II	7,—13,000	6 3/4
III	12,—16,000	12 1/4
IV	12,—18,500	> 12 1/4

* Referenced to air of 0.075 lb/ft^3 at 70°F, and 29.92 ft. Hg Barometer.
**Depends upon wheel design, type bearings and maximum operating temperature

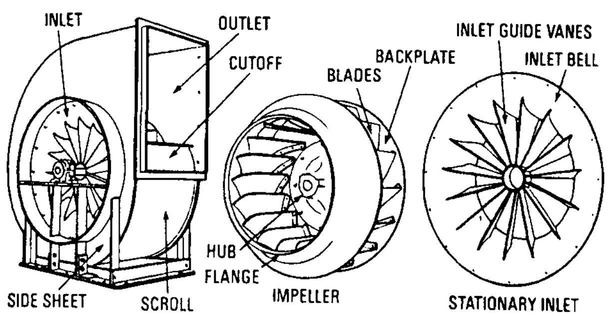


Figure 12-118B. Exploded view of a centrifugal fan. (Used by permission: Jorgenson, R., Ed. *Fan Engineering,* ©1983. Buffalo Forge Co., Div. Howden Fan Co. All rights reserved.)

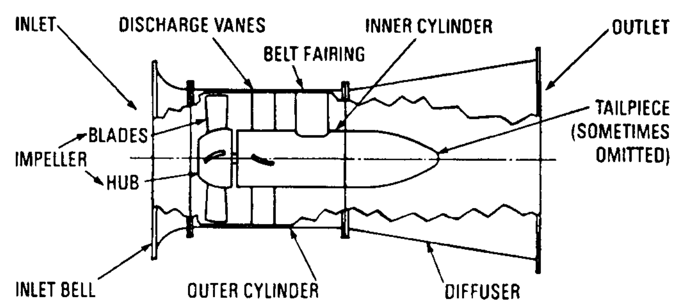


Figure 12-118D. Cutaway view of a vaneaxial fan. (Used by permission: Jorgenson, R., Ed. *Fan Engineering,* ©1983. Buffalo Forge Co., Div. Howden Fan Co., Inc. All rights reserved.)

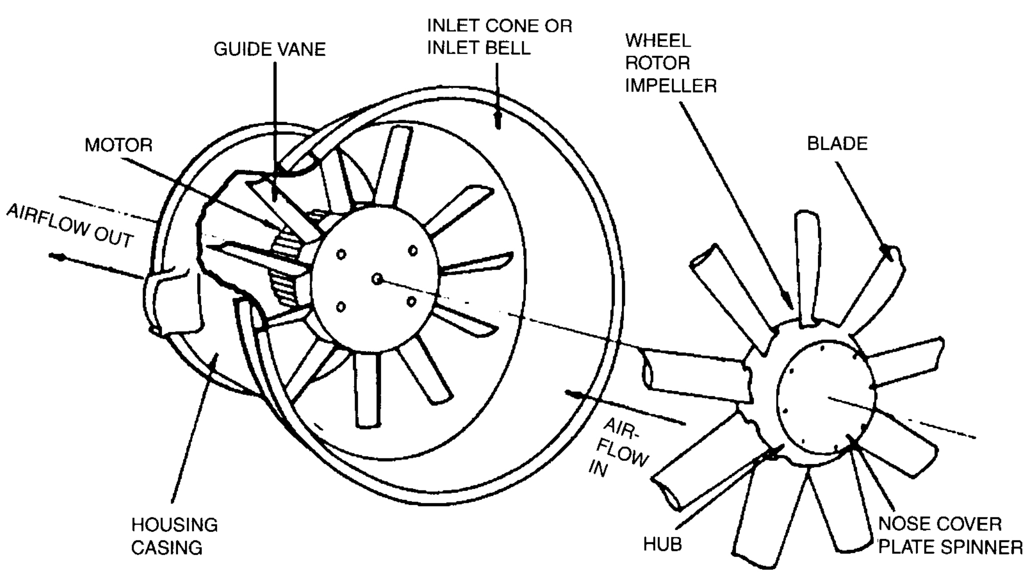
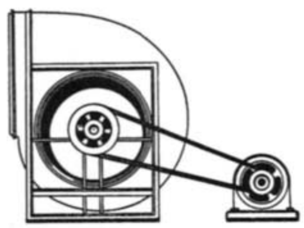
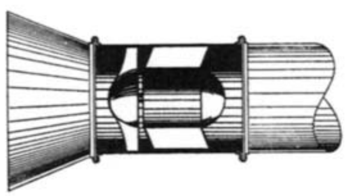
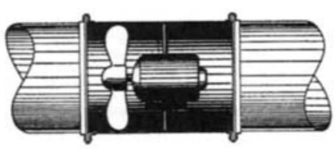
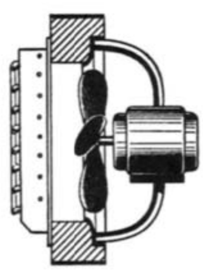


Figure 12-118C. Common names associated with axial fan components. (Used by written permission: *ASHRAE Handbook and Product Directory,* ©1979, pp. 3.4. American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc. All rights reserved.) Also see 1996 Edition, *HVAC Systems and Equipment.*

Table 12-11
Classification of Ventilating and Industrial Fans

Types of Air Moving Equipment		Group Classification	Description
Ventilating and Industrial Fans — Centrifugal, Axial and Propeller Types		**Centrifugal Fan** Either Belt Drive or Direct Connection	A centrifugal fan consists of a fan rotor or wheel within a scroll type of housing. The centrifugal fan is designed to move air or gases over a wide range of volumes and pressures. The fan wheel may be furnished with straight, forward curve, backward curve, or radial tip blades. The fan housing may be constructed of sheet metal or cast metals with or without protective coatings such as rubber, lead, enamel, etc.
		**Vaneaxial Fan** Either Belt Drive or Direct Connection	A vaneaxial fan consists of an axial flow wheel within a cylinder combined with a set of air guide vanes located either before or after the wheel. The vaneaxial fan is designed to move air or gases over a wide range of volumes and pressures. It is generally constructed of sheet metal, although cast metal fan wheels are sometimes furnished.
		**Tubeaxial Fan** Either Belt Drive or Direct Connection	A tubeaxial fan consists of an axial flow wheel within a cylinder. The tubeaxial fan is designed to move air or gas through a wide range of volumes at medium pressures. Its construction is similar to the vaneaxial fan.
		**Propeller Fan** Either Belt Drive or Direct Connection	A propeller fan consists of a propeller or disc wheel within a mounting ring or plate. The propeller fan is designed to move air from one enclosed space to another or from indoors to outdoors or vice versa in a wide range of volumes at low pressure. (The automatic type of shutter illustrated in the cutaway opposite is not a part of the propeller fan but is an auxiliary device to protect the fan when not operating by keeping out wind, rain, snow, and cold).

Reprinted from AMCA Bul B. 110, ©1952. National Association of Fan Manufacturers, with written permission from Air Movement and Control Assoc., Inc.

Of these several types, the centrifugal fan is probably the most used for industrial process applications. The other types are used primarily in ventilating or exhausting service. Centrifugal fans are made in three general types:

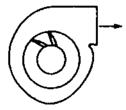
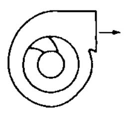
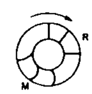
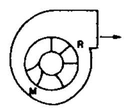
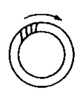
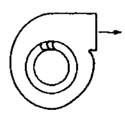
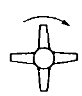
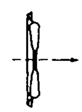
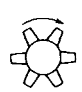
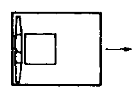
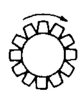
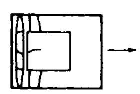
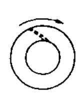
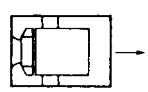
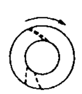
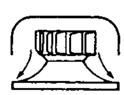
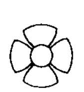
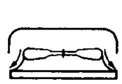
a. Radial blades (or as a modified radial tip blade)
b. Forward curved blades
c. Backward curved blades

Figures 12-118A–D, 12-119D, 12-120A–D, and 12-121 show blade construction and wheel assembly for these types.

The assembly of the centrifugal fan is shown in Figure 12-122. For special applications of dirty gases, see details in Figures 12-123A, 12-123B-1, and 12-123B-2.

The axial fan is being built in larger capacities and for higher pressure characteristics and is finding increasing application. Figure 12-124 shows a vaneaxial unit. A tubeax-

Table 12-12A
Types of Fans

	TYPE	IMPELLER DESIGN		HOUSING DESIGN	
CENTRIFUGAL FANS	AIRFOIL		HIGHEST EFFICIENCY OF ALL CENTRIFUGAL FAN DESIGNS. 10 TO 16 BLADES OF AIRFOIL CONTOUR CURVED AWAY FROM THE DIRECTION OF ROTATION. AIR LEAVES THE IMPELLER AT A VELOCITY LESS THAN ITS TIP SPEED AND RELATIVELY DEEP BLADES PROVIDE FOR EFFICIENT EXPANSION WITHIN THE BLADE PASSAGES. FOR GIVEN DUTY THIS WILL BE THE HIGHEST SPEED OF THE CENTRIFUGAL FAN DESIGNS.		SCROLL TYPE, USUALLY DESIGNED TO PERMIT EFFICIENT CONVERSION OF VELOCITY PRESSURE TO STATIC PRESSURE THUS PERMITTING A HIGH STATIC EFFICIENCY, ESSENTIAL THAT CLEARANCE AND ALIGNMENT BETWEEN WHEEL AND INLET BELL BE VERY CLOSE IN ORDER TO REACH THE MAXIMUM EFFICIENCY CAPABILITY. CONCENTRIC HOUSINGS CAN ALSO BE USED AS IN POWER ROOF VENTILATORS, SINCE THERE IS EFFICIENT PRESSURE CONVERSION IN THE WHEEL.
	BACKWARD INCLINED BACKWARD CURVED		EFFICIENCY IS ONLY SLIGHTLY LESS THAN THAT OF AIRFOIL FANS. BACKWARDLY INCLINED OR BACKWARDLY CURVED BLADES ARE SINGLE THICKNESS. 10 TO 16 BLADES CURVED OR INCLINED AWAY FROM THE DIRECTION OF ROTATION. EFFICIENT FOR THE SAME REASONS GIVEN FOR THE AIRFOIL FAN ABOVE.		UTILIZES THE SAME HOUSING CONFIGURATION AS THE AIRFOIL DESIGN.
	RADIAL		SIMPLEST OF ALL CENTRIFUGAL FANS AND LEAST EFFICIENT HAS HIGH MECHANICAL STRENGTH AND THE WHEEL IS EASILY REPAIRED. FOR A GIVEN POINT OF RATING THIS FAN REQUIRES MEDIUM SPEED. THIS CLASSIFICATION INCLUDES RADIAL BLADES (R) AND MODIFIED RADIAL BLADES (M). USUALLY 6 TO 10 IN NUMBER.		SCROLL TYPE, USUALLY THE NARROWEST DESIGN OF ALL CENTRIFUGAL FAN DESIGNS DESCRIBED HERE, DUE TO LESS EFFICIENT WHEEL CAPABILITIES DIMENSIONAL REQUIREMENTS OF THIS HOUSING ARE NOT AS CRITICAL AS FOR AIRFOIL AND BACKWARDLY INCLINED FANS.
	FORWARD CURVED		EFFICIENCY IS SOMEWHAT LESS THAN AIRFOIL AND BACKWARDLY CURVED BLADED FANS. USUALLY FABRICATED OF LIGHTWEIGHT AND LOW COST CONSTRUCTION. HAS 24 TO 64 SHALLOW BLADES WITH BOTH THE HEEL AND TIP CURVED FORWARD. AIR LEAVES WHEEL AT VELOCITY GREATER THAN WHEEL TIP SPEED AND PRIMARY ENERGY TRANSFERRED TO THE AIR IS BY USE OF HIGH VELOCITY IN THE WHEEL. FOR GIVEN DUTY, WHEEL WILL BE THE SMALLEST OF ALL CENTRIFUGAL TYPES AND OPERATE AT LOWEST SPEED.		SCROLL IS SIMILAR TO AND OFTEN IDENTICAL TO OTHER CENTRIFUGAL FAN DESIGNS. THE FIT BETWEEN THE WHEEL AND INLET IS NOT AS CRITICAL AS ON AIRFOIL AND BACKWARDLY INCLINED BLADED FANS.
AXIAL FANS	PROPELLER		EFFICIENCY IS LOW. IMPELLERS ARE USUALLY OF INEXPENSIVE CONSTRUCTION AND LIMITED TO LOW PRESSURE APPLICATIONS. IMPELLER IS OF 2 OR MORE BLADES, USUALLY OF SINGLE THICKNESS ATTACHED TO RELATIVELY SMALL HUB. ENERGY TRANSFER IS PRIMARILY IN FORM OF VELOCITY PRESSURE.		SIMPLE CIRCULAR RING, ORIFICE PLATE OR VENTURI DESIGN. DESIGN CAN SUBSTANTIALLY INFLUENCE PERFORMANCE AND OPTIMUM DESIGN IS REASONABLY CLOSE TO THE BLADE TIPS AND FORMS A SMOOTH INLET FLOW CONTOUR TO THE WHEEL
	TUBEAXIAL		SOMEWHAT MORE EFFICIENT THAN PROPELLER FAN DESIGN AND IS CAPABLE OF DEVELOPING A MORE USEFUL STATIC PRESSURE RANGE. NUMBER OF BLADES USUALLY FROM 4 TO 8 AND HUB IS USUALLY LESS THAN 50% OF FAN TIP DIAMETER. BLADES CAN BE OF AIRFOIL OR SINGLE THICKNESS CROSS-SECTION.		CYLINDRICAL TUBE FORMED SO THAT THE RUNNING CLEARANCE BETWEEN THE WHEEL TIP AND TUBE IS CLOSE. THIS RESULTS IN SIGNIFICANT IMPROVEMENT OVER PROPELLER FANS.
	VANEAXIAL		GOOD DESIGN OF BLADES PERMITS MEDIUM TO HIGH PRESSURE CAPABILITY AT GOOD EFFICIENCY. THE MOST EFFICIENT FANS OF THIS TYPE HAVE AIRFOIL BLADES. BLADES ARE FIXED OR ADJUSTABLE PITCH TYPES AND HUB IS USUALLY GREATER THAN 50% OF FAN TIP DIAMETER.		CYLINDRICAL TUBE CLOSELY FITTED TO THE OUTER DIAMETER OF BLADE TIPS AND FITTED WITH A SET OF GUIDE VANES, UPSTREAM OR DOWNSTREAM FROM THE IMPELLER. GUIDE VANES CONVERT THE ROTARY ENERGY IMPARTED TO THE AIR AND INCREASE PRESSURE AND EFFICIENCY OF FAN.
SPECIAL DESIGNS	TUBULAR CENTRIFUGAL		THIS FAN USUALLY HAS A WHEEL SIMILAR TO THE AIRFOIL, BACKWARDLY INCLINED OR BACKWARDLY CURVED BLADE AS DESCRIBED ABOVE. (HOWEVER THIS FAN WHEEL TYPE IS OF LOWER EFFICIENCY WHEN USED IN FAN OF THIS TYPE.) MIXED FLOW IMPELLERS ARE SOMETIMES USED.		CYLINDRICAL SHELL SIMILAR TO A VANEAXIAL FAN HOUSING EXCEPT THE OUTER DIAMETER OF THE WHEEL DOES NOT RUN CLOSE TO THE HOUSING. AIR IS DISCHARGED RADIALLY FROM THE WHEEL AND MUST CHANGE DIRECTION BY 90 DEG. TO FLOW THROUGH THE GUIDE VANE SECTION.
	POWER ROOF VENTILATORS CENTRIFUGAL		MANY MODELS USE AIRFOIL (A) OR BACKWARD INCLINED (B) IMPELLER DESIGNS. THESE HAVE BEEN MODIFIED FROM THOSE MENTIONED ABOVE TO PRODUCE A LOW PRESSURE, HIGH VOLUME FLOW RATE CHARACTERISTIC. IN ADDITION MANY SPECIAL CENTRIFUGAL IMPELLER DESIGNS ARE USED INCLUDING MIXED FLOW DESIGN.		DOES NOT UTILIZE A HOUSING IN A NORMAL SENSE SINCE THE AIR IS SIMPLY DISCHARGED FROM THE IMPELLER IN A 360 DEG. PATTERN AND USUALLY DOES NOT INCLUDE A CONFIGURATION TO RECOVER THE VELOCITY PRESSURE COMPONENT.
	POWER ROOF VENTILATORS AXIAL		A GREAT VARIETY OF PROPELLER DESIGNS ARE EMPLOYED WITH THE OBJECTIVE OF HIGH VOLUME FLOW RATE AT LOW PRESSURE.		ESSENTIALLY A PROPELLER FAN MOUNTED IN A SUPPORTING STRUCTURE WITH A COVER FOR WEATHER PROTECTION AND SAFETY CONSIDERATIONS. THE AIR IS DISCHARGED THROUGH THE ANNULAR SPACE AROUND THE BOTTOM OF THE WEATHER HOOD.

*These performance curves reflect the general characteristics of various fans as commonly employed. They are not intended to provide the complete selection criteria for application purposes, since other parameters such as diameter, speed, etc. are not defined.

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Table 12-12A
Types of Fans (Continued)

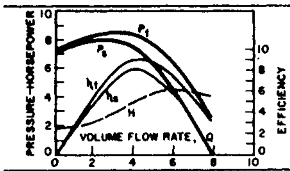
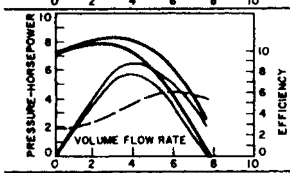
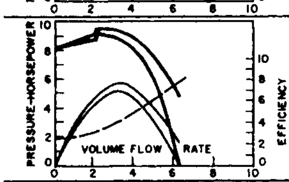
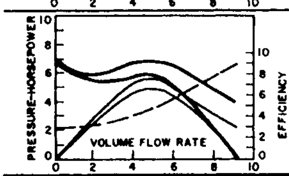
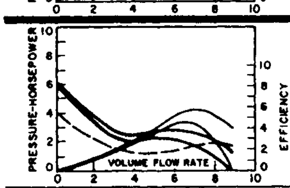
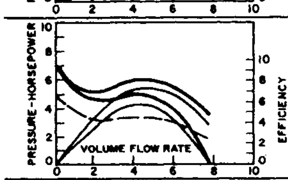
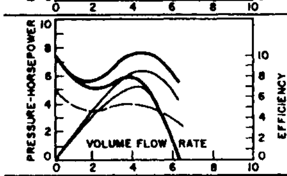
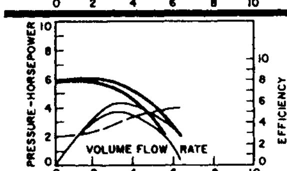
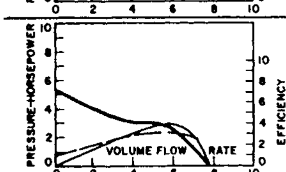
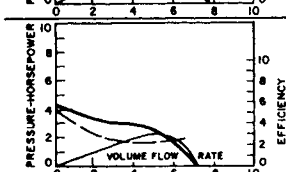
PERFORMANCE CURVES	PERFORMANCE CHARACTERISTICS*	APPLICATIONS*
	HIGHEST EFFICIENCIES OCCUR 50 TO 65% OF WIDE OPEN VOLUME. THIS IS ALSO THE AREA OF GOOD PRESSURE CHARACTERISTICS; THE HORSEPOWER CURVE REACHES A MAXIMUM NEAR THE PEAK EFFICIENCY AREA AND BECOMES LOWER TOWARDS FREE DELIVERY, A SELF LIMITING POWER CHARACTERISTIC AS SHOWN.	GENERAL HEATING, VENTILATING AND AIR CONDITIONING SYSTEMS. USUALLY APPLYING ONLY TO LARGE SYSTEMS WHERE THE SAVINGS IN POWER WILL BE SIGNIFICANT. CAN BE USED ON LOW, MEDIUM AND HIGH PRESSURE SYSTEMS. USED IN LARGE SIZES FOR CLEAN AIR INDUSTRIAL APPLICATIONS WHERE POWER SAVINGS WILL BE SIGNIFICANT.
	OPERATING CHARACTERISTICS OF THIS FAN ARE SIMILAR TO THE AIRFOIL FAN MENTIONED ABOVE. PEAK EFFICIENCY FOR THIS FAN IS SLIGHTLY LOWER THAN THE AIRFOIL FAN.	SAME. HEATING, VENTILATING AND AIR CONDITIONING APPLICATIONS AS THE AIRFOIL FAN. IS ALSO USED IN SOME INDUSTRIAL APPLICATIONS WHERE THE AIRFOIL BLADE IS NOT ACCEPTABLE DUE TO CORROSIVE AND/OR EROSION ENVIRONMENT.
	HIGHER PRESSURE CHARACTERISTICS THAN THE ABOVE MENTIONED FANS.CURVE MAY HAVE A BREAK LEFT OF PEAK PRESSURE BUT THIS USUALLY IS NOT SUFFICIENT TO CAUSE DIFFICULTY. POWER RISES CONTINUALLY TO FREE DELIVERY.	USED PRIMARILY FOR MATERIAL HANDLING APPLICATIONS IN INDUSTRIAL PLANTS. WHEEL CAN BE OF RUGGED CONSTRUCTION AND IS SIMPLE TO REPAIR IN THE FIELD. WHEEL IS SOMETIMES COATED WITH SPECIAL MATERIAL. THIS DESIGN ALSO USED FOR HIGH PRESSURE INDUSTRIAL REQUIREMENTS. NOT COMMONLY FOUND IN HVAC APPLICATIONS.
	PRESSURE CURVE IS LESS STEEP THAN THAT OF BACKWARD CURVED BLADED FANS. THERE IS A DIP IN THE PRESSURE CURVE LEFT OF THE PEAK PRESSURE POINT AND HIGHEST EFFICIENCY OCCURS TO THE RIGHT OF PEAK PRESSURE, 40 TO 50% OF WIDE OPEN VOLUME. FAN SHOULD BE RATED TO THE RIGHT OF PEAK PRESSURE. POWER CURVE RISES CONTINUALLY TOWARD FREE DELIVERY AND THIS MUST BE TAKEN INTO ACCOUNT WHEN MOTOR IS SELECTED.	USED PRIMARILY IN LOW PRESSURE HEATING, VENTILATING AND AIR CONDITIONING APPLICATIONS SUCH AS DOMESTIC FURNACES, CENTRAL STATION UNITS AND PACKAGED AIR CONDITIONING EQUIPMENT FROM ROOM AIR CONDITIONING UNITS TO ROOF TOP UNITS.
	HIGH FLOW RATE BUT VERY LOW PRESSURE CAPABILITIES AND MAXIMUM EFFICIENCY IS REACHED NEAR FREE DELIVERY. THE DISCHARGE PATTERN OF THE AIR IS CIRCULAR IN SHAPE AND THE AIRSTREAM SWIRLS DUE TO THE ACTION OF THE BLADES AND THE LACK OF STRAIGHTENING FACILITIES.	FOR LOW PRESSURE, HIGH VOLUME AIR MOVING APPLICATIONS SUCH AS AIR CIRCULATION WITHIN A SPACE OR VENTILATION THROUGH A WALL WITHOUT ATTACHED DUCT WORK. USED FOR MAKE-UP AIR APPLICATIONS.
	HIGH FLOW RATE CHARACTERISTICS WITH MEDIUM PRESSURE CAPABILITIES. PERFORMANCE CURVE INCLUDES A DIP TO THE LEFT OF PEAK PRESSURE WHICH SHOULD BE AVOIDED. THE DISCHARGE AIR PATTERN IS CIRCULAR AND IS ROTATING OR WHIRLING DUE TO THE PROPELLER ROTATION AND LACK OF GUIDE VANES.	LOW AND MEDIUM PRESSURE DUCTED HEATING, VENTILATING AND AIR CONDITIONING APPLICATIONS WHERE AIR DISTRIBUTION ON THE DOWNSTREAM SIDE IS NOT CRITICAL. ALSO USED IN SOME INDUSTRIAL APPLICATIONS SUCH AS DRYING OVENS, PAINT SPRAY BOOTHS AND FUME EXHAUST SYSTEMS.
	HIGH PRESSURE CHARACTERISTICS WITH MEDIUM VOLUME FLOW RATE CAPABILITIES. PERFORMANCE CURVE INCLUDES A DIP, CAUSED BY AERODYNAMIC STALL, TO THE LEFT OF PEAK PRESSURE WHICH SHOULD BE AVOIDED. GUIDE VANES CORRECT THE CIRCULAR MOTION IMPARTED TO THE AIR BY THE WHEEL AND IMPROVE PRESSURE CHARACTERISTICS AND EFFICIENCY OF THE FAN.	GENERAL HEATING, VENTILATING AND AIR CONDITIONING SYSTEMS IN LOW, MEDIUM AND HIGH PRESSURE APPLICATIONS IS OF ADVANTAGE WHERE STRAIGHT THRU FLOW AND COMPACT INSTALLATION ARE REQUIRED, AIR DISTRIBUTION ON DOWNSTREAM SIDE IS GOOD. ALSO USED IN INDUSTRIAL APPLICATION SIMILAR TO THE TUBEAXIAL FAN. RELATIVELY MORE COMPACT THAN COMPARABLE CENTRIFUGAL TYPE FANS FOR SAME DUTY.
	PERFORMANCE IS SIMILAR TO BACKWARD CURVED FAN EXCEPT LOWER CAPACITY AND PRESSURE. DUE TO THE 90 DEG. CHANGE IN DIRECTION OF THE AIRFLOW IN THE HOUSING THE EFFICIENCY WILL BE LOWER THAN THE BACKWARD CURVED FAN. SOME DESIGNS MAY HAVE A DIP IN THE CURVE SIMILAR TO THE AXIAL FLOW FAN.	USED PRIMARILY FOR LOW PRESSURE RETURN AIR SYSTEMS IN HEATING, VENTILATING AND AIR CONDITIONING APPLICATIONS. HAS STRAIGHT THRU FLOW CONFIGURATION.
	USUALLY INTENDED TO OPERATE WITHOUT ATTACHED DUCTWORK AND THEREFORE TO OPERATE AGAINST A VERY LOW PRESSURE HEAD. IT IS USUALLY INTENDED TO HAVE A RATHER HIGH VOLUME FLOW RATE CHARACTERISTIC. ONLY STATIC PRESSURE AND STATIC EFFICIENCY ARE SHOWN FOR THIS TYPE OF PRODUCT.	FOR LOW PRESSURE EXHAUST SYSTEMS SUCH AS GENERAL FACTORY, KITCHEN, WAREHOUSE, AND SOME COMMERCIAL INSTALLATIONS WHERE THE LOW PRESSURE RISE LIMITATION CAN BE TOLERATED. UNIT IS LOW IN FIRST COST AND LOW IN OPERATING COST AND PROVIDES POSITIVE EXHAUST VENTILATION IN THE SPACE WHICH IS A DECIDED ADVANTAGE OVER GRAVITY TYPE EXHAUST UNITS. THE CENTRIFUGAL UNIT IS SOMEWHAT QUIETER THAN THE AXIAL UNIT DESCRIBED BELOW.
	USUALLY INTENDED TO OPERATE WITHOUT ATTACHED DUCTWORK AND THEREFORE TO OPERATE AGAINST VERY LOW PRESSURE HEAD. IT IS USUALLY INTENDED TO HAVE A HIGH VOLUME FLOW RATE CHARACTERISTIC. ONLY STATIC PRESSURE AND STATIC EFFICIENCY ARE SHOWN FOR THIS TYPE PRODUCT.	FOR LOW PRESSURE EXHAUST SYSTEMS SUCH AS GENERAL FACTORY, KITCHEN, WAREHOUSE AND SOME COMMERCIAL INSTALLATIONS WHERE THE LOW PRESSURE RISE LIMITATION CAN BE TOLERATED. UNIT IS LOW IN FIRST COST AND LOW IN OPERATING COST AND PROVIDES POSITIVE EXHAUST VENTILATION IN THE SPACE WHICH IS A DECIDED ADVANTAGE OVER GRAVITY TYPE EXHAUST UNITS.

Table 12-12B
Relative Characteristics of Centrifugal Fans

	Backwardly Curved	Radial Blade	Forwardly Curved
First cost*	High	Medium	Low
Efficiency	High	Medium	Low
Stability of operation	Good	Good	Poor
Space required	Medium	Medium	Small
Tip speed	High	Medium	Low
Resistance to abrasion	Medium	Good	Poor
Ability to handle sticky materials	Medium	Good	Poor

*For normal-duty applications, first cost is essentially the same for all types. Heavy duty applications reflect the comparison listed.

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ial design would be similar; see Table 12-11. The propeller fan is used for high volume and very low static pressure systems, Figure 12-125.

Construction

Wheels

For most general-service noncorrosive applications, the wheels use medium- to heavy-gauge carbon or alloy steel. For the centrifugal types, a die-formed entrance shroud provides smooth entrance flow of the air into the wheel. A solid steel plate serves as a back plate of the single entrance wheel, but of course, cannot be used on a double-entry wheel. Here, both sides of the wheel have entrance shrouds, Figure 12-120C.

For most fans, blades are riveted to the back plate or hub respectively. For centrifugals, the blades are usually welded to the shroud; however, they can be welded to both shroud and backplate. For some designs and classes of service, an extra reinforcing ring is welded to the blades between the backplate and the shroud. This is particularly necessary for large heavy-duty units; see Figure 12-120B.

For corrosive service the wheel can be made of stainless or nonferrous alloys; can be all plastic in some designs; or may be covered with rubber, lead, or plastic. Rubber-covered fans are usually limited in wheel peripheral velocity to 12,000–15,000 ft/min, although some wheels with certain types of rubber can run higher.[8]

Rubber-covered fan temperatures are usually limited to 130–180°F; again these limitations are a function of gas atmosphere and the specific rubber compound.

Lead is applied as a homogeneous coating to a thickness of $^1/_8$–$^1/_4$-in.maximum on the wheel and as a coating or lining of $^1/_8$–$^1/_4$-in. to the inside of the housing. The wheel is limited to a peripheral velocity of 12,000 ft per min at a maximum temperature of 200°F, and at 9,000 ft per min the temperature can reach an allowable maximum of 400°F.[8]

Wheels are also made of solid sheet bronze or aluminum for nonsparking service. The ventilating and industrial fan identification is given in Table 12-11.

Housing or Casing

The housing is streamlined in design to give good air flow characteristics. It may be of riveted welded or bolted metal sheets. The usual material is steel, although construction is available to meet corrosion conditions described for the wheel.

Evase Discharge

See Figure 12-126. An evase is a diffuser at the fan outlet that gradually increases in area to decrease velocity and to convert kinetic energy to static pressure (regain).[128] This aids in providing optimum performance of the fan; however, the evase must usually be requested as it is not normally built onto the fan, but the owner's design engineer can design this same effect into the attaching duct from the fan to the system.

Inlet Bell

The inlet bell may be arranged in several ways or attached by different methods, but generally it is dieformed to contours that direct the air into the entrance of the wheel. It seals with an overlap of the wheel shroud over the inlet bell.

Stationary or moveable inlet vanes, if used, attach to the inlet bell. These are also dieformed for uniformity and are welded in place for the stationary type and attached to a spider that is attached to the bell for the movable type, Figures 12-127A–B. The discharge damper, Figure 12-127C, is used for system volume and pressure control and is discussed later in regard to comparative performance with inlet vanes.

Specifications

To properly recommend and quote fans for a specific application, the process engineer must either (a) evaluate the manufacturer's rating tables or (b) furnish proper data for others to do this. The basic information includes

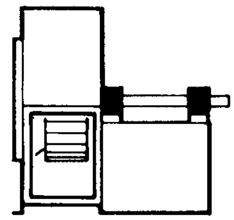
A. Gas or vapor

1. Inlet temperature and pressure, density (lb/ft^3)
2. Discharge pressure, absolute or static pressure, in. water.
3. Nature of gas: corrosion, entrained solid particles, entrained fibers, and entrained moisture.
4. Capacity, cfm at 60°F and 14.7 psia or at other *specified* conditions.

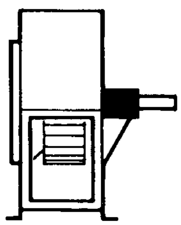
SW - Single Width **DW** - Double Width
SI - Single Inlet **DI** - Double Inlet

Arrangements 1, 3, 7 and 8 are also available with bearings mounted on pedestals or base set independent of the fan housing

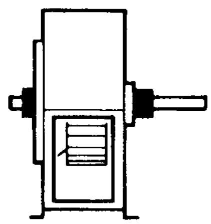
For designation of rotation and discharge,	see 99-2406.
For motor position, belt or chain drive,	see 99-2407.
For designation of position of inlet boxes,	see 99-2405.



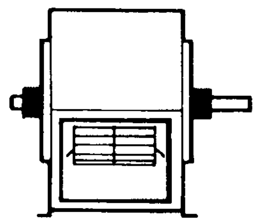
ARR. 1 SWSI For belt drive or direct connection. Impeller overhung. Two bearings on base



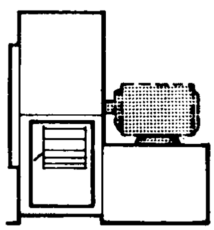
ARR. 2 SWSI For belt drive or direct connection. Impeller overhung. Bearings in bracket supported by fan housing.



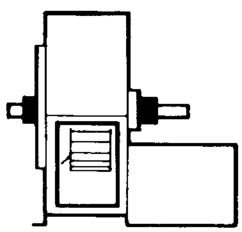
ARR. 3 SWSI For belt drive or direct connection. One bearing on each side and supported by fan housing.



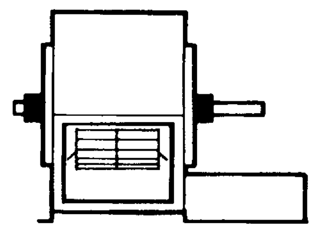
ARR. 3 DWDI For belt drive or direct connection. One bearing on each side and supported by fan housing.



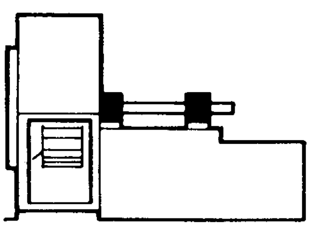
ARR. 4 SWSI For direct drive. Impeller overhung on prime mover shaft. No bearings on fan. Prime mover base mounted or integrally directly connected.



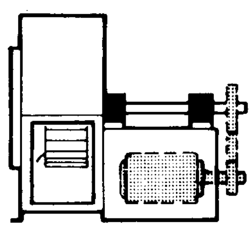
ARR. 7 SWSI For belt drive or direct connection. Arrangement 3 plus base for prime mover.



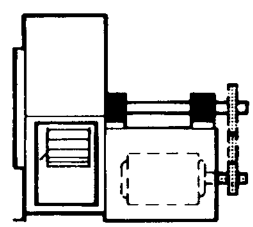
ARR. 7 DWDI For belt drive or direct connection. Arrangement 3 plus base for prime mover.



ARR. 8 SWSI For belt drive or direct connection. Arrangement 1 plus extended base for prime mover.



ARR. 9 SWSI For belt drive. Impeller overhung, two bearings, with prime mover outside base.



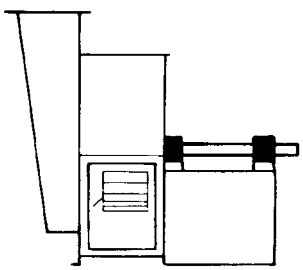
ARR. 10 SWSI For belt drive. Impeller overhung, two bearings, with prime mover inside base.

AMCA STANDARD 99-2404-78 PAGE 1 OF 2	DRIVE ARRANGEMENTS FOR CENTRIFUGAL FANS	Adopted 10-9-78 / Reviewed 1983

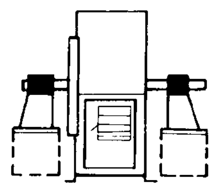
Figure 12-119A, Page 1 and 2 of 2. AMCA standard drive arrangements for centrifugal fans. (Used by permission: Reprinted from *AMCA Publication 99-86 Standards Handbook,* ©1986, Standard AMCA No. 99-2404-78, with written permission from Air Movement and Control Association International, Inc., ©1986. All rights reserved.)

SW - Single Width **DW** - Double Width
SI - Single Inlet **DI** - Double Inlet

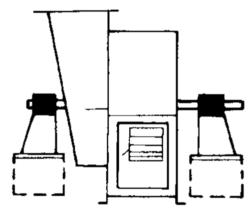
For designation of rotation and discharge, see 99-2406.
For motor position, belt or chain drive, see 99-2407.
For designation of position of inlet boxes, see 99-2405.



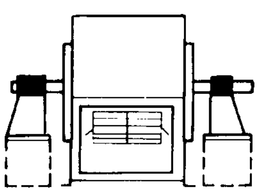
ARR. 1 SWSI WITH INLET BOX For belt drive or direct connection. Impeller overhung, two bearings on base. Inlet box may be self-supporting.



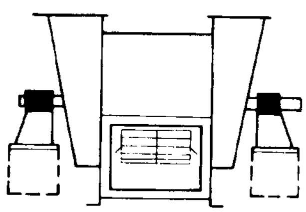
ARR. 3 SWSI WITH INDEPENDENT PEDESTAL For belt drive or direct connection fan. Housing is self-supporting. One bearing on each side supported by independent pedestals.



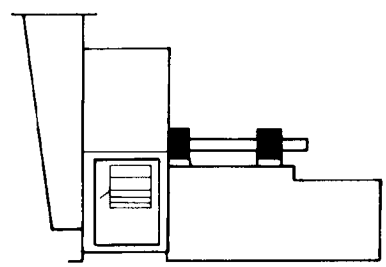
ARR. 3 SWSI WITH INLET BOX AND INDEPENDENT PEDESTALS For belt drive or direct connection fan. Housing is self-supporting. One bearing on each side supported by independent pedestals with shaft extending through inlet box.



ARR. 3 DWDI WITH INDEPENDENT PEDESTAL For belt drive or direct connection fan. Housing is self-supporting. One bearing on each side supported by independent pedestals.



ARR. 3 DWDI WITH INLET BOX AND INDEPENDENT PEDESTALS For belt drive or direct connection fan. Housing is self-supporting. One bearing on each side supported by independent pedestals with shaft extending through inlet box.

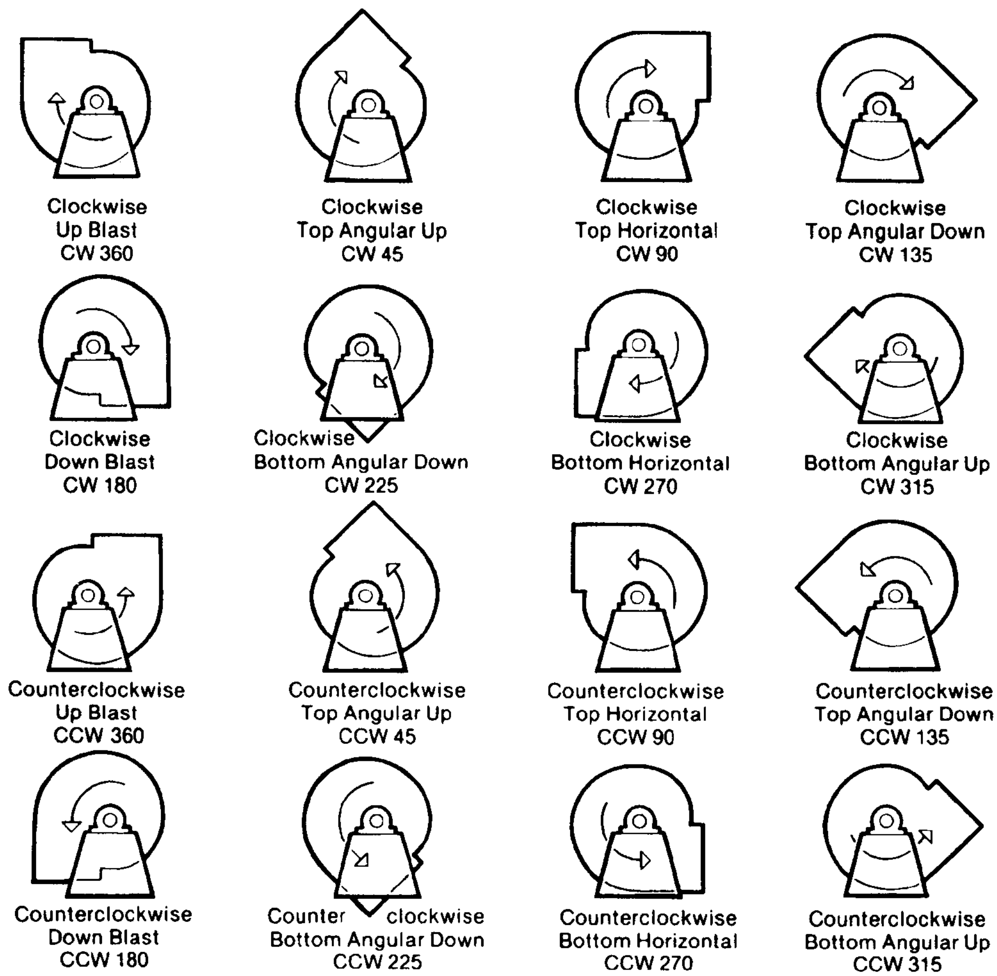


ARR. 8 SWSI WITH INLET BOX For belt drive or direct connection. Impeller overhung, two bearings on base plus extended base for prime mover. Inlet box may be self-supporting.

AMCA STANDARD 99-2404-78 PAGE 2 OF 2	**DRIVE ARRANGEMENTS FOR CENTRIFUGAL FANS**	**Adopted 10-9-78**
		Reviewed 1983

Supersedes 2404-78

Figure 12-119A, Page 2 of 2. AMCA standard drive arrangements for centrifugal fans. (Reprinted from: *AMCA Publication 99-86 Standards Handbook,* ©1986, Standard AMCA No. 99-2404-78, with written permission from Air Movement and Control Association International, Inc., ©1986. All rights reserved.)



Notes:

1. Direction of rotation is determined from drive side of fan.
2. On single inlet fans, drive side is always considered as the side opposite fan inlet.
3. On double inlet fans with drives on both sides, drive side is that with the higher powered drive unit.
4. Direction of discharge is determined in accordance with diagrams. Angle of discharge is referred to the vertical axis of fan and designated in degrees from such standard reference axis. Angle of discharge may be any intermediate angle as required.
5. For fan inverted for ceiling suspension, or side wall mounting, direction of rotation and discharge is determined when fan is resting on floor.

AMCA STANDARD 99-2406-83	**DESIGNATIONS FOR ROTATION AND DISCHARGE OF CENTRIFUGAL FANS**	**Adopted 2-22-83** **Reviewed**

Figure 12-119B. AMCA standard designation of rotation and discharge of centrifugal fans. Reprinted with permission: *AMCA Publication 99-86 Standards Handbook,* ©1986, Standard AMCA No. 99-2406-83, with written permission from Air Movement and Control Association International, Inc., ©1986. All rights reserved.

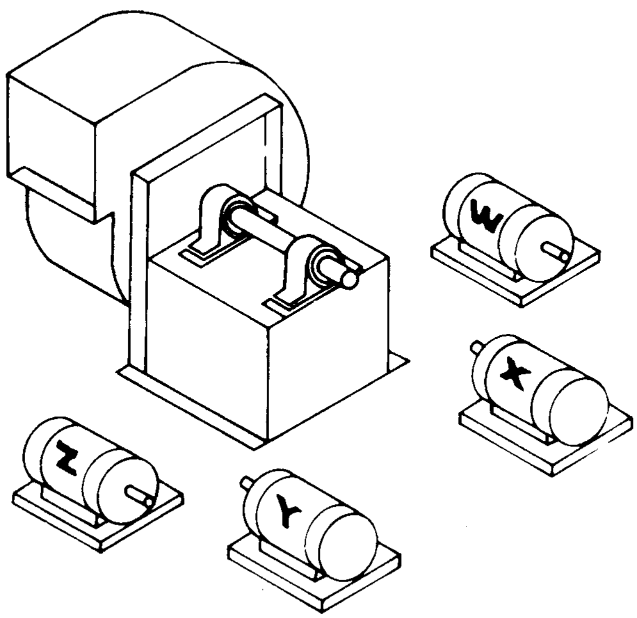


Figure 12-119C. AMCA standard motor positions for belt or chain drive centrifugal fans. Reprinted from *AMCA Publication 99-86 Standards Handbook*, ®1986, Standard AMCA No. 99-2407-66, with written permission from Air Movement and Control Association International, Inc., ©1986. All rights reserved.)

B. Operation and control

1. Normal expected static pressure based on system resistance at
   a. Maximum point of operation.
   b. Minimum point of operation.
   c. System diagram indicating type of resistances on suction and discharge sides of fan.
2. Physical diagram indicating fan arrangement with respect to other parts of system
3. Type of control of air flow
   a. Fixed or automatic inlet vanes
   b. Discharge damper
   c. Combination
4. Operation: continuous, intermittent, hours per day

C. Recommended materials of construction

1. All parts in contact with gas
2. Acceptable coatings, linings, and plastic

D. Noise level at fan, at 10 ft from fan

E. Type fan preferred or recommended

1. Centrifugal
   a. Type of wheel blades (radial, backward, or forward)
   b. Arrangement

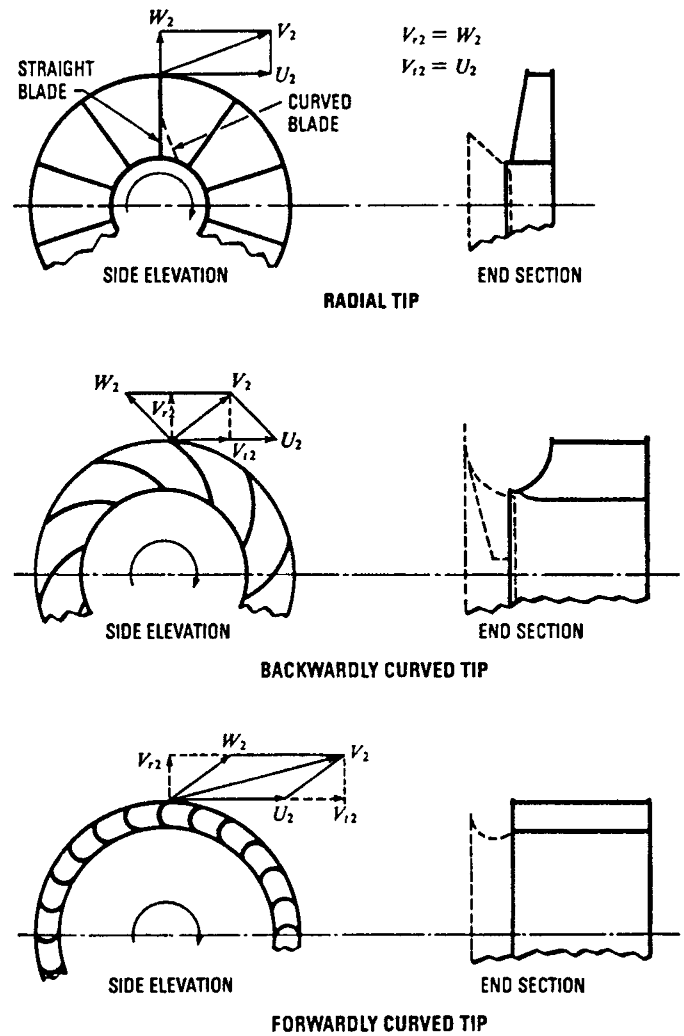
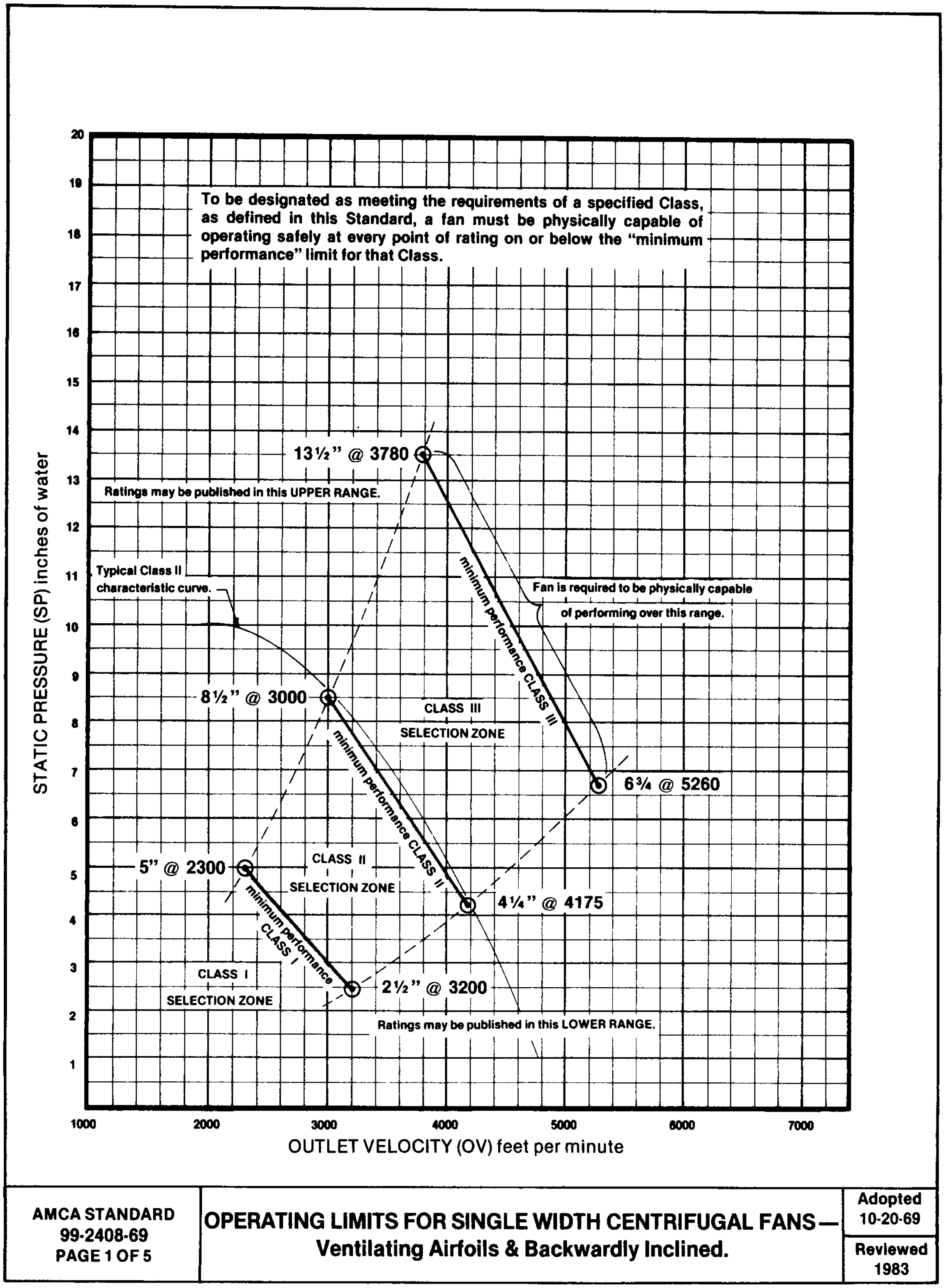


Figure 12-119D. Centrifugal blade design for backward, radial, and forward tips. (Used by permission: *Fan Engineering*, 8th Ed., ©1983. Buffalo Forge Co. , Howden Fan Co. All rights reserved.)

   c. Rotation: clockwise, counterclockwise
   d. Speed: rpm
   e. Discharge location
   f. Width wheel (single, double)
   g. Inlet (single, double); type connection
   h. Dampers (inlet, outlet)
   i. Access openings in casing: clean out (size), drains (size)
   j. Type bearings: water-cooled, cooling wheel
   k. Shaft seal: type
2. Vaneaxial or tubeaxial
   a. Arrangement
   b. Rotation
   c. Speed: rpm
   d. Mounting (vertical, horizontal)



Supersedes 2408-69

Figure 12-119E. AMCA standard operating limits classes for single width centrifugal fans—ventilating airfoils and backwardly inclined. Other designs with different strength parameters are defined in the same publication noted. (Reprinted by permission: *AMCA Publication 99-86 Standards Handbook,* ©1986, Standard AMCA No. 99-2408-69, with written permission from Air Movement and Control Association International, Inc., ©1986. All rights reserved.)

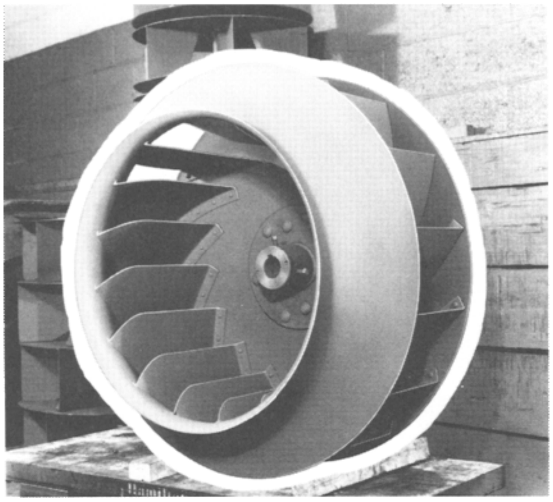


Figure 12-120A. Backwardly curved blades, Class 1 rotor. (Used by permission: The Howden Fan Co.)

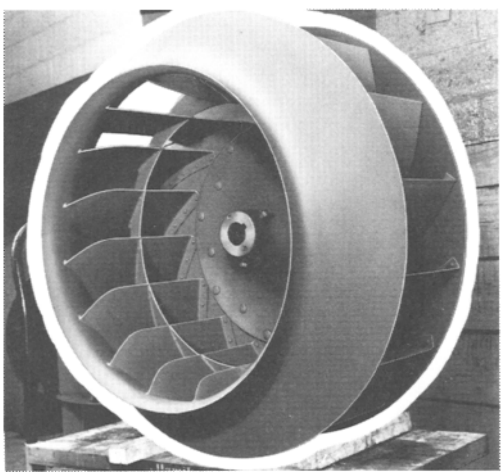


Figure 12-120B. Backwardly curved blades, Class 11 rotor. (Used by permission: The Howden Fan Company.)

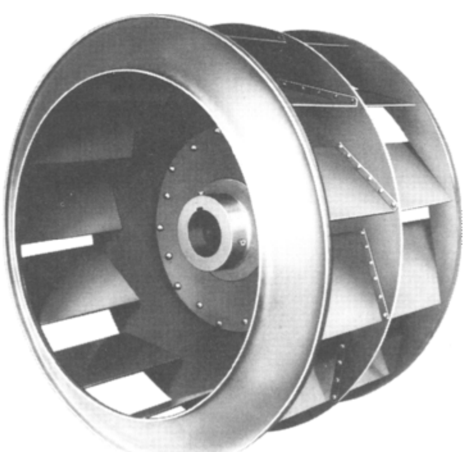


Figure 12-120C. Double-inlet, double-width wheel, backward blades. (Used by permission: The Howden Fan Company.)

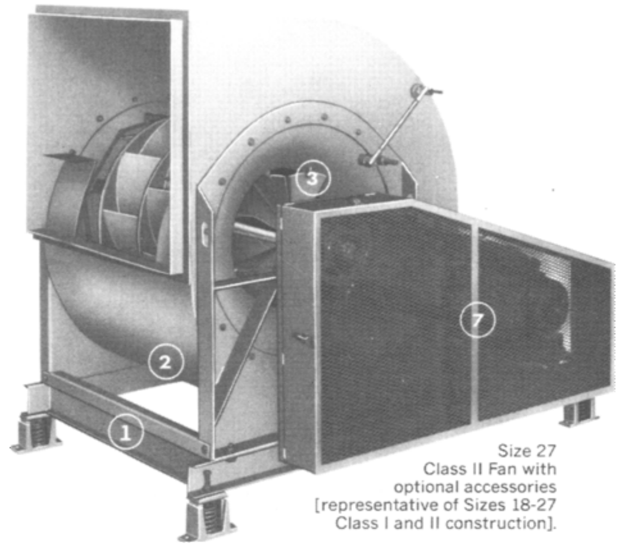


Figure 12-120D. Fan assembly with double-width wheel, inlet vane dampers, and belt drive with guard. (Used by permission: Bul. 141, ©1995. The New York Blower Co.® For more information, contact the company at www.nyb.com.)

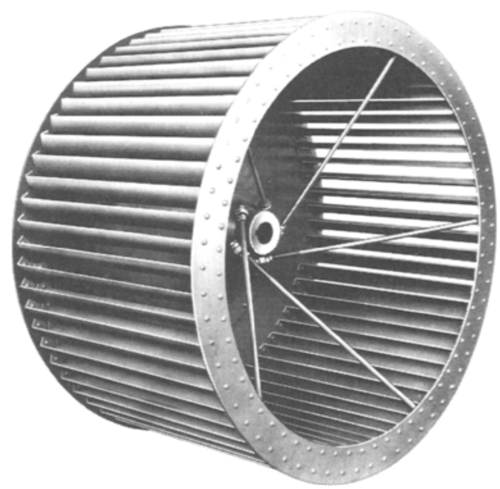
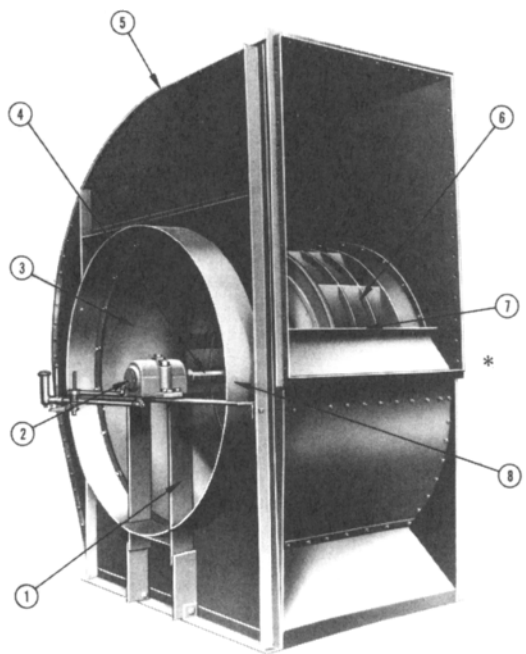


Figure 12-121. Single-width wheel (or impeller) with forward curved blades. (Used by permission: The Howden Fan Company.)

   e. Inlet and outlet: special tapered, inlet cone, outlet cone, type connection
   f. Access openings
   g. Supports
3. Propeller
   a. Mounting: vertical, horizontal
   b. Propeller: type, make, and material
   c. Speed: rpm
   d. Propeller tip speed
   e. Safety guard



1. Bearing supports
2. Bearings. Can be either sleeve or antifriction type as desired. Sleeve bearings shown.
3. Inlet
4. Shaft
5. Housing
6. Wheel
7. Cut-off. Continues the scroll shape of fan housing.
8. Inlet collar

*Note: driver connection to shaft, this side.
*Note: driver connection to shaft, this side.

Figure 12-122. Typical inlet side view, Arrangement 3, Class 1, single-width, single-inlet, counterclockwise rotation with top horizontal discharge centrifugal fan. (Used by permission: Clarage, A Twin Cities Fan Co., Birmingham, Ala.)

F. Mechanical

1. Brake horsepower required
2. Motor or other driver horsepower recommended and speed
3. Gas velocity at fan outlet discharge
4. Wheel: diameter, material, and thickness, type construction
5. Blades: material and thickness, how attached to wheel
6. Guide vanes: material and how operated
7. Casing and components: give material, thickness
8. Bearings: make, type, and how lubricated
9. Belt: static-free, number required, and guard
10. Gears: make, model, horsepower rating, and load factor

Fan Drivers

V-belts using electric motors are the most common drive for process applications. Belts should be selected to be

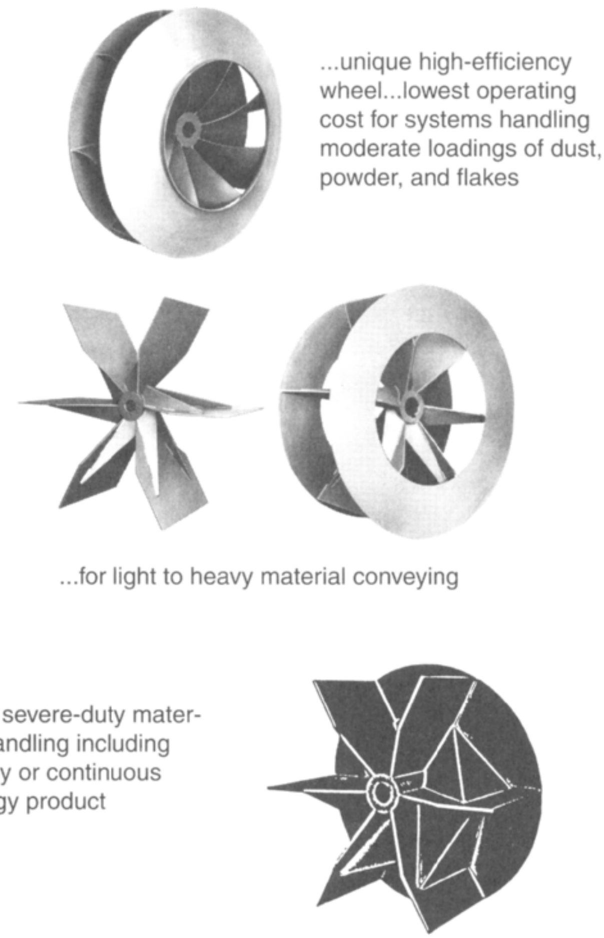
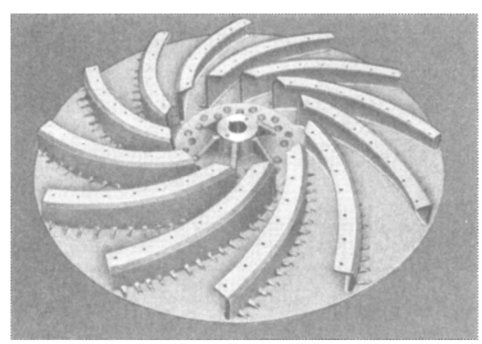


Figure 12-123A. Fan wheel choices for handling dust and other materials-handling applications. (Used by permission: The New York Blower Co.®, Bull. "Quality Equipment Guide," pg. 9. For more information, contact the company at www.nyb.com.)



Close-up view of the D wheel showing the method of attachment of the chrome carbide hardfaced plate covering the entire blade working surface. This application included a high pressure development capability which required a very narrow width wheel design. Maintainability was enhanced by incorporating a removable flange (shroud) design (not shown).

Figure 12-123B-1. Special blade construction for handling dirty gases. (Used by permission: Bul. C-5200. The Howden Fan Co.)

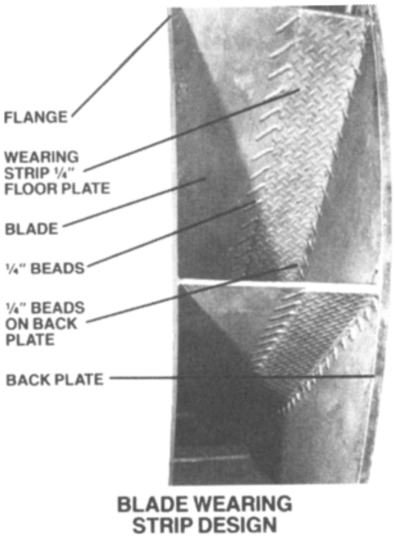


Figure 12-123B-2. Wear strip construction on induced draft fan for dirty gases. The inertia of suspended dust particles carries them toward the backplate or centerplate where the wear plate withstands the abrasion normally affecting the blade. (Used by permission: Bul. 2-5100. The Howden Fan Co.)

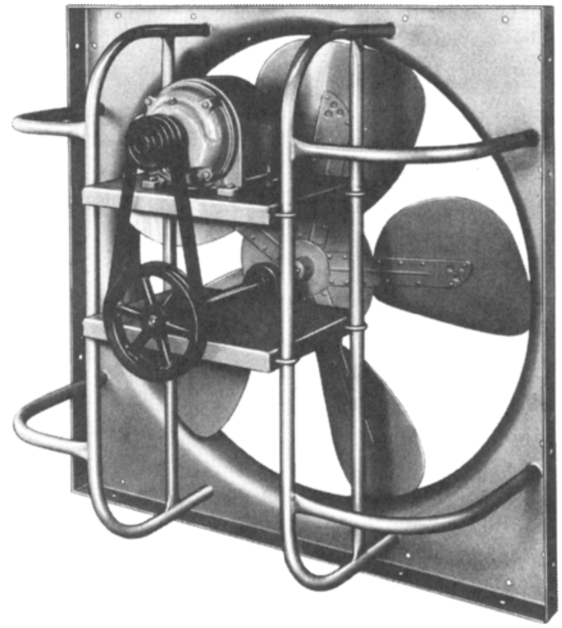
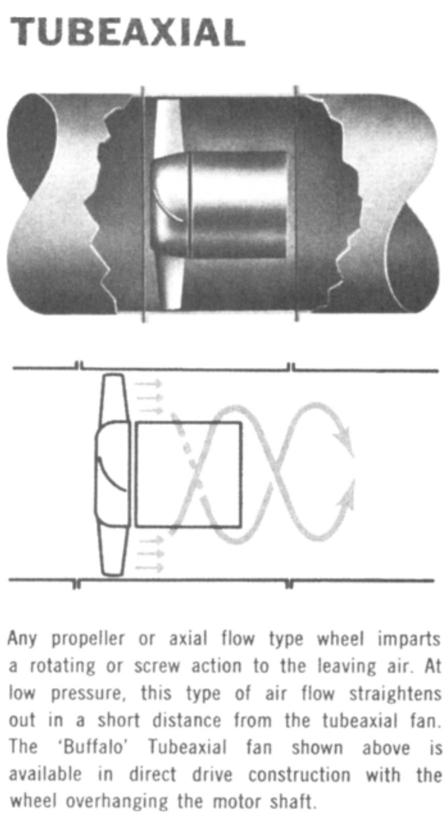


Figure 12-125. Square panel, v-belt drive propeller fan. (Used by permission: ©The New York Blower Co. For more information, contact the company at www.nyb.com.)



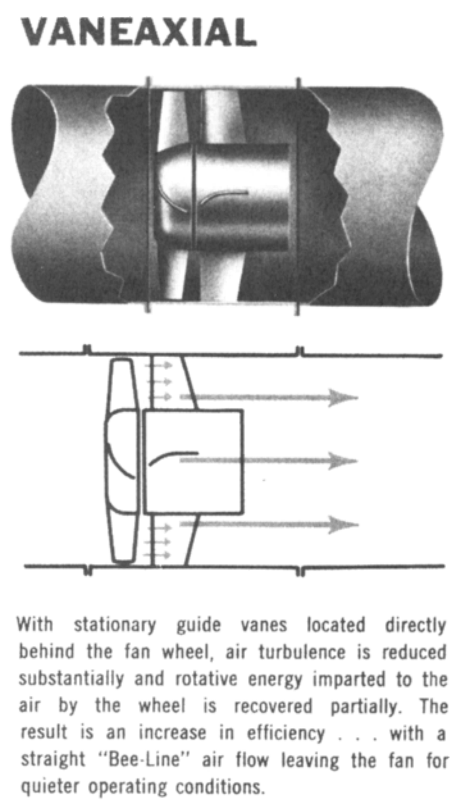


Figure 12-124. Comparison of tubeaxial and vaneaxial fans. Drives can be belt or direct motor. (Used by permission: Bul. F-305 D, 1991, ©1968. The Howden Fan Co.)

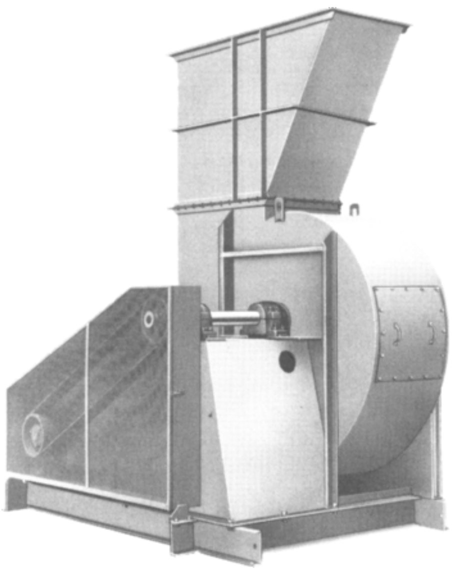


Figure 12-126. Evase duct attachment on fan discharge to improve the regain of static pressure. Fans can be purchased with or without the evase section. (Used by permission: Bul. 331, ©1992. The New York Blower Co.® For more information, contact the company at www.nyb.com.)

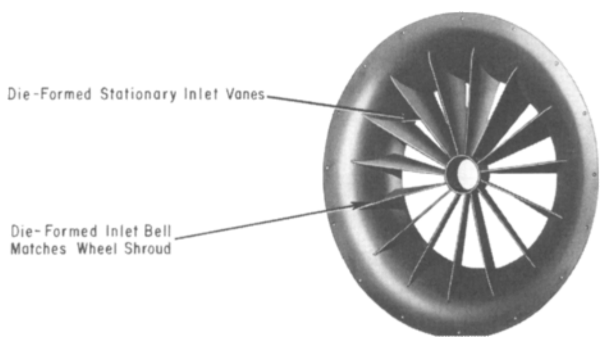


Figure 12-127A. Stationary inlet vanes. (Used by permission: The Howden Fan Co.)

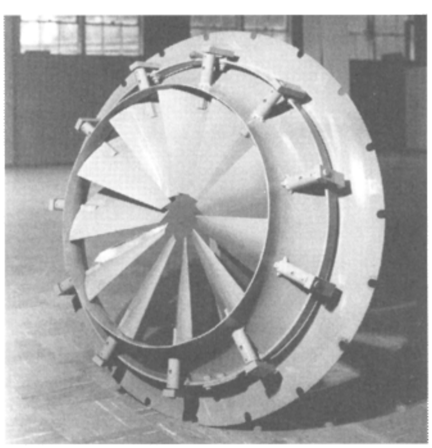


Figure 12-127B. The reverse side of the variable inlet vane shows the operating mechanism. (Used by permission: The Howden Fan Co.)

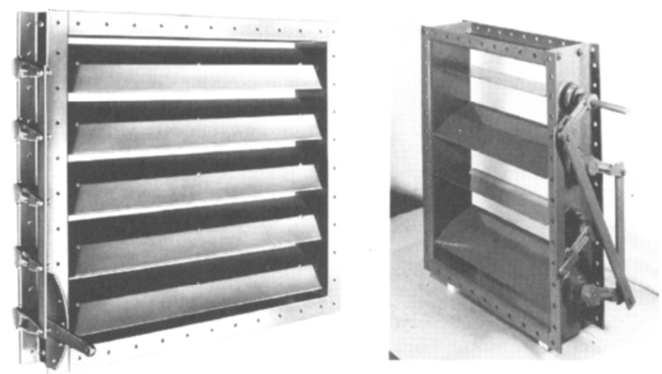


Figure 12-127C. Outlet dampers: left, standard outlet damper; right, stream flow outlet damper. (Used by permission: The Howden Fan Co.)

capable of safely handling maximum horsepower. The fan shaft requires a certain horsepower for proper operation, and to this must be added mechanical losses of belts, gears or any other device imposed between the fan and its driver. The driver shaft output rating should be larger than the requirement at this point. Other drives include steam turbine, gas pressure reducing turbine, gas engines, and variable-speed motors.

Performance

Performance details and definitions are given in the *Standards* of the Air Movement and Control Association International, Inc.[49] Previously, the National Association of Fan Manufacturers merged with this organization.

For fan pressure conditions, see Figure 12-129.

Summary of Fan Selection and Rating

1. Before a fan selection can be completed either by the owner's engineer or the manufacturer, a system resistance calculation must be made at several selected fan volume flow rates. This will be discussed in the section, "Summary of Fan System Calculations," Figure 12-128.[128, 132]
2. Determine the air or gas density at inlet to fan.
   a. Standard air density for many fan ratings = 0.075 lb/ft^3 corresponding to 70°F and 29.92 in. Hg. barometer.
   b. Actual *air* density = (0.075) (air density ratio, Table 12-13 for actual elevation, barometer, and inlet temperature).
   c. Gas or mixture other than air, calculated inlet conditions density, lb/ft^3.

3. Determine fan inlet air or gas/mixture (including water vapor) *actual* volume (acfm) to enter the fan, cfm

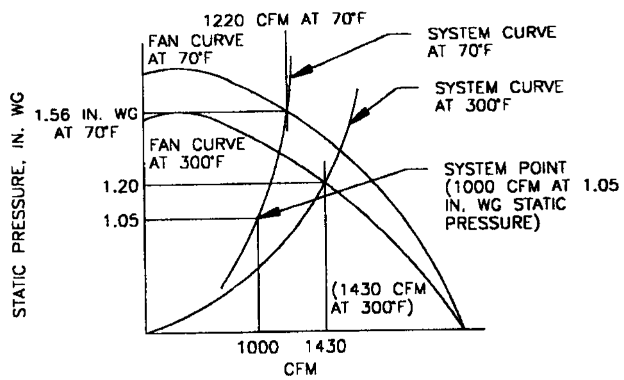


Figure 12-128. Fan-system curve relationship with fan at different temperatures. (Used by permission: Engineering Letter No. 4, p. 5. ©The New York Blower Co. For more information, contact the company at www.nyb.com.)

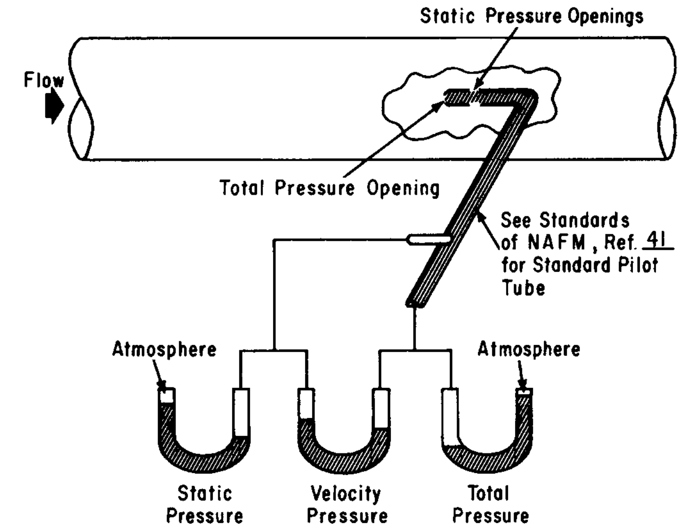


Figure 12-129. Pressure readings from a pitot tube.

Table 12-13
Air Density Ratio

At Selected Altitude or Barometer

$$\text{Ratio} = \frac{(530°\text{R})\,(\text{Actual Barometer})}{(\text{Actual.°R})\,(29.92)}$$

Thus, New Density = (Standard 0.075)(Ratio), Lb/ft³

Air Temperature °F	Altitude in Ft Above Sea Level: 0	1,000	2,000	3,000	4,000	5,000	6,000	7,000	8,000	9,000	10,000	15,000	20,000
	Barometric Pressure in In.: 29.92	**28.86**	**27.82**	**26.81**	**25.84**	**24.89**	**23.98**	**23.09**	**22.22**	**21.38**	**20.58**	**16.88**	**13.75**
70°	1.000	.964	.930	.896	.864	.832	.801	.772	.743	.714	.688	.564	.460
100°	.946	.912	.880	.848	.818	.787	.758	.730	.703	.676	.651	.534	.435
150°	.869	.838	.808	.770	.751	.723	.696	.671	.646	.620	.598	.490	.400
200°	.803	.774	.747	.720	.694	.668	.643	.620	.596	.573	.552	.453	.369
250°	.747	.720	.694	.669	.645	.622	.598	.576	.555	.533	.514	.421	.344
300°	.697	.672	.648	.624	.604	.580	.558	.538	.518	.498	.480	.393	.321
350°	.654	.631	.608	.586	.565	.544	.524	.505	.486	.467	.450	.369	.301
400°	.616	.594	.573	.552	.532	.513	.493	.476	.458	.440	.424	.347	.283
450°	.582	.561	.542	.522	.503	.484	.466	.449	.433	.416	.401	.328	.268
500°	.552	.532	.513	.495	.477	.459	.442	.426	.410	.394	.380	.311	.254
550°	.525	.506	.488	.470	.454	.437	.421	.405	.390	.375	.361	.296	.242
600°	.500	.482	.465	.448	.432	.416	.400	.386	.372	.352	.344	.282	.230
650°	.477	.460	.444	.427	.412	.397	.382	.368	.354	.341	.328	.269	.219
700°	.457	.441	.425	.410	.395	.380	.366	.353	.340	.326	.315	.258	.210

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at actual temperature and pressure. The acfm is the conventional catalog rating unit for fans[128] because a fan handles *the same volume* of air/gas at any density. Acfm should be used when specifying a fan.[128]

4.

Corrected static pressure (SP)

$$= \frac{\text{actual SP}}{\text{air sensity ratio (Table 12-13)}} = \text{in. water} \qquad (12\text{-}158)$$

Use this corrected SP with the catalog rating tables of the manufacturer.

Scfm is standard cubic feet per minute corrected to standard density conditions. For example, to determine the scfm of 10,000 acfm at 600°F and 6 in. WG for air:

Static pressure: new scfm =

$$10{,}000\,\frac{\text{(density at 600°F and 6 in. WG)}}{0.075\ \text{(air)}} \tag{12-159}$$

$$= \text{acfm}\,\frac{\text{(density at nonstandard temperature and pressure)}}{0.075},$$

scfm (12-160)

This represents that by correcting the weight of air at 600°F and 6 in. WG. The new volume would be reduced to the calculated value.

Acfm means actual cubic feet per minute. It is the volume of gas flowing, and the value is not dependent on density.

How to Use Manufacturers' Capacity Tables[133]

1. Correct gas/air stream for other than standard density (0.075 lb/ft^3) because the manufacturer's fan performance tables are based on dry air at 70°F at sea level at a density of 0.075 lb/ft^3. For flow densities other than 0.075, corrections need to be made.
   a. Temperature correction only to 70°F (standard): From Table 12-14A, select factor (F_1) at actual temperature, select fan from manufacturer's rating tables at static pressure of (F_1) (required fan static pressure output) as corrected to 70°F.
   b. Altitude correction for elevations above sea level: From Table 12-14B, select factor (F_2) at the elevation of fan location, then (F_2) (required SP at operating temperature) = SP, in. WG at 70°F.
   c. Correction for density rarefaction (negative pressure, inlet side): For negative pressure less than 20 in. WG, the correction is considered negligible. Referring to Table 12-14C, the correction factors for negative static pressure, at SP in Table, select factor (F_3), the corrected static pressure (SP) = (F_3) (required fan static pressure output based on temperature corrected SP of Par. (a)).
   d. Brake horsepower (Note: most manufacturers' fan bhp tables include the effect of bearing drag.[133]): Referring to Table 12-14A, look to factors F_1, F_2, and F_3 when applicable, multiply them together to get the combined factor, F_4, and divide the fan tables bhp by F_4 to obtain the required bhp for actual conditions.

Table 12-14A
Correction Factors for Centrifugal Fan Operation

Chart IV SP and bhp Correction Factors for Temperature [°F]

Temp.	Factor
−50°	.77
−25°	.82
0°	.87
20°	.91
40°	.94
60°	.98
70°	1.00
80°	1.02
100°	1.06
120°	1.09
140°	1.13
160°	1.17
180°	1.21
200°	1.25
225°	1.29
250°	1.34
275°	1.39
300°	1.43
325°	1.48
350°	1.53
375°	1.58
400°	1.62
450°	1.72
500°	1.81
550°	1.91
600°	2.00
700°	2.19
800°	2.38
900°	2.56
1000°	2.76

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Select a fan from the tables at (1) required inlet cfm at inlet temperature and pressure, (2) at fan tables required operating rpm approved by manufacturers for fan size, correcting (reducing) for manufacturer's safe speed for temperature and materials of wheel construction. Generally, standard high-strength steel construction has a correction on speed of 1.0 up to 400°F and graduated down to 0.87 for 800°F, and only a 1.0 factor for aluminum at 70°F and 0.97 at 200°F.

   e. Handling gases other than standard air requires the determination of the actual gas density, lb/ft^3, including the amount of moisture or other material. It may be advisable to consult the manufacturer for proper selection of fan under these conditions.

Table 12-14B
Correction Factors for Centrifugal Fan Operation

Chart V SP and bhp Correction Factors for Altitude [feet above sea level]

Altitude	Factor
0	1.00
500	1.02
1,000	1.04
1,500	1.06
2,000	1.08
2,500	1.10
3,000	1.12
3,500	1.14
4,000	1.16
4,500	1.18
5,000	1.20
5,500	1.22
6,000	1.25
6,500	1.27
7,000	1.30
7,500	1.32
8,000	1.35
9,000	1.40
10,000	1.45

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Table 12-14C
Correction Factors for Centrifugal Fan Operation

Chart VI SP and bhp Correction Factors for Rarefication [negative inlet pressure]

SP	Factor
5 in.	1.01
10 in.	1.03
15 in.	1.04
20 in.	1.05
25 in.	1.07
30 in.	1.08
35 in.	1.09
40 in.	1.11

NOTE: When more than one correction is made, the factors are combined by multiplying them.

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Example 12-13. Fan Selection

This example is used by permission of Bul. 864, New York Blower Co., [133]

A fan is required for 9,600 cfm, 24 in. SP, 5,000 fpm outlet velocity at 300°F sea level, and 20 in. negative inlet pressure, handling clean air.

Table 12-14D
Correction Factors for Centrifugal Fan Operation

Chart 1 Maximum Safe Speeds of DH, LS, or RIM Wheels at 70°F

Size	Rpm
224	3,800
264	3,600
294	3,090
334	2,770
364	2,500
404	2,285
454	2,000
504	1,810
574	1,590
644	1,420
714	1,280
784	1,170
854	1,070

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1. Table 12-14A gives a 1.43 factor for 300°F, and Table 12-14C gives a 1.05 factor for rarefication. The combined factor is 1.5 [1.43 × 1.05]. Multiply 24 in. SP by 1.5 = 36 in. SP at standard density. Select from capacity tables for 9,600 cfm at 36 in. SP.
2. A size 334 Series 45 GI fan with a DH wheel is selected for 9,600 cfm, 36 in. SP, 5,000 fpm outlet velocity at 2,447 rpm and 79.8 bhp. (Bul. 864, New York Blower[133])
3. Catalog chart Table 12-14D indicates that the safe speed of the fan is 2,770 rpm and the catalog chart, Table 12-14, indicates a 1.0 factor for the standard steel wheel at 300°F. The fan is satisfactory to operate at 2,447 rpm.
4. The performance at operating conditions would be 9,600 cfm, 24 in. SP, 5,000 fpm outlet velocity at 2,447 rpm and 53.2 bhp [79.8 ÷ 1.5].

Pressures

Rating Pressure

Fans are rated independently of their systems. The net sizing rating for selection includes velocity pressure, vp, to account for acceleration. The fan rating static pressure, sp:[128]

Total net fan sp = sp outlet − sp inlet − vp inlet

Pressure is produced by the rotating fan blades as they change the air/gas velocity by imparting kinetic energy. These velocity changes are the results of tangential and radial velocity components for centrifugal fans and of axial and tangential velocity components for axial fans.[130]

Table 12-14E
Correction Factors for Centrifugal Fan Operation

Chart II Temperature Correction Factors for Maximum Safe Speeds of LS or RIM Wheels

Tem- pera- ture °F	Construction Materials: Mild Steel	Alumi- num	A-572-50/ 60*	304	316	347
70°F	1.0	1.0	1.0	1.0	.95	1.0
200°F	1.0	.97	1.0	.89	.92	1.0
300°F	1.0	—	1.0	.82	.88	.99
400°F	1.0	—	1.0	.78	.86	.97
500°F	.97	—	1.0	.75	.83	.97
600°F	.94	—	1.0	.73	.80	.97
700°F	.91	—	1.0	.71	.78	.96
800°F	.82	—	1.0	.70	.77	.96
900°F	—	—	—	.68	.76	.95
1,000°F	—	—	—	—	.75	.94

***nyb** option on material.

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Table 12-14F
Correction Factors for Centrifugal Fan Operation

Chart III Temperature Corrections Factors for Maximum Safe Speeds of DH Wheels

Temp. °F	Construction Materials: Standard High-Strength Steel	Aluminum
70°F	1.0	1.0
200°F	1.0	.97
300°F	1.0	—
400°F	1.0	—
500°F	.99	—
600°F	.95	—
700°F	.91	—
800°F	.87	—

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For centrifugal fans, the rotating impellers (wheels) develop the fan's pressure by (a) the centrifugal force created by rotating the volume of air/gas enclosed between the blades (see Figures 12-119D and 12-122) and (b) the kinetic energy imparted to the air by its velocity leaving the impeller/wheel.[130] From Reference 128,

$$\text{sp fan} = (\text{sp}_{\text{outlet}})\left(\frac{0.075}{\text{density}_{\text{outlet}}}\right) - (\text{sp}_{\text{inlet}})\left(\frac{0.075}{\text{density}_{\text{inlet}}}\right) - \text{vp}_{\text{inlet}}, \text{ in W.G.} \quad (12\text{-}161)$$

$$\text{vp}_{\text{inlet}} = \left(\frac{\text{cfm}_{\text{inlet}}}{(1{,}096)(\text{inlet area, ft}^2)}\right)(\text{density}_{\text{ inlet}}) \quad (12\text{-}162)$$

scfm = acfm (actual inlet density/standard density)

The sum of the static and velocity pressures is the total pressure. See Figure 12-129.

Total Pressure

The total pressure, p_t, is measured as the increase in total pressure given to a gas passing through a fan. It is a measure of the total energy increase per unit volume imparted to the flowing gas by the fan.

$$p_t = p_s + p_v \quad (12\text{-}163)$$

where p_t = total system pressure, usually expressed as in. of water
p_s = static pressure, in. of water
p_v = velocity pressure, in. of water
sp = static pressure, in. of water (or, in. water gage)
vp = velocity pressure, in. water
ρ = gas density, lb/ft³
g = acceleration due to gravity, 32.2 ft/sec-sec
v_m = gas velocity, ft/min

Velocity Pressure

Velocity pressure is a measure of the velocity pressure in the fan outlet. It is indicated by a differential reading of an impact tube facing the direction of air flow in the fan outlet and by a static reading normal to air flow in the fan outlet. It is a measure of the kinetic energy per unit volume of gas, existing at the fan outlet.[11] Also see Table 12-15.The portion of the pressure developed by a fan due to the air or gas velocity is termed *velocity pressure*.

$$p_v = \frac{\rho(v_m)^2}{1.203(10^6)}, \text{ in. of water} \quad (12\text{-}164)$$

$$\text{or, } v = 1{,}096.5\,(p_v/\rho)^{1/2}, \text{ ft/min}$$

For "standard air" of density, ρ = 0.075 lb/ft³

$$v = 4{,}004\,(\rho_v)^{1/2}, \text{ ft/min} = v_m \quad (12\text{-}165)$$

Static Pressure

The portion of the fan pressure necessary to overcome friction in the flowing system of gas/air is termed *static pressure*.[128]

The static pressure is the fan total pressure less the fan outlet velocity pressure. It is indicated, Figure 12-129, by the differential of an impact tube facing the direction of air flow in the fan inlet and by a static reading normal to the air flow in the fan outlet.[11]

Table 12-15
Velocity Pressures for Standard Air

Density = 0.075 lb/ft³	
Duct Velocity ft/min	**Velocity Pressure, in. water**
800	0.040
1,000	0.063
1,200	0.090
1,400	0.122
1,600	0.160
1,800	0.202
2,000	0.250
2,200	0.302
2,400	0.360
2,600	0.422
2,800	0.489
3,000	0.560
3,200	0.638
3,400	0.721
3,600	0.808
3,800	0.900
4,000	0.998
4,200	1.100
4,400	1.21
4,600	1.32
4,800	1.44
5,000	1.56
5,200	1.69
5,400	1.82
5,600	1.95
5,800	2.10
6,000	2.24

Example 12-14. Fan Selection Velocities

A fan conveying standard air at 3,000 acfm is to be selected for a system. (Using the duct loss calculations in Chapter 2, "Flow of Fluids" V. 1, 3rd Ed., of this series). Establish at assumed 4,400 ft/min (Table 12-15).

Fan outlet sp = 3.5 in. water
Fan inlet sp = −6.75 in. water
Velocity pressure, vp, inlet = 1.21 (Table 12-15) at 4,400 ft/min for standard air (or use Equation 12-164)

Net total sp = 3.5 − (−6.75) − 1.21 = 9.04 in. water

Therefore, a fan should be selected for 3,000 acfm at 9.04 in. water sp and have an outlet velocity of at least 4,400 ft/min (if suspended material is not present); otherwise, a lower velocity unit could be selected but should be checked with the manufacturers.

Operational Characteristics and Performance

Centrifugal Fan

The principal types of fans for industrial, chemical, and petrochemical applications are centrifugal and axial-flow. (See Figures 12-118B and 12-118D). For the wheel, blades are (a) forward curved (Figure 12-119D), in which the kinetic energy and the centrifugal force velocity of the rotating wheel/blades of these two velocities are additive and (b) backward curved, in which the velocities oppose each other, and the result is less than the forward wheel. For general service, the backward curved wheel blades are more efficient than the forward curved blades.[130]

The three types of centrifugal fan blades—radial, backward, and forward—give three characteristic performances. Table 12-12B gives a quick comparison.

Figures 12-130A, 12-130B, 12-131A, 12-131B, and 12-132 present the typical performance curves for radial, backward, and forward bladed fans, respectively. Exact performance for a given fan can be obtained only on test or by established design data for a given style and wheel configuration. The typical curves show what to expect for any particular blade design, because the blading sets the performance. Modifications, such as blade angle, affect the specific performance; however, these follow the characteristics of the blade type. Certain streamlined blade designs offer improved mechanical efficiencies over the regular designs. Exact or guaranteed performance can be identified only by a manufacturer for a particular unit.

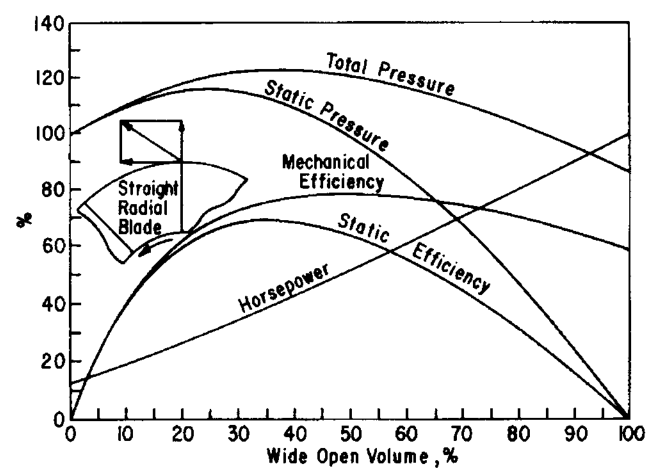


Figure 12-130A. Characteristic curves for straight radial blade. (Used by permission: The Howden Fan Company.)

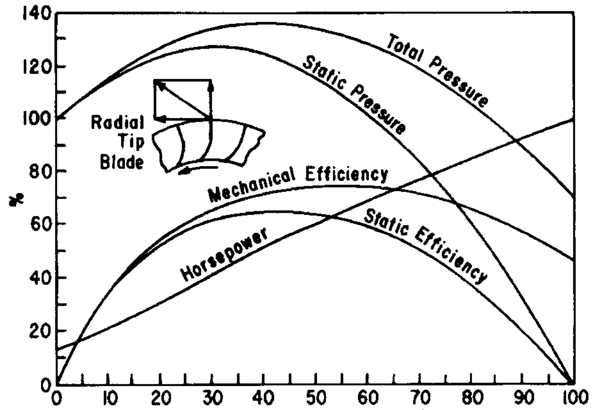


Figure 12-130B. Characteristic curves for radial tip blade. (Used by permission: The Howden Fan Company.)

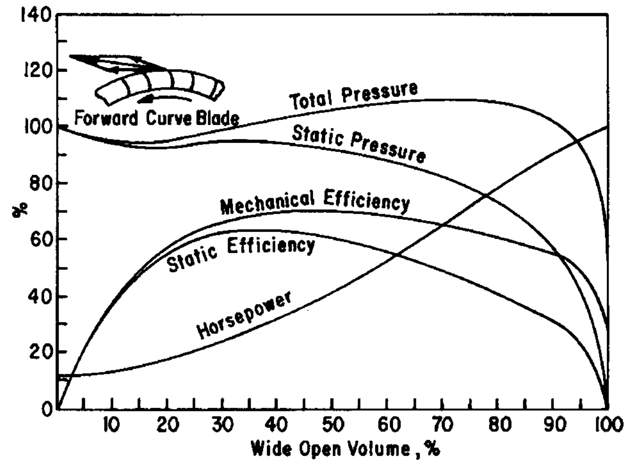


Figure 12-132. Characteristic curves for forward curved blade. (Used by permission: The Howden Fan Company.)

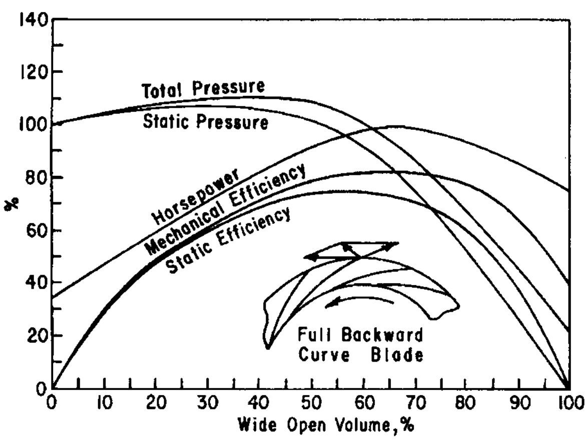


Figure 12-131A. Characteristic curves for backward curved blade. (Used by permission: The Howden Fan Company.)

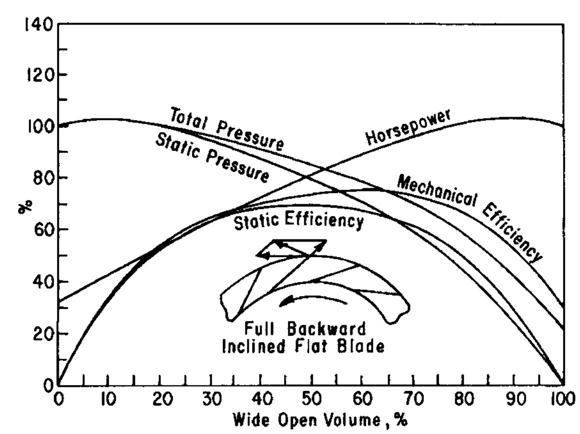


Figure 12-131B. Characteristic curves for full backward inclined flat blade. (Used by permission: The Howden Fan Company.)

Radial Blade

This type of blade is not close fitting in the housing, as it is usually used for handling suspended materials; abrasive dust collecting; and exhausting of fumes from dirty, greasy, or acid environments. It is easily covered or coated with protective material to withstand a particular condition. It has an overloading horsepower characteristic curve. Efficiency of this blade depends quite a bit upon the manufacturer and his field of application for the fan. In any case, it is not the highest efficiency blade. A fair running static efficiency is 50–70% for both the straight radial blade and radial tip blade.

The rather sharply rising static pressure curve of the radial blade centrifugal allows for small changes in volume as the resistance of the system changes considerably. The rising smooth horsepower characteristic gives efficient performance at dampered or higher resistance conditions. This is well suited for induced draft conditions using outlet dampers for volume control and also gives good performance with fixed or variable inlet vanes.

Backward Blade

This type of blade is well suited to streamline flow conditions and is used extensively on ventilating, air conditioning, and clean and dirty process gas streams. The backward blade does not catch dirt easily. The outstanding and important characteristic is the nonoverloading horsepower. This is the only commonly used blade style with this feature. It is important in process control and eliminates the need for oversized motors or other drivers. Speed of operation is high, which allows direct or belt connection to the driver. Certain streamlined blade designs provide the same basic characteristics with more efficient and quieter operation. The usual operating static efficiency range for the regular blade is 65–80% and 80–92% for the streamlined design. This type of blade is preferred on double-width, double-inlet fan installations due to the capability of the blade to not overload for nonuniform flow.

Forward Blade

This type of blade is usually shallow, operates at slow speed for a given capacity, and usually has low outlet velocity. It is the quietest running of the three types and is relatively small for its capacity. Its operating characteristics are poor for many applications, because the horsepower rises sharply with a decrease in static pressure after the peak pressure for the fan has been reached. This requires a careful examination of the system to avoid overloading the driver. Due to the shape of the blade and its direction of rotation, this type should not be used on dirty service. Material tends to lodge on the blades and can cause serious loss in capacity as well as imbalance. This blade is used in fixed systems, such as package arrangements in which the conditions are not only known, but set. The blade is not well suited to protective coatings due to the multiplicity of blades and their deep cup. The operating static efficiency range is 55–75%.

Axial Flow Fan

The performance of the axial flow fan is represented in Figures 12-133A–B.

For these fans, pressure is produced by the change in velocity passing through the impeller wheel. No pressure is produced by centrifugal force. (See Figures 12-118C and 12-119D and Tables 12-11, 12-12A, and 12-12B).

The horsepower characteristic is nonoverloading and is important in many applications. Because so many blade types, from the axial to the propeller type, exist, it is difficult to present a typical analysis of these fans. Note also that usually the peak horsepower is reached in the blocked-tight or no-flow condition. Figures 12-133A and 12-133B indicate the relative noise level compared to the average centrifugal, indicating an axial as less desirable from this point at low flow conditions. The usual pressure range of the application is 0–3 in. water static pressure.[46]

The vane axial and tube axial can be selected for higher outlet velocities than the centrifugal, approximately 2,000–4,000 ft/min through the fan casing as compared to 2,000–3,500 (a few to 6,600) ft/min outlet velocity for the centrifugal. The axial fans should be connected to ducts by tapered cone connections.

The use of inlet guide vanes tends to flatten the horsepower curve for axials.

The peak efficiency range of the tube axial is 30–50% and 40–65% for the vane axial. Under some conditions, this may be 5% higher than a corresponding centrifugal unit.

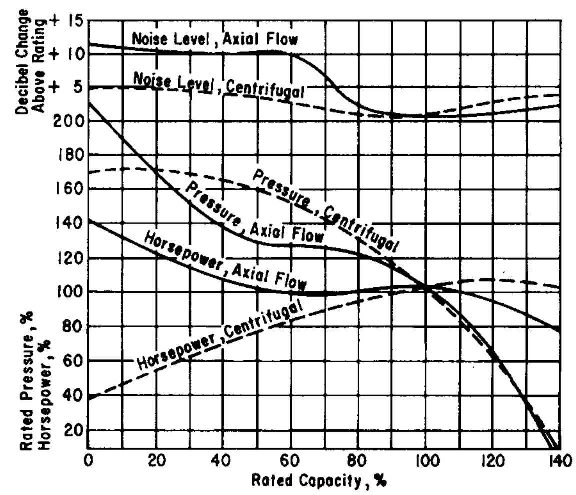


Figure 12-133A. Performance curve for axial-type fans. (Used by permission: The Howden Fan Company.)

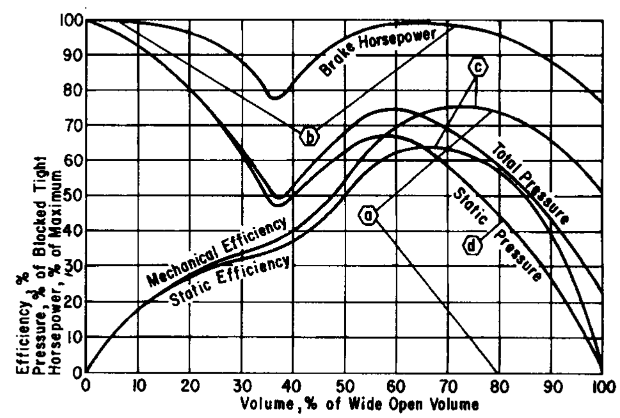


Figure 12-133B. Performance curves for streamlined designs of axial-type fans. (Used by permission: The Howden Fan Company.)

More care must be given to selection of the axials because the best operating range for efficiency and noise is relatively narrow.

Propeller Fans

See Figure 12-125.

These fans usually operate with no piping or duct work on either side and move air or gas from one large open area to another. Pressures are usually very low, and volumes depend on size, blade pitch, the number of blades, and speed.

Static efficiencies run from 10–50% depending on the fan and its installation. With a well-designed inlet ring and discharge diffuser, the efficiencies may be 50–60%.

Fans are tested by the manufacturers to the applicable industry codes of ASHRAE 51 and AMCA 214—latest editions. Tests are conducted between shut-off (no flow at blanked off discharge) and free delivery (no flow resistance on discharge).[130]

Fan Control

Methods of fan volume control follow.

Variable Speed Drive. This can be accomplished by turbines, direct current motors, variable-speed motors, or slip-ring motors. With the changing speed of the driver, the fan output capacity and pressure can be varied. For capacity reductions below 50%, an outlet damper is usually added to the system.

Outlet Damper with Constant Fan Speed. The system resistance is varied with this damper. The volume of gas delivered from the fan is changed as a function of the movement of the damper. It is low in first cost and simple to operate but does require more horsepower than other methods of control.

Variable Inlet Vane with Constant Fan Speed. Horsepower required is less at reduced loads than with the outlet damper. The angle and/or extent of closure of the inlet vanes controls the volume of gas admitted to the inlet of the wheel. This popular control is simple and can be operated manually or automatically. Figure 12-134 shows the performance curve of a typical situation using the variable inlet vanes that control or limit the inlet volume and spin the air at the fan inlet. The amount of volume change and spin can be varied by changing the vane positions. The pressure from the fan and its horsepower requirement (within certain limitations) depend on this spin. The vane positions are shown as A, B, and C. At 74% rated volume capacity, the hp for position C is 64% of the rated value, and the sp is 58% of rating. Note that if an outlet damper were used, hp would have been 88% of rating, Figures 12-135A and 12-135B, assuming the curves represented the same fan and system. Inlet vane control is more expensive than outlet dampers, but this can be justified usually by lower power costs, especially on large horsepower installations.

Fluid Drive

This method allows fan speed to be adjusted 20–100% with corresponding volume changes. A constant-speed motor is used, which simplifies the electrical requirements. The motor can be started disconnected from the fan. This scheme probably saves more power than the others, although its initial cost can be quite high for specialized installations, Figure 12-135B. Note that curve F of Figure 12-135B is the actual power input to the fan shaft. The hydraulic or fluid drive has about 3% in losses, so its power input at 100% load is actually about 103% to allow for this.

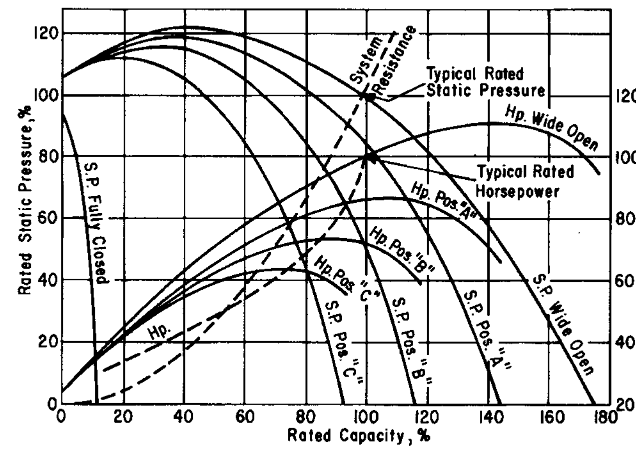


Figure 12-134. Typical variable inlet vane performance curves. (Used by permission: Elliott® Company, Jeannette, PA.)

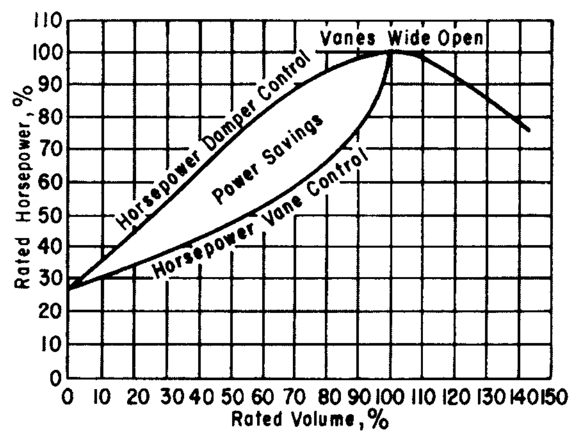


Figure 12-135A. Power savings comparison between inlet vane control and outlet dampers. (Used by permission: The Howden Fan Company.)

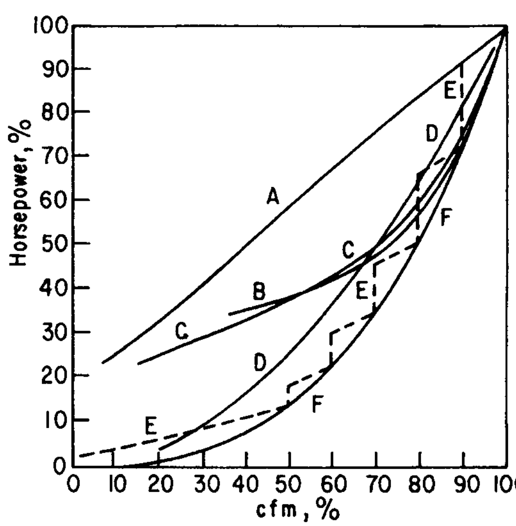


Figure 12-135B. Comparison of efficiencies of five principal methods of controlling fan output. (Used by permission: The Howden Fan Company.)

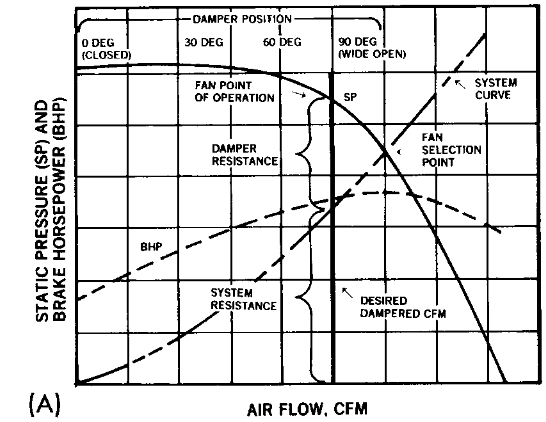
Curves B and C are for variable vane inlet dampening, and Curve A is for outlet dampening of a backward blade fan. Curve E shows an outlet damper with a multiple step speed slip-ring motor. This has an outlet damper for final control from 89–100%. From this graph, a reasonably accurate selection can be made of the control features to consider for most installation conditions.

Control Methods

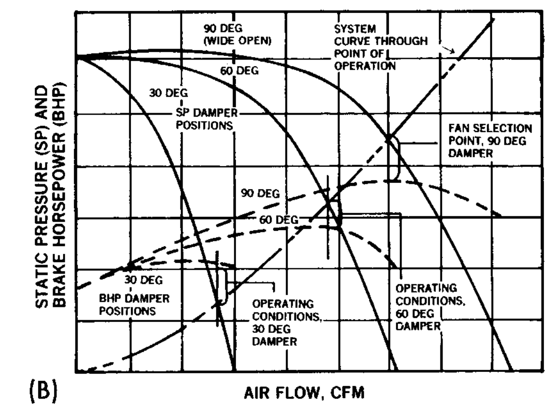
Outlet dampers, Figure 12-127C, control the air flow just after it has passed through the fan, but all may not necessarily actually enter the outlet duct system; thus, the resistance of the damper causes the air/gas to circulate within the fan wheel/blades and limits the quantity that can actually leave the fan casing or outlet flange. Figures 12-136A and 12-136B illustrate several settings (full, open, closed) of the damper for a backward inclined centrifugal fan. Also see Figure 12-137. The damper control affects cfm, sp, and bhp. From full open position at the intersection of the system resistance curve with the fan sp curve at "Fan Selection Point," as the damper is closed, it can reach the 90 (wide open) mark, and the fan's performance moves left up the sp curve to the "Fan Operation Point." On the bhp curve, the operation moves to the left (lower cfm than "Selection Point"), and the bhp is read at the dark line up from the reduced cfm at the intersection with the bhp curve.

The volume and pressure control methods with electric motor drive for centrifugal fans are:[135]

A. Constant Speed
  1. Dampers, louver type (simplest)
     a. Unidirectional louvers
     b. Reverse-directional louvers
  2. Inlet vanes



- Static pressure and brake horsepower curves for backwardly inclined fan with outlet damper. As the damper closes, the point of operation - brake horsepower and static pressure - moves to the left of the original fan selection point to the 90-degrees (wide open) damper setting.



- Effect of applying inlet dampers to the fan in Figure 1. Separate SP and BHP curves are developed for each vane setting. Fan operating points at these settings are determined by system resistance (points where system curve intersects SP and BHP fan curves).

Figure 12-136A, B. Effects of fan dampers on air flow, static pressure, and brake horsepower. (A) Backwardly inclined fan with outlet damper. (B) Inlet damper applied to condition of (A). (Used by permission: "Engineering Letter No.11," ©1996. The New York Blower Co.® For more information, contact the company at www.nyb.com.)

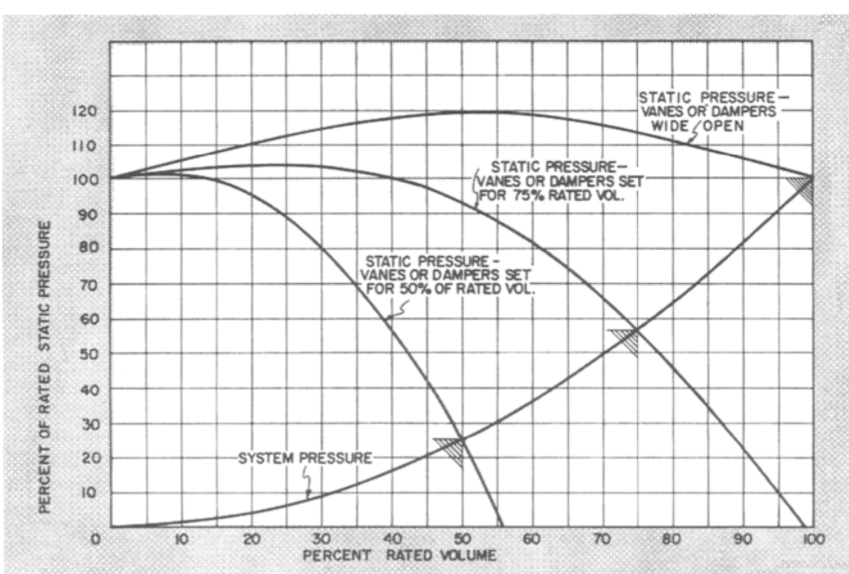


Figure 12-137. Effect of partially closed outlet vanes or dampers on fan pressure-volume curve. (Used by permission: Tobin, J. F. *Power Engineering*, p. 82, Sept. 1954. ©PennWell Publishing Company/*Power Engineering Magazine.* All rights reserved.)

B. Variable speed
   1. Variable-speed motors
   2. Fluid drives
   3. Magnetic couplings

Constant speed fans have the lowest first cost, quickest response, least efficiency, greater noise, and hardest start. Variable-speed fans have more efficiency, less noise, an easier start, greater first cost, and slower response.

Inlet dampers reduce the gas/air entering the fan. Figure 12-136 illustrates inlet dampers on a backward inclined centrifugal fan. As the inlet vane angle on the damper is changed (for example, wide open, 90°, 60°, or 30°), new sp and bhp curves are generated. The new point of operation is defined by the system. Starting with the initial "Fan Selection Point, 90° Damper," the change to the 60° damper setting establishes a new sp curve and a new bhp curve.

Figure 12-137 illustrates the separate effect of a partially closed outlet damper or vane on a pressure-volume curve.

Haines[136] presents a helpful discussion on this subject, emphasizing the controls of the fan itself plus the impact of an air flow duct system terminal zone control from the distribution duct. The VAV shown on Figure 12-138 represents the pressure drop through the zone control device. Referring to the figures, note the sudden drop in system pressure from the fan static pressure-volume operating curve to the system resistance curve, indicating that the zone control pressure drop is not the same as the system resistance static pressure loss, as would be treated for other duct, valves, etc. The figures represent discharge damper, inlet vane damper, and speed control. The discharge damper is the simplest control. The preferred dampering method is by use of the inlet vane dampers at the fan, resulting in a reduction in fan operating pressure and a savings of fan horsepower.[136] The best method to save horsepower is by speed control of the fan using an eddy current clutch or motor speed controller. Each change in fan speed produces a new fan curve parallel to the other fan curve. Note that system and fan sp do not vary in strict accordance with fan laws, because the pressure drop through the zone terminal control device must increase to decrease the cfm to its zone. The resulting bhp reduction, as cfm is reduced, is not as great as theory would predict. Note that this applies for this air conditioning zone control.

Fan Laws

The performance of a fan is usually obtained from a manufacturer's specific curve or performance tables. Expected performance for a change from one condition of operation to another, or from one fan size to another, is given in Table 12-16 for geometrically similar fans.

Fan laws apply to blowers, exhausters, centrifugal, and axial flow fans. The relations are satisfactory for engineering calculations as long as the pressure rise is not great (not over 1 psi). Theoretically, a 4-in. H_2O pressure rise affects air density to cause a 1% deviation.[38] Where greater accuracy is required, the familiar adiabatic horsepower relations are used.

These laws are applicable *only* for geometrically similar fans and to the *same point of rating* on the performance curve. Fans in the same homologous series have the same geometrical shape, and their curves also are similar in shape. Fan performance is expressed as the rating for one condition of operation. This includes fan size, speed, capacity, pressure, and horsepower.[38] When two similar fans are compared at the same point of rating, they have the same efficiency, even though they are of different size and operate at different speeds. Therefore, these laws allow the examination of performance for changing the operating conditions on a single fan or the prediction of performance for the change from operations using one fan to those using a different size but of the same type and relative geometrical arrangement. It is *not* possible to predict backward blade fan performance from the operation of a forward blade unit. Laws 1 through 10 are expressed on a volume basis, and Laws 11, 12, and 13 are the three principal laws restated on a weight basis.[38]

Examination of Table 12-16 shows that Fan Law 1, for example, can be written for geometrically similar fans:

$$\frac{V_1}{V_2} = \left(\frac{D_1}{D_2}\right)^3 \left(\frac{rpm_1}{rpm_2}\right) \qquad (12\text{-}166)$$

where D = fan wheel diameter or its equivalent, in.
V = volume, cfm

The subscripts 1 and 2 represent conditions 1 and 2, respectively, or initial and final.

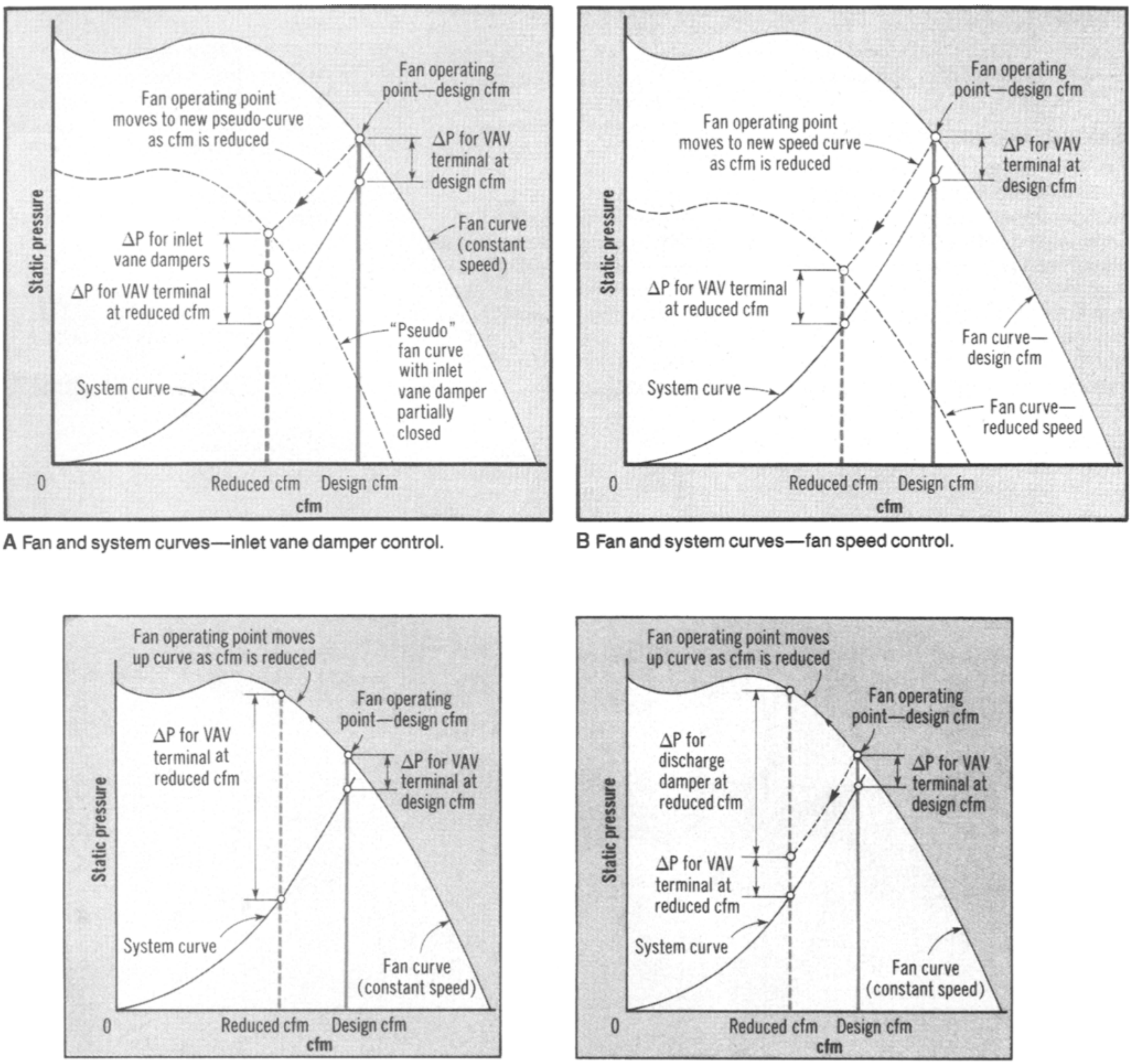
$$\frac{P_1}{P_2} = \left(\frac{D_1}{D_2}\right)^2 \left(\frac{rpm_1}{rpm_2}\right)^2 \left(\frac{\delta_1}{\delta_2}\right) \qquad (12\text{-}167)$$

Note $\left(\frac{\delta_1}{\delta_2}\right)$ may be replaced by $\left(\frac{\rho_1}{\rho_2}\right)$

p = static or total pressure, in. of water. When static pressure changes, the total pressure changes proportionately; also = p_s.
δ = relative density of gas referred to air at standard conditions. It may also be used as ratio of absolute density of gas to absolute density of air.
hp = horsepower input to fan, (or $(hp)_s$, or $(hp)_t$)

Compressibility K_p can often be ignored at usual low fan static pressures.

$$\frac{(hp)_1}{(hp)_2} = \left(\frac{D_1}{D_2}\right)^5 \left(\frac{n_1}{n_2}\right)^3 \left(\frac{\delta_1}{\delta_2}\right) \qquad (12\text{-}168)$$



A Fan and system curves—inlet vane damper control.

B Fan and system curves—fan speed control.

C Fan and system curves—no fan volume control.

D Fan and system curves—discharge damper control.

Figure 12-138A, B, C, D. Centrifugal fan system control methods. Note the effect of air distribution zone control (Δ for VAV) terminal on sudden jump in system resistance to fan operating static pressure curve. (Used by permission: Haines, R. W. *Heating/Piping/Air Conditioning*, p. 107, Aug. 1983. ©Penton Media, Inc. All rights reserved.)

Fan Law 7 c:

$$\left(\frac{hp}{hp}\right)_2^1 = \left(\frac{p_1}{p_2}\right)^{5/2}\left(\frac{n_1}{n_2}\right)^2\left(\frac{\delta_1}{\delta_2}\right)^{3/2} \qquad (12\text{-}169)$$

It is important to be able to properly interpret these laws; referring to the first law, (a), (b), and (c) show that for constant fan size (wheel diameter as the best representation), the capacity varies as the speed (rpm); pressure varies as the square of the speed; and the horsepower required varies as the cube of the speed. Now, looking at a constant speed interpretation of this same law, the capacity varies as the cube of the size; pressure varies as the square of the size; and horsepower varies as the fifth power of size—all for constant gas density. If the gas density varies too, then its ratio of change must be considered as expressed by the ratios shown in Table 12-16.

Table 12-16
Fan Laws

No.	Dependent Variables	Independent Variables							
1a	$Q_a = Q_b$	$\times$	$\left(\frac{D_a}{D_b}\right)^3$	$\times$	$\left(\frac{N_a}{N_b}\right)^1$	$\times$	(1)	$\times$	$\left(\frac{K_{pa}}{K_{pb}}\right)^{-1}$
1b	$p_{FTa} = p_{FTb}$	$\times$	$\left(\frac{D_a}{D_b}\right)^2$	$\times$	$\left(\frac{N_a}{N_b}\right)^2$	$\times$	$\left(\frac{\rho_a}{\rho_b}\right)^1$	$\times$	$\left(\frac{K_{pa}}{K_{pb}}\right)^{-1}$
1c	$P_{ia} = P_{ib}$	$\times$	$\left(\frac{D_a}{D_b}\right)^5$	$\times$	$\left(\frac{N_a}{N_b}\right)^3$	$\times$	$\left(\frac{\rho_a}{\rho_b}\right)^1$	$\times$	$\left(\frac{K_{pa}}{K_{pb}}\right)^{-1}$
1d	$p_{FVa} = p_{FVb}$	$\times$	$\left(\frac{D_a}{D_b}\right)^2$	$\times$	$\left(\frac{N_a}{N_b}\right)^2$	$\times$	$\left(\frac{\rho_a}{\rho_b}\right)^1$	$\times$	(1)
1e	$L_{Wa} = L_{Wb}$	$+ 70 \log$	$\left(\frac{D_a}{D_b}\right)$	$+ 50 \log$	$\left(\frac{N_a}{N_b}\right)$	$+ 20 \log$	$\left(\frac{\rho_a}{\rho_b}\right)$		
2a	$Q_a = Q_b$	$\times$	$\left(\frac{D_a}{D_b}\right)^2$	$\times$	$\left(\frac{p_{FTa}}{p_{FTb}}\right)^{1/2}$	$\times$	$\left(\frac{\rho_a}{\rho_b}\right)^{-1/2}$	$\times$	$\left(\frac{K_{pa}}{K_{pb}}\right)^{-1/2}$
2b	$N_a = N_b$	$\times$	$\left(\frac{D_a}{D_b}\right)^{-1}$	$\times$	$\left(\frac{p_{FTa}}{p_{FTb}}\right)^{1/2}$	$\times$	$\left(\frac{\rho_a}{\rho_b}\right)^{1/2}$	$\times$	$\left(\frac{K_{pa}}{K_{pb}}\right)^{1/2}$
2c	$P_{ia} = P_{ib}$	$\times$	$\left(\frac{D_a}{D_b}\right)^2$	$\times$	$\left(\frac{p_{FTa}}{p_{FTb}}\right)^{3/2}$	$\times$	$\left(\frac{\rho_a}{\rho_b}\right)^{-1/2}$	$\times$	$\left(\frac{K_{pa}}{K_{pb}}\right)^{1/2}$
2d	$p_{FVa} = p_{FVb}$	$\times$	(1)	$\times$	$\left(\frac{p_{FTa}}{p_{FTb}}\right)^1$	$\times$	(1)	$\times$	$\left(\frac{K_{pa}}{K_{pb}}\right)^1$
2e	$L_{Wa} = L_{Wb}$	$+ 20 \log$	$\left(\frac{D_a}{D_b}\right)$	$+ 25 \log$	$\left(\frac{p_{FTa}}{p_{FTb}}\right)$	$- 5 \log$	$\left(\frac{\rho_a}{\rho_b}\right)$		
3a	$N_a = N_b$	$\times$	$\left(\frac{D_a}{D_b}\right)^{-3}$	$\times$	$\left(\frac{Q_a}{Q_b}\right)^1$	$\times$	(1)	$\times$	$\left(\frac{K_{pa}}{K_{pb}}\right)^1$
3b	$p_{FTa} = p_{FTb}$	$\times$	$\left(\frac{D_a}{D_b}\right)^{-4}$	$\times$	$\left(\frac{Q_a}{Q_b}\right)^2$	$\times$	$\left(\frac{\rho_a}{\rho_b}\right)$	$\times$	$\left(\frac{K_{pa}}{K_{pb}}\right)^1$
3c	$P_{ia} = P_{ib}$	$\times$	$\left(\frac{D_a}{D_b}\right)^{-4}$	$\times$	$\left(\frac{Q_a}{Q_b}\right)^3$	$\times$	$\left(\frac{\rho_a}{\rho_b}\right)$	$\times$	$\left(\frac{K_{pa}}{K_{pb}}\right)^2$
3d	$p_{FVa} = p_{FVb}$	$\times$	$\left(\frac{D_a}{D_b}\right)^{-4}$	$\times$	$\left(\frac{Q_a}{Q_b}\right)^2$	$\times$	$\left(\frac{\rho_a}{\rho_b}\right)$	$\times$	$\left(\frac{K_{pa}}{K_{pb}}\right)^2$
3e	$L_{Wa} = L_{Wb}$	$- 80 \log$	$\left(\frac{D_a}{D_b}\right)$	$+ 50 \log$	$\left(\frac{Q_a}{Q_b}\right)$	$+ 20 \log$	$\left(\frac{\rho_a}{\rho_b}\right)$		

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Table 12-16
Fan Laws (Continued)

No.	Dependent Variables		Independent Variables						
4a	$Q_a = Q_b$	$\times$	$\left(\frac{D_a}{D_b}\right)^{4/3}$	$\times$	$\left(\frac{P_{ia}}{P_{ib}}\right)^{1/3}$	$\times$	$\left(\frac{\rho_a}{\rho_b}\right)^{-1/3}$	$\times$	$\left(\frac{K_{pa}}{K_{pb}}\right)^{-2/3}$
4b	$p_{FTa} = p_{FTb}$	$\times$	$\left(\frac{D_a}{D_b}\right)^{-4/3}$	$\times$	$\left(\frac{P_{ia}}{P_{ib}}\right)^{2/3}$	$\times$	$\left(\frac{\rho_a}{\rho_b}\right)^{1/3}$	$\times$	$\left(\frac{K_{pa}}{K_{pb}}\right)^{-1/3}$
4c	$N_a = N_b$	$\times$	$\left(\frac{D_a}{D_b}\right)^{-5/3}$	$\times$	$\left(\frac{P_{ia}}{P_{ib}}\right)^{1/3}$	$\times$	$\left(\frac{\rho_a}{\rho_b}\right)^{-1/3}$	$\times$	$\left(\frac{K_{pa}}{K_{pb}}\right)^{1/3}$
4d	$p_{FVa} = p_{FVb}$	$\times$	$\left(\frac{D_a}{D_b}\right)^{-4/3}$	$\times$	$\left(\frac{P_{ia}}{P_{ib}}\right)^{2/3}$	$\times$	$\left(\frac{\rho_a}{\rho_b}\right)^{1/3}$	$\times$	$\left(\frac{K_{pa}}{K_{pb}}\right)^{2/3}$
4e	$L_{Wa} = L_{Wb}$	$-$ 13.3 log	$\left(\frac{D_a}{D_b}\right)$	$+$ 16.6 log	$\left(\frac{P_{ia}}{P_{ib}}\right)$	$+$ 3.3 log	$\left(\frac{\rho_a}{\rho_b}\right)$		
5a	$D_a = D_b$	$\times$	$\left(\frac{Q_a}{Q_b}\right)^{1/2}$	$\times$	$\left(\frac{p_{FTa}}{p_{FTb}}\right)^{-1/4}$	$\times$	$\left(\frac{\rho_a}{\rho_b}\right)^{1/4}$	$\times$	$\left(\frac{K_{pa}}{K_{pb}}\right)^{1/4}$
5b	$N_a = N_b$	$\times$	$\left(\frac{Q_a}{Q_b}\right)^{-1/2}$	$\times$	$\left(\frac{p_{FTa}}{p_{FTb}}\right)^{3/4}$	$\times$	$\left(\frac{\rho_a}{\rho_b}\right)^{3/4}$	$\times$	$\left(\frac{K_{pa}}{K_{pb}}\right)^{1/4}$
5c	$P_{ia} = P_{ib}$	$\times$	$\left(\frac{Q_a}{Q_b}\right)^{1}$	$\times$	$\left(\frac{p_{FTa}}{p_{FTb}}\right)^{1}$	$\times$	$\left(1\right)$	$\times$	$\left(\frac{K_{pa}}{K_{pb}}\right)^{1}$
5d	$p_{FVa} = p_{FVb}$	$\times$	$\left(1\right)$	$\times$	$\left(\frac{p_{FTa}}{p_{FTb}}\right)^{1}$	$\times$	$\left(1\right)$	$\times$	$\left(\frac{K_{pa}}{K_{pb}}\right)^{1}$
5e	$L_{Wa} = L_{Wb}$	$+$ 10 log	$\left(\frac{Q_a}{Q_b}\right)$	$+$ 20 log	$\left(\frac{p_{FTa}}{p_{FTb}}\right)$	$+$ 0 log	$\left(\frac{\rho_a}{\rho_b}\right)$		
6a	$D_a = D_b$	$\times$	$\left(\frac{Q_a}{Q_b}\right)^{1/3}$	$\times$	$\left(\frac{N_a}{N_b}\right)^{-1/3}$	$\times$	$\left(1\right)$	$\times$	$\left(\frac{K_{pa}}{K_{pb}}\right)^{1/3}$
6b	$p_{FTa} = p_{FTb}$	$\times$	$\left(\frac{Q_a}{Q_b}\right)^{2/3}$	$\times$	$\left(\frac{N_a}{N_b}\right)^{4/3}$	$\times$	$\left(\frac{\rho_a}{\rho_b}\right)^{1}$	$\times$	$\left(\frac{K_{pa}}{K_{pb}}\right)^{-1/3}$
6c	$P_{ia} = P_{ib}$	$\times$	$\left(\frac{Q_a}{Q_b}\right)^{5/3}$	$\times$	$\left(\frac{N_a}{N_b}\right)^{4/3}$	$\times$	$\left(\frac{\rho_a}{\rho_b}\right)^{1}$	$\times$	$\left(\frac{K_{pa}}{K_{pb}}\right)^{2/3}$
6d	$p_{FVa} = p_{FVb}$	$\times$	$\left(\frac{Q_a}{Q_b}\right)^{2/3}$	$\times$	$\left(\frac{N_a}{N_b}\right)^{4/3}$	$\times$	$\left(\frac{\rho_a}{\rho_b}\right)^{1}$	$\times$	$\left(\frac{K_{pa}}{K_{pb}}\right)^{2/3}$
6e	$L_{Wa} = L_{Wb}$	$+$ 23.3 log	$\left(\frac{Q_a}{Q_b}\right)$	$+$ 26.6 log	$\left(\frac{N_a}{N_b}\right)$	$+$ 20 log	$\left(\frac{\rho_a}{\rho_b}\right)$		

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Table 12-16
Fan Laws (Continued)

No.	Dependent Variables		Independent Variables						
7a	$D_a = D_b$	$\times$	$\left(\frac{p_{FTa}}{p_{FTb}}\right)^{1/2}$	$\times$	$\left(\frac{N_a}{N_b}\right)^{-1}$	$\times$	$\left(\frac{\rho_a}{\rho_b}\right)^{-1/2}$	$\times$	$\left(\frac{K_{pa}}{K_{pb}}\right)^{1/2}$
7b	$Q_a = Q_b$	$\times$	$\left(\frac{p_{FTa}}{p_{FTb}}\right)^{3/2}$	$\times$	$\left(\frac{N_a}{N_b}\right)^{-2}$	$\times$	$\left(\frac{\rho_a}{\rho_b}\right)^{-3/2}$	$\times$	$\left(\frac{K_{pa}}{K_{pb}}\right)^{1/2}$
7c	$P_{ia} = P_{ib}$	$\times$	$\left(\frac{p_{FTa}}{p_{FTb}}\right)^{5/2}$	$\times$	$\left(\frac{N_a}{N_b}\right)^{-2}$	$\times$	$\left(\frac{\rho_a}{\rho_b}\right)^{-3/2}$	$\times$	$\left(\frac{K_{pa}}{K_{pb}}\right)^{3/2}$
7d	$p_{FVa} = p_{FVb}$	$\times$	$\left(\frac{p_{FTa}}{p_{FTb}}\right)^{1}$	$\times$	$\left(1\right)$	$\times$	$\left(1\right)$	$\times$	$\left(\frac{K_{pa}}{K_{pb}}\right)^{1}$
7e	$L_{Wa} = L_{Wb}$	$+ 35 \log$	$\left(\frac{p_{FTa}}{p_{FTb}}\right)$	$- 20 \log$	$\left(\frac{N_a}{N_b}\right)$	$- 15 \log$	$\left(\frac{\rho_a}{\rho_b}\right)$		
8a	$D_a = D_b$	$\times$	$\left(\frac{P_{ia}}{P_{ib}}\right)^{-1/4}$	$\times$	$\left(\frac{Q_a}{Q_b}\right)^{3/4}$	$\times$	$\left(\frac{\rho_a}{\rho_b}\right)^{1/4}$	$\times$	$\left(\frac{K_{pa}}{K_{pb}}\right)^{1/2}$
8b	$N_a = N_b$	$\times$	$\left(\frac{P_{ia}}{P_{ib}}\right)^{3/4}$	$\times$	$\left(\frac{Q_a}{Q_b}\right)^{-5/4}$	$\times$	$\left(\frac{\rho_a}{\rho_b}\right)^{-1/4}$	$\times$	$\left(\frac{K_{pa}}{K_{pb}}\right)^{-1/2}$
8c	$p_{FTa} = p_{FTb}$	$\times$	$\left(\frac{P_{ia}}{P_{ib}}\right)^{1}$	$\times$	$\left(\frac{Q_a}{Q_b}\right)^{-1}$	$\times$	$\left(1\right)$	$\times$	$\left(\frac{K_{pa}}{K_{pb}}\right)^{-1}$
8d	$p_{FVa} = p_{FVb}$	$\times$	$\left(\frac{P_{ia}}{P_{ib}}\right)^{1}$	$\times$	$\left(\frac{Q_a}{Q_b}\right)^{-1}$	$\times$	$\left(1\right)$	$\times$	$\left(1\right)$
8e	$L_{Wa} = L_{Wb}$	$+ 20 \log$	$\left(\frac{P_{ia}}{P_{ib}}\right)$	$- 10 \log$	$\left(\frac{Q_a}{Q_b}\right)$	$+ 0 \log$	$\left(\frac{\rho_a}{\rho_b}\right)$		
9a	$D_a = D_b$	$\times$	$\left(\frac{P_{ia}}{P_{ib}}\right)^{1/2}$	$\times$	$\left(\frac{p_{FTa}}{p_{FTb}}\right)^{-3/4}$	$\times$	$\left(\frac{\rho_a}{\rho_b}\right)^{1/4}$	$\times$	$\left(\frac{K_{pa}}{K_{pb}}\right)^{-1/4}$
9b	$N_a = N_b$	$\times$	$\left(\frac{P_{ia}}{P_{ib}}\right)^{-1/2}$	$\times$	$\left(\frac{p_{FTa}}{p_{FTb}}\right)^{5/4}$	$\times$	$\left(\frac{\rho_a}{\rho_b}\right)^{-1/4}$	$\times$	$\left(\frac{K_{pa}}{K_{pb}}\right)^{3/4}$
9c	$Q_a = Q_b$	$\times$	$\left(\frac{P_{ia}}{P_{ib}}\right)^{1}$	$\times$	$\left(\frac{p_{FTa}}{p_{FTb}}\right)^{-1}$	$\times$	$\left(1\right)$	$\times$	$\left(\frac{K_{pa}}{K_{pb}}\right)^{-1}$
9d	$p_{FVa} = p_{FVb}$	$\times$	$\left(1\right)$	$\times$	$\left(\frac{p_{FTa}}{p_{FTb}}\right)^{1}$	$\times$	$\left(1\right)$	$\times$	$\left(\frac{K_{pa}}{K_{pb}}\right)^{1}$
9e	$L_{Wa} = L_{Wb}$	$+ 10 \log$	$\left(\frac{P_{ia}}{P_{ib}}\right)$	$+ 10 \log$	$\left(\frac{p_{FTa}}{p_{FTb}}\right)$	$+ 0 \log$	$\left(\frac{\rho_a}{\rho_b}\right)$		

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Table 12-16
Fan Laws (Continued)

No.	Dependent Variables		Independent Variables						
10a	$D_a = D_b$	×	$\left(\frac{P_{ia}}{P_{ib}}\right)^{1/5}$	×	$\left(\frac{N_a}{N_b}\right)^{-3/5}$	×	$\left(\frac{\rho_a}{\rho_b}\right)^{-1/5}$	×	$\left(\frac{K_{pa}}{K_{pb}}\right)^{1/5}$
10b	$Q_a = Q_b$	×	$\left(\frac{P_{ia}}{P_{ib}}\right)^{3/5}$	×	$\left(\frac{N_a}{N_b}\right)^{-4/5}$	×	$\left(\frac{\rho_a}{\rho_b}\right)^{-3/5}$	×	$\left(\frac{K_{pa}}{K_{pb}}\right)^{-2/5}$
10c	$p_{FTa} = p_{FTb}$	×	$\left(\frac{P_{ia}}{P_{ib}}\right)^{2/5}$	×	$\left(\frac{N_a}{N_b}\right)^{4/5}$	×	$\left(\frac{\rho_a}{\rho_b}\right)^{3/5}$	×	$\left(\frac{K_{pa}}{K_{pb}}\right)^{-3/5}$
10d	$p_{FVa} = p_{FVb}$	×	$\left(\frac{P_{ia}}{P_{ib}}\right)^{2/5}$	×	$\left(\frac{N_a}{N_b}\right)^{4/5}$	×	$\left(\frac{\rho_a}{\rho_b}\right)^{3/5}$	×	$\left(\frac{K_{pa}}{K_{pb}}\right)^{2/5}$
10e	$L_{Wa} = L_{Wb}$	+ 14 log	$\left(\frac{P_{ia}}{P_{ib}}\right)$	+ 8 log	$\left(\frac{N_a}{N_b}\right)$	+ 6 log	$\left(\frac{\rho_a}{\rho_b}\right)$		*(Page 4 of 4)*

Note that an entire set of dependent variables must be calculated whenever a particular set of independent variables is changed.[74]

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Nomenclature:

$P_{ia} = P_i$ = fan input power
D = fan size (impeller or wheel diameter)
N = fan speed
N_s = fan specific speed
ρ = fan air density
Q = fan flow rate
K_p = compressibility coefficient
L_w = sound power level
p_{FT} = fan total pressure
p_{FV} = fan velocity pressure
p_{FS} = fan static pressure
η_S = fan static efficiency
η_T = fan total efficiency

Subscripts:

a = fan performance to be predicted
b = known fan performance
i = impeller power
s = shaft power
o = output power

Note: Fan laws Nos. 1, 2, 3, or 4 depend on the design situation.

For all the fan laws: $\eta_{Ta} = \eta_{Tb}$ and (point of rating)$_a$ = (point of rating)$_b$

In considering fan performance, it is incorrect to calculate a change for one characteristic and ignore the others in the Fan Law Group, *because it takes all three characteristics to properly define the operational point of the fan.*

Effects of Temperature

A fan moves the same volume of air or gas regardless of the density or molecular weight. Thus, a fan moving 600 cfm at 70°F will also move 600 cfm at 500°F. Because gases including air weigh less at 500°F than at 70°F, the fan will require less horsepower, but for the same reason with the gas/air weighing less, reduced static and velocity pressure will be generated. The reduction in static pressure is proportional to the reduction in horsepower. Therefore, the overall fan efficiency remains unchanged.[128]

From reference 128, total efficiency

$$= \frac{(\text{cfm})(\text{total pressure})}{(6{,}356)(\text{brake horsepower})} \tag{12-170}$$

Example 12-15. Change Speed of Existing Fan

A fan handling 85°F air is installed but has insufficient capacity. The unit is v-belt driven, and the available on hand sheaves to place on the unit will run the speed to 1,108 rpm. Use the same fan wheel.

The existing unit:

Wheel diameter = 24.5 in.
cfm = 8,708
Outlet velocity = 1,400 ft/min

Static pressure = 2 in. water
rpm = 957
Shaft hp = 3.78
Density = 0.075 lb/ft^3

The piping system and air density remain the same after conversion to the new speed. Determine the new operating conditions (continued).

Example 12-15/16. Fan Law 1

$$Q_a = (Q_b)(D_a/D_b)^3(N_a/N_b)(\rho_a/\rho_b) \quad (12\text{-}171)$$

Volume, Q_{new}

$$= 8{,}708(24.5/24.5)^3(1{,}108/957)(1) = 10{,}081 \text{ cfm}$$

$$\text{Static pressure, } P_{Fta} = P_{FIb}(D_a/D_b)^2(N_a/N_b)^2(\rho_a/\rho_b) \quad (12\text{-}172)$$

$$2(24.5/24.5)^2(1{,}108/957)^2(1) = 2.68 \text{ in. water}$$

$$\text{Shaft horsepower, } P_{ia} = P_{ib}(D_a/D_b)^5(N_a/N_b)^3(\rho_a/\rho_b) \quad (12\text{-}173)$$

$$= 3.78(24.5/24.5)^5(1{,}108/957)^3(1) = 5.86 \text{ shp}$$

Example 12-17. Change Pressure of Existing Fan, Fan Law 2

For the fan existing in Example 12-15, what will be the conditions if the static pressure must be changed from 2 to 3.5 in. due to a change in the system pressure level into which the fan discharges? The piping system, air density, and fan size remain unchanged. What will be the new operating conditions for this fan?

$$Q_a = Q_b(D_a/D_b)^2(p_{Fta}/p_{FTb})^{1/2}(\rho_a/\rho_b)^{-1/2} \quad (12\text{-}174)$$

$$\text{New capacity} = Q_a = 8{,}708(3.5/2)^{1/2}(1)^{-1/2} = 11{,}519 \text{ cfm}$$

New speed

$$= N_a = 95.7(D_a/D_b)^{-1}(3.5/2)^{1/2}(1)^{-1/2} = 1{,}265 \text{ rpm}$$

Shaft horsepower, P_{ia}

$$= 3.78(D_a/D_b)^2(3.5/2)^{3/2}(1)^{-1/2} = 8.75 \text{ SHP}$$

Example 12-18. Rating Conditions on a Different Size Fan (Same Series) to Correspond to Existing Fan

Use the existing fan of Example 12-15 as the reference unit. Determine the equivalent or same rating on a 27-in. diameter wheel fan available to operate at 858 rpm.

Fan Law 1: Refer to Table 12-16.

$$\text{Capacity, } Q_a = 8{,}708\left(\frac{27}{24.5}\right)^3\left(\frac{858}{957}\right) = 10{,}449 \text{ cfm}$$

$$\text{Static pressure, } p_{FTa} = 2\left(\frac{27}{24.5}\right)^2\left(\frac{858}{957}\right)^2 = 1.95 \text{ in. water}$$

$$\text{Shaft bhp} = 3.78\left(\frac{27}{24.5}\right)^5\left(\frac{858}{957}\right)^3 = 4.42$$

Note that this gives conditions of the same point of rating for the two fans, and this is the only manner in which fan laws apply to two different units.

Example 12-19. Changing Pressure at Constant Capacity

Use the data for the fan in Example 12-15.

The system pressure requires a change to 4-in. static pressure, but the cfm must remain the same. Determine the fan size, speed, and bhp.

Fan Law 5: Refer to Table 12-16.

Capacity, $Q_a = Q_b$ constant at 8,708 cfm

$$\text{Fan size, } D_a = D_b(Q_a/Q_b)^{1/2}(p_{FTa}/p_{FTb})^{-1/4}(\rho_a/\rho_b)^{1/4} \quad (12\text{-}175)$$

$D_a = 24.5(4/2)^{-1/4}(1) = 20.6$ in. wheel diameter

Speed, $N_a = 957(Q_a/Q_b)^{-1/2}(4/2)^{3/4}(1) = 7{,}609$ rpm

Shaft hp, $P_{ia} = 3.78(Q_a/Q_b)^1(4/2)^1(1) = 7.56$

Note that the wheel diameter of 20.6 in. is probably not found in manufacturers' standard units. Therefore, select a standard wheel diameter that is closest to the 20.6 in. and then recalculate by Fan Law 1 the change of that wheel's performance to the desired or necessary conditions just calculated. This can be accomplished by changing speed. Most manufacturers have a standard wheel of 20 in. with the next size being 22.25 in.

Example 12-20. Effect of Change in Inlet Air Temperature

The fan of Example 12-15 has been operating at 85°F so that the effect of inlet air density is not significantly different than at 70°F. Operations now require that the process air be heated to 175°F. What effect will this have on the fan operation?

Fan Law 1 or 6:
cfm = 8,708 (constant)
rpm = 957 (constant)
Air density at 175°F:

$$\rho = 0.075\left(\frac{460 + 70}{460 + 175}\right) = 0.0626 \text{ lb/ft}^3$$

$$\text{Pressure} = 2\left(\frac{0.0626}{0.075}\right) = 1.66 \text{ in. water}$$

$$\text{Shaft bhp} = 3.78\left(\frac{0.0626}{0.075}\right) = 3.16$$

Note that this less dense air requires less horsepower, and also that the fan can produce only 1.66 in. static as compared to 2 in. with standard air. The system resistance must be adjusted to accommodate this lower static pressure; otherwise, the fan will follow its characteristic curve by reducing its flow until it discharges at the static pressure of 1.66 in.

Peripheral Velocity or Tip Speed

The peripheral velocity of the fan wheel or impeller is expressed as:

$$V_p = \pi D' \,(\text{rpm}) \qquad (12\text{-}176)$$

where V_p = peripheral velocity, ft per min
D' = wheel diameter, ft
rpm = speed, revolutions per minute
π = 3.1416

Horsepower

1. The output power of P_o of a fan is the rate at which useful energy is delivered to the gas stream, based on polytropic compression.[74]

$$P_o = \frac{Q\, p_{FT} K_p}{C_Q}\text{, see Fan Law 5C, Table 12-16.}$$

Fan horsepower based on total pressure: Mechanical hp output[38],

$$(\text{hp})_t = \frac{0.0001573\, V_1\, p_t}{e_t} \qquad (12\text{-}177)$$

2. Fan horsepower based on static pressure output

$$(\text{hp})_s = \frac{0.0001573\, V_1\, p_s}{e_s} \qquad (12\text{-}178)$$

3. The power input to the impeller, P_i, is determined from the power output and the polytropic efficiency, η_p.[74]

$$P_i = \frac{Q\, p_{FT}\, K_p}{C_Q\, \eta_p}$$

Air horsepower (output)

$$(\text{hp})_a = 0.0001573 V_1 p_t \qquad (12\text{-}179)$$

4. Fan input power, P_s, is the sum of the power input to the impeller and the mechanical losses of the drive train:

$$P_s = P_i + P_m \qquad (12\text{-}180)$$

The drive losses are generally based on the use of the V-belt drives, because the use of these types of drives is quite popular; see losses from Figure 12-139. The mechanical losses in the drive train, P_m, must be considered separately, because they cannot be predicted by Fan Laws.

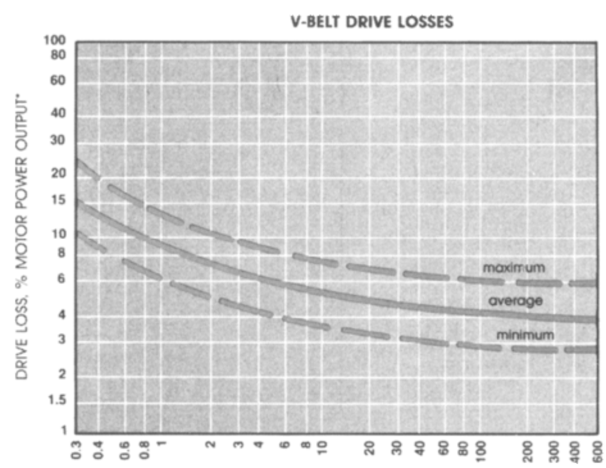
When Fan Law considerations are not involved:

$$P_s = \frac{Q\, p_{FT}\, K_p}{\eta_T\, C_Q} \qquad (12\text{-}181)$$

where η_T = total fan efficiency
= ratio of fan output power to fan input power
C_Q = 6,354, engineering units
P_s = fan input power, sum of power to impeller and mechanical losses; also = P_{ib}
P_m = mechanical loses, such as belt drive, gear drive, bearings
P_o = fan output power = total output power
Q = fan flow rate, cfm
p_{FT} = fan total pressure
K_p = compressibility coefficient
γ = isentropic coefficient

Shaft or brake horsepower (input) based on direct current motor[49]

$$\text{bhp} = \frac{(\text{Amps.})(\text{volts})(\text{motor efficiency})}{746} \qquad (12\text{-}182)$$



Range of Drive Loss for Standard Belts
Higher Fan Speeds Tend To Have Higher Losses
Than Lower Fan Speeds at the Same Horsepower

*Drive losses are based on the conventional V-belt, which has been the "workhorse" of the drive industry for several decades.

Figure 12-139. V-belt drive losses for fans. (Used by permission: Cat. C-2000, Dec. 1990. ©The Howden Fan Co.)

5. Shaft or brake horsepower (input) based on alternating current (3-phase) motor[49]

$$bhp = \frac{(\sqrt{3})(\text{amps})(\text{volts})(\text{motor eff.})(\text{power factor})}{746} \quad (12\text{-}183)$$

where p_t = total pressure of fan, in. of water
p_s = static pressure of fan, in. of water
e_t = total efficiency of fan, fraction
e_s = static efficiency of fan, fraction
V_1 = inlet volume, ft^3/min
hp = actual fan horsepower
bhp = brake horsepower

Efficiency

Total air horsepower *output* of a fan (i.e. the useful energy added to the flowing air/gas stream):

$ahp_s = 0.000157\, Q\, p_s$
or, $ahp_t = 0.000157\, Q\, p_t$

$$\text{Mechanical efficiency, } \eta_s = \frac{ahp_s}{bhp} = \frac{0.000157\, Q\, p_s}{bhp} \quad (12\text{-}184)$$

or,

$$\eta_T = \frac{ahp_t}{bhp} = \frac{0.000157\, Q\, p_t}{bhp} \quad (12\text{-}185)$$

where bhp = input horsepower to drive the fan, brake horsepower
ahp_s = air horsepower, static
ahp_t = air horsepower, total
Q = flow volume through fan, cfm
p_s = static pressure, in. water
p_t = total pressure, in. water
η_s = static efficiency, fraction
η_t = total efficiency, fraction

The mechanical efficiency of a fan is the ratio of the horsepower output to the horsepower input at the fan shaft.[131] The input horsepower to drive the fan consists of the air horsepower, the energy losses in the fan, fluid dynamic losses, shock losses, leakage, disk friction, and bearing losses (all as horsepower).[131] The fan outlet velocity pressure loss has been included in the fluid dynamic losses.

$$\text{Mechanical (total), } e_t = \frac{(hp)_a}{bhp} = \frac{0.0001573\, V_1\, p_t}{bhp} \quad (12\text{-}186)$$

$$\text{Static, } e_s = e_t \frac{p_s}{p_t} = \frac{0.0001573\, V_1\, p_s}{bhp} \quad (12\text{-}187)$$

Example 12-21. Fan Power and Efficiency

A fan is to handle 150,000 cfm air at 28 in. water gage (WG), 790 hp power input to the impeller. The inlet pressure is 29.92 in. Hg.

The isentropic coefficient is 1.4, air density = 0.070 lb/ft³, speed = 1,185 rpm. What is output power and total fan efficiency?[74]

p_{Ti} = 29.92 (13.62) = 407.5 in. WG.
Ratio: x = 28/407.5 = 0.0687

$$z = \frac{(\gamma - 1)(C_Q P_i)}{(\gamma)(Q\, p_{Ti})} = \frac{(1.4 - 1)}{(1.4)} \frac{[(6{,}354)(790)]}{[150{,}000(407.5)]} =$$

0.02346

$$K_p = \frac{z \ln(1 + x)}{x \ln(1 + z)} = \frac{(0.02346) \ln(1 + 0.0687)}{(0.0687) \ln(1 + 0.02346)}$$

$$K_p = 0.3414 \frac{(0.06644)}{(0.02318)} = 0.978$$

$$P_o = \frac{150{,}000(28)(0.978)}{6{,}354} = 652.4$$

η_T = fan total efficiency = 652/790 = 0.825 (fraction)

Temperature Rise

The temperature rise as the gas passes through a fan is

$$(T_2 - T_1) = T_1 \frac{[(p_2)^{n - 1/n} - 1]}{(p_1)} \quad (12\text{-}188)$$

$\Delta = (T_2 - T_1)$, temperature rise, °F

where Δt = temperature rise, °F
T_1 = air or gas temperature at fan inlet, °R
p_2 = fan outlet static pressure, in. water abs or other absolute units
p_1 = atmospheric pressure or fan inlet pressure (if not atmospheric), in. water absolute or other absolute units
T_2 = air or gas temperature at outlet, °R
P_v = fan outlet velocity pressure, in. water abs or other absolute units
n = polytropic coefficient
e_s = fan static efficiency, fraction

Fan Noise

In many installations the noise of fan operation is important. This is particularly true in heating and air conditioning applications and is a point to consider in industrial applications.

Sound power is the total energy emitted from a fan that is a function of the fan's speed and point of operation and is independent of the fan's installation and surrounding environment.[128] Sound power level is the acoustical power expressed in decibels (dB) radiating from a source.[128] Sound power can be converted into predictable pressure levels (dBA) after the acoustical environment surrounding the fan is defined.[128] Sound pressure for a specific fan varies with change in air volume, pressure, or efficiency. Note that "fan sound power ratings and corrections only reflect noise created by air turbulence within the fan," and do not include other mechanical or vibration noise.[128] Reference to the Air Movement and Control Association International, Inc., standards will provide specific details that the manufacturers use as guides. The AMCA[134] recommends "never ask for a dBA (sound pressure) fan rating as it is not clear what information will be offered, as dBA depends on too many variables. Sound pressure can only be calculated from fan sound power ratings with known variables. Refer to the respective manufacturers."

Fan noise is a function of fan speed, air velocity, and the system in which a fan is operating. Thus, there can be a choice of low fan speed and high air velocity or vice versa. Noise is proportional to the pressure developed and is not affected appreciably by the type of blading of the wheel.

As expected, fans operate quietest at the point of maximum efficiency. The highest noise level will be for frequencies below 100 cycles per second.[11] At the higher pressures, the fan should be selected in the region of peak efficiency for quiet operation. Table 12-17 presents some guide information for noise level with respect to fan operation.

Sound and the sound laws for fans are thoroughly discussed by Madison.[38, 74]

Table 12-17
Generally Accepted Fan Outlet Velocities for Quiet Operation

Static Pressure, in. Water	Outlet Velocity, ft/min
$^1/_4$	700–1,000
$^3/_8$	800–1,100
$^1/_2$	900–1,200
$^5/_8$	975–1,300
$^3/_4$	800–1,400
$^7/_8$	900–1,500
1	850–1,600
1 $^1/_4$	900–1,750
1 $^1/_2$	1,150–1,900
1 $^3/_4$	1,350–2,050
2	1,400–2,200
2 $^1/_2$	1,500–2,500
3	1,700–2,500
4	1,900–2,500
5	2,100–2,600
6	2,300–2,600

Note: The lower values offer the quietest operation. Values lower than those indicated will usually ensure quiet operation for the average system.

Used by permission and compiled from: Bul. 221, Clarage, A Twin Cities Fan Company, and "Applications Manual" No. 1121, The Howden Fan Company.

Fan Systems

An operating fan is always a part of some system. This system may be simple, such as a propeller fan exhausting an area to the atmosphere, or it may be quite complex and consist of different sizes of duct, elbows, transition pieces, dust collectors, etc. Regardless of the system, the fan cannot be selected until the flow and resistance characteristics have been analyzed. Fan selection for the system is based on the static pressure for a given volume of gas flowing. Because most fans operate at relatively low pressures, the effect of uncertainty or error in resistance calculations can have a large percentage effect on horsepower and operational characteristics. Despite considerable work by many investigators, it is essentially impossible to determine exact figures for the system resistances. It is usually a good practice to add 10–20% to the calculated static pressure as a safety factor. Some control should be installed on the inlet or discharge side of the fan, Figure 12-135B.

As a general guide for the average system, if the actual system pressure requirement is known for one flow capacity, the system can be calculated assuming the pressure varies as the square of the volume. The curve is parabolic going through the origin of a pressure-volume plot.

A fan can operate only along its characteristic curve, but after that fan is placed in a fixed system, it can operate only at the one point where pressure-volume conditions match the pressure-volume system curve calculated based on the system resistance, see Figure 12-134. Thus, if the fan characteristic curve is superimposed on the plot of the system, the point of intersection will be the point of operation. To change this point requires changing at least one condition on the fan or the system.

The fan discharge pressure necessary to overcome the system resistance is composed of friction loss plus losses due to changes in velocity (accelerations and decelerations) in the duct and connections of the system. If the fan must discharge into or maintain a system pressure at a constant level greater than that resulting from system losses, a constant pressure must be added to the calculated losses to obtain the true fan discharge pressure requirements.[25] That is, if the fan discharges through a duct system into a chamber to be held at 1.5 in. water positive pressure, then this is a fixed pressure level that is added to the calculated flow losses. See Figure 12-138. In general, accelerations do not contribute any appreciable portion of the velocity change losses. On the other hand, decelerations contribute a considerable portion.

In order to analyze the total system resistance and its relation to fan performance, Figure 12-140 is used. Without defining what comprises the system resistance, but representing it by Curve A–A, this system is to flow 13,000 cfm of air at 1.1 in. static pressure. A fan has been selected that operates at 600 rpm and is represented by its static pressure Curve C–C. The intersection of these two curves, Point 1, is the only point of

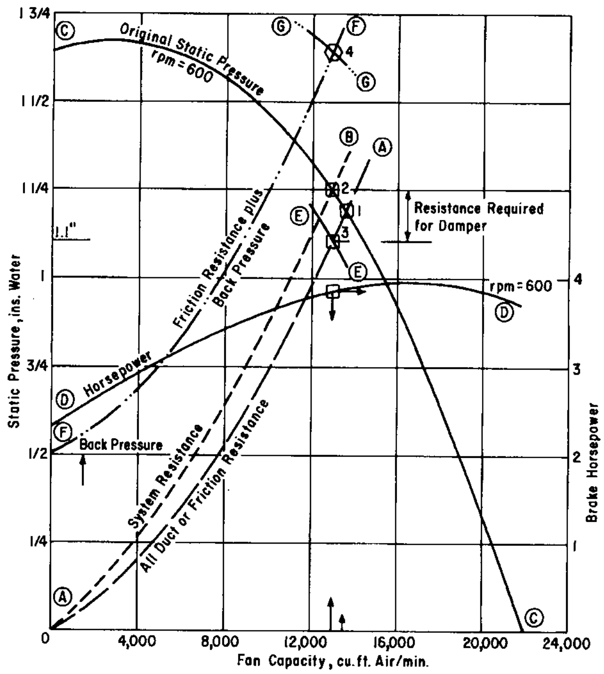


Figure 12-140. System resistances.

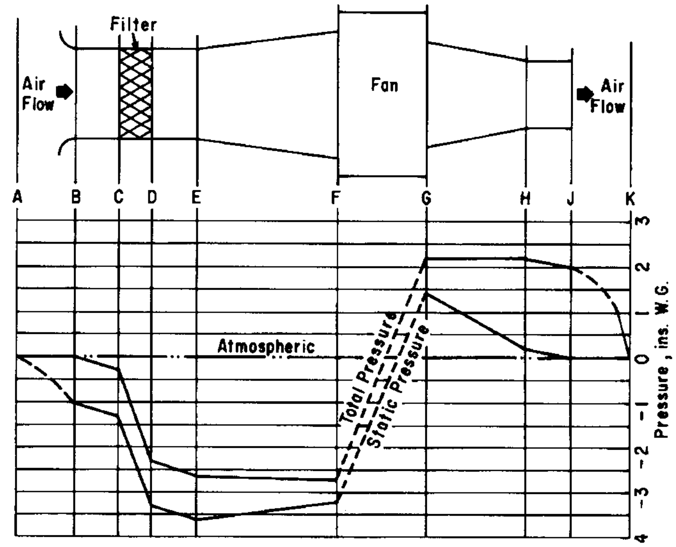


Figure 12-141. Typical gradient curves of pressure through a fan system. (Used by permission: The Howden Fan Company.)

operation for the system. It is not the exact point required for this system, as Point 1 represents a flow of 13,600 cfm at 1.19 in. static pressure and the requirement of 3.87 hp. To obtain the exact conditions of the problem, the fan discharge may be dampered. The damper resistance will have to be equivalent to the fan pressure at 13,000 cfm (1.19 in. static pressure) minus the required 1.1 in. static pressure, which is equal to 0.09 in. The horsepower from Curve D–D will be 3.83 hp.

The fan will now have a system resistance Curve A–B and operate at Point 2. As an alternate approach to securing a system balance at the point required, the motor speed can be changed by a suitable means. If the fan speed is reduced by 13,000/13,600, the new speed should be (0.889)(600) = 573 rpm. A new fan Curve E–E will go through the desired point conditions. The new horsepower for this operation will be $(3.87)(573/600)^3 = 3.35$ hp.

Note that the second scheme requires less running horsepower but does require adjustment to a new fan speed to produce Curve E–E. They will now operate at Point 3 without a damper. If the resistance of the system had included a back-pressure of $^1/_2$ -inch water as shown by system Curve F–F, the volume of 13,000 cfm would be reached at some speed greater than 600 rpm and located at Point 4 on the static pressure Curve G–G.

System Component Resistances

The total system pressure loss or resistance is the sum of the resistances of individual component parts, such as ducts, enlargements, contractions, filters, etc.

The total system loss is made up of the sum of the friction pressure losses and the velocity change or dynamic pressure losses. At any point in the system:

$$p_t = p_s + p_v \tag{12-189}$$

Velocity pressure:

$p_v = (v_m/4005)^2$, for air, see Table 12-15. $\rho = 0.075$ lb/ft^3

See Equation 12-164 for other gases.

v_m = mean velocity of flow, ft/min = (ft^3/min)/A_d
A_d = cross-sectional area of duct, ft^2

As the air or gas flows through a system, the total pressure decreases in the direction of the flow. Figure 12-141 shows this for a typical system.

Total system pressure = sum of all duct friction on suction and discharge side of fan + the sum of all acceleration and deceleration losses on suction and discharge side of fan + (velocity head of system exit − velocity head of system inlet) + (static backpressure of system on discharge − existing static pressure on suction side of system).

Note that for straight duct flow at constant cross-section, the total and static pressures decrease together (constant resistance). At the contraction section, the total pressure decreases very little, but static pressure is converted to velocity pressure, because static and velocity pressures are mutually convertible. At the sudden enlargement, the process of changing the velocity pressure to static pressure is inefficient, and a total pressure loss occurs. At J the static pressure is 0, and the total pressure is the velocity pressure as the gas stream leaves the duct.

Although the term *static pressure* is generally used in designing fan and duct systems and is the performance quoted by fan manufacturers in tables and charts, the *total* pressure of the system is a characteristic for which the total mechanical energy must be supplied for the *system*. Note that when the velocity in a duct remains constant, the velocity head remains constant.

Duct Resistance

For most low-pressure air systems, the flow is through sheet metal duct, either circular or rectangular. For very large flows and/or pressures higher than light gage duct work can withstand, the duct is usually made of fabricated metal plate and in some instances may be coated or lined for protection from corrosive process gases.

Air

Most air conditioning and ventilating data is established for air handling. Figure 12-142 gives the friction losses for air in straight ducts and is based on standard air of 0.075 lb/ft^3 density flowing at 70°F and 14.7 psia through clean, round galvanized metal ducts having approximately 40 joints per 100 ft. No safety factor is provided in this chart as it is based on[30]

$$h_f = f\frac{l\, v_f^2}{D_i(2g)} \tag{12-190}$$

where h_f = head loss due to friction, in ft of *fluid* flowing
l = length of duct, ft
D_i = I.D. of duct, ft
v_f = fluid velocity, ft per sec
g = acceleration of gravity, 32.17 ft per sec/sec
f = friction factor dependent upon Reynolds number and relative roughness of the duct. (See the "Fluid Flow Chapter," V. 1, for N_{Re} vs. Friction Factor, f, chart or standards of the Hydraulic Institute.)

At low pressures encountered in ventilation and other fan applications, friction in ducts may be corrected for changes in air or gas density without serious error by[30]

$$h_o = h_{fs}\left(\frac{\rho_o}{\rho_s}\right) \tag{12-191}$$

where (in consistent units)
h_o = friction or head loss under actual operating conditions, ft fluid, or in fluid
h_{fs} = friction or head loss under standard air conditions, same units as h_o
ρ_o = density of air under actual operating conditions, lb/ft^3
ρ_s = density of air under standard conditions, lb/ft^3

Other Gases

Air charts are not applicable to process gases having properties and conditions different than air. To determine such losses, the pressure drops should be calculated taking the gas properties into account.

Duct Velocity

Table 12-18 gives recommended and maximum duct velocities for a variety of conditions. Note that for industrial process applications for air or gas handling the average values to consider are 1,600–2,400 FPM with 2,000 FPM being a fair value. Of course, specific conditions should dictate the use of such guideline figures.

High-velocity air-handling systems for ventilation (heating and cooling) are sometimes designed as high as 6,000 FPM, but these conditions require special study as the noise level is high without sound-deadening arrangements.

Summary of Fan System Calculations

1. Lay out system, showing connections, duct lengths, all fixed resistances, etc.
2. Calculate individual *total pressure* losses for duct, elbows, expansions, etc. For fittings, the losses are *total* when calculated as indicated by the equations, and the duct loss is friction only. Also see references 11 and 128 and Chapter 2, V. 1, 3rd Ed., of this series.
3. Total values calculated in (2). This is the system total pressure loss, static losses plus velocity head.
4. For fan size determinations use
   a. Value of *total pressure* of (3) if *total pressure* tables or charts are available.
   b. *Static pressure* tables or charts. For static pressure value, determine velocity pressure at fan discharge, $(v_m/4{,}005)^2$ for air and subtract from system total pressure of (3).

Series Operation

The operation of low-pressure fans in series is essentially a condition of a constant weight of flow of gas through the units with the final discharge total pressure being the sum of the total pressures of the individual fans. It is somewhat unusual to have more than two fans in a series.

Usually fans are placed in series to obtain more pressure than any reasonable single fan will produce. They may be required to boost pressures as dictated by system resistance changes. In this latter case, it is important to pick fans near peak efficiency to allow for deviations in operations without paying a premium in horsepower. It is important to keep in mind that the addition of fans in series does not increase the capacity of flow by the additive values of the individual fans.

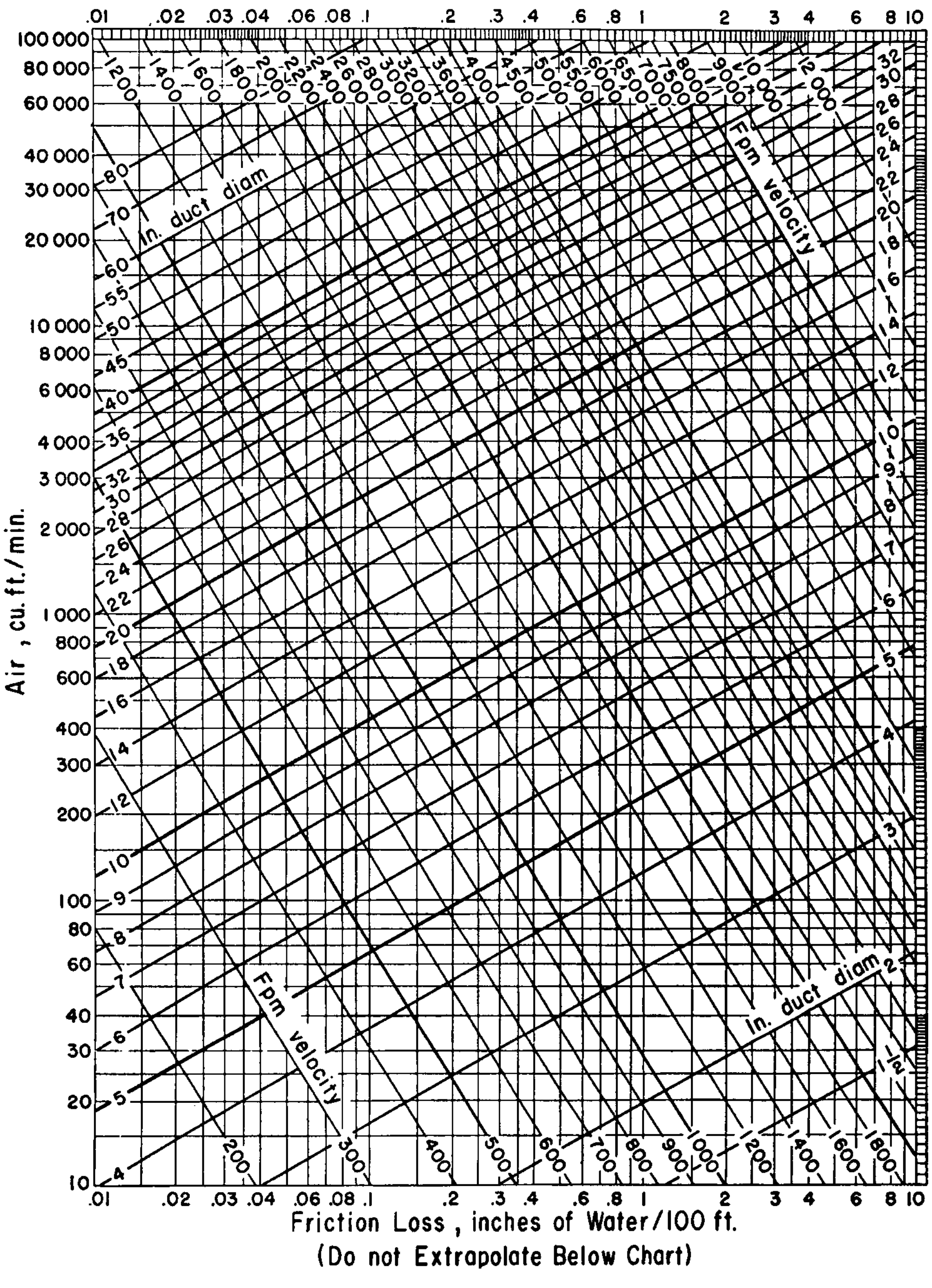


Figure 12-142. ASHVE Friction chart for air in duct. (Used by permission: *Heating, Ventilating and Air Conditioning Guide,* ©1949. American Society of Heating and Ventilating Engineers, reprinted© with permission; Air Movement and Control Association International, Inc. All rights reserved.)

This chart applies to smooth, round galvanized iron ducts. See the following table for corrections to apply when using other pipe.

Type of Pipe	Degree of Roughness	Velocity ft/min	Roughness Factor (Use as multiplier)
Galv. steel	Med. rough	1,000–3,000	1.43
Concrete	Med. rough	1,000–2,000	1.4
Riveted steel	Very rough	1,000–2,000	1.9
Tubing	Very smooth	1,000–2,000	.9

EXAMPLE: Find friction loss 6,000 ft^3 per min (cfm) through 100 ft of 16-in. diameter pipe. Select 6,000 cfm on left scale and move horizontally right to diagonal line marked 16 in. The other intersecting diagonal shows that velocity in the pipe is 4,300 ft per min. Vertically below,the friction per 100 ft is 1.35 in.

Table 12-18
Recommended and Maximum Duct Velocities

	Recommended Velocities, fpm			Maximum Velocities, fpm		
Designation	Residences	Schools, Theaters, Public Buildings	Industrial Buildings	Residences	Schools, Theaters, Public Buildings	Industrial Buildings
Outside air intake*	500	500	500	800	900	1,200
Filters*	250	300	350	300	350	350
Heating coils*	450	500	600	500	600	700
Air washers	500	500	500	500	500	500
Suction connections	700	800	1,000	900	1,000	1,400
Fan outlets	1,000–1,600	1,300–2,000	1,600–2,400	1,700	1,500–2,200	1,700–2,800
Main ducts	700–900	1,000–1,300	1,200–1,800	800–1,200	1,100–1,600	1,300–2,200
Branch ducts	600	600–900	800–1,000	700–1,000	800–1,300	1,000–1,800
Branch risers	500	600–700	800	650–800	800–1,200	1,000–1,600

*These velocities are for total face area not the net free area; other velocities in table are for net free area.

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For Two Low-Pressure Identical Fans

Figure 12-143 shows the individual static pressure curve p_{fs} and total pressure curve p_{ft}. If pressure losses between the two fans are neglected (and they should be very low for good design), the combined total pressure curve is twice the value of curve p_{ft}, 2 p_{ft}. The new operating static pressure also should be twice the individual total pressure value minus the velocity pressure, $2\,p_{ft} - p_{fv}$; for *identical* fans, the new operating static pressure is equal to $2\,p_{fs} + p_{fv}$. The operation of the series fans will be along the system resistance curve, and the resultant point of operation will be at the intersection of the system curve with the curve for $(2\,p_{ft} - p_{fv})$.

For Two Low-Pressure Different Size Fans

Figure 12-143 shows the curves for identical fans. When unlike fans are placed in series, the individual static and total pressure curves are placed on the graph. The individual curves for assumed fans No. 1 and No. 2 and the system resistance curve define the system. The combined total pressure curve is the sum of the individual values at the same capacity. Then total combined pressure curve = p_{ft} (No. 1) + p_{ft} (No. 2). The new combined operating static pressure curve is the sum of the individual static pressure valves at the same capacity and exists only at the outlet of the second fan; then, the total combined static pressure = $p_{ft1} + p_{ft2} - p_{fv2}$. Any losses in connections between the fans will reduce the values of the total static and system total pressures.

High-Pressure Fans

The effect of density of the air or gas entering the suction of the second (and/or third, etc.) fan is significant for high-pressure units operating individually above about 15 in. of water static. The static pressure rise in the second fan corrected for suction pressure is as follows, neglecting losses.[38]

Actual pressure rise of a second fan in series

$$= \left(\frac{\text{absolute suction pressure of second fan}}{\text{absolute atmospheric pressure}}\right) \times \left(\frac{\text{design pressure rise of second fan}}{\text{with atmospheric suction}}\right)$$

Parallel Operation

Parallel operation of many fans in a system is common; yet, it is a condition that must be analyzed to avoid improper and inefficient operation.

Fans are used in parallel to obtain increased capacity in preference to a single large installation, to increase capacity at constant pressure, and for low-resistance systems requiring large capacities. It is important to study the effect of the addition or removal of fans on the system. This is done using the system resistance and the fan characteristics.

For two or more fans, the capacity for the system is the sum of the individual capacities at a given static or total

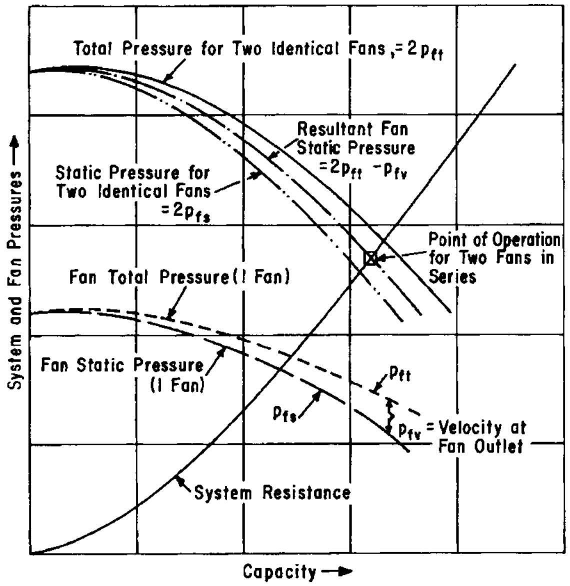

Figure 12-143. System for two identical fans operating in series.

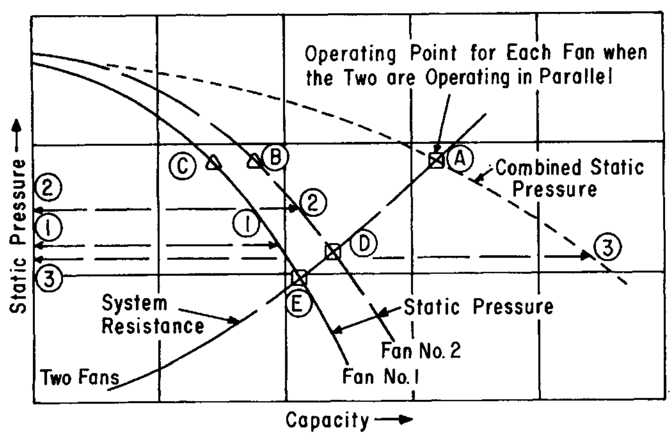

Figure 12-144. Parallel operation of two unlike fans.

pressure. This is shown in Figure 12-144 for different fans. The static pressure total is often the one used for system study in preference to total pressure, including velocity pressures. The calculated system resistance is plotted and intersects the total combined static pressure at Point A. This is the operational point for the two fans in this particular system and is composed of the capacities at Points B and C for the two fans. If only one fan runs in this same system, the calculated resistance at the volume flow of one fan is either Point D or E, for fans No. 2 and No. 1, respectively.

With one fan running at Point E, assume that the second fan is started. As it picks up speed, it follows its static pressure curve toward Point B (see Figure 12-144). It is operating at the intersection points of system resistance equivalent to that calculated for one full fan volume plus some equivalent in changing friction of the second fan's capacity. Such hypothetical points might be anywhere along D–B as well as giving a total effect along the combined static pressure curve. In the actual situation, although the result is the same, the first fan falls off some in capacity as the second fan picks up load, and *together* they follow up the system resistance curve toward A, their final point of steady operation for this fixed system.

A qualitative and somewhat quantitative indication of the case with which the two (or more) fans will parallel is represented by their limit curve of Figure 12-145.[27] This curve is constructed by starting at pressure P corresponding to the system intersection A, and plotting the increments of Volume x, y, etc., to define the limit of available volume that the duct can accept and which the second fan must pick up as it comes on the line. For good parallel operation, the limit curve must intersect the combined static pressure curve, SP, at only one point.

This must be a clear and definite intersection and not the situation represented in Figure 12-146 for a forward bladed fan system. Note that the limit curve parallels the static curve and has a very poor long tangential intersection. The

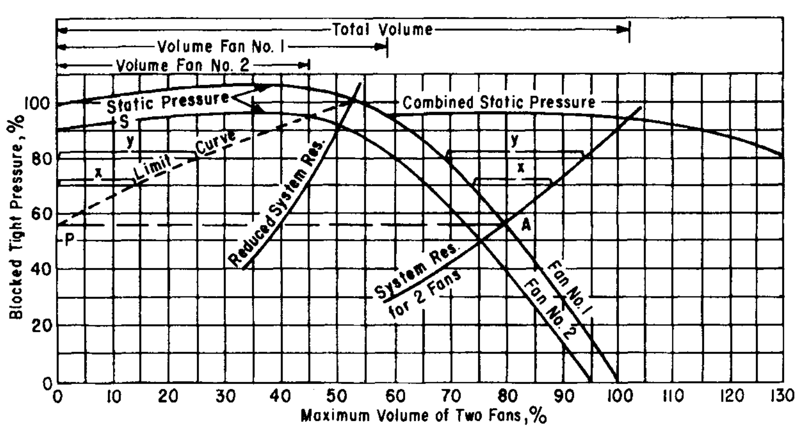

Figure 12-145. Curve for fans operated in parallel at different speeds. (Adapted, revised, and used by permission: The Howden Fan Company.)

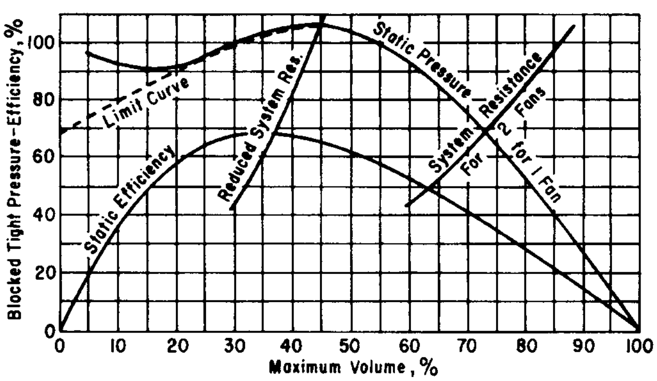

Figure 12-146. Parallel analysis applied to forward curved blade fan. (Used by permission: The Howden Fan Company.)

greater the combined static pressure curve is above the limit curve, the greater is the certainty of good parallel operation. Forward curve blade fans can be operated in parallel; however, the operation must be at pressures considerably lower than the peak, and thus, high efficiency and good operation of this fan are almost incompatible.

Note that the total static pressure curve of Figure 12-145 is limited by the lowest output pressure of the multifan system. The limit curve is established using the fan curve (No. 1 in this example) having the smallest volume increment to the system resistance curve. In this situation fan No. 2 cannot add to the system until its pressure-volume relation reaches the peak point on its curve.

Although the analysis of parallel operation indicates that a fan may not operate satisfactorily, often it actually will operate, but under modified conditions. The effect of a slight difference in the individual fan ductwork can be enough to allow operation, or sometimes a change in damper setting will allow operation. Usually in such situations, efficiency will be reduced with a higher horsepower consumption. If the fans discharge toward each other in such a way as to affect each other's operation, the fans may actually operate at a reduced pressure, somewhere between the static and total pressure curves.[38]

Fan Selection

To have the fan represent the best possible selection considering the particular circumstances and requirements, it is important to study the fan type curves and to recognize whether a small change in system resistance would be easily handled by a particular fan, whether speed variations and the resulting volume and pressure changes are acceptable, and whether the fan can be protected against corrosion, etc. References 19, 31, and 38 will be helpful. Specifications should be submitted to several manufacturers for their recommendations. In this way full advantage is received from specialized knowledge of applications and the associated problems. Of particular importance is the evaluation of various volume and pressure control schemes. The process engineer must be familiar with the manufacturers' rating tables in the catalogs and be in a position to make specific selections as well as to check manufacturers' recommendations.[51,52]

The volume of a fan should be determined by (1) the process material balance plus reasonable extra (about 20%) plus volume for control at possible future requirements; (2) generous capacity for purging; and (3) process area ventilation composed of fume hoods, heat dissipation, and normal comfort ventilation. Table 12-19 gives suggested air changes for area ventilation, but not air conditioning. Excellent details for evaluation and the design of ventilating, air conditioning, and heating can be found in Reference 31.

Table 12-19
Average Air Changes Required for Good Ventilation

	Minutes per Change		Minutes per Change
Assembly halls	2–10	Generator rooms	2–5
Auditoriums	2–10	Gymnasiums	2–10
Bakeries	2–3	Kitchens, hospital	2–5
Banks	3–10	Kitchens, residential	2–5
Barns	10–20	Kitchens, restaurant	1–3
Bars	2–5	Laboratories	1–5
Beauty parlors	2–5	Laundries	1–3
Boiler rooms	1–5	Markets	2–10
Bowling alleys	2–10	Offices	2–10
Churches	5–15	Packing houses	2–5
Clubs	2–10	Plating rooms	1–5
Dairies	2–5	Pool rooms	2–5
Dance halls	2–10	Projection rooms	1–3
Dining rooms	3–10	Recreation rooms	2–10
Dry cleaners	1–5	Residences	2–5
Engine rooms	1–3	Sales rooms	2–10
Factories	2–5	Theaters	2–8
Forge shops	2–5	Toilets	2–5
Foundries	1–5	Transformer rooms	1–5
Garages	2–10	Warehouses	2–10

Used by permission: Bul. A-108, Hartzell Fan Engineering Data. Hartzell Fan Company, Piqua, OH.

Multirating Tables

The multirating tables of the fan manufacturers are convenient for selecting any of the many types of fans. Figure 12-147 is one portion of such a table. Usually cfm values can be found close enough to requirements to be acceptable. Direct interpolation in the table for volume, rpm, and bhp is acceptable for narrow ranges; otherwise the Fan Laws must be used.

CFM	OUTLET VELOCITY	1 3/4" STATIC RPM	1 3/4" STATIC BHP	2" STATIC RPM	2" STATIC BHP	2 1/4" STATIC RPM	2 1/4" STATIC BHP	2 1/2" STATIC RPM	2 1/2" STATIC BHP	3" STATIC RPM	3" STATIC BHP	3 1/2" STATIC RPM	3 1/2" STATIC BHP	4" STATIC RPM	4" STATIC BHP	4 1/2" STATIC RPM	4 1/2" STATIC BHP	5" STATIC RPM	5" STATIC BHP
2660	1400	1206	.96	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
3040	1600	1251	1.12	*[illegible]	*[illegible]	1371	1.41	—	—	—	—	—	—	—	—	—	—	—	—
3420	1800	1304	1.29	1364	1.46	*[illegible]	*[illegible]	*[illegible]	*[illegible]	1573	2.11	—	—	—	—	—	—	—	—
3800	2000	1368	1.50	1419	1.66	1473	1.84	1520	2.01	*[illegible]	*[illegible]	1710	2.74	—	—	—	—	—	—
4180	2200	1434	1.74	1485	1.91	1534	2.11	1578	2.28	1667	2.66	*1755	*3.05	1845	3.46	1927	3.87	—	—
4560	2400	1506	2.02	1552	2.20	1600	2.40	1640	2.58	1744	2.98	1808	3.39	*1894	*[illegible]	*1970	*4.27	2046	4.71
4940	2600	1583	2.35	1625	2.54	1666	2.73	1705	2.92	1786	3.33	1865	3.76	1945	4.22	2020	4.69	*2093	*[illegible]
5320	2800	1661	2.71	1700	2.90	1741	3.11	1780	3.30	1854	3.74	1930	4.19	1999	4.64	2074	5.13	2142	5.62
5700	3000	1742	3.11	1779	3.31	1816	3.53	1854	3.74	1923	4.20	1995	4.66	2059	5.11	2131	5.63	2199	6.13
6080	3200	1822	3.56	1857	3.77	1893	4.02	1929	4.24	1996	4.70	2064	5.18	2124	5.64	2194	6.18	2257	6.70
6460	3400	1893	3.99	1939	4.28	1974	4.54	2005	4.78	2070	5.26	2140	5.76	2198	6.23	2261	6.79	2320	7.33
6840	3600	1988	4.60	2020	4.84	2054	5.11	2085	5.35	2149	5.86	2214	6.39	2270	6.89	2330	7.47	2387	8.01
7220	3800	2072	5.19	2101	5.44	2136	5.73	2166	5.99	2226	6.53	2290	7.07	2344	7.60	2401	8.20	2456	8.76
7600	4000	2160	5.85	2188	6.11	2218	6.40	2249	6.68	2307	7.24	2364	7.81	2420	8.38	2477	8.97	2529	9.54

Rated in accordance with NAFM Bulletin 110, Plate 1
*Points of maximum mechanical efficiency
Based on Standard Air of 0.075 lb/ft³ (70°F at sea level)

Single Width, Single Inlet

Outlet outside dimensions 15 $^1/_2$ in. × 18 $^1/_2$ in.
Wheel 18 $^1/_4$ in. diameter
Outlet area = 1.90 ft^2 inside
Tip speed = 4.78 × rpm, ft/min

$$\text{Max. bhp} = .591 \left(\frac{\text{rpm}}{1000}\right)^3$$

Density = .0750 lb/ft^3
Max. rpm = 3,020
Max. temp. = 200°F, air free from abrasive particles
Wheel uncoated except for paint

Figure 12-147. Typical manufacturer's rating table for backward blade wheel. (Original use of table for first two edtions used by permission: ILG Electric Ventilating Co. Note: This company cannot be located in business in 1999.)

Performance tables are based on standard dry air at 70°F at sea level (barometric pressure 29.92 in. Hg) with a density of 0.075 lb per ft^3. When the fans are required to handle gases at other conditions at the inlet, corrections must be made for temperature, altitude, and air or gas density.

The system resistance must be calculated in the usual manner and at the actual operating conditions of the fan. Corrections are then applied to convert this condition to "standard" for use in reading the rating tables.

Performance at Other Than Standard Conditions: Corrections from Charts

1. Calculate actual density of gas (or air) under operating conditions.
   For *air*, Figure 12-148 is convenient to use:
   Read temperature correction factor, F_1.
   Read altitude correction factor, F_2.

$$\text{Actual density} = \frac{(F_1)(F_2)}{0.075}, \text{ lb/ft}^3 \text{ at operating conditions} \tag{12-192}$$

For gases other than air the density must be calculated because the curves of Figure 12-148 are for 0.075 lb/ft^3 density air. If the gas handled has a density *at* 70°F and 14.7 psia close to that of air under these conditions, then the curves could be read for convenience. Otherwise, calculate the actual gas density by the gas laws.

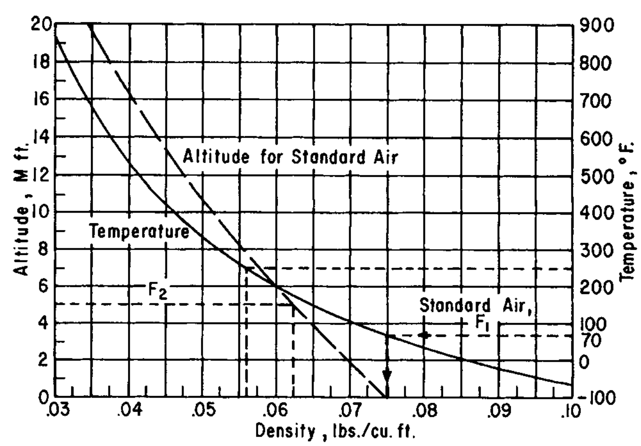

Figure 12-148. Correction factors for effects of altitude and temperature on standard air. (Used by permission: Clarage, A Twin City Fan Company.)

2. Calculate the equivalent static pressure:

$$= (\text{Required static pressure})\frac{0.075}{(\text{actual density})} \tag{12-193}$$

3. From manufacturers' rating tables for air or gas, at the required cfm *at inlet operating conditions* and the equivalent static pressure calculated in (2), read the rpm and bhp. Interpolate if necessary.
4. The rpm is the correct value for the actual operating conditions.
5. The bhp must be corrected for density:

$$\text{Actual bhp} = (\text{bhp from table})\left(\frac{\text{actual density}}{0.075}\right) \qquad (12\text{-}194)$$

6. The correct performance at the actual operating conditions will be as follows:
cfm as set at inlet conditions
static pressure as set at inlet conditions, in. of water
temperature as set at inlet conditions
rpm as read from manufacturers' tables
bhp as corrected by (5)

Alternate Correction for Performance at Other Than Standard Air Conditions

This is essentially the same as the previous method; however, for air systems only it offers some convenience.

1. Read correction factor from Figure 12-149.
2. Divide the actual static pressure required by the correction factor to obtain the equivalent static pressure.
3. Using actual cfm at intake conditions to fan and *equivalent* static pressure (2), read manufacturers' rating tables for bhp and rpm. Note that if exact cfm or pressure value is not listed in tables, it may be reached by (a) interpolation as previously described for approximation or (b) by using fan laws to correct one set of table values.
4. The rpm is correct as read from the table (or as modified by 3a or 3b).
5. The bhp at operating condition

$$= (\text{bhp from tables})\ (\text{factor from Figure 12-149})$$

Example 12-22. Fan Selection for Hot Air

A system requires 6,060 cfm of air at 400°F against a 2.04 static pressure. The installation is at an elevation of 1,400 ft.

Determine what type of fan blading should be used. A backward-curved blade fan will be selected for this installation because the following are not known: (1) the accuracy with

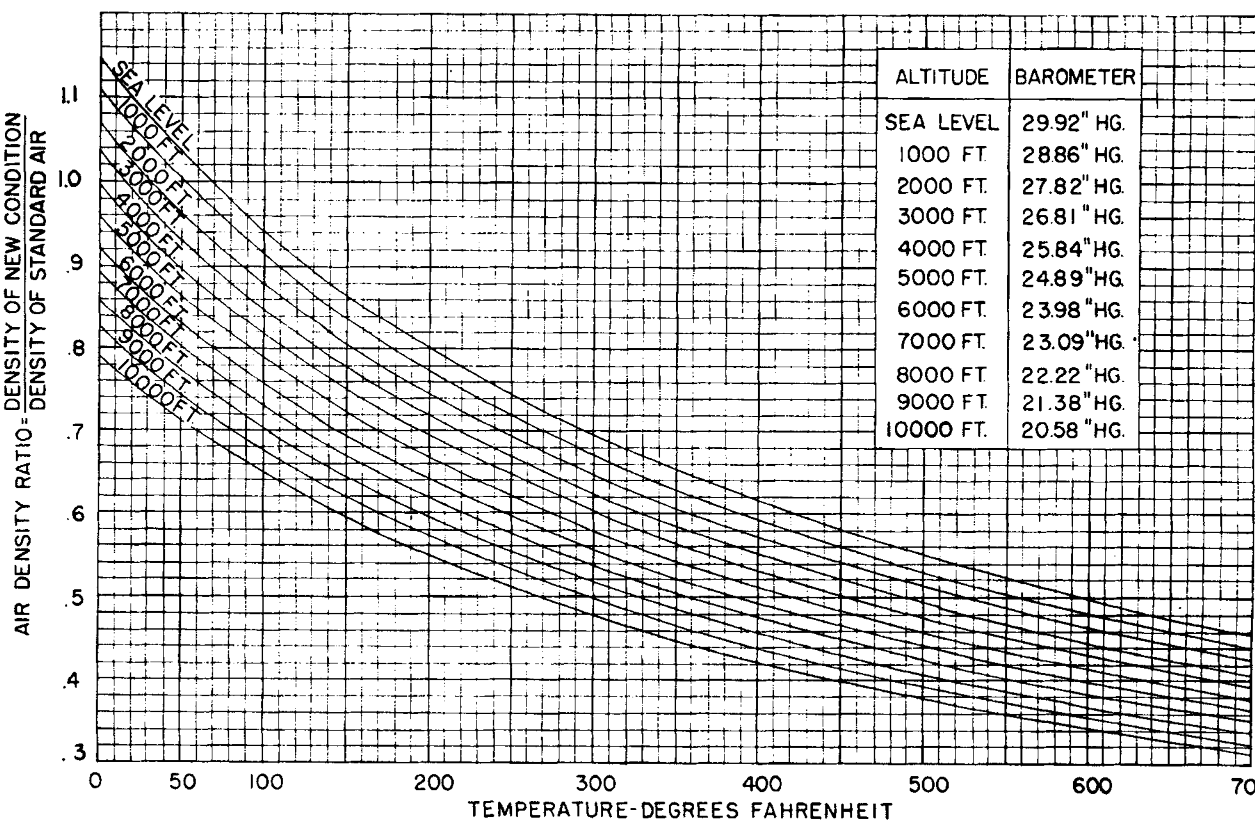

Figure 12-149. Correction factor for air only; use with nonstandard air conditions. (Used by permission: Clarage, A Twin City Fan Co.)

which the system characteristic of 2.04 in. of water at 6,060 cfm was determined, (2) the type of process control to be used, and (3) the possible system variation.

A backward-curved blade will take care of the preceding unknowns. It will have

1. High efficiencies. It is a blade that offers flexibility by inherently providing high efficiencies over a wide range. It has its highest efficiency near its maximum horsepower. This gives flexibility above and below the design point.
2. Nonoverloading characteristics. A backward-curve blade will allow close "motoring" without fear of overloading in the event of process upsets.
3. Steep static pressure curves. It offers a wide range of static pressures with a small change in capacity.

Determine whether the 6,060 cfm at 400°F and against $1\,{}^1/_2$-in. static pressure can be used to select a fan from the manufacturers' tables. The manufacturers' tables are prepared in accordance with the industry standard set up by the Air Movement and Control Association.[49] These tables are based on standard air.

Operating conditions other than these must be corrected before going into the table.

Because this is an *air* system, the density correction chart in Figure 12-148 is used.

1. Actual density of air at operating conditions:

From chart, read at 400°F, factor $F_1 = 0.046$.
Also read at 1,400-ft altitude, factor $F_2 = 0.0713$.
Actual density $= (0.046)(0.0713)/0.075 = 0.0433\ \text{lb/ft}^3$

$$\text{Air density ratio} = \left(\frac{0.0433}{0.075}\right) = 0.584$$

2. Equivalent static pressure (at standard conditions):

$$= 2.04\left(\frac{0.075}{0.0433}\right) = 3.53\ \text{in. water}$$

3. Select fan from manufacturers' performance table, Figure 12-147, at 3.53 in. and 6,060 cfm, because a constant speed fan will deliver the same volume against 2.04 in. at the density of 0.0433 as it will at 3.53 in. and standard air conditions. Note that this particular table limits selections to 200°F operating temperature. Usually a manufacturer will have the same style fan available in the next class, which will allow the needed higher temperature operation. Sometimes the limit is only in the type of bearings and their cooling arrangements. For this example assume that an acceptable unit is available, which will perform essentially according to the tables given.

   For practical *estimating* purposes interpolation can be overlooked here, because the values of the table are so close to the actual values.

cfm = 6,080
Speed = 2,064 rpm at 3.5 in.
bhp = 5.18 for standard air (would be only slightly higher for actual conditions)

4. Actual rpm = 2,064 (slightly higher for 3.53 in.)
5. Actual bhp of fan = 5.18 (0.0433/0.075) = 3.0
6. Performance at 1,400 ft elevation and 400°F inlet air temperature will be as follows (approximately, this can be improved by applying the Fan Laws to data read from table)

cfm = 6,080
rpm = 2,064 (±)
bhp = 3.0

7. Max. nonoverloaded bhp (from rating table)

$$= 0.591\left(\frac{2{,}064}{1{,}000}\right)^3 = 5.2$$

8. Driver
   This is too hot an installation for V-belt drives. However, they may be used if ventilation is good and perhaps an insulated hot wall is interposed between the sheaves and fan housing. Allowance must be made for belt losses from manufacturers' tables and also any other mechanical losses of the driver. If a motor is used, the shaft output should be 5 hp to cover losses and allow for nonoverload. The 0.2 overload at peak conditions does not justify a 7.5 hp motor because expected operations will be at 3.0, and a 5 hp motor can usually be overloaded 10% without difficulty.

Determine whether any other fan could be used in this application. The next larger or smaller fan size should be examined. Other manufacturers could possibly give a different size that might be more efficient. The final selection should be based on an analysis of several different manufacturers' fans.

Determine the tip speed of this fan. Wheel diameter = 18.25 in.

Tip speed = $(18.25/12)(\pi)(2{,}064) = 9{,}880$ ft/min or from manufacturer's table: tip speed = $4.78(2{,}064) = 9{,}880$ ft/min.

This fan is in Class II according to Figure 12-147 but might be Class I in some other design.

Determine the outlet velocity of this fan. What is the significance of the outlet velocity?

From the manufacturers' table, the outlet velocity is 3,200 ft/min.

When quietness is important, the outlet velocity should be in the range of 1,200 to 2,100 fpm. The low outlet velocity corresponds to low outlet velocity pressure, and this factor directly influences power consumption. The velocity should be kept to a minimum, particularly when the static pressure is low. However, very low outlet velocities (less than 1,000 fpm) are not really desirable, because they produce no advantages, not even quietness. Actual decibel ratings can be attained from the manufacturer, and these are the best indication of actual noise level to be expected.

Example 12-23. Fan Selection Using a Process Gas

A fan is to handle 49,500 cfm (at suction conditions) of a process gas at a suction condition of 120°F and 13.5 psia and is to discharge at 2.5 in. water. The gas density at these suction conditions is 0.085 lb/ft^3.

Because manufacturers' tables are based on standard 0.075 lb/ft^3 air, this density difference must be recognized. According to Fan Law No. 6, if the rpm (speed) and cfm (capacity) are constant, the pressure and hp vary directly as the relative density.

1. Equivalent static pressure $= 2.5\left(\dfrac{0.075}{0.085}\right) = 2.21$ in.

2. Reading a manufacturers' table (not illustrated) shows a fan as follows:

Wheel dia.: 60 in.

$$\text{Max. peak bhp} = 187.6\left(\frac{\text{rpm}}{1{,}000}\right)^3$$

Outlet area = 20.70 in.^{2}
cfm listed = 49,680
Outlet velocity = 2,400 ft/min
Static pressure = 2.25 in. water
bhp = 26.1
rpm = 521

Because the capacity of the fan is slightly over the requirements, interpolation and correction of speed and bhp will not be made (0.3%). However, if desired they could be corrected as given in a previous example.

3. Correct rpm = 521 (not interpolating)
4. Actual cfm = 49,680

5. Actual bhp $= (26.1)\left(\dfrac{0.085}{0.075}\right) = 29.6$ (for gas)

6. Max. peak bhp $= 187.6\left(\dfrac{521}{1{,}000}\right)^3 = 26.4$ (for air)

$$\text{Max. peak bhp} = 26.4\left(\frac{.085}{.075}\right) = 29.9 \text{ (for gas)}$$

This indicates that the selection is operating at near peak condition.

Bard[137] presents an unusual analysis with typical performance curves of the problems associated with various conditions that can develop with parallel fan operation.

Blowers and Exhausters

Blowers and exhausters are centrifugal machines that vary considerably between manufacturers as to performance and physical size. Many have a single wheel with conventional side end inlet and vertical top discharge and range approximately to 16,000 cfm at 90 in. WG static pressure operating at 3,550 rpm. Figure 12-150 illustrates the style of blower/exhauster, and Figure 12-151 is a typical performance curve. Figure 12-152 illustrates multiwheel blower/exhausters. Figure 12-153 shows a different design of impeller for the Turbotron® compressor and exhausters. An assembled unit is illustrated in Figure 12-154 with a range of performance curves, showing pressure to 15 psig, vacuum to 16 in.Hg., no timing gears, pulse free, low vibration, and capacities to 900 cfm.

Nomenclature

A = constant
A_d = cross-section area of duct, ft^2
A_p = cross-section area of piston, in.^{2}
A_r = cross-section area of piston rod, in.^{2}
acfm = actual ft^3/min, actual temperature and pressure
B = constant
bhp = brake horsepower

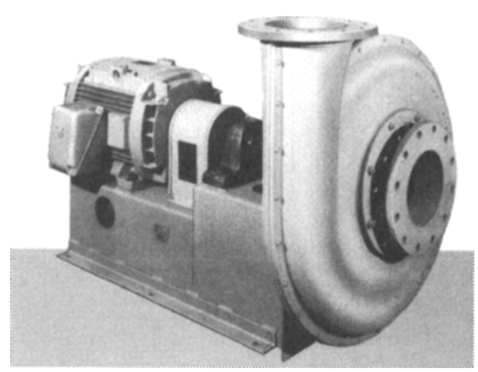

Figure 12-150. This standard arrangement 8 cast iron (C.I.) "E" blower, with motor and coupling factory mounted, is fitted with optional flanged inlet and outlet and coupling guard. (Used by permission: Bul. FI-411C (11/90), ©1988. The Howden Fan Co.)

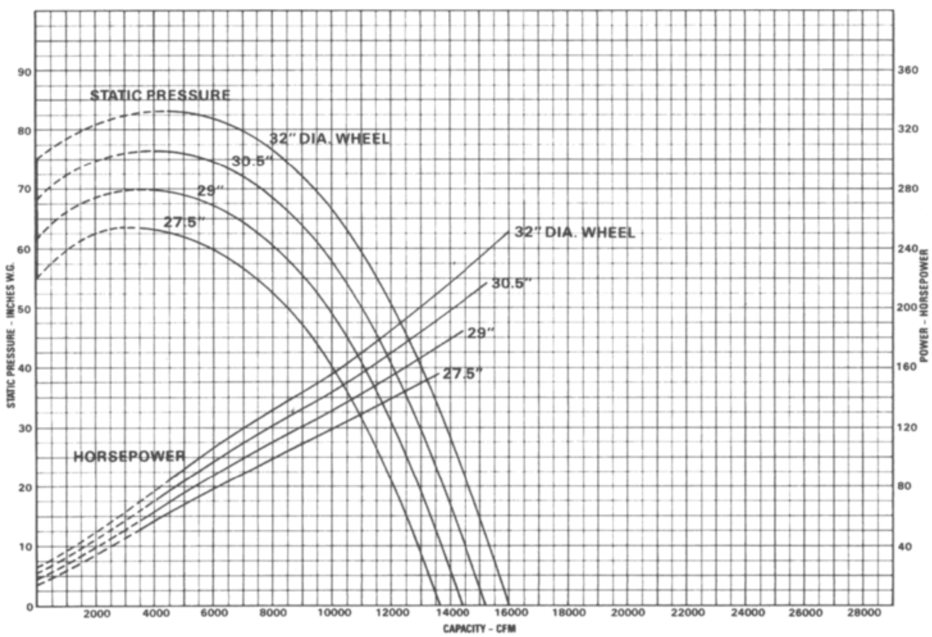

Figure 12-151. Large size 8E blower performance chart, running at 3,570 rpm, up to 32 in. dia. wheel (or impeller), producing static pressure ≅ 90 in. WG at 6,000 cfm, and higher cfm at lower sp. (Used by permission: Bul. FI-411C, ©1988. The Howden Fan Co.)

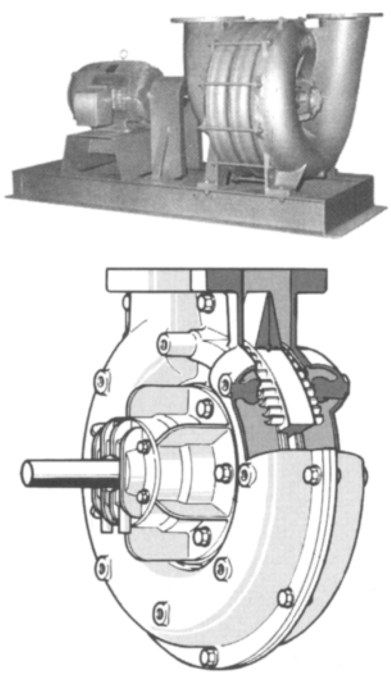

Figure 12-152. Multiwheel blower built in many aspects similarly to a centrifugal compressor—top inlet and top discharge, range to 24,000 cfm, 17 psig discharge, or 16 in. Hg. vacuum. (Used by permission: Bul. 101-2-2, 9/92. ©Lamson Corp.)

Figure 12-153. The Turbotron® impeller represents a new performance approach for air and gas handling in compressors and exhausters. (Used by permission: File No. TBT-2, 9/92. ©Lamson Corp.)

bhp/MM CFD = brake horsepower required/1,000,000 ft^3/day of gas, measured at a base pressure of 14.4 psia and suction temperature

BNV = blower net volume at inlet conditions, ft^3/min

C = capacity of gas to be compressed, referenced to 14.4 psia and suction temperature, ft^3/day

C* = constant

C′ = experimentally determined dynamic loss coefficient

cfr = ft^3/revolution, displacement of blower

c_p = specific heat at constant pressure, Btu/lb(°F)

c_v = specific heat at constant volume, Btu/lb(°F)

c′,c″,c‴ = constants in compression process

D = impeller diameter, in. or fan wheel diameter, in.

D′ = blower displacement, ft^3/revolution

D′ = wheel diameter, ft, or = impeller diameter, in.

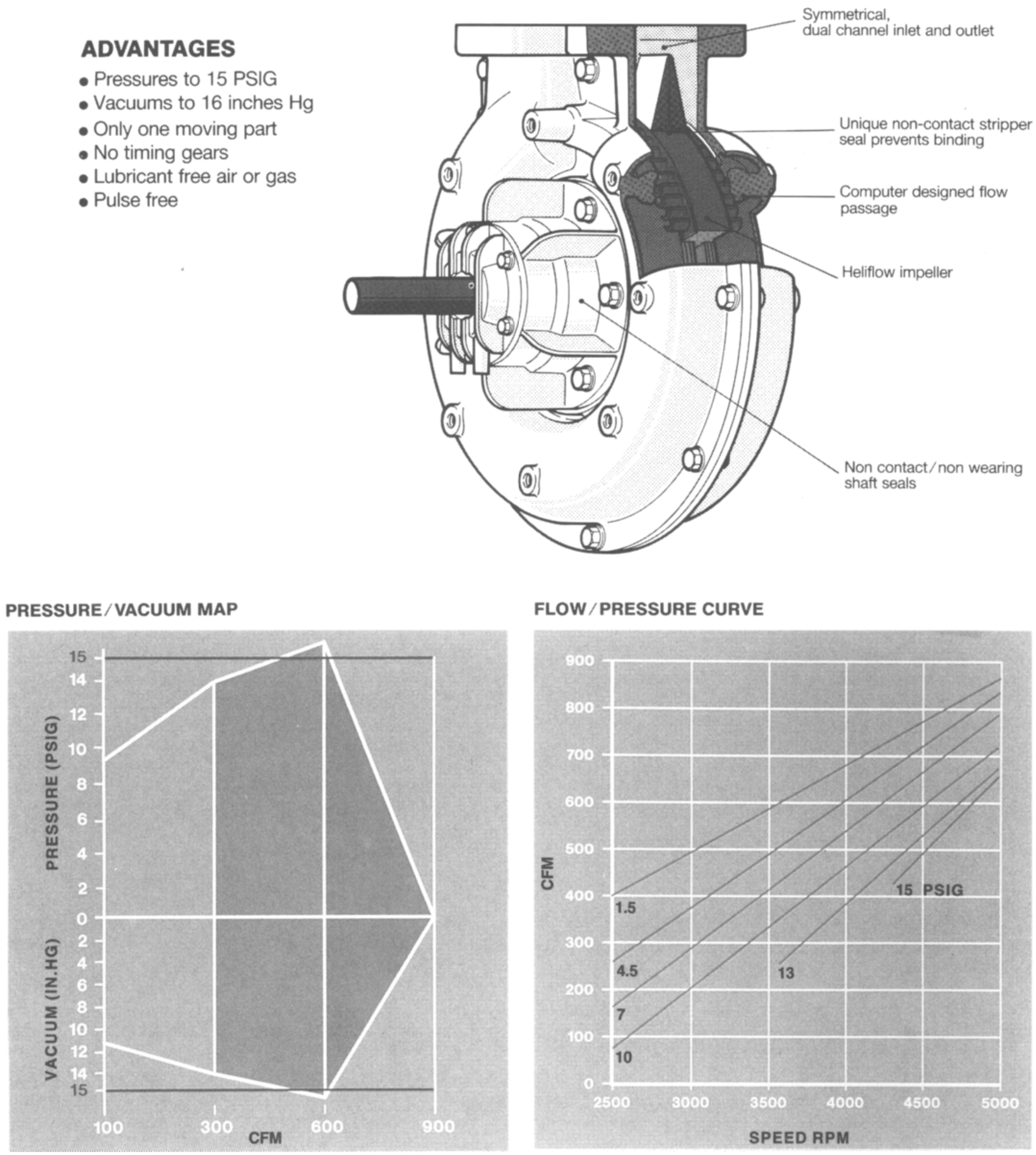

Figure 12-154. Assembled blower with Turbotron® impeller wheel and a range of performance shown on the charts. (Used by permission: File No. TBT-2, 9/92. ©Lamson Corp.)

D_1 = I.D. of duct, ft
db = decibels sound level
E_p = polytropic efficiency, = m/m′
E_v = volumetric efficiency (actual) used as a fraction, but may be calculated in equation as percent
E_v' = theoretical volumetric efficiency, fraction
e_{ad} or e_a = adiabatic efficiency, fraction
e_{ar} = combined adiabatic and reversible compression efficiencies, fraction
e_{as} = adiabatic shaft efficiency, fraction
e_{cm} = combined compression and mechanical efficiencies of a compressor (not including driver), fraction
e_p = polytropic or hydraulic efficiency, fraction
e_s = static efficiency of fan, fraction
e_t = total efficiency of fan, fraction
Fan size ≅ wheel diameter
F_L = frame loss for motor-driven compressors, fraction
F_w = theoretical horsepower factor
f = friction factor dependent upon Reynolds number and relative roughness of the duct, dimensionless
ghp = gas horsepower
g = gravitational constant, 32.2 ft/sec/sec.
H = total head in ft, equal to work of compression in ft-lb/lb
H′ = head/stage, ft of fluid
H_a = adiabatic head, ft-lb/lb, or ft
H_{ad} = adiabatic head, ft
H_p = polytropic head, ft-lb/lb, or ft
$(hp)_a$ = air horsepower
hp_g = gas compression horsepower
$(hp)_t$ = fan horsepower based upon total pressure (or hp)
$(hp)_s$ = fan horsepower based on static pressures (or hp)
h = enthalpy of gas, Btu/lb
h_f = head loss due to friction, ft of fluid flowing
h_{fs} = friction or head loss under standard air conditions, same as h_o
h_o = friction or head loss for actual operating conditions, ft of fluid, or other consistent units
h_s = friction or head loss for standard air conditions, ft of fluid, or other consistent units.
h_v = total shock pressure loss, in. of water
h_1, h_2 = enthalpy of inlet gas and exit gas respectively, Btu/lb
Δh = enthalpy change, Btu/lb
hp = horsepower
ihp = indicated horsepower
i = interstage pressure at stage discharge conditions, psi
J = Joule constant = 778 ft-lb/Btu
K = constant
k = adiabatic exponent, ratio of specific heats, = c_p/c_v
k′ = pseudo-compression coefficient correcting for deviation from ideal gas law
L_o = loss factor, comprised of losses due to pressure drop through friction of piston rings, rod packing, valves, and manifold, Figure 12-19
l = length of duct, ft
M = mass flow, lb/min
MMcf = one million ft^3 gas at 14.7 psia and 60°F.
M′ = Mach number, dimensionless
M_{cp} = molal heat capacity at constant pressure, Btu/lb-mol(°R)
MW = molecular weight
m = isentropic or adiabatic exponent
m′ = polytropic exponent, or n = polytropic exponent
N = number of lb-mol, or rotational speed, revolutions/min, rpm
N_m = number of lb-mol gas/hr
N_s = specific speed, dimensionless
n = polytropic exponent or coefficient for compression or expansion, or n = rpm, speed in Fan Laws
P or p = pressure, absolute, psia, or inlet pressure, psia
P_c = critical pressure, absolute
P_f = final pressure of multistage set of cylinders
P_r = reduced pressure = P/P_c
P_{s2} = fan outlet static pressure, in. water abs
P_t = total system pressure psia
P_v = fan outlet velocity pressure, in. water abs
P_v'' = vapor pressure of moisture in saturated gas, psia
P_1 = atmospheric pressure or fan inlet pressure (if not atmospheric), in. water abs, or initial suction condition, abs
P_2 = discharge pressure, abs
PD = piston displacement, ft^3/min
PD′ = piston displacement, $in.^3$
psi = pounds per $in.^2$
p = static or total pressure, in. of water
p′ = pressure, lb/ft^2, abs
Prime (′) = interstage discharge condition, reduced by the pressure drop through the intercoolers, valves, piping, etc., represents *actual* pressure to suction of successive cylinders in multistage compression system
p_s = fan static pressure, in. water
P_{fs} = p_s = fan static pressure, in. water
p_{ft} = fan total pressure, in. water
p_{fv} = fan velocity pressure, in. water
p_t = total pressure, in. water
p_v = velocity pressure, in. water
Δp = pressure drop through intercooler, psi
Q or Q_s = inlet (or volume) flow of gas (or suction), ft^3/min, or ft^3/sec (as marked), also = Q_1, or = heat
Q_a or Q_b = fan capacity, ft^3/min (cfm)
R = universal gas constant = 1,545 ft-lb_{Force}/(lb-mol)(°R), units depend upon units of pressure, volume, and temperature = 1.986(Btu)/(lb-mol)(°R)
R_i, or R′ = specific gas constant for gas involved, = 1,545/mol wt
R_o = universal gas constant = 1,545 = same for all gases using P = lb/ft^2abs, or = 10.729 when P = lb-$in.^2$ abs (see "R")
R_c = ratio of compression across a single cylinder, or total compressor
rpm = revolutions per minute, or = n in Fan Laws
RH = relative humidity, fraction
R_t = overall ratio of compression across multistage unit = P_f/P_1
r = radius of elbow, in.
S = stroke, ft
= entropy, Btu/lb
S′ = slip rpm for rotary positive blower
S_p = slip rpm at a specified discharge pressure for rotary positive blower.
scfm = standard ft^3/min at 14.7 psia, 68°F and 36% relative humidity

shp = shaft horsepower input to fan, or compressor
SP, or sp = static pressure, in. of water (or in. water gage)
s = stroke length, in.
T = temperature, abs, °R = °F + 460
T_c = critical temperature, °R
T_1 = air or gas temperature at fan inlet, °Rankine
T_r = pseudo-reduced temperature
= °R/[(mol avg T_c (abs)] = °R/°T_c
t = temperature, °F
t_c = temperature after adiabatic compression, °F
t_w = water temperature
u = peripheral velocity of wheel, ft-sec
u′ = gas velocity at any point, ft/sec
V = volume, ft^3 (Note: may be ft^3/min, when stated)
VP or vp = velocity pressure, in. of water
V_a = actual capacity or actual delivery, referenced to intake or suction conditions of cylinder, ft^3/min
V_c = clearance volume, $in.^3$
V_d = volume of dry gas, cfm
V_p = peripheral velocity of fan wheel, ft-min
V_{pc} = % clearance or used as a fraction
V_{pc}' = fraction clearance = % clearance/100
V_s = sonic or acoustic velocity, ft/sec
V_s' = slip, cfm
V_w = volume of gas containing moisture or other condensable vapor, cfm
V_1 = suction or inlet volume, ft^3/min
v = specific volume, ft^3/lb
v_f = fluid velocity, ft/sec
v_m = fluid velocity, ft-min, or = v
W or w = mass flow of gas, lb/min or = power
w = vapor pressure of water at temperature, psia
y = mol fraction of a component in the vapor phase
Z = compressibility factor, dimensionless

Greek Symbols

α = blade angle, with plane of rotation, degrees
γ = number of compression stages, or specific weight of gas, lb/ft^3, or isentropic coefficient
Φ or ϕ = flow coefficient
Ψ = pressure or head coefficient
μ = pressure or head coefficient
δ = relative density of gas referred to air at standard conditions
Δ = change, interval
ν = pressure coefficient, ft/sec
υ = tip speed of impeller or rotor wheel, ft/ sec
ν = gas velocity, ft/sec
η = total fan efficiency
ρ = density, lb/ft^3
ρ_g = gas density, lb/ft^3
ρ_o = density of air under actual operating conditions, lb/ft^3
ρ_s = density of air under standard conditions, lb/ft^3
π = 3.1416

Subscripts

1 = first condition
2 = second condition
ad = adiabatic
f = final or last stage
d = discharge condition
i = interstage condition
ib = fan input power
m = mean value
p = polytropic
s = suction condition, or = static
T or t = total
y = conditions of gas across a cylinder, represented by 1 for first stage, 2 for second stage, etc.

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Chapter 13

Reciprocating Compression Surge Drums

Proper performance of a reciprocating compressor cylinder depends upon the required quantity and condition of gas available at the suction flange of the cylinder. At the same time, conditions must be satisfactory for proper removal of the compressed gas without back pressure effects, gas pulsation, vibration, and/or noise. Many installations have failed to perform satisfactorily, as evidenced by excessive horsepower consumption and overall adjacent system piping vibration, due to inadequate pipe sizes, gas pulsations, and/or gas surge volumes. Conventional pipe sizing does not recognize the significant changes in gas flow rate that take place as a compressor piston moves from head-end to crank-end or vice versa.

It is beyond the scope of this chapter to provide detailed design procedures for systems that may be required to prevent or reduce gas pulsations in a reciprocating compressor system. However, this chapter does attempt to (1) alert the design engineer that the topic does need to be addressed, (2) provide preliminary design methods for parts of the system design, and (3) direct the reader to references for the most effective design approach known at this writing.

A complete evaluation of the surge capacity and pulsation frequencies of the system (i.e., suction header and suction surge drum, Figure 13-1A), compressor, discharge surge drum, and discharge header is necessary before reaching an understanding of the performance of the compressor in this system. The only known effort in this direction is the work of the Southern Gas Association's compressor analog computer at Southwest Research Institute, a study by the National Advisory Committee for Aeronautics (now NASA)[4] on mufflers, the American Petroleum Institute Standard 618, paragraph 3.3.

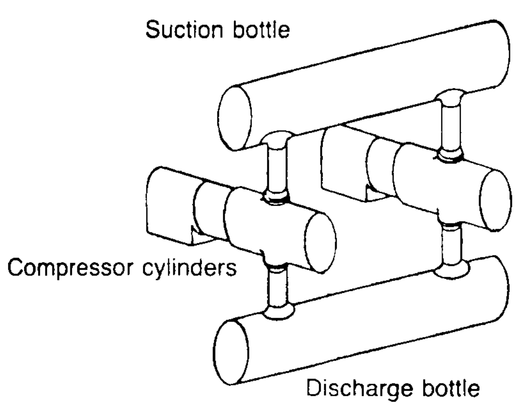


Figure 13-1A. Parallel acting compressor cylinders with common suction and discharge surge bottles or drums.

More recently, the Southern Gas Association's Gas Machinery Research Council and Pipeline and Compressor Research Council cooperated with Southwest Research Institute to develop a software package that "enhances reciprocating compressor operation."[17] Several major American and foreign compressor manufacturers (and some other interested companies) have cooperated with Southwest Research Institute or developed their own proprietary pulsation-reduction design computer programs.

An excellent review and evaluation of three methods of pulsation and surge control is given in Reference 10. Before attempting to evaluate or design a system, a thorough understanding of what takes place during gas compression is necessary, as well as an understanding of the method's capabilities.

Pulsations may cause

1. Vibration of the compressor, pipe, and vessels, even to the extreme of causing mechanical and foundation failure.
2. Design of oversized piping to overcome the higher pressure drop induced by pulsative flow.
3. Lower compressor efficiency by requiring more horsepower to overcome the higher internal pressure peaks.
4. Piping vibration resulting in fatigue failure.
5. Mechanical damage to other equipment in the system.
6. Erratic flow through meters resulting in poor accuracy.
7. Induced noise throughout the piping system.
8. Overstressed piston rods due to unbalanced or dynamic forces.

Pulsation Dampener or Surge Drum

"A pulsation dampener is an acoustic filter designed for minimum transmission of all the frequencies generated as compressor or engine speed varies from minimum to maximum."[10] Actually, this cannot be achieved; however, units can be selected to transmit low frequencies up to a cut-off frequency with little if any reduction in power. They undergo attenuation of frequencies from the cut-off region and up to a band-pass of high transmission where some frequencies may completely pass through the unit. The pass band frequencies develop from the resonance of the surge drum system.[20]

Duhe, Eckhardt, and Smalley[23] have analyzed the reciprocating compressor pulsation drum sizing recommendations of the American Petroleum Institute Standard No. 618 and compared its results for pulsation bottle/drum design with the results from the Southwest Research Institute's digital computer design program, which has been used worldwide to analyze reciprocating compressor systems and bottle/drum sizing. The results suggest the following:[23]

1. The API estimation formulae are useful approximations but need some refinement.
2. Insufficient information is currently available to propose a change in design methods.
3. Two changes were proposed—one for each of the suction and discharge bottle design equations.

At present, the API procedure is suggested by the authors as a reasonable approximation technique but is not as thorough nor as specific as the design computer technique of Southwest Research Institute.[17, 18]

Common Design Terminology

- *Volume Bottles:* Connected directly to compressor cylinders, volume bottles are empty vessels free of internal mechanical components. Usually these simple vessels do not properly reduce gas pulsations. For a pulsation-induced vibration of "f" cycles per second, harmonics at 2–3 times "f" can be expected.
- *Low Pass Filter:* Two chambers are connected by a "choke tube" or pipe. At least one of the chambers should be connected to the compressor cylinder, Figures 13-1B and 13-1C. Melton[8] offers a standardized, simple design that is useful for some types of package compressor arrangement.

A practical unit for good performance must have a low frequency cut-off below the lowest significant frequency being generated and must not have pass bands greater than the cut-off frequency, which coincides with significant har-

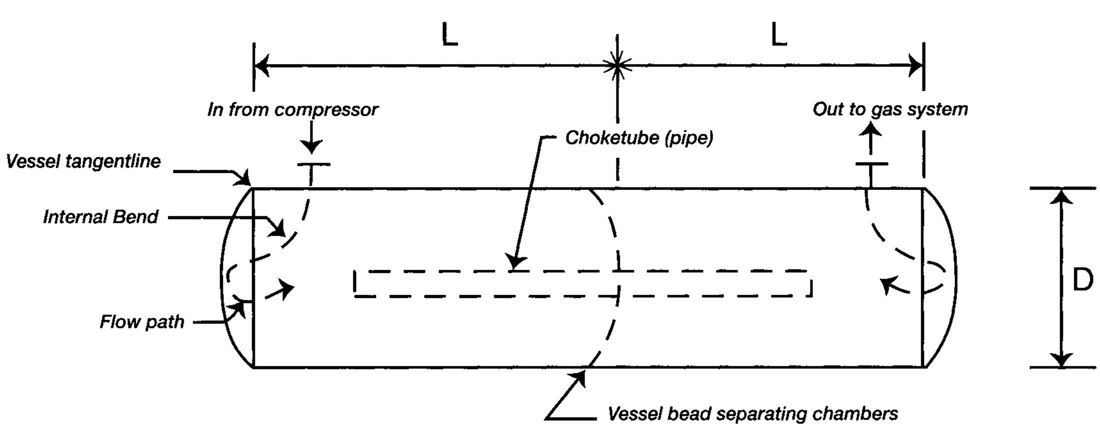


Figure 13-1B. Low pass filter. Two equal (usually) surge chambers connected by a "choke-tube." This type of filter does not have to be a common vessel as shown here. See Figure 13-1C.

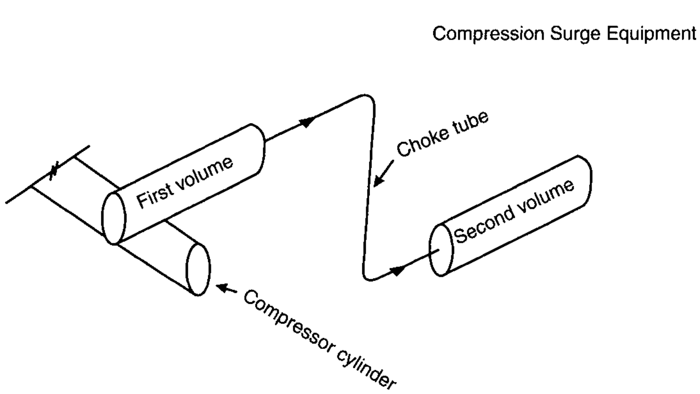


Figure 13-1C. Pulsation dampener schematic with "choke-tube." See Example 13-3.

monics of the fundamental frequency being generated.[10] The surge drum must alternately store energy from the various frequencies to be removed or reduced and then release that energy in a manner that will present a relatively smooth and continuous flow from the system.

In many instances, the design of suction and discharge pulsation dampening drums (or bottles) for reciprocating compressors is based on piston displacement and volumetric efficiency, and this design normally will suffice to reduce peak pulsation to approximately 5% of the line pressure. In special or other cases, experience has shown that operational difficulties (vibrations, meter pulsations, etc.) may indicate that the peak pulse pressure of 5% line pressure is inadequate. Thus, the pressure in pulsation-reduction design selection is

1. Surge bottles (drums), shown in Figure 13-1A, designed by using piston displacement and volumetric efficiency to 5% or lower of peak pulsations. The bottles designed by this method can become quite large, because the 5% is reduced to 3% or 2%, and the limitation becomes physical and economic. This is a nonacoustical design.
2. Filter systems designed by acoustical principles, Figure 13-1D and 13-1E.
3. Piping and filter systems designed by acoustical principles, using a simulation technique on an analog computer.

In practice, the only positive way to evaluate performance of a unit is to install it in the compressor system and observe its operation relative to noticeable system vibration, compressor performance, and measured frequency information. More recent (circa 1996) developments[10, 18] have continued to use the earlier analog techniques,[14] and digital computer programs now allow an evaluation of the system prior to fabrication and erection and enable the designer to readjust parameters to effect a harmonious operating compressor-piping system. The piping arrangement (and size) and its *anchor* points are important in a well-designed system. Whenever possible, this system approach is preferred to an individual surge drum design only.

It is important to note that all designers do not agree that the problem of pulsation is acoustical in nature, and hence the approach to solving the problems will necessarily be different.

Use a specification sheet (Figure 13-2) to enquire system design information from a design service or a qualified engineering concern.

A general appreciation of some of the effect of gas pulsation on the performance and impact on the compressor pulsation drums, their nozzles, the piping system, and the cylinder valve performance, as well as possible other effects in some unique systems or items of equipment, can be discovered by examining the cylinder performance using an indicator card. This examination (see Chapter 12) can reveal acceptable and unacceptable performance in terms of pressure variations within the cylinder as the piston passes through its cycle. Hicks[25] presents a helpful analysis; see Figures 13-3 and 13-4.

The pulsations can cause the use of excess horsepower when compared to the ideal or a system design that reduces pulsations and thereby improves cylinder performance and efficiency. The pulsation shaking forces in the suction and discharge dampeners (bottles) can be evaluated by computer analysis, and the magnitude and frequency in hertz can be reduced to an acceptable level by adjusting the dimensions (size) of the dampeners.[18] The magnitude of the internal forces directly affects the mechanical stress on the nozzles of the cylinder and of the dampeners. Compressor

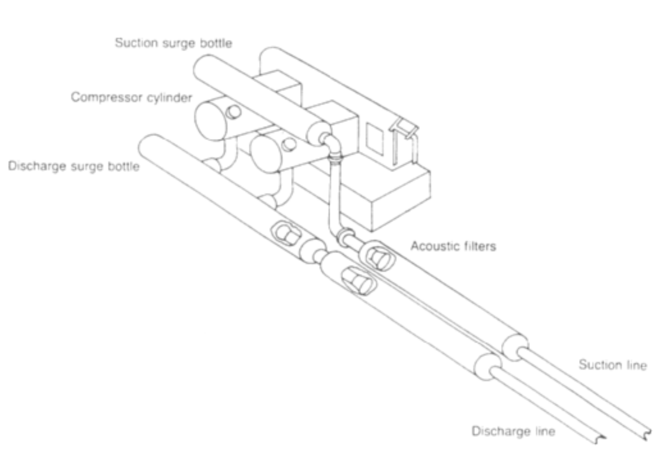


Figure 13-1D. Compressor installation with surge bottles and acoustic filters.

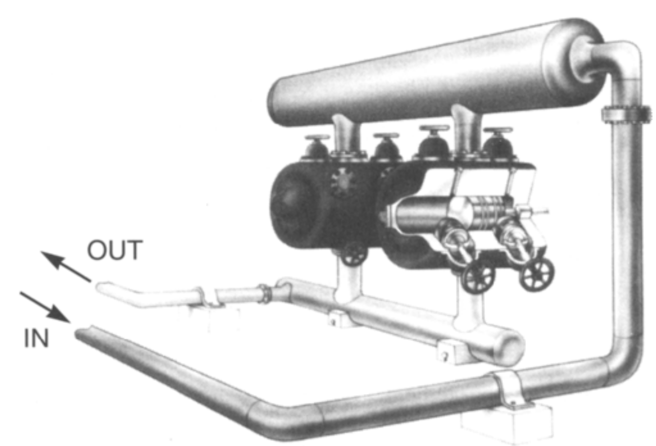


Figure 13-1E. A computer facility performs dynamic simulation and analysis of reciprocating compressor installations such as this two-cylinder compressor. Interactive pulsation and mechanical analysis ensures trouble-free operation. (Used by permission: Southern Gas Association's Gas Machinery Research Council.)

performance can be negatively affected by poor performance, such as leaks, poor seating of valves, piston rod leaks, rod seal leaks, heat transfer, and incorrectly assumed cylinder clearances (loading);[18] see Chapter 12. It is necessary to distinguish pulsation effects and compressor valve problems, such as losses and flutter. Note that valve flutter occurs when the flow is insufficient to force the valve full open or to the stop position during the full piston cycle. Figures 13-5 and 13-6A–C illustrate the effects of pulsation on valve behavior.

Job Reference ____
Installation for ____ Plant Location ____
Make and Model Compressors ____ No. of Compressors ____
Power Cyls: Bore & Stroke ____ No. of Cyls. ____ RPM ____
COMPRESSOR SPECIFICATIONS
1st STAGE
Bore ____ Stroke ____ No. of Cyls. ____ Disp. CFM ____ S.A. ____ D.A. ____
If Multiple Cyls.: Crank Angles? ____ Clearance ____ %
2nd STAGE
Bore ____ Stroke ____ No. of Cyls. ____ Disp. CFM ____ S.A. ____ D.A. ____
If Multiple Cyls.: Crank Angles? ____ Clearance ____ %
3rd STAGE
Bore ____ Stroke ____ No. of Cyls. ____ Disp. CFM ____ S.A. ____ D.A. ____
If Multiple Cyls.: Crank Angles? ____ Clearance ____ %
4th STAGE
Bore ____ Stroke ____ No. of Cyls. ____ Disp. CFM ____ S.A. ____ D.A. ____
If Multiple Cyls.: Crank Angles? ____ Clearance ____ %
OPERATING CONDITIONS, EACH COMPRESSOR: (Standard Conditions 14.7 psia and 60°F.)
Type of Gas ____ Corrosive? ____
Moisture Included in Flow Rate [Yes] [No], Saturated at ____.

1st STAGE ____ SCFD or SCFM
Intake ____ CFM at ____ psia and ____ °F.
Discharge ____ CFM at ____ psia and ____ °F.
Molecular Wt. ____ Specific Gravity ____ Cp/Cv ____

2nd STAGE ____ SCFD or SCFM
Intake ____ CFM at ____ psia and ____ °F.
Discharge ____ CFM at ____ psia and ____ °F.
Molecular Wt. ____ Specific Gravity ____ Cp/Cv ____

3rd STAGE ____ SCFD or SCFM
Intake ____ CFM at ____ psia and ____ °F.
Discharge ____ CFM at ____ psia and ____ °F.
Molecular Wt. ____ Specific Gravity ____ Cp/Cv ____

4th STAGE ____ SCFD or SCFM
Intake ____ CFM at ____ psia and ____ °F.
Discharge ____ CFM at ____ psia and ____ °F.
Molecular Wt. ____ Specific Gravity ____ Cp/Cv ____

Figure 13-2. Gas surge drum inquiry specifications. (Used and adapted by permission: Burgess-Manning, Subsidiary of Nitram Energy, Inc.)

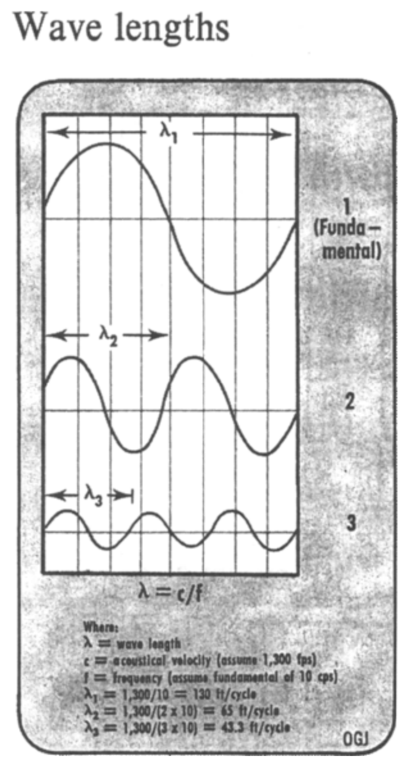


Figure 13-4. Wave lengths for the frequencies shown in Figure 13-3, assuming an acoustical velocity of 1,300 fps. (Used by permission: Hicks, E. J. *Oil and Gas Journal,* p. 38, July 24, 1978. ©PennWell Publishing Company. All rights reserved.)

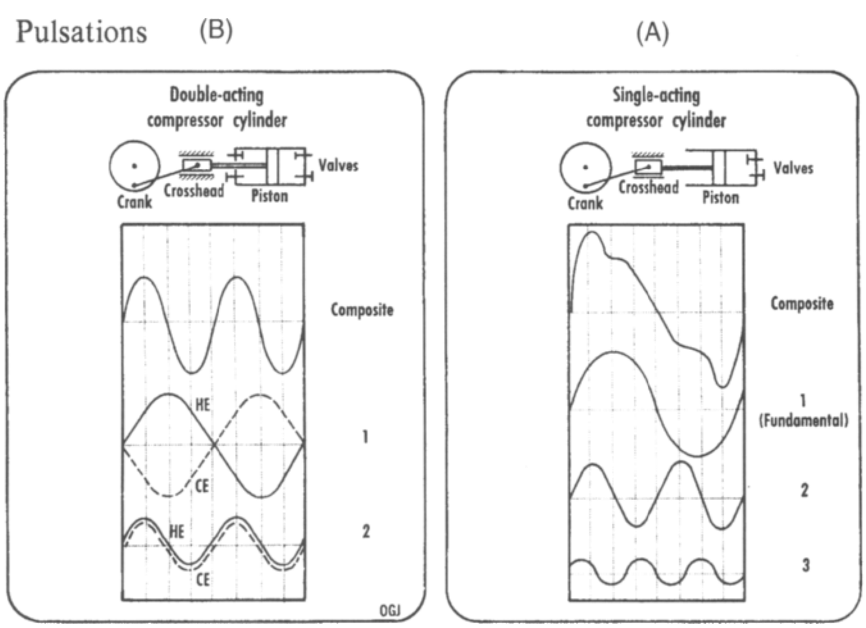


Figure 13-3. Typical pressure wave in the suction and discharge piping shown as composite of (A) waves that are multiples of fundamental sine wave, (B) waves (1), (2), and (3) representing multiple frequencies of pulsations that act as exciting forces for vibration. Note: Vertical axis is pressure. (Used by permission: Hicks, E. J. *Oil and Gas Journal,* p. 38, July 24, 1978. ©PennWell Publishing Company. All rights reserved.)

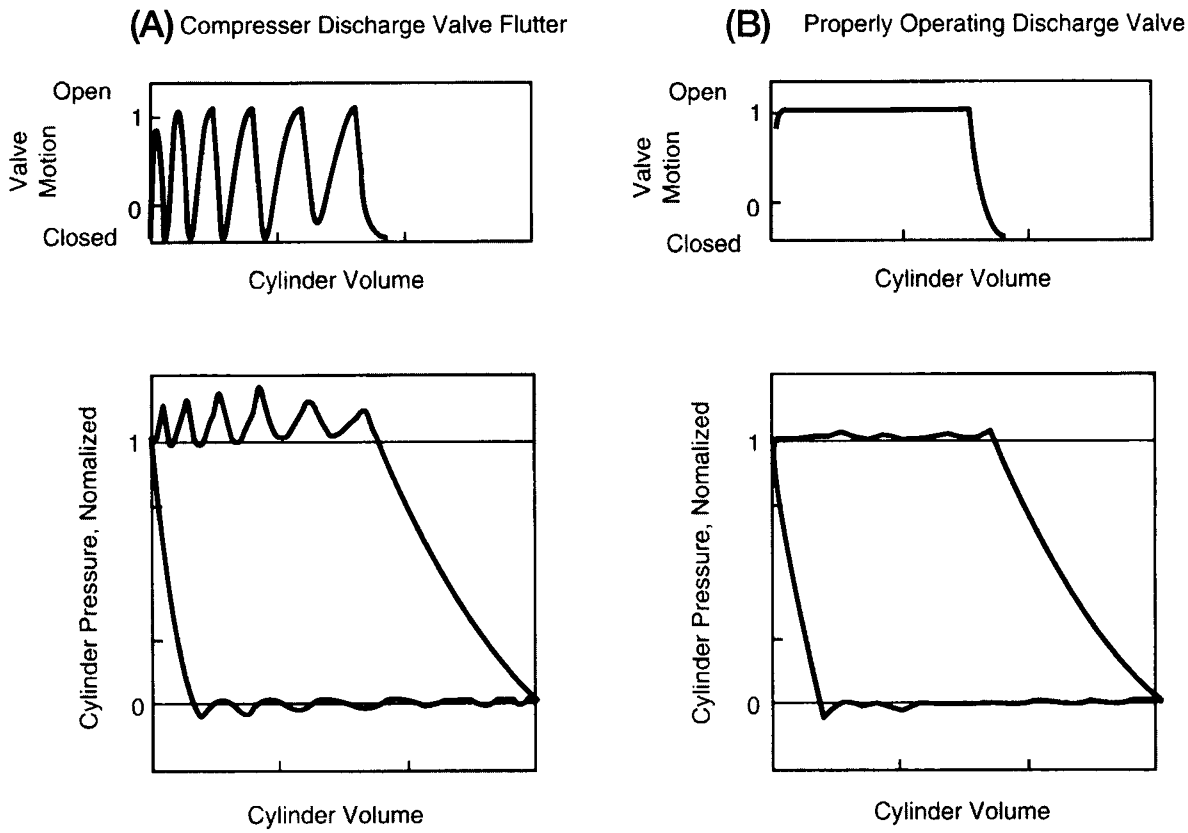


Figure 13-5. Compressor valve diagnostics and optimization procedure: (A) compressor discharge valve flutter and (B) properly operating discharge valve. (Used by permission: Southern Gas Association's Gas Machinery Research Council.)

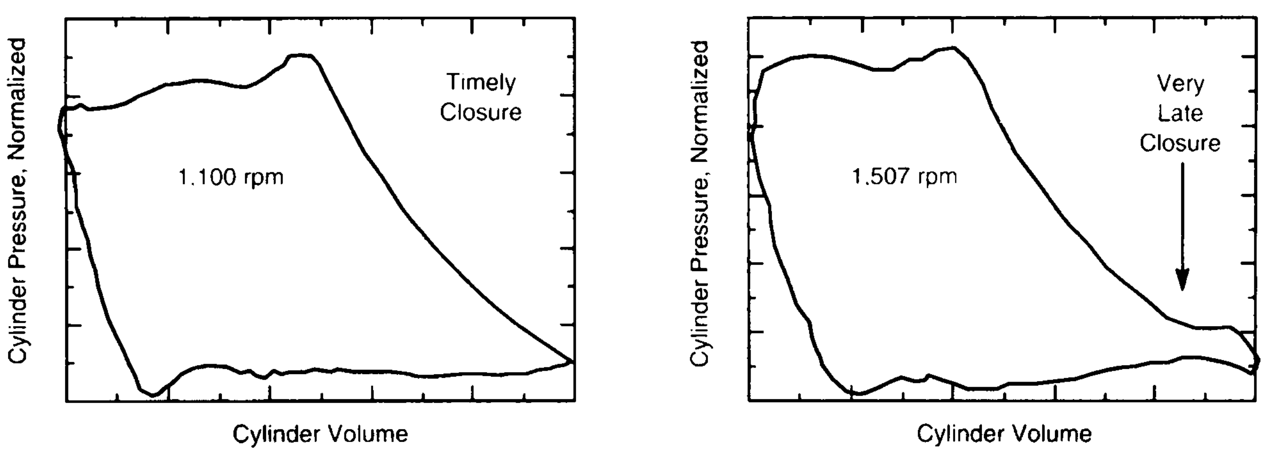


Figure 13-6A. Influence of pulsations on valve behavior. (Used by permission: "Compressor and Piping System Simulation," Southern Gas Association's Gas Machinery Research Council.)

Applications

The control of acoustic resonances should always be considered in piping design. That is, acoustic resonant frequencies of piping elements should be separated from the frequencies of prominent engine harmonics as much as possible.[13–16] Acoustic resonance control is important in compressor manifold piping in which quite high amplitude dynamic unbalanced forces can exist because of acoustic resonances, and in choke tubes, laterals, and by-pass stubs in which a half or quarter-wave acoustic resonance can cause excessive acoustic wave amplitude.

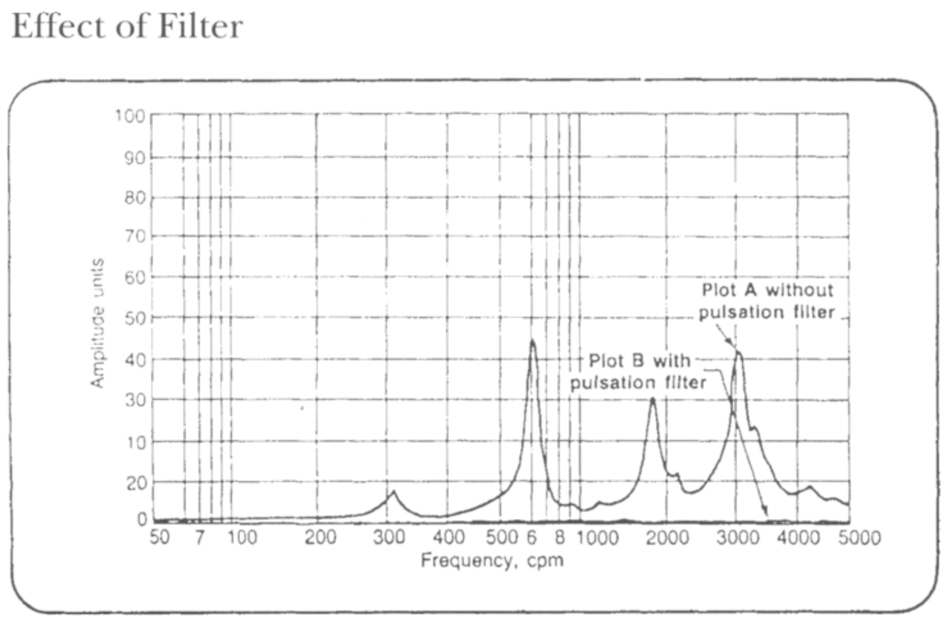


Figure 13-6B. Effect of pulsation filter on gas compression pulsations at orifice meter from 200–550 psig. (Used by permission: von Nimitz, W. W., and O. Flanigan, *Oil and Gas Journal,* p. 68, Sept. 8, 1980. ©PennWell Publishing Company. All rights reserved.)

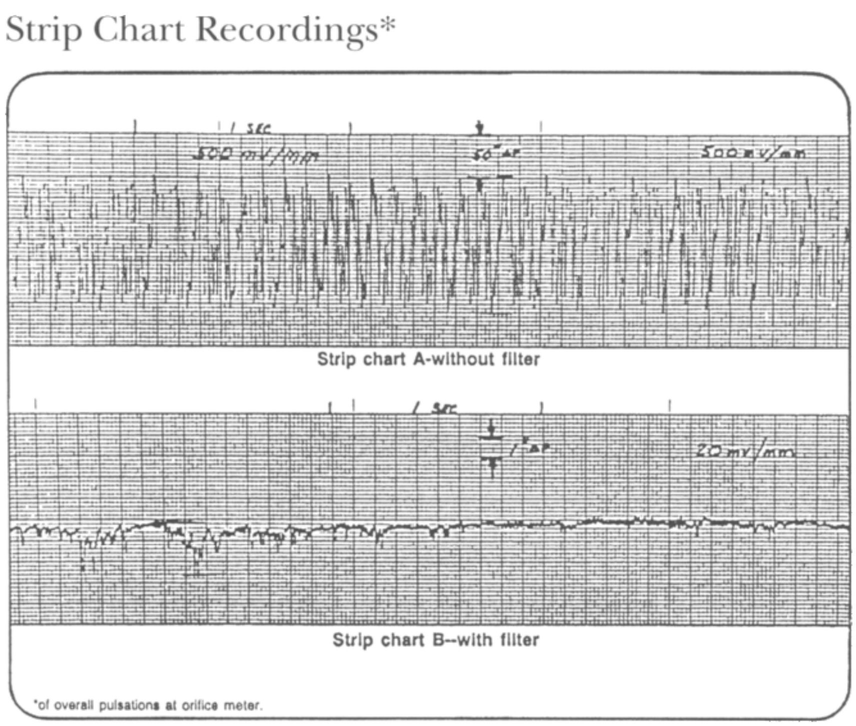


Figure 13-6C. Effects of pulsation filter on orifice flow meter charts at 300 rpm compressor speed. (A) Before peak-to-peak differential pulsations were 160 psi, and (B) after installation of filter pulsation, levels dropped to 1.5 psi. (Used by permission: von Nimitz, W. W., and O. Flanigan, *Oil and Gas Journal,* p. 60, Sept. 8, 1980. ©PennWell Publishing Company. All rights reserved.)

The two most-often used techniques for suppressing unbalanced forces, particularly in compressor manifolds, are center feeding of the suppression bottle and using multichamber bottles (Figures 13-7A, 13-7B, and 13-1C) and flow direction reversal.

For the application and design of a reduced pulsation-vibration system, the acoustic computer technique developed by Southwest Research Institute for the Gas Machinery Research Council in cooperation with the Pipeline and Compressor Research Council is considered the most prominent technique available through public and commercial institutions. This technique is recommended as the most reliable over manual and graphical techniques for creating a final detailed design of a surge drum or acoustic filter and system piping analysis and reciprocating valve performances.[17, 18] Figures 13-6B and 13-6C illustrate the effectiveness of the pulsation filter system using actual field data and the beneficial effects of the design of the acoustic filter using the earlier[10, 14] Southern Gas Association/Southwest Research analog computer system, which has now been

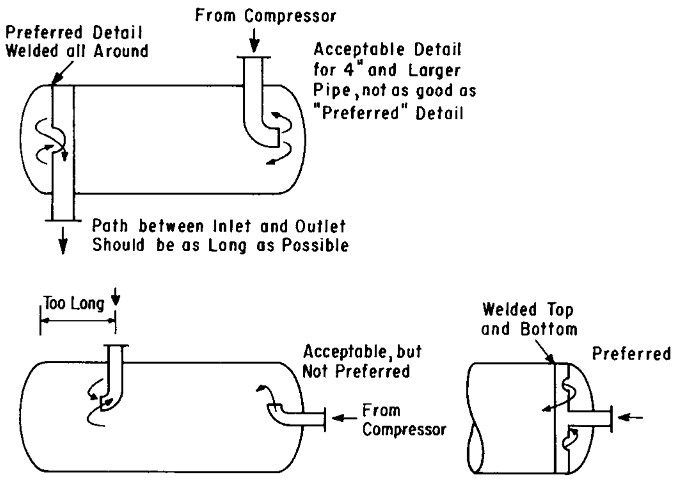


Figure 13-7A. Surge drum internal details. Note: The design indicated may not correspond to the internal details recommended by a professional computer analysis design. This diagram represents a concept suitable for some applications.

superseded by a digital system. An actual gas orifice meter was used to represent the pulsation impact in the figures.

If pulsations leave the surge or suppression bottles, which are usually located right at the suction and discharge of reciprocating compressors, force pulsations create vibrations (mechanical) in the piping systems, which can lead to fatigue and instrument control and metering problems. The intent of the control is to limit vibrations, although a direct relation between the overall pulsation levels and the vibration they might produce does not necessarily exist. The highest vibration levels may not correspond to the highest pulsation peaks at a given frequency.

As von Nimitz points out,[13] only cyclic stresses are directly related to failure probability. These stresses are often produced by pulsations in the fluid system, by mechanical vibrations produced by the mechanical movement of certain equipment components, and as a result of the fluid pulsations. Figure 13-8 lists the sequence of events that leads to most failures of equipment and piping.

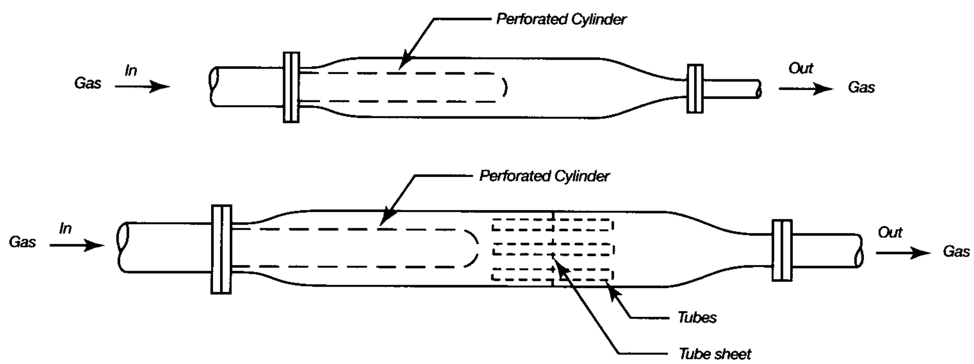


Figure 13-7B. Alternate surge drum internal details.

Pulsations	Generated by compressor or pump action with the amplitudes controlled by the location of acoustic resonances and damping in pulsation filters and piping system.
Acoustical Shaking Forces	Produced by coupling of energy at bends, closed valves or capped ends, restrictions, and similar points in the system with levels dependent on the degree of coupling.
Vibrations	Levels dependent on location of mechanical resonances relative to the frequency of shaking forces, point of coupling relative to the mode shape and amount of damping.
Cyclic Stresses	Strains due to vibration of components relative to each other or to supports result in stresses the levels of which are also dependent on stress intensification factors present.
Failure Probability	Probability of fatigue failures determined not only on the basis of calculated cyclic stresses but also their cumulative effects during projected installation life.

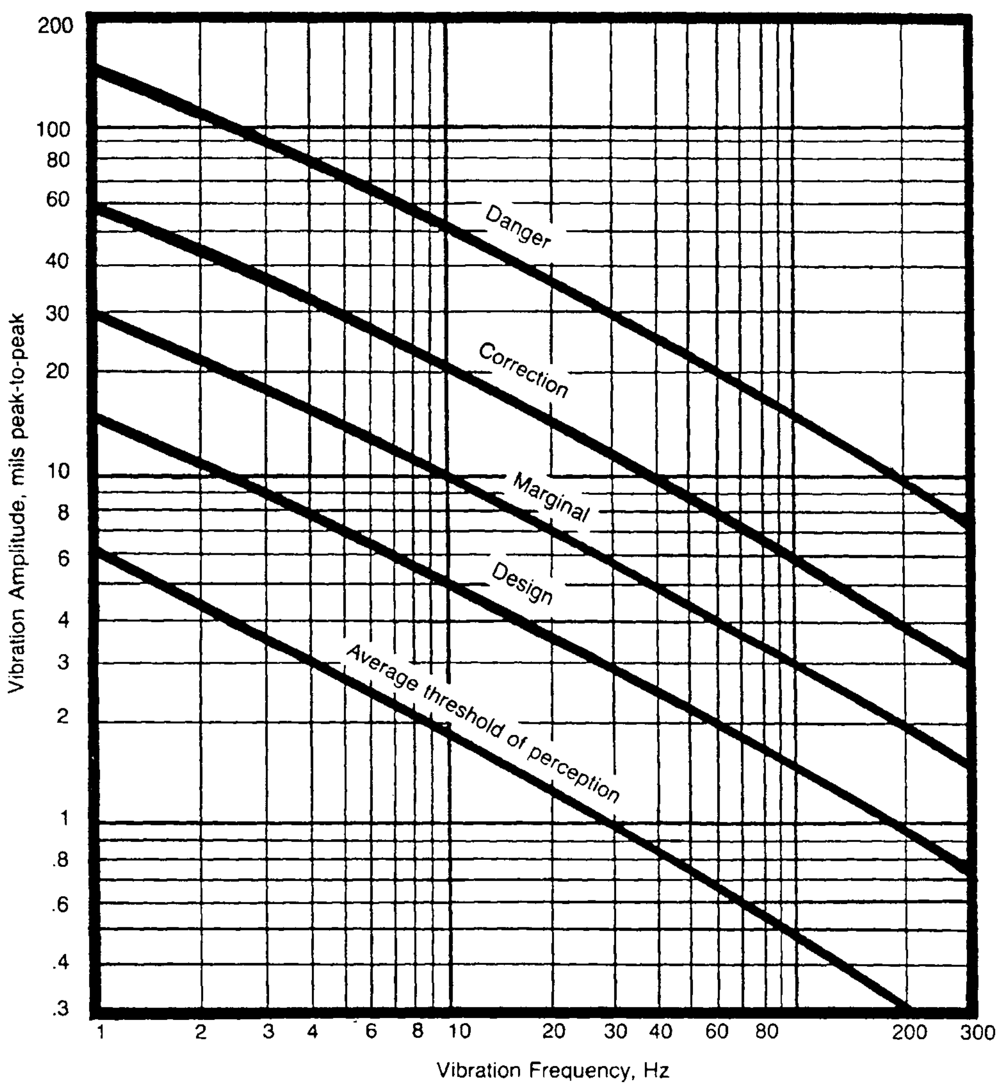
Figure 13-8. Relationship of pulsations to vibrations to cycle stresses. (Used by permission: von Nimitz, W. W. Lecture of Reference 13, Part 1, Table 1, proceedings of the 1974 Purdue Compressor Technology Conference.)

Vibration control is one of the key objectives behind any pulsation suppression. Therefore, the end result of much effort is to reduce the magnitude of the measured mechanical vibration. Figure 13-9 provides "frequency-variable allowable vibration level criteria." A typical fixed maximum peak-to-peak vibration movement of 8 mil is a reasonable reference; however, for many critical applications, a vibration of 2–4 mil is all that can safely be tolerated, Figure 13-10.

For compressor horsepower less than 500, the suggested design techniques for surge bottle design included here can often be satisfactory; however, due to the wide variations in the equipment and system arrangements, no real assurance can be given. For equipment greater than 500 hp, and even for critical applications at lower horsepower, the SGA (Southern Gas Association) Compressor Design System is the best available technology for system analysis and design. Refer to the work of von Nimitz.[13–16, 17, 18]

The more important techniques for controlling acoustic resonance are selected depending on the required purpose.[13–16, 17, 18]

1. Surge bottles (drums) for compressor's cylinders.
2. Addition of acoustic filtering.



Note: Indicated vibration limits are for average piping systems constructed in accordance with good engineering practices. Make additional allowances for critical applications, unreinforced branch connections, etc.

Figure 13-9. Frequency-variable allowable vibration level criteria. (Used by permission: von Nimitz, W. W. From Reference 13, proceedings of the 1974 Purdue Compressor Technology Conference.)

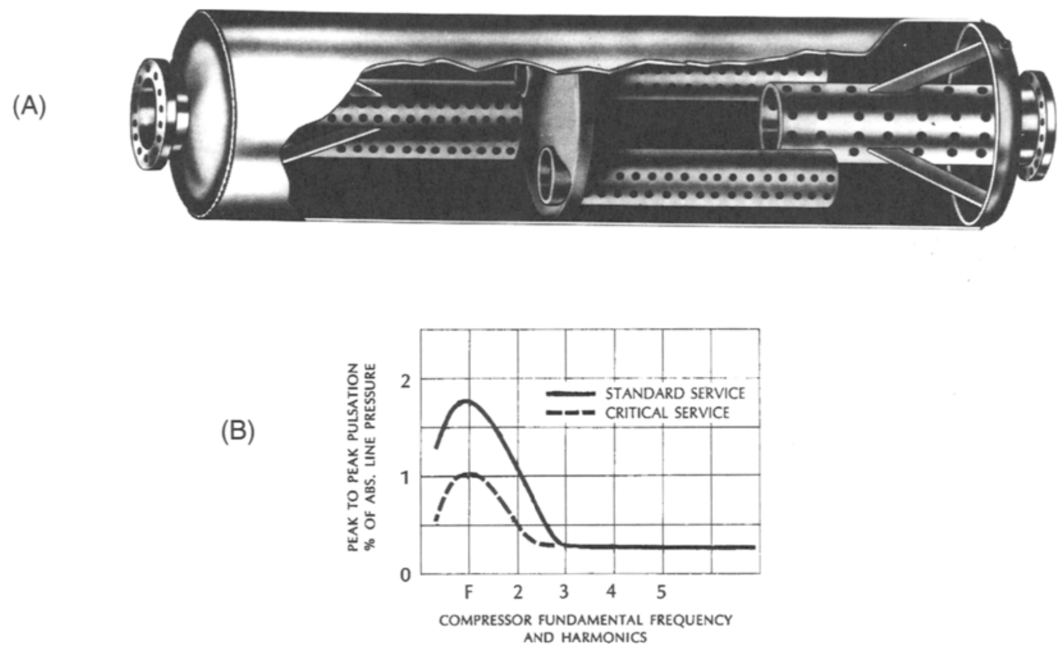


Figure 13-10. Commercial gas pulsation snubber showing (A) typical internal arrangement and (B) typical pulsation performance. (Used by permission: Bul. 20-1-2. Burgess-Manning, Inc., subsidiary of Nitram Energy, Inc.)

3. Variation of piping lengths or the use of baffles and choke-tubes in surge bottles.
4. Use of side branch resonators.
5. Use of dissipative components, such as orifice plates, perforations, absorbing walls, etc.
6. Compressor valve problems:[18]
   a. Excessive losses.
   b. Flutter.

Surge Drums

Generally, when placed close to the compressor cylinder, surge drums will minimize acoustic wave amplitudes, but they do not eliminate high-frequency acoustic response (Figures 13-11 and 13-12).

Acoustic Filters

Properly designed, they effectively can eliminate the transmission of high-frequency response. The filter should prevent the transmission of all acoustic response with as little restriction to steady flow as possible. Filters can be designed for almost any desired degree of acoustic response control.

Cut-Off Frequency Formula

This technique is simple, provides rapid calculation of the lowest frequency at which significant dampening begins,

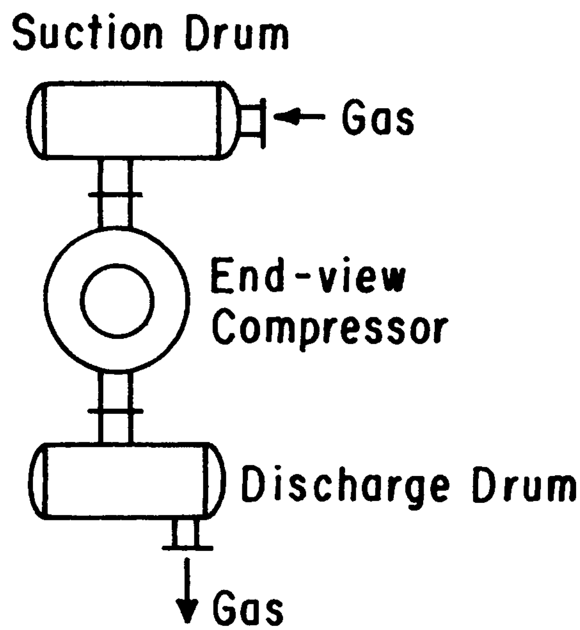


Figure 13-11. Single compressor cylinder with surge drums.

and allows rapid determination of pipe size to cause dampening.

The disadvantages are that the low-frequency cut-off is usually defined as the frequency above which more than 90% dampening occurs, but the results may be in error by

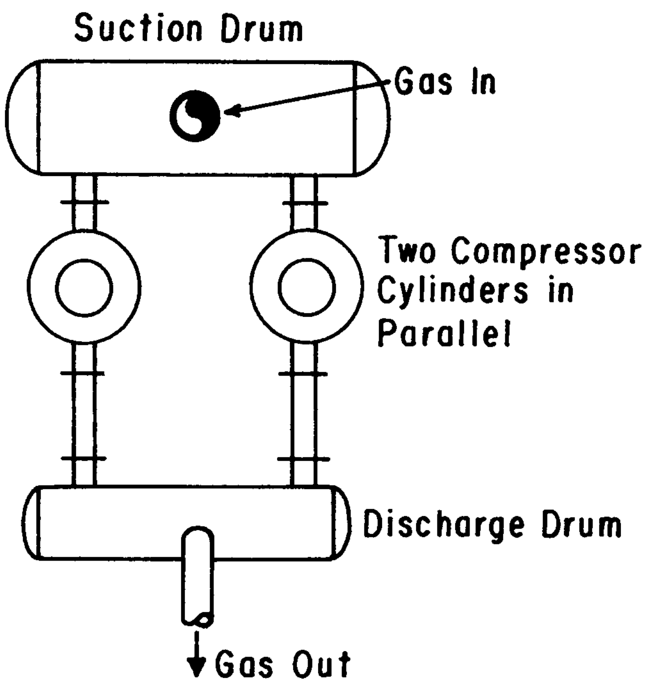


Figure 13-12. Parallel compressor cylinders with surge drums.

1,000%; it ignores the form or shape of the surge volume and the relative volumes of the bottle and the choke; it ignores the existence of pipe resonances and standing waves; it fails to predict pass-bands; and it leads to indiscriminate use of surge volume, choke diameter, and choke length to achieve a desired cut-off without considering resonance effects. The cut-off frequency for a low-pass filter is given by

$$f_c = \frac{C}{\pi}\sqrt{\frac{S}{LV_1}} \tag{13-1}$$

where C = velocity of sound, ft/sec at pressure, sp gr and temperatures
S = cross-sectional area of the choke, ft^2
L = length of choke, ft
V_1 = volume of the filter bottle, ft^3
f_c = frequency at cut-off, all greater than eliminated (almost), cycles/sec

Sharp and Henderson[10] summarize the advantages and disadvantages of the three techniques for attempting to design improved pulsation control into compressor suction and discharge systems. As they point out, these methods deal solely with the prevention of compressor pulsation transmission to the suction and discharge headers and do not provide a means of preventing or controlling the pulsations in the system piping, preventing cylinder overload, or improving cylinder efficiency (they may limit efficiency losses). The only techniques that can examine the entire system—compressor, piping, and associated equipment—use the analog or digital computer systems described earlier. Therefore, the following comments do not refer to the computerized techniques.

Cos-W Method

The advantages are that this method is relatively simple to use, permits the determination of low-frequency cut-off, indicates the frequency of the major pass-bands above cut-off, and permits the determination of pipe sizes to provide dampening.

The disadvantages include not allowing the calculation of the amount of dampening in any frequency, which can result in cut-off frequencies for all pass-bands being in error; the inaccurate prediction of pass bands above the low-frequency cut-off for some piping configurations; failure to predict pass-bands; and the indiscriminate use of surge volume, choke diameter, and choke length to achieve a desired cut-off without considering resonance effects.

Surge drums should be located as close to the compressor cylinders as possible (Figures 13-11 and 13-12). Direct connection is often used. The more pipe that is installed between the point of origin (cylinder) of the pressure waves and the surge drum, the less effective the drum seems to be. At the same time these lengths of pipe are subject to extreme pressure surges, and the possibility of mechanical rupture, weld failure, etc., is great. Bends, particularly vertical, should be well braced so as not to vibrate, Figures 13-1C and 13-1D.

Compressor buildings are often elevated to allow the necessary clearance for the installation of drums directly below the cylinders. Suction drums are usually located on top of the cylinders. Installation outside the compressor building is considered poor practice, except in special cases.

The piping system layout and anchoring is very important, as often the first recognition of pulsation problems is from the vibration of the piping. Increasing the number of anchors and their rigidity may not be the proper answer to the problem. Some vibrations may be caused by the gas pulsation, and others may be produced by the mechanical looseness of the compressor on its foundation, by the unbalanced forces within the compressor cylinder, or by improper foundation. Before making changes or corrections, it is important to determine the primary source of the vibration —gas pulsations or induced mechanical forces.[21] Instruments can aid in sorting out the main source of the problem. Piping design specialists can thoroughly analyze by sophisticated computer programs the piping sizes, the pipe routing, the pipe anchoring, the impact of valves in the system, exactly where anchors should be installed, and the forces and stresses on all the piping components. These specialists also can provide a meaningful analysis of where the vibration problems might occur and the level of the piping mechanical stress along the pipe layout. It is important to avoid 90° or other sharp bends or "dead" ends in the pipe. Reference 21 offers some sound experience on this topic.

Drums are used from vacuum conditions up to very high pressures of 5,000–10,000 psi or more.

Some situations bring into question the necessity of a surge drum in the system. This becomes more evident in the low flow rate, high compression ratio units in which the drum is calculated to be only slightly larger than the usual pipe size. In some of these cases, it has been found satisfactory to enlarge the pipe size and eliminate the drum. Each system must be carefully evaluated as generalities cannot solve the variety of situations.

When an intercooler is required and mounted between stages of compression, some designs will perform as a satisfactory surge drum. Caution should be used because long, small diameter coolers probably will not serve to reduce pulsation and surge. The vibration of tubes against tube supports and baffles has caused tube failures. Baffles should fit snugly on the tube when used in this application. Substitution of intercoolers for surge drums is not a recommended practice.

Internal Details

Many special internal designs can be used for these drums, with each design attempting to reduce the energy of certain disturbing frequencies from passing through the unit. In a sense, the internal pulsations "phase shift."[5] Patented arrangements are offered by several engineering organizations.

Excellent results in some petrochemical and light hydrocarbon applications have been achieved with the rather simple arrangements of Figures 13-7A and 13-7B. Pressure drop must be calculated as this can be an important part of drum performance.

A principle of application relative to internals is to obtain as many 180° or (second preference) 90° turns of the gas flow after it enters the drum as reasonably possible. Flat plate pieces, baffles, etc., welded to the side or nozzles are not recommended, because the vibration forces tend to fatigue and crack the welds. Baffle plates have been known to break loose on one face and rattle around in the drum. This can be potentially dangerous. The vessel itself should be designed for rugged service.

The ends of the drums should be elliptical or dished heads and never flat.

Design Method—Surge Drums (Nonacoustic)

The method presented here[2, 12] does meet the requirements necessary to ensure a good understanding of the entire system around a compressor. This method has given excellent results from suction pressures of atmospheric to discharge pressures up to 5,500 psi. Drum sizes are reasonable, and no serious piping vibration problems have been attributed to the drums. This method is presented as a guide and recommendation, but not as a substitute for a thorough computer piping system analysis, as might be performed by such world-recognized firms as Southwest Research Institute (San Antonio, Texas), Dresser-Rand Co. (USA), Burckhardt Engineering Works (Basel, Switzerland), Burgess Manning, Inc. (USA), and others.

The techniques described here are usually suitable for noncritical applications, as well as for adding pulsation-reduction equipment to existing systems to further reduce the magnitude of piping system pulsations, including metering problems. A careful, complete computer analysis of the system of equipment and piping upstream and downstream of the compressor(s) is essential for the performance and safety design of a system for high-pressure gases, such as ethylene, from about 3,000 psig to 50,000 psig and higher, for example.

Single-Compression Cylinder

Surge Drum Volume

$$V = \frac{(PD/\text{stroke})(\Delta V)}{(P'/P)^{1/k} - 1} \tag{13-2}$$

For single-action cylinders: PD/stroke = PD/rpm
For double-action cylinders: PD/stroke = PD/(2)(rpm)

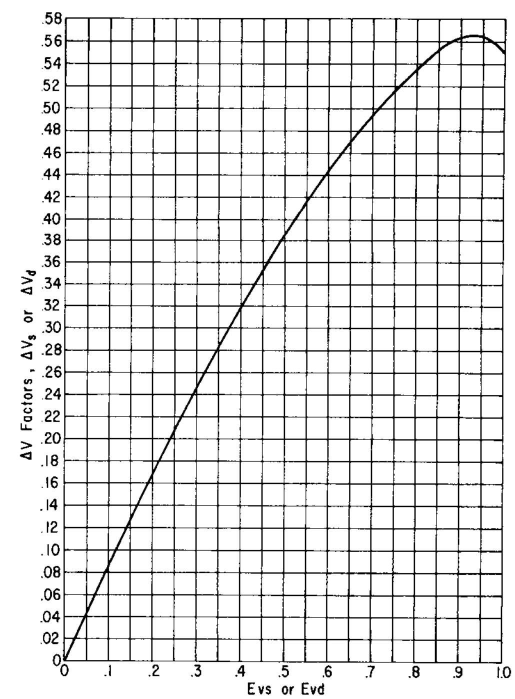


Figure 13-13. ΔV factor curve for single-acting cylinders. (Used by permission: Cooper Cameron Corporation.)

Volume Rate of Change

The volume rate of change factor, ΔV, is obtained from Figure 13-13 for single-acting cylinders and from Figure 13-14 for double-acting cylinders.

For suction drums, ΔV_s is evaluated using the suction volumetric efficiency, E_{vs}. E_{vs} = cfm at suction conditions/PD at the selected rpm, or E_{vs} can be obtained from curves of volumetric efficiency versus the compression ratio for varying cylinder clearances.

For discharge drums, the ΔV_d is evaluated using the discharge volumetric efficiency, E_{vd}.

$$E_{vd} = E_{vs}/(R_c)^{1/k} \quad (13\text{-}3)$$

Pressure Fluctuation Ratio

The allowable pressure fluctuation ratio, P′/P, within the surge drum is usually set at 1.05, representing a design effort for 5% pressure fluctuation from average system pressure. Values larger than this are seldom worth using, although an effort to reduce fluctuations to about 2–3% can be attempted by using P′/P at 1.02 or 1.03. The drums become larger in size as the P′/P ratio decreases, and it is difficult in the *average* installation to detect the difference in performance below 1.05. For special metering problems, P′/P of 1.02 should be considered.

Table 13-1 is convenient for ratios of 1.05.

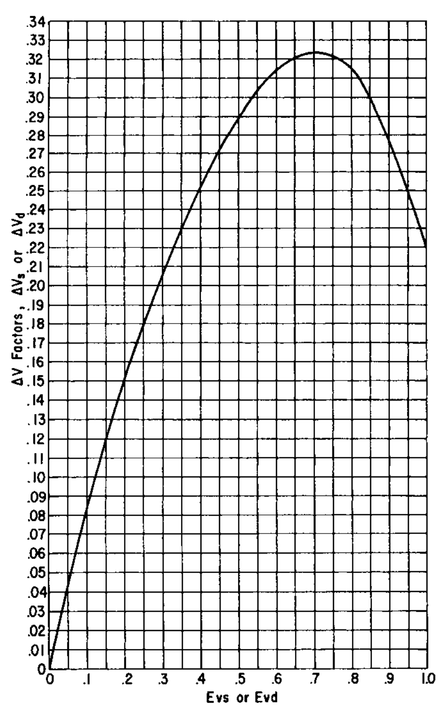


Figure 13-14. ΔV curve for double-acting cylinders. (Used by permission: Cooper Cameron Corporation.)

Table 13-1
Convenient Listing of k Values

k	$(1.05)^{1/k} - 1$
1.40	0.0355
1.35	0.0368
1.30	0.0383
1.26	0.0395
1.25	0.0398
1.20	0.0415
1.15	0.0434

Surge Drum Diameter

$$D = 10.32\,(V)^{1/3} \text{ in.}$$

where D is the minimum drum I.D. For usual low-to-medium pressure, D can be used as vessel O.D. if desired. This D is based on the most satisfactory drum length being twice the diameter.

Surge Drum Length

The preferable, overall length is L = 2D

If interferences occur with calculating the length, set the length and recalculate the diameter, being careful not to accept a diameter smaller than the original design when based on D = L/2.

Parallel Multicylinder Arrangement Using Common Surge Drum

This arrangement is based on compressor cylinders operating in parallel and being of the same size and characteristics.

Surge Drum Volume

Determine the volume for a single cylinder and then multiply by the number of cylinders in parallel on any particular compression service (or ratio).

$$V\,(\text{total}) = (V_{\text{single}})\,(\text{number of cylinders in parallel})$$

Surge Drum Diameter

Calculate the minimum diameter using the volume calculated for a single cylinder only.

Surge Drum Length

$$L = 2Dn, \text{ in.} \quad (13\text{-}4)$$

where n = the number of cylinders in parallel.

If the calculated length is too short to span the multiple cylinders (if mounted directly above or below), then increase the overall length to provide the required assembly length and keep the diameter the same as originally calculated.

If the length is too long, establish an acceptable length and recalculate the diameter using the calculated total drum volume. This new diameter must not be less than the originally determined value.

The drum length for multiple cylinder units will be determined by the center-to-center distances of the outer-most cylinders plus a minimum length for the installation of the flanges to connect to the cylinders and the mechanical clearances for good welding and fabrication.

Pipe Sizes for Surge Drum Systems[2, 12]

The minimum pipe diameter is given by

$$d = 13.54\left[\frac{(\text{PD per cyl.})(\text{rate factor})}{2{,}000}\right]^{1/2}, \text{ in.} \qquad (13\text{-}5)$$

The piping of concern in this design is given in Table 13-2 together with the approximate rate factor.

Pipe diameters are rounded up to the next standard size, and proper wall thickness (schedule) is determined consistent with the operating pressure of that portion of the system. For the rate factor, see Figure 13-15.

For the pipe size between the cylinder and its suction or discharge drum, the diameter of the cylinder connection may be used, provided that the pipe length between the cylinder and drum is not more than twice the cylinder flange diameter and provided that the calculated diameter does not exceed the cylinder connection by more than two standard pipe diameter increments.

Table 13-2
Pipe Evaluation in Compressor System

Pipe Section	Rate Factor for Single-Acting and Double-Acting Cylinders
Suction header	(E_{vs}) (No. cylinders per machine) (No. units or machines)
Header to suction drum	(E_{vs}) (No. cylinders per machine)
Suction drum to compressor cylinder	Read Figure 13-15; use E_{vs} and the proper curve for cylinder
Compressor cylinder to discharge drum	Read Figure 13-15; use E_{vd} and the proper curve for cylinder
Discharge drum to discharge header	(E_{vd}) (No. cylinders per machine)
Discharge header	(E_{vd}) (No. cylinders per machine) (No. units or machines)

Used by permission: Cooper Cameron Corporation.

Average Flow Rate of Gas from Cylinder

Double-acting cylinders:

Average flow rate = (PD) (rate factor), cfm

Single-acting cylinders:

Average flow rate = (PD) (2) (rate factor), cfm

Minimum Pipe Flow Area

$$\text{minimum area} = \frac{\text{average flow rate, cfm}}{\text{allowable velocity, ft/min}}$$

For average designs, allowable velocity = 2,000 ft/min

Example 13-1. Surge Drums and Piping for Double-Acting, Parallel Cylinder, Compressor Installation

A multistage compressor is to be installed in a process plant. The following data defines the compression:

rpm = 250
bhp = 2,000
First-stage conditions:
No. cylinders = 2, double-acting, 90° crank angle
Cylinder diameter = 34 in.

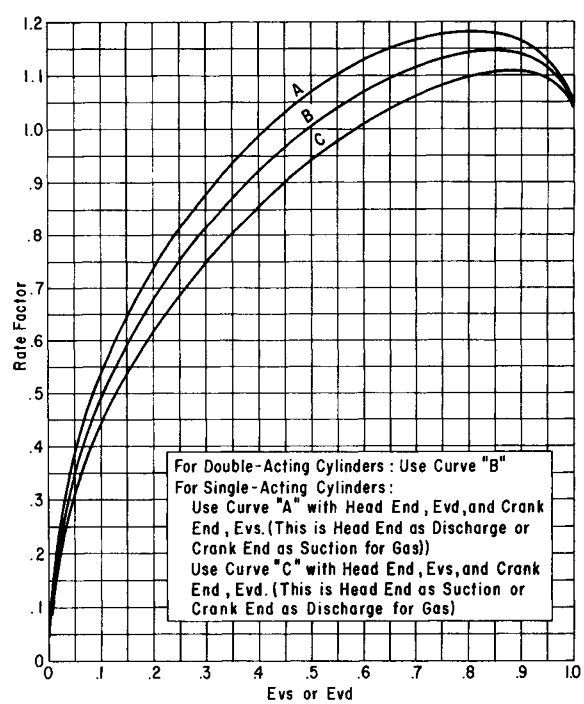


Figure 13-15. Rate factor curves. (Used by permission: Cooper Cameron Corporation.)

Piston rod diameter = 4 in.
Stroke = 20 in.
Clearance = 14.4%
$c_p/c_v = 1.35$
Suction = 14.2 psia at 104°F
Discharge = 34.2 psia at 249°F
Compression ratio = 2.41
Actual capacity (total) = 8,240 cfm (dry) at 14.2 psia and 104°F

1. Piston displacement, double-acting cylinder.
   This is usually given by the manufacturer. If not available, it may be calculated.

PD per cylinder
$= \pi\,(34)^2(20)(2)/(4)(1{,}728) - \pi(4)^2(20)/(4)(1{,}728)$
$= 21.0 - 0.146 = 20.85\ \text{ft}^3/\text{rev}$
At 250 rpm, total PD per cylinder = (250)(20.85) = 5,210 cfm

2. Rated capacity.
   The gas has 7.5% by volume water vapor at suction conditions (saturated).

Actual rated capacity (saturated) = 8,240/0.925
= 8,900 cfm at 14.2 psia and 104°F
Capacity *per* cylinder = 8,900/2 = 4,450 cfm (sat)

3. Volumetric efficiency (for suction).

$E_{vs} = 4{,}450/5{,}210 = 0.852$

Note: This checks curve of E_{vs} versus ratio of compression at 14.4% clearance.

4. ΔV_s from Figure 13-14.

ΔV_s (at $E_{vs} = 0.852$) = 0.299

5. Suction drum volume, for $(P'/P) = 1.05$

$$V = \frac{(PD/\text{stroke})(\Delta V_s)}{(P'/P)^{1/k} - 1} \qquad (13\text{-}6)$$

No. strokes for double-acting cylinder, at 250 rpm = (2)(250)
= 500

PD/stroke = 5,210/500 = 10.45 CF/stroke

$$V = \frac{(10.45)(0.299)}{(1.05)^{1/1.35} - 1} = \frac{3.12}{1.0368 - 1} = 84.8\ \text{ft}^3 \text{ per cylinder}$$

6. Suction drum diameter.

$D = 10.32\,(V)^{1/3} = 10.32(84.8)^{1/3} = 45.4\ \text{in.} = 3.78\ \text{ft}$

7. Suction drum length.

$L = 2Dn$

Using a 48-in. O.D. vessel and assuming a $^3/_8$-in. wall:

D = 47.25 in. I.D.
L = 2(47.25)(2 cylinders in parallel)
L = 189 in. = 15 ft, 9 in.

Because the compressor has the cylinder space 6 ft 8 in. apart (centerline), this would make the overhang on each side (15 ft, 9 in. − 6 ft, 8 in.)/2 = 4 ft, 6 in. approximately. This may be too much, particularly if the second-stage cylinder is on the same side of the crankshaft and adjacent on a 6 ft, 8 in. center.
Based on the layout spacing, select a drum length of 6 ft, 8 in. + 2 (2 ft overhang) = 10 ft, 8 in. overall length (approximate, to be adjusted on final detail drawing). The new diameter corresponding to this length is

New cross section = (84.8)(2)/10.67 = 15.9 ft^2
D = 4.48 ft = 53.8 in.
Use suction surge drum, Figure 13-16:
4 ft, 6 in. dia. × 10 ft, 8 in. overall straight shell length
Note that the diameter is greater than the minimum of 3.78 ft.

8. Pipe sizes.
   Assume that this installation requires three separate compressors with suction as just stated.
   Two of the machines will operate full-time; the third will serve as a standby. For such a situation, the suction header would be sized for the suction capacity of two machines. In practice, it may be better to run all three machines at reduced speed. Then when one comes down for some reason, the other two can speed up to carry the full load. Of course, this type of operation cannot be accomplished with a fixed speed drive. For the situation described, the design load is still the full capacity of two machines or four cylinders.

   a. Suction header.

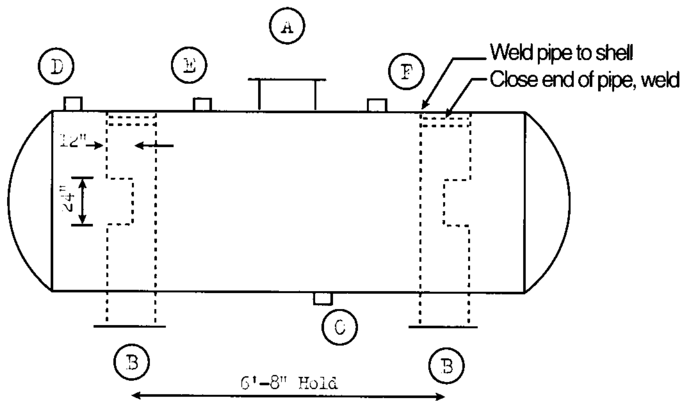
Rate factor = (E_{vs})(No. cyl.)(No. units or machines)
= (0.852)(2)(2) = 3.408
Minimum pipe diameter, d = 13.54 [(PD)(rate factor)/2,000]$^{1/2}$
d = 13.54[(5210)(3.498)/2,000]$^{1/2}$
= 40.7 ft

Use either a 40-in. I.D. header for these three compressors (two running or equivalent with three) or bring in two parallel headers, cross tied, of about half the flow area each, which equals (5,210)(3.408)/(2,000)(2) or 4.53 ft^2. This corresponds to a pipe diameter of 2 ft, 5 in., say 2 ft, 6 in. This is still a large pipe, and space arrangements may dictate which is preferred. If this header is coming to the compressors from a great distance, the pressure drop must be checked to be certain that the system drop will ensure specified pressure at the suction of the cylinders. In some cases, the header size can be made slightly smaller if the pressure

	SPEC. DWG. NO. A-
Job No.	Page 1 of 1 Pages
B/M No.	No. Units 3
DRUM OR TANK SPECIFICATIONS	Item No. D - 65

Service: Process Gas Compressor, First Stage Suction Drum

Size: 4'-6" O.D. x 10'-8" straight shell Type: Horizontal



DESIGN DATA

Operating Pressure 14.2 PSI A Operating Temp. Max. 150 °F.

Design Press. 14 PSIA to 40 PSI G Design Temp. 450 °F.

Code ASME Stamp Yes Lethal Const. No Density of Contents — Lbs/cu. ft

Materials: Shell Steel Heads Steel Supports Steel

Lining; Metal ____ Rubber or Plastic ____

Brick ____ Cement ____

Internal Corrosion Allowance 1/16 Self Supporting No Insulation: ☐ Yes ☒ No, Class ____

NOZZLES

Service	No. Req'd.	Size	Press Class	Facing	Mark No.
Inlet	1	30"	150	Raised	A
Vapor Out to compressor cyl.	2	24"	150	Raised	B
Liquid Out					
Drain	1	1"	6000	Screwed	C
Safety Valve					
Level Control					
Pressure Tap	1	1"	6000	Screwed	F
Vent	1	1"	6000	Screwed	D
Gage Glass					
Manhole					
Thermowell	1	1"	6000	Screwed	E

REMARKS

Adjust straight shell length to clear adjacent cylinders and aisle walkway, then recheck volume.

Drum to be mounted directly on suction cylinders.

By	Chk'd	Rev.	Rev.	Rev.	Rev.
Date					

P.O. To: ____

Figure 13-16. Surge drum design for Example 13-1.

drop is negligible. This must be handled with care and must not raise the flowing velocity over 50% greater than the 2,000 fpm. In all systems, a pressure drop check should be made.

b. Header to suction drum of each compressor.

Rate factor = (E_{vs}) (No. cylinders per machine)
= (0.852) (2) = 1.704

$d = 13.54[(5{,}210)(1.704)/2{,}000]^{1/2} = 28.6$ in.

Use diameter, d = 30-in. pipe.

c. Suction drum to compressor cylinder.

Read rate factor from Figure 13-15 for E_{vs} of 0.852 and double-acting cylinder, factor = 1.145.

$d = 13.54[(52.10)(1.145)/2{,}000]^{1/2} = 22.4$ in.

Use 22-in. or 24-in. pipe, or check size of suction opening and if reasonably close (in building, close to compressor), within about 2 in., use the same pipe size as cylinder suction. Do not use this approach if the suction drum is removed from the compressor and the suction connection is smaller than the size calculated. Use the larger pipe size.

9. Discharge drums.
These are evaluated in the same manner as for the suction drums, using the calculated E_{vd} and proper value of ΔV_d.
10. Discharge piping.
Use the rate factor indicated in Table 13-2 and follow the pattern illustrated for the suction piping.

Example 13-2. Single Cylinder Compressor, Single Acting

Size the discharge surge drum and associated piping for a single-acting, single-cylinder, motor-driven compressor with the following performance characteristics:

PD (given by manufacturer): 136.1 cfm
E_{vs} (given by manufacturer): 75.8%
rpm: 300
c_p/c_v: 1.40 for carbon monoxide
Ratio of compression: 3.15

Discharge is at the head end of the cylinder; suction is at crank end (toward crankshaft).

1. Discharge volumetric efficiency.

$$E_{vd} = E_{vs}/R_c^{1/k} = 0.758/(3.15)^{1/1.40} = 0.334$$

2. ΔV_d for single-acting cylinder at E_{vd} of 0.334:

From Figure 13-13, $\Delta V_d = 0.268$

3. Use P′/P = 1.05

Then $(P'/P)^{1/k} - 1 = (1.05)^{1/1.40} - 1 = 0.0355$

4. PD/stroke = PD/rpm = 136.1/300 = 0.454
5. Volume of discharge surge drum.

$$V = \frac{(PD/stroke)(\Delta V_d)}{(P'/P)^{1/k} - 1} = \frac{(0.454)(0.268)}{0.0355}$$

$$V = 3.43 \text{ ft}^3$$

6. Drum diameter, for L = 2D

$$D = 10.32(V)^{1/3} = 10.32(3.43)^{1/3} = 15.6 \text{ in.}$$

Use 16-in. diameter; mount drum under compressor cylinder.

7. Length.

L = 2D = 2(16) = 32 in., straight shell.

This length appears satisfactory.

8. Piping sizes.

a. Compressor to discharge drum.
At E_{vd} = 0.334, read Figure 13-15 curve A, because the discharge of the cylinder is at the head end (or outboard end), rate factor = 0.915

$d = 13.54\ [(136.1)(0.915)/2{,}000]^{1/2} = 3.38$ in.

Use 4-in. pipe

b. Discharge drum to process system (no header for a single compressor).

Rate factor = (E_{vd}) (No. cyl) = (0.334) (1) = 0.334

$d = 13.54\ [(136.1)(0.334)/2{,}000]^{1/2} = 2.04$ in.

Use 2-in. sch. 40 steel pipe if the length of run to other equipment is short; however, if the length is long, 3-in. pipe might be preferred. In either case, check the pressure drop.

Frequency of Pulsations

See Figures 13-3 and 13-4.

Although it is beyond the present scope of this presentation to include the details, reference to some of the points for an understanding of the Cosine (Cos)-W method of system analysis is suggested.[7, 11]

All frequencies probably exist in the pressure pulsations from a compressor; however, some are more basic than others and carry the greatest pressure energy peaks.

For a single-acting cylinder, the base frequency will be

$$f = rpm/60$$

Harmonics or multiples of 2, 3, 4, etc., of this frequency will exist and be dominant for two-cycle gas engines, and one half multiples will be dominant for four-cycle engines.[3] If several single-acting cylinders are operating on the same system in parallel, the magnitude of the pulses will depend upon the combination of cylinders and crank throws, and this magnitude is additive for the simultaneous waves in phase.

For a double-acting cylinder, the base frequency will be twice that of the single-acting cylinders, with its harmonics as additional predominant frequencies. Thus, one double-acting cylinder running at 300 rpm has a frequency of (300)(2)/60 or 10 cycles/sec. If the machine could slow down, as in gas engine operation, to say 260 rpm, then the base frequency would be (260)(2)/60 = 8.67 cycles/sec. This brings out the importance of examining the operation, as the filter or pulsation removal system should cut-off at about 9 cycles rather than 10 cycles, as the low troublesome frequency is 8.67. Actually, no design can cut-off exactly, but rather, a curved reduction in power occurs for any frequency. If two machines are in parallel, then the base frequencies are 10 and 20 cycles/sec, plus the harmonics up to about the eighth.

For any arrangement of equal angular piston spacing (on the crankshaft), here are the fundamental or base frequencies.[9]

For Odd Number of Cylinders

$$f = \frac{(\text{rpm})(\text{No. cylinders})(\text{action* of cylinders})}{60} \qquad (13\text{-}7)$$

*Use 1 for single-acting and 2 for double-acting cylinders.

For Even Number of Cylinders

$$f = (\text{rpm})(\text{No. cylinders})/(60) \qquad (13\text{-}8)$$

Compressor Suction and Discharge Drums

Design Method—Acoustic Low Pass Filters

The method outlined here presents the design analysis for acoustic filters after the descriptions of Taylor,[11] Sharp and Henderson,[10] von Nimitz,[13–16] and Hicks.[25]

Frequency of Pulsation

The first consideration for establishing a filter system is to determine what frequencies will exit for a given compressor unit:

1. For single-acting cylinder, rpm/60 will be the base frequency, and all filters should be sized below this base frequency. In addition, multiples of this base frequency (harmonics): 2 rpm/60, 3 rpm/60, 4 rpm/60, etc., will exist.
   Other frequencies will also exist, but the harmonics of the base frequency will be predominant. The largest pulse will depend upon the combination of cylinders and crank throws and will occur at the point where two or more cylinders discharge simultaneously, because successive waves in phase are additive.
2. In a double-acting cylinder, the base frequency will be double that of a single-acting cylinder with its harmonics as additional predominant frequencies.

With compressors of even angular piston spacing, the fundamental (or base) frequencies may be computed as follows:

$$fp = \frac{\text{rpm (N)}}{60}\ \text{(for even number of cylinders)} \qquad (13\text{-}9)$$

$$fp = \frac{\text{rpm (N)(A)}}{60}\ \text{for odd number of cylinders)} \qquad (13\text{-}10)$$

where N = number of cylinders
A = action of cylinders (single or double)

The effect of frequency and its amplitude upon vibrational stresses is of interest; therefore, let

s = displacement from rest position
t = time
A″ = peak amplitude of displacement
w = $2\pi f$ (f = frequency of vibration)
m = particle of mass being displaced and assuming a sinusoidal relation

Then,

$$s = A'' \sin wt$$

And the velocity of the mass is

$$\frac{ds}{dt} = wA'' \cos wt$$

$$\frac{d^2s}{dt^2} = w^2A'' \sin wt$$

And the force associated with this acceleration is

$$F = -(\Delta m)w^2 A'' \sin wt \qquad (13\text{-}11)$$

It is apparent that vibrational stresses are a function of the square of the frequency as compared with the first power of the amplitude. Thus, a structure vibrating at a high frequency but with a very small amplitude actually may be producing stresses far greater than one having a lower frequency but a larger amplitude of oscillation. It is important to bear this in mind when the band-pass frequencies are determined.

The wave form associated with the fundamental frequencies is primarily the result of the pulse produced by the stroke of the compressor piston, which is, in turn, modified by the action of the intake or discharge valve. In most cases the wave form is shaped by valve action and is partially modified by the characteristics of the piping downstream of the valve. The chief disturbing frequencies lie in the range of 4–100 cycles/sec.

Velocity of Sound in the Gas

The determination of the velocity of sound in the gas flowing may be made from either of the following equations:

$$C = \sqrt{\frac{144\,k\,P\,g}{\rho}} \qquad (13\text{-}12)$$

$$C = 222.5\sqrt{\frac{kT}{M}} \qquad (13\text{-}13)$$

where C = velocity of sound, ft/sec
k = ratio of specific heats, c_p/c_v
P = psia, absolute pressure of gas flowing
g = acceleration due to gravity, 32.2 ft/sec^2
ρ = gas density, lb/ft^3
T = absolute temperature of gas flowing, °R
M = molecular weight of gas flowing

Cutoff Frequency

After the anticipated disturbing frequencies have been determined, it is necessary to begin the sizing of the filter system. The cutoff frequency should be set *at least* one cycle per second below the lowest frequency to be filtered. The base frequency is determined from compressor speed and should be determined at the lowest anticipated compressor speed for variable-speed compressors. The following equation is used to determine cut-off frequency for a low-pass filter:

$$f_c = \frac{C}{\pi}\sqrt{\frac{S}{LV_1}} \qquad (13\text{-}14)$$

where f_c = cut-off frequency, cycles/sec
C = velocity of sound, ft/sec
S = flow area of choke-tube, ft^2
L = choke-tube length, ft
V_1 = first drum volume, ft^3
A_1 = cross-sectional flow area of first drum volume, ft^2
A_2 = cross-sectional flow area of second drum volume, ft^2

This equation holds only for the configuration shown in Figure 13-17.

Values to use for V_1, first drum volume—L, choke-tube length, and S, choke-tube flow area—will depend primarily on physical limitations and the degree of attenuation. Attenuation will be discussed later. Generally, the first drum volume may be sized according to piston displacement and volumetric efficiency considerations with a check of the physical limitations (nonacoustic method previously described) as to the choke-tube length and the pressure drop associated with the choke-tube. If the previously mentioned considerations will meet with design requirements, a further check on band-pass frequencies and the degree of attenuation are in order.

The Helmholtz resonance frequency is

$$f_n = \frac{C}{2\pi}\sqrt{\frac{S}{LV_1}} = 2f_c \qquad (13\text{-}15)$$

Band-Pass Frequencies

In a low band-pass type filter, certain frequencies exist for a given filter system that will pass through the filter unattenuated. These frequencies are known as band-pass frequencies and are determined as follows:

$$\text{First band-pass frequency} = \left(\frac{180}{\frac{2\pi L}{C}}\right) \div 57.296$$

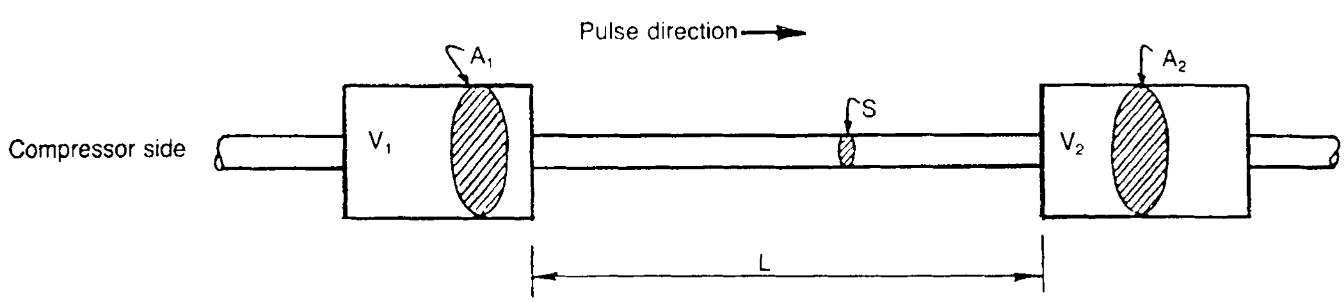


Figure 13-17. Series type acoustical filter system (volumes V_1 and V_2 may be separate vessels or one drum).

$$\text{or} = \frac{C}{2L} \tag{13-16}$$

$$\text{Second band-pass frequency} = \left(\frac{360}{\frac{2\pi L}{C}}\right) \div 57.296$$

$$\text{or} = \frac{C}{L} \tag{13-17}$$

The criterion for design is that none of the disturbing frequencies are within five cycles/sec from the band-pass frequencies.

Attenuation

1. By area ratio.
   Referring to the series filter, the degree of attenuation that can be anticipated from a given area ratio may be evaluated by the following expression:

$$(P_{ra})^2 = \frac{4A_2/A_1}{\left(\frac{A_2}{A_1} + 1\right)^2 \frac{\cos^2 2\pi fL}{C} + \left(\frac{S}{A_1} + \frac{A_2}{S}\right)^2 \sin^2 \frac{2\pi fL}{C}} \tag{13-18}$$

where P_{ra} = ratio of the average pulse energy flowing out of the filter to the average pulse energy flowing into the filter.

A_1, A_2 = cross-sectional flow areas respectively for the two drum volumes, ft^2

A plot of P_{ra} versus frequency will yield a curve as shown in Figure 13-18.

An examination of the attenuation by an area ratio equation will indicate that for any practical installation, the term

$$\left(\frac{S}{A_1} + \frac{A_2}{S}\right)^2 \tag{13-19}$$

will be somewhat larger than the term

$$\left(\frac{A_2}{A_1} + 1\right)^2 \tag{13-20}$$

Using this as a criterion, a quick calculation at a frequency such that

$$\cos \frac{2\pi fL}{C}$$

is equal to 1.0 or −1.0 will indicate the minimum attenuation to be expected.

Thus,

$$(P_{ra})^2 \text{ (maximum)} = \frac{4A_2/A_1}{\left(\frac{A_2}{A_1} + 1\right)^2} \tag{13-21}$$

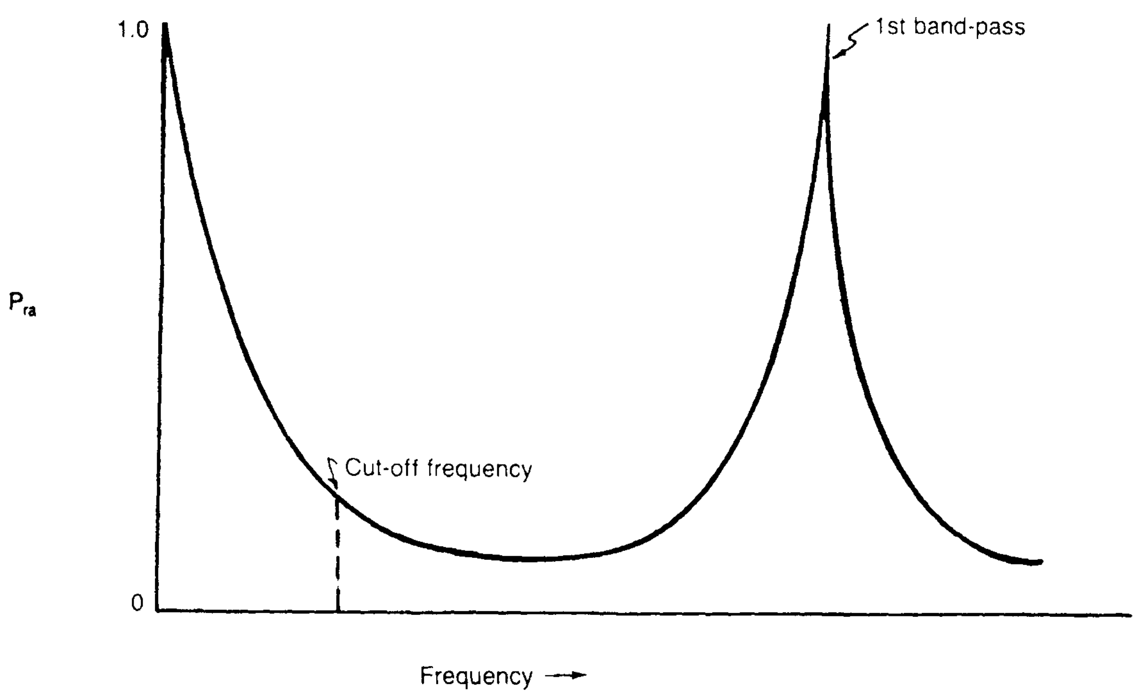


Figure 13-18. Pulse energy ratio for frequency variations.

Conversely, the maximum attenuation to be anticipated can be determined by setting

$$\sin \frac{2\pi fL}{C}$$

equal to 1.0 or -1.0, and

$$(P_{ra})^2 \text{ (minimum)} = \frac{4A_2/A_1}{\left(\frac{S}{A} + \frac{A_2}{S}\right)^2} \quad (13\text{-}22)$$

These equations should give a check as to the degree of attenuation that may be expected from a given filter system.

2. By cos-W method.[10]

The preceding methods are the criteria in making a selection of the proper volume and choke-tube combination. A more complex solution is now in order. Here, the following relation is used:

$$\cos W = \cos\frac{2\pi fL}{C} - \frac{\pi fV}{SC}\sin\frac{2\pi fL}{C} \quad (13\text{-}23)$$

where V = volume of the smaller chamber, ft^3
f = frequency in cycles/sec
L = choke-tube length, ft
C = velocity of sound in gas, ft/sec
S = cross-sectional area of choke-tube, ft^2

Cos W is used as an indicator to determine the frequency ranges of attenuation and transmission. Transmission regions are defined as those wherein cos W has a value of +1 and −1. Values of a higher order indicate regions in which attenuation will occur. Cos W should be plotted as shown in Figures 13-19A and 13-19B.

Because cos W has limits of +1 and −1, the *transmission through the filter can occur only for values that lie between the upper and lower limits with the maximum transmission occurring at cos W = 0.*

By fixing the quantities V, L, and S from the foregoing paragraphs, cos W may now be plotted by solving the cos W equation for any given frequency. The cos W curve may be more rapidly determined by calculating the maximum, minimum, and 0 points.

Maximum and minimum points occur where

$$\frac{-V}{(2LS + V)}\left(\frac{2\pi fL}{C}\right) = \frac{\tan 2\pi fL}{C} \quad (13\text{-}24)$$

Zero points occur where

$$\frac{2SL}{V}\left(\frac{1}{2\pi fL/C}\right) = \frac{\tan 2\pi fL}{C} \quad (13\text{-}25)$$

These equations may be quickly solved from the plot yielding the values of frequency at maximum, minimum, and 0 points, shown in Figures 13-20A and 13-20B.

Intersections of the tangent curve Z_1 and Z_2 correspond to 0 values of cos W. Intersections of the tangent curve Z_1 with

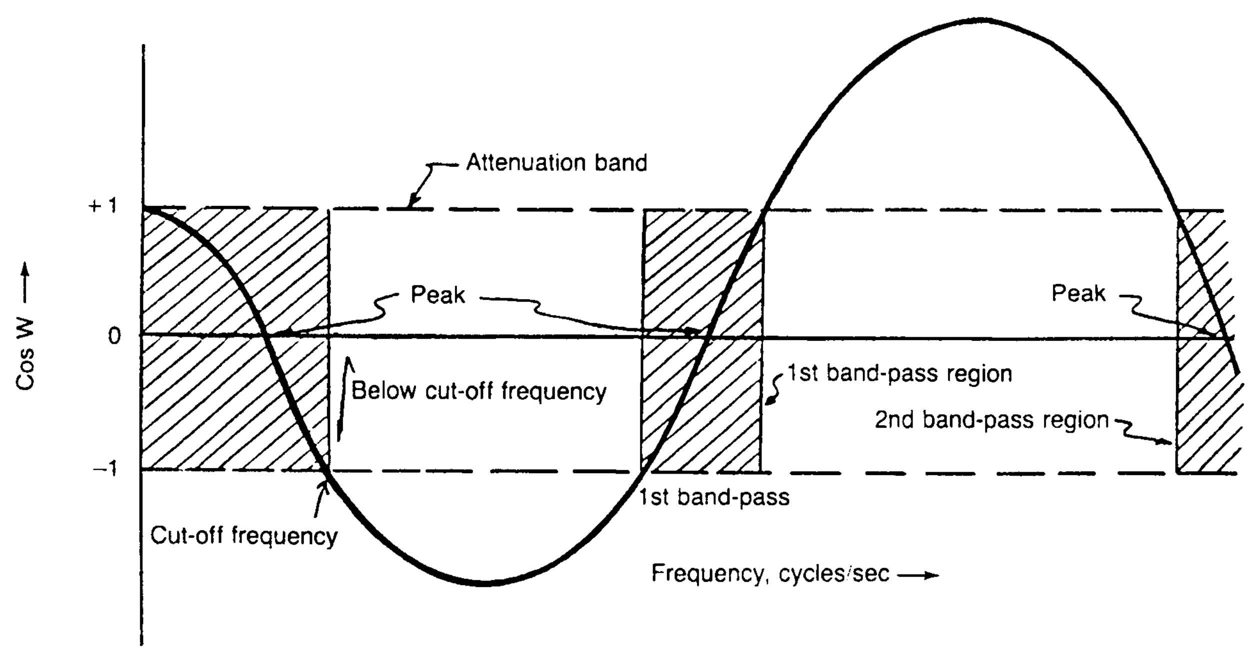


Figure 13-19A. Cos W plot.

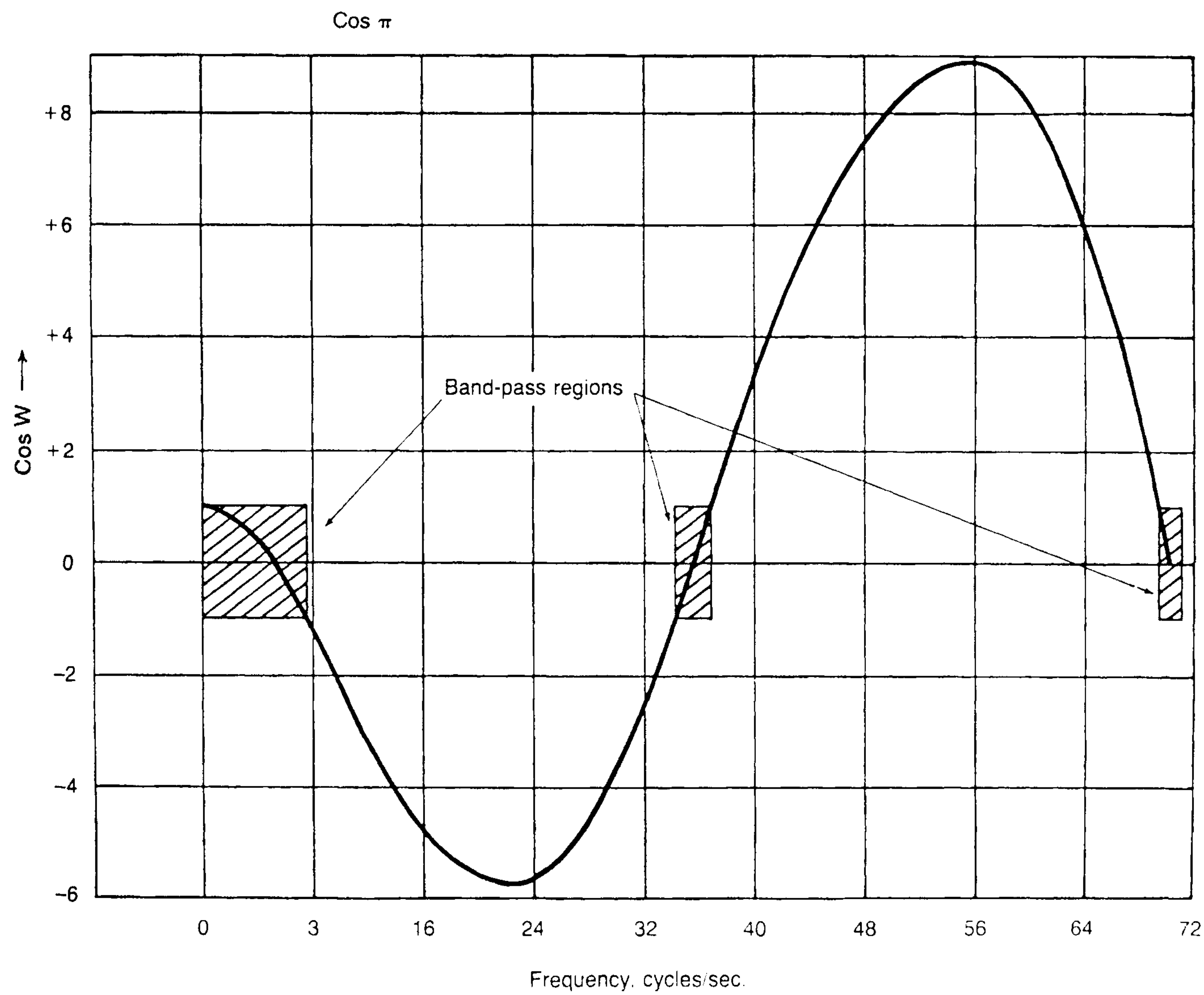


Figure 13-19B. Cos W curve for Example 13-3 calculation.

Z_3 correspond to the maximum and minimum values of cos W. Solving the abscissa value, $2\pi fL/C$ for $\int$ yields the frequency at which maximum, minimum, and 0 values of cos W occur. The cos W curve may now be constructed. Intermediate values of cos W may be approximated from this curve to construct the power ratio curve, shown in Figures 13-21A and 13-21B.

The power ratio P_r is determined by the relationship:

$$P_r = \frac{1}{1 + \left(\frac{2\pi fV}{SC}\right)^2 \cos^2 W} \qquad (13\text{-}26)$$

where P_r = power ratio (i.e., the percentage of power passing through the filter element, the ratio of the average pulse energy flowing out of the filter to that flowing into the filter)

f = frequency, cycles/sec

V = volume of smaller drum, ft^3

S = cross-sectional area of choke-tube, ft^2

C = velocity of sound in gas, ft/sec

It will be noted from the foregoing plot that the compressor should be operated between successive peaks (i.e., at a low P_r value).

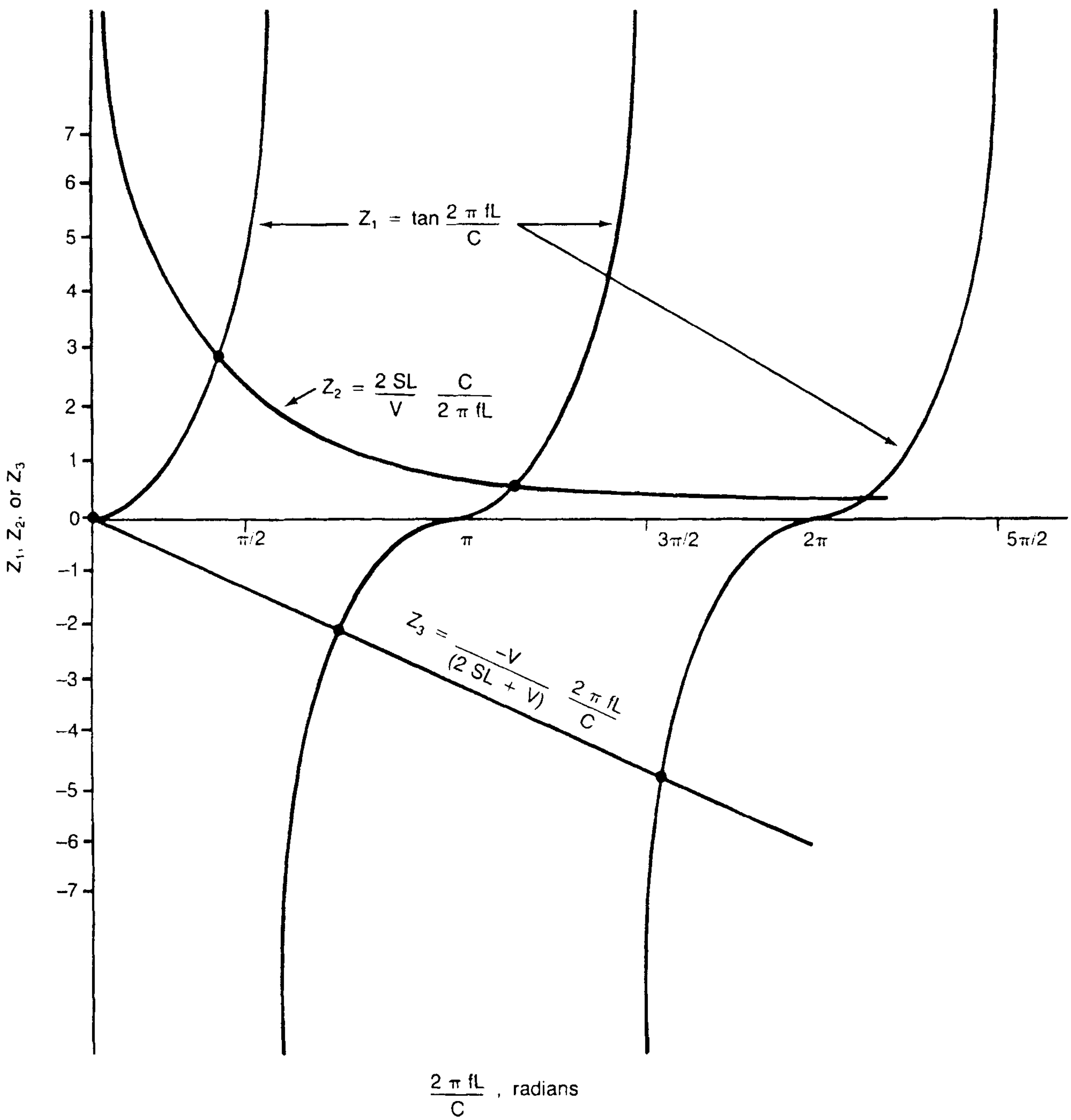


Figure 13-20A. Plot for solving cos W limits.

Example 13-3. Sizing a Pulsation Dampener Using Acoustic Method

See Figure 13-1C.

Given: An engine-driven compressor to handle 244 cfm natural gas (gravity = 0.658) at intake conditions of 90 psig and 80°F, to discharge at 150 psig; k = 1.27.

Compressor data: Single-cylinder, double-acting, 10-in. bore, 14-in. stroke, piston displacement of 382 cfm at a rated speed of 300 rpm. The lowest anticipated operating speed is 260 rpm.

To size a pulsation dampener so that vibrations may be held to a very minimum (size dampener for discharge side only for this example):

1. Approximate first volume size.

   Figure by surge bottle method allowing peak pulse pressure to be 5% of the line pressure (P′/P) = 1.05. Make both bottles the same size.

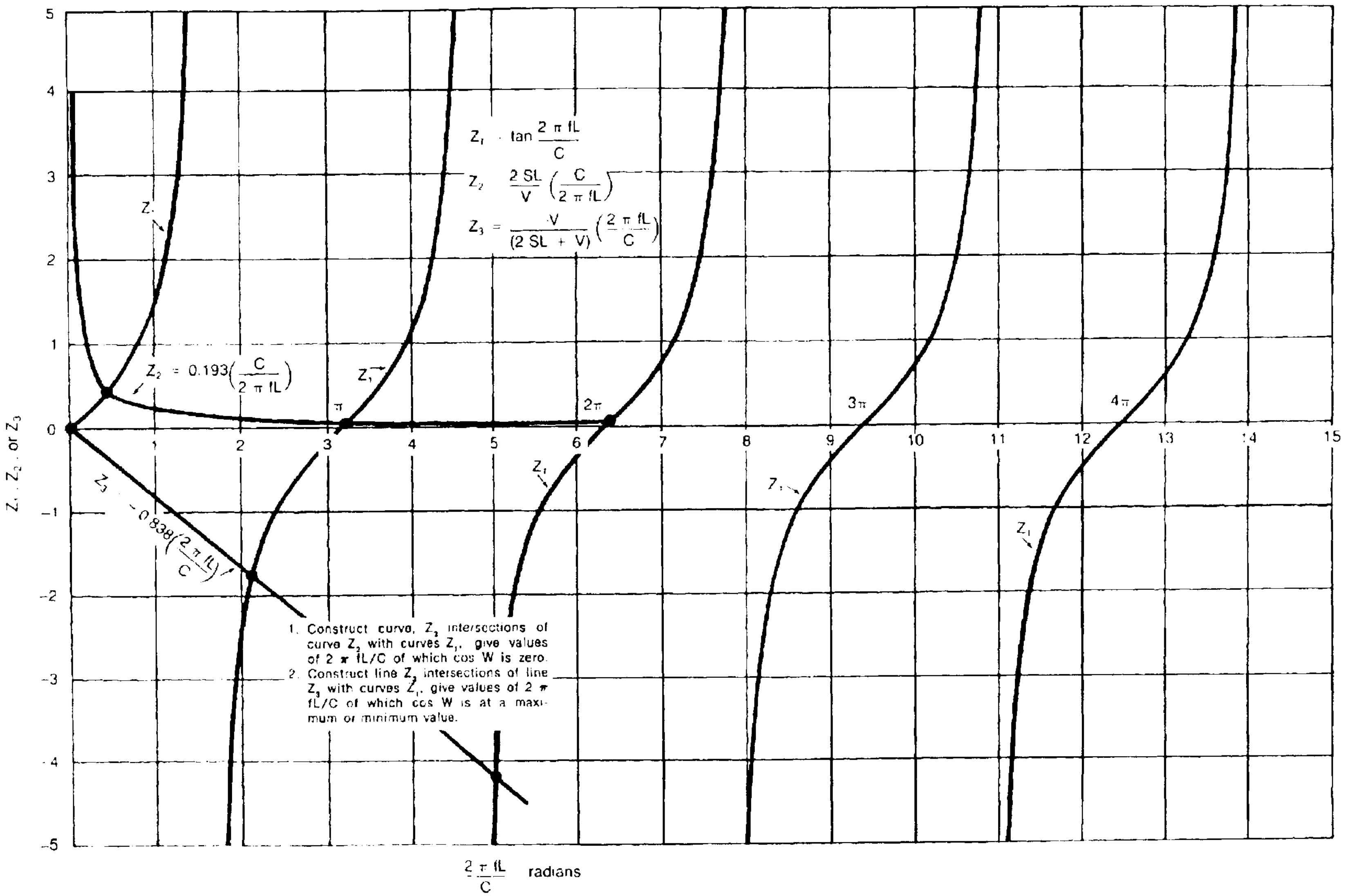


Figure 13-20B. Tangent curve for determining maximum, minimum, and zero points of cos W curve for example calculation.

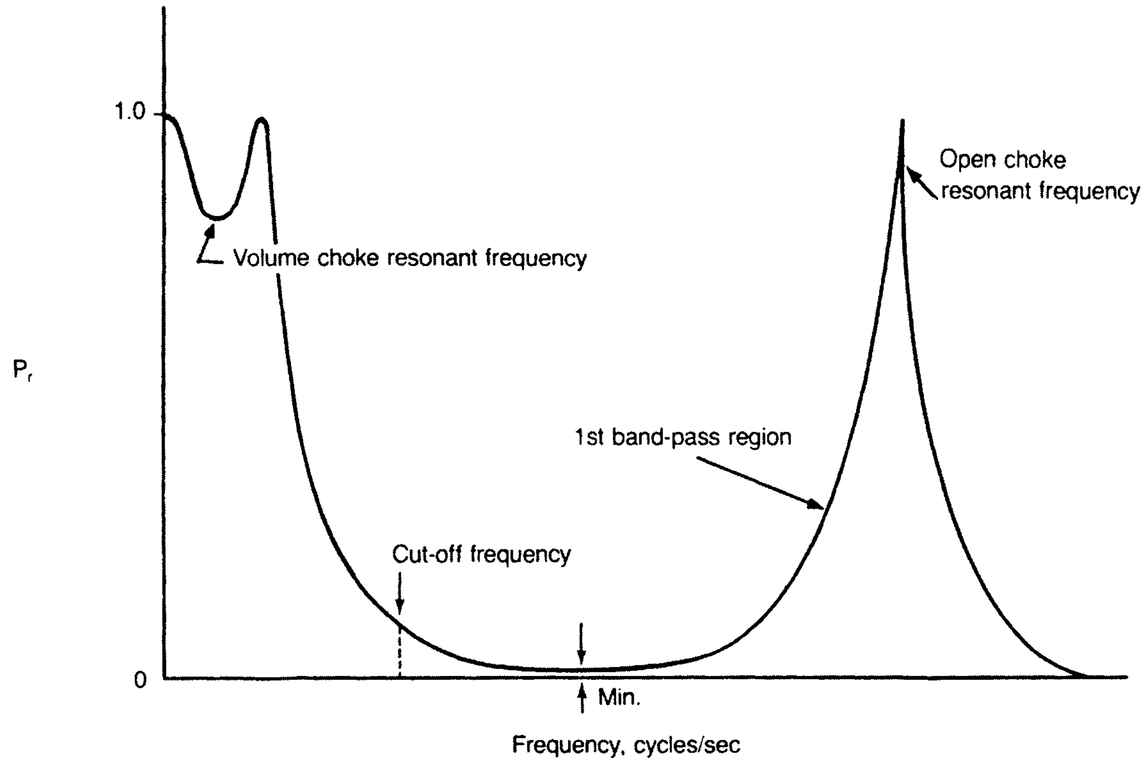


Figure 13-21A. Typical power ratio plot.

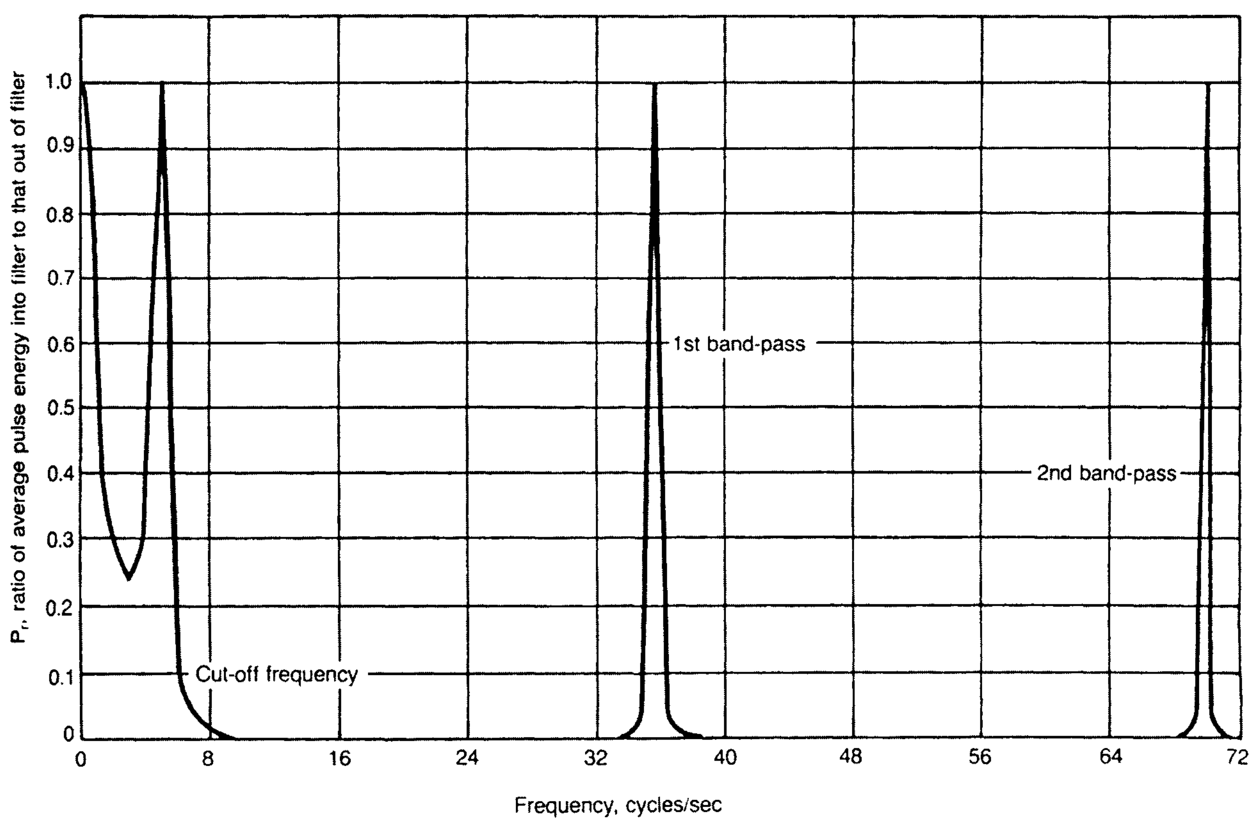


Figure 13-21B. P_r curve for example calculation.

a. Suction volumetric efficiency:

$$E_{vs} = \frac{\text{Rated capacity}}{\text{Piston displacement}} = \frac{244}{382} = 64\%$$

b. Discharge volumetric efficiency:

$$E_{vd} = \frac{E_{vs}}{R^{1/k}}$$

where R = compression ratio = 1.57
k = ratio of specific heats, c_p/c_v = 1.27

$$E_{vd} = \frac{64}{(1.57)^{1/1.27}} = 45\%$$

c. Discharge volume change factor from Figure 13-14:

$f_d = 0.273$

d. Volume of first bottle:

$$V = \frac{f_d \times \text{Piston displacement per stroke}}{(P'/P)^{1/k} - 1} \quad (13\text{-}27)$$

$$V = \frac{0.273(0.636)}{1.05^{1/k} - 1} = 4.43 \text{ ft}^3$$

Use 16 in., $^3/_8$ in. W. T. pipe by 3 ft-6 in. long with weld caps. Actual volume is approximately 4.83 ft^3.

2. Velocity of sound in natural gas:

$$C = 223\sqrt{\frac{kT}{M}}$$

where T = absolute temperature, °K
M = molecular weight = 19.07
k = ratio of specific heats, 1.27
C = velocity of sound, ft/sec

$$C = 223\sqrt{\frac{(1.27)(593)}{19.07}} = 1{,}405 \text{ ft/sec}$$

a. Discharge temperature:

$$T_2 = T_1\left(R^{\frac{1-k}{k}}\right) \quad (13\text{-}28)$$

where T_1 = intake temperature, °R
T_2 = discharge temperature, °R
R = compression ratio
k = ratio of specific heats

$$T- = 540(1.57)^{\frac{0.27}{1.27}} = 593 \text{ °R}$$

3. Disturbing frequencies for double-acting cylinder:

$$f = \frac{SNA}{60} = \frac{(rpm)nA}{60} \quad (13\text{-}29)$$

where rpm = compressor speed
n = number of cylinders
A = cylinder action (1 or 2 for single- or double-acting cylinders)

$$f\,(\text{at rated speed}) = \frac{300\,(1.0)(2)}{60} = 10 \text{ cycles/sec}$$

$$f\,(\text{at lowest speed}) = \frac{260(1)(2)}{60} = 8.67 \text{ cycles/sec}$$

The pulsation dampener must be capable of filtering the base frequency, or 10, 20, 30, 40 cycles/sec based on the rated speed, with a cut-off frequency at least one cycle/sec below the low speed frequency (8.67 cycles/sec).

$$f_c = \frac{1}{\pi}\sqrt{\frac{C^2S}{LV}} \quad (13\text{-}30)$$

where f_c = cut-off frequency, cycles/sec
C = velocity of sound, ft/sec
L = choke-tube length, ft
V = smaller volume, ft^3
S = cross-sectional area of choke-tube, ft^2
C = 1,405 fps from Item 2 calculation
V = 4.83 ft^3, from Item 1 calculation
S = 2-in. Sch. 40 pipe (area = 0.0233 ft^2)
Note: This is to be checked later for excessive pressure drop.
L = 20 ft
Note: This is to be checked later for the effect on band-passes.

then, $f_c = \frac{1}{\pi}\sqrt{\frac{(1{,}405)^2(.0233)}{20\,(483)}} = 6.95$ cycles/sec.

This 1.7 cycles/sec below the lowest anticipated operating speed should be acceptable.

4. Pressure drop check (should include entrance and exit velocity losses)

Gas flow through 2 in. choke-tube 20 ft long
Pressure = 150 psig
Temperature = 133°F

$$R_e = \frac{6.32W}{d\,\mu} \quad (13\text{-}31)$$

where W = flow, lb/hr
d = pipe I.D., ins.
μ = viscosity, centipoise

$$R_e = \frac{6.32(4{,}860)}{2.067\,(0.012)} = 1.24 \times 10^6$$

f = 0.017 (Chapter 2, V. 1)

$$\Delta P_{100'} = \frac{0.000366\ fW^2}{d^5\ \rho}$$

where f = friction factor
W = flow, lb/hr
d = I.D. of pipe
ρ = density, lb/ft³

$$\Delta P_{100'} = \frac{(0.000366)(0.017)(4{,}860)^2}{37.7(0.49)}$$

= 7.3 psi/100 eq ft
= 1.5 psi for 2 in. choke-tube. This is not excessive, and a 2 in. choke-tube will be used.

5. Band-pass frequency check.
Band-Pass frequencies should lie at least 5 cycles/sec from disturbing frequencies.

a. First band-pass frequency:

$$f_{1st} = \frac{C}{2L}$$

where L = choke-tube length, ft
C = velocity of sound, ft/sec

$$f_{1st} = \frac{1{,}405}{2(20)} = 35.3 \text{ cycles/sec}$$

35.3 cycles/sec + approximately 2 cycles/sec will be only partially filtered. As 35.3 cycles/sec lies between the 2nd and 3rd harmonic of 10 cycles/sec, this band-pass should not prove detrimental to the dampener.

b. Second band-pass:

$$f_{2nd} = \frac{C}{L} = \frac{1{,}405}{20} = 70.6 \text{ cycles/sec}$$

6. Attenuation by area ratio method

$$(P_{ra})^2 = \frac{4A_2/A_1}{\left(\frac{A_2}{A_1} + 1\right)^2 \cos^2 \frac{2\pi fL}{C} + \left(\frac{S}{A_1} + \frac{A_2}{S}\right)^2 \sin^2 \frac{2\ fL}{C}} \qquad (13\text{-}32)$$

where A_1 = cross-sectional area of 1st volume, ft²
A_2 = cross-sectional area of 2nd volume, ft²
S = cross-sectional area of choke tube, ft²
f = frequency, cycles/sec
L = choke-tube length, ft
C = velocity of sound, ft/sec

P_{ra} will be at a maximum value at cos $(2\pi fL/C) = 1.0$ and at a minimum value at sin $(2\pi fL/C) = 1.0$. Intermediate points of the P_{ra} curve may be determined; however, for this example, consider only maximum and minimum values. The P_{ra} curve may be plotted as shown in Figure 13-22.

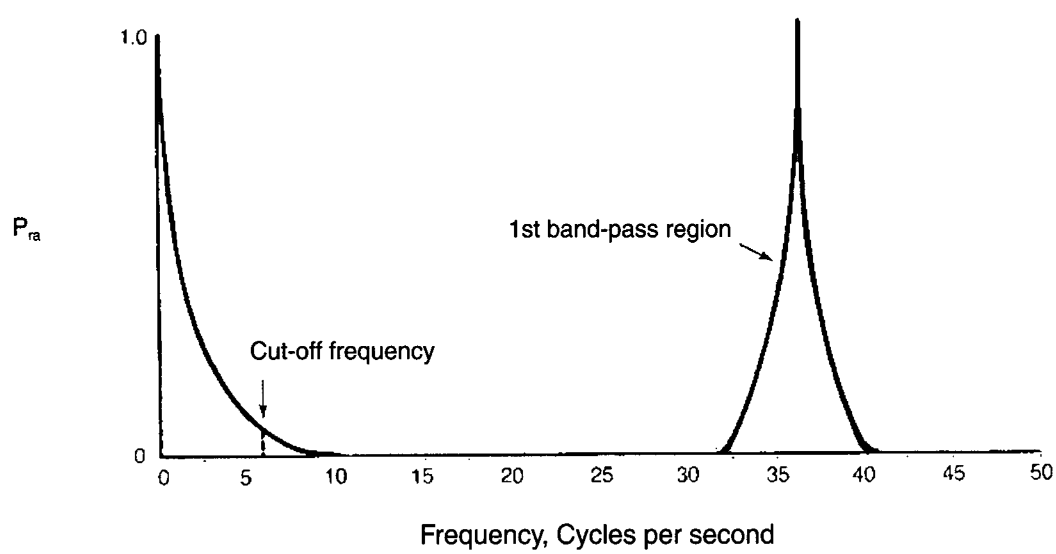


Figure 13-22. P_{ra} curve for Example 13-3.

a. Minimum attenuation ($\cos 2\pi fL/C = 1.0$):

$$(P_{ra})^2 = \frac{4A_2/A_1}{\left(\frac{A_2}{A_1} + 1\right)^2}$$

$A_1 = A_2$ and $P_{ra} = 1.0$

b. Maximum attenuation ($\sin 2\pi fL/C = 1.0$):

$$(P_{ra})^2 = \frac{4A_2/A_1}{\left(\frac{S}{A_1} + \frac{A_2}{S}\right)^2} = \frac{4}{\left(\frac{0.0223}{1.268} + \frac{1.268}{0.233}\right)^2} = 0.00135$$

7. Attenuation by the Cos W method
The cos W curve may be approximated by determining only the maximum, minimum, and 0 values of cos W.

$$\cos W = \cos\frac{2\pi fL}{C} - \frac{\pi fV}{SC}\sin\frac{2\pi fL}{C} \qquad (13\text{-}33)$$

Zero points of cos W occur where

$$Z_2 = \frac{2SL}{V}\left(\frac{C}{2\pi fL}\right) = \tan\frac{2\pi fL}{C} = Z_1 \qquad (13\text{-}34)$$

and the maximum and minimum points occur where

$$Z_3 = \frac{-V}{(2SL + V)}\left(\frac{2\pi fL}{C}\right) = \tan\frac{2\pi fL}{C} = Z_1$$

$$Z_2 = \frac{2(0.0233)(20)}{4.83}\left(\frac{C}{2\pi fL}\right) = 0.193\frac{C}{2\pi fL}$$

$$Z_3 = \frac{-4.83}{(2(0.0233)(20) + 4.83)}\left(\frac{2\pi fL}{C}\right) = -0.838\frac{2\pi fL}{C}$$

Referring to Figure 13-20B, the following table may be set up, using

$$\frac{2\pi fL}{C} = \frac{2\pi f(20)}{1{,}405} = 0.0895f$$

$$\frac{\pi fV}{SC} = \frac{\pi f(4.83)}{0.0233\,(1{,}405)} = 0.463f$$

Cos W Point	$\frac{2\pi fL}{C}$	f,cps	$\cos\frac{2\pi fL}{C}$	$\sin\frac{2\pi fL}{C}$	$\frac{\pi fV}{SC}$	Cos W
+max	0	0	+1.0	0	0	+1.0
0	0.44	4.9	+0.905	+0.425	2.27	−0.06
−max	2.05	22.9	−0.887	+0.462	10.6	−5.79
0	3.21	35.8	−0.998	−0.070	16.6	+0.172
+max	5.02	56.2	+0.952	−0.306	26.0	+8.91
0	6.3	70.5	+1.00	+0.030	32.6	+0.022

(Read from sin and cos tables for columns 4 and 5.)

Note: Maximum and minimum (+, −) read from Z_3 intersections with Z_1, and 0 points read from Z_2 intersections with Z_1. Using these maximum, minimum, and 0 points, the cos W curve may now be plotted. See Figure 13-19B.

As the transmission region occurs for values of cos W from +1 to −1, the P_r curve may now be determined from the relation (see Figure 13-21B):

$$P_r = \frac{1}{1 + \left(\frac{2\pi fV}{SC}\right)^2 \cos^2 W} \qquad (13\text{-}35)$$

$$\frac{2\pi fV}{SC} = \frac{2\pi f\,(4.83)}{0.0233\,(1405)} = 0.926f$$

f, cps	Cos W	$\frac{SC}{2\pi fV}$	$\left(\frac{2\pi fV}{SC}\right)^2 \cos^2 W$	P_r
0	1.0	0	0	1.0
1	0.96	0.926	0.79	0.56
2	0.83	1.85	2.36	0.297
3	0.65	2.78	3.26	0.235
4	0.37	3.70	1.87	0.348
5	0	4.63	0	1.0
6	−0.35	5.56	3.80	0.208
7	−0.78	6.49	25.6	0.0376
8	−1.26	7.41	87.2	0.0113
34	−1.10	31.5	1200	0.00083
35	−0.40	32.4	168	0.00595
36	+0.36	33.3	144	0.00695
69	+1.40	63.9	8000	0.000125
71	−0.90	65.7	3500	0.000286

8. Second volume and choke-tube resonance frequency and beat frequencies

a. Resonant frequency of second volume and choke-tube:

$$f_r = \frac{1}{2\pi}\sqrt{\frac{C^2S}{LV}}$$

where V = volume of second drum, ft^3 (make same as first volume = 4.83 ft^2)
C = velocity of sound, ft/sec
S = cross-sectional area of choke-tube, ft^2
L = choke-tube length, ft

$$f_r = \frac{1}{2\pi}\sqrt{\frac{(1,405)^2(0.0233)}{(20)\,(4.83)}} = 3.48 \text{ cycles/sec}$$

b. Beat frequencies:
A beat frequency exists as the sum or difference of two frequencies. For a base of 10 cycles/sec and a resonant frequency of 3.48 cycles/sec, beat frequency = 10 − 3.48 = 6.52 cycles/sec, which is below the compressor base frequency of 10 cycles/sec and is acceptable. Beat frequencies of the harmonics of the base frequency will be filtered.

9. Summary
Consider any risers that may be installed in the piping system. In considering risers, use resonant calculations for a closed tube (C/L) and investigate the open-end tube (C/2L).
The filter system designed will filter all frequencies lying above the cut-off frequency of 6.95 cycles/sec to the first band-pass of 35.8 ± 2 cycles/sec, and the second band-pass of 70.5 ± 2 cycles/sec. This arrangement should perform satisfactorily.
10. Recommendation
Install a two-bottle filter system for the 10 in. × 14 in. double-acting compressor, each bottle to be 16 in. ϕ × $^3/_8$ in. W.T. × 3 ft-6 in. long with weld caps. Install a 2 in. Sch. 40 choke-tube between bottles 20 ft total length. Install 90° elbows or baffles at entrance and exit of each bottle.

Design Method—Modified NACA Method for Design of Suction and Discharge Drums

The sizing of pulsation drums is based on the National Advisory Committee for Aeronautics (NACA) Technical Note 2893. The single expansion chamber type has been chosen because of cost, ease of installation, and effectiveness.

When using this method, it is necessary to assume an allowable pressure drop though the pulsation drum. This pressure drop does not need to be large ($^1/_4$ –1 lb). However, the higher the allowable pressure drop, the smaller the resulting bottle, because the diameter and the length of the drum vary as (pressure drop)$^{-1/6}$ or the volume of the drum varies as (pressure drop)$^{-1/2}$.

Compressor piping should be sized in the normal manner, and the pressure drop taken through an orifice plate or a short pipe spool placed on the inlet end of the suction bottles or on the outlet end of the discharge bottles.

The suction and discharge drums should be installed as close to the compressor cylinder inlet and outlet flanges as practical.

Do not grossly oversize compressor piping. Compressor manufacturers generally recommend gas velocities in the range of 2,000 ft per min. Compressor piping should be sized to meet compressor pressure requirements and allowable pressure drops between stages. Avoid vertical loops in compressor piping.

In this design procedure, Steps 1 through 4 are based on two assumptions that will be close approximations for normal compressor surge drums. These assumptions are t hat d_2^2/d_1^2 will always be large (20–40) and that the $\sin^2(2\pi f l_e/C)$ will always be small enough that the sine of the angle and the angle in radians are nearly equal.

If these two assumptions are incorrect, the solution given here will be incorrect. This will show up in the design procedure, Step 5, "Check actual value of P_r." If the calculated value of P_r is more than 1.25 times the assumed value of P_r, dimensions should be adjusted to bring the calculated and assumed values into line. If the calculated value of P_r cannot be brought into line by reasonable dimensional adjustment, refer to the NACA Technical Note 2893.

Data required

$k = c_p/c_v$
T = temperature of gas in drum, °R
M = average mol wt
Q = gas flow cfm at standard conditions (14.7 psia and 60°F)
P = pressure in drum, psia
C_o = orifice or short pipe coefficient
= 0.62 for sharp-edged orifice
= 0.81 for short pipe
h = allowable pressure drop, psi
$\bar{V}$ = specific volume, ft³/lb, at 14.7 psia and 60°F
f = frequency, cycles/sec

$$= \frac{\text{rpm}}{60} \text{ for single cylinder, single acting}$$

$$= \frac{\text{rpm} \times 2}{60}$$

for single-cylinder, double-acting; or two cylinders, double-acting, 180° apart, where two cylinders constitute a single stage

$$= \frac{\text{rpm} \times 4}{60}$$

for two cylinders, double-acting, 90° or 270° apart, where two cylinders constitute a single stage

$$P_r = \text{Power ratio} = \frac{\text{sound power leaving bottle}}{\text{sound power entering bottle}}$$

= 0.05 for reducing piping vibration

= 0.03 for reducing pulsation for metering

Design Procedure

1. Determine velocity of sound

$$C = \sqrt{\frac{49{,}781\ k\ T}{M}}\text{, velocity of sound, ft/sec} \qquad (13\text{-}36)$$

$k = c_p/c_v$
T = °R, operating temperature
M = average mol weight

2. Determine orifice or choke diameter

d_1 = choke or orifice diameter, I.D.

$$= \left(\frac{Q}{187.2\ C_o}\right)^{1/2} \left(\frac{T}{h\bar{V}\ P}\right)^{1/4} \qquad (13\text{-}37)$$

Q = cfm at 14.7 psia and 60°F
h = differential pressure drop, psi
$\bar{V}$ = specific volume, ft³/lb, at 14.7 psia and 60°F
P = operating pressure, psia
C_o = orifice coefficient
= 0.62 for sharp-edged orifice
= 0.81 for short pipe

3. Determine the operating frequency

$$f = \frac{rpm}{60} \text{ for single cylinder, single acting}$$

$$= \frac{rpm \times 2}{60}$$

for single cylinder, double-acting if two cylinders are used for a single stage and their cranks are 0° or 180° apart

$$= \frac{rpm \times 4}{60}$$

for two cylinders used as a single stage and their cranks are 90° or 270° apart

4. Determine approximate diameter and length of bottle

d_2 = I.D. of bottle, ins.

$$d_2 \cong \left(\frac{6\ d_1^2 C}{\pi\ f\ (P_r)^{1/2}}\right)^{1/3}$$

$$l_e = \frac{2\ d_2}{12}$$

$$P_r = \text{power ratio} = \frac{\text{sound power leaving bottle}}{\text{sound power entering bottle}}$$

Assume P_r = 0.05 for reducing pipe vibration
= 0.03 for reducing pulsation for metering
l_e = length of bottle, ft (for dished heads, the effective length may be considered as 1/2 I.D.D.)

5. Check actual value of P_r

Alter d_2 and l_e to standard nominal dimensions but do not reduce the volume of the bottle and solve:

$$P_{ra} = \frac{1}{1 + 1/4\left(\frac{d_2^2}{d_1^2} - \frac{d_1^2}{d_2^2}\right)^2 \sin^2 \frac{2\pi\ f l_e}{C}} \qquad (13\text{-}38)$$

Note: $\left(\frac{2\pi\ f\ l_e}{C}\right)$ is in radians

If calculated P_r is much different than assumed P_r, change bottle dimensions to make them approximately equal. Make the bottle smaller if P_r is too small or larger if P_r is too large. Bottle dimensions may be partially controlled by physical conditions, but the P_r should be kept reasonably close to the assumed value.

6. Check for first band-pass frequency

$$f_{1bp} = \frac{C}{2\ l_e} \text{ first band-pass frequency}$$

For best results, the first band-pass frequency should *not* be an integer multiple of the base frequency.

Example 13-4. Sample Calculation

Problem

Design suction and discharge bottles for an ethylene compressor. Suction drum is to be sized for an orifice meter a short distance upstream. Discharge drum is to be sized for prevention of piping vibration.

Material	Ethylene
Capacity	130,000 lb/day
Suction pressure	300 psig
Suction temperature	80°F
Discharge pressure	500 psig
Discharge temperature	126°F
Compressor speed	260 rpm
Single-cylinder, double-acting	

Data

k = 1.21
T_s = suction, 80 + 460 = 540°R
T_d = discharge, 126 + 460 = 586°R
M = 28
Q = 1,220 scfm (14.7 psia and 60°F)
P_s = suction, 315 psia
P_d = discharge, 515 psia
C_o = 0.62
h = 5 psi
$\bar{V}$ = 13.5 ft³/lb

$f = 8.67$
$P_r = 0.03$ for suction
$\quad = 0.05$ for discharge

Suction Drum

1. Velocity of sound

$$C = \sqrt{\frac{49{,}781\ k\,T}{M}}$$

$$= \sqrt{\frac{49{,}781 \times 1.21 \times 540}{28}}$$

$= 1{,}080$ ft/sec

2. Orifice diameter

$$d_1 = \left(\frac{Q}{187.2\ C_o}\right)^{1/2} \left(\frac{T}{hV\,P}\right)^{1/4}$$

$$= \left(\frac{1.220}{187.2 \times .62}\right)^{1/2} \left(\frac{540}{5 \times 13.5 \times 315}\right)^{1/4}$$

$= 1.29$ in. (orifice diameter)

3. Frequency

$$f = \frac{\text{rpm} \times 2}{60} = \frac{260 \times 2}{60} = 8.67 \text{ cps}$$

4. Approximate diameter and length of drum

$$d_2 \cong \left(\frac{6\,d_1^2\,C}{\pi f\,(P_r)^{1/2}}\right)^{1/3}$$

$$\cong \left(\frac{6 \times 129^2 \times 1080}{3.14 \times 8.67 \times (0.03)^{1/2}}\right)^{1/3}$$

$\cong 13.2$ in. (drum I.D.)

$$l_e = \frac{2\,d_2}{12}$$

$$= \frac{2 \times 13.2}{12} = 2.21 \text{ ft}$$

$= 2$ ft. $2\frac{1}{2}$ in.

5. Check actual value of P_r
Use 14 in. O.D. $^3/_8$ in.-wall pipe with pipe caps 2 ft, 6 in. overall. Effective length 2 ft, 6 in.–0 ft, 3 $^1/_2$ in. = 2 ft, 2 $^1/_2$ in.

$$P_r = \frac{1}{1 + 1/4\left(\frac{d_2^2}{d_1^2} - \frac{d_1^2}{d_2^2}\right)^2 \sin^2\frac{2\pi f\,l_e}{C}}$$

$$= \frac{1}{1 + 1/4\left[\left(\frac{13.25}{1.29}\right)^2 - \left(\frac{1.29}{13.25}\right)^2\right]^2 \sin^2\frac{2\pi\ 8.67 \times 2.21}{1080}}$$

$= 0.0284$ This is satisfactory.

6. First band-pass frequency

$$f_{1bp} = \frac{C}{2\,l_e}$$

$$= \frac{1{,}080}{2 \times 2.21} = 244$$

$$\frac{244}{8.67} = 28.1$$ This is satisfactory.

Discharge Drum

1. Velocity of sound

$$C = \sqrt{\frac{49{,}781\ k\,T}{M}}$$

$$= \sqrt{\frac{49{,}781 \times 1.21 \times 586}{28}}$$

$= 1{,}120$ feet per second

2. Orifice diameter

$$d_1 = \left(\frac{Q}{187.2\ C_o}\right)^{1/2} \left(\frac{T}{h\bar{V}P}\right)^{1/4}$$

$$= \left(\frac{1{,}220}{187.2 \times .62}\right)^{1/2} \left(\frac{586}{5 \times 13.5 \times 515}\right)^{1/4}$$

$= 1.17$ in. (orifice diameter)

3. Frequency

$$f = \frac{\text{rpm} \times 2}{60}$$

$$= \frac{260 \times 2}{60} = 8.67 \text{ cps}$$

4. Approximate diameter and length of drum

$$d_2 \cong \left(\frac{6\,d_1^2\,C}{\pi f\,(P_r)^{1/2}}\right)^{1/3}$$

$$\cong \frac{(6 \times 1.17^2 \times 1{,}120)^{1/3}}{3.14 \times 8.67 \times (0.05)^{1/2}}$$

$$\cong 11.42$$

$$l_e = \frac{2 \times 11.42}{12} = 1.9 \text{ ft} = 1 \text{ ft}, 10\tfrac{3}{4} \text{ in.}$$

5. Check actual value of P_r
   Use 12 in. schedule 40 pipe 11.94 in. I.D. with pipe caps 2 ft, 3 in. overall. Effective length 2 ft, 3 in.–0 ft, $3^3/_8$ in. = 1 ft, $11^5/_8$ in. = 1.97 ft.

$$P_r = \frac{1}{1 + 1/4\left(\frac{d_2^2}{d_1^2} - \frac{d_1^2}{d_2^2}\right)^2 \sin^2 \frac{2\pi f\, l_e}{C}}$$

$$= \frac{1}{1 + 1/4\left[\left(\frac{11.94}{1.17}\right)^2 - \left(\frac{1.17}{11.94}\right)^2\right]^2 \sin^2 \frac{2\pi\, 8.67 \times 1.97}{1{,}120}}$$

$= 0.0394$ This is satisfactory but could be larger.

6. First band-pass frequency

$$f_{1bp} = \frac{C}{2\, l_e}$$

$$= \frac{1{,}120}{2 \times 1.97} = 284$$

$\frac{284}{8.67} = 31.8$ This is satisfactory.

Pipe Resonance

Although it is not the intent of this chapter to delve into pipe stress and vibration analysis, a few alerting comments appear to be in order. From this, the design engineer should be able to seek the proper qualified technical help to analyze the stresses associated with the piping layout.

It is important to investigate possible situations of harmful resonance. Although all pipes and pipe-volume combinations will resonate, it is necessary to investigate only the cases that might cause mechanical problems. This refers to a frequency that will given a objectionable beat frequency or to a length of pipe that is tuned to a stimulating frequency.

A beat can occur as the sum or difference of two frequencies. The sum or difference of any resonant frequency and a compressor driving frequency must be equal to some low value, possibly only $^1/_2$ cycle per sec or less. Therefore, only two frequencies or possible harmonics of these frequencies, which vary by a value of less than 2 cycles per sec, will be harmful as far as vibration is concerned for many, not all cases. It is important that risers be given close attention in regard to resonance. Vertical loops should be avoided. The first volume bottle or drum of the surge or filter system should be connected directly to the compressor cylinder.

If no practical single filter can be designed to filter all necessary frequencies, then it may be advisable to design two filters that have overlapping characteristics so that a broad attenuation band is yielded by the combination. The attenuation characteristics of two separate filter systems probably will be additive (i.e., if the first section showed a 30% attenuation at 15 cycles per sec and the second section yielded 10% attenuation at 15 cycles per sec, the combined filter arrangement would yield 40% attenuation at 15 cycles per sec).

The attenuation is increased as the number of sections is increased; however, economics prohibit increasing the number by greater than two, usually.

Although the following comments do not substitute for a complete analysis of each pipe system around a compressor, they serve to identify a few key points.

The fundamental resonant frequency of a pipe closed at one end:[11,12]

$$f = v_s/(4\, L_p) = (2N - 1)v_s/(4L_p) \tag{13-39}$$

This can be used for the pipe between the compressor cylinder and the surge drum.[12] The length should be such that the fundamental and second harmonic are avoided, and these should differ by several cycles, preferably more than five, from the base and harmonics produced by the compressor.

The fundamental resonant frequency of a pipe open at both ends:

$$f = v_s/(2\, L_p) = Nv_s/2L_p \tag{13-40}$$

This can be used for the pipe between the surge drum and the pipe header. The length of pipe should be such that the fundamental and third harmonic are avoided, and these should differ by at least five cycles from the base and harmonics produced by the compressor.

Everett[22] discusses pulsation associated with reciprocating compression and presents several points of analysis and diagrams primarily related to the pipe vibration.

Estimating the magnitude of gas pulsations, suction, and discharge:[22]

$$P_P = 0.0017P_L \frac{(n)(\gamma_G)(B/D)^2(S)(N)}{(\gamma_G^c)} = \text{psi} \tag{13-41}$$

where P_P = pressure pulsation magnitude, peak to peak, psi
P_L = line pressure, psi abs
C = speed of sound in fluid, ft/sec
γ_G = specific density of gas to standard air
B = cylinder bore, in.

S = piston stroke, in.
D = nozzle diameter, in.
G = number of cylinders
n = ratio of specific heats for gas, c_p/c_v
N = rotative speed, rpm

Figure 13-23 suggests recommended pulse level (peak-to-peak) pressure pulsations for acceptable pipe vibration. Figure 13-24 presents allowable machinery and pipe vibration at safe limits and damage levels, and Figure 13-25 presents allowable pressure pulsations for various pipe spans between rigid supports when a 5 mil peak-to-peak vibration is allowed at the center of the pipe spans for the pipe sizes noted.

Mechanical Considerations: Drums/Bottles and Piping

Experience has proven that pulsation drums/bottles/dampeners must be designed and fabricated using exceptionally heavy construction and good welding techniques and that the addition of support gusset plates to reinforce nozzles and other attachments is essential.

Plates and pipe inside these vessels must be of heavy design and welded (not bolted) to the vessel itself. Otherwise, vibration forces within the vessels and piping can create fatigue cracks and equipment failure, Figure 13-26. Even if applicable vessel codes do not require such strength reinforcement, it is advisable for the designer to insist on these extra rugged details.

The drums/dampeners must be provided with reinforced support plates as the vessel is anchored to a strong/heavy concrete support. If the vessel is not mounted to the compressor, extreme care must be taken to prevent vibrational cracking of the vessel as well as the connecting nozzle of the compressor. The vessel must be so anchored to prevent its own vibration due to the strong vibration/pulsations taking place inside as a part of the dampening process.

Pipe nozzles attached to the pulsation drum/dampener should be as large as practical, as short as possible, have several welded gussets along the length of the nozzle, and be welded continuously to both sides of the gusset and onto the vessel shell. The gussets should be generous and provide a condition for good transition of stresses. Obviously, all this welding on pipe and/or drum should be stress relieved. The pipe to and from the compressor and its drum/dampeners must be heavily anchored into concrete and not to building frames, floors, etc. The routing should be as straight as practical—no abrupt changes in direction and no operating shut-off valves close to any drum. Generous bends are better than 90° standard bends and T-intersections must be completely avoided. Refer to earlier discussion on recommendations for a detailed computer analysis of the entire compressor-vessels-piping relationship.

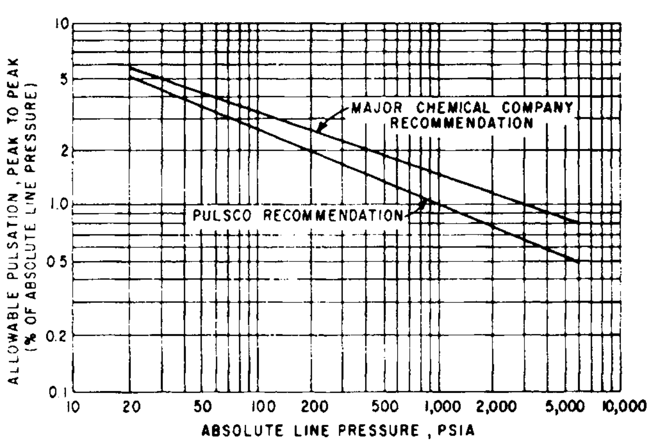


Figure 13-23. Recommended pulse level for acceptable pipe vibration (assumes adequate pipe support and anchors). (Used by permission: Everett, W. S. *Hydrocarbon Procession and Petroleum Refiner*, V. 43, No. 8, p. 117, ©1964. Gulf Publishing Company. All rights reserved.)

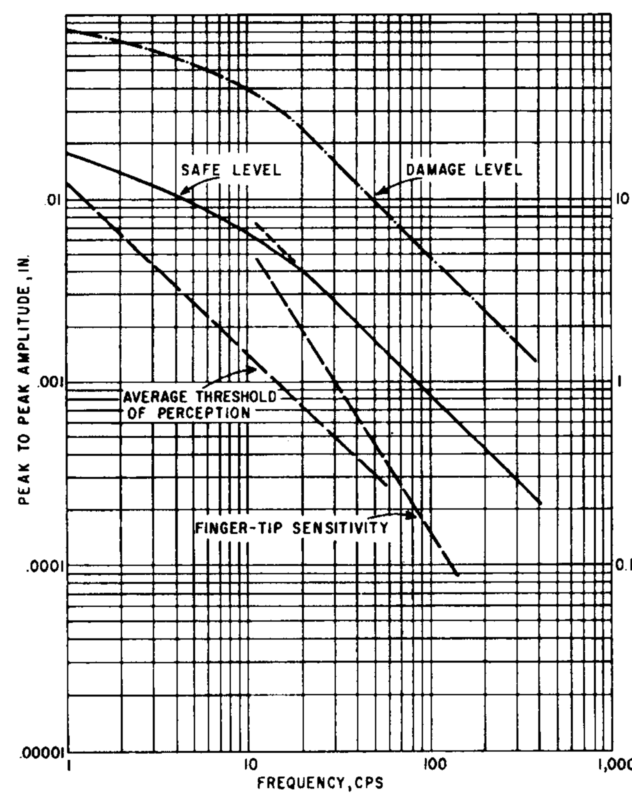


Figure 13-24. Allowable machinery and pipe vibration at safe limits and damage levels. (Used by permission: Everett, W. S. *Hydrocarbon Processing and Petroleum Refiner*, V. 43, No. 8, p. 117, ©1964. Gulf Publishing Company. All rights reserved.)

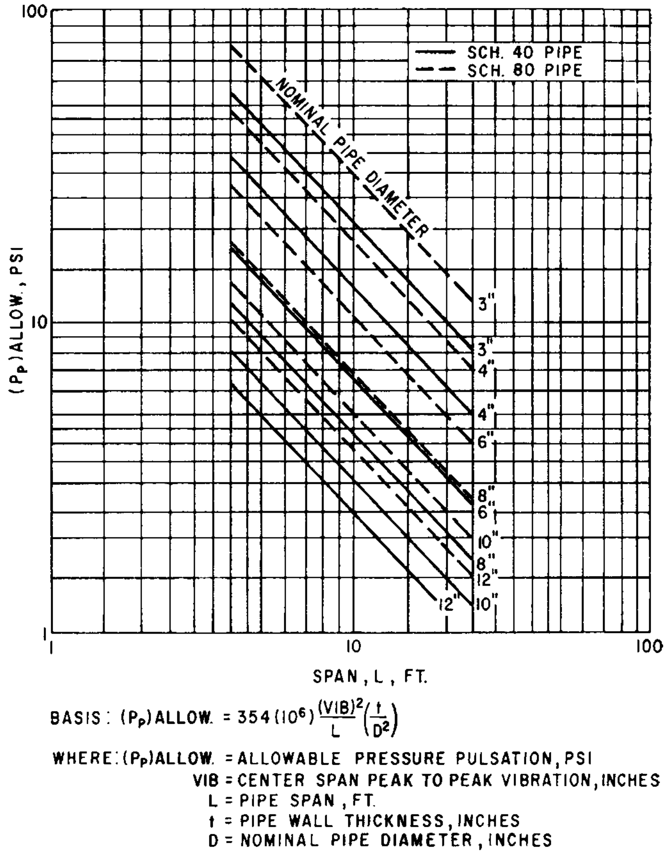


Figure 13-25. Allowable pressure pulsations for various pipe spans between rigid supports when a 5-mil peak-to-peak vibration is allowed at center of span. (Used by permission: Everett, W. S. *Hydrocarbon Processing and Petroleum Refiner,* V. 43, No. 8, p. 117, ©1964. Gulf Publishing Company. All rights reserved.)

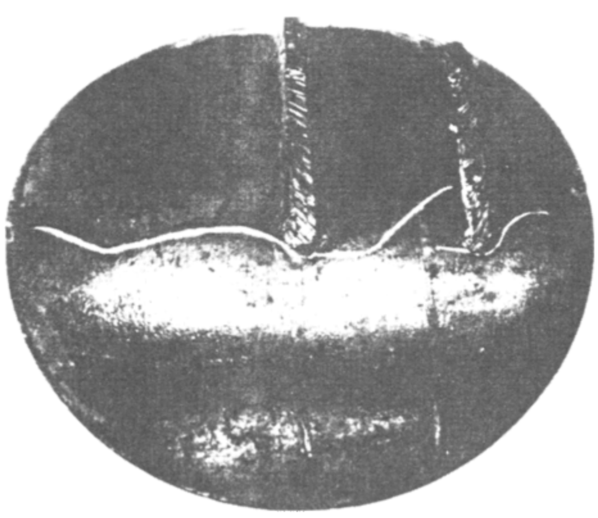


Figure 13-26. Magnaflux powder shows fatigue cracks that occurred at toes of gussets on a compressor bottle. (Used by permission: Hicks, E. J. *Oil and Gas Journal,* p. 38, July 24, 1978. ©PennWell Publishing Company. All rights reserved.)

Nomenclature

A = action of cylinder, single(1) or double(2), or cross-sectional flow area of drum, ft^2
A″ = peak amplitude of displacement
A_1, A_2 = cross-section flow area of first drum and second drum, respectively, ft^2
B = cylinder bore, in.
cfm = ft^3 gas per min
C = velocity of sound in fluid at pressure and temperature, ft/sec
C_o = orifice or short pipe coefficient:
= 0.62 for sharp edge orifice
= 0.81 for short pipe
CE = crank end of cylinder
D = surge drum I.D., in., or = nozzle diameter, in., or = nominal pipe diameter, in. (Figure 13-25)
d = pipe I.D., in., or, I.D. of bottle, in., or = orifice diameter, in.
E_{vd} = discharge volumetric efficiency, fraction or percent
E_{vs} = suction volumetric efficiency, fraction or percent
F = force of acceleration
f = frequency, cycles/sec, or friction factor as specified
f_c = cut-off frequency, cycles/sec; also used as resonant frequency
G = number of cylinders
HE = head end of cylinder
Hz = frequency of vibration
g = acceleration of gravity, 32.2 ft/sec/sec
k or n = ratio of specific heats, c_p/c_v (gas)
L = choke-tube length, ft
L_p = length of pipe, ft
L′ = surge drum length, in.
l_e = length of bottle, ft (for dished heads on vessels, the effective length of a head may be considered as $^1/_2$ (I.D.D.) or $^1/_2$ inside diameter of the dish.
M = molecular weight of gas
m = particle of mass being displaced and assuming a sinusoidal relation
N = harmonic number, number of cylinders, or = rotative speed, rpm
n = number of cylinders in parallel, or = just number of cylinders, or = ratio of specific heats of gas, c_p/c_v
P = absolute pressure, psia, at flowing gas conditions, or = may be operating pressure, psia
P_L = line pressure, psi abs
P_P = pressure pulsation magnitude, peak-to-peak, psi
P_{ra} = pulsation attenuation by area-ratio method, or = ratio of the average pulse energy flowing out of the filter to the average pulse energy flowing into the filter, or = P_r, power ratio (i.e., the percentage of power passing through the filter element).
P_r = power ratio (i.e. the percentage of power passing through the filter element). The ratio of the average pulse energy flowing out of the filter to that flowing into the filter.
PD = piston displacement, cfm, or ft^3/rev
P′/P = absolute pressure fluctuation ratio
V_1 = volume of filter bottle, ft^3

ΔP = pipe pressure drop, psi/100 eq ft, or termed differential or allowable pressure drop, psi = h.
Q = flowing cfm at 14.7 psia and 60°F
R_e = Reynolds Number
R_c = ratio of compression of compressor cylinder
rpm = revolutions per minute
S = cross-sectional area of choke-tube, ft^2, or = piston stroke, in.
s = displacement from rest position, or used as piston stroke, in.
T = absolute temperature of gas flowing, °R or °K as specified
t = time, or = pipe wall thickness, in. (Figure 13-25)
V = (or V_1) volume of surge drum, or filter bottle, ft^3
V = specific volume at 14.7 psia and 60°F, ft^3/lb
ΔV = volume rate of change factor, $= \Delta V_s$ or ΔV_d, $= f_d$ or f_s
v_s = velocity of sound in gas at temperature and pressure, ft/sec

Greek

ρ = density of gas, lb/ft^3
π = pi = 3.1416
γ_G = ratio of specific density of gas to air
λ = wave length

Subscripts

1,2, etc. = number of drums or items
s = suction
d = discharge

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Chapter 14

Mechanical Drivers

Electric Motors

Electric motors are the most common drivers for the majority of pumps, compressors, agitators, and similar equipment in the process industries. Process engineers should obtain the assistance of a qualified electrical engineer before completing motor *specifications* for the wide variety of equipment applications and respective power sources. The use of standard specifications for the various types and classes of motors is helpful and reduces repetitious details. Be certain that the type of motor is properly matched to the service, atmosphere, load characteristics, and available type and power factor of the electrical energy to drive the motor. Some basic guides are summarized, but they cannot be used as all-inclusive rules to fit all plant or equipment conditions.[4, 7, 8, 49, 54, 59, 85]

This chapter gives primary attention to the commonly used general-purpose alternating current motors, although others are noted to acquaint the engineer with the variety of motors, motor enclosures, and mechanical drive turbines available and sometimes used in the chemical and petrochemical industries. Because direct-current motors are not frequently used in the process industries (other than in heavy-duty processes), they are not examined further in this chapter.

Terminology

Although many terms are generally obvious to engineers, a few selected terms should be reviewed when referring to motor-driven systems.

- Voltage: electric pressure from power system, which is rated *at the motor;* 440–460v are most common above 1.5 hp.
- Current or full-load current: expressed as amps (amperes) and drawn by motor at a stated voltage.
- Phase: describes power source as single- (1) or three- (3) phase, which are essentially the only phases used in U.S. industry.
- Frame: refers to standard motor housing frame as dimensioned by the NEMA (National Electrical Manufacturers Association) Code.
- Cycles/sec or Hertz (Hz): the frequency of the electrical source. The U.S. standard is 60 Hz, but in some other countries this standard varies, with 50 Hz being somewhat common.
- Torque: the rotating force developed by a motor for starting and carrying loads. Some loads require high torque motors.
- Power: electric power is expressed in watts.
- Service factor: the multiplier greater than standard design (1.0) that expresses the capability of the motor to carry more continuous duty horsepower than the basis of design for the design voltage, current, and Hz. Usually, a service factor of 1.05–1.15 is designed into higher quality motors, but do not assume this; contact the manufacturer.
- Speed (motor rpm): the rotating speed of the motor when driving the "name plate" horsepower. For induction motors, a difference exists between the synchronous speed of the motor (the speed of the revolving magnetic field) and the operating speed, which is called *slip.* For example, a motor rated at 1,800 rpm usually will operate at 1,750 or 1,760 rpm due to slip. This is important to recognize when rating the actual output of a pump (see Vol. 1, Chapter 3) or a compressor, as the actual operating speed under load must be taken into account.
- Motor horsepower: the mechanical shaft output horsepower, which represents the actual horsepower *delivered* to the drive-consuming equipment. Thus, for example, the horsepower input to an agitator or mixer assembly is the horsepower at the motor shaft connected to and cumulatively drawn by the gear box-to-agitator shaft and may *not* be the (1) motor nameplate horsepower or (2) rated horsepower of the gear box or agitator design horsepower.

Refer to the many details of standardization, applications, and design features in NEMA Standard MG1-1993, Rev. 1995 (*Motors and Generators*)[31] and the safety and environmental requirements of the National Fire Protection Association Codes.[28, 29, 40, 42]

Load Characteristics

The characteristics of the loads on motors vary widely with the application. The three basic parts are

1. Work load. This usually can be calculated by the work required at the application, such as lifting loads with a crane, compressing a gas, etc.
2. Friction load. This load is determined by the manufacturer of the equipment being driven. In one actual case in driving a centrifugal compressor handling a gas with high specific volume (requiring an unusually large frame), the friction load was 50 hp out of a total of 350 hp. Belts, gears, and so on, must be considered in this load.
3. Inertia load (torque). This is usually determined by the manufacturer of the driven equipment and is evaluated by determining the time for the machinery to drift to a stop, being retarded by its own friction.[37] Peak, fluctuating, or shock loads and their cycles or repetition must be considered.

Quoted from Westinghouse Motor Co. Handbook:[46]

"Referencing to time in seconds, minutes, etc. then convert to horsepower:

$$\text{horsepower} = \frac{(\text{ft-lb torque})(\text{rev/min})}{5{,}250} \tag{14-1}$$

For one minute, one horsepower will lift 33,000 pounds one foot, or as a unit of power, this equals a rate of 33,000 foot-pounds of work per minute, or 550 foot-pounds per second. One horsepower equals 746 watts of power. The resistance to flow of electrical current through a wire or conductor is termed an ohm. One ohm is the amount of resistance in a wire through which one ampere of current flows under one volt of electrical pressure. Useful relationships are:"

$$\text{Volts, } E = P/I = (I)(R) \tag{14-2}$$

$$\text{Amps, } I = P/E = E/R \tag{14-3}$$

$$\text{Watts, } P = (E)(I) \tag{14-4}$$

$$\text{Ohms, } R = E/I \tag{14-5}$$

Energy is converted from electrical energy into mechanical energy using the power from electrical transmission lines.

Basic Motor Types: Synchronous and Induction

See Figures 14-1A–E and 14-2A–D.

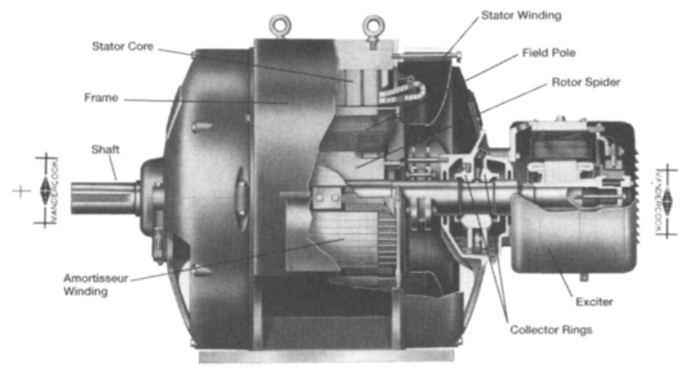

Figure 14-1A. A six-pole (1,200 rpm at 60 cycles) synchronous motor with direct-connected exciter. (Used by permission: *E-M Synchronizer,* Issue 200 SYN-42, ©1955. Dresser-Rand Company.)

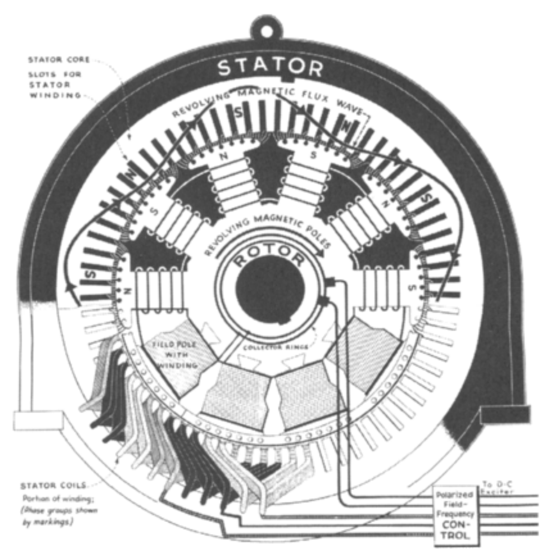

Figure 14-1B. Electrical connections in synchronous motors. (Used by permission: *E-M Synchronizer*, Vol. 7, No. 1, ©1944. Dresser-Rand Company.)

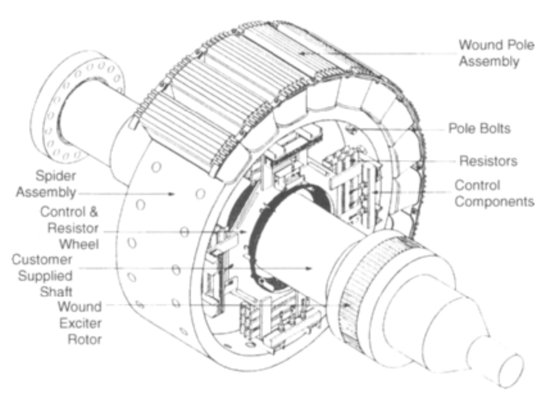

Figure 14-1C. Engine-type synchronous motor. Compressor manufacturer furnishes the motor shaft and motor bearings. A solid hub is used for maximum mechanical strength. (Used by permission: Bul. WMC-04A-94. TECO-Westinghouse Motor Co.)

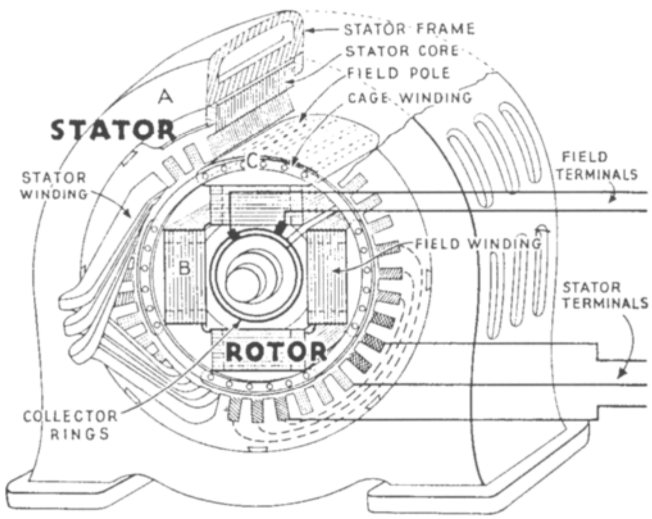

Figure 14-1D. Cutaway and partly developed illustration of 4-pole synchronous motor to show elements of construction and external wiring. (Used by permission: Bul. 200-Tec-1120 ©1955, Dresser-Rand Company.)

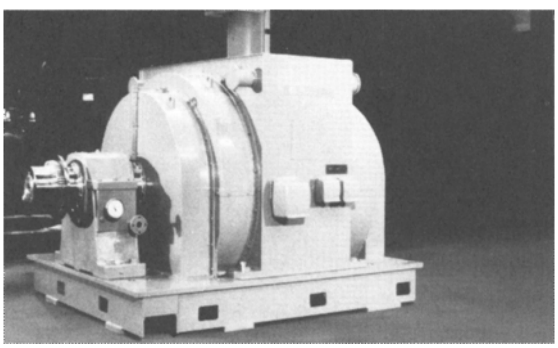

Figure 14-1E. Heavy-duty synchronous motor. (Used by permission: Bul. WMC-04A-94. TECO-Westinghouse Motor Co.)

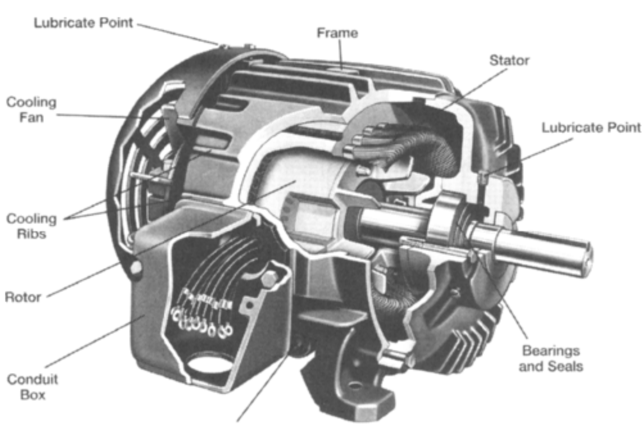

Figure 14-2A. Alternating current (AC) induction motor cross-sectional view of 220/440v, 3-phase, 60-cycle, totally enclosed fan-cooled standard motor. (Used by permission: Reliance Electric Co., Div. Rockwell Automation.)

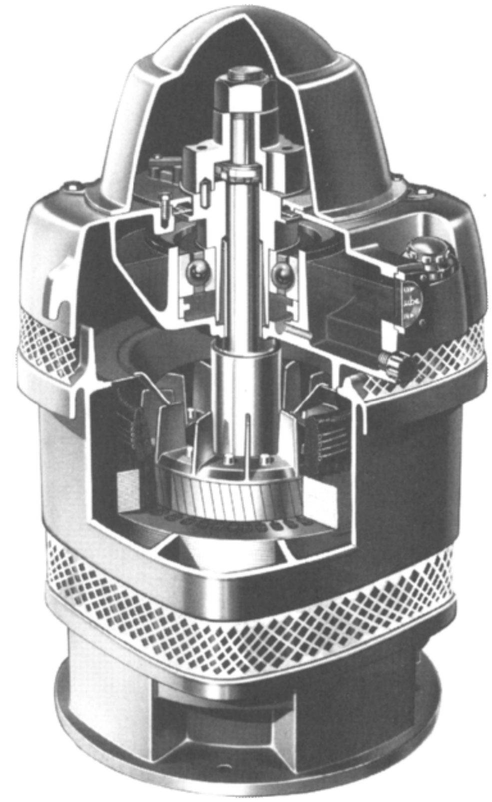

Figure 14-2B. Typical Holloshaft® construction for vertical induction motor, weather-protected construction. (Used by permission: Bul. BR 509-91B ©1991, U.S. Electrical Motors, Div. Emerson Electric Co.)

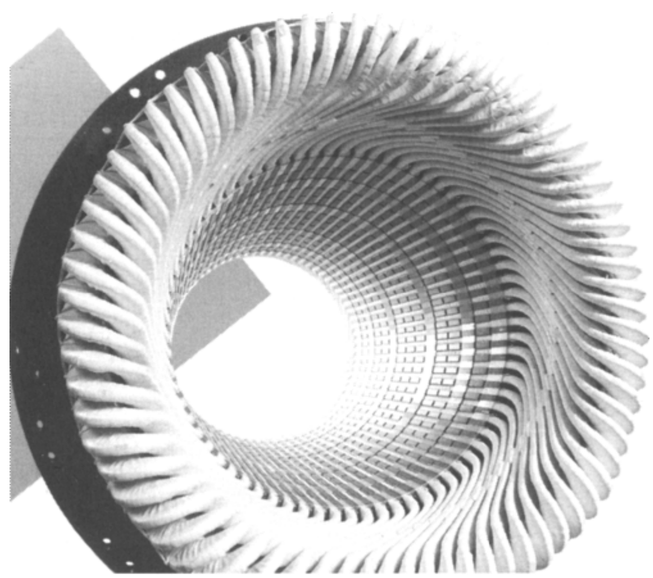

Figure 14-2C. Siemens form wound stator with Siemens MICLAD-sealed epoxy mica insulation system. (Used by permission: Bul. M&DD 3323 ©1994, Siemens Corporation, Motors and Drives Division.)

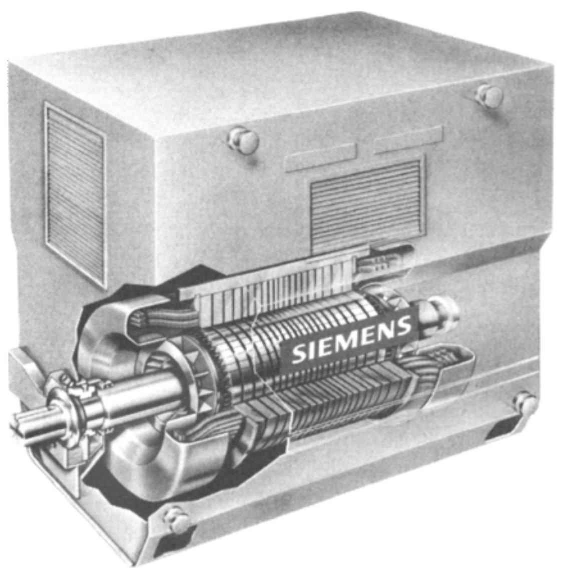

Figure 14-2D. Large horizontal induction motor. Motors can meet API 541 requirements and ISO 9001 Certification for petroleum and chemical industry applications. (Used by permission: Bul. M&DD 3226-3. Siemens Corporation, Motors and Drives Division.)

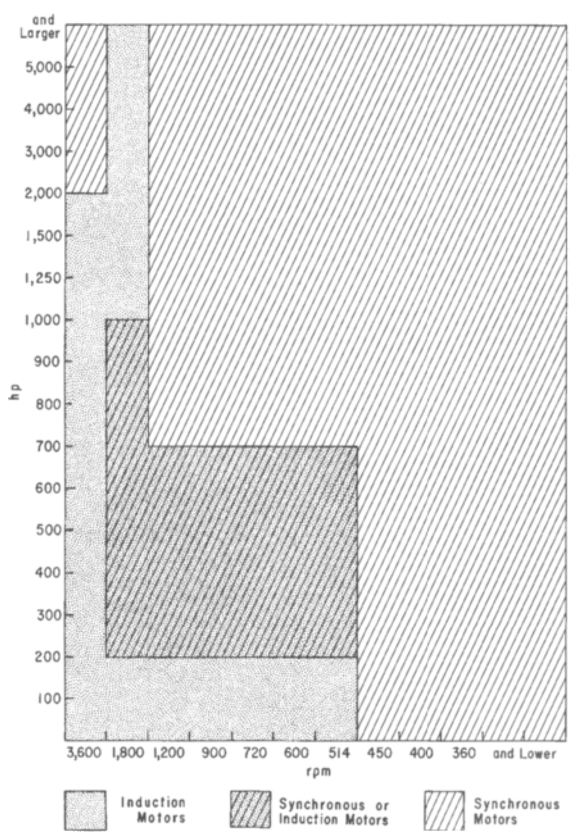

Figure 14-3. General areas of application of synchronous motors and induction motors. (Used by permission: *E-M Synchronizer*, 200-SYN-33. ©Dresser-Rand Company.)

NEMA motor classifications from Reference 31 for usual chemical and petrochemical applications are as indicated in Figure 14-3.

I. Open motors
   Open drip-proof
   Open drip-proof guarded
II. Weather protected
   Weather protected I
   Weather protected II
III. Totally enclosed
   Totally enclosed, nonventilated
   Totally enclosed, fan-cooled
   Totally enclosed, fan-cooled, tube type
   Totally enclosed, water-air cooled
   Totally enclosed, force ventilated, or air-to-air cooled
   Totally enclosed, explosion-proof
   Totally enclosed, dust-ignition proof
   Totally enclosed, water-proof

A. Synchronous Motors. See Figures 14-1A–E.

1. Higher efficiency at any speed rating, particularly medium and low speeds.
2. Suitable for direct connection to medium- (less than 1,000 rpm) and low-speed applications.
3. Available with unity or a leading power factor.
4. Constant speed, no slip.
5. Less sensitive to bearing wear or misalignment than induction motor.
6. Require source of direct-current excitation.
7. Costs of control equipment higher than for induction motor.

B. Synchronous Motors. See Figures 14-1A–E. From an assembly standpoint, three types of synchronous motors exist:[55]

1. Engine type: consists of a stator and rotor, and the motor shaft and bearings are not furnished by the motor manufacturer. These are usually used to drive compressors.
2. Pedestal-bearing motors: consists of shaft, pedestal-type bearings, rotor, stator, and base.
3. Bracket-bearing motors: complete motor with bearings bolted to ends of motor frame, ready to mount on foundation.

For all types, avoiding torsional vibration problems[61] is essential.

The discussion to follow is reproduced by written permission of TECO-Westinghouse Motor Co.[46]

Synchronous Motor Theory

"The synchronous motor is a **constant-speed** machine. Unlike the induction motor which inherently has slip from losses, the synchronous motor uses an excitation system to continually keep the rotor in synchronous speed with current flowing through the stator. Within its designed torque characteristics, it will operate at synchronous speed regardless of load variations.

The synchronous motor starts and accelerates the connected load like an induction motor, and is designed to give the best possible induction-motor operation during starting, consistent with good synchronous-motor performance.

In general, the stator of a synchronous motor is quite similar to the stator of an induction motor. The polyphase current flowing in the stator winding sets up a rotating magnetic field in the same way as the induction motor.

The difference between synchronous and induction motors can be seen in the rotor. The synchronous rotor is generally equipped with **salient poles**. For starting, a winding which corresponds to the squirrel-cage winding of an induction motor is placed on the pole heads. The winding is usually referred to as the **damper winding** because it helps damp out oscillations of the rotor about its normal position, caused by load pulsations when the motor is operating at synchronous speed. The damper winding consists of a number of bars which are pushed through partly closed slots in the pole head. The ends of the bars are brazed to copper segments.

Besides the damper winding, there is also a **field-coil** winding on each rotor pole. It is excited from a DC source (see Product Focus section for different kinds of exciters) when the motor is running at synchronous speed. During starting, the field winding is almost always short circuited through a starting or "**field-discharge**" resistor.

The damper and field windings each produce an effect on accelerating torque. The resistance of the damper winding is usually selected to produce its maximum torque during the first part of the starting period and a field-discharge resistor is selected so that the field-circuit torque peaks near pull-in speed.

Low inrush current can normally be obtained with a synchronous motor, because the starting winding is not used as the running winding.

Power-factor can be rated at unity, leading, or even lagging. The synchronous motor can supply corrective kvar to counteract lagging power factor caused by induction motors or other inductive loads.

Torque is a force that tends to produce rotation.[46] If a force of 50 pounds is applied to the handle of a 2' crank, this force produces 100 pounds of torque (twist-ability) when it is at right angles to the crank arm. Torque may be converted into **horsepower** when the element of **time** is considered. If the torque in foot-pounds is measured over a given period of time it becomes foot pounds per second or minute, and may be converted into horsepower.

When torque, in foot pounds, is multiplied by the speed in revolutions per minute, it may be divided by the constant 5,250 to find horsepower, using this formula:

$$\text{horsepower} = \frac{\text{foot pounds torque} \times \text{revolutions per minute}}{5{,}250}$$

In terms of a minute, one horsepower is the power required to lift 33,000 pounds one foot. A unit of power, then, is equal to a rate of 33,000 foot-pounds of work per minute, or 550 foot-pounds per second."

From Dresser-Rand[55] by permission: "the stator of the synchronous motor has a distributed polyphase winding which is connected to the power supply. The rotor has an even number of projecting field poles each wound with a coil. The field coils are excited by direct current from an exciter or from some other source of direct current. Imbedded in the faces of the field poles is the squirrel-cage starting winding. When current is applied to the stator of the synchronous motor, it creates a revolving magnetic field which acts on the cage winding the same as in a squirrel-cage induction motor, to start and accelerate the rotor. At synchronizing speed, usually about 95% synchronous speed, d-c excitation is applied to the field winding and motor pulls into synchronism, the north and south poles of the rotor 'locking in with' and rotating in synchronism with the revolving poles of the stator magnetic field." Oscarson[65] presents a helpful discussion of the application of synchronous motors to reciprocating compressors, Figures 14-4A and 14-4B.

Selection of Synchronous Motor Speeds

Range: 3,000–3,600 rpm. These speeds are not often used due to the high cost of construction. Economics favor an induction motor at a slower speed with a gear speed increaser or an induction motor in the 2,500–22,000 hp range.

Range: 900–1,800 rpm. These speeds are used for pumps above 3,000 hp and for centrifugal compressors using speed increasers. The efficiency, power-factor correction, and other factors may favor motors below 3,000 hp.

Range: 514–720 rpm. These speeds are for motors selected above 4 hp/rpm.

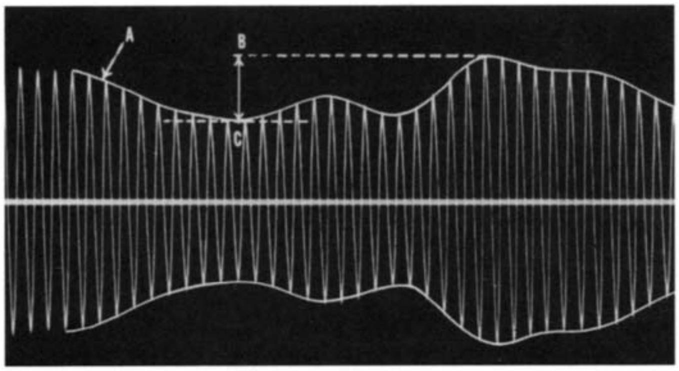

Figure 14-4A. Oscillogram shows variation of current to a synchronous motor driving a reciprocating compressor, The compressor is two-cylinder, horizontal, double-acting, and operates at 257 rpm. Line "A" is the envelope of the current wave. Difference B–C is current variation. Value B–C divided by the rated full load current is the percentage of current variation. (Used by permission: Oscarson, G. L. *E-M Synchronizer*, 200 SYN 52, p. 11. ©Dresser-Rand Company.)

Range: below 514 rpm. Down to 1,000 hp, these motors are considered due to higher efficiency, an improved power factor, and possibly for the price. For high voltages, 4kv and higher, the synchronous motor becomes economical at even lower hp.

Squirrel-Cage Induction Motors. See Figures 14-2A–14-2D.

This polyphase (usually three-phase compared to single-phase) motor is the basic "work horse" of the process industries for general-purpose applications. Characteristics of polyphase squirrel-cage induction motors are

1. Simple, rugged construction.
2. Available with 3,600 synchronous rpm at 60 cycles (two-pole) and 1,800 synchronous rpm at 60 cycles (four-pole) in large horsepower ratings, and 1,200 synchronous rpm at 60 cycles (six-pole). For 50 cycles

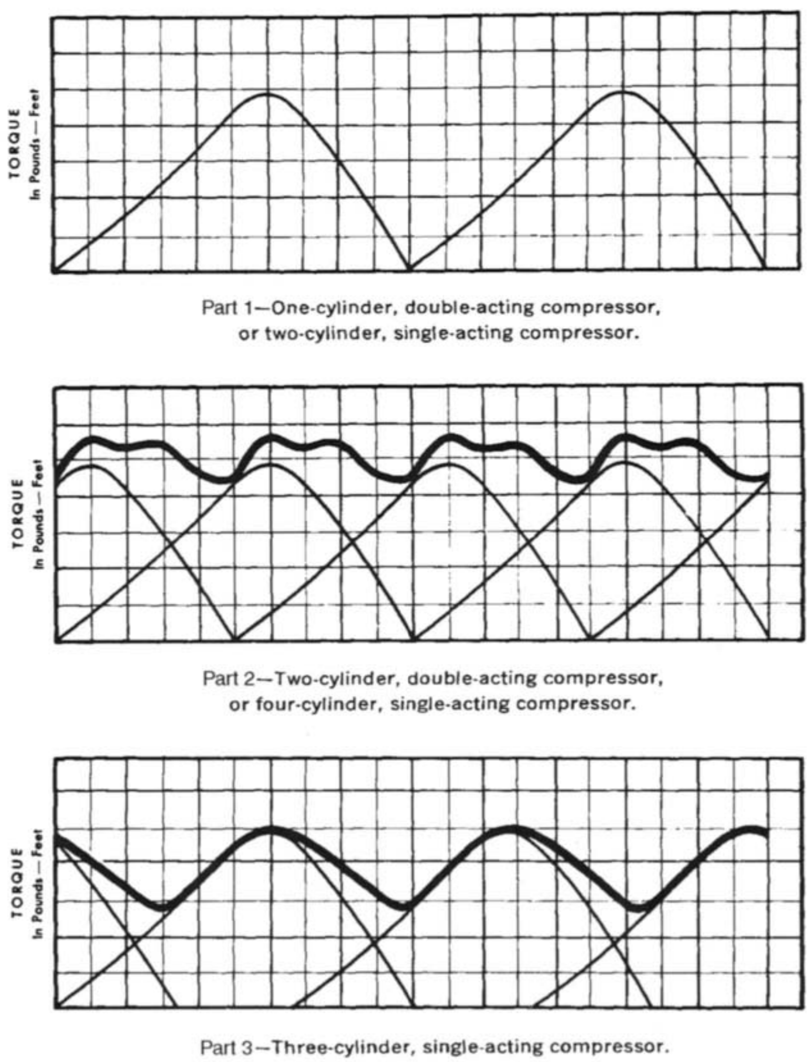

Figure 14-4B. Crank effort varies with the type of compressor. (Used by permission: Oscarson, G. L. *E-M Synchronizer*, 200 SYN 52, p. 11. ©Dresser-Rand Company.)

power, the speed is 0.8334 times the cycle speed (or 16.66% less).
3. Less efficient than synchronous motors, primarily due to slip.
4. A low power factor below 500 rpm.
5. High starting current.
6. Low power factor on starting and at fractional loads.

Alternating current motors are either single-phase current or polyphase (three-phase current). The polyphase motors are

1. Squirrel cage
2. Wound-rotor
3. Synchronous

The running speed of these motors is

rpm = [(100 − s)/100] [120 f/n}

where s = percent slip
f = frequency, cycles/sec
n = number of poles

For squirrel-cage motors, slip ranges to 5% of the synchronous or constant speed. For variable-frequency motor drives, see Reference 89.

The calibration curve (three-phase induction motor) of Figure 14-5 is essential to determine the load change based on the current change for motors below 200 hp or with load variations of more than 5–10%.[48]

Induction Motors. See Figures 14-2, 14-6, 14-7, 14-8A, and 14-8B.

This is the most common kind of alternating current motor used by the industry.[75] The term *squirrel cage* is usually

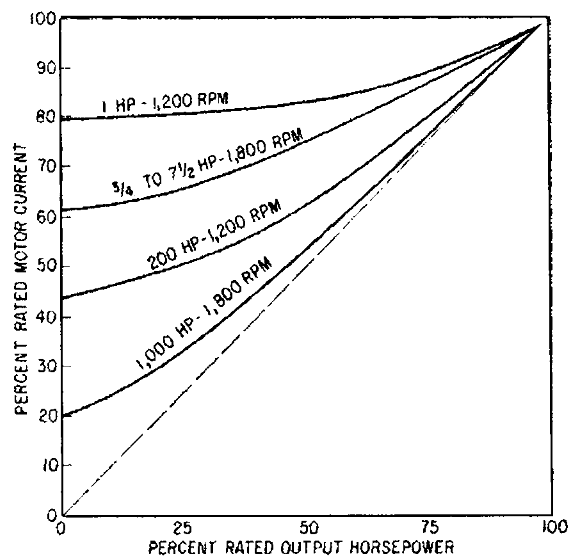

Figure 14-5. Dashed line denotes a linear or direct relationship between input current and output horsepower for induction motor. Solid curves show how far from reality that relationship is for actual motors. Without such a *calibration curve* for the motor in question, determining the change in load from a measured change in current is not possible. (Used by permission: Nailen, R. L. *Hydrocarbon Processing*, p. 205, Sept. 1973. ©Gulf Publishing Co. All rights reserved.)

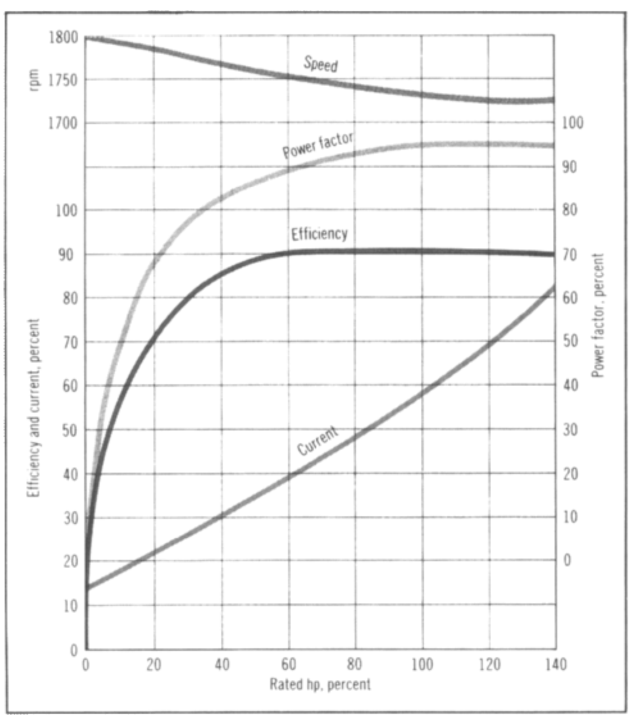

Figure 14-6. Typical squirrel cage induction motor performance curves. (Used by permission: Bell, C. J., and Hester L. R. *Heating/Piping/Air Conditioning*, p. 51, Dec. 1981. ©Penton Media, Inc. All rights reserved.)

associated with induction motors because the electric circuit resembles a rotating squirrel cage. The term *synchronous* speed for an induction motor refers to the fixed-design operating speed and should not be confused with the synchronous speed of a synchronous motor. The induction motor synchronous speed drops off due to load, and this is termed *slip*. The following description is reproduced by permission from TECO-Westinghouse Motor Co.:[46]

Induction Motor Theory

"Electricity is the result of electrons flowing through a conductor, or wire. Current is the flow of electricity when a pressure or voltage is applied to the conductor. If the wire is formed into a loop or coil, the coil is placed around or in a steel core and voltage is applied to the coil, current will flow through the wire and produce magnetic flux. This is what happens in the stationary part of the induction motor.

This stationary part of the motor includes the iron, coils, and magnetic circuit and is called the **stator**. The **rotor**, or squirrel cage, is a number of bars all connected together.

The rotor is placed in the center of the stator. An A.C. (alternating current) power supply is applied (voltage) to the stator coils. A magnetic flux is produced in the small space between the rotor and stator

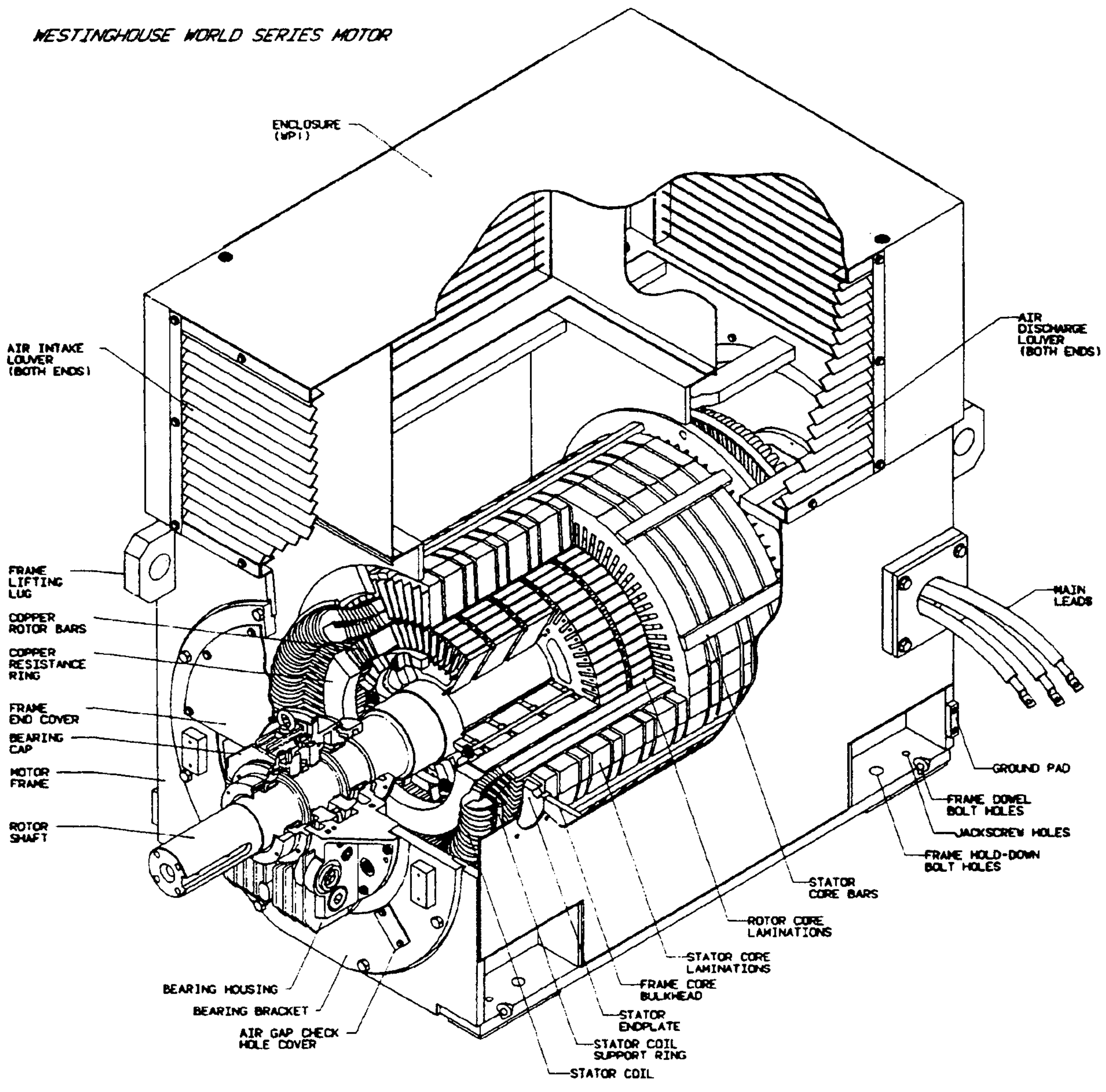

Figure 14-7. Internal details of large induction motor. (Used by permission: *Reference Handbook*, TECO-Westinghouse Motor Co.)

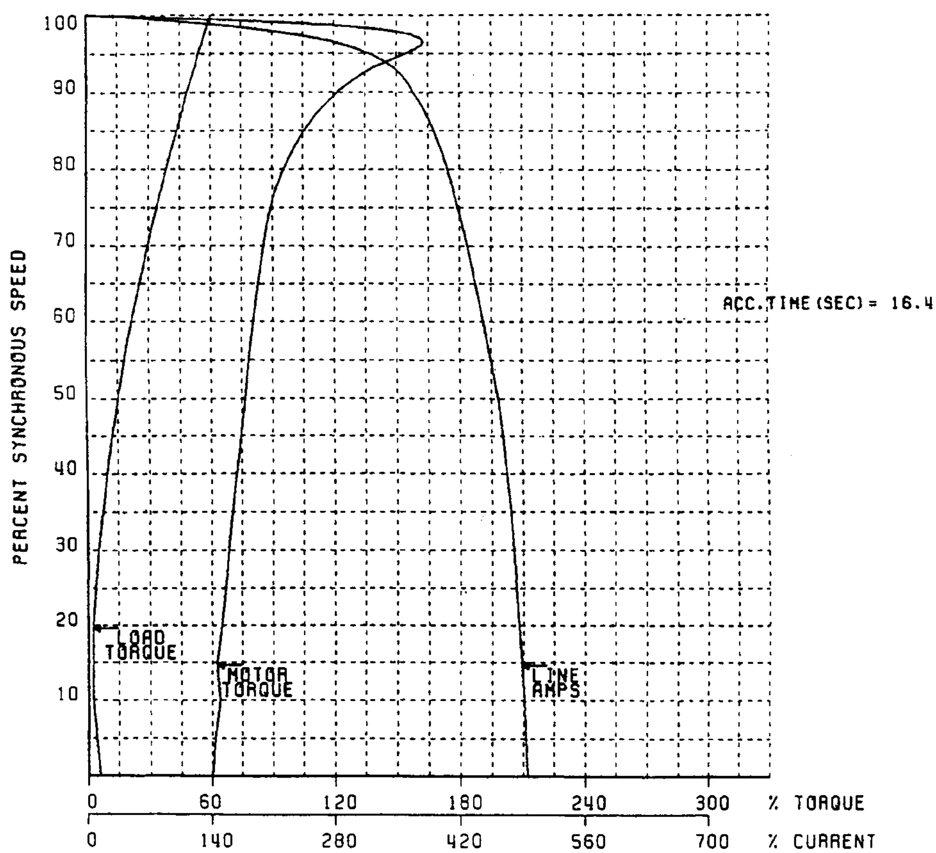

Figure 14-8A. Induction motor starting characteristics (calculated) at 80% line voltage. (Used by permission: Curve: MPRO822890. TECO-Westinghouse Motor Co. 8/89)

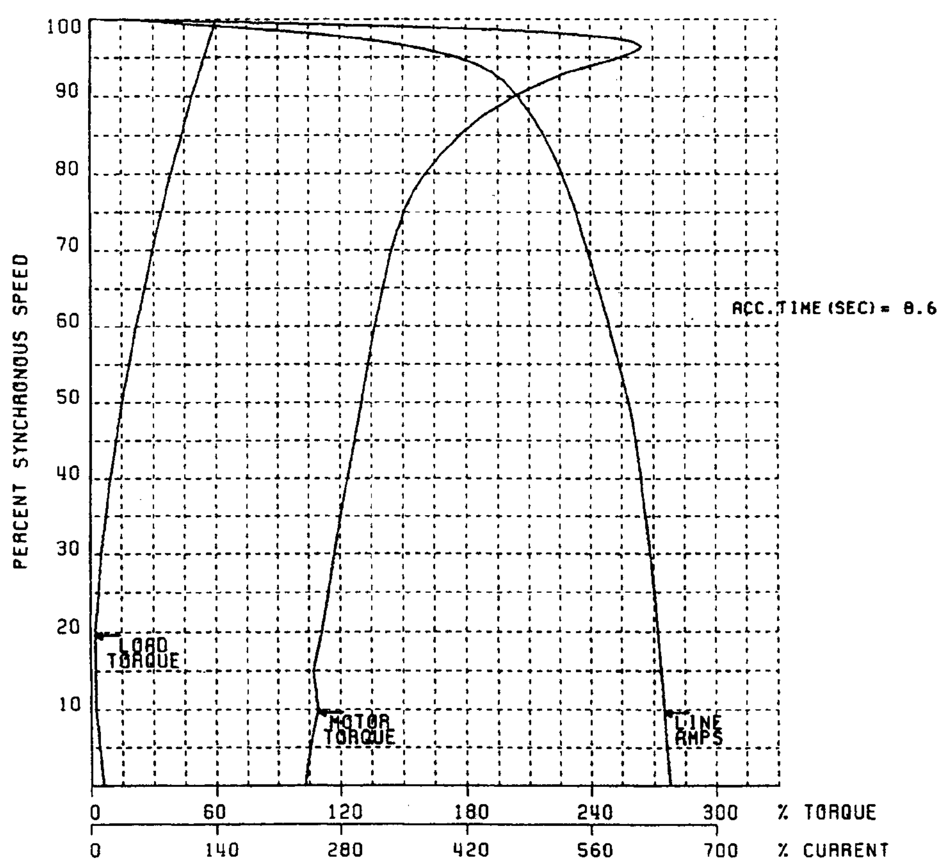

Figure 14-8B. Induction motor starting characteristics (calculated) at 100% line voltage. (Used by permission: Curve: MPRO82289D. TECO-Westinghouse Motor Co. 8/89) (**air gap**). The flux also penetrates the squirrel cage rotor bars. Because the bars are all connected at the ends (**end-rings**) there is an electrical circuit in which current flows. This current causes heat and the bars get hot, but do not melt.

When the motor first starts there is a sudden surge of current into the stator winding (**inrush current**). This is usually 6 to 6.5 times the amps the motor sees when it is running. This causes similar surge in the squirrel cage rotor bars, so great care must be taken to size all the bars, WMC uses copper bars, correctly. This magnetic circuit in the rotor induces a current; thus, the name **induction** motors.

One of the basic electromagnetic theories is that when a flux exists, a wire in which current flows (rotor bars) will be subjected to a force (**motion**). By applying the voltage around the circumference of the stator, a rotating magnetic field travels through the stationary core (stator). But, because the force on a portion of the rotor bars is also magnetically rotating, physical rotation of the rotor also occurs. In order for this rotation to occur, the rotor must be mounted on a shaft which is placed in bearings at the ends of the stator.

The rotating magnetic field in the stator travels around the stator at what is called **synchronous speed**. By grouping stator coils together in what is called **poles**, the motor rotor can be designed to turn at a certain speed (revolutions per minute/rpm). On an induction motor the number of poles cannot be seen or counted without the drawings.

Because the squirrel cage is magnetically connected to the stator flux, it also rotates at near synchronous speed. It is only "near" because its rotation is slightly slower than the synchronous speed in the stator. The difference is called **slip**. Slip is caused because of less than perfect magnetic coupling in the air gap, bearing friction, air resistance, and most importantly by applying load. Most induction motors experience a low slip, usually 1 to $1^1/_2$ percent difference, but high slip motors can be designed for a 5–13% difference.

The motor synchronous speed is determined by the **frequency** of the rotating field in the stator (60 cycles in the U.S.) and the number of **poles** (coil connections in the stator) such that motor synchronous speed

$$(\text{rpm}) = \frac{120 \times \text{frequency}\,(60)}{\text{number of poles}}"$$

Number of Poles	rpm
2	3,600
4	1,800
6	1,200
8	900
10	720
12	600
14	514
16	450
18	400
20	360

See Tables 14-1 and 14-2.

Typical induction motor starting characteristics for 80% and 100% of line voltage are shown in Figures 14-8A and 14-8B.

Table 14-1
(See Figure 14-3.)
General Application of Synchronous and Induction Motors*

	rpm	
Horsepower	**Induction**	**Synchronous**
0–200	514–3,600	450–lower
200–2,000 up	1,800–3,600	–
200–700	514–1,800	514–1,800
200–700	–	450–lower
700–1,000	1,800	1,800
700–1,000	–	1,200–lower
1,000–2,000 up	–	1,200–lower
2,000–5,000	1,800	3,600–lower

*60 Hz.

Table 14-2
Rpm for Synchronous and Induction Motors

Number of Poles	Cycles 50 Rpm	Cycles 60 Rpm	Standard Horsepower Induction	Standard Horsepower Synchronous
2	3,000	3,600	$1\,^1/_2$–5,000	1,000–larger
4	1,500	1,800	1–5,000	30–5,000
6	1,000	1,200	$^3/_4$–5,000	30–10,000
8	750	900	$^1/_2$–10,000	30–30,000
10	600	720	$^1/_2$–10,000	40–30,000
12	500	600	$^1/_2$–10,000	50–30,000
14	429	514	3–22,500	100–30,000
16	375	450	3– any practical hp	20–10,000
18	333	400	50–	same as above
20	300	360	50–	
22	273	327	50–	
24	250	300	50–	
26	231	277	75–	
28	214	257	100–	
30	200	240	125–	
32	188	225	200–	

$$\text{rpm} = \frac{120\,(\text{frequency})}{\text{no. poles}}$$

The moment of inertia of a driven load is very important in the proper selection of a motor for those situations in which the motor accelerates a heavy load or makes frequents starts.[52] Moment of inertia is WR^2 as lb-ft^2. This is the product of the weight of an object and the square of the radius of gyration (i.e., $WR^2 = Wk^2$, where W = weight in lb, and k is the radius of gyration in ft; where R = radius of a disk, and $k = R/(2)^{1/2}$. Other details exist for calculating the radius of gyration of other shaped objects.[52]

For centrifugal compressors, any standard motor of NEMA design B rating should be satisfactory.[59] For reciprocating compressors, the compressor load requires a variable torque during each revolution, which causes current fluctuation in the driving motor. The stator current fluctuations can be damaging to the motor and are limited to 60% of full-load current (NEMA MG-1-20.82).[59] The number of successive starts of the motor is also limited per NEMA MG-1-20.43.[59] Typical squirrel-cage induction motor performance curves are shown in Figure 14-6.

Duty

The chemical and petrochemical industries specify continuous duty service. This means that the motor can operate indefinitely when handling the specified horsepower (rated) load at the proper voltage. To specify less than continuous duty is uneconomical and not good design practice.

The induction motor is usually used in industrial service; however, many important large horsepower applications still exist for the synchronous motor.

Types of Electrical Current

Direct Current (DC). This current is transmitted for industrial uses only in exceptional situations. The most common sources of direct current are storage batteries and industrial devices called *rectifiers,* in which alternating current is changed (rectified) to direct current, as is used in electrolytic cells[81] for the manufacture of chlorine gas, magnesium, aluminum, and a few other chemicals. The direct current is flowing from the source through the user application and back to the source, in one direction.[46] The motor is primarily used for speed control of selected equipment.

Energy is converted from electrical energy into mechanical energy using the power from electrical transmission lines.

For alternating current:

Operating cost

$$= \frac{(\text{wattage})(\text{hours used})}{1{,}000} = \text{kwh (kilowatt-hr)} \qquad (14\text{-}6)$$

$$= (\text{kwh})(\text{rate, cents/kilowatt-hr})$$

Alternating Current (AC). This current flows back and forth in the wire between the source and user application; see Figure 14-9. Alternating current is produced by a device called a generator and is the most common type of current in homes, offices, industrial factories, and other applications.

This alternating current can be generated at a power plant and distributed by transmission lines to distribution stations where a transformer turns the high-transmission voltage into a lower voltage, and this power will usually be again transformed to a lower voltage for use by local equipment. Normal use voltage for residences, most industrial and business offices, and some specific plant instruments and small equipment is 115–120v (up to 130v). Larger motors of 1 $^1/_2$ hp and up to about 250 hp normally use 230/460v to 2,300–4,000v and even higher for some special equipment. Although the voltage has been changed several times from the generating station (or power plant), the frequency of the alternating current still remains at 60

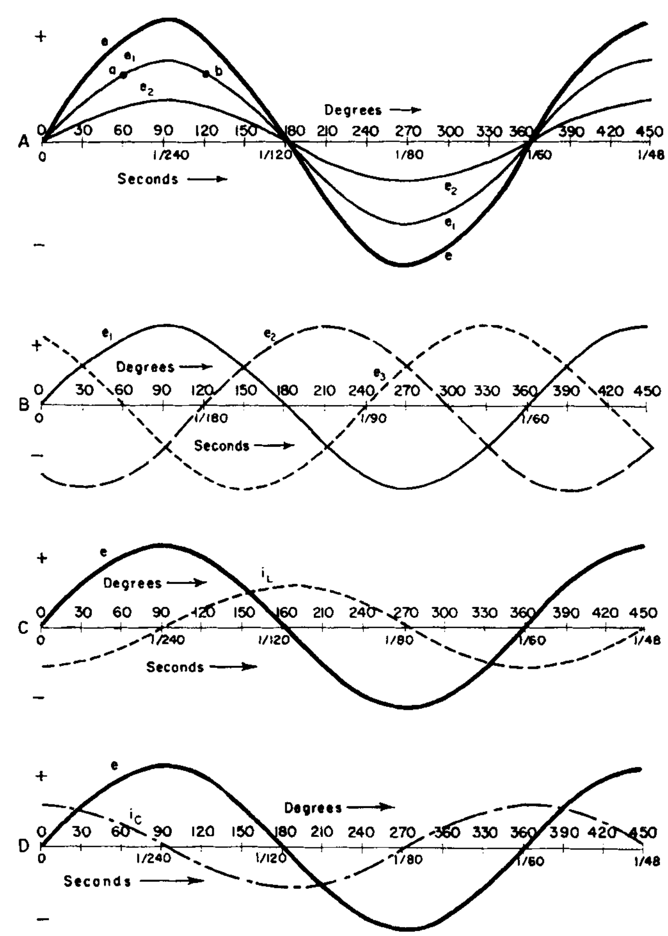

Figure 14-9. Phase Relationship in alternating circuits. (Used by permission: Peach, N. *Power*, p. 138. July, 1957. ©McGraw-Hill, Inc. All rights reserved.)

cycles/sec (in the United States). The frequency is determined by the speed of the engine or turbine driving the generator's shaft.[46]

The generator action produces the alternating current, i.e., a coil mounted on a shaft rotates between the north and south poles of electromagnets mounted in the frame. As the coil moves toward the north magnetic pole, the electric current is produced in the wire, and the strength of the current reaches a maximum when the center of the coil is in line with the center of the north pole of the magnet. Then as it continues to rotate, the current decreases, reaching 0 when the coil is halfway between the north and south magnet poles. As the coil continues to rotate toward the south pole, the current increases, but at the time the direction of the current in the wire is reversed.[46]

The continually changing directions, as the coil moves past the north and south poles at high speeds, produces an electric current that changes direction or alternates twice during each revolution of the generator shaft (driven by steam turbine, gas turbine, gas engine, hydraulic turbine, or other independent source of mechanical power). The number of cycles through which the current passes in a given period of time is the measure of the number of revolutions the coil makes in the same time;[46] this is the frequency of the current. The standard in the United States for most alternating current is 60 cycles/sec or 60 Hz, but in a few places 50-cycle current is generated to match certain equipment. Also see reference 88. When a single coil is rotating on a shaft, then only one current is generated, single-phase current. When three coils of wire are mounted on the shaft and are equal distances apart, each coil produces an alternating current, and this is termed three-phase current (or polyphase). As the coils are equally spaced around the circle of rotation, each coil will have a different amount of current at a particular moment.[46] The three-phase current is most popular for use in U.S. industry; also see reference 58.

Fractional and small integral horsepower motors usually operate at 115–230v (AC). Often these come with preassembled or predesigned packages of equipment (or systems) purchased for a specific purpose, and the process engineer may not be involved; however, it is essential that the company electrical engineer review these motors and their control specifications for both safety and quality."

The details of alternating current phase relationship are presented by Peach, N.[58] and used by permission:

"Strictly speaking, a phase is any point on such a curve corresponding to a certain number of degrees of rotation. Points *a* and *b* on curve e_1 are different phases of the curve, sketch *A*, . . . [see Figure 14-9]. Degrees of rotation correspond to an actual amount of time. On a 60-cycle curve, one cycle or 360 deg equal $^1/_{60}$ sec, 180 deg equal $^1/_{120}$ sec, etc. Difference in phase represents difference in time. Two curves are *in phase* if corresponding points on each curve occur at the same time, as on curves e_1 and e_2 in sketch *A*. Note that these voltages are at a positive maximum at the same time (90 deg or $^1/_{240}$ sec), both zero at 180 deg or $^1/_{120}$ sec, both a negative maximum at 270 deg or $^1/_{80}$ sec, etc. Suppose that e_1 and e_2 represent the voltages of two generators in a series. Then at any instant the total voltage would be the sum of the two voltage waves, shown by the resultant curve *e*.

Now if a generator were connected in such a manner that it produced three emf waves corresponding points on which were 120 deg apart, a *three-phase* system of voltages would be generated. Considering the 3-phase 60-cycle emf curves in sketches *B*, e_2 begins to increase in a positive direction 120 deg or $^1/_{180}$ sec later than e_1, and e_3 begins 120 deg later than e_2 (or $^1/_{90}$ sec later than e_1). This is our familiar 3-phase system. The resultant of the three emfs is always zero, that is, the sum of the emfs in the positive direction is equal to the sum of the emfs in the negative direction. This makes it possible to transmit 3-phase current in a 3-wire system. If the three wires are labeled *a*, *b* and *c*, e_1 appears as a voltage between *a* and *b*, e_2 between *b* and *c*, and e_3 between *c* and *a*. Each of these voltages (e_1, e_2, e_3) constitutes a *single-phase* voltage, as would *e* in sketch *A*. Single-phase voltage obtained from a 3-phase system is referred to as a "phase" of the 3-phase system, as e_1 might be called "phase *a–b*." To add to the confusion, the three conductors of the system are often called "phase conductors" or "phase wires," and may be labeled "phase *a*, phase *b*, phase *c*." It's a common error to confuse *phase voltage* with *phase conductor*, and suppose that by disconnecting one phase conductor (let's say phase conductor *c*) you disconnect one phase voltage and leave a 2-phase system. Actually, disconnecting phase conductor *c* disconnects two phase voltages ($e_2 = b - c$ and $e_3 = c - a$), leaving only $e_1 = a - b$.

Current produced by a voltage wave can also be represented by a curve. If impedance of the circuit consists only of resistance, the current will be in phase with the voltage. If the impedance is wholly inductive reactance, the current will be 90 deg out of phase *lagging*. Current in an inductive circuit *lags* the voltage that produces it; that is, points on the current curve occur later than on the voltage curve. Current i_L in sketch *C* represents current produced by *e* in a wholly inductive circuit. Note that i_L begins to increase in a positive direction 90 deg or $^1/_{240}$ sec later than *e*. Thus i_L is zero when *e* is maximum and i_L is maximum when *e* is zero. This occurs because inductive reactance is a counter-emf proportional to the rate of change of the magnetic flux. The flux is changing fastest when the current is zero. (Remember, when discussing current and voltage waves we're talking about *instantaneous* values, not the effective values which we measure on instruments and use in practical calculations.) In a circuit having resistance as well as inductive reactance, the current will lag the voltage by an angle less than 90 deg.

In capacitive circuits the current *leads* the voltage producing it. Here current is maximum 90 deg earlier than the voltage, as shown by i_c in sketch *D*. Again, if the circuit has resistance, the angle of the phase difference (displacement) is less than 90 deg."

Characteristics

Figure 14-10 compares the efficiencies of the synchronous and induction motors. For a synchronous motor designed with an 0.8 power factor, the motor delivers a leading magnetizing kva component equal to 60% of the motor kva rating. The power factor of an induction motor is always lagging. This means that it requires the electrical system to furnish it a magnetizing kva component.[10] The decision to choose a synchronous rather than an induction motor often may hinge on the power saving, including the power factor correction; see Figure 14-10 and Tables 14-1 and 14-2.

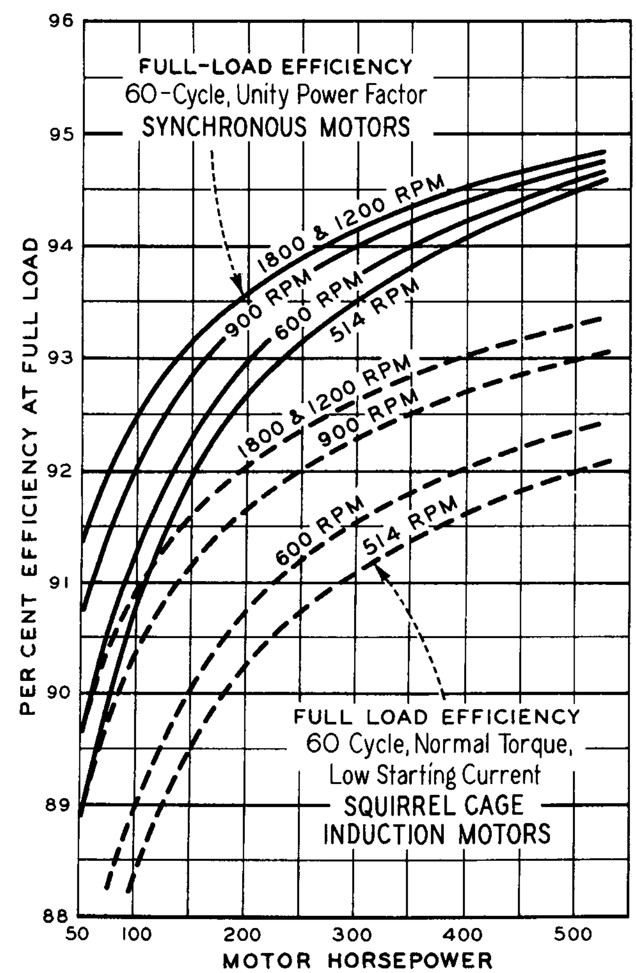

Figure 14-10. Synchronous and induction motor efficiencies: Full load efficiencies of high-speed 0.8 power factor synchronous motors in the ratings shown are 1–2% lower than unity power factor motors. (Used by permission: *E-M Synchronizer*, 200-SYN-33. ©Dresser-Rand Company.)

$$\text{Motor kilowatts input} = (\text{hp})(0.746)/\text{motor efficiency} \qquad (14\text{-}7)$$

$$\text{Motor kva input} = \text{kilowatts/power factor} \qquad (14\text{-}8)$$

Motor efficiency is usually highest at full load and falls off as the load is reduced. Voltage and frequency variations also affect the efficiency.[24, 27] Large motors have a higher efficiency than small motors. For the same horsepower, high-speed motors are more efficient than low-speed motors. Except for larger sizes, high-voltage (2,300 and larger) motors are less efficient than the low-voltage motors for the same horsepower.

The speeds of induction motors are slightly less than for the synchronous motors due to slip. For example, a synchronous speed of 1,800 rpm will actually be 1,750–1,734 rpm in an induction motor; 1,200 rpm becomes 1,150 rpm and 3,600 rpm becomes 3,450 rpm.

Table 14-3 indicates the derating factors for the effect of altitude on standard Class B motors. Motors are designed to operate within Class B temperature rise limits when operated at rated horsepower at altitudes up to 3,300 ft. For operation of this class of motor at altitudes greater than 3,300 ft at less than the rated horsepower, the derating factors shown in Table 14-3 should be used.

Table 14-3
Effect of Altitude on Operation of Large 200–2,000 hp Induction Motors (for Altitudes Greater Than 3,300 ft)

Altitude (ft)	Derating Factor
3,300– 5,000	.97
5,000– 6,600	.94
6,600– 8,300	.90
8,300– 9,900	.86
9,900–11,500	.82

NOTE: Refer all explosion-proof and dust ignition-proof applications greater than 3,300 ft altitude to the factory.

Motors are designed to operate within Class B temperature rise limits when operated at rated horsepower at altitudes up to 3,300 ft.

For operation between 3,300 and 5,500 ft altitude at rated horsepower using Class F temperature rise limits, add 3% to the basic motor price.

For operation greater than 5,500 ft altitude, refer to factor for frame size and price.

For operation of a standard Class B motor at altitudes greater than 3,300 ft at less than rated horsepower, use the derating factor.

It is important to verify performance with the manufacturer.

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Voltage: Alternating Current. Standard voltages for induction motors are 110, 208, 220, 440, 550, 2,300, 4,000, 4,600, 6,600, and 13,200. For synchronous motors, the voltages are the same except that 110 volts is not used.

Single-phase, 110v conditions are usually used only for fractional to 1 $^{1}/_{2}$ hp loads. From 1–100 hp, 220–440 (460v) and 550v, 3-phase, are most common; and from 75–250 hp the voltage is 2,300v or 440v (460v). Above 200 hp up to 1,750–2,500 hp, the voltage is 2,300v or 4,000v, and above this, 13.2kv is used. In a 4.16kv system, 440v is used for 75–250 hp, and 4,000v is used for all above this rating.[44]

A summary[8] of the chemical process industry's use of motors according to size and horsepower is shown in Table 14-4.

Direct Current. Standard voltages are 115, 230, 250, and 600.

Horsepower Ratings.[31] Standard NEMA ratings for induction motors are

General purpose: $^{1}/_{2}$, $^{3}/_{4}$, 1, 1 $^{1}/_{2}$, 2, 3, 5, 7 $^{1}/_{2}$, 10, 15, 20, 25, 30, 40, 50, 60, 75, 100, 125, 150, 200, 250, 300, 350, 400, 450, and 500.

Large motors: 250, 300, 350, 400, 450, 500, 600, 700, 800, 900, 1,000, 1,250, 1,500, 1,750, 2,000, 2,250, 2,500, 3,000, 3,500, 4,000, 4,500, 5,000 and up to 30,000.

The ratings for synchronous motors are

General purpose: Same as preceding, omitting the $^{1}/_{2}$ through 25 hp sizes.

Large high speed: Same as preceding, except start with a 200 (0.8 power factor) motor.

Low speed: 20 through 30,000 hp in sizes as listed previously.

Table 14-4
Average Percentages of Use Identified by Voltage and Horsepower Range

	Range of Horsepower Motor		
Voltage	**5–200**	**201–2,000**	**Greater Than2,000**
208–230	3–10		
460	40–55	10–14	
2.3kv		10–15	6
4.0kv	10–15	6	
13.2kv			3

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For direct-current motors, the standard horsepower ratings are essentially the same as the previous, except the largest rating is about 8,000 hp.

Energy Efficient (EE) Motor Designs

New energy efficient (EE) motors[2, 8, 26, 66, 98] for the most-used horsepower ranges have been developed by the Arthur D. Little Co.[17] The National Electrical Manufacturers Association has adopted a new motor efficiency standard; MG 1-12.53 Test B, which provides a standard efficiency test method; and a labeling standard. The efficiency results for full load for a given motor design are considered a band of efficiency with a minimum/maximum expected and nominal or average expected for that design. These efficiencies are significantly improved over most previous standard motor designs. Full-load efficiencies are given in Table 14-5.

The NEMA standard specifies that the procedure is to be the latest revision of the Institute of Electrical and Electronics Engineers (IEEE) Standard 841-1994. It is much more rigorous and uniform than the International (IEC 34-2), British (BS-269), and Japanese (JEC-37) methods, based on the U.S. evaluation of all the methods.

Many of the major motor manufacturers discovered on comparison of their own specifications to the IEEE Standard 841-1994 that their designs already exceeded the requirements of the new standard. Others made a few modifications, and their units satisfied the new standard. The standard required, among other items, (a) a no-load vibration limit of 0.08 in./sec and (b) a temperature rise of 80°C maximum with Class B insulation at rated load. The life of the motor is essentially controlled by the life of its internal insulation and is represented by Figure 14-11.[53]

Many manufacturers are registered to ISO (International Standards Organization), which is reported to be the toughest industrial quality assurance standards in the world and covers design, development, production, installation, and service.

For alternating current systems, as are primarily used in the United States in the chemical and related industry:[8]

Power Consumed: Active or actual power and is considered energy used by a resistive load.

Reactive Loads: Make demands on electrical system but yield no useful work.

Where the inductive circuits equal the capacitive circuits, the apparent power coincides with the active power. Thus, the power factor for the system is 1.0, and all power is consumed usefully.

Power factor is another factor that should be considered in the selection of a motor. Power factor can be improved by the design of the motor or by the external addition of a

Table 14-5
Full-Load Efficiencies of Energy-Efficient Motors

Open Motors

	2 Pole		4 Pole		6 Pole		8 Pole	
Hp	Nominal Efficiency	Minimum Efficiency	Nominal Efficiency	Minimum Efficiency	Nominal Efficiency	Minimum Efficiency	Nominal Efficiency	Minimum Efficiency
1.0	—	—	82.5	80.0	80.0	77.0	74.0	70.0
1.5	82.5	80.0	84.0	81.5	84.0	81.5	75.5	72.0
2.0	84.0	81.5	84.0	81.5	85.5	82.5	85.5	82.5
3.0	84.0	81.5	86.5	84.0	86.5	84.0	86.5	84.0
5.0	85.5	82.5	87.5	85.5	87.5	85.5	87.5	85.5
7.5	87.5	85.5	88.5	86.5	88.5	86.5	88.5	86.5
10.0	88.5	86.5	89.5	87.5	90.2	88.5	89.5	87.5
15.0	89.5	87.5	91.0	89.5	90.2	88.5	89.5	87.5
20.0	90.2	88.5	91.0	89.5	91.0	89.5	90.2	88.5
25.0	91.0	89.5	91.7	90.2	91.7	90.2	90.2	88.5
30.0	91.0	89.5	92.4	91.0	92.4	91.0	91.0	89.5
40.0	91.7	90.2	93.0	91.7	93.0	91.7	91.0	89.5
50.0	92.4	91.0	93.0	91.7	93.0	91.7	91.7	90.2
60.0	93.0	91.7	93.6	92.4	93.6	92.4	92.4	91.0
75.0	93.0	91.7	94.1	93.0	93.6	92.4	93.6	92.4
100.0	93.0	91.7	94.1	93.0	94.1	93.0	93.6	92.4
125.0	93.6	92.4	94.5	93.6	94.1	93.0	93.6	92.4
150.0	93.6	92.4	95.0	94.1	94.5	93.6	93.6	92.4
200.0	94.5	93.6	95.0	94.1	94.5	93.6	93.6	92.4
250.0	94.5	93.6	95.4	94.5	95.4	94.5	94.5	93.6
300.0	95.0	94.1	95.4	94.5	95.4	94.5	—	—
350.0	95.0	94.1	95.4	94.5	95.4	94.5	—	—
400.0	95.4	94.5	95.4	94.5	—	—	—	—
450.0	95.8	95.0	95.8	95.0	—	—	—	—
500.0	95.8	95.0	95.8	95.0	—	—	—	—

Enclosed Motors

	2 Pole		4 Pole		6 Pole		8 Pole	
Hp	Nominal Efficiency	Minimum Efficiency	Nominal Efficiency	Minimum Efficiency	Nominal Efficiency	Minimum Efficiency	Nominal Efficiency	Minimum Efficiency
1.0	75.5	72.0	82.5	80.0	80.0	77.0	74.0	70.0
1.5	82.5	80.0	84.0	81.5	85.5	82.5	77.0	74.0
2.0	84.0	81.5	84.0	81.5	86.5	84.0	82.5	80.0
3.0	85.5	82.5	87.5	85.5	87.5	85.5	84.0	81.5
5.0	87.5	85.5	87.5	85.5	87.5	85.5	85.5	82.5
7.5	88.5	86.5	89.5	87.5	89.5	87.5	85.5	82.5
10.0	89.5	87.5	89.5	87.5	89.5	87.5	88.5	86.5
15.0	90.2	88.5	91.0	89.5	90.2	88.5	88.5	86.5
20.0	90.2	88.5	91.0	89.5	90.2	88.5	89.5	87.5
25.0	91.0	89.5	92.4	91.0	91.7	90.2	89.5	87.5
30.0	91.0	89.5	92.4	91.0	91.7	90.2	91.0	89.5
40.0	91.7	90.2	93.0	91.7	93.0	91.7	91.0	89.5
50.0	92.4	91.0	93.0	91.7	93.0	91.7	91.7	90.2
60.0	93.0	91.7	93.6	92.4	93.6	92.4	91.7	90.2
75.0	93.0	91.7	94.1	93.0	93.6	92.4	93.0	91.7
100.0	93.6	92.4	94.5	93.6	94.1	93.0	93.0	91.7
125.0	94.5	93.6	94.5	93.6	94.1	93.0	93.6	92.4
150.0	94.5	93.6	95.0	94.1	95.0	94.1	93.6	92.4
200.0	95.0	94.1	95.0	94.1	95.0	94.1	94.1	93.0
250.0	95.4	94.5	95.0	94.1	95.0	94.1	94.5	93.6
300.0	95.4	94.5	95.4	94.5	95.0	94.1	—	—
350.0	95.4	94.5	95.4	94.5	95.0	94.1	—	—
400.0	95.4	94.5	95.4	94.5	—	—	—	—
450.0	95.4	94.5	95.4	94.5	—	—	—	—
500.0	95.4	94.5	95.8	95.0	—	—	—	—

Used by permission: *Motors and Generators*, NEMA Standard MG 1-1993, Rev. 1, Section II, Part 12, p. 25–26, 1995. National Electrical Manufacturers Association. (Note: All references to NEMA Standards are reprinted by Permission of the National Electrical Manufacturers Association from NEMA Standards Publication No. MG-1-1993, *Motors and Generators* ©(1995) by the National Electrical Manufacturers Association. Rev. No. 2, Apr. 1, (1995),) capacitor. As a part of the total economic solution of a motor driver, the cost benefits of the total evaluation of the various features relating to short-term and long-term motor costs should be examined. The reference by D. C. Montgomery[26] is one good presentation for this purpose.

$$\text{E efficiency (fraction)} = \frac{746\ (\text{hp output})}{\text{watts input}} = \frac{\text{input} - \text{losses}}{\text{input}} \tag{14-9}$$

PF_1 power factor (fraction) (for 3-phase system)

$$= \frac{\text{watts in}}{(\text{volts})(\text{amps})(1.73)} \tag{14-10}$$

NEMA Design Classifications

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Classification According to Size

1.02 Machine. As used in this standard a machine is an electrical apparatus which depends on electromagnetic induction for its operation and which has one or more component members capable of rotary movement. In particular, the types of machines covered are those generally referred to as motors and generators . . . .

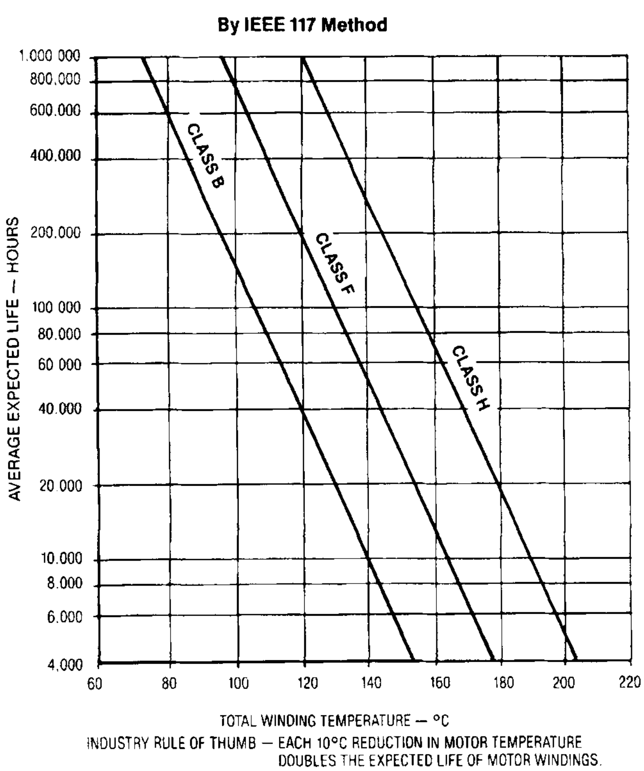

Figure 14-11. Temperature versus life curves for insulation systems. (Used by permission: Bul. E-7, May 1993. ©The Lincoln Electric Co., Motor Division.

1.03 Small (Fractional) Machine. A small machine is either: (1) a machine built in a two-digit frame number series in accordance with 11.01.1 (or equivalent for machines without feet) or (2) a machine built in a frame smaller than that frame of a medium machine (see 1.04), which has a continuous rating at 1700–1800 rpm of 1 horsepower for motors or 0.75 kilowatt for generators; or (3) a motor rated less than $^1/_3$ horsepower and less than 800 rpm.

1.04 Medium (Integral) Machine

1.04.1 Alternating-Current Medium Machine. An alternating-current medium machine is a machine: (1) built in a three- or four-digit frame number series in accordance with 11.01.2 (or equivalent for machines without feet) and (2) having a continuous rating up to and including the information in Table 14-6.

1.04.2 Direct-Current Medium Machine. A direct-current medium machine is a machine: (1) built in a three- or four-digit frame number series in accordance with 11.01.2 (or equivalent for machines without feet) and (2) having a continuous rating up to and including 1.25 horsepower per rpm for motors 1.0 kilowatt per rpm for generators.

1.05 Large Machine

1.05.1 Alternating-Current Large Machine. An alternating-current large machine is: (1) a machine having a continuous power rating greater than that given in 1.04.1 for synchronous speed ratings above 450 rpm; or (2) a machine

Table 14-6
Classification According to Size
Alternating-Current Medium Machine

Synchronous Speed, rpm	Motors hp	Generators, Kilowatt at 0.8 Power Factor
1,201–3,600	500	400
901–1,200	350	300
721–900	250	200
601–720	200	150
515–600	150	125
451–514	125	100

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having a continuous power rating greater than that given in 1.03 for synchronous speed ratings equal to or below 450 rpm.

1.05.2 Direct-Current Large Machine. A direct-current large machine is a machine having a continuous rating greater than 1.25 horsepower per rpm for motors or 1.0 kilowatt per rpm for generators.

A. Alternating Current

1. Polyphase squirrel-cage induction motor.
   This is the most generally applicable motor to the majority of process plant applications. NEMA[31] rule MG1-1.25 states, "A general-purpose motor is any open motor having a continuous 40°C rating and designed, listed and offered in standard ratings with the standard operating characteristics and mechanical construction for use under usual service conditions without restriction to a particular application or type of application." Usual practice is to limit general-purpose ratings to 200 hp and smaller with speeds of 450 rpm and greater.[34] Designs B and C are the most commonly considered as general-purpose applications. Table 14-6 indicates the horsepower ranges for medium-sized motors, from NEMA earlier reference.
2. Synchronous motor.
   This is a very good motor for direct connection to certain loads, particularly where constant speed is required. NEMA[31] defines it as a "synchronous machine which transforms electrical power from an alternating-current system into mechanical power." It usually has direct-current field excitation by a separately driven direct-current generator or one directly connected to the motor. This motor remains synchronous with the supply frequency and is not affected by the load. Proper application requires consideration of the following:[24, 27]
   a. Power factor.
   b. Locked-rotor (static) torque.
   c. Pull-up (accelerating) torque.
   d. Pull-in torque.
   e. Pull-out torque.
   f. Effect of load inertia on pull-in torque and thermal capacity of the amortisseur winding.
   g. Effect of voltage variation on torques.
   Table 14-7 identifies many of the applications for synchronous motors.
3. Wound rotor induction motor.
   This is a special induction-type motor that can be operated essentially as an induction motor after starting or as a high slip machine, if needed. When the slip rings are short circuited after starting, the speed of a wound motor is essentially constant regardless of load, as a squirrel-cage motor.[23] When the rings are not shorted but connected to a resistor, the motor will accommodate high slip conditions. The speed then varies with the load.
   Applications requiring unusually high starting torque with very low starting current, adjustable speed, or severe reversing are best suited for this motor. When power sources require a close limit on starting currents, the wound rotor is often the choice. They may be used to start equipment such as conveyors, reciprocating pumps, cranes, hoists, bending rolls, etc., under load. They are also used for severe reversing service, plugging service, and frequent starting and stopping.
4. Variable speed motor.
   This is a special design for specific and usually special application.

B. Direct Current. These motors are general purpose and used (1) for continuous operation under fairly constant load, (2) when fine speed adjustment is needed, and (3) when d–c current is readily available or the characteristics are required for proper operation of equipment.

Hazard Classifications: Fire and Explosion*

It is important that the fire and explosion hazards of an area be carefully examined, because the expense of consistent installation of all the motors, controls, switches, instruments, and wiring can be considerable. Tables 14-8A and 14-8B summarize the National Fire Code[28, 29] for hazardous locations. It is equally important to be consistent and not install explosion-proof motors with nonexplosion proof wiring, because a failure in the conduit can still cause considerable damage.

Tables 14-8B-1–4 are a selected group of National Electrical Code Articles that recognize certain subjects with which the process engineer should be acquainted. These subjects

(Text continues on page 634)

*The standard reference for manufacturers as well as the application of electrical equipment in process plants is The National Electrical Code.[28, 29, 42, 91] The Introduction Article 90 and the Purpose and Scope of the National Electrical Code® (NFPA-70; NEC) and all other selections (Articles) from the Code® are quoted and reprinted by permission from the National Electrical Code®, 1996 Ed. ©1995. Throughout this text, the information identified as "reprinted from the National Electrical Code®" is not the complete and official position of the NFPA on the referenced subject, which is represented only by the Standard in its entirety. National Electrical Code® and NEC® are registered trademarks of the National Fire Protection Association, Inc., Quincy, MA 02269. The material is used here by permission with the understanding that it represents informational extracts from the code, and the code is valid only when used in its entirety.

Table 14-7
Typical Torque Requirements for Synchronous Motor Applicationsϕ

In individual cases, lower values may be satisfactory or higher values may be necessary, depending upon the characteristics of the particular machine and the effect of the locked-rotor kva on the line voltage.
For applications having higher inertia, the Wk^2 of the load may require a motor design that cannot be determined from the torque requirements alone. For such applications, the motor manufacturer should always be provided with the actual value of the Wk^2 of the load.

		Torques in Percent of Motor Full-Load Torque			
Item No.	Application	Locked-Rotor	Pull-In	Pull-Out	Ratio of Wk^2 of Load to Normal Wk^2 of Load
1	Attrition mills (for grain processing)—starting unloaded	100	60	175	3–15
2	Ball mills (for rock and coal)	140	110	175	2–4
3	Ball mills (for ore)	150	110	175	1.5–4
4	Banbury mixers	125	125	250	0.2–1
5	Band mills	40	40	250	50–110
6	Beaters, standard	125	100	150	3–15
7	Beaters, breaker	125	100	200	3–15
8	Blowers, centrifugal—starting with				
	a. Inlet or discharge valve closed	30	40–60*	150	3–30
	b. Inlet or discharge valve open	30	100	150	3–30
9	Blowers, positive displacement, rotary—by-passed for starting	30	25	150	3–8
10	Bowl mills (coal pulverizers)—starting unloaded				
	a. Common motor for mill and exhaust fan	90	80	150	5–15
	b. Individual motor for mill	140	50	150	4–10
11	Chippers—starting empty	60	50	250	10–100
12	Compressors, centrifugal—starting with				
	a. Inlet or discharge valve closed	30	40–60*	150	3–30
	b. Inlet or discharge valve open	30	100	150	3–30
13	Compressors, Fuller Company				
	a. Starting unloaded (by-pass open)	60	60	150	0.5–2
	b. Starting loaded (by-pass closed)	60	100	150	0.5–2
14	Compressors, Nash-Hytor—starting unloaded	40	60	150	2–4
15	Compressors, reciprocating—starting unloaded				
	a. Air and gas	30	25	150	0.2–15
	b. Ammonia (discharge pressure 100–250 psi)	30	25	150	0.2–15
	c. Freon	30	40	150	0.2–15
16	Crushers, Bradley-Hercules—starting unloaded	100	100	250	2–4
17	Crushers, cone—starting unloaded	100	100	250	1–2
18	Crushers, gyratory—starting unloaded	100	100	250	1–2
19	Crushers, jaw—starting unloaded	150	100	250	10–50
20	Crushers, roll—starting unloaded	150	100	250	2–3
21	Defibrators (see Beaters, standard)				
22	Disintegrators, pulp (see Beaters, standard)				
23	Edgers	40	40	250	5–10
24	Fans, centrifugal (except sintering fans)—starting with				
	a. Inlet or discharge valve closed	30	40–60*	150	5–60
	b. Inlet or discharge valve open	30	100	150	5–60
25	Fans, centrifugal sintering—starting with inlet gates closed	40	100	150	5–60
26	Fans, propeller type—starting with discharge valve open	30	100	150	5–60
27	Generators, alternating current	20	10	150	2–15
28	Generators, direct current (except electroplating)				
	a. 150 kw and smaller	20	10	150	2–3
	b. Greater than 150 kw	20	10	200	2–3
29	Generators, electroplating	20	10	150	2–3
30	Grinders, pulp, single, long magazine-type—starting unloaded	50	40	150	2–5
31	Grinders, pulp, all except single, long magazine-type—starting unloaded	40	30	150	1–5
32	Hammer mills—starting unloaded	100	80	250	30–60
33	Hydrapulpers, continuous type	125	125	150	5–15
34	Jordans (see Refiners, conical)				
35	Line shafts, flour mill	175	100	150	5–15
36	Line shafts, rubber mill	125	110	225	0.5–1
37	Plasticators	125	125	250	0.5–1

Item No.	Application	Torques in Percent of Motor Full-Load Torque: Locked-Rotor	Pull-In	Pull-Out	Ratio of Wk^2 of Load to Normal Wk^2 of Load
38	Pulverizers, B&W — starting unload				
	a. Common motor for mill and exhaust fan	105	100	175	20–60
	b. Individual motor for mill	175	100	175	4–10
39	Pumps, axial flow, adjustable blade—starting with				
	a. Casing dry	5–40**	15	150	0.2–2
	b. Casing filled, blades feathered	5-40**	40	150	0.2–2
40	Pumps, axial flow, fixed blade—starting with				
	a. Casing dry	5–40**	15	150	0.2–2
	b. Casing filled, discharge closed	5–40**	175–250**	150	0.2–2
	c. Casing filled, discharge open	5–40**	100	150	0.2–2
41	Pumps, centrifugal, Francis impeller—starting with				
	a. Casing dry	5–40**	15	150	0.2–2
	b. Casing filled, discharge closed	5–40**	60–80*	150	0.2–2
	c. Casing filled, discharge open	5–40**	100	150	0.2–2
42	Pumps, centrifugal, radia impeller—starting with				
	a. Casing dry	5–40**	15	150	0.2–2
	b. Casing filled, discharge closed	5–40**	40–60*	150	0.2–2
	c. Casing filled, discharge open	5–40**	100	150	0.2–2
43	Pumps, mixed flow—starting with				
	a. Casing dry	5–40**	15	150	0.2–2
	b. Casing filled, discharge closed	5–40**	82–125*	150	0.2–2
	c. Casing filled, discharge open	5–40**	100	150	0.2–2
44	Pumps, reciprocating—starting with				
	a. Cylinders dry	40	30	150	0.2–15
	b. By-pass open	40	40	150	0.2–15
	c. No by-pass (three cylinder)	150	100	150	0.2–15
45	Refiners, conical (Jordan, Hydrafiners, Claflins, Mordens)—starting with plug out	50	50–100†	150	2–20
46	Refiners, disc type—starting unloaded	50	50	150	1–20
47	Rod mills (for ore grinding)	160	120	175	1.5–4
48	Rolling mills				
	a. Structural and rail roughing mills	40	30	300–400††	0.5–1
	b. Structural and rail finishing mills	40	30	250	0.5–1
	c. Plate mills	40	30	300-400††	0.5–1
	d. Merchant mill trains	60	40	250	0.5–1
	e. Billet, skelp, and sheet bar mills, continuous, with lay-shaft drive	60	40	250	0.5–1
	f. Rod mills, continuous with lay-shaft drive	100	60	250	0.5–1
	g. Hot strip mills, continuous, individual drive roughing stands	50	40	250	0.5–1
	h. Tube piercing and expanding mills	60	40	300–400††	0.5–1
	i. Tube rolling (plug) mills	60	40	250	0.5–1
	j. Tube reeling mills	60	40	250	0.5–1
	k. Brass and copper roughing mills	50	40	250	0.5–1
	l. Brass and copper finishing mills	150	125	250	0.5–1
49	Rubber mills, individual drive	125	125	250	0.5–1
50	Saws, band (see Band mills)				
51	Saws, edger (see Edgers)				
52	Saws, trimmer	40	40	250	5–10
53	Tube mills (see Ball mills)				
54	Vacuum pumps, Hytor				
	a. With unloader	40	30	150	2–4
	b. Without unloader	60	100	150	2–4
55	Vacuum pumps, reciprocating—starting unloaded	40	60	150	0.2–15
56	Wood hogs	60	50	250	30–100

*The pull-in torque with the designs and operating conditions. The machinery manufacturer should be consulted.

**For horizontal shaft pumps and vertical shaft pumps having no thrust bearing (entire thrust load carried by the motor), the locked-rotor torque required is usually between 5 and 20%, and for vertical shaft machines having their own thrust bearing a locked-rotor torque as high as 40% is sometimes required.

†The pull-in torque required varies with the design of the refiner. The machinery manufacturer should be consulted. Furthermore, even though 50% pull-in torque is adequate with the plug out, it is sometimes considered desirable to specify 100% to cover the possibility that a start will be attempted without complete retraction of the plug.

††The pull-out torque varies depending upon the rolling schedule.

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Table 14-8A
Typical Hazardous Classifications Summary According to the National Electric Code (NEC)

Classification	Hazardous Environment
Class I	Potentially explosive flammable gases or vapors in the air
Class II	Combustible dust in the air
Class III	Ignitable fibers or flyings (dust) in the air
Group A	Acetylene
Group B	Hydrogen, gases, or vapors of manufactured origin
Group C	Ethyl-ether vapors, ethylene, or cyclopropane
Group D	Gasoline, hexane, naphtha, benzene, butane, propane, alcohols, acetone, lacquer, solvent vapors, or natural gas (methane)
Group E	Conductive dust and metal dust: aluminum, magnesium, and their commercial alloys
Group F	Carbon black, coal, or coke dust
Group G	Flour, starch, grain dusts
Division 1	Locations where hazardous material exists (always or periodically) during operating conditions
Division 2	Locations where hazardous material exists only in the case of a fault situation (leaky valve, burst pipe, faulty equipment)

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include those represented in this chapter but do not limit the scope of code information and requirements. The selected Articles here are: 90-1 through 90-7, 500-1 through 500-7, 505-2 through 505-10, 510-1, 510-2; 515-1 through 515-6, Table 515-2, and Figure 515-2. The reader is referred to the complete National Electrical Code, ©1996 or later. The material presented here is intended as a guide and cannot cover all the conditions defined in the full code.

It is important that the designer/project engineer become familiar with all process-related features of the full code and to recognize and identify the requirements for hazardous plant-related situations:

A. Classes and Groups
B. Classes and Divisions

Table 14-8A briefly identifies the key hazardous classifications established by the National Electrical Code, Articles 500–505.[28, 29, 94] A more detailed copy of selected portions of the NEC is included in Tables 14-8B and 14-8B-1–4. Depending on the scope of any particular project, the engineer should examine the topics covered in the entire NEC for applicable requirements.

Plant areas are classified with respect to the possibilities of fire and explosion hazards existing or developing in the area while electrical equipment is in operation. The locations are

(Text continues on page 647)

Table 14-8B
National Electrical Code®: (NEC®), NFPA®-70, Article 90, Introduction

90-1. Purpose.

(a) Practical Safeguarding. The purpose of this *Code* is the practical safeguarding of persons and property from hazards arising from the use of electricity.

(b) Adequacy. This *Code* contains provisions considered necessary for safety. Compliance therewith and proper maintenance will result in an installation essentially free from hazard but not necessarily efficient, convenient, or adequate for good service or future expansion of electrical use.

(FPN): Hazards often occur because of overloading of wiring systems by methods or usage not in conformity with this *Code*. This occurs because initial wiring did not provide for increases in the use of electricity. An initial adequate installation and reasonable provisions for system changes will provide for future increases in the use of electricity.

(c) Intention. This *Code* is not intended as a design specification nor an instruction manual for untrained persons.

90-2. Scope.

(a) Covered. This *Code* covers:

(1) Installations of electric conductors and equipment within or on public and private buildings or other structures, including mobile homes, recreational vehicles, and floating buildings; and other premises such as yards, carnival, parking, and other lots,

and industrial substations.

(FPN): For additional information concerning such installations in an industrial or multibuilding complex, see the *National Electrical Safety Code*, ANSI C2-1993.

(2) Installations of conductors and equipment that connect to the supply of electricity.

(3) Installations of other outside conductors and equipment on the premises.

(4) Installations of optical fiber cable.

(5) Installations in buildings used by the electric utility, such as office buildings, warehouses, garages, machine shops, and recreational buildings that are not an integral part of a generating plant, substation, or control center.

(b) Not Covered. This *Code* does not cover:

(1) Installations in ships, watercraft other than floating buildings, railway rolling stock, aircraft, or automotive vehicles other than mobile homes and recreational vehicles.

(2) Installations underground in mines and self-propelled mobile surface mining machinery and its attendant electrical trailing cable.

(3) Installations of railways for generation, transformation, transmission, or distribution of power used exclusively for operation of rolling stock or installation used exclusively for signaling and communications purposes.

(4) Installations of communications equipment under the exclusive control of communications utilities located outdoors or in building spaces used exclusively for such installations.

(5) Installations, including associated lighting, under the exclusive control of electric utilities for the purpose of communications, metering, generation, control, transformation, transmission, or distribution of electric energy. Such installations shall be located in buildings used exclusively by utilities for such purposes; outdoors on property owned or leased by the utility; on or along public highways, streets, roads, etc.; or outdoors on private property by established rights such as easements.

(c) Special Permission. The authority having jurisdiction for enforcing this Code may grant exception for the installation of conductors and equipment that are not under the exclusive control of the electric utilities and are used to connect the electric utility supply system to the service-entrance conductors of the premises served, provided such installations are outside a building or terminate immediately inside a building wall.

90-3. Code Arrangement. This *Code* is divided into the Introduction and nine chapters. Chapters 1, 2, 3, and 4 apply generally; Chapters 5, 6, and 7 apply to special occupancies, special equipment, or other special conditions. These latter chapters supplement or modify the general rules. Chapters 1 through 4 apply except as amended by Chapters 5, 6, and 7 for the particular conditions.

Chapter 8 covers communications systems and is independent of the other chapters except where they are specifically referenced therein.

Chapter 9 consists of tables and examples.

Material identified by the superscript letter "x" includes text extracted from other NFPA documents as identified in Appendix A.

90-4. Enforcement. This *Code* is intended to be suitable for mandatory application by governmental bodies exercising legal jurisdiction over electrical installations and for use by insurance inspectors. The authority having jurisdiction for enforcement of the *Code* will have the responsibility for making interpretations of the rules, for deciding upon the approval of equipment and materials, and for granting the special permission contemplated in a number of the rules.

The authority having jurisdiction may waive specific requirements in this *Code* or permit alternate methods where it is assured that equivalent objectives can be achieved by establishing and maintaining effective safety.

This *Code* may require new products, constructions, or materials that may not yet be available at the time the *Code* is adopted. In such event, the authority having jurisdiction may permit the use of the products, constructions, or materials that comply with the most recent previous edition of this *Code* adopted by the jurisdiction.

90-5. Mandatory Rules and Explanatory Material. Mandatory rules of this *Code* are characterized by the use of the word "shall." Explanatory material is in the form of Fine Print Notes (FPN).

90-6. Formal interpretations. To promote uniformity of interpretation and application of the provision of this *Code*, Formal Interpretation procedures have been established.

(FPN): These procedures may be found int he "NFPA Regulations Governing Committee Projects."

(continued)

Table 14-8
(Continued)

90-7. Examination of Equipment for Safety. For specific items of equipment and materials referred to in this Code, examinations for safety made under standard conditions will provide a basis for approval where the record is made generally available through promulgation by organizations properly equipped and qualified for experimental testing, inspections of the run of goods at factories, and service-value determination through field inspections. This avoids the necessity for repetition of examinations by different examiners, frequently with inadequate facilities for such work, and the confusion that would result from conflicting reports as to the suitability of devices and materials examined for a given purpose.

It is the intent of this *Code* that factory-installed internal wiring or the construction of equipment need not be inspected at the time of installation of the equipment, except to detect alterations or damage, if the equipment has been listed by a qualified electrical testing laboratory that is recognized as having the facilities described above and that requires suitability for installation in accordance with this *Code*.

(FPN No. 1): See Examination, Identification, Installation, and Use of Equipment, Section 110-3.

(FPN No. 2): See definition of "Listed," Article 100.

90-8. Wiring Planning.

(a) Future Expansion and Convenience. Plans and specifications that provide ample space in raceways, spare raceways, and additional spaces will allow for future increases in the use of electricity. Distribution centers located in readily accessible locations will provide convenience and safety of operation.

(b) Number of Circuits in Enclosures. It is elsewhere provided in this *Code* that the number of wires and circuits confined in a single enclosure be varyingly restricted. Limiting the number of circuits in a single enclosure will minimize the effects from a short-circuit or ground fault in one circuit.

90-9. Metric Units of Measurement. For the purpose of this *Code*, metric units of measurement are in accordance with the modernized metric system known as the International System of Units (SI).

Values of measurement in the *Code* text will be followed by an approximate equivalent value in SI units. Tables will have a footnote for SI conversion units used in the table.

Conduit size, wire, size, horsepower designation for motors, and trade sizes that do not reflect actual measurements, e.g., box sizes, will not be assigned dual designation SI units.

(FPN): For metric conversion practices, see *Standard for Metric Practice*, ANSI/ASTM E380-1993.

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Table 14-8B-1
NFPA-70 Article 500 — Hazardous (Classified) Locations

500-1. Scope—Articles 500 Through 505. Articles 500 through 505 cover the requirements for electrical equipment and wiring for all voltages in locations where fire or explosion hazards may exist due to flammable gas or vapors, flammable liquids, combustible dust, or ignitable fibers or flyings.

500-2. Location and General Requirements. Locations shall be classified depending on the properties of the flammable vapors, liquids or gases or combustible dusts or fibers that may be present and the likelihood that a flammable or combustible concentration or quantity is present. Where pyrophoric materials are the only materials used or handled, these locations shall not be classified.

Each room, section, or area shall be considered individually in determining its classification.

(FPN): Through the exercise of ingenuity in the layout of electrical installations for hazardous (classified) locations, it is frequently possible to locate much of the equipment in less hazardous or in nonhazardous locations, and, thus, to reduce the amount of special equipment required.

All other applicable rules contained in this *Code* shall apply to electrical equipment and wiring installed in hazardous (classified) locations.

Exception: As modified in Articles 500 through 505.

All threaded conduit referred to herein shall be threaded with an NPT standard conduit cutting die that provides $^3/_4$-in. taper per ft. Such conduit shall be made wrenchtight to (1) prevent sparking when fault current flows through the conduit system, and (2) ensure the explosionproof or dust-ignitionproof integrity of the conduit system where applicable.

(FPN): Some equipment provided with metric threads will need suitable adapters to permit connection to rigid conduit with NPT threads.

Optical fiber cables and fiber optic devices approved as an intrinsically safe system suitable for the hazardous (classified) location involved shall be installed in accordance with Sections 504-20 and 770-52.

Exception: Optical fiber cables or fiber optic devices that are conductive shall be installed in accordance with Articles 500 through 503.

(a) Protection Techniques. The following shall be acceptable protection techniques for electrical and electronic equipment in hazardous (classified) locations.

(1) Explosionproof Apparatus. The protection technique shall be permitted for equipment in those Class I, Division 1 and 2 locations for which it is approved.

(FPN): Explosionproof apparatus is defined in Article 100. For further information, see *Explosionproof and Dust-Ignitionproof Electrical Equipment for Use in Hazardous (Classified) Locations,* ANSI/UL 1203-1988.

(2) Dust-ignitionproof. This protection technique shall be permitted for equipment in those Class II, Division 1 and 2 locations for which it is approved.

(FPN): Dust-ignitionproof equipment is defined in Section 502-1. For further information, see *Explosionproof and Dust-Ignitionproof Electrical Equipment for Use in Hazardous (Classified) Locations,* ANSI/UL 1203-1988.

(3) Purged and Pressurized. This protection technique shall be permitted for equipment in any hazardous (classified) location for which it is approved.

(FPN No. 1): In some cases, hazards may be reduced or hazardous (classified) locations limited or eliminated by adequate positive-pressure ventilation from a source of clean air in conjunction with effective safeguards against ventilation failure.

(FPN No. 2): For further information, see *Standard for Purged and Pressurized Enclosures for Electrical Equipment,* NFPA 496-1993 (ANSI).

(4) Intrinsically Safe Systems. Intrinsically safe apparatus and wiring shall be permitted in any hazardous (classified) location for which it is approved, and the same provisions of Articles 501 through 503, 505, and 510 through 516 shall not be considered applicable to such installations, except as required by Article 504.

Installation of intrinsically safe apparatus and wiring shall be in accordance with the requirements of Article 504.

(FPN): For further information, *see Intrinsically Safe Apparatus and Associated Apparatus for Use in Class I, II, and III, Division 1, Hazardous Locations,* ANSI/UL 913–1988.

(5) Nonincendive Circuits. This protection technique shall be permitted for equipment in those Class I, Division 2, Class II, Division 2, and Class III locations for which it is approved.

(FPN): Nonincendive circuit is defined in Article 100. For further information, see *Electrical Equipment for Use in Class I, Division 2 Hazardous (Classified) Locations,* ANSI/ISA-S12.12-1984.

(6) Nonincendive Component. A component having contacts for making or breaking an incendive circuit and the contacting mechanism shall be constructed so that the component is incapable of igniting the specified flammable gas- or vapor-air mixture. The housing of a nonincendive component is not intended to (1) exclude the flammable atmosphere or (2) contain an explosion.

This protection technique shall be permitted for current-interrupting contacts in those Class I, Division 2, Class II, Division 2, and Class III locations for which the equipment is approved.

(FPN): For further information, see *Electrical Equipment for Use In Class I and II, Division 2, and Class III Hazardous (Classified) Locations,* UL 1604–1988.

(7) Oil Immersion. This protection technique shall be permitted for current-interrupting contacts in Class I, Division 2 locations as described in Section 501-6(b)(1)(2).

(FPN): See Sections 501-3(b)(1), Exception a.; 501-5(a)(1), Exception b., 501-6(b)(1); 501-14(b)(1), Exception a.; 502-14(a)(2), Exception; and 502-14(a)(3), Exception. For further information, see *Industrial Control Equipment for Use In Hazardous (Classified) Locations,* ANSI/UL 698-1991.

(8) Hermetically Sealed. A hermetically sealed device shall be sealed against the entrance of an external atmosphere and the seal shall be made by fusion, e.g., soldering, brazing, welding, or the fusion of glass to metal.

This protection technique shall be permitted for current-interrupting contacts in Class I, Division 2 locations.

(FPN): See Sections 501-3(b)(1), Exception b.; 501-5(a)(1), Exception a.; 501-6(b)(1); and 501-14(b)(1), Exception b. For further information, see *Electrical Equipment for Use in Class I, Division 2 Hazardous (Classified) Locations,* ANSI/ISA-S12.12-1984.

(b) Reference Standards.

(FPN No. 1): It is important that the authority having jurisdiction be familiar with recorded industrial experience as well as with standards of the National Fire Protection Association, the American Petroleum Institute, and the Instrument Society of America that may be of use in the classification of various locations, the determination of adequate ventilation, and the protection against static electricity and lightning hazards.

(continues)

Table 14-8B-1
(Continued)

(FPN No. 2): For further information on the classification of locations, see *Flammable and Combustible Liquids Code,* NFPA 30-1993; *Standard for Drycleaning Plants,* NFPA 32–1990; *Standard for Spray Application Using Flammable and Combustible Materials,* NFPA 33-1995; *Standard for Dipping and Coatings Processes Using Flammable or Combustible Liquids,* NFPA 35-1995; *Standard for the Manufacture of Organic Coatings,* NFPA 35-1995; *Standard for Solvent Extraction Plants,* NFPA 36-1993; *Standard on Fire Protection for Laboratories Using Chemicals,* NFPA 45-1991; *Standard for Gaseous Hydrogen Systems at Consumer Sites,* NFPA 50A-1994; *Standard for Liquefied Hydrogen Systems at Consumer Sites,* NFPA 50B-1994; *Standard for the Storage and Handling of Liquefied Petroleum Gases,* NFPA 58-1995; *Standard for the Storage and Handling of Liquefied Petroleum Gases at Utility Gas Plants,* NFPA 59-1995; *Recommended Practice for Classification of Class I Hazardous (Classified) Locations for Electrical Installations in Chemical Process Areas,* NFPA 497A-1992; *Recommended Practice for the Classification of Class II Hazardous (Classified) Locations for Electrical Installations in Chemical Process Areas,* 497B-1991; *Manual for Classification of Gases, Vapors, and Dusts for Electrical Equipment in Hazardous (Classified) Locations,* NFPA 497M-1991; *Recommended Practice for Fire Protection in Wastewater Treatment and Collection Facilities,* NFPA 820-1995; *Classification of Locations for Electrical Installations At Petroleum Facilities,* ANSI/API 500-1992; *Area Classification In Hazardous (Classified) Dust Locations,* ANSI/ISA-S12.10-1988.

(FPN No. 3): For further information on protection against static electricity and lightning hazards in hazardous (classified) locations, see *Recommended Practice on Static Electricity,* NFPA 77–1993; *Standard for the Installation of Lightning Protection Systems,* NFPA 780-1995; and *Protection Against Ignitions Arising Out of Static Lightning and Stray Currents,* API RP 2003-1991.

(FPN No. 4): For further information on ventilation, see *Flammable and Combustible Liquids Code,* NFPA 30-1993; and *Recommended Practice for Classification of Locations for Electrical Installations at Petroleum Facilities,* API RP 500-1991, Section 4.6.

(FPN No. 5): For further information on electrical system for hazardous (classified) locations on offshore oil and gas platforms, see *Design and Installation of Electrical Systems for Offshore Production Platforms,* ANSI/API RP 14F-1991.

500-3. Special Precaution. Articles 500 through 504 require equipment construction and installation that will ensure safe performance under conditions of proper use and maintenance.

If Article 505 is used, area classification, wiring, and equipment selection shall be under the supervision of a qualified Registered Professional Engineer.

(FPN No. 1): It is important that inspection authorities and users exercise more than ordinary care with regard to installation and maintenance.

(FPN No. 2): Low ambient conditions require special consideration. Explosionproof or dust-ignitionproof equipment may not be suitable for use at temperatures lower than –25°C (–13°F) unless they are approved for low-temperature service. However, at low ambient temperatures, flammable concentrations of vapors may not exist in a location classified Class I, Division 1 at normal ambient temperatures.

For purposes of testing, approval, and area classification, various air mixtures (not oxygen-enriched) shall be grouped in accordance with Sections 500-3(a) and 500-3(b).

Exception No. 1: Equipment approved for specific gas, vapor, or dust.

Exception No. 2: Equipment intended specifically for Class I, Zone 0, Zone 1, or Zone 2 locations shall be grouped in accordance with Section 5 05-5.

(FPN): This grouping is based on the characteristics of the materials. Facilities have been available for testing and approving equipment for use in the various atmospheric groups.

(a) Class I Group Classifications. Class I groups shall be as follows:

(1) Group A. Atmospheres containing acetylene.

(2) Group B. Atmospheres containing hydrogen, fuel and combustible process gases containing more than 30 percent hydrogen by volume or gases or vapors of equivalent hazard such as butadiene, ethylene oxide, propylene oxide, and acrolein.

Exception No. 1: Group D equipment shall be permitted to be used for atmospheres containing butadiene if such equipment is isolated in accordance with Section 501-5(a) by sealing all conduit $^1/_2$-in. size or larger.

Exception No. 2: Group C equipment shall be permitted to be used for atmospheres containing ethylene oxide, propylene oxide, and acrolein if such equipment is isolated in accordance with Section 501-5(a) by sealing all conduit $^1/_2$ -in. size or larger.

(3) Group C. Atmospheres such as ethyl ether, ethylene, or gases or vapors of equivalent hazard.

(4) Group D. Atmospheres such as acetone, ammonia, benzene, butane, cyclopropane, ethanol, gasoline, hexane, methanol, methane, natural gas, naphtha, propane, or gases or vapors of equivalent hazard.

Exception: For atmospheres containing ammonia, the authority having jurisdiction for enforcement of this Code shall be permitted to reclassify the location to a less hazardous location or a nonhazardous location.

(FPN No. 1): For additional information on the properties and group classification of Class I materials, see *Manual for Classification of Gases, Vapors, and Dusts for Electrical Equipment in Hazardous (Classified) Locations,* NFPA 497M-1991, and *Guide to Fire Hazard Properties of Flammable Liquids, Gases, and Volatile Solids,* NFPA 325–1994.

(FPN No. 2): The explosion characteristics of air mixtures of gases or vapors vary with the specific material involved. For Class I locations, Groups A, B, C, and D, the classification involves determinations of maximum explosion pressure and maximum safe clearance between parts of a clamped joint in an enclosure. It is necessary, therefore, that equipment be approved not only for class but also for the specific group of the gas or vapor that will be present.

(FPN No. 3): Certain chemical atmospheres may have characteristics that require safeguards beyond those required for any of the above groups. Carbon disulfide is one of these chemicals because of its low ignition temperature[100°C (212°F)] and the small joint clearance permitted to arrest its flame.

(FPN No. 4): For classification of areas involving ammonia atmosphere, see *Safety Code for Mechanical Refrigeration,* ANSI/ASHRAE 15-1992, and *Safety Requirements for the Storage and Handling of Anhydrous Ammonia,* ANSI/CGA G2.1-1989.

(b) Class II Group Classifications. Class II groups shall be as follows:

(1)* Group E. Atmospheres containing combustible metal dusts, including aluminum, magnesium, and their commercial alloys, or other combustible dusts whose particle size, abrasiveness, and conductivity present similar hazards in the use of electrical equipment.

(FPN): Certain metal dusts may have characteristics that require safeguards beyond those for atmospheres containing the dusts of aluminum, magnesium, and their commercial alloys. For example, zirconium, throium, and uranium dusts have extremely low ignition temperatures [as low as 20°C (68°F)] and minimum ignition energies lower than any material classified in any of the Class I or Class II Groups.

(2)* Group F. Atmospheres containing combustible carbonaceous dusts, including carbon black, charcoal, coal, or dusts that have been sensitized by other material so that they present an explosion hazard.

(FPN): See *Standard Test Method for Volatile Materials in the Analysis Sample for Coal and Coke,* ASTM D 3175-1989.

(3) Group G. Atmospheres containing combustible dusts not included in Group E or F, including flour, grain, wood, plastic, and chemicals.

(FPN No. 1): For additional information on group classification of Class II materials, see *Manual for Classification of Gases, Vapors, and Dusts for Electrical Equipment in Hazardous (Classified) Locations,* NFPA 497M-1991.

(FPN No. 2): The explosion characteristics of air mixtures of dust vary with the materials involved. For Class II locations, Groups E, F, and G, the classification involves the tightness of the joints of assembly and shaft openings to prevent the entrance of dust in the dust-ignitionproof enclosure, the blanketing effect of layers of dust on the equipment that may cause overheating, and the ignition temperature of the dust. It is necessary, therefore, that equipment be approved not only for the class, but also for the specific group of dust that will be present.

(FPN No. 3): Certain dusts may require additional precautions due to chemical phenomena that can result in the generation of ignitable gases. See *National Electrical Safety Code,* ANSI C2-1993, Section 127A-Coal Handling Areas.

(c) Approval for Class and Properties. Equipment, regardless of the classification of the location in which it is installed, that depends on a single compression seal, diaphragm, or tube to prevent flammable or combustible fluids from entering the equipment, shall be for a Class I, Division 2 location.

Exception: Equipment installed in a Class I, Division 1 location shall be suitable for the Division 1 location.

(FPN): See Section 501-5(f)(3) for additional requirements.

Equipment shall be approved not only for the class of location but also for the explosive, combustible, or ignitable properties of the specific gas, vapor, dust, fiber, or flyings that will be present. In addition, Class I equipment shall not have any exposed surface that operates at a temperature in excess of the ignition temperature of the specific gas or vapor. Class II equipment shall not have an external temperature higher than that specified in Section 500-3(f). Class III equipment shall not exceed the maximum surface temperatures specified in Section 503-1.

Equipment that has been approved for a Division 1 location shall be permitted in a Division 2 location of the same class or group.

Where specifically permitted in Articles 501 through 503, general-purpose equipment or equipment in general-purpose enclosures shall be permitted to be installed in Division 2 locations if the equipment does not constitute a source of ignition under normal operating conditions.

Unless otherwise specified, normal operating conditions for motors shall be assumed to be rated full-load steady conditions.

Where flammable gases or combustible dusts are or may be present at the same time, the simultaneous presence of both shall be considered when determining the safe operating temperature of the electrical equipment.

(FPN): The characteristics of various atmospheric mixtures of gases, vapors, and dusts depend on the specific material involved.

Optical fiber cables and fiber optic devices approved for hazardous (classified) locations shall be installed in accordance with Sections 504-20 and 770-52.

Exception: Optical fiber cables or devices that are conductive shall also be installed in accordance with Articles 500 through 503.

(d) Marking. Approved equipment shall be marked to show the class, group, and operating temperature or temperature range referenced to a 40°C ambient.

(FPN): Equipment not marked to indicate a division, or marked "Division 1" or "Div. 1," is suitable for both Division 1 and 2 locations. Equipment marked "Division 2" or "Div. 2" is suitable for Division 2 locations only.

The temperature range, if provided, shall be indicated in identification numbers, as shown in Table 500-3(d).

Exception: As required in Section 505-10(b).

Identification numbers marked on equipment nameplates shall be in accordance with Table 500-3(d).

(continued)

Table 14-8B-1
(Continued)

Exception: As required in Section 505-10(b).

Equipment that is approved for Class I and Class II shall be marked with the maximum safe operating temperature, as determined by simultaneous exposure to the combinations of Class I and Class II conditions.

Exception No. 1: Equipment of the nonheat-producing type, such as junction boxes, conduit, and fittings, and equipment of the heat-producing type having a maximum temperature not more than 100°C (212°F) shall not be required to have a marked operating temperature or temperature range.

Exception No. 2: Fixed lighting fixtures marked for use in Class I, Division 2 or Class II, Division 2 locations only shall not be required to be marked to indicate the group.

Exception No. 3: Fixed general-purpose equipment in Class I locations, other than fixed lighting fixtures, that is acceptable for use in Class I, Division 2 locations shall not be required to be marked with the class, group, division, or operating temperature.

Exception No. 4: Fixed dusttight equipment other than fixed lighting fixtures that are acceptable for use in Class II, Division 2 and Class III locations shall not be required to be marked with the class, group, division, or operating temperature.

Exception No. 5: Electric equipment suitable for ambient temperatures exceeding 40°C shall be marked with both the maximum ambient temperature and the operating temperature or temperature range at that ambient temperature.

Table 500-3(d). Identification Numbers

Maximum Temperature		
Degrees C	**Degrees F**	**Identification Number**
450	842	T1
300	572	T2
280	536	T2A
260	500	T2B
230	446	T2C
215	419	T2D
200	392	T3
180	356	T3A
165	329	T3B
160	320	T3C
135	275	T4
120	248	T4A
100	212	T5
85	185	T6

(FPN): Since there is no consistent relationship between explosion properties and ignition temperature, the two are independent requirements.

(e) Class I Temperature. The temperature marking specified in (d) above shall not exceed the ignition temperature of the specific gas or vapor to be encountered.

Exception: Where area classification is in accordance with Article 505, the temperature marking in Section 505-10(b) shall not exceed ignition temperature of the specific gas or vapor to be encountered.

(FPN): For information regarding ignition temperature of gases and vapors, see *Manual for Classification of Gases, Vapors, and Dusts for Electrical Equipment in Hazardous (Classified) Locations,* NFPA 497M-1991, and *Guide to Fire Hazard Properties of Flammable Liquids, Gases, and Volatile Solids,* NFPA 325-1994.

(f) Class II Temperature. The temperature marking specified in (d) above shall be less than the ignition temperature of the specific dust to be encountered. For organic dusts that may dehydrate or carbonize, the temperature marking shall not exceed the lower of either the ignition temperature or 165°C (329°F). See *Manual for Classification of Gases, Vapors, and Dusts for Electrical Equipment in Hazardous (Classified) Locations,* NFPA 497M-1991., for minimum ignition temperatures of specific dusts.

The ignition temperature for which equipment was approved prior to this requirement shall be assumed to be as shown in Table 500-3(f).

500-4. Specific Occupancies Articles 510 through 517 cover garages, aircraft hangars, gasoline dispensing service stations, bulk storage plants, spray application, dipping and coating processes, and health care facilities.

500-5. Class I Locations. Class I locations are those in which flammable gases or vapors are or may be present in the air in quantities sufficient to produce explosive or ignitable mixtures. Class I locations shall include those specified in (a) and (b) below.

(a) Class I, Division 1. A Class I, Division 1 location is a location (1) in which ignitable concentrations of flammable gases or vapors can exist under normal operating conditions; or (2) in which ignitable concentrations of such gases or vapors may exist frequently because of repair or maintenance operations or because of leakage; or (3) in which breakdown or faulty operation of equipment or processes might release ignitable concentrations of flammable gases or vapors, and might also cause simultaneous failure of electric equipment.

Table 500-3(f)

	Equipment That is Not Subject to Overloading		Equipment (Such as Motors or Power Transformers) That May Be Overloaded			
			Normal Operation		Abnormal Operation	
Class II Group	Degrees C	Degrees F	Degrees C	Degrees F	Degrees C	Degrees F
E	200	392	200	392	200	392
F	200	392	150	302	200	392
G	165	329	120	248	165	329

(FPN No. 1): This classification usually includes locations where volatile flammable liquids or liquefied flammable gases are transferred from one container to another; interiors of spray booths and areas in the vicinity of spraying and painting operations where volatile flammable solvents are used; locations containing open tanks or vats of volatile flammable liquids; drying rooms or compartments for the evaporation of flammable solvents; locations containing fat and oil extraction equipment using volatile flammable solvents; portions of cleaning and dyeing plants where flammable liquids are used; gas generator rooms and other portions of gas manufacturing plants where flammable gas may escape; inadequately ventilated pump rooms for flammable gas or for volatile flammable liquids; the interiors of refrigerators and freezers in which volatile flammable materials are stored in open, lightly stoppered, or easily ruptured containers; and all other locations where ignitable concentrations of flammable vapors or gases are likely to occur in the course of normal operations.

(FPN No. 2): In some Division I locations, ignitable concentrations of flammable gases or vapors may be present continuously or for long periods of time. Examples include the inside of inadequately vented enclosures containing instruments normally venting flammable gases or vapors to the interior of the enclosure, the inside of vented tanks containing volatile flammable liquids, the area between the inner and outer roof sections of a floating roof tank containing volatile flammable fluids, inadequately ventilated areas within spraying or coating operations using volatile flammable fluids, and the interior of an exhaust duct that is used to vent ignitable concentrations of gases or vapors. Experience has demonstrated the prudence of (a) avoiding the installation of instrumentation or other electric equipment in these particular area altogether or, (b) where it cannot be avoided because it is essential to the process and other locations are not feasible (see Section 500-2, first FPN), using electric equipment or instrumentation approved for the specific application or consisting of intrinsically safe systems as described in Article 504.

(b) Class I, Division 2. A Class I, Division 2 location is a location (1) in which volatile flammable liquids or flammable gases are handled, processed, or used, but in which the liquids, vapors, or gases will normally be confined within closed containers or closed systems from which they can escape only in case of accidental rupture or breakdown of such containers or systems, or in case of abnormal operation of equipment; or (2) in which ignitable concentrations of gases or vapors are normally prevented by positive mechanical ventilation, and which might become hazardous through failure or abnormal operation of the ventilating equipment; or (3) that is adjacent to Class I, Division 1 location, and to which ignitable concentrations of gases or vapors might occasionally be communicated unless such communication is prevented by adequate positive-pressure ventilation from a source of clean air, and effective safeguards against ventilation failure are provided.

(FPN No. 1): This classification usually includes locations where volatile flammable liquids or flammable gases or vapors are used but that, in the judgment of the authority having jurisdiction, would become hazardous only in case of an accident or of some unusual operating condition. The quantity of flammable material that might escape in case of accident, the adequacy of ventilating equipment, the total area involved, and the record of the industry or business with respect to explosions or fires are all factors that merit consideration in determining the classification and extent of each location.

(FPN No. 2): Piping without valves, checks, meters, and similar devices would not ordinarily introduce a hazardous condition even though used for flammable liquids or gases. Locations used for the storage of flammable liquids or of liquefied or compressed gases in sealed containers would not normally be considered hazardous unless also subject to other hazardous conditions.

500-6. Class II Locations. Class II locations are those that are hazardous because of presence of combustible dust. Class II locations shall include those specified in (a) and (b) below.

(a) Class II, Division 1. A Class II, Division 1 location is a location (1) in which combustible dust is in the air under normal operating conditions in quantities sufficient to produce explosive or ignitable mixtures; or (2) where mechanical failure or abnormal operation of machinery or equipment might cause such explosive or ignitable mixtures to be produced, and might also provide a source of ignition through simultaneous failure of electric equipment, operation of protection devices, or from other causes; or (3) in which combustible dusts of an electrically conductive nature may be present in hazardous quantities.

(FPN): Combustible dusts that are electrically nonconductive include dusts produced in the handling and processing of grain and grain products, pulverized sugar and cocoa, dried egg and milk powders, pulverized spices, starch and pastes, potato and woodflour, oil meal from beans and seed, dried hay, and other organic materials that may produce combustible dusts when processed or handled. Only Group E dusts are considered to be electrically conductive for classification purposes. Dusts containing magnesium or aluminum are particularly hazardous, and the use of extreme precaution will be necessary to avoid ignition and explosion.

(continued)

Table 14-8B-1
(Continued)

(b) Class II, Division 2. A Class II, Division 2 location is a location where combustible dust is not normally in the air in quantities sufficient to produce explosive or ignitable mixtures, and dust accumulations are normally insufficient to interfere with the normal operation of electrical equipment or other apparatus, but combustible dust may be in suspension in the air as a result of infrequent malfunctioning of handling or processing equipment and where combustible dust accumulations on, in or in the vicinity of the electrical equipment may be sufficient to interfere with the safe dissipation of heat from electrical equipment or may be ignitable by abnormal operation or failure of electrical equipment.

(FPN No. 1): The quantity of combustible dust that may be present and the adequacy of dust removal systems are factors that merit consideration in determining the classification and may result in an unclassified area.

(FPN No. 2): Where products such as seed are handled in a manner that produces low quantities of dust, the amount of dust deposited may not warrant classification.

500-7. Class III Locations. Class III locations are those that are hazardous because of the presence of easily ignitable fibers or flyings, but in which such fibers or flyings are not likely to be in suspension in the air in quantities sufficient to produce ignitable mixtures. Class III locations shall include those specified in (a) and (b) below.

(a) Class III, Division 1. A Class III, Division 1 location is a location in which easily ignitable fibers or materials producing combustible flyings are handled, manufactured, or used.

(FPN No. 1): Such locations usually include some part of rayon, cotton, and other textile mills; combustible fiber manufacturing and processing plants; cotton gins and cotton-seed mills; flax-processing plants; clothing manufacturing plants; woodworking plants; and establishments and industries involving similar hazardous processes or conditions.

(FPN No. 2): Easily ignitable fibers and flyings include rayon, cotton (including cotton linters and cotton waste), sisal or henequen, istle, jute, hemp, tow, cocoa fiber, oakum, baled waste kapok, Spanish moss, excelsior, and other materials of similar nature.

(b) Class III, Division 2. A Class III, Division 2 location is a location in which easily ignitable fibers are stored or handled.

Exception: In process of manufacture.

After the 1996 NEC was issued, Section 500-3 was appealed. See p. 20.

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Table 14-8B-2
NFPA-70 Article 505—Class I, Zone 0, 1, and 2 Locations

505-2. General Requirements. The general rules of this *Code* shall apply to the electrical wiring and equipment in locations classified as Class I, Zone 0, Zone 1, or Zone 2.

Exception: As modified by this article.

505-5. Grouping and Classification. For purposes of testing, approval, and area classification, various air mixtures (not oxygen enriched) shall be grouped as follows:

(FPN): Group I electric apparatus is intended for use in underground mines. See Section 90-2(b)(2).

Group II is subdivided according to the nature of the gas atmosphere, as follows.

(a) Group IIC. Atmospheres containing acetylene, hydrogen, or gases or vapors of equivalent hazard.

(FPN): This grouping is equivalent to Class I, Groups A and B, as described in Sections 500-3(a)(1) and (a)(2).

(b) Group IIB. Atmospheres containing acetaldehyde, ethylene, or gases or vapors of equivalent hazard.

(FPN): This grouping is equivalent to Class I, Group C, as described in Section 500-3(a)(3).

(c) Group IIA. Atmospheres containing acetone, ammonia, ethyl alcohol, gasoline, methane, propane, or gases or vapors of equivalent hazard.

(FPN No. 1): This grouping in (c) above is equivalent to Class I, Group D, as described in Section 500-3(a)(4).

(FPN No. 2): The gas subdivision in (a), (b), and (c) above is based on the maximum experimental safe gap, minimum igniting current, or both. The test apparatus for determining the maximum experimental safe gap is described in IEC Publication 79-1A (1975) and UL Technical Report No. 58 (1993).

(FPN No. 3): The classification of mixtures of gases or vapors according to their maximum experimental safe gaps and minimum igniting currents is described in IEC publication 79-12 (1978).

(FPN No. 4): It is necessary that the meaning of the different equipment markings and Group II classifications be carefully observed to avoid confusion with Class I, Divisions 1 and 2, Groups A, B, C, and D.

505-7. Zone Classification. The classification into zones shall be in accordance with the following:

(a) Class I, Zone 0. A Class I, Zone 0 location is a location (1) in which ignitable concentrations of flammable gases or vapors are present continuously; or (2) in which ignitable concentrations of flammable gases or vapors are present for long periods of time.

(FPN No. 1): As a guide in determining when flammable gases are present continuously, for long periods, or under normal conditions, refer to, *Recommended Practice for Classification of Locations for Electrical Installations of Petroleum Facilities,* API RP 500-1991, *Electrical Apparatus for Explosive Gas Atmospheres, Classifications of Hazardous Areas,* IEC 79-10 and, *Institute of Petroleum Area Classification Code for Petroleum Installations,* IP 15.

(FPN No. 2): The classification includes locations inside vented tanks or vessels containing volatile flammable liquids; inside inadequately vented spraying or coating enclosures where volatile flammable solvents are used; between the inner and outer roof sections of a floating roof tank containing volatile flammable liquids; inside open vessels, tanks, and pits containing volatile flammable liquids; the interior of an exhaust duct that is used to vent ignitable concentrations of gases or vapors; and inside inadequately ventilated enclosures containing normally venting instruments utilizing or analyzing flammable fluids and venting to the inside of the enclosures.

(FPN No. 3): It is not good practice to install electrical equipment in Zone 0 locations except when the equipment is essential to the process or when other locations are not feasible. (See Section 500-2.) If it is necessary to install electrical systems in Zone 0 locations, it is good practice to install intrinsically safe systems as described by Article 504.

(FPN No. 4): Normal operations is considered the situation when plant equipment is operating within its design parameters. Minor releases of flammable material may be part of normal operations. Minor releases include the releases from seals that rely on wetting by the fluid being pumped, handled, or processed. Failures that involve repair or shutdown (such as the breakdown of pump seals and flange gaskets, and spillages caused by accidents) are not considered normal operation.

(b) Class I, Zone 1. A Class I, Zone 1 location is a location (1) in which ignitable concentrations of flammable gases or vapors are likely to exist under normal operating conditions; or (2) in which ignitable concentrations of flammable gases or vapors may exist frequently because of repair or maintenance operations or because of leakage; or (3) in which equipment is operated or processes are carried on, of such a nature that equipment breakdown or faulty operations could result in the release of ignitable concentrations of flammable gases or vapors and also cause simultaneous failure of electrical equipment in a mode to cause the electrical equipment to become a source of ignition; or (4) that is adjacent to a Class I, Zone 0 location from which ignitable concentrations of vapors could be communicated, unless communication is prevented by adequate positive-pressure ventilation from a source of clean air and effective safeguards against ventilation failure are provided.

(FPN): This classification usually includes locations where volatile flammable liquids or liquefied flammable gases are transferred from one container to another, in areas in the vicinity of spraying and painting operations where flammable solvents are used; adequately ventilated drying rooms or compartments for the evaporation of flammable solvents; adequately ventilated locations containing fat and oil extraction equipment using volatile flammable solvents; portions of cleaning and dyeing plants where volatile flammable liquids are used; adequately ventilated gas generator rooms and other portions of gas manufacturing plants where flammable gas may escape; inadequately ventilated pump rooms for flammable gas or for volatile flammable liquids; the interiors of refrigerators and freezers in which volatile flammable materials are stored in the open, lightly stoppered, or easily ruptured containers; and other locations where ignitable concentrations of flammable vapors or gases are likely to occur in the course of normal operation, but not classified Zone 0.

(c) Class I, Zone 2. A Class I, Zone 2 location is a location (1) in which ignitable concentrations of flammable gases or vapors are not likely to occur in normal operation and if they do occur will exist only for a short period; or (2) in which volatile flammable liquids, flammable gases, or flammable vapors are handled, processed, or used, but in which the liquids, gases, or vapors normally are confined within closed containers or closed systems from which they can escape only as a result of accidental rupture or breakdown of the containers or system, or as a result of the abnormal operation of the equipment with which the liquids or gases are handled, processed, or used; or (3) in which ignitable concentrations of flammable gases or vapors normally are prevented by positive mechanical ventilation, but which may become hazardous as the result of failure or abnormal operation of the ventilation equipment; or (4) that is adjacent to a Class I, Zone 1 location, from which ignitable concentrations of flammable gases or vapors could be communicated, unless such communication is prevented by adequate positive-pressure ventilation from a source of clean air, and effective safeguards against ventilation failure are provided.

(FPN): The Zone 2 classification usually includes locations where volatile flammable liquids or flammable gases or vapors are used, but which would become hazardous only in case of an accident or of some unusual operating condition.

505-10. Listing and Marking.

(a) Listing. Equipment that is listed for a Zone 0 location shall be permitted in a Zone 1 or Zone 2 location of the same gas group. Equipment that is listed or is otherwise acceptable for a Zone 1 location shall be permitted in a Zone 2 location of the same group.

(b) Marking. Equipment shall be marked to show the class, zone, gas group, and temperature class referenced to a 40°C (104°F) ambient.

(continued)

Table 14-8B-2
(Continued)

Exception: Electric equipment approved for operation at ambient temperatures exceeding 40°C shall be marked with the maximum ambient temperature for which the equipment is approved, and the operating temperature or temperature range at that ambient temperature.
The temperature class marked on equipment shall be as shown in Table 505-10(b).

Table 505-10(b).
Classification of Maximum Surface Temperature for Group II Electric Apparatus

Temperature Class	T1	T2	T3	T4	T5	T6
Maximum Surface Temperature (°C)	≤ 450	≤ 300	≤ 200	≤ 135	≤ 100	≤ 85

After the 1996 *NEC* was issued, Article 505 was appealed. See p. 20.

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Table 14-8B-3
NFPA-70 Article 510—Hazardous (Classified) Locations—Specific

510-1. Scope. Articles 511 through 517 cover occupancies or parts of occupancies that are or may be hazardous because of atmospheric concentrations of flammable liquids, gases, or vapors, or because of deposits or accumulations of materials that may be readily ignitable.
510-2. General. The general rules of this *Code* shall apply to electric wiring and equipment in occupancies within the scope of Articles 511 through 517, except as such rules are modified in those articles. Where unusual conditions exist in a specific occupancy, the authority having jurisdiction shall judge with respect to the application of specific rules.

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Table 14-8B-4
NFPA-70 Article 515—Bulk Storage Plants

Including Class 1 Locations Table, and NFPA-70 Figure 515.2, "Marine Terminal Handling Flammable Liquids"

515-1. Definition. A bulk storage plant is that portion of a property where flammable liquids are received by tank vessel, pipelines, tank car, or tank vehicle, and are stored or blended in bulk for the purpose of distributing such liquids by tank vessel, pipeline, tank car, tank vehicle, portable tank, or container.
(FPN): For further information, see *Flammable and Combustible Liquids Code,* NFPA 30-1993 (ANSI).
515-2.* Class I Locations. Table 515-2 shall be applied where Class I liquids are stored, handled, or dispensed and shall be used to delineate and classify bulk storage plants. The Class I location shall not extend beyond a floor, wall, roof, or other solid partition that has no communicating openings.
(FPN): The area classifications listed in Table 515-2 are based on the premise that the installation meets the applicable requirements of *Flammable and Combustible Liquids Code,* NFPA 30-1993 (ANSI), Chapter 5, in all respects. Should this not be the case, the authority having jurisdiction has the authority to classify the extent of the classified space.
515-3. Wiring and Equipment within Class I Locations. All electric wiring and equipment within the Class I locations defined in Section 515-2 shall comply with the applicable provision of Article 501.
Exception: As permitted in Section 515-5.
515-4. Wiring and Equipment Above Class I Locations. All fixed wiring above Class I locations shall be in metal raceways or PVC schedule 80 rigid nonmetallic conduit, or equivalent, or be Type MI, TC, or Type MC cable. Fixed equipment that may produce arcs,

sparks, or particles of hot metal, such as lamps and lampholders for fixed lighting, cutouts, switches, receptacles, motors, or other equipment having make-and-break or sliding contacts, shall be of the totally enclosed type or be so constructed as to prevent escape of sparks or hot metal particles. Portable lamps or other utilization equipment and their flexible cords shall comply with the provisions of Article 501 for the class of location above which they are connected or used.

515-5. Underground Wiring.

(a) Wiring Method. Underground wiring shall be installed in threaded rigid metal conduit or threaded steel intermediate metal conduit or, where buried under not less than 2 ft (610 mm) of cover, shall be permitted in rigid nonmetallic conduit or an approved cable. Where rigid nonmetallic conduit is used, threaded rigid metal conduit or threaded steel intermediate metal conduit shall be used for the last 2 ft (610 mm) of the conduit run to emergence or to the point of connection to the aboveground raceway. Where cable is used, it shall be enclosed in threaded rigid metal conduit or threaded steel intermediate metal conduit from the point of lowest buried cable level to the point of connection to the aboveground raceway.

(b) Insulation. Conductor insulation shall comply with Section 501-13.

(c) Nonmetallic Wiring. Where rigid nonmetallic conduit or cable with a nonmetallic sheath is used, an equipment grounding conductor shall be included to provide for electrical continuity of the raceway system and for grounding of noncurrent-carrying metal parts.

515-6. Sealing. Approved seals shall be provided in accordance with Section 501-5. Sealing requirements in Sections 501-5(a)(4) and (b)(2) shall apply to horizontal as well as to vertical boundaries of the defined Class I locations. Buried raceways under defined Class I locations shall be considered to be within a Class I, Division 1 location.

515-2.* Class I Locations—Bulk Plants

Location	Class I, Division	Extent of Classified Location
Indoor equipment installed in accordance with *Flammable and Combustible Liquids Code,* NFPA 30-1993 (ANSI), Section 5-3.3.2 where flammable vapor-air mixtures may exist under normal operation.	1	Space within 5 ft of any edge of such equipment, extending in all directions.
	2	Space between 5 ft and 8 ft of any edge of such equipment, extending in all directions. Also, space up to 3 ft above floor or grade level within 5 ft to 25 ft horizontally from any edge of such equipment.*
Outdoor equipment of the type covered in *Flammable and Combustible Liquids Code,* NFPA, 30-1993 (ANSI), Section 5-3.3.2, where flammable vapor-air mixtures may exist under normal operation	1	Space within 3 ft of any edge of such equipment, extending in all directions.
	2	Space between 3 ft and 8 ft of any edge of such equipment, extending in all directions. Also, space up to 3 ft above floor or grade level within 3 ft to 10 ft horizontally from any edge of such equipment.
Tank—Aboveground**	1	Space inside dike where dike height is greater than the distance from the tank to the dike for more than 50 percent of the tank circumference.
Shell, Ends, or Roof and Dike Space	2	Within 10 ft from shell, ends, or roof of tank. Space inside dikes to level of top of dike.
Vent	1	Within 5 ft of open end of vent, extending in all directions.
	2	Space between 5 ft and 10 ft from open end of vent, extending in all directions.
Floating Roof	1	Space above the roof and within the shell
Underground Tank Fill Opening	1	Any pit, box, or space below grade level, if any part is within a Division 1 or 2 classified location.
	2	Up to 18 in. above grade level within a horizontal radius of 10 ft from a loose fill connection, and within a horizontal radius of 5 ft from a tight fill connection.
Vent—Discharging Upward	1	Within 3 ft of open end of vent, extending in all directions.
	2	Space between 3 ft and 5 ft of open end of vent, extending in all directions.
Drum and Container Filling		
Outdoors, or Indoors with Adequate Ventilation	1	Within 3 ft of vent fill openings, extending in all directions.
	2	Space between 3 ft and 5 ft from vent or fill opening, extending in all directions. Also, up to 18 in. above floor or grade level within a horizontal radius of 10 ft from vent or fill openings.

(continued)

Class I, Location	Division	Extent of Classified Location
Pumps, Bleeders, Withdrawal Fittings, Meters, and Similar Devices		
Indoors	2	Within 5 ft of any edge of such devices, extending in all directions. Also up to 3 ft above floor or grade level within 25 ft horizontally from any edge of such devices.
Outdoors	2	Within 3 ft of any edge of such devices, extending in all directions. Also up to 18 in. above grade level within 10 ft horizontally from any edge of such devices.
Pits		
Without Mechanical Ventilation	1	Entire space within pit if any part is within a Division 1 or 2 classified location.
With Adequate Mechanical Ventilation	2	Entire space within pit if any part is within a Division 1 or 2 classified location.
Containing Valves, Fittings, or Piping, and not within a Division 1 or 2 Classified Location	2	Entire pit
Drainage Ditches, Separators, Impounding Basins		
Outdoor	2	Space up to 18 in. above ditch, separator, or basin. Also up to 18 in. above grade within 15 ft horizontally from any edge.
Indoor		Same as pits.
Tank Vehicle and Tank Car†		
Loading through Open Dome	1	Within 3 ft of edge of dome, extending in all directions.
	2	Space between 3 ft and 15 ft from edge of dome, extending in all directions.
Loading through Bottom Connections with Atmospheric Venting	1	Within 3 ft of point of venting to atmosphere, extending in all directions.
	2	Space between 3 ft and 15 ft form point of venting to atmosphere, extending in all directions. Also up to 18 in. above grade within a horizontal radius of 10 ft from point of loading connection.
Office and Rest Rooms	Ordinary	If there is any opening to these rooms within the extent of an indoor classified location, the room shall be classified the same as if the wall, curb, or partition did not exist.
Loading through Closed Dome with Atmospheric Venting	1	Within 3 ft of open end of vent, extending in all directions.
	2	Space between 3 ft and 15 ft from open end of vent, extending in all directions. Also within 3 ft of edge of dome, extending in all direction.
Loading through Closed Dome with Vapor Control	2	Within 3 ft of point of connection of both fill and vapor lines, extending in all directions.
Bottom Loading with Vapor Control Any Bottom Unloading	2	Within 3 ft of point of connections, extending in all directions. Also up to 18 in. above grade within a horizontal radius of 10 ft from point of connections.
Storage and Repair Garage for Tank Vehicles	1	All pits or spaces below floor level.
	2	Space up to 18 in. above floor or grade level for entire storage or repair garage.
Garages for other than Tank Vehicles	Ordinary	If there is any opening to these rooms within the extent of an outdoor classified location, the entire room shall be classified the same as the space classification at the point of opening.
Outdoor Drum Storage	Ordinary	
Indoor Warehousing Where There Is No Flammable Liquid Transfer	Ordinary	If there is any opening to these rooms within the extent of an indoor classified location, the room shall be classified the same as if the wall, curb, or partition did not exist.
Piers and Wharves		See Figure 515-2.

*The release of Class I liquids may generate vapors to the extent that the entire building, and possibly a zone surround it, should be considered a Class I, Division 2 location.

**For Tanks—Underground, see Section 514-2.

When classifying extent of space, consideration shall be given to fact that tank cars or tank vehicles may be spotted at varying points. Therefore, the extremities of the loading or unloading positions shall be used.

For SI units: 1 in. = 25.4 mm; 1 ft = 0.3048 m.

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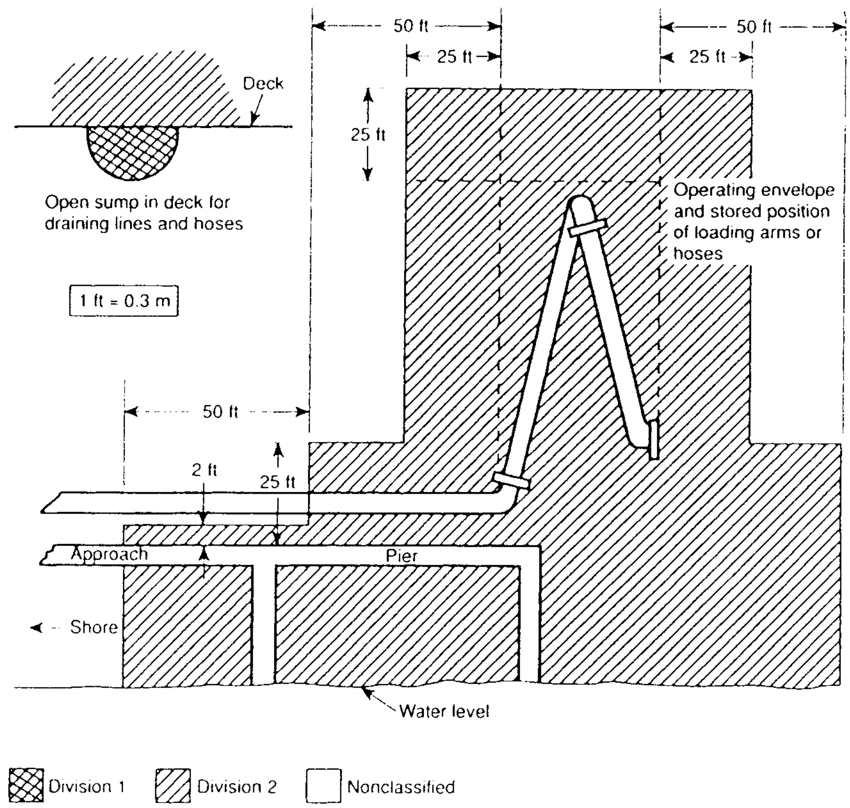
Notes:



(1) The "source of vapor" shall be the operating envelope and stored position of the outboard flange connection of the loading arm (or hose).
(2) The berth area adjacent to tanker and barge cargo tanks is to be Division 2 to the following extent:
a. 25 ft (7.6 m) horizontally in all directions on the pier side from that portion of the hull containing cargo tanks.
b. From the he water level to 25 ft (7.6 m) above the cargo tanks at their highest position.
(3) Additional locations may have to be classified as required by the presence of other sources of flammable liquids on the berth, or by Coast Guard or other regulations.

Figure 512-2.x Marine Terminal Handling Flammable Liquids

then more specifically classified with respect to the properties of the flammable vapors, liquids, gases, combustible dusts, and fibers or any special materials that can be brought into the area by any means. In addition, the possibility of these potentially hazardous materials contributing to combustion levels must be considered. You must understand the flammability range of all possible hazardous materials—that is, the lower and upper flammable limits. Published tables are available for a wide listing of organic and inorganic compounds including dusts. In determining the hazard classification, each local plant area must be considered separately, as various areas can be classified differently, and the presence of barrier walls can influence the establishment of a classification. See Volume 1, 3rd Ed., Chapter 7 of this series.

After an area classification has been established, then *all* electrical equipment, instruments, and wiring must adhere to that classification. Therefore, some motor enclosure types would not be electrically acceptable for specific area classifications.

Motors are designed according to a 40°C ambient air temperature and are marked to show the *operating* temperature or temperature range (NEMA, Article 500-2 (b)) for operation. These temperatures are not to exceed the ignition temperatures of the specific gas, vapor, or dust expected in the environment. The maximum temperatures for which motors are marked or identified by temperature class is given in Table 14-8B-1 (NFPA Table 500-3d), also see Figure 14-11.

Note that no consistent relationship exists between explosion properties and ignition temperatures.

Electrical Classification for Safety in Plant Layout

See Tables 14-8B-1, 14-8B -2, 14-8B -3, 14-8B -4 and 14-9.

The process designer and project engineer should classify the various areas of a plant following NFPA-70, Article 505[28, 29] in order to advise the electrical and other project team members of the degree of electrical hazards anticipated. The appropriate equipment (motors, instruments, conduit, wiring, etc.) should be specified according to NFPA-70, Article 500 and others as applicable,[28] the ASME Code and the API Code as appropriate. See NFPA-497A and

497B for *Classifications of Class I and Class II Hazardous Locations for Electrical Installations in Chemical Process Areas,* reference 29; NFPA-321 for *Classification of Flammable and Combustible Liquids,* reference 29, and NFPA-325 for *Guide to Fire Hazards Properties of Flammable Liquids, Gases and Volatile Solids,* reference 29. Major fires and explosions have occurred in chemical plants, petrochemical plants, and refineries when flammable liquids leaked into local plant drainage (recessed trench in floor) and traveled hundreds of feet to ultimately reach a nonhazardous classified area, where it then became ignited by a spark or hot surface.

As you can see, it is important to plan the extent of the hazardous areas and various methods, such as fire walls, to prevent the transfer or propagation of volatile/flammable materials from one plant area to another. When in doubt, err on the side of being safe and conservative. Also keep in mind that dust can be just as hazardous as flammable liquids/vapors and must be classified and handled accordingly. Tables 14-9 and 14-10 identify the NEMA Standard for Motors.[31]

To aid in establishing the hazard classification by the designer, see NFPA-Code 321 and 497A and 497B.[29]

Table 14-9
Classification According to Environmental Protection and Methods of Cooling

Details of protection (IP) and methods of cooling (IC) are defined in Part 5 and Part 6, respectively. They conform to IEC Standards.

1.25a OPEN MACHINE (IP00, IC01)

An open machine is one having ventilating openings which permit passage of external cooling air over and around the windings of the machine. The term "open machine," when applied in large apparatus without qualification, designates a machine having no restriction to ventilation other than that necessitated by mechanical construction.

1.25.1a Dripproof Machine (IP12, IC01)

A dripproof machine is an open machine in which the ventilating openings are so constructed that successful operation is not interfered with when drops of liquid or solid particles strike or enter the enclosure at any angle from 0 to 15 degrees downward from the vertical. The machine is protected against solid objects greater than 2 inches.

1.25.2a Splash-Proof Machine (IP13, IC01)

A splash-proof machine is an open machine in which the ventilating openings are so constructed that successful operation is not interfered with when drops of liquid or solid particles strike or enter the enclosure at any angle not greater than 60 degrees downward from the vertical.

The machine is protected against solid objects greater than 2 inches.

1.25.3a Semi-Guarded Machine (IP10, IC01)

A semi-guarded machine is an open machine in which part of the ventilating openings in the machine, usually in the top half, are guarded as in the case of a "guarded machine" but the others are left open.

1.25.4a Guarded Machine (IP2, IC01)

A guarded machine is an open machine in which all openings giving direct access to live metal or rotating parts (except smooth rotating surfaces) are limited in size by the structural parts or by screens, baffles, grilles, expanded metal, or other means to prevent accidental contact with hazardous parts.

The openings in the machine enclosure shall be such that (1) a probe such as that illustrated in Figure 1-1a, when inserted through the openings, will not touch a hazardous rotating part; (2) a probe such as that illustrated in Figure 1-2a when inserted through the openings, will not touch film-coated wire; and (3) an articulated probe such as that illustrated in Figure 1-3a, when inserted through the openings, will not touch an uninsulated live metal part.

1.25.5a Dripproof Guarded Machine (IP22, IC01)

A dripproof guarded machine is a dripproof machine whose ventilating openings are guarded in accordance with 1.25.4a.

1.25.6a Open Independently Ventilated Machine (IC06)

An open independently ventilated machine is one which is ventilated by means of a separate motor-driven blower mounted on the machine enclosure. Mechanical protection shall be as defined in 1.25.1a to 1.25.5a, inclusive. This machine is sometimes known as a blower-ventilated machine.

1.25.7a Open Pipe-Ventilated Machine

An open pipe-ventilated machine is an open machine except that openings for the admission of the ventilating air are so arranged that inlet ducts or pipes can be connected to them. Open pipe-ventilated machines shall be self-ventilated (air circulated by means integral with the machine (IC11) or forced-ventilated (air circulated by means external to and not a part of the machine) (IC17). Enclosures shall be as defined in 1.25.1a to 1.25.5a, inclusive.

1.25.8a Weather-Protected Machine

1.25.8.1a Type I (IC01)

A weather-protected Type I machine is a guarded machine with its ventilating passages so constructed as to minimize the entrance of rain, snow and air-borne particles to the electric parts.

1.25.8.2a Type II (IC01)

A weather-protected Type II machine shall have, in addition to the enclosure defined for a weather-protected Type I machine, its ventilating passages at both intake and discharge so arranged that high-velocity air and air-borne particles blown into the machine by storms or high winds can be discharged without entering the internal ventilating passages leading directly to the electric parts of the machine itself. The normal path of the ventilating air which enters the electric parts of the machine shall be so arranged by baffling or separate housings as to provide at least three abrupt changes in direction, none of which shall be less than 90 degrees. In addition, an area of low velocity not exceeding 600 feet per minute shall be provided in the intake air path to minimize the possibility of moisture or dirt being carried into the electric parts of the machine.

NOTE—Removable or otherwise easy to clean filters may be provided instead of the low velocity chamber.

1.26a TOTALLY ENCLOSED MACHINE (IP5X)

A totally enclosed machine is one so enclosed as to prevent the free exchange of air between the inside and outside of the case but not sufficiently enclosed to be termed air-tight and dust does not enter in sufficient quantity to interfere with satisfactory operation of the machine.

1.26.1a Totally Enclosed Nonventilated Machine (IP54, IC410)

A totally enclosed nonventilated machine is a frame-surface cooled totally enclosed machine which is only equipped for cooling by free convection.

1.26.2a Totally Enclosed Fan-Cooled Machine (IP14, IC411)

A totally enclosed fan-cooled machine is a frame-surface cooled totally enclosed machine equipped for self-exterior cooling by means of a fan or fans integral with the machine but external to the enclosing parts.

1.26.3a Totally Enclosed Fan-cooled Guarded Machine (IP54, IC411)

A totally-enclosed fan-cooled guarded machine is a totally-enclosed fan-cooled machine in which all openings giving direct access to the fan are limited in size by the design of the structural parts or by screens, grilles, expanded metal, etc., to prevent accidental contact with the fan. Such openings shall not permit the passage of a cylindrical rod 0.75 inch diameter, and a probe such as that shown in Figure 1-1a shall not contact the blades, spokes, or other irregular surfaces of the fan.

1.26.4a Totally Enclosed Pipe-ventilated Machine (IP44)

A totally enclosed pipe-ventilated machine is a machine with openings so arranged that when the inlet and outlet ducts or pipes are connected to them there is no free exchange of the internal air and the air outside the case. Totally enclosed pipe-ventilated machines may be self-ventilated (air circulated by means integral with the machine (IC31)) or forced-ventilated (air circulated by means external to and not a part of the machine (IC37)).

1.26.5a Totally Enclosed Water-cooled Machine (IP54)

A totally-enclosed water-cooled machine is a totally enclosed machine which is cooled by circulating water, the water or water conductors coming in direct contact with the machine parts.

1.26.6a Water Proof Machine (IP55)

A water-proof machine is a totally enclosed machine so constructed that it will exclude water applied in the form of a stream of water from a hose, except that leakage may occur around the shaft provided it is prevented from entering the oil reservoir and provision is made for automatically draining the machine. The means for automatic draining may be a check valve or a tapped hole at the lowest part of the frame which will serve for application of a drain pipe.

1.26.7a Totally Enclosed Air-to-Water-Cooled Machine (IP54)

A totally enclosed air-to-water-cooled machine is a totally enclosed machine which is cooled by circulating air which, in turn, is cooled by circulating water. It is provided with a water-cooled heat exchanger, integral (IC7_W) or machine mounted (IC8_W), for cooling the internal air and a fan or fans, integral with the rotor shaft (IC_1W) or separate (IC_5W) for circulating the internal air.

1.26.8a Totally Enclosed Air-to-Air Cooled Machine (IP54)

A totally enclosed air-to-air cooled machine is a totally enclosed machine which is cooled by circulating the internal air through a heat exchanger which, in turn, is cooled by circulating external air. It is provided with an air-to-air heat exchanger, integral (IC5_), or machine mounted (IC6_), for cooling the internal air and a fan or fans, integral with the rotor shaft (IC_1), or separate, but external to the enclosing part or parts (IC_6), for circulating the external air.

1.26.9a Totally Enclosed Air-Over Machine (IP54, IC417)

A totally enclosed air-over machine is a totally enclosed frame-surface cooled machine intended for exterior cooling by a ventilating means external to the machine.

1.26.10a Explosion-Proof Machine[1]

An explosion-proof machine is a totally enclosed machine whose enclosure is designed and constructed to withstand an explosion of a specified gas or vapor which may occur within it and to prevent the ignition of the specified gas or vapor surrounding the machine by sparks, flashes or explosions of the specified gas or vapor which may occur within the machine casing.

1.26.11a Dust-Ignition-Proof Machine[2]

A dust-ignition-proof machine is a totally enclosed machine whose enclosure is designed and constructed in a manner which will exclude ignitable amounts of dust or amounts which might affect performance or rating, and which will not permit arcs, sparks, or heat otherwise generated or liberated inside of the enclosure to cause ignition of exterior accumulations or atmospheric suspensions of a specific dust on or in the vicinity of the enclosure.

Successful operation of this type of machine requires avoidance of overheating from such causes as excessive overloads, stalling, or accumulation of excessive quantities of dust on the machine.

---

[1]See ANSI/NFPA 70, National Electrical Code, Article 500—For Hazardous Locations, Class I, Groups, A, B, C, or D.

[2]See ANSI/NFPA 70, National Electrical Code, Article 500—For Hazardous Locations, Class II, Groups, E, F, or G.

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Table 14-10A
NEMA MG 1-1993, Rev. 1, Section 1, Part 1:
Classification According to Application

(Some of the definitions in this section apply to only specific types or sizes of machines.)

1.06 GENERAL PURPOSE MOTOR

1.06.1 General-Purpose Alternating-Current Motor

A general-purpose alternating-current motor is an induction motor, rated 200 horsepower and less, which incorporates all of the following:

a. Open or enclosed construction
b. Rated continuous duty
c. Service factor in accordance with 12.47
d. Class A or higher rated insulation system with a temperature rise not exceeding that specified in 12.42 for Class A insulation for small motors or Class B or higher rated insulation system with a temperature rise not exceeding that specified in 12.43 for Class B insulation for medium motors.

It is designed in standard ratings with standard operating characteristics and mechanical construction for use under usual service conditions without restriction to a particular application or type of application.

1.06.2 General-Purpose Direct-Current Small Motor

A general-purpose direct-current small motor is a small motor of mechanical construction suitable for general use under usual service conditions and has ratings and constructional and performance characteristics applying to direct-current small motors as given in Parts 10, 11, 12, and 14.

1.07 INDUSTRIAL SMALL MOTOR

An industrial small motor is an alternating-current or direct-current motor built in either NEMA frame 42, 48, or 56 suitable for industrial use.

It is designed in standard ratings with standard operating characteristics for use under usual service conditions without restriction to a particular application or type of application.

1.08 INDUSTRIAL DIRECT-CURRENT MEDIUM MOTOR

An industrial direct-current motor is a medium motor of mechanical construction suitable for industrial sue under usual service conditions and has ratings and constructional and performance characteristics applying to direct current medium motors as given in Parts 10, 11, 12, and 14.

1.09 INDUSTRIAL DIRECT-CURRENT GENERATOR

An industrial direct-current generator is a generator of mechanical construction suitable for industrial use under usual service conditions and has ratings and constructional and performance characteristics applying to direct current generators as given in Parts 11 and 15.

1.10 DEFINITE-PURPOSE MOTOR

A definite-purpose motor is any motor designed in standard rating with standard operating characteristics or mechanical construction for use under service conditions other than usual or for use on a particular type of application.

1.11 GENERAL INDUSTRIAL MOTORS

See 23.01.

1.12 METAL ROLLING MILL MOTORS

See 23.02.

1.13 REVERSING HOT MILL MOTORS

See 23.03.

1.14 SPECIAL-PURPOSE MOTOR

A special-purpose motor is a motor with special operating characteristics or special mechanical construction, or both, designed for a particular application and not falling within the definition of a general purpose or definite-purpose motor.

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Motor Enclosures

Table 14-9 summarizes the more common motor enclosures by types and environment.

Tables 14-10A and 14-10B identify motor enclosures associated with applications.

Special designs of some of these types have been created to fit a particular situation. The motor manufacturer should be advised of the operating conditions and environment.

Motors must be classified to correspond to the proper electrical hazards of Tables 14-8A and 8B, and the manufacturer should be so advised.

For high moisture or humidity conditions, internal heaters are recommended for most motors to prevent condensation during idle periods. Some special insulations may not require this precaution; however, it should be discussed with the manufacturer.

Table 14-10B
Motor Enclosures: Horizontal or Vertical Shafts

Type	Motor	Std. Temp. Rise Class A Insul.	Approx. Cost Increase	Application-Protection
Open, Drip proof	I, S	40°C	0 to 10%	Dripping liquids or falling particles
Protected	I	50°C	10%	Metal chips in machine shops
Splash-proof	I, S	40°C	10 to 15%	Dripping and splashing liquids, breweries, food plants, dairies, etc.
Enclosed collector rings	S	40°C	4 to 30%	Explosive and nonexplosive atmospheres
Enclosed, forced ventilated	I, S	40°C	4 to 20%	Same as fan-cooled
Enclosed, self-ventilated	I, S	40°C	15 to 40%	Same as fan-cooled
Totally-enclosed non-ventilated	I	55°C	10 to 40%	Same as fan-cooled
Totally-enclosed fan-cooled	I, S	55°C	40 to 115% (135% for S)	Abrasive dust, dirt, grit, corrosive fumes too severe for other types
Explosion-proof	I	55°C	10 to 20% higher than fan-cooled	Flammable, volatile liquids atmospheres as in oil refineries, varnish plants, and solvent plants

Note: I = Induction Motor
S = Synchronous Motor

Used by permission: Lincoln, E. S., Ed. *Electrical Reference Book*, Motors and Generators chapter, H-I. Electrical Modernization Bureau; Kropf, V. J., "Motors and Motor Control," *Chem. Eng.*, p. 123, July 1951.

It should be recognized that the Underwriters Laboratories and other testing agencies have not certified all types of electrical equipment (including motors) for operation in hydrogen and acetylene atmospheres. This means that such areas must be arranged for proper ventilation[15] or a non-electrical, nonsparking means of providing the motive power for various pumps, compressors, etc.

Motor Torque

Torque is the turning effort developed by the motor or the resistance to turning exerted by the load.[54] Usually torque is expressed in ft-lb; however, the usual expression is as a percentage of the full load torque. Synchronous motors usually offer several types of torque. *Starting* or breakaway (called *locked rotor*) torque is developed at the instant of starting, see Figure 14-12.

The torque requirements of the driven equipment determine the torque specifications of the motor from initial start to shutdown. NEMA[31] definitions are adapted as follows:

Full-load torque: The torque necessary to produce the rated horsepower at full load speed. In lb at 1 ft radius, it is equal to the horsepower × 5,250 divided by the full load speed.

Locked-rotor (static) torque, starting, or breakaway: The minimum torque that a motor will develop at rest for all angular positions of the rotor, with rated voltage applied at rated frequency.

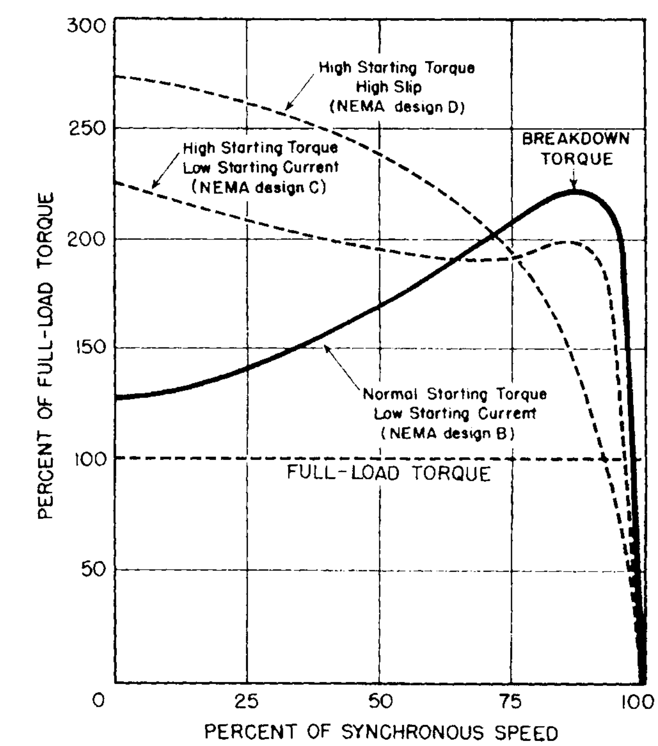

Figure 14-12. Typical torque curves for NEMA design B, C and D; induction motors having synchronous speeds below 1,800 rpm. These curves also apply to some 1,800 rpm design motors. (Used by permission: *E-M Synchronizer*, Bul. 200-TEC-1120, p. 4, ©1955. Dresser-Rand Co.

Pull-up torque: For an alternating current motor, this is the minimum external torque developed by the motor during the period of acceleration from rest to the speed at which breakdown torque occurs. For motors that do not have a definite breakdown torque, the pull-up torque is the minimum torque developed up to the rated speed.

Breakdown torque: For an alternating current motor, this is the maximum torque that it will develop with rated voltage applied at rated frequency, without an abrupt drop in speed.

Pull-out torque: For a synchronous motor, this is the maximum sustained torque that the motor will develop at synchronous speed for one minute with rated voltage applied at rated frequency and with normal excitation.

Pull-in torque: For a synchronous motor, this is the maximum constant torque under which the motor will pull its connected inertia load into synchronism, at rated voltage and frequency, when its field excitation is applied. The speed to which a synchronous motor will bring its load depends on the power required to drive it, and whether the motor can pull the load into step from this speed depends on the inertia of the revolving parts. So, the pull-in torque cannot be determined without having the Wk^2 as well as the torque of the load.

For proper application of a motor:[23, 24]

1. The locked-rotor torque must be 10–250% of full-load torque, depending on the driven equipment.
2. The torque after breakaway for acceleration to full speed must be considerably greater than the torque required by the driven machine. The greater the margin, the shorter will be the time to accelerate the inertia (Wk^2) of the driven equipment and of the motor rotor to full speed. The Wk^2 of the driven equipment must be obtained from the equipment manufacturer and given to the motor manufacturer.
3. The pull-out or breakdown torque must be greater than the maximum torque required by the driven equipment to prevent stalling, usually 150% of full load torque for unity-power factor motors and 220–225% for 0.8 leading power factor motors.[32]

$$\text{Torque} = \frac{5{,}250\ \text{hp}}{\text{rpm}}, \text{ ft-lb} \qquad (14\text{-}11)$$

where torque = tangential effort in lb at 1 ft radius, ft-lb
hp = horsepower developed
rpm = revolutions per minute

All of the torque developed by the motor must either accelerate inertia or overcome load torque.

NEMA normal load Wk^2 values:

$$\text{normal load } Wk^2 = \frac{(3.75)\,(100{,}000)\,(\text{hp})^{1.15}}{(\text{rpm})^2} \qquad (14\text{-}12)$$

Power Factor for Alternating Current

The power factor is the factor by which the apparent kva power is multiplied to obtain the actual power, kw, in an alternating current system. It is the ratio of the in-phase component of the line current to the total current.[32]

In induction motors, the magnetizing component of the current always lags 90°. Therefore, the line current lags at all loads; the magnitude depends upon the magnetizing current load.

In synchronous motors, the excitation is supplied by a separate direct current source, either as a separate motor-generator (M-G) set or as an exciter mounted directly on the motor shaft. The current can be made to lead to various degrees by varying the magnitude of the field strength.

The power factor for motors is rated as lagging, unity, or leading. Using alternating current, the power consumed, called the *active* or *actual power,* is considered the energy used by the resistive load.[47] The synchronous motor supplies a unity or leading factor, and an induction motor provides a unity or lagging factor. "By applying the proper amount of d-c excitation to the field poles of a synchronous motor, it operates at unity power factor. Unity power factor synchronous motors are designed to operate in this way. A full load, with excitation they require no lagging reactive kva from the line, nor do they supply leading reactive kva to the line; they run at unity power factor with a minimum amount of stator current, and hence at highest efficiency."[55]

Review the types of motors proposed for a process plant with a qualified electrical engineer; thereby evaluating whether the mix of synchronous and induction motors will help the net power factor for the plant, because a net lagging factor for plants means that all power to that plant will cost more than if the factor were unity or leading. From Brown and Cadick:[47]

Apparent power = EI, or va, or kva
Active power = EICosθ, or W, or kw
Note: θ = vector diagram angle of current between apparent power and active power
Reactive power = EISinθ, or VAR, or kVAR
Compute power factor:
F_p = active power/apparent power
F_p = EICosθ/(EI) = cosθ
F_p = W/(VAR) = (kw)/(kVAR)

Note that reactive power makes demands on the power system, but does not produce any useful work.

$$\text{Rated motor kva} = \frac{(\text{hp})\,(0.746)}{(\text{Eff})\,(\text{power factor})} \qquad (14\text{-}13)$$

Power changes are based on kVAR demand; thus, the lower the power factor, the higher the demand charge. See Plankenhorn,[67] Valoda,[76] and Lazar[77] for a helpful discussion of this subject.

Power charges are based on VAR demand; thus, the lower the power factor the higher the demand charge.

Most process plants must be careful to maintain a favorable power factor for their system; otherwise a penalty may be placed on power costs. If the power factor falls below some set value—say 0.8—power costs increase because the actual power (as current) going into work (horsepower) is considerably less than the total supplied to the plant system. The difference is that which goes into the magnetizing field (reactive current), which does not represent actual work. By adding synchronous motors or capacitors to an otherwise all-induction load system, you may raise the power factor from a lagging condition to unity (or nearly so). The synchronous motors may be designed to furnish varying amounts of leading power factor. This is a study or balance that must be recognized at the time of plant design, and recommendations should be prepared by competent electrical power engineers.

The usual synchronous motor power factors are unity (1.0) or 0.8 leading. Values of 0.7 or 0.6 leading will give more leading correction to an otherwise lagging system.

Figure 14-13 illustrates the power factor operation of various types of equipment.

The induction motor usually *requires* from 0.3 to 0.6 reactive magnetizing kva per hp of operating load, but an 0.8 leading power factor synchronous motor will *deliver* from 0.4–0.6 corrective magnetizing kva per hp depending on the mechanical load carried. Thus, equal connected hp in induction and 0.8 leading power factor synchronous motors will result in an approximate unity power factor for the system.[32]

$$\text{reactive kva} = \sqrt{(\text{total kva})^2 - (\text{kw})^2} \tag{14-14}$$

This is always lagging for an induction motor. For a synchronous motor of power factor (PF) = 1.0, the kva and kw are equal, and for any PF less than 1.0—that is 0.9, 0.8, 0.7, etc., the PF is leading. Also see references 71, 80, and 82.

Motor Selection

The factors presented in the preceding paragraphs must be evaluated in the light of a given motor application. In general, they may be summarized for a general-purpose application. Also see references 47, 57, and 87.

A. Atmospheric Temperature around Motor

The allowable temperature rise of a motor during operation also implies a maximum safe temperature limit. Temperature cut-off switches usually are incorporated in the motor and control circuit.

B. Horsepower Requirement of Driven Equipment, Including Losses through Gears, Belts, Etc.

The horsepower demand over a reasonable range of operation must be known (i.e., peaks, variations, etc.). The horsepower delivered by the motor from its shaft must be adequate to carry all expected loads.

C. Torque Requirements of Driven Equipment

The motor manufacturer must be given the torque and Wk^2 of the driven load. If not, he must assume this data and run the chance that the motor may not match the requirements. For simple equipment, such as centrifugal pumps or fans, it is usually sufficient to state the service.

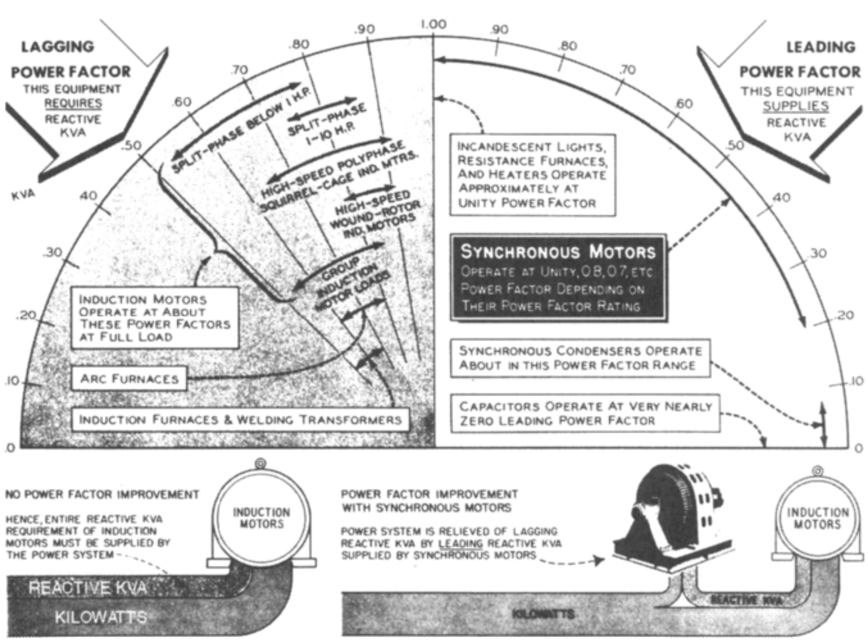

Figure 14-13. Power factor operation of various apparatus and how synchronous motors improve power factor. (Used by permission: *E-M Synchronizer*, 200-SYN-42, ©1955. Dresser-Rand Company.)

D. Voltage, Frequency, Phase

The available operating voltage, phase, and frequency must be given to the motor manufacturer. When the question of off-standard conditions arises, every effort should be made to use the standard or stock motors.

Variations in voltage and frequency must be identified to the manufacturer.

E. Power Factor

The existing and/or required power factor (leading or lagging) must be stated for a synchronous motor.

F. Environment

The geographical location as well as atmospheric conditions (moisture, vapors, dust, etc.) in which the motor must operate should be stated, because the manufacturer has experience in a wide variety of conditions and can recommend the type of motor and materials of construction, such as for fan and insulation. The process engineer must specify the nature of the hazard—that is, whether an explosion-proof motor is required. If it is required, the group hazard must also be identified.

G. Speed

The importance of constant or variable speed will influence the type of motor, motor-gear, motor-belt, or motor-magnetic drive combination.

Table 14-11A is a valuable guide in motor selection.

Table 14-11B is useful in comparing motor temperature rise versus class of insulation. See section NEMA MG-1, Rev. 1, Part 1, Section 1, p. 15, "Classification of Insulation Systems."

Table 14-12 is recommended for summarizing data to and from the manufacturer. All data need not be furnished by the purchaser, but the manufacturer must be given the environment conditions or the equivalent standards established by the purchaser. Table 14-13[41] is a very complete check list for specifying motors.

Figure 14-14 illustrates combinations that might be used to drive a centrifugal compressor through a speed increaser.

Figure 14-15 illustrates a synchronous motor used as the drive for a reciprocating compressor. This is a common arrangement.

Speed Changes

Variations in required speed or speed increase or decrease from the driver may be handled by gear boxes, belt or chain drive, magnetic drive, magnetic or fluid couplings, or variable-speed motors.

The belt drive must be properly grounded. In explosive or hazardous atmospheres, belts are not recommended, but if used, they must be of the anti-sparking design with special brushes to ensure the grounding of static charges.

Table 14-11A
Motor Selection Table

	Motor Symbol	
Application	**Alternating Current**	**Direct Current**
Agitator	1A-1B-2B	6A
Baler (power)	1D	6B-7
Ball mill	1C-2B-3A	6B
Blower (positive pressure)	1A-1B-2B-3A-4	6A
Boring mill	2A-3A	6A-8
Buffer	1A-1B-2A	6A
Cement kiln	3A	8
Compressor	1A-1B-1C-3A-4	6B-8
Conveyor	1A-1C-2B-3A	6B-8
Crane	1D-2A-3B	7
Crusher	1A-1C-1D	6A-6B
Dough mixer	1A-1B-1C-2B	6A-6B
Drilling machine	1A-1B-2A	6A-8
Drying tumbler	1A-1B-1D	6A
Fan (centrifugal and propeller)	1A-1B-2C-3A-4	6A-8
Finishing stand	3B	8
Grinder	1A-1B-2A	6A
Hammer (power)	1D	6B
Hammer mill	1C	6A
Hoist	1D-2A-3B	7
Jordan	1A-1B-4	6A
Keyseater	1A-1B	6A
Lathe	1A-1B-2A	6A-8
Laundry extractor	1C-1D	6B
Laundry washer	1A-1B-1D	6A
Line shaft	1A-1B	6A
Metal grinder	1A-1B	6A
Metal saw	1A-1B	6A
Milling machine	1A-1B-2A	6A-8
Mill table	3A	8
Mine hoist	3B	8
Molder	1A-1B	6A
Ore grinder	3A	8
Pipe threader	1A-1B	6A
Planer	1A-1B	6A
Polisher	1A-1B-2A	6A
Printing press (job)	1A-1B-3A	6B-8
Printing press (rotary and offset)	3A	6B-8
Pulverizer	1C	6B
Pump (centrifugal)	1A-1B-2B-3A-4	6B
Pump (displacement)	1C-2B-3A	6B
Rock crusher	3A	6B-7
Sander	1A-1B	6B
Sand mixer (centrifugal)	1C	6A
Saw (circular)	1A-1B	6A
Saw (band)	1A-1B-1C-3A	6A-6B
Screw machine	1A-1B	6A
Shaper	1A-1B	6A

	Motor Symbol	
Application	Alternating Current	Direct Current
Spinning and weaving machinery	1A-1B	6A
Stoker	1A-1B-1C-2B	6A-8
Tumbling barrel	1C	6A
Winch	1D-3A	6B-8

Explanation of Symbols

1. Squirrel-Cage, Constant-Speed
   A. Normal torque, normal starting current
   B. Normal torque, low starting current
   C. High torque, low starting current
   D. High torque, high slip
   E. Elevator
2. Squirrel-Cage, Multi-Speed
   A. Constant horsepower
   B. Constant torque
   C. Variable torque
   Note: Classes A, B, and C listed under "1" are also applicable to multispeed motors. The listing (A, B, or C) for the constant-speed motor indicates the form of motor to use under "2." For example: if the listing shows 1C-2B, then the multispeed, constant torque motor should also be high torque, low starting current.
3. Wound-Rotor
   A. General-purpose
   B. Crane and hoist
4. Synchronous
5. Direct-Current, Constant-Speed
   A. Shunt-wound
   B. Compound-wound
6. Direct-Current, Variable-Speed, Series-Wound
   Note: Series motors must be connected directly to the load (not belted).
7. Direct-Current, Adjustable-Speed

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Table 14-11B
Classification of Insulation Systems

Insulation systems are divided into classes according to the thermal endurance of the system for temperature rating purposes. Four classes of insulation systems are used in motors and generators, namely, classes A, B, F, and H. These classes have been established in accordance with IEEE Std. 1.

Insulation systems shall be classified as follows:

Class A—An insulation system that, by experience or accepted test, can be shown to have suitable thermal endurance when operating at the limiting Class A temperature specified in the temperature rise standard for the machine under consideration.

Class B—An insulation system that, by experience or accepted test, can be shown to have suitable thermal endurance when operating at the limiting Class B temperature specified in the temperature rise standard for the machine under consideration.

Class F—An insulation system that, by experience or accepted test, can be shown to have suitable thermal endurance when operating at the limiting Class F temperature specified in the temperature rise standard for the machine under consideration.

Class H—An insulation system that, by experience or accepted test, can be shown to have suitable thermal endurance when operating at the limiting Class H temperature specified in the temperature rise standard for the machine under consideration.

"Experience," as used in this standard, means successful operation for a long time under actual operating conditions of machines designed with temperature rise at or near the temperature rating limit.

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Major motor companies perform commercial tests on their motors at various stages of manufacture and for final conformance to design and specifications. TECO-Westinghouse Motor Co.[46] lists the following, which are reproduced by permission:

> Each motor is given a test per NEMA MG 1-20.46 to prove freedom from electrical or mechanical defects and provide assurance that it meets design specifications. This test consists of the following:
>
> No-load running current and power
> Check current balance
> Measure winding resistance
> High potential test
> Vibration test per NEMA MG 1-20.53
>
> If a commercial test witness is desired, a commercial test described above must be made in all cases. A re-test is necessary when a witness test is specified. The "Complete Test" will only be taken on request. It consists of the Commercial test plus the following:
>
> Full load heat run
> Percent slip
> No-load running current
> Checking of current balance
> Pull out torque
> Locking rotor current
> Starting torque
> Efficiency at full, $^3/_4$ and $^1/_2$ load
> Power factor at full, $^3/_4$ and $^1/_2$ load
> Winding resistance measurement
> Bearing inspection
>
> Tests in the complete test will be performed uncoupled as follows:
>
> No load saturation curve measuring starting torque, current and power.

(Text continues on page 659)

Table 14-11C
Insulation and Temperature Rise for Medium and Polyphase Induction Motors

Class of insulation system (see 1.65)	A	B	F*	H*†
Time rating (shall be continuous or any short-time rating given in 10.36)				
Temperature rise (based on a maximum ambient temperature of 40°C), °C				
a. Windings, by resistance method				
1. Motors with 1.0 service factor, other than those given in items a.3 and a.4	60	80	105	125
2. All motors with 1.15 or higher service factor	70	90	115	...
3. Totally-enclosed nonventilated motors with 1.0 service factor	65	85	110	130
4. Motors with encapsulated windings and with 1.0 service factor, all enclosures	65	85	110	...
b. The temperatures attained by cores, squirrel-cage windings, and miscellaneous parts (such as brushholders, brushes, poles, tips, etc.) shall not injure the insulation or the machine in any respect.				

*Where a Class F or H insulation system is used, special consideration should be given to bearing temperatures, lubrication, etc.

†This column applies to polyphase induction motors only.

Notes
1—Abnormal deterioration of insulation may be expected if the ambient temperature of 40°C is exceeded in regular operation. See Note 3.
2—The foregoing values of temperature rise are based upon operation at altitudes of 3,300 ft (1,000 meters) or less. For temperature rises for motors intended for operation at altitudes above 3,300 ft (1,000 meters), see 14.04.
3—The temperature rises given in the preceding table are based upon a reference ambient temperature of 40°C. Motors intended for use in higher ambient temperatures should have temperature rises not exceeding the value calculated from the appropriate formula, rounded off to the nearest 5 degrees.
For motors given in items a.1 and a.2—Temperature rise = 0.9 ($T_c - T_a$)
For motors given in items a.3 and a.4—Temperature rise = 0.965 ($T_c - T_a$)

Where: T_a = ambient temperature
T_c = for items a.1 and 1.4
105°C for Class A insulation system
130°C for Class B insulation system
155°C for Class F insulation system
180°C for Class H insulation system
for item a.2
115°C for Class A insulation system
140°C for Class B insulation system
165°C for Class F insulation system
T = for item a.3
105°C for Class A insulation system
130°C for Class B insulation system
155°C for Class F insulation system
175°C for Class H insulation system

When a higher ambient temperature than 40°C is required, preferred values of ambient temperature are 50°C, 65°C, 90°C, and 115°C.

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Table 14-12
Information Required for Selecting Motors

Motor Data

General
Type of motor (cage, wound-rotor, synchronous, or dc) ....
Quantity ....hp.........rpm...........Phase...........
Cycles.......................Voltage.........................
Time rating (continuous, short-time, intermittent)..................
Overload (if any)........% for........Service factor.........%
Ambient temperature............C temperature rise.............C
Class of insulation: Armature.....Field.....Rotor of w-r motor.....
Horizontal or vertical..............Plugging duty...............
Full- or reduced-voltage or part-winding starting(ac)................
If reduced voltage—by autotransformer or reactor.................
Locked-rotor starting current limitations...........................
Special characteristics..
Induction Motors
Locked-rotor torque........%........Breakdown torque........%
or for general-purpose cage motor: NEMA Design (A, B, C, D)........

Synchronous Motors
Power factor.......Torques: Locked-rotor.......% Pull-in.......%
Pull-out........% Excitation........volts dc. Type of exciter.......
If m-g exciter set, what are motor characteristics?.................
Motor field rheostat.........Motor field discharge resistor.........
Direct-Current Motors
Shunt, stabilized shunt, compound, or series wound.................
Speed range............Nonreversing or reversing..............
Continuous or tapered-rated....................................
Mechanical Features
Protection or enclosure...............Stator shift...............
Number of bearings..............Type of bearings...............
Shaft extension: Flanged........Standard or special length.........
Press on half-coupling...............Terminal box..............
NEMA C or D flange...........Round-frame or with feet..........
Vertical: External thrust load.......lbs. Type of thrust bearing.......
Base ring type.........Sole plates..........Accessories..........

Load Data

Type of Load..
If compressor drive, give NEMA application number................
Direct-connected, geared, chain, V-belt, or flat-belt drive.............
Wk^2 (inertia) for high inertia drives..........................lb-ft²

Starting with full load, or unloaded...If unloaded by what means?....
For variable-speed or multi-speed drives, is load variable, torque, constant torque, or constant horsepower?.........................
Operating conditions...

Used by permission: *Allis Chalmers Motor and Generator Reference Book*, Bul. 51R7933. A C Compressor Corporation. E. S. Lincoln, Ed. "Electrical Reference Book," Electrical Modernization Bureau, Colorado Springs, CO.

Table 14-13
Data Required When Ordering Electric Motors

The following listing of data to be supplied to manufacturers when ordering motors or requesting bids is based on the requirements and recommendations of a number of representative manufacturers. This list is split into vital and desirable information and is intended as a checklist in preparing both requisitions and purchase orders. By using this list, you will give the manufacturer a clear picture of your needs for most motor applications in the oil and petrochemical industries and eliminate call-backs, requests for supplementary information, and so on.
Do not mix special with standard motors on a single purchase order. If changes must be made later in specials, *some* delay can be avoided.

Electrical Specifications

Vital Information (Also see references 83, 85, 86)

A. All Motors. (Power Characteristics)
1. Voltage, AC or DC.
2. Cycles per second, AC only.
3. Number of phases, AC only.

B. Induction Motors.
1. Type and horsepower. (Type often signifies motor design. Horsepower should equal the maximum anticipated requirement of the driven machine, or under upset conditions if the service is vital. Also give horsepower requirements of driven device. (If in doubt, give application details.)
2. Speed and direction of rotation.
a) Speed, rpm
1) Fixed speed. Advise synchronous speed desired (i.e., 3,600, 1,800, etc.) for AC motors. For DC motors, advise desired basic speed at full, load, *and* maximum speed by field control.
2) Multispeed (AC polyphase motors). Specify whether single or dual winding,* the two or more speeds desired, and choose:
(a) Constant horsepower.
(b) Constant torque.
(c) Variable torque.
*NOTE: Certain speed combinations require dual windings, and others may be either one or two. In the latter case, compare costs of both motors *plus* their controls.
3) Variable speed (AC wound-rotor motor).
(a) Maximum speed.
(b) Number of steps of speed control.
(c) Speed at each step (or allow motor manufacturer to select from performance data for driven machine).
(d) Performance data on driven machine (i.e., torque requirements from design data curves on pump, centrifugal compressor, blower, winch, hoist, etc.) Specifying NEMA design for motor, under torque, may be omitted if design curves are furnished.
b) Gear motor. So specify and state the output shaft speed or speeds. Also specify class of gear as follows:
1) Class I gear—not more than 8–10 hr/day: *no* shock loads.
2) Class II gear—more than 8–10 hr/day *or* shock loads.
3) Class III gear—more than 8–10 hr/day *and severe* shock loads.
c) Direction of rotation. State reference point. (Example: "clockwise facing the end opposite drive" or coupling end.) If service to be reversible via control, so state.
3. Torque. Specify NEMA design (A, B, C, or D) or manufacturer's designation if known. For special applications, advise when load peaks occur and severity, starting, pull-in, and maximum torques. For fractional hp, some manufacturers give you the choice of capacitor-split phase, reactance-split phase, or repulsion-start.
4. Locked rotor current (especially for fractional hp motors). Specify National Electrical Code rating (NFPA). (Should not exceed H, or 7.09 KVA/hp for high-quality motors. Advise if nameplate is to be code-stamped. Large motors usually are so marked.)
5. Application. (Very necessary for special motors and desirable for all others; especially large motors driving centrifugal and reciprocating pumps and compressors. Advise whether these start under load. If a replacement and previous motor have failed frequently, give all details and reconsideration of electrical specifications as in order.)

C. Synchronous Motors. These are normally used only when speed constancy is important, "power factor correction" is desired, or application is very low speed.
1. Section A and Items 1 and 2, Section B.
2. Advise of exciter power characteristics available, or specify manufacturer to provide required exciter (small synchronous motors may use permanent magnets) as: direct-connected, belt-driven, or motor-generator set.
3. Torque. Specify required starting, pull-in, pull-out, and maximum torques.
4. Complete application details and data on driven machine usually required by most manufacturers.

(continued)

Table 14-13
(Continued)

Desirable Supplementary Information, All Motors

A. All motor drives for use in process plant applications should meet the requirements of the Electrical and Electronics Engineers (IEEE) Standard 841-1994 (or latest edition), The National Electrical Manufacturers Association Standard MG-1 (latest edition) and the National Fire Protection Association applicable Standards/Codes (latest edition).

B. Protective provisions if not to be provided by usual overload device in controller. (Manufacturer may make recommendations based on application details.) Buyer may specify.

1. Special temperature-sensing windings or elements; always optional with customer. These may be thermocouples embedded in the winding, listed under various trade names, or resistance windings, operating through motor control device. Advise whether contacts are to be normally open or closed and whether to operate alarm or shutdown motor. In fractional horsepowers, these are often bimetallic strips interrupting power supply and resetting automatically or manually.

2. Heating element for moisture protection. Specify:
   a) Embedded in coil.
   b) Installed within motor enclosure.
   c) Power characteristics available or to be provided for operating a) or b).
   NOTE: Some manufacturers do not supply such provisions; see catalog or contact representative.

3. Insulation. (Normally specified for "tough" environments.)
   a) If requirement known, specify NEMA Class (A, B, and H or manufacturer's special catalog options. Some manufacturers require special application information when NEMA Class H is specified).
   b) High-temperature service. (Advise anticipated maximum ambient temperature.)
   c) Chemical resistant. (For marine and coastal applications; chemical plants; locations where abrasive dusts are common; high humidity inland locations where motor duty is cyclic indoors or outdoor. Some manufacturers supply so-called "special Class A Gulf Coast insulation," "chemical insulation," etc.; others do not.)

C. Starting provision. (Although most normal motors now are capable of across-the-line starts, some designs are not; check catalog carefully on this point and specify requirement if different.)

D. Performance characteristics. (Usually includes following electrical characteristics: efficiency, power-factor, locked rotor KVA, full-load speed, temperature rise and starting, break-down, and full-load torques.) Be sure to advise whether special test certification is to be furnished; this *always* extends delivery time and increases cost.

E. Replacement motors. (Advisable to state why original motor failed, especially if an electrical failure. Given details for manufacturer's study and recommendation.)

F. Installation: Motors may be installed into process plants in several ways:

1. For primary process equipment drives; For this situation the company project or process engineer should establish the key process-related requirements, including the safety environment such as explosive vapors, liquids or dusts, etc. See standards this chapter. Then depending on the company organization, the mechanical or electrical engineer should become involved with the motor specifications, motor testing before delivery, and the overall requirements of the motor to be suited for the application, and company and/or NEMA specifications should become involved in ordering the motor.

2. For "package" process system drivers; Unless the company (termed "owner") takes a firm hand in specifying the motors—even small HP or fractional HP—the motors furnished may often be (a) cheap construction, (b) minimum HP, (c) not suited for the process environment of the plant (see NFPA Codes) from a safety point of view, such as drip-proof guarded, totally enclosed fan cooled, or explosion proof motor, or others and may have to be replaced before the plant can safely start up. Here again the process, mechanical and electrical engineers may need to become involved in firming up the small, medium or large motor specifications for the "package." Some "packages" such as process refrigeration systems may involve large motors driving centrifugal compressors, and these cannot be overlooked. Sometimes it is wise to specify the "make{ of the motor for consistency of parts and maintenance.

Mechanical Specifications

Vital Information.

A. Enclosure.

1. Hazard. Specify explosion-proof and always class and group (i.e., Class 1 Group D for most petroleum refinery applications).

2. Protective. Options are open (no protection), open drip-proof, splash-proof, totally-enclosed, forced-ventilated, self-ventilated, and sanitary.

3. Advise whether corrosive conditions preclude use of standard materials in motor construction.

B. Bearings. See catalog only as to options and manufacturer's standard as to grease and oil-lubricated ball and sleeve bearings, also force-feed. *Do not specify* size and make of bearing or factory-filled lubricant.

C. Mounting.

1. Standard. Advise mounting position: vertical, horizontal, wall, etc. (Some motors cannot be mounted other than horizontal because of bearing and/or lubrication arrangements without special provisions. Others are universal mounted, and still others can be suitably modified in the field.)

2. Flange. Advise type; see catalog. Specify horizontal or vertical position.
3. Vertical hollow-shaft. (Usually a pump application.) Some manufacturers prefer these to be ordered through manufacturer of driven machine due to complex requirements.
4. Special. Advise requirements in detail; manufacturer may have an available standard ordinarily unlisted.

D. Shaft Connection. (See catalog.)
1. Specify type: direct (specify type and make of coupling, keyed or screwed, and whether or not manufacturer is to supply), and single or double-ended; spiral threads, supplying details; V-belt or flat-belt sheave.
2. Shaft extension. Specify only for special applications; note catalog dimension data before ordering.
3. Special connections. Consult manufacturer.

E. Thrust and unusual bearing loading.
1. End-thrust. Advise magnitude in both directions, such as in centrifugal pump applications, especially when motor is vertically mounted. Application details, when furnished, usually give a clue to whether or not these data are required.
2. Lateral thrust. Advise only in cases of unusual bearing loads.

F. Duty.
1. Continuous or intermittent operation.
2. If intermittent duty, is it occasionally overloaded?

G. Environment.
1. Location, inside or out-of-doors; Is location damp or humid? Is water likely to drip or splash on motor?
2. Maximum air temperature (advise whether sun shines on motor). See reference 60 for comments on motor temperature rise.
3. Presence of corrosive fumes or liquids; specify nature.
4. Dusts; present or absent. If present, state nature—corrosive, abrasive, or flammable.

Desirable Supplementary Information

A. Breathers and drains.
1. Specify when option given in catalog (some manufacturers do not supply these).
2. Standard or explosion-proof.
3. Material of construction (under corrosive conditions).

B. Frame size. (Usually applicable only when ordering replacement motors or for special applications. Size usually must be at least as large as manufacturer's standard for motor size and speed.)
NOTE: Beginning in 1954, NEMA Standard frame size will be one number smaller for motor of 1–3 hp than previously, due to improvements in insulation requiring less space for wiring. It, therefore, will be advisable in ordering replacements in this range to state whether or not driven equipment was sized to take the former standard frame for this horsepower.

C. Nameplate material. (Specify stainless steel when standard brass will not resist anticipated corrosion.)

D. Fan material. (Normally may be left to manufacturer if environment—Item 7—details are supplied. Otherwise, advise brass, high silica, bronze, aluminum, "textolite" or micarta, etc.)

E. Special frame or mounting construction. (Consult manufacturer.)

F. Nameplate data. Advise whether Locked Rotor KVA and Underwriter Code stamps required.

G. Certified drawings. Advise whether required and to whom supplied.

H. Special shipping instructions.

I. Space Heaters. A resistive heating device installed in the motor to prevent condensation when the motor is not operating. This requires a special control system.

J. Pressure ventilated large induction or synchronous motors: A Pressure Ventilated motor requires a closed pressure system to force filtered air or nitrogen into the motor casing (housing) to avoid corrosive or explosive conditions internally. Constance reference 56 describes some of the details of such a system; also see Ecker et al. [69].

Ordering Replacement Parts

A. Nameplate data. Give *all* nameplate data, regardless of how inconsequential it may seem; the manufacturer is virtually helpless without it.

B. Use name, number, or other designation in parts list, as well as naming part.

C. Bearings. Most bearings are standard as to dimensions and may be ordered from the bearing manufacturer. (Note: Within the last few years motor bearing ratings have been changed, generally toward reduced rated loads.)

Used by permission: Thornton, D. P., Jr. *Petroleum Processing*, p. 1321, Sept. 1953, and also adapted and modified by this author.

(Text continued from page 655)

The heat run will be equivalent load method.

Efficiency at full, $^3/_4$ and $^1/_2$ load and power factor at full, $^3/_4$ and $^1/_2$ load and breakdown torque will be determined by equivalent circuit calculation (IEEE 112, Method F) using calculated values for slip and stray current losses.

Adjustable Speed Drives

This is an important and useful motor drive arrangement for many process and nonprocess applications.[73, 74, 93] For a variable-frequency controller, see reference 94.

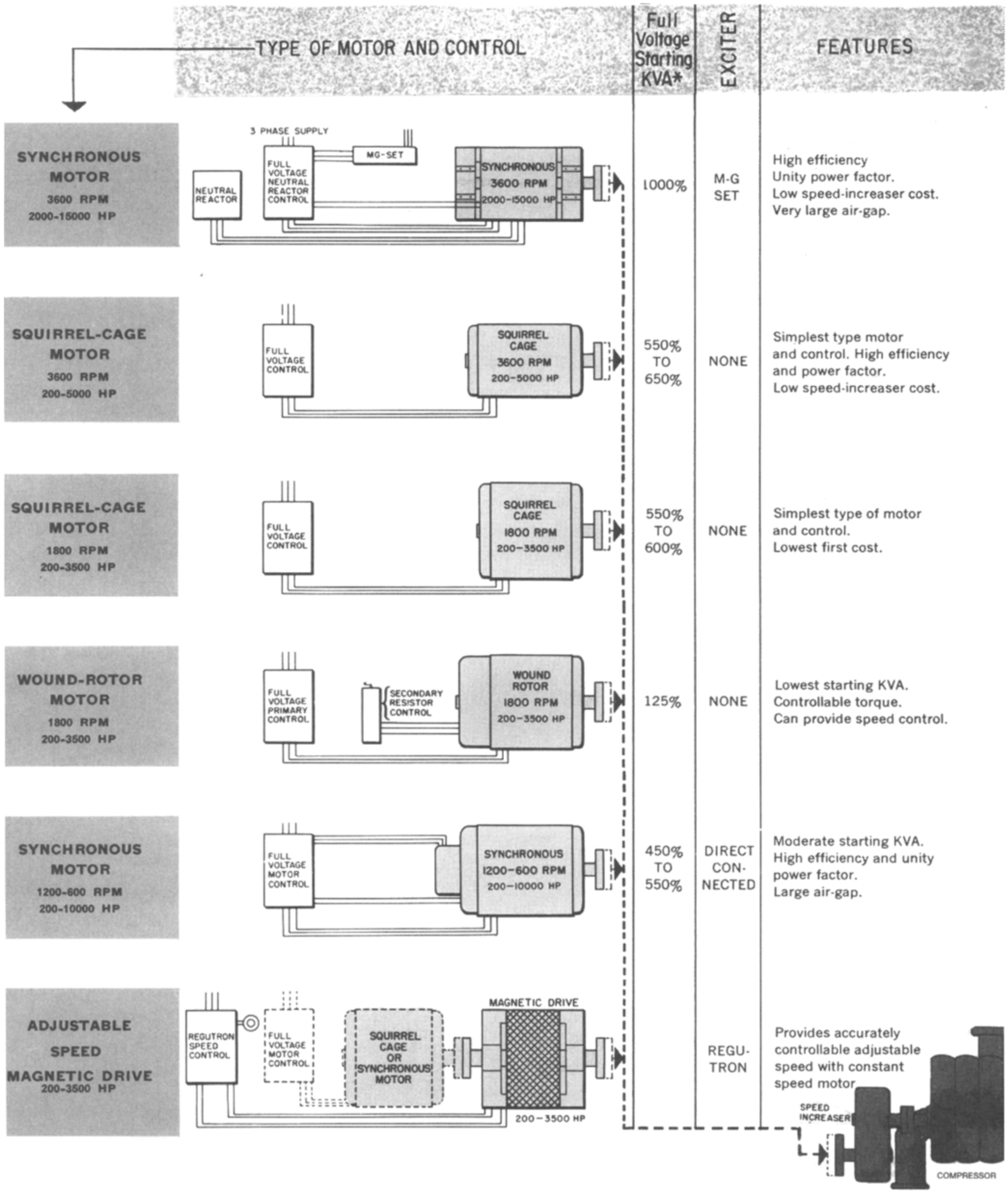


Figure 14-14. Selection chart for centrifugal compressor drive. (Used by permission: *E-M Synchronizer*, 200-SYN-52, ©1958. Dresser-Rand Company.)

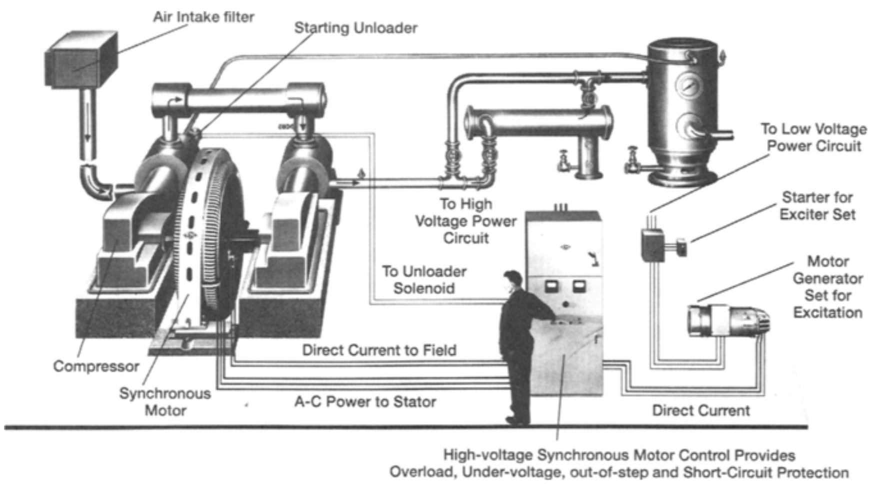


Figure 14-15. Synchronous motor drive for reciprocating compressor. (Used by permission: *E-M Synchronizer*, 200-SYN-52, ©1958. Dresser-Rand Company.)

Table 14-14
Useful Electrical Formulas

$$\text{hp} = \frac{\text{Torque (in ft-lb)} \times \text{rpm}}{5{,}250}$$

$$\text{hp} = \text{kw}/0.746$$

$$\text{Motor synchronous rpm}^{*} = \frac{120 \times \text{Hz}}{\text{number of poles}}$$

* Synchronous rpm can be for an induction motor

$$\text{Full-load amps}^{*} = \frac{\text{hp} \times 0.746}{1.73 \times \text{kv} \times \text{Eff'cy} \times \text{power factor}}$$

*Three-phase

$$\text{rpm} = \frac{120 \times \text{frequency in Hz}}{\text{no. of poles}}$$

$$\text{rpm (for 60 Hz)} = 7{,}200/\text{no. poles}$$

$$\text{rpm (for 50 Hz)} = 6{,}000/\text{no. poles}$$

To find: ac three-phase

$$\text{Amperes when hp is known} = \frac{\text{hp} \times 746}{1.73 \times E \times \text{Eff} \times PF}$$

$$\text{Amperes when kilowatts is known} = \frac{KW \times 1{,}000}{1.73 \times E \times PF}$$

$$\text{Amperes when kva is known} = \frac{\text{kva} \times 1{,}000}{1.73 \times E}$$

$$\text{Kilowatts} = \frac{1.72 \times I \times E \times PF}{1{,}000}$$

$$\text{kva} = \frac{1.73 \times I \times E}{1{,}000}$$

$$\text{hp (output)} = \frac{1.73 \times I \times E \times EFF \times PF}{746}$$

I = amperes; E = volts; Eff. = efficiency;
PF = power factor; kva = kilovolt-amperes;
kw = kilowatts.

A = amperes

Used by permission: *Motor Reference Handbook.* TECO-Westinghouse Motor Co.

Table 14-14 provides useful general formulas (for estimating purposes) and is used by permission of TECO-Westinghouse Motor Co.

Mechanical Drive Steam Turbines

The mechanical drive steam turbine uses expanding steam to drive a rotating wheel with a power shaft. See Figures 16-16A–H.

This type of turbine is used for driving various mechanical rotating equipment through a direct-connected coupling to the driven equipment or through a speed increaser or decreaser between the turbine and the driven equipment (such as, centrifugal compressor, high speed pump, blower, fan, or other rotating equipment).

The evaluation of steam turbine drives versus electric motor selections for an application is reviewed by Ranade, et al.[101]

This type of turbine is similar in all respects to the steam power-generator turbine but is considerably simpler, not having as many wheels, nor the same power plant economics to consider. For a general description, see Table 14-15. Neerken[30] is an excellent reference to steam turbine

Table 14-15
General Description for Steam Turbines

Component	Use	Materials of Construction
Casing (horizontally split, vertical split available)	Bolted case and cover	Cast iron, cast steel
Rotor	Holds blades or buckets	Steel, usually forging shrunk or keyed to shaft
Blades or buckets	Handles steam expansion	Stainless steel, chrome alloy
Governor centrifugal-weight (fly-balls) or other designs	Automatically controls speed	Steel, stainless and/or monel
Overspeed trip	Releases trip valve when speed exceeds 115% of maximum rating	Stainless steel
Trip valve	Shuts off steam on overspeed	Steel, stainless, and nickel iron
Nozzle block and nozzles	Establish steam jets	High tensile carbon silicon steel
Shaft	Carries rotor and transmits power	Heat treated alloy steel forging
Shaft packing	To keep steam leakage to a minimum where the shaft extends through the casing	Spring-backed carbon rings

performance analysis; see also references 7, 97, 18, 99, 100, 89, and 56.

The two basic types of turbine are illustrated in Figures 14-16A, 14-16B, and 14-16C and their thermal performance is described by Reese and Carlson[97] using the following terms:

1. Reaction
2. Impulse

Most steam turbines operate in the condensing (of steam on exhaust from turbine) mode or noncondensing or back-pressure mode. (Steam is exhausted or extracted from the turbine at preselected exhaust pressure for other uses.) See Figures 14-17A–C, 14-18A, 14-18B, 14-19A, 14-19B, 14-20A, and 14-20B.

The majority of designs in process plants use the impulse type (Figure 14-20A) with a single wheel with one or two velocity stages. The multistage unit for larger applications is shown in Figure 14-20B.

Standard Size Turbines

No. Stages	HorsePower Range	Lb Steam Required/hr/bhp
Small 1	0.7–3,300	575–45
Medium 1 or 2	5–5,500	50–15
Large 2+	500–70,000	25–5

Speed Range

< 1,000 rpm to 30,000 rpm
Usual: 2,000 rpm to 15,000 rpm

Efficiency Range

Single stage = 30%
Multi-stage = 60–80%

Motive Steam

Inlet pressure:	Essentially atmospheric up to nearly critical
Discharge pressure:	Vacuums to 29.5 in. Hg (30-in. bar.) up to several hundred pounds gage
Inlet temperatures:	Saturation corresponding to inlet steam up to 1,000–1,050°F total temperature

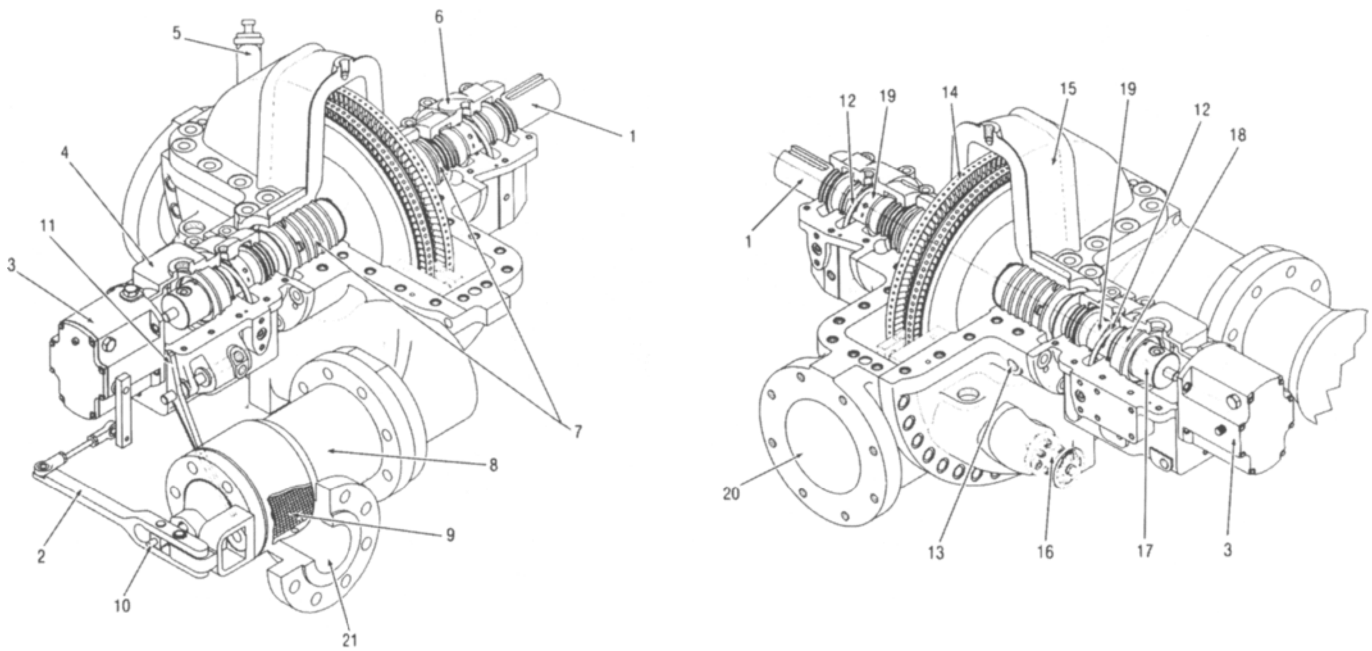
Applications

Mechanical drive turbines, although designed for steam, can be adapted and modified to operate with high pressure gas as the motive power. This is particularly profitable where the gas is needed at reduced pressure. The driven applications can be the same as for the turbine when steam is the motive medium.

The steam turbine is operated as a condensing unit when the exhaust steam is condensed as it leaves the turbine. This is usually below atmospheric pressure and often advisable in order to (1) maintain proper plant-wide steam and condensate balance and (2) use all reasonable energy in steam, Figure 14-18A.

The steam turbine is operated noncondensing when the exhaust steam is not condensed but passes into a low-pressure distribution system for additional use and heat recovery, Figure 14-18B.

The turbine applied to driving mechanical equipment is not operated (very often) with extraction or bleed streams. Here again, this depends upon the plant steam balance, and this is one of the fine features of steam turbine drive. The flexibility of design and application allow it to be set in the proper place for the economic balance of a system. The decision as to condensing or noncondensing should not be



LEGEND

1.	Turbine Shaft	12.	Oil Rings (2)
2.	Governor Lever	13.	Packing Case Leakoffs (2)
3.	Woodward TG Governor	14.	Turbine Wheels
4.	Steam End Bearing Case	15.	Turbine Case
5.	Sentinel Warning Valve	16.	Hand Valve
6.	Exhaust End Bearing Case	17.	Overspeed Cup
7.	Carbon Packing Rings	18.	Thrust Bearing
8.	Steam Chest	19.	Main Bearings (2)
9.	Steam Strainer	20.	Exhaust
10.	Governor Valve Stem	21.	Inlet
11.	Trip Lever		

Figure 14-16A. Single-Stage Turbine with Woodward Speed Governor Arrangements. (Used by permission: Instruction Manual 102-P, ©1994 Dresser-Rand Company.)

arbitrary, because the turbine can become very uneconomical if it does not fit the system. Figures 14-19A and 14-19B illustrate a few combinations relative to the steam.

Major Variables Affecting Turbine Selection and Operation

The key variables to consider are used by permission of Althearn[95].

1. Horsepower and speed of driven machine.
2. Steam pressure and temperature available or to be decided.
3. Steam needed for process (if sufficient, consider a back-pressure turbine).
4. Steam cost and value of turbine efficiency. Should it be single-stage or multistage? Should it be single-valve or multivalve? Is steam an inexpensive process by-product, or is the entire cost of generating the steam chargeable to the driver?
5. Should extraction for feed-water heating or multi-process pressure levels be considered?
6. Should condensing turbine with extraction for process be considered?
7. Control systems, speed control, pressure control, process control: If remote control, will it be pneumatic or electric? And, what speed or pressure variation can be tolerated, and how fast must the system respond?
8. Safety features such as overspeed trip, low oil pressure trip, remote solenoid trip, vibration monitor, or other

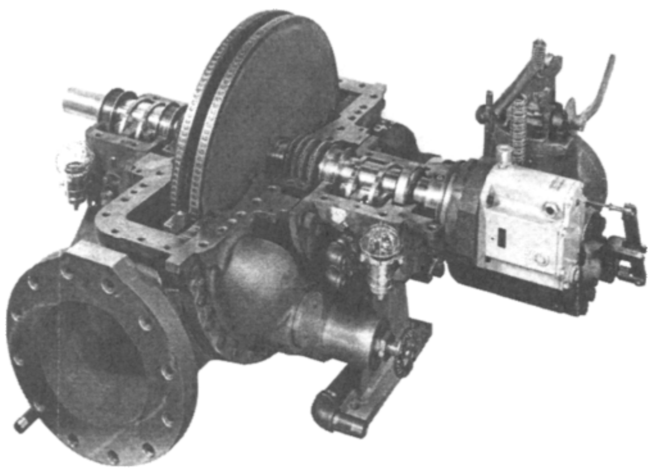


Figure 14-16B. View of Elliott YR Single Stage Steam Turbine, with top cover removed. (Used by permission: ©Elliott Co.)

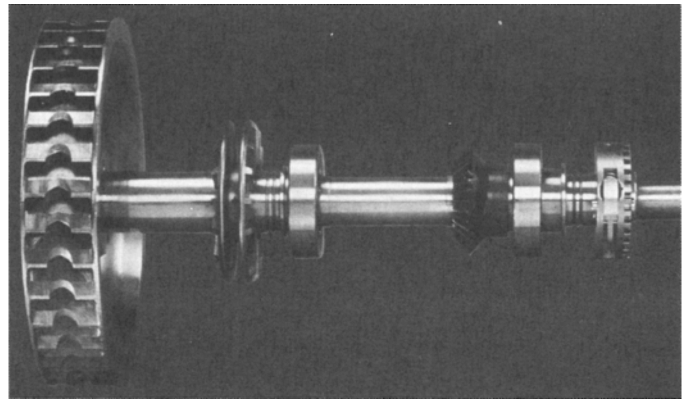


Figure 14-16C. Single Stage Steam Turbine Rotor: The GLT turbine features an overhung solid wheel that eliminates bearing box alignment problems and costs associated with installing steam separators or moisture traps commonly required for quick start applications. This overhung arrangement requires only one packing box, reducing the gland maintenance costs and potential leakage by half. (Used by permission: Bul. 8902-GLT.Dresser-Rand Company.)

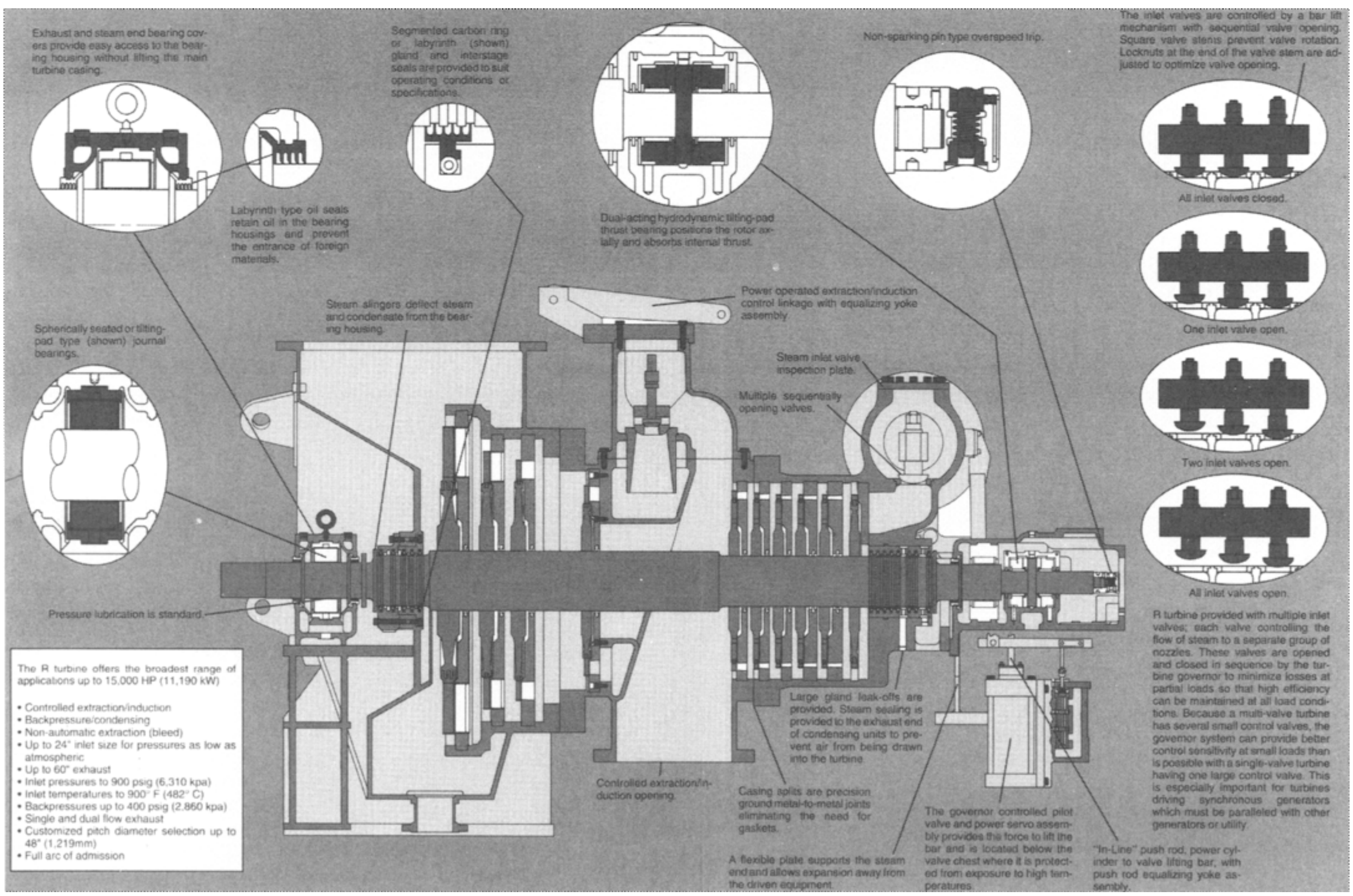


Figure 14-16D. Murray "R" Turbine; multistage, multivalve steam turbine, hp range to 15,000; speed 1,000 to 15,000 rpm; max. inlet steam 900 psig and 900°F; max. exhaust 400 psig; no. stages: 1 to 15. (Used by permission: Bul. TT 22 ©1991. Murray Turbomachinery Div., Tuthill Corporation.)

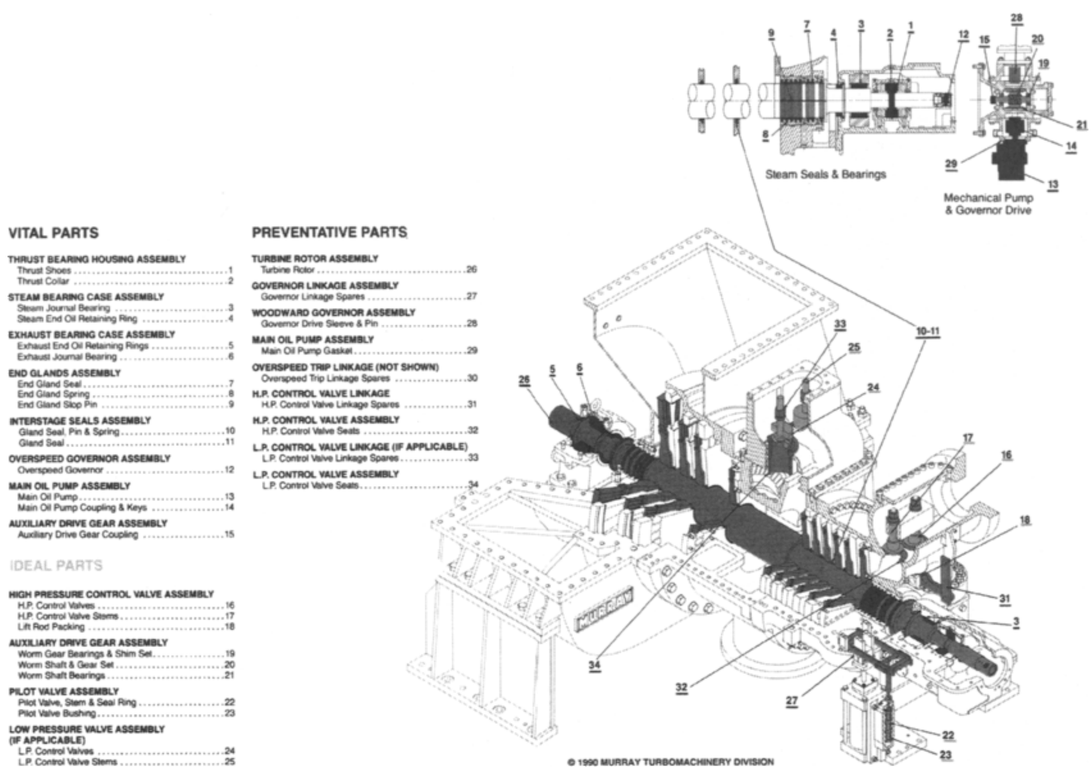


Figure 14-16E. Section view multistage, multivalve steam turbine, same as Figure 14-16D. (Used by permission: Bul. VIP 901. ©Murray Turbomachinery Div., Tuthill Corporation.)

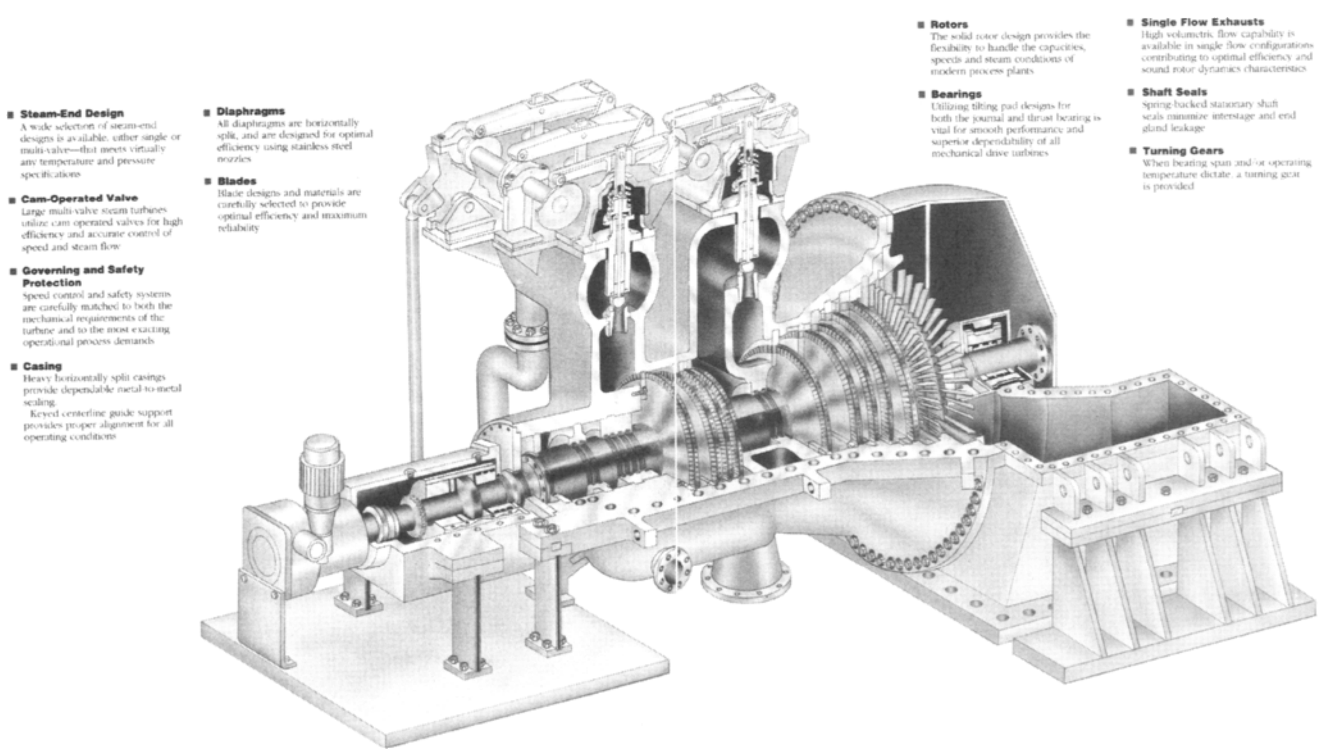


Figure 14-16F. Cutaway of large steam turbine, multistaged, multivalve, for driving mechanical rotating equipment. Connection to mechanical driven equipment shaft shown as a flange joint on right end of turbine shaft. Exhaust steam is at lower right, inlet steam is at bottom center near smaller wheels. (Used by permission: Bul. 8908-E0MD. Dresser-Rand Company.)

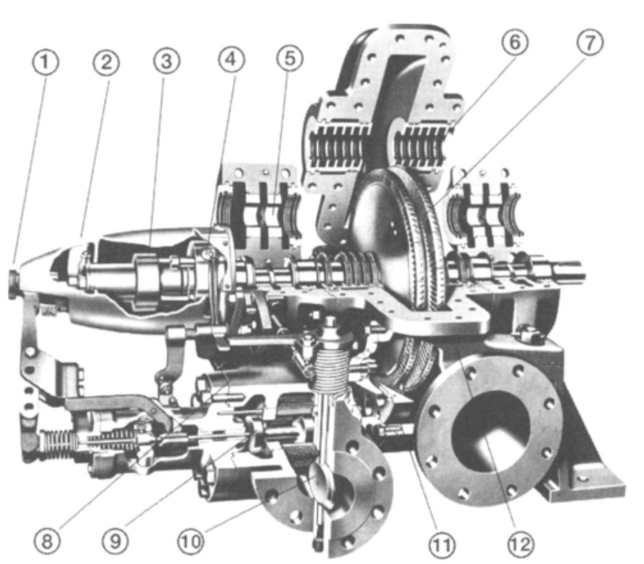
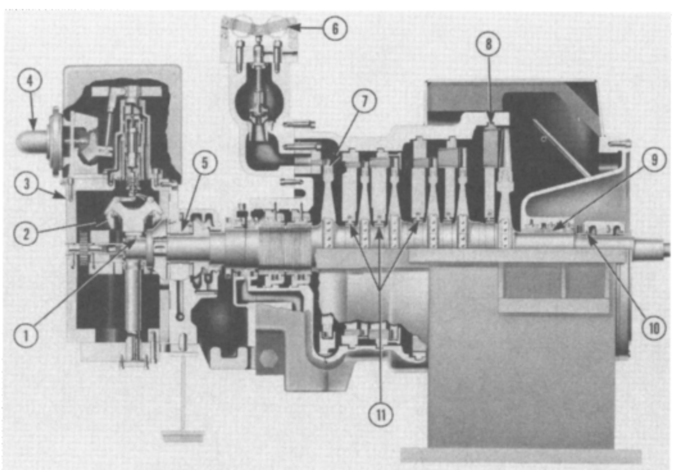


Figure 14-16G. Impulse type single stage steam turbine with shaft type governor. (Used by permission: Westinghouse Electric Corp., Steam Div.)



1. Emergency trip mechanism
2. Speed governor
3. Front standard
4. Diaphragm-type air motor
5. Ball-seat bearings
6. Torque-tube bar-lift valve gear
7. External dovetail-type buckets
8. All-welded diaphragms
9. Metallic labyrinth packing
10. Balancing ring and oil deflector
11. Metallic interstage packing

Figure 14-16H. Impulse type multistage, multivane high speed steam turbine. (Used by permission: General Electric Co.)

special monitoring of temperature, temperature changes, casing, and rotor expansion.

9. The price range from the minimum single-stage turbine to the most efficient multistage is quite wide.
10. Reliable operation.*
11. Low maintenance.*
    (* added this author)

Example 14-1

Selection

This procedure is used from Althearn with permission.[95]

1. Estimate steam flow. Refer to the Mollier diagram for steam in Figure 14-21, which shows the available enthalpy in Btu/lb of steam.

   A = Single-stage turbine
   B = Five-stage turbine
   C = Seven-stage turbine
   D = Nine-stage turbine
   Blank = Isentropic expansion

2. Read the inlet drive or throttle steam pressure and temperature on the diagram and note the enthalpy value. For example:

For 750°F and 600 psig (615 psia), read enthalpy = 1,375 Btu/lb
If the isentropic discharge or exhaust steam is to be at 50 psig (65 psia), read enthalpy = 1,160 Btu/lb
Then Btu/lb available = 1,375 – 1,160 = 215.
For a single-stage turbine with an efficiency of 30%:
Effective Btu/lb steam = 215 (30/100) = 64.5 Btu/lb
If the power required is 1,000 hp, it will require:
lb/hr steam = (2,547 Btu/hp hr) (1,000 hp/64.5 Btu/lb)
= 39,488 lb/hr. steam

For a stage with a heat drop of about 35 Btu, the steam velocity through the nozzle of the turbine will be about 1,300 ft/sec.[95] For an efficient stage, with a wheel speed of about 600 ft/sec, the nozzle will be between 3,500 and 4,000 rpm.

Theoretical steam rate = TSR = 3,413/Δh = lb/kwh
For the preceding example, TSR = 3,413/215 = 15.87 lb/kwh

The TSR can be obtained from published manufacturers' tables. For estimating condensing steam rates for single-stage turbines, refer to Figures 14-21 and 14-22.

First, for Figure 14-22, enter at the top at rpm and move to the first estimating turbine wheel diameter; then read down to the TSR (calculated or from tables) at lb/kw-hr; read across to base steam rate in lb/hp-hr. Note that the base steam rate is per hp-hr, and the TSR is per kwh (or kw-hr). Now correct the base steam rate for the horsepower loss (i.e., the portion of blades of turbine spinning outside the nozzle arc, creating friction and windage).[95] From Figure 14-23, at the top read rpm at the exhaust pressure on curved lines noted "cond," read down to the estimated wheel diameter, and read the horsepower loss on the left vertical axis.

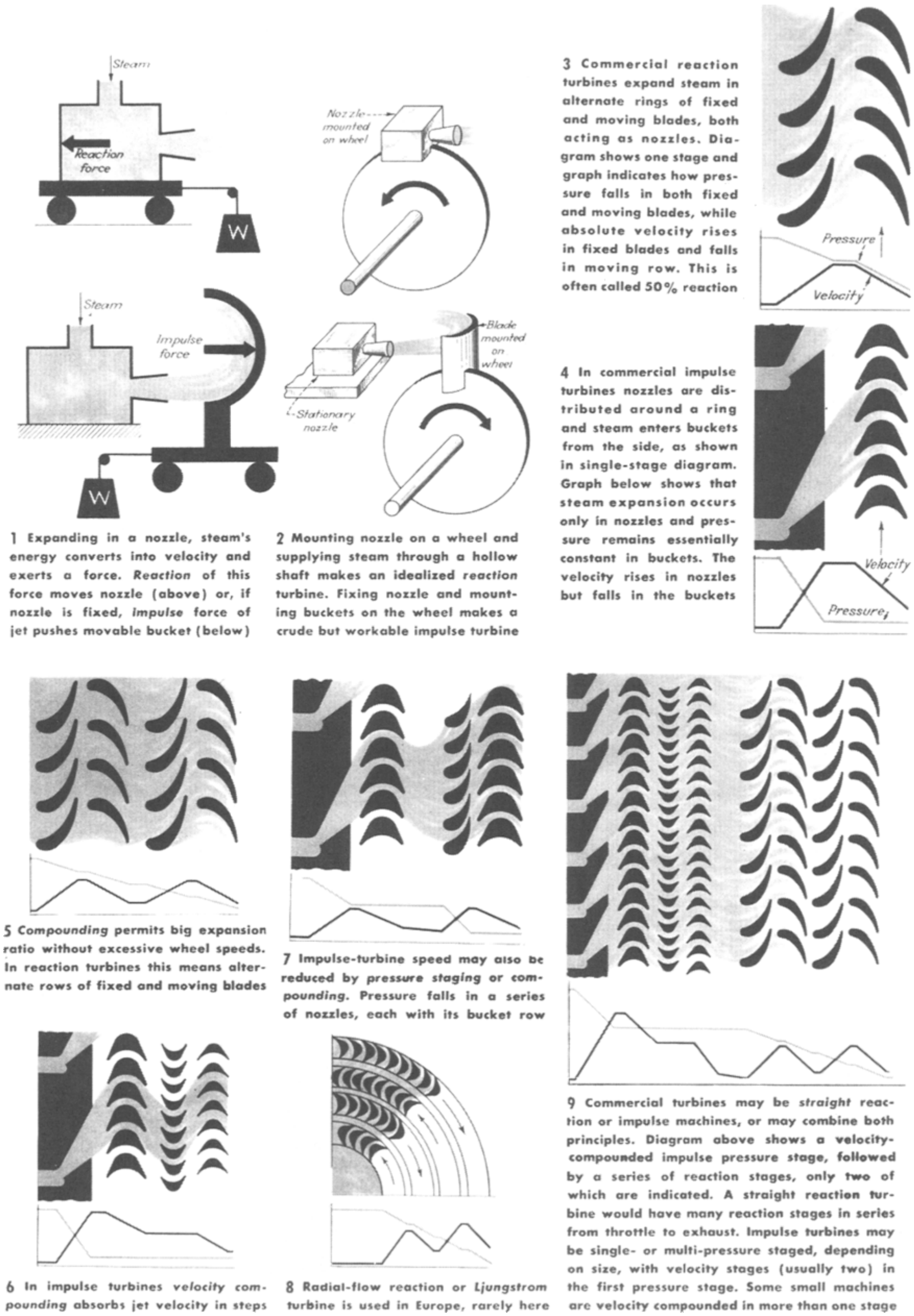


Figure 14-17A. Basic reaction and impulse turbine principles. (Used by permission: Rowley, L. N., B. G. A. Skrotzki and W. A. Vopat. *Power*, Dec. 1945. ©McGraw-Hill, Inc. All rights reserved.)

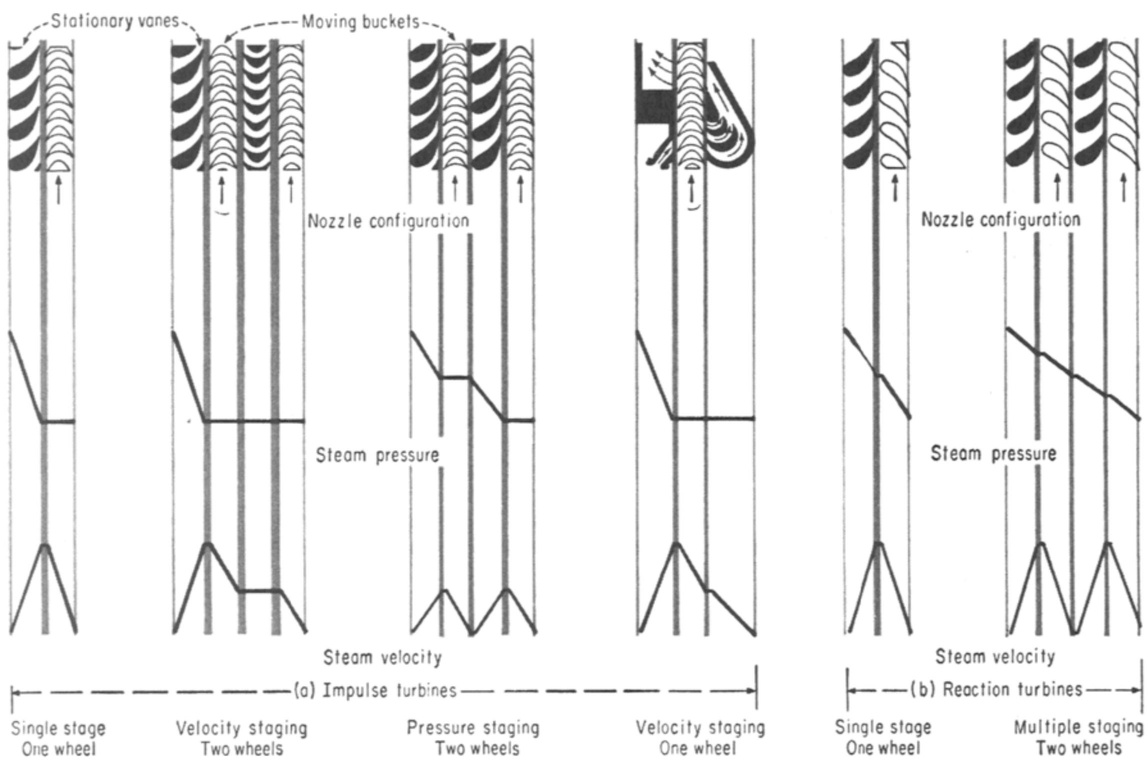
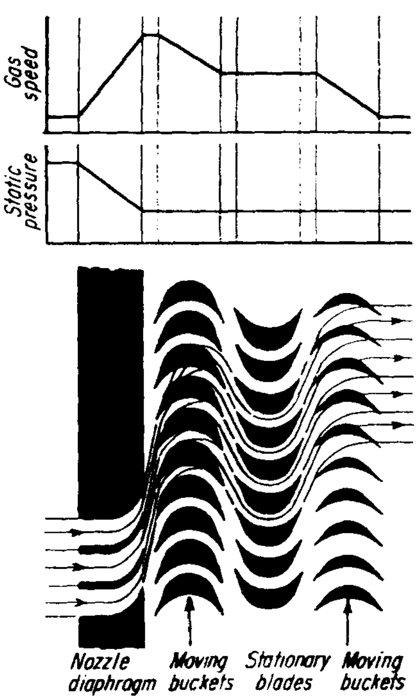
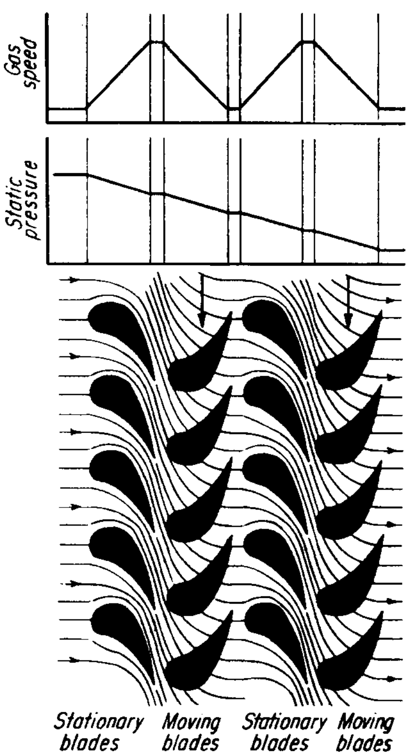


Figure 14-17B. Wheel arrangements and performance of (a) impulse turbines and (b) reaction turbines. (Used by permission: Breseler, S. A. *Chemical Engineering*, p. 124, May 23, 1966. ©McGraw-Hill, Inc. All rights reserved.)



Static pressure stays constant in the three blading rows of compounded stage



Gas pressure drops in both moving and stationary blading of reaction turbine

Figure 14-17C. Steam vapor flow through impulse and reaction turbines. (Used by permission: Skrotzki, B. G. A. *Power*, p. 170, Sept. 1959. ©McGraw-Hill, Inc. All rights reserved.)

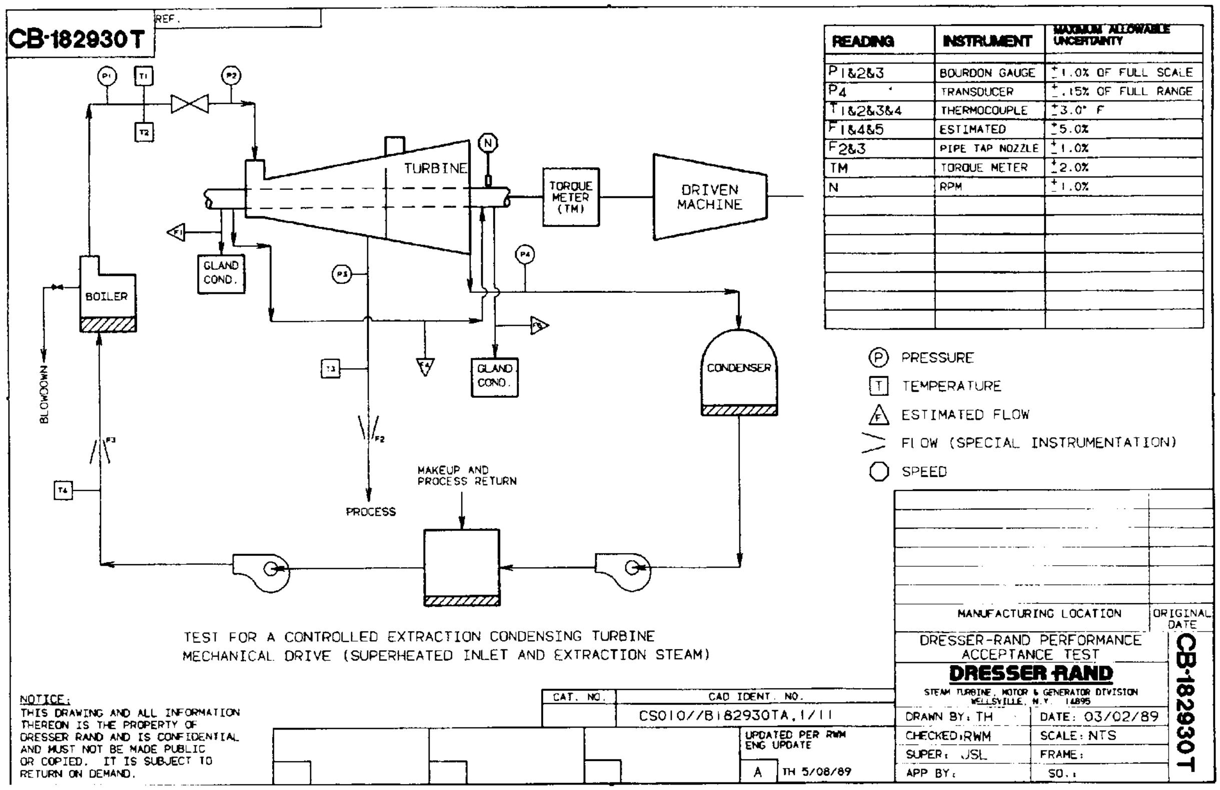


Figure 14-18A. Condensing, extraction mechanical drive steam turbine. (Used by permission: Lamberson, J. and Moll, R. "Technology Report ST 18.6." ©Dresser-Rand Company.

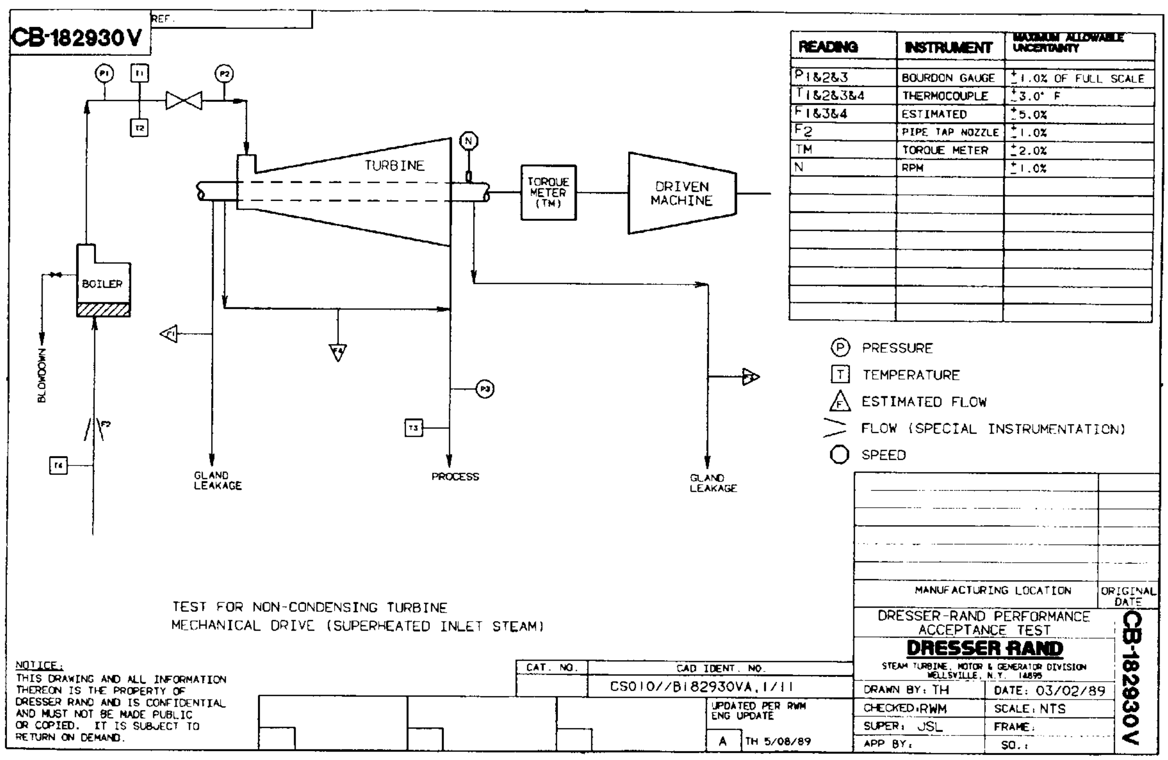


Figure 14-18B. Noncondensing steam turbine mechanical drive. (Used by permission: Lamberson, J. and Moll, R. "Technology Report ST 18.1." ©Dresser-Rand Company.)

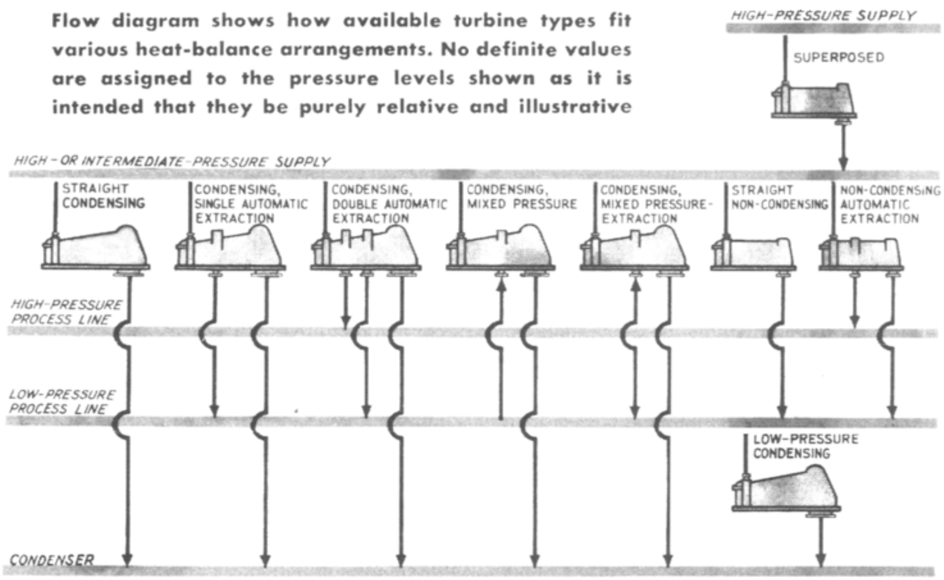


Figure 14-19A. Heat balance arrangements for steam turbines. (Used by permission: Rowley, L. N., B. G. A. Skrotzki and W. A. Vopat. *Power*, Dec. 1945. ©McGraw-Hill, Inc. All rights reserved.)

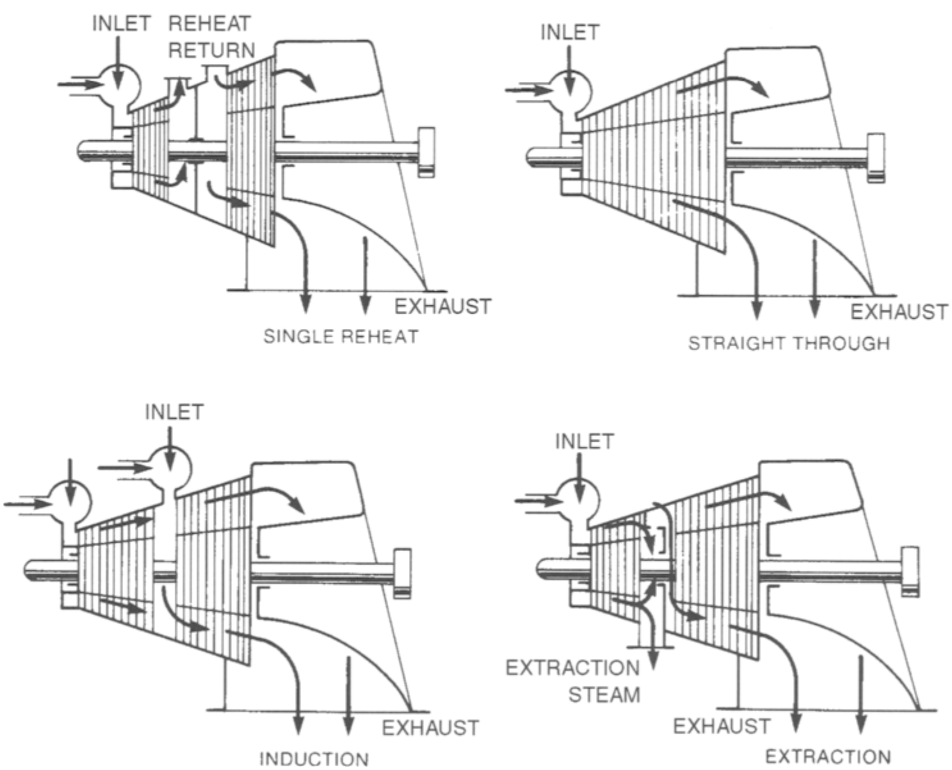


Figure 14-19B. Schematic flows of steam through mechanical drive steam turbines of various types, primarily but not exclusively for power-generation applications. This is quite typical of industrial plants that generate their own power internally. (Used by permission: "Turbines and Diesels, A Century of Power Progress," *Power*, p. 339, April 1982. ©McGraw-Hill, Inc. All rights reserved.)

Other turbine wheel diameters can be explored using the same approach.[95] The horsepower limit is established by reading across from the speed rpm and intersection of the the trial wheel diameter and by reading down to interpolate the horsepower limit. If the wheel diameter selected at your rpm is lower or greater than the required horsepower for the application, another speed and/or wheel diameter must be selected. After the horsepower loss is established, Figure 14-23, this loss must be added to the total hp design requirement for the application.

The turbine has the features of (1) variable speed operation, (2) ease of controls, (3) nonsparking operation for hazardous environment, (4) enclosed operation suitable for corrosive atmospheres, (5) operation without electrical power, (6) small space requirements per horsepower, (7) steam use under almost any condition to suit plant heat bal-

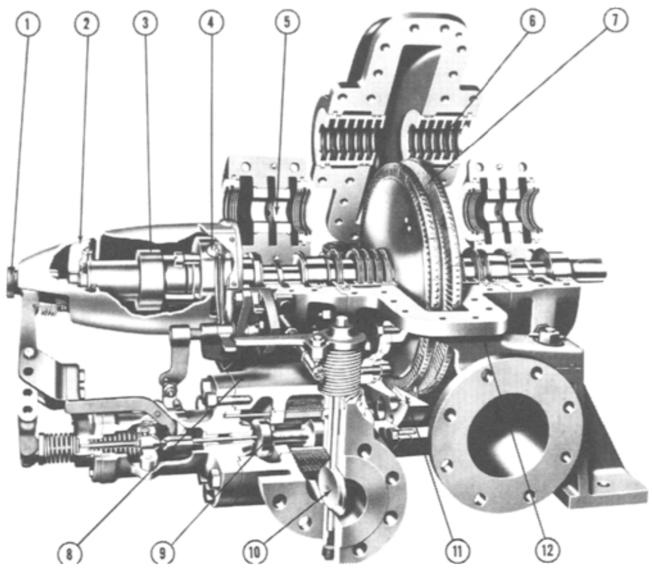


Figure 14-20A. Impulse type single-stage steam turbine with shaft-type governor. (Used by permission: Westinghouse Electric Corp., Steam Div.)

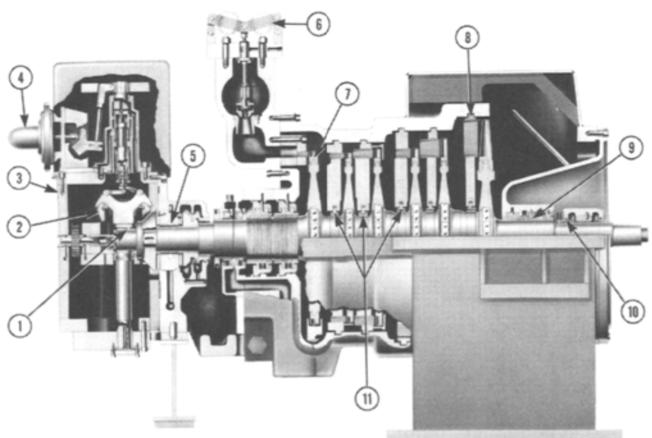


Figure 14-20B. Impulse type multistage, multivalve, high-speed steam turbine. (Used by permission: General Electric Co.)

ance, (8) particularly adaptable for use with high speed equipment by direct correction, (9) low first cost, (10) reliable operation, and (11) very low maintenance.

Operation and Control

A governor valve controls the flow of steam to the turbine.[96, 98] This valve is actuated by the governor, which is operated by the speed of the machine. When the speed exceeds the set value of the governor, a trip-valve is actuated to completely shut off the steam supply. The trip valve may

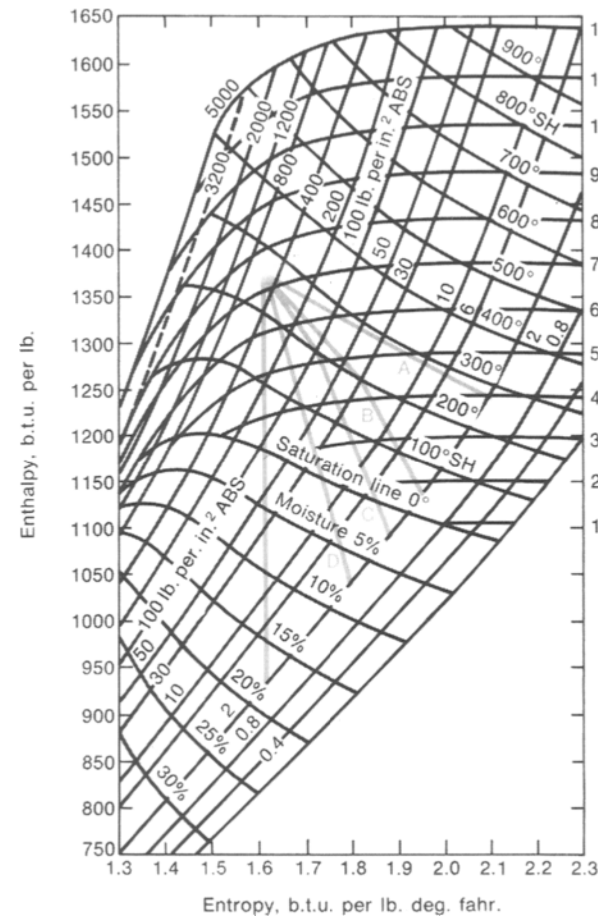


Figure 14-21. Mollier diagram with expansion lines drawn for turbines with different numbers of stages. Note right vertical axis. Temperatures correspond to heavy Mollier lines and not the uniform graph paper lines. (Used by permission: Althearn, F. H. *Hydrocarbon Processing,* p. 83, Aug. 1979. ©Gulf Publishing Co. All rights reserved.)

be a separate valve, or it may be in combination with the governor or throttle valve.

The steam enters the nozzles of the turbine and expands to the buckets or blades. Hand-operated or automatic valves are usually available to control the flow to groups of these nozzles. This allows more efficient operation at reduced loads.

When the steam leaves a condensing turbine, it passes to a surface-type condenser for recovery of the condensate. Vacuum equipment (jets or pumps) are necessary to achieve high vacuums on the condenser.

Turbines can be controlled in a number of ways, depending upon the sensitivity and reliability required. [13, 14, 18, 19, 36]

The lubricating oil system for a turbine is very important and is nearly always provided with a dual pumping arrangement.[33] One pump can be driven directly off the turbine shaft and the other by separate electric motor or steam turbine. In another arrangement, one pump can be separately electric driven and the other separately steam turbine driven. Twin coolers are often provided in the dual system to

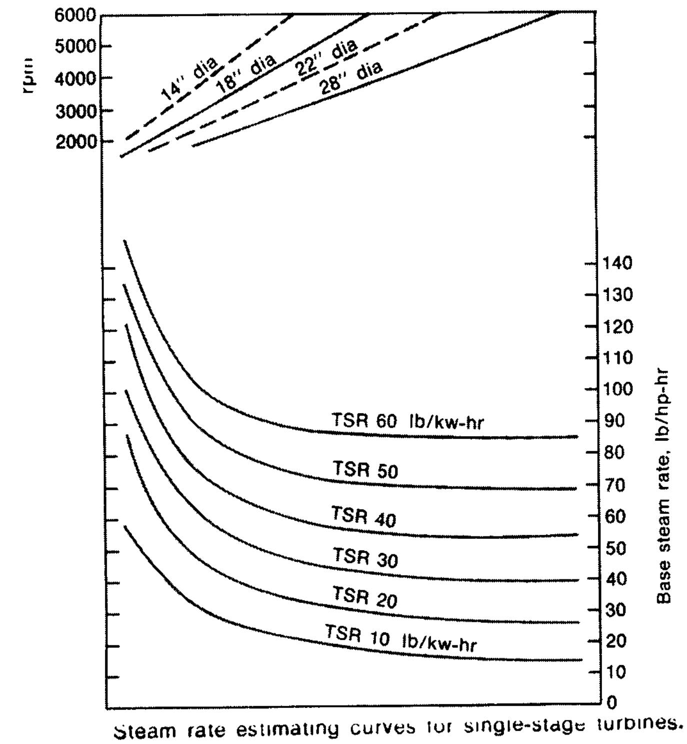


Figure 14-22. Steam rate estimating curves for single-stage turbines. (Used by permission: Althearn, F. H. *Hydrocarbon Processing,* p. 87, Aug. 1979. ©Gulf Publishing Co. All rights reserved.)

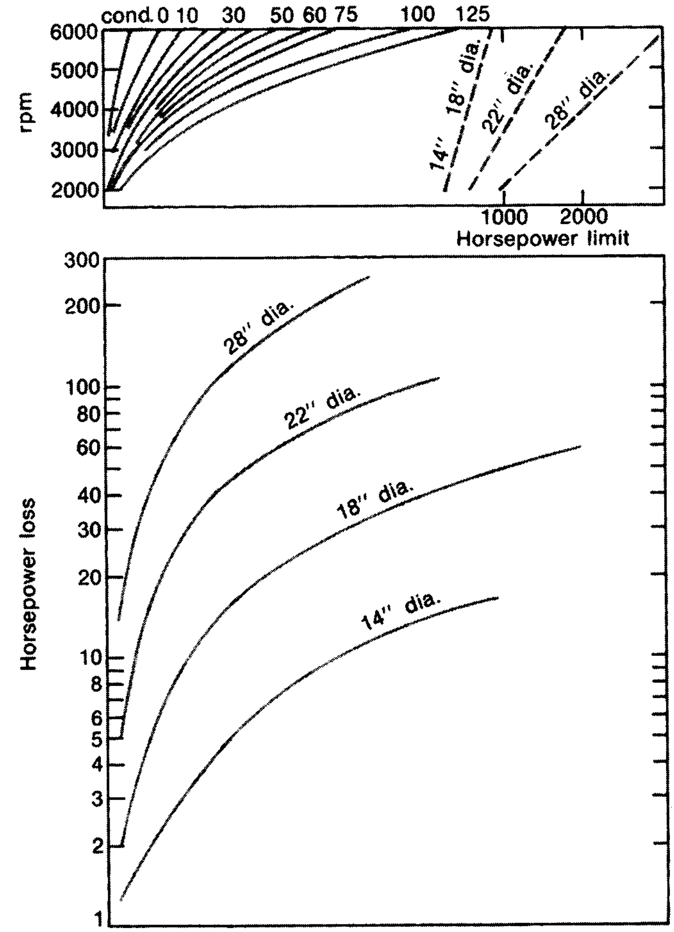


Figure 14-23. Horsepower loss in single-stage turbines can be estimated from these curves. (Used by permission: Althearn, F. H. *Hydrocarbon Processing,* p. 87, Aug. 1979. ©Gulf Publishing Co. All rights reserved.)

allow for continuous operation. The oil is used to lubricate the moving parts, to carry heat from hot turbine parts, and to operate steam control valves in the governor system.

A safety valve is usually needed on the steam exhaust side of the turbine to protect against high pressure on shut down. Most turbine case designs will not safely handle inlet steam pressure on the exhaust side, as the case is not designed to withstand intake pressure throughout. These valves are normally rated for 110% of the design steam rate.

It is important to seal the glands on the turbine shaft, and a typical arrangement is illustrated in Figure 14-24. Thrust bearing failures can be serious problems for steam turbines and other mechanical drives, as well as the bearings used on the driven equipment.

Specifications

The best compilation of standardized specifications is presented in the *API Specification For Mechanical-Drive Steam Turbines For General Refinery Services.*[1] Its suggested standard datasheet is given in Figure 14-25.

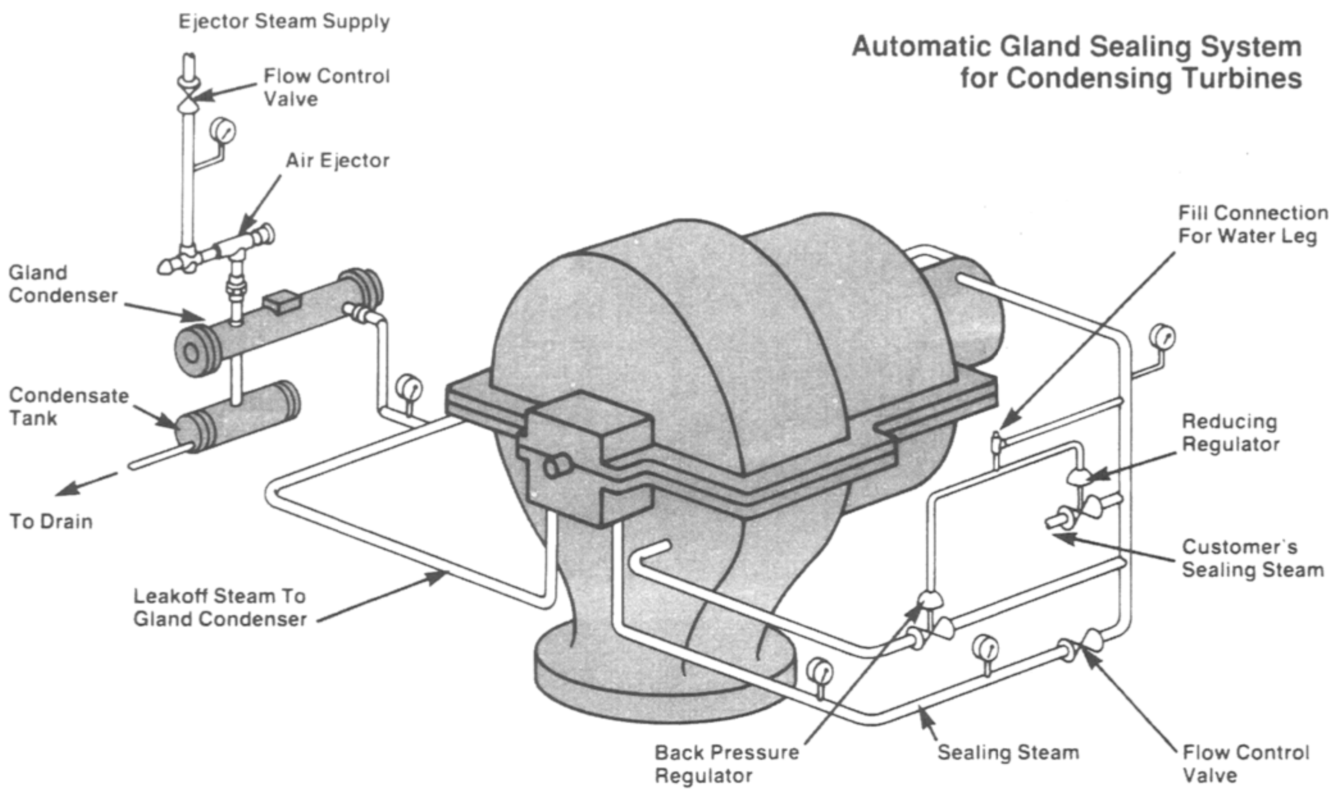
Other useful data forms are given in Figures 14-26A and 14-26B.

The turbine manufacturer should be given any information regarding partial loads, change in loads and speeds, speed range, changes in steam pressure (not just variations) and temperature, steam cost evaluation data, preference for economies in capital or operating costs, etc.

Performance

Exact performance can be given only by the manufacturer for a specified turbine selected to operate at a particular set of conditions. However, estimates can be made which are usually quite satisfactory for general evaluations and comparisons. The most useful criteria are the steam rate and the system cost. Steam rate is the flow of steam in pounds per brake horsepower output per hour through the turbine. It is established for a definite shaft horsepower output, given steam pressure and temperature, exhaust system pressure, and shaft rpm.[39]

Hand valves are used to reduce the load and to maintain reasonably good efficiencies. In general, one hand valve closed allows operation at 60–75% of rated load for a single valve unit. When the turbine has two hand valves, with both valves closed, the unit gives 50–60% of rated load; with one



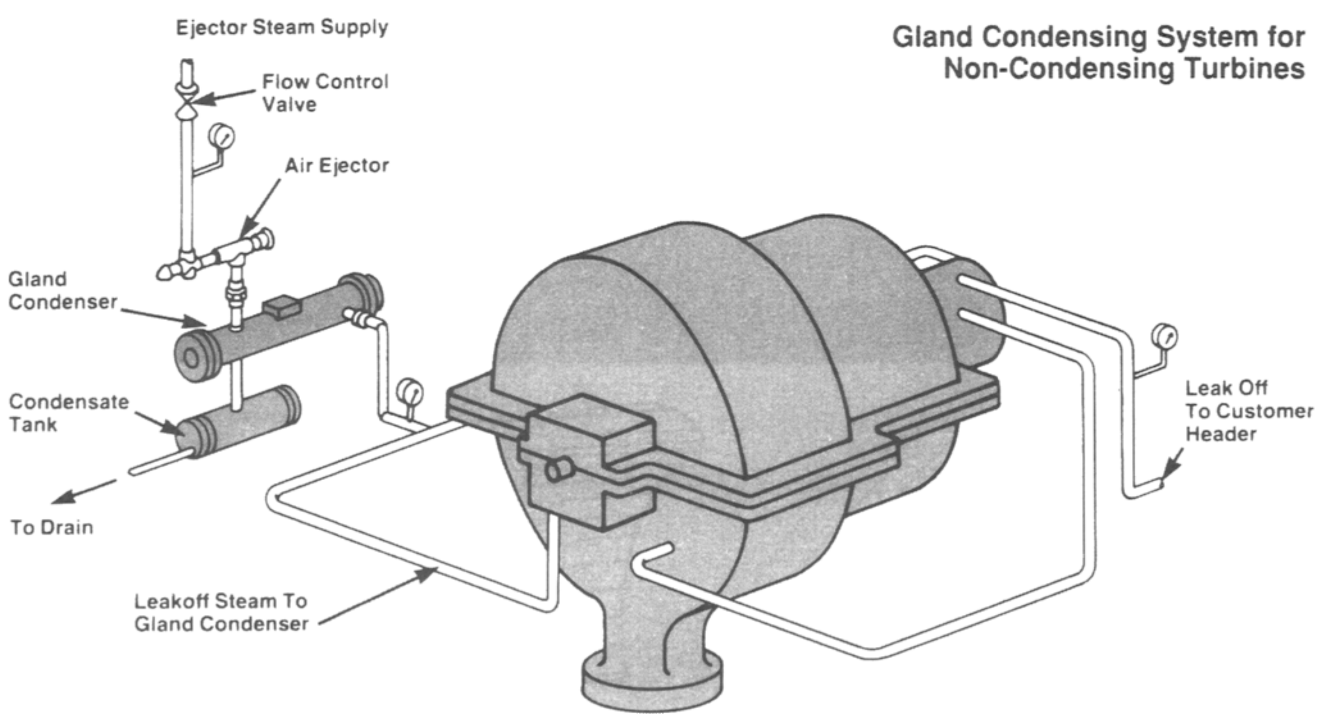


Figure 14-24. Steam sealing systems for condensing and noncondensing steam turbines. (Used by permission: Bul. 8908-EMD. Dresser-Rand Company.)

STANDARD DATA SHEET FOR STEAM TURBINES (AND GEARS)

PURCHASER ___	1	MANUFACTURER ___
DESTINATION ___	2	JOB NO. ___ ITEM NO. ___
ITEM NO. ___ SERVICE ___	3	QUOTE NO. ___ DATE ___
NO. REQUIRED ___ DRIVEN EQUIPMENT ___	4	SERIAL NO. ___
OPERATING CONDITIONS	5	**TURBINE SPECIFICATIONS**
HORSEPOWER: Rated ___ Normal ___	6	DESIGNATION ___ NO. OF STAGES ___
SPEED (rpm): Rated ___ Normal ___ Max Cont ___	7	POTENTIAL MAXIMUM HORSEPOWER ___
INITIAL PRESS. (psig): Max ___ Normal ___ Min ___	8	SIZE STEAM: Inlet ___ Exhaust ___
INLET TEMP (F tt): Max ___ Normal ___ Min ___	9	GOVERNOR TYPE ___ NEMA Class ___
EXH PRESS. (psig/in. Hg abs): Max ___ Normal ___ Min ___	10	NO. AUTO. VALVES ___ TYPE LIFT: Cam ___ Bar ___
MAX CASING PRESSURE ___ psig	11	ROTOR TYPE: Solid ___ Built-Up ___
COOLING-WATER SUPPLY: Temp ___ F Press. ___ psig	12	SPEED (rpm): Max Allow ___ 1st Crit ___ Trip ___
INDOOR ___ OUTDOOR ___ ROOF: Yes ___ No ___	13	BEARINGS: Radial Type and Size ___
WINTERIZATION: Yes ___ No ___	14	THRUST: Type and Size ___
DUTY: Cont ___ Intermit ___ Standby ___ Hrs/Yr ___	15	LUBRICATION: Type ___
APPROX STEAM RATE DESIRED ___ lb/bhp/hr	16	PACKING TYPE: End-Glands ___
STEAM COST ___ $/M# PAYOUT PERIOD ___ Years	17	INTERSTAGE DIAPHRAGMS ___
CONSTRUCTION FEATURES	18	NET WEIGHT ___ lb
TYPE: Vert ___ Horiz ___ NO. STAGES: Single ___ Multiple ___	19	MAIN OIL PUMP: Make ___ Type ___ gpm ___
ROTATION: (From Gov End) cw ___ ccw ___	20	Drive ___ hp ___
GOVERNOR: Mech ___ Hydr ___ Oil Relay ___ NEMA Class ___	21	AUX OIL PUMP: Make ___ Type ___ gpm ___
GOVERNOR VALVE TYPE: Multiple ___ Single ___	22	AUX OIL PUMP TURBINE: Make ___ Size ___
HAND VALVE: Min Steam Press. ___ Economy ___	23	FILTER: Make ___ Type ___ Size ___ Rating ___
AIR HEAD FOR INSTRUMENT CONTROL ___ JACKSCREW ___	24	**GEAR SPECIFICATIONS**
Range ___ Max rpm @ ___ psig Min rpm @ ___ psig	25	MANUFACTURER ___ DESIGNATION ___
HAND SPEED CHANGER: ___ Range ___ Max ___ Min rpm	26	BUILT-IN ___ SEPARATE ___
SEPARATE TRIP THROTTLE VALVE: Hydraulic ___ Mech ___	27	NO. OF REDUCTIONS ___ RATIO ___
Remote Trip ___ Actuation ___	28	CL. TO CL. OF SHAFTS ___ FACE WIDTH ___
BASE PLATE UNDER: Turbine ___ Turbine and Gear ___	29	TYPE OF BEARINGS ___ TYPE LUBR ___
OIL RESERVOIR: In Base ___ Separate ___	30	AGMA SERVICE FACTOR ___
LUBE SYSTEM: Separate ___	31	ROTATION L.S. SHAFT (From Gov End) cw ___ ccw ___
With Gear ___ With Driven Unit ___	32	**MATERIALS**
MAIN OIL PUMP: Integral ___ Separate ___ Drive ___	33	STEAM INLET PARTS ___
AUX OIL PUMP: Vert ___ Horiz ___ Turb ___ Motor ___	34	CASING ___ SHAFT ___
CURRENT ___ v ___ ph ___ cy	35	NOZZLES ___ NOZZLE RINGS ___
STEAM: Inlet ___ psig ___ F tt Exh ___ psig/in. Hg abs	36	WHEELS ___ BLADING ___
OIL COOLERS: Twin ___ Single ___	37	SHROUDS ___ GOV VALVE TRIM ___
STRAINERS OR FILTERS: Twin ___ Single ___	38	SHAFT MATERIAL UNDER PACKING ___
CONTROL PANEL: Yes ___ No ___	39	APPLIED BY: Spraying ___ Plating ___
PRESS. GAGES: Oil ___ Steam Chest ___	40	OIL COOLERS: No. ___ Type ___ Mfr ___ Rating ___
TACHOMETER: Vibr Reed ___ Electric ___	41	TUBES: No. ___ Dia ___ BWG ___ Lgt ___ Mtl ___
INSULATION: Yes ___ No ___ JACKET: Yes ___ No ___	42	MTL: Tube Sheet ___ Channel ___ Cover ___
COUPLING: Make ___ Type ___	43	**ALLOWABLE PIPING FORCES AND MOMENTS**
MOUNT COUPLING HALF: Yes ___ No ___	44	
STEAM INLET: Rating ___ Facing ___ Orient ___	45	INLET FLANGE / EXHAUST FLANGE
EXHAUST: Rating ___ Facing ___ Orient. ___	46	Force Lb / Moment Ft-Lb / Force Lb / Moment Ft-Lb
GEAR UNIT REQUIRED: Yes ___ No ___	47	PARALLEL TO SHAFT ___ ___ ___ ___
GEAR OUTPUT SPEED ___ rpm AGMA SERVICE FACTOR ___	48	VERTICAL ___ ___ ___ ___
ROTATION L.S. SHAFT (From Gov. End) cw ___ ccw ___	49	HORIZ 90° TO SHAFT ___ ___ ___ ___
ELECTRICAL EQUIPMENT HAZARD CLASS ___	50	**STEAM RATES**
TESTS REQUIRED (S = MFR SHOP, W = WITNESSED)	51	RATED ___ NORMAL ___ lb/bhp/hr
HYDRO ___ RUNNING ___ SPECIAL ___ INSP ___	52	___
REMARKS	53	MAX STEAM THRU NOZZLES ___ lb/hr
___	54	MAX ALLOWABLE PRESS. ON EXHAUST END ___ psig
___	55	**EXCEPTIONS TO SPECIFICATION**
___	56	___
___	57	___

Figure 14-25. Standard datasheet for steam turbines (and gears). (Used by permission: API Standard 615, *Mechanical Drive Steam Turbines for General Refinery Service*, 1st Ed., Appendix 1, ©1958. American Petroleum Institute.)

valve open the rated load is 75–80%. Both hand valves open gives 100% rated load.

Single-stage turbines are lower in cost, but usually lower in efficiency than a multistage unit. The small horsepower loads are generally best suited to the single-stage machines. Where the ratio of the inlet steam pressure (throttle) to exhaust pressure is small and the corresponding available energy is low, the single-stage turbine may be more efficient than a multistage unit, particularly when operating speeds are high.[44]

Steam Rates

Theoretical steam rate $= 2{,}544.1/(h_1 - h_2)$,
lb steam per hr per hp (14-15)

Because, (kilowatt hour) (0.746) = hp-hr, (14-16)

Theoretical steam rate $= 3{,}412.7/(h_1 - h_2)$,
lb steam per hr per kilowatt (14-17)

The theoretical values for enthalpy may be read from a Mollier chart, where h_1 is the enthalpy of the steam at the turbine inlet, and h_2 is the enthalpy of the steam at the exhaust pressure and at the inlet entropy. The expansion of steam through the turbine is theoretically at constant entropy. These theoretical rates must be corrected for performance inefficiencies of the particular turbine. The calculations presented here are good for the average design, but exact values for a particular make and model turbine must be quoted by

SPEC. DWG. NO.
A-

Page ___ of ___ Pages	
Unit Price	
No. Units	
Item No.	

Job No. ______________________

____

B/M No. ______________________

MECHANICAL DRIVE TURBINE SPECIFICATIONS

Make ______________________ Type ______________________

H.P. Rating ____________ At ____________ RPM. ______________________

OPERATING CONDITIONS

	Guarantee	Alternate
Speed, RPM		
Steam Rate, Lbs/Hp-Hr.		
Hand Valve(s) Position		
Steam Press. @ Throttle, PSIG		
Total Temp. @ Throttle, °F.		
Exhaust Steam Press. (PSIA) (Inches Hg Abs.)		
Extraction Steam Press, PSIG		
Extraction Steam Quantity, Lbs/Hr.		

Overspeed Test (No Load) ____________ RPM. Max. Allowable Continuous Operating Speed ____________ RPM

First Critical RPM ____________ . No. Stages ____________ .

Flange Size & Rating: Inlet ______________________ Exhaust ______________________

MATERIAL

Casing: Steam Chest ______________________ Exhaust End ______________________

Shaft ____________ Wheels ____________ Rotor Blades ____________

Diaphragm Blade ____________ Diaphragm ____________ Nozzle ____________

Journal Bearing Type ______________________ Thrust Bearing Type ______________________

Packing: Shaft ______________________ Shaft Sleeve ______________________

AUXILIARY EQUIPMENT

Trip Throttle Valve: Rating ____________ Make ____________ Material ____________

Governor: NEMA Class ____________ Variable Speed ________ Oil Relay ________ Hydraulic Orifice ____________

Constant Speed ____________ Direct Acting ____________ Mechanical ____________

Speed Changer: Range ______________________ Manual ______________________

Air Diaphragm Motor ______________________ Hydraulic ______________________

Hand Nozzle Valve(s). Number ________ (Total). Overload ____________ Partial Load ____________

Steam Admission Valve(s) Governor Controlled: Single ______________________ Multiple ______________________

Emergency Trip Valve: Combined with Throttle Valve ______________________ Separate ____________

Emergency Overspeed Governor: Independent ____________ . Set at ________ RPM. Non Sparking ____________

Tachometer: Electric ____________ Vibrating Reed ______________________

Casing Relief Valve: Set ____________ PSIG. Sentinel Warning ____________ Full Relief ____________

Steam Strainer: Integral ____________ Separate ____________ "Y" Type ____________

Vacuum Breaker ______________________ Steam Seal Piping ______________________

Steam Gage(s): Steam Chest ____________ Each Nozzle Chamber ____________ Exhaust ____________

Panel Mounted ____________ Mounted on Turbine ______________________

Insulation & Lagging: Complete Casing ______________________ Down to Exhaust End ____________

Steel Base for: Turbine Only ______________________ Turbine & (Pump) (compressor) ____________

Sole Plate Mounting on Purchasers Foundation ______________________

By	Chk'd.	App.	Rev.	Rev.	Rev.
Date					

P.O. To: ______________________

Figure 14-26A. Mechanical drive turbine specifications, part 1.

	SPEC. DWG. NO. A-
Job No.______	Page of Pages
______	Unit Price
B/M No.______	No. Units
	Item No.

MECHANICAL DRIVE TURBINE SPECIFICATIONS

TURBINE LUBRICATING SYSTEM: FORCED FEED

Independent of (Pumps) (Compressor)______ Combined ______

Oil Cooler: Tubes______ Water ______ @ ______ °F.

External Oil Cooler(s)______ O.D. Tubes. Single ______ Dual ______

External Oil Filter(s): Single ______ Dual ______

Oil Temp. Indicator(s): Dial Type ______ Stem Type ______ (After) (Before) Cooler ______

Bearing Thermometer(s): Dial Type ______ Stem Type ______ For Turbine ______

Oil Pressure Gauge(s): Total ______. Mounted: Piping ______ Panel ______

Sight Flow Indicator in Oil Line______ Leaving each Bearing ______

Main Oil Pump: Integral, Driven from Turbine Shaft; Separate Turbine Driven; Separate Motor Driven; Motor Class ______

Steam Turbine Driven Auxiliary Oil Pump for ______ PSIG ______ °F. T T Steam Exhaust ______ PSIG, Including Control.

Electric Motor Driven Auxiliary Oil Pump______ Starter ______

Low Oil Pressure Failure Switch(es): Electric ______ Pneumatic ______

Hydraulic ______ to Actuate Auxiliary Oil Pump Driver ______ Solenoid Trip ______

Pneumatic Dump______ , Classification ______

Low Oil Pressure Alarm Switch ______

Prefabricated Interconnecting Oil Piping ______

Oil Reservoir Located: Separate ______ In Base ______

All Electrical Equipment: Open Construction ______ Explosion Proof. ______

Motors & Starters: ______ Volts ______ Phase ______ Cycle

Protective Devices to be for______ Volts ______ Phase ______ Cycle

Air Supply Required @ ______ PSIG

HEAT LOADS

Turbine: Oil ______ BTU/Hr. ______ GPM ______ PSIG

Cooling Water______ °F. ______ GPM ______ PSIG

Compressor: Oil ______ BTU/Hr. ______ GPM ______ PSIG

Cooling Water______ °F. ______ GPM ______ PSIG

REMARKS

By	Chk'd.	App.	Rev.	Rev.	Rev.
Date					

P.O. To: ______

Figure 14-26B. Mechanical drive turbine specifications, part 2.

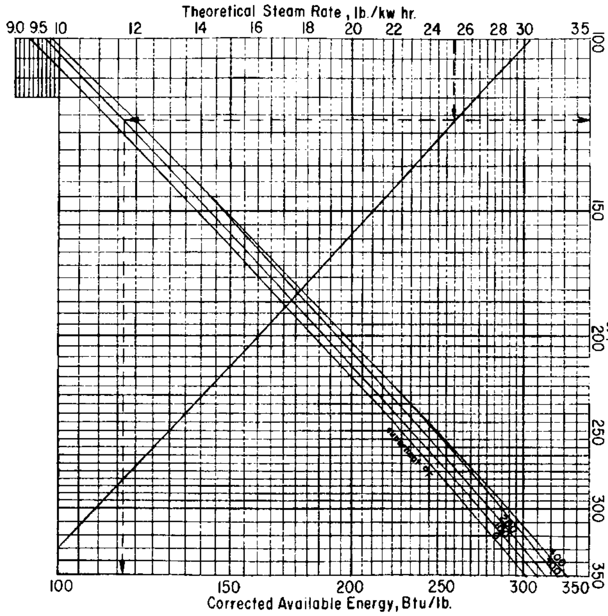


Figure 14-27A. Available energy in steam-theoretical steam rates, 9 to 35 lb/kw-hr, for single-stage general-purpose turbine. (Used by permission: Westinghouse Electric Corp., Steam Division.)

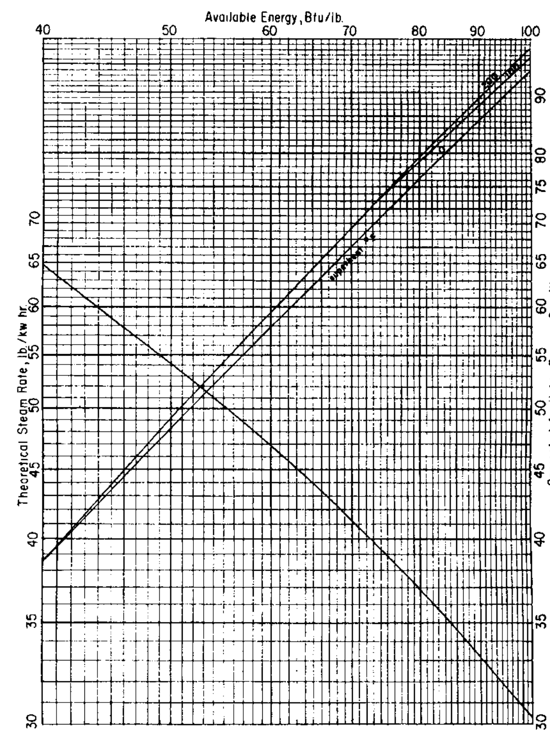


Figure 14-27B. Available energy in steam-theoretical steam rates, 30 to 70 lb/kw-hr, for single-stage general-purpose turbines. (Used by permission: Westinghouse Electric Corp., Steam Division.)

its manufacturer. In general, the difference between the rates will be less than 10%. Steam rates should be examined and compared for a variety of conditions in order to aid in an economical and efficient turbine selection.[43]

Theoretical system rate tables have been prepared by Keenan and Keyes,[22] or the values may be calculated as indicated.

Single-Stage Turbines

The calculation procedure for single-stage noncondensing general-purpose turbine[20] is as follows:

1. Determine theoretical steam rate.
2. Determine available energy in steam from Figures 14-27A and 14-27B.
3. Determine corrected available energy using superheat of steam, Figures 14-27A and 14-27B.
4. Determine basic turbine efficiency, Figure 14-28.
5. Determine horsepower losses from Figure 14-29A, 14-29B, or 14-29C.
6. Full load steam rate at rated speed:

$$= \left[\frac{2{,}545}{(\text{corrected available energy})\,(\text{efficiency})}\right]\left[\frac{\text{hp} + \text{hp loss}}{\text{hp}}\right] \tag{14-18}$$

where hp = rated horsepower

7. Total full load steam flow = (hp) (full load steam rate).

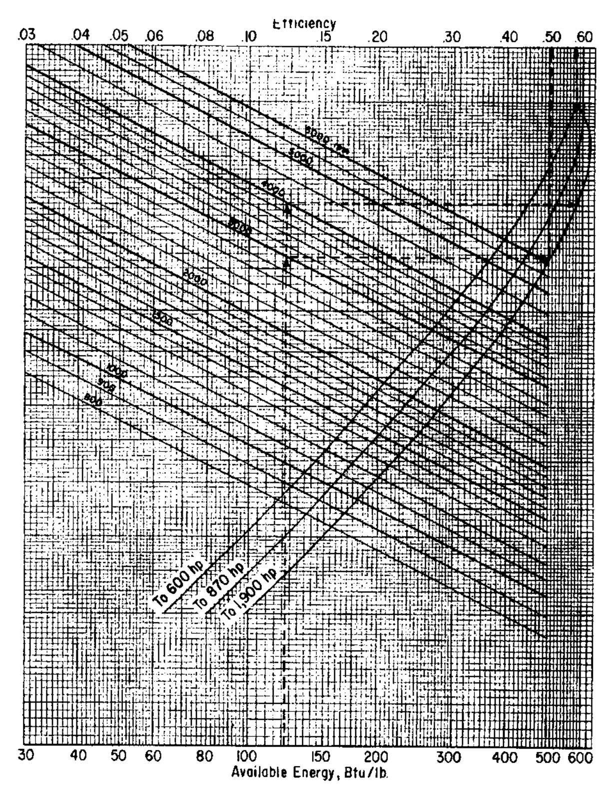


Figure 14-28. Basic turbine efficiency for single-stage general-purpose turbine. (Used by permission: Westinghouse Electric Corp., Steam Division.)

Single-Stage Noncondensing Partial Load at Rated Speed with No Hand Valve.[20] The steam rate at any partial load is obtained by using the full-load steam flow and the no-load steam flow. A Willans line (straight) is drawn between these points, and then the partial load can be determined at any rate.

1. Determine no-load flow factor, Figure 14-30. Determine 10^6/(full load steam rate) (rpm). Read no-load factor corresponding to the general size of turbine.
2. No load flow = (no load factor) (full load flow).
3. Plot full load flow at rated hp and no load flow at zero hp. Read steam flow at the required hp load.
4. Partial load steam rate, Figure 14-31, = partial load steam flow/partial load hp.

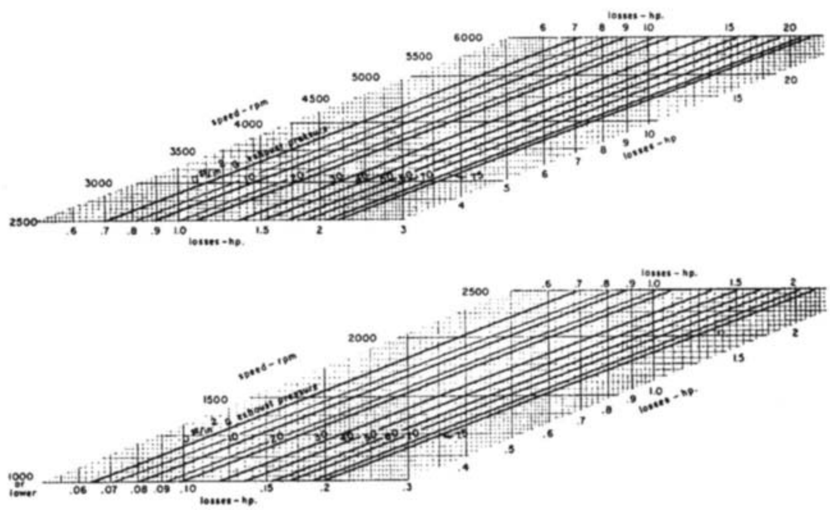
Single Stage Noncondensing Partial Load at Reduced Speed with No Hand Valve.[20]

1. Obtain basic efficiency of turbine at desired speed using Figure 14-28 together with the available energy as determined in Step 2 of the rated conditions. Determine corrected available energy in Step 3.
2. Determine the internal steam rate for the steam conditions and reduce speed by

Internal steam rate

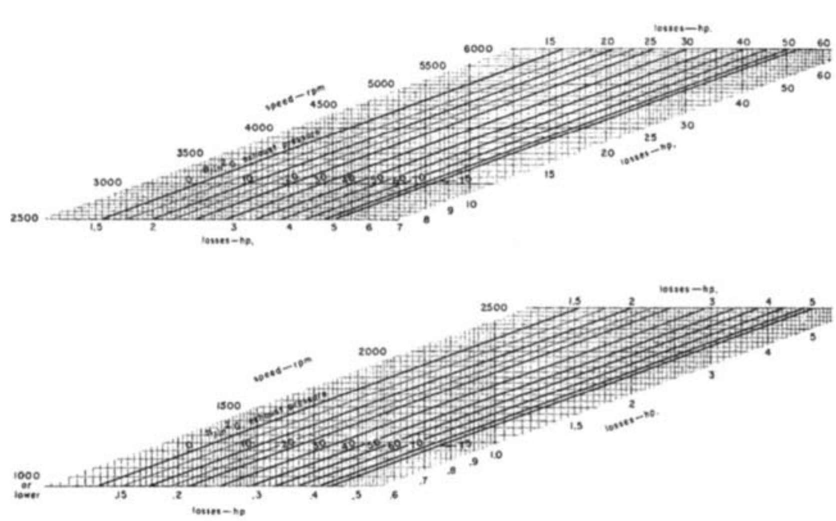
= 2,545/(corrected available energy) (efficiency)

3. Determine horsepower loss from Figures 14-29A, 14-29B, and 14-29C at the reduced turbine speed and exhaust pressure.



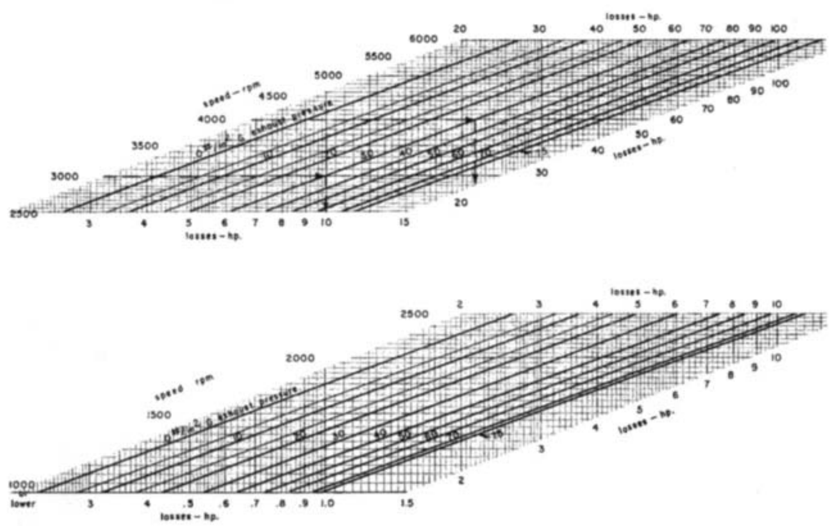
Inlet diameter of 3 inches, 250–600 psig max. at 500°F to 750°F maximum

Figure 14-29A. Horsepower losses, rated hp to 600. (Used by permission: Westinghouse Electric Corp., Steam Division.)



Inlet diameter of 4 inches, 250–400 psig max. at 500°F to 750°F maximum, or 3-inch, 600 psig maximum at 750°F maximum

Figure 14-29B. Horsepower losses, rated hp to 870. (Used by permission: Westinghouse Electric Corp., Steam Division.)



Inlet diameter of 6 inches, 250–400 psig max. at 500°F to 750°F maximum, or 600 psig maximum at 750°F maximum

Figure 14-29C. Horsepower losses, rated hp to 1,900. (Used by permission: Westinghouse Electric Corp., Steam Division.)

4. Determine net horsepower available at the reduced speed:

Net hp available

$$= \frac{\text{flow (lb/hr) at rated load and rated speed}}{\text{internal steam rate, Step 2}} - \text{hp loss} \quad (14\text{-}19)$$

5. Determine steam rate of the net hp available at the reduced speed and full-load rated flow:

$$= \frac{\text{full-load rated flow, lb/hr}}{\text{net hp at reduced speed}}, \text{lb/hr/hp}$$

6. Determine no-load flow factor from Figure 14-30. Calculate $10^6/$ (steam rate at net hp) (reduced rpm).
7. New no-load flow
   = (full load flow at rated speed) (no-load factor).
8. Plot Willans line, full-load steam flow at new hp and no load flow at zero hp. Read desired partial load flow at partial hp. Figure 14-31.
9. Steam rate at desired reduced load and speed:

$$= \frac{\text{steam flow at partial load}}{\text{hp at reduced speed and partial load}} \quad (14\text{-}20)$$

Partial Load at Rated or Reduced Speed with Hand Valve.[20] The hand valves are considered to give the turbine additional ratings (as percent of original rating), and these values may be used as rating points following the outline given for rated and also for partial loads.

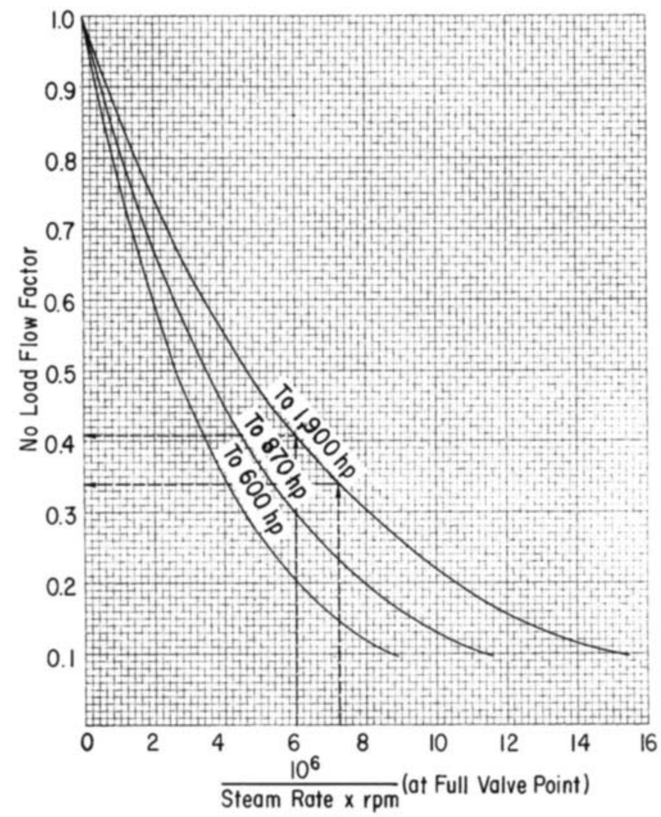


Figure 14-30. No-load flow factors for single-stage general-purpose turbine. (Used by permission: Westinghouse Electric Corp., Steam Division.)

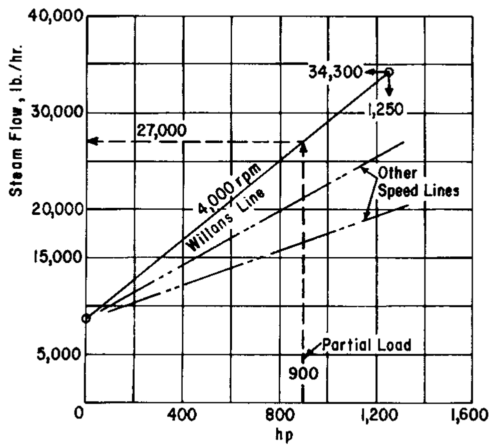


Figure 14-31. Willans line plot for partial load steam turbine..

Example 14-2*: Full Load Steam Rate, Single-Stage Turbine

Turbine: rated hp = 1,250
rated speed = 4,000 rpm
Steam: 350 psig at 600°F total temperature
saturation temperature = 435.6°F
Exhaust: 30 psig
Turbine: heavy-duty type

1. Theoretical steam rate at 350 psig and 600°F TT, and 30 psig exhaust = 18.31 lb steam/hr/KW (from tables)

$$\text{lb steam/hr/hp} = (18.31)(0.746) = 13.68$$

2. Available energy (Figure 14-27) = 174 Btu/lb.
3. Corrected available energy at 165°F superheat (Figure 14-27) = 163 Btu/lb.
4. Basic turbine efficiency (Figure 14-28) = 0.58.
5. Horsepower losses (heavy-duty type) = 21.4 hp.
6. Full load steam rate at rated speed:

$$= \left[\frac{2{,}545}{(163)(0.58)}\right]\left[\frac{1{,}250 + 21.4}{1{,}250}\right] = 27.4 \text{ lb/hr/hp}$$

7. Total full-load steam flow = (1,250)(27.4)
= 34,300 lb/hr.

Example 14-3*: Single-Stage Turbine Partial Load at Rated Speed

For the turbine in Example 14-2, determine the steam rate when the unit is loaded to only 900 hp but must run at a rated speed of 4,000 rpm.

Draw a Willans line chart, Figure 14-31.

1. No-load flow factor, Figure 14-30.

$$\frac{10^6}{(\text{full load steam rate})(\text{rpm})} = \frac{10^6}{(27.4)(4{,}000)} = 9.12$$

Reading curve, factor = 0.26 (heavy-duty turbine)

2. No load flow = (34,300) (0.26) = 8,930 lb/hr.
3. Willans line plot Figure 14-31 at partial load of 900 hp, the steam flow should be 27,000 lb/hr.
4. Steam rate at partial load and rated speed
= 27,000/900 = 30.0 lb/hr hp.

Calculation Procedure for Single-Stage Condensing General-Purpose Turbine.[20] Calculate as shown for a noncondensing turbine using the proper theoretical steam rate corresponding to the exhaust pressure. The hp losses are approximated by using the 0 psig exhaust line of Figure 14-29A, 14-29B, or C. The resulting steam rate, when calculated using the procedure outlined, must be multiplied by a correction factor.

Duty	Factor
Light	1.08
Medium	1.05
Heavy	1.00

The results are approximate, but satisfactory for most calculations and studies.

Using Turbines with Reduction Gears. When gear reducers are connected to steam turbines, the steam rates must be increased:

1. At rated horsepower and speed: 2%
2. At constant speed, the loss in percent is inversely proportional to the hp:

$$\text{New steam rate} = \text{partial load rate}\left[1 + 0.02\left(\frac{\text{rated hp}}{\text{reduced hp}}\right)\right] \tag{14-21}$$

3. At reduced speed, loss in percent is directly proportional to speed and inversely proportional to load:

$$\text{New steam rate} = [\text{partial load rate}]\left[1 + 0.02\left(\frac{\text{rated hp}}{\text{reduced hp}}\right)\left(\frac{\text{reduced rpm}}{\text{rated rpm}}\right)\right] \tag{14-22}$$

Effect of Wet Steam. Steam turbines should not be operated with wet steam. The manufacturers will not guarantee performance when the moisture is greater than 3%.

Steam rates calculated for dry saturated steam are increased as follows:

*The above examples used by permission of Westinghouse Electric Corp., Steam Division.

Two percent for each 1% of moisture, up to a maximum of 3%.

Multistage Turbines

See Figures 14-16E– H and 14-20B.

Multistage steam turbines are used for higher horsepower loads and often higher rotating speeds than the single-stage units. Figure 14-32 is a guide in the application range for single-stage and multistage units.

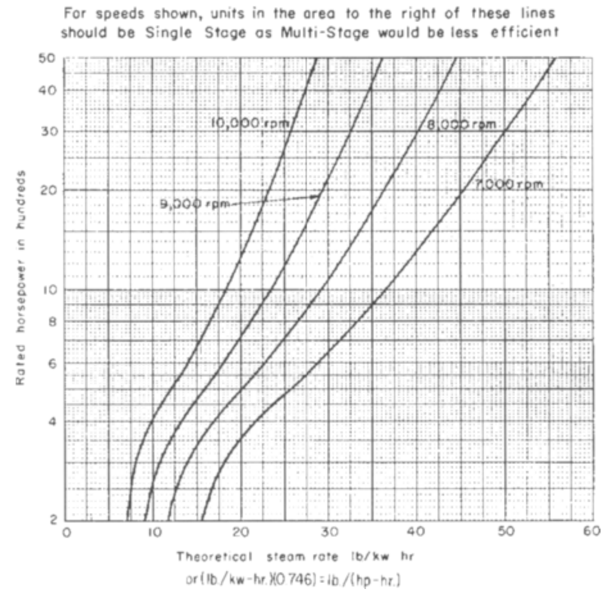


Figure 14-32. Limits of multistage steam turbine application. (Used by permission: General Electric Company.)

Gas and Gas-Diesel Engines

Gas engines are two- and four-cycle piston-operated internal combustion machines using natural or other gas mixtures for firing, Figures 14-33. These engines are available in horsepowers starting at about 25 hp, Figure 14-34 and reaching up to 10,000 hp and higher. Gasoline and butane-propane fueled engines are available in the horsepowers below 200. The gas-diesel engine is primarily adapted to the large horsepower loads and can operate either as a gas engine or as a diesel, usually using the gas for starting and diesel fuel for continuous operation.

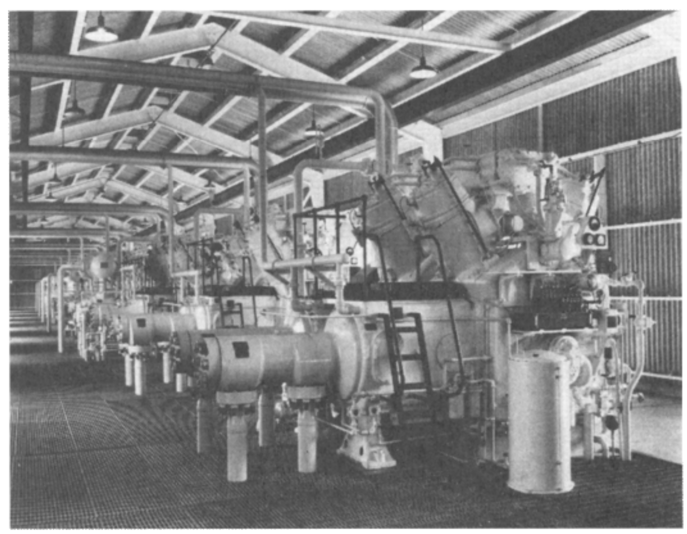


Figure 14-33. Gas engine driven parallel compression cylinders in process gas plant service. Note that the front side of gas engines are on the right with high-pressure compressor cylinders extending horizontally left. Also note the suction side pulsation drums on top of compressor cylinders, mid-way. (Used by permission: Cooper-Cameron Corporation, Reciprocating Products Division.)

Application

Engine drivers are well adapted for reciprocating compressor cylinders, direct connection to power generators, direct or through gear connection to fans, centrifugal compressors, pumps, etc.[5, 6]

In addition to the engine, facilities are needed for cylinder jacket cooling, exhaust manifold cooling, lube oil cooling and cleaning, air pressure starting, and electrical ignition. A variety of arrangements are used for each of these, depending upon the application, type of engine, horsepower, and geographical location of the equipment.

Foundations must be designed to take the weight loads and overturning moments without transmitting vibration to other equipment and buildings.

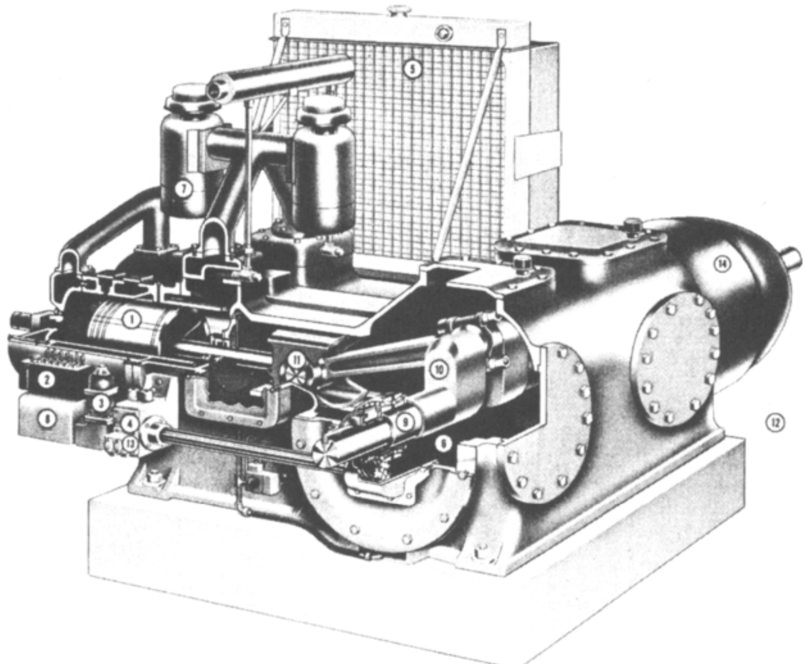
Engine Cylinder Indicator Cards

The power indicator cards of typical two- and four-cycle gas engines are compared with an accepted goal for a fuel-air cycle. The performance depends upon the type and composition of the fuel, Figure 14-35.

The four-cycle engine takes two complete piston strokes for exhaust, scavenging, and charging. The two-cycle engine exhausts, scavenges, and charges for about 25% of its piston travel before bottom center, and until about 25% after bottom center.[21] The two-cycle machine does not have intake and exhaust valves but uses ports.

Referring to Figure 14-35, the stroke generally represented by CD is the power portion; ABC is the compression. In a four-cycle gas-diesel, the fuel is injected along BC, and the exhaust valves open at D. For a two-cycle gas engine, the exhaust ports close at about E on the AB compression stroke. The exhaust ports open at F.

The indicator card is useful in balancing the load per cylinder. The area of the card and the pressure-length scale are used to determine the horsepower.



(1) Two-cycle design eliminates troublesome valves, rocker arms, tappets, push rods and cams. A power stroke every revolution, slow speed, low BMEP and port scavenging are Ajax design features.

(2) Automatic lubricator force feeds lubrication to power cylinders. Oil level gauge, automatic fill valve and low-oil-level shutdown are standard.

(3) Heavy-duty governor provides constant-speed operation. Overspeed shutdown switch is standard. Pneumatic governor operator available for speed and capacity control.

(4) Hydraulic fuel injection system is standard on DP-125 and DP-165 and optional on EA-30 through DP-81. Positive pressure injection yields up to 35 percent in fuel savings.

(5) Thermosyphon cooling provides optimum cylinder cooling in a closed system without the use of a thermostatic bypass valve and water pump. High engine jacket water temperature shutdown is standard on all units. A hydrogen sulphide corrosion-resistant radiator is optional.

(6) Splash lubrication system provides lubrication to crossheads, crosshead pin bearings and crankpin bearings. The crankcase is sealed from products of combustion, reducing oil changes to one-year intervals. Eliminates the requirements for a troublesome oil pump, cooler and filter. Crankcase oil level gauge, automatic fill valve and low-oil-level shutdown are standard.

(7) Oil bath air filters eliminate frequent cleaning and replacement. Oversize design allows extra-long service in difficult atmospheres.

(8) High-reliability Altronic ignition system features only one moving part, the alternator, which runs on sealed ball bearings. This component eliminates the magneto and breaker points. The units are timed at the factory and no further adjustments are necessary.

(9) Tapered roller bearings are double-row and sized above the load-carrying ability actually required.

(10) Closed-die-forged components, including crankshaft and connecting rods, are forged from high-alloy steel in precision dies.

(11) Crosshead guide absorbs thrust vectors and precludes misalignment.

(12) Rugged construction assures long life with low maintenance. Compare our ten- and twenty-year overhauls to annual and semiannual overhauls on high-speed multicylinder engines.

(13) Air/gas starting equipment is standard on DP-60 through DP-165 and optional on EA-22 through E-42. Electric starting is optional on EA-22 through DP-81.

(14) Clutch power takeoff is oversized for extended life and standard on all units.

Figure 14-34. Ajax model rated 22–165 bhp gas engine for driving remote oil field equipment, 2-cycle, fuel injection. Driven load attached to power take-off, item (14). (Used by permission: Bul. 2-214. ©Cooper Cameron Corporation Cooper Energy Services, Ajax Superior.)

$$\text{Indicated hp} = \frac{P\,L\,A\,N}{33{,}000}, \text{ per power cylinder end} \tag{14-23}$$

where L = length of piston stroke, ft
P = mean indicated pressure by measurement of the indicator card

$$P = \frac{(\text{area of card, in.}^2)\,(\text{pressure scale, lb/in.})}{\text{length of card, in.}}$$

A = area of power piston, in.^{2}
N = number of power strokes per minute. For two cycle engines, N = engine rpm, for four-cycle engine, N = engine rpm/2

For a multicylinder engine, the total horsepower is the sum of the power of all cylinders.

$$\text{Mechanical efficiency} = \frac{\text{bhp delivered}}{\text{indicated hp}} \tag{14-24}$$

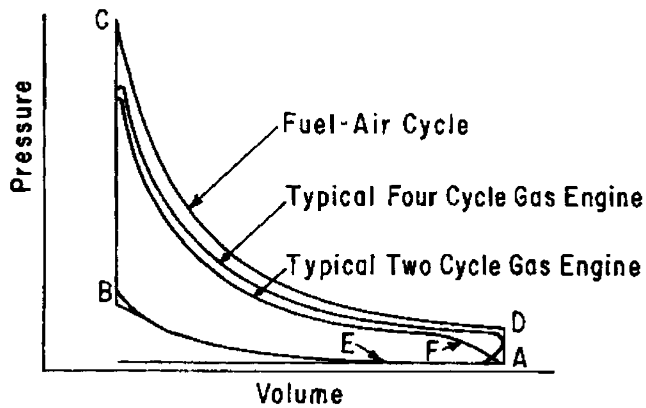


Figure 14-35. Theoretical and actual gas engine indicator cards. (Used by permission: Newcomb, W. K., 24th Conference, Oil and Gas Div.. ©American Society of Mechanical Engineers.)

Speed

Normal speeds range from about 200–800 rpm, with lower speeds usually associated with the larger engines. For horsepower requirements around 800–1,500 hp, the rated speeds might be 300, 330, 400, and 440. From any given rated speed the engine will satisfactorily drop to 50% speed and rise to about 110–115% overspeed. This varies with the engine and its application.

Turbocharging and Supercharging

The power available from the power cylinders can be increased over conventional design by adding additional air per stroke as well as exhaust cooling, etc.

Specifications

Figure 14-36 is a convenient summary sheet for most of the pertinent specifications for a gas engine. Supplemental information to the manufacturer at the time of inquiry should include the following:

1. Type of application; equipment to be driven.
2. Duty, whether continuous or intermittent.
3. Type and make of any direct-connected (to shaft) gears, blowers, etc. Give torque requirements.
4. Type of fuel.
5. Speed control characteristics.

Combustion Gas Turbine

The gas combustion turbine, more commonly called the gas turbine, uses natural or other gas or liquid fuel for combustion and the generation of hot gases for expansion through a power turbine. The industrial gas turbine is preferably suitable and economical for applications of 50,000 and higher horsepower, although some designs are applicable for lower hp requirements.

Compressed air for the combustion is supplied by a centrifugal compressor drive by the power turbine. The extra power is available for mechanical drive or other equipment or for power generation, see Figures 14-37A–D.[105, 106, 107]

A combustion system detail is shown in Figure 14-38A–B and in Molick.[25].

The gas turbine most often used for industrial plant applications is the simple cycle, single-shaft unit shown[103] in Figure 14-38A and 14-38B. The compressor is a centrifugal on the common shaft with the turbine itself. Sometimes an electric motor (see Figure 14-38A) is set up to drive the air compressor in order to allow the system to get started, because air is needed for the combustion, even though the startup conditions may be on limited scale, and then builds up as the gas fired turbine becomes more operational. The load is then connected by direct coupling to the single shaft of the drive arrangement or through a gear train connecting to a process centrifugal compressor or large blower, or other large mechanical equipment.

Rowley and Skrotzki[35] provide an excellent summary of the operation and performance of the gas turbine, see Figure 14-40. The following is copied by permission:

> "The simplest form of gas-turbine plant, Figure 14-40, consists of an *air compressor, a combustion chamber, and a turbine.* The compressor takes in atmospheric air and raises its pressure. In the combustion chamber, fuel burns in the compressed air, raising its temperature and increasing its heat energy. This produces a working fluid that can be expanded in the turbine to develop mechanical energy, just as the expansion of steam does in the more familiar steam turbine. Part of the gas turbine's energy output goes to drive the compressor and the remainder is available as useful work. The fact that the working fluid is a gas gives the gas turbine its name; there is no connection with the fuel burned, which may be a liquid, gaseous, or possibly solid. To know how gas turbines behave requires a clear understanding of the various steps in the cycle. Because the only known way to make a heated fluid produce mechanical energy is to allow it to expand from a higher pressure to lower one in an engine or turbine, we must start with a compressor to create the higher pressure. It is conceivable that expansion of compressed air in the turbine could generate energy to drive the compressor and the machines could operate without burning fuel in a combustion chamber. But both turbine and compressor are less than 100% efficient and hence turbine output cannot match compressor load because of the losses incurred in compression and expansion."

SPEC. DWG. NO.
A-

Job No. ____________

B/M No. ____________

Page	of Pages
Unit Price	
No. Units	
Item No.	

COMPRESSOR DRIVE SPECIFICATIONS

GAS ENGINE

Make ____ Model ____ Type ____

No. Power Cylinders ____ No. Cycle ____ (Supercharged) (Turboflow) (____)

(Direct) (Belt) (Gear) Drive to Compressor Cylinders) Altitude ____ Ft.

Rated BHP at Sea Level ____ @ ____ °F: ____ RPM. Speed Range ____ Max. ____ Min.

Diameter of Power Cyl. ____ inches. Length of Stroke ____ Inches.

Fuel Consumption @ Full Load Torque ____, ¾ Load ____ ½ Load ____ BTU/BHP. Hr.

Brake MEP of Power Cyl. @ Full Load Torque ____

Fuel Gas: Press. ____ PSIG. Volume of Tank ____ cu. Ft. (By Purchaser)

Starting Air: Press ____ PSIG. Cu. ft. Free Air/Start ____ Based on ____ Rev.

Tank Capacity, Cu. ft./one Start ____.

Lubricating Oil System: Sump Capacity ____ Gal. Pump Capacity ____ GPM at Rated Speed.

Oil Outlet Temp. ____ °F. Oil Pressure ____ PSI. Heat Rejected to Oil ____ BTU/BHP/Hr.

Cooling System: Capacity of Engine Jackets ____ Total Gals.

Heat Rejected to Engine Jackets ____ BTU/BHP/Hr.

ΔP Engine Jackets: GPM ____ ____ ____ ____ ____ ____ ____ ____

ΔP, Ft. Water ____ ____ ____ ____ ____ ____ ____ ____

Rec. Inlet Temp. to Jackets ____ °F. Max. ΔT Across Jackets ____ °F.

ACCESSORIES

Lube Oil Cooler ____ Lube Oil Pump ____ Oil Cooled Power Pistons ____

Full-Force Feed Lubrication to Power and Compressor Cyl. ____ Governor ____

Full-Flow Lube Oil Filter and Strainer ____ Ignition Wiring ____

Ignition Type ____

Automatic Air Starting Valves and Quick Opening Main Air Valve ____

Automatic Fuel Gas Shutdown for (a) High Jacket Water Temp. ____

(b) Low Lube Oil Press. ____ (c) Overspeed ____

Pyrometer and Individual Cyl. Thermocouples ____ Fuel Gas Throttle Valve ____

Hand Operated Barring Over Device ____ Fuel Gas Shut-Off Valve ____

Diaphragm Operated Governor ____ Tachometer (Electrical, Mechanical) with Clutch ____

Flywheel Guard ____ Lubricating Oil Priming Pump ____

Alarm for High Jacket Water Temp. and Low Lube Oil Press. ____

Special Piston Rings ____

Exhaust Silencers: Make ____ Model ____ Size ____

Oil Bath Air Filter: Make ____ Model ____ Size ____

Backfire Relief Valve ____ Size ____

Gauge Board ____ Access Stairs, Platforms etc. to Engine ____

Pipe Connections, (Diameter Inches): Starting Air ____ Exhaust ____

Fuel Gas ____ Lube Oil Inlet ____ Lube Oil Outlet ____

Water Inlet ____ Water Outlet ____ Scavenging Air Inlet ____

REMARKS

Furnish Make, Model, Size, Type, Materials of Construction, etc. on all Vendor Furnished Equipment.

By	Chk'd.	App.	Rev.	Rev.	Rev.
Date					

P.O. To: ____

Figure 14-36. Compressor drive specifications—gas engine.

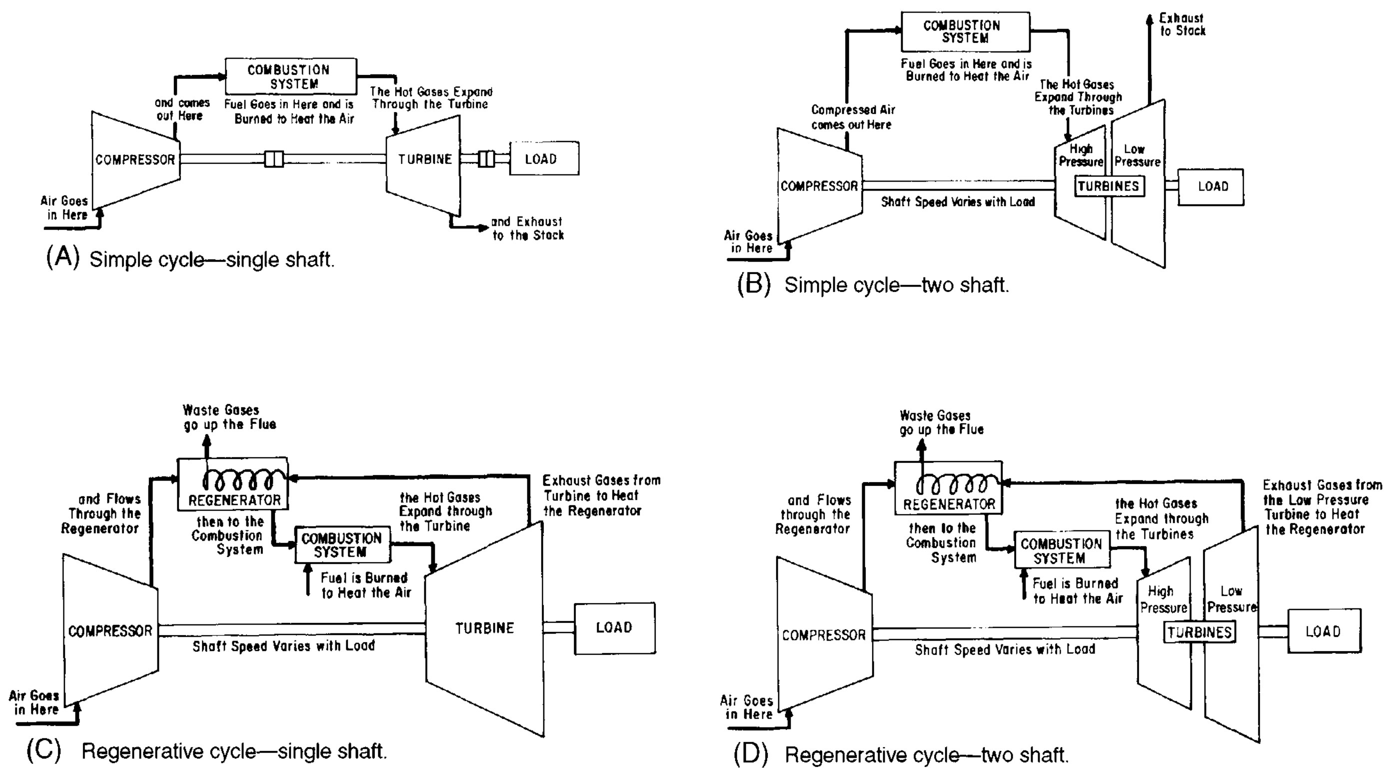


Figure 14-37. Gas turbine cycles A, B, C, and D. (Used by permission: General Electric Company.)

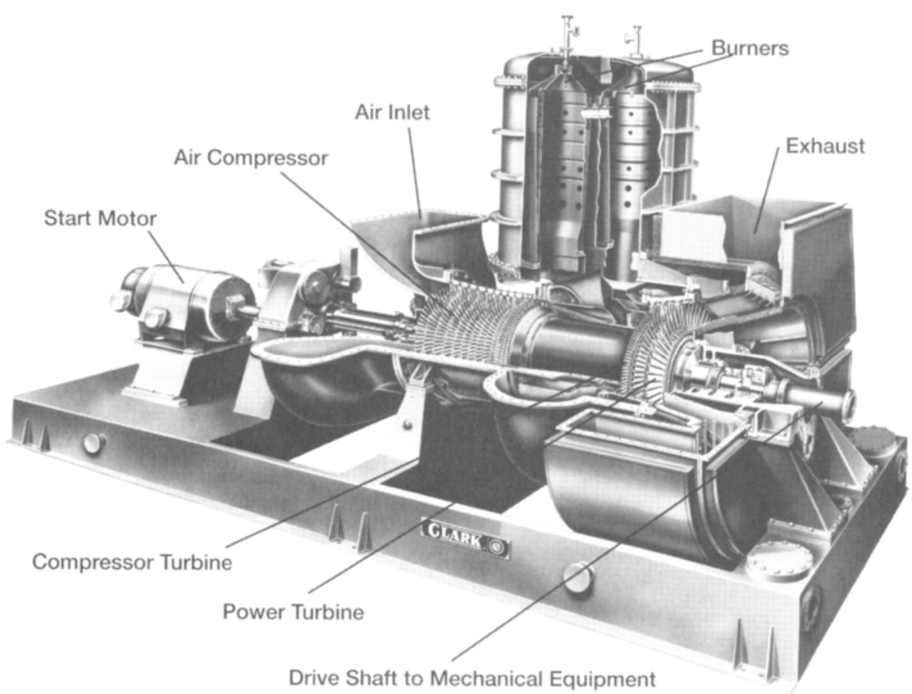


Figure 14-38A. Section of gas turbine showing basic features. (Used by permission: Dresser-Rand Company.)

Several common arrangements have been developed to improve the efficiency and/or to accomplish specific purposes, such as furnishing hot flue gas to boilers or to process gas exchangers in addition to the simultaneous generation of power or driving equipment.[35] Figures 14-37A–D illustrate several common arrangements or cycles.

Figure 14-41 illustrates a couple of power capacity styles of gas turbine generators for providing the required hot gas to drive a power turbine connected to mechanical applications such as power generation, gas compression, cogeneration, oil pumping, and others.

Figure 14-38B. Example of gas turbine operating cycle. (Used by permission: General Electric Company.)

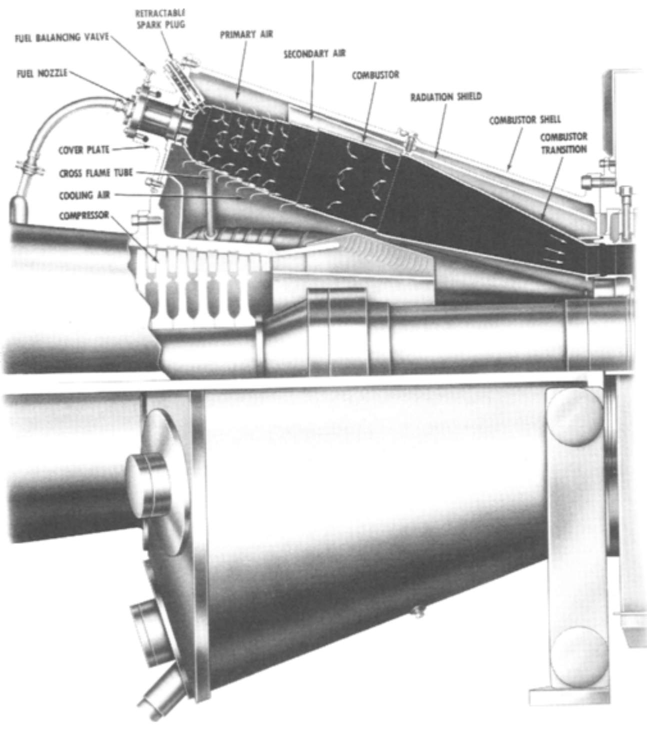


Figure 14-39. Gas turbine combustion unit. (Used by permission: Westinghouse Electric Corporation.)

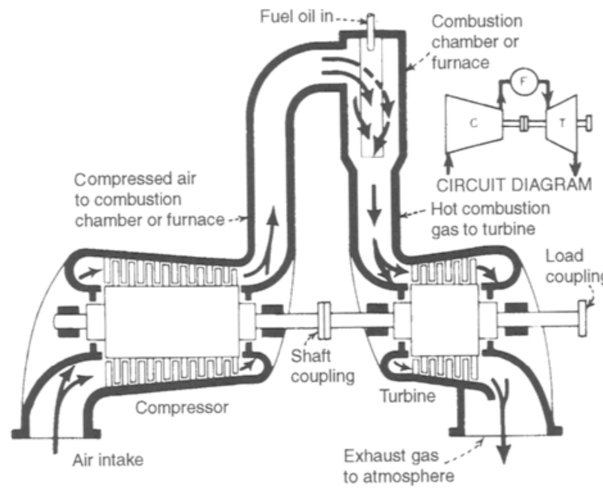


Figure 14-40. Basic simple gas-turbine consists of three principal pieces of equipment: (1) compressor that raises pressure of atmospheric air and discharges it to (2) combustion chamber or furnace where the burning of fuel raises air temperature before it enters (3) turbine through which the heated gas expands and does work on the turbine blades. Major part of turbine output goes to drive compressor; remainder is available power to drive shaft-connected mechanical equipment such as centrifugal compressor or other. (Used by permission: Rowley, L. N. and B. G. A. Skrotzki. "Gas Turbines," *Power*, p. 79, Oct. 1946. ©McGraw-Hill, Inc. All rights reserved.)

Nomenclature

A = area of power piston, in.^{2}
ac = alternating current
dc = direct current
E = volts
F_p = power factor; also see PF = active power
f = frequency,cycles/sec.
h_1 = enthalpy of steam at inlet conditions, Btu/lb
h_2 = enthalpy of steam at exhaust conditions, Btu/lb
hp = horsepower, or HP
I = current, amperes; OR, = induction motor
kv = kilovolts
kwh = kilowatt hours
k = radius of gyration, ft
kva = kilovolt-amperes, or KVA
kw = kilowatts
L = length of piston stroke, ft
N = number of power strokes per min
n = number of poles for motor
PF = power factor; OR, = F_p
P = power, or work, watts, or kW (kilowatts), also
P = mean indicated pressure by measurement of the indicator card, lb
rpm = revolutions per minute, or RPM
R = resistance, ohms; OR, radius of disk
S = synchronous motor
s = percentage slip
Steam rates = lb steam/hr/hp
TSR = theoretical steam rate, lb/kwh
V = volts or voltage
W = watts power; ALSO, P; OR, = weight, lb
Wk^2 = flywheel effect
θ = vector diagram angle of current between apparent power and active power

References

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Models	Power bhp / kW	Speed rpm	Fuel Rate Btu/hph / kJ/kWh	Thermal Efficiency %	Exhaust Temp °F / °C	Overall Weight[1] lb / kg
Coberra 2000	20,240 / 15,093	5,500	8,570 / 12,124	29.6	825 / 441	50,000 / 22,680
Coberra 6000	35,000 / 26,100	4,800	6,825 / 9,653	37.3	870 / 465	58,000 / 26,300

Rolls-Royce Gas Generator (GG) Model	GG Speed rpm	GG Compression Ratio	GG Compressor Stages	GG Turbine Stages	GG Exhaust Mass Flow lb/sec / kg/sec	GG Combustors	GG Turbine Inlet Temp °F / °C
Avon	7,600	9.2:1	17	3	171 / 78	8	1608 / 876
RB211	9,255[3] / 6,560[4]	20:1	7–6	1–1	198 / 90	1[2]	2128 / 1164

Cooper-Bessemer® Power Turbine Model	Power Turbine Stages
RT-48	2
RT-62	2

[1]Approximate weight excluding intake and exhaust system
[2]Annular [3]HP compressor [4]IP compressor

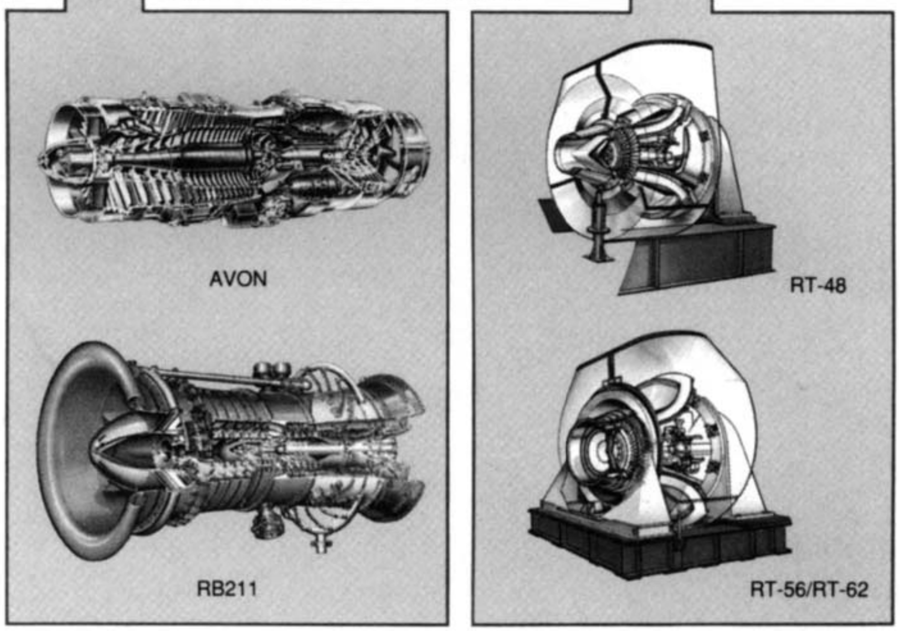

Figure 14-41. Performance specifications for one manufacturer's gas turbine generators that drive the power turbine unit for mechanical and power applications. (Used by permission: Bul. 6-204B. ©Cooper Cameron Corporation, Cooper Rolls Division.)

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