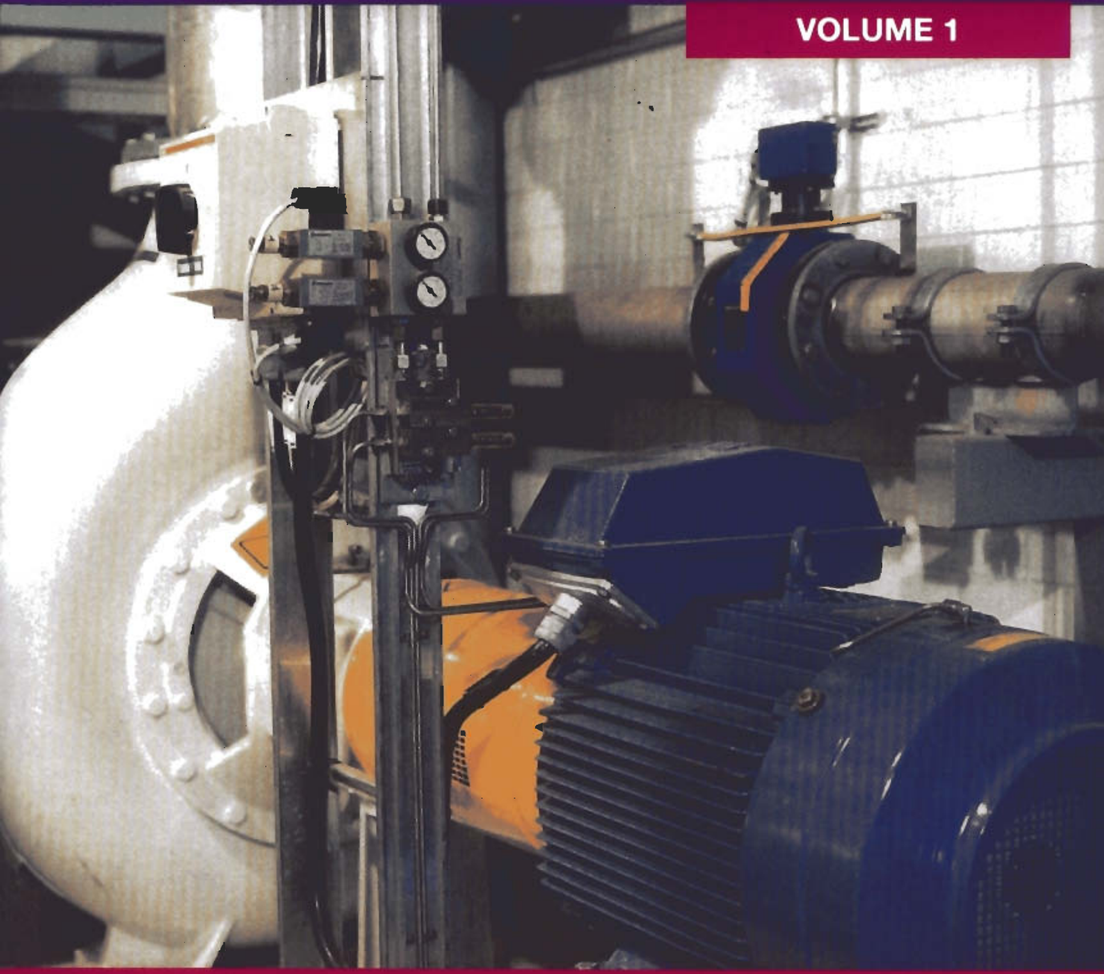
THIRD EDITION

Practical Machinery Management for Process Plants

Improving Machinery Reliability

VOLUME 1



Heinz P. Bloch

Practical Machinery Management for Process Plants

VOLUME 1 ▪ THIRD EDITION

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Practical Machinery Management for Process Plants:

Volume 1: Improving Machinery Reliability, 3rd edition
Volume 2: Machinery Failure Analysis and Troubleshooting, 3rd edition
Volume 3: Machinery Component Maintenance and Repair, 2nd edition
Volume 4: Major Process Equipment Maintenance and Repair, 2nd edition

Other Machinery Engineering Texts from the Same Author:

Introduction to Machinery Reliability Assessment, 2nd edition
Reciprocating Compressors: Operation and Maintenance

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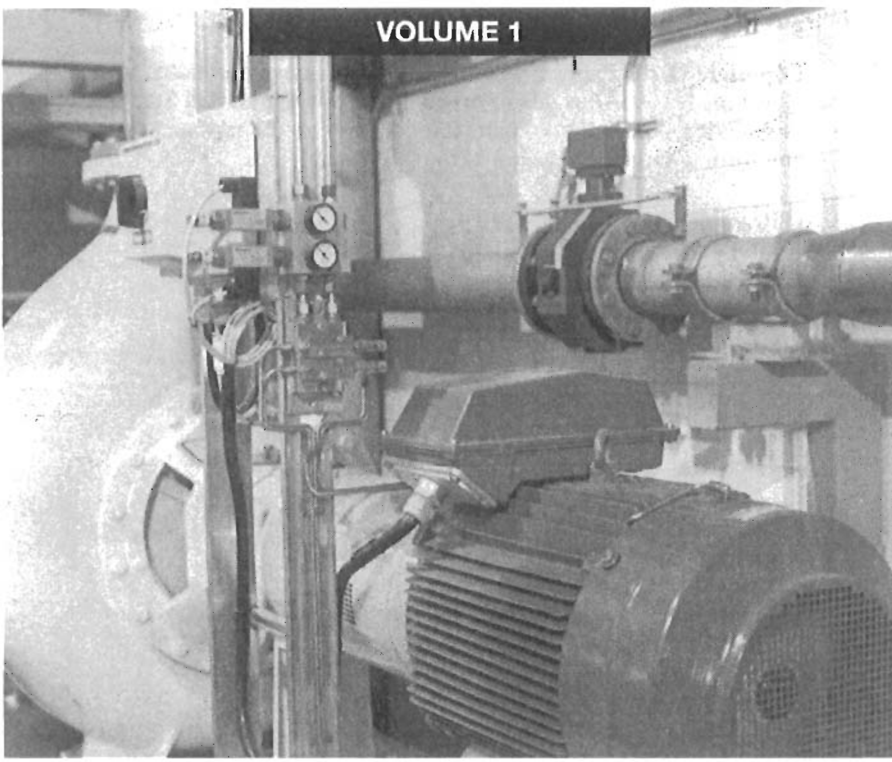

Heinz P. Bloch

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Preface

Most of today's process plants proudly display a Company Vision statement. Sadly, relatively few pursue the kinds of action needed to reach their often lofty visions. Conversely, it should be clear to us that a serious company will take steps *today* to identify and implement the science and technology "investments" necessary for modern petrochemical plants to remain competitive into the *next* decade and beyond.

Based on my observation or perception of trends among the trendsetters and the forward thinking of the "Best-of-Class" companies, I would like to alert the reader to a few of the work processes, organizational realities, lineups or interfaces, as well as hardware and software systems that have been implemented by the most profitable process plants in my career, dating from the 1960s to the present.

I will summarize by giving a few important explanations. First, none of the items I highlighted in this third edition were concocted for the sake of compiling a wish list of far-fetched goals. Every one of the various observations and recommendations either reflects current practice or has been implemented by one or more plants in the United States or overseas.

Second, no single plant presently applies or implements all the recommendations or practices given here. It is nevertheless of real importance to acknowledge that some companies come surprisingly close to practicing these reliability concepts or will soon implement them. The future belongs to them.

Third, it may not be realistic to expect every company to have the same priorities for implementing what is perceived to be the ideal path toward high reliability and profitability. However, it would be equally unrealistic to assume that a company can pick and choose from a smorgasbord of easy items and forget about the politically difficult ones. Measuring up to tough competition will require an uncompromising and single-minded desire to pursue excellence. Paying lip service to reliability and profitability concepts without implementing the difficult and sometimes unpopular steps necessary to get there is a costly exercise in futility and is doomed to failure.

Finally, we should all recognize the interwoven relationship of so many of the requirements and issues. It is important to realize that we can logically hold someone accountable for quality and solid performance only after training that person. Progress implies change. Change implies risk and extra effort to manage the risk. We can better justify, specify, purchase, install, operate, and maintain process plant machinery *only* if we invest time and money up front in reading and learning about best available practices. That, of course, is what this book is all about.

Many of my colleagues in process plants, machinery manufacturing facilities, or in the consulting field are practitioners of the various reliability improvement or assurance approaches. And for allowing me to include some of their work in this revised and updated text, sincere thanks go to Paul Barringer, whose work on life cycle costing and reliability assessment is truly unique; Lou Bewig for some excellent work on benchmarking; Gary Bostick (Woodward Governor) for a concise write-up on modern turbomachinery controls; R. Ellis and M. Galley (Dow) for documenting task descriptions used in best-of-class maintenance; Galen Evans (Ludeca) for quantifying the reliability impact of laser-optic alignment issues; S. Gupta and John Paisie (Sun Oil Company) for groundbreaking work on the value-related definition of turnaround scope; Bill Key (Flowserve), W. Schoepplein, and J. Nasowicz (Dichtungswerke Feodor Burgmann), Bill Adams, W. Binning, and R. Phillips (Flowserve), Jim Netzel and P. Shah (John Crane) all of whom contributed lucid material on modern sealing technology; John S. Mitchell for his always authoritative and equally compelling summary of the direction in which maintenance efforts must be channelled in the twenty-first century; L. C. Peng for his contribution on pipe stress issues; Jean Revelt (Lincoln Electric) for neatly explaining important reliability aspects of electric motors; R. Ricketts (Solomon Associates) for shedding considerable light on rigorous benchmarking; and to Paul Smith for his observations on the "knowledge worker" who is certain to be needed to deal with reliability issues from this day on.

Their contributions and those of others whose personal and/or company names are mentioned in footnotes and captions are gratefully acknowledged.

Heinz P. Bloch, P.E.

Introduction

The View of an Advocate for Change*

Machinery reliability management in the process industries can be divided into three phases: equipment selection and pre-erection reliability assurance, preparation for effective startup, and post-startup reliability assurance and maintenance cost reduction. All of these phases are important; they are intertwined and merit equal attention. The techniques and procedures described in this text cover essential elements of each phase; they have been critically examined and have led to substantially improved reliability and maintenance efficiencies. Adoption of applicable techniques and procedures at your plant is certain to result in similar benefits.

In the quest for increased reliability, multiple dimensions must be considered.

The first is whether maximum profitability for a given enterprise and maximum reliability are one and the same. Recent interviews with experienced individuals indicate a growing awareness that business-operating and profit models often change dramatically with time and other factors. The latter include status of product sales (sold out or not sold out), level of inventory, alternative sources, and facility design life. The funds necessary to maintain and improve reliability must fit profitably within the enterprise business model.

Next are the methods and practices that must be established and maintained to assure optimum reliability. Good design and installation practices, improved components and materials, condition-directed maintenance, root cause identification and correction are the subjects dealt with in this book. All are vital and must be addressed. Success demands more than awareness. Acceptance and top-down commitment to optimizing reliability are mandatory.

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*Contributed by John S. Mitchell, San Juan Capistrano, California. Adapted by permission.

Organizational and administrative aspects of every function must be streamlined and optimized. In the reliability area this means bringing maintenance and operations closer together in a supportive partner relationship rather than the common adversarial hierarchical organization.

Finally, information creation and effective communications are essential to measure performance, assure conformance to enterprise objectives and best-of-class benchmarks. Within a typical enterprise there are at least four classifications of information. Senior executives require information such as costs-per-unit output and production availability. At the MRP (manufacturing resource planning) level, long-term prediction of equipment lifetime—the ability to meet contractual obligations—is essential. Operations must have detailed, real-time knowledge of equipment condition and any immediate threats to production. At the detail level, condition assessment, maintenance management, and information systems must function together. Tasks include gathering and managing data, creating and exchanging information as well as directing appropriate information to other levels in the organization. Accomplishing this ambitious, crucial objective requires generically open systems and a common method of communications.

In many industrial enterprises, senior management appears to be growing increasingly aware that maintenance and reliability improvement, or more broadly, lifetime asset management, is the "final frontier" of maximizing profitability. Thus far, most of the focus seems to be on reducing costs by re-engineering the administrative process and eliminating personnel. Requirements for real success include awareness that maintenance and reliability improvement are strategic contributors to income and profitability. Investment to optimize reliability and reduce the need for maintenance is imperative. From a strategic perspective, maintenance cost reduction is a result—not an action.

Optimized practices such as pre-procurement equipment reliability audits, installation reviews, and condition-directed or predictive maintenance have been in use since the 1960s. All have proven highly effective toward improving availability and reducing unexpected failures and costs. Unfortunately, results have not been communicated effectively in financial terms to senior management. As a consequence, many successful condition-directed maintenance programs are being curtailed or, in some cases, terminated altogether as cost cutting measures.

Are arbitrary cost reductions and changes for change sake the way to greater maintenance efficiency? In most cases the answer is no. Arbitrary downsizing, eliminating proven programs such as condition-directed maintenance to end the ongoing operating cost may well have the opposite effect—reduced availability, reduced efficiency, and increased maintenance costs.

The answers are in three areas: value, organization, and information. These issues are addressed in the text.

Reliability improvement and maintenance activities must be reoriented from a cost-centered to a value- or profit-centered mentality. Within a cost-centered framework there are no incentives for improvement. In fact, there are disincentives for improvement! Everyone knows what happens if a maintenance budget is underspent and how those responsible for the achievement are rewarded. "Spend it or lose it" is known to all. As a result, many expenditures occur at year end—some unwise—to make certain budgeted funds are all spent. It would be far better to shift to a value orientation that encourages continuous improvement and rewards increased effectiveness.

Many leading enterprises are shifting to multi-functional team-based organizations. Benefits include single-person accountability for a readily identifiable process or area, pride of ownership, and elimination of counterproductive trade mindsets.

Success with the necessary changes requires enabling technology. Technology includes designing in reliability, designing out maintenance, and implementation of productivity-improving information systems that make the remaining maintenance tasks easier and more efficient. Planning, scheduling, tracking workflow, and providing time, materials, and cost information are vital functions of computerized management and information systems. Technology is indispensable for condition assessment and for clearly conveying equipment status to operators, maintenance, and production planners. Technology also plays a vital role in assembling and communicating planning and performance information, value and benefits to senior executives and financial managers.

There must be an overall vision or concept that unifies individual changes into an optimized whole fabric. Profit-centered maintenance, the first stage in the unification process, establishes value as the prime objective. Value is achieved by maximizing quality, efficiency, and commercial availability while permanently reducing the need for maintenance. Add an optimized organization and crucial information made available at every level of the organization and the result is value-driven asset management[1]—a totally new concept for achieving maximum value from industrial production and manufacturing assets.

Asset Management and the Maintenance Process. Industrial production and manufacturing equipment are the specific assets of interest to reliability and maintenance professionals. Machinery, solids-handling equipment, heat exchangers, valves, etc. are examples. Vital integrity tests and condition-assessment measurements include mechanical and fluid condition, operating efficiency, safety checks, operational and electrical tests, thickness, and cathodic/anodic voltage measurements, temperature profiling (thermography), and leak detection. The preceding examples and others form the basis of asset management.

There is one incontestable law of maintenance: The only way to permanently reduce maintenance cost is to *reduce the need for maintenance.* Examples of this principle are as close as our television. Many of today's automobiles advertise a guaranteed 100,000 miles between tune-ups. How have maintenance requirements been reduced so dramatically—by at least a factor of 20 in the past ten years? The answer is clear: Design for least maintenance. That means better materials, fewer parts, greater attention to lubrication, and extensive use of low-maintenance components such as fuel injection and electronic ignition. Trends in the automobile industry demonstrate that reduced maintenance has real value. There is another law of maintenance familiar to television viewers: maintenance neglected or ignored always reappears—multiplied in cost and effect. "Pay me now or pay me much more later."

Another bit of wisdom bears repeating: Many senior business and financial executives fail to recognize high availability, normal operation, and the absence of problems as direct results of continuing action. Examples include continuing action by conscientious, committed individuals and systematic programs such as cost-justified equipment improvements, condition-directed maintenance, and proactive problem solving. In some cases, successful maintenance programs are curtailed or even terminated because high availability and fewer problem incidents lead to the conclusion that benefits have been largely captured and thus the means employed to gain these results are no longer necessary.

For the past several years, reliability has been the primary focus of maintenance professionals. However, as most are aware, reducing load and the rate of production increases reliability. But is reduced production an option? If not, perhaps reliability is not the final objective. Reliability is a maintenance-oriented objective and the means to an end—but perhaps not the end itself. The ability to operate when required at specified production output and quality while gaining maximum value is an outward objective directed to the success and well-being of the enterprise as a whole. If this conclusion is true, perhaps processes and priorities should be reviewed and reconsidered.

The concept of value leads to equipment selection and maintenance processes that look outward and are results oriented. As this text will demonstrate, overall characteristics include:

- design for reliability and maintainability
- require best-practice installation
- identify and correct root-cause deficiencies
- eliminate chronic problems
- invest for continuing, value-directed, permanent improvements
- create value-oriented measures of performance
- conduct continuing workforce training
- monitor and test to verify condition and assess and measure results
- report value gained to senior management in credible financial terms that compel support
- establish and maintain a process of continuous improvement

All the preceding are directed to gain maximum value. Prioritization and careful consideration of production availability requirements and life cycle cost are mandatory. With this philosophy, sacrificing the future for short-term gain is a relic of past inefficiencies.

Specific examples in the machinery area include:

- select components and materials for an optimum balance between production availability and *life cycle* cost—not just least cost
- properly level and attach baseplates to the foundation; properly align piping
- provide easy access to components requiring maintenance; avoid interfering with efficient removal and reinstallation
- assure quality lubrication, exclude contaminants, perhaps by utilizing oil mist

- maximize condition directed maintenance
- eliminate unnecessary scheduled maintenance
- employ situational use of operate-to-failure
- employ precision balancing and alignment; include comprehensive alignment training
- eliminate overrepairs
- require post-repair quality assurance such as motor testing under load
- increase common spare parts, e.g., impellers, shafts, bearings, couplings

The change in philosophy and perspective necessary to gain maximum value from maintenance must be driven from the very top of an enterprise. And this likely requires education. Education for plant management, senior corporate, and financial management is necessary to illuminate the potential and benefits to be derived from optimized maintenance. Education should focus on ways to build the compelling vision of how equipment-asset management can and must contribute to twenty-first century success.

Profit-centered maintenance[2] is a name given to a concept constructed around creating value. Profit-centered maintenance has the following attributes:

- a mindset, not an accounting method
- oriented to create greatest value—not least cost
- addresses physical, administrative, and organizational processes
- uses enabling technology to the fullest
- addresses the relationship between maintenance and other functional areas

Profit-centered maintenance is a continuing, regenerating process. It includes a combination of life cycle optimized, value-oriented design, root-cause identification and correction, proactive, condition-directed, scheduled and reactive maintenance, and streamlined administration. All are assembled to create maximum value and operating profitability.

There are numerous benefits to be gained by adopting profit-centered maintenance. The central benefit is the concept of value that drives the entire process to new levels of maintenance effectiveness.

Implementing profit-centered maintenance requires a long-term, life cycle perspective and a commitment and willingness to invest for the identification and root-cause correction of deficiencies. These are aimed toward life extension and reduced life cycle costs *by permanently decreasing the need for and cost of maintenance.*[3] At each step along the way it is imperative to formulate and communicate credible financial justification and results of profit-centered maintenance to senior executives and financial management.

Condition assessment, condition-directed maintenance, and proactive problem avoidance are vital elements of profit-centered maintenance. Much has been written about condition assessment and condition-directed maintenance; only the crucial value or profit-centered principles will be repeated here. These include:

- The mix of maintenance types, e.g., reactive, preventive and condition-directed, is determined by value considerations. Some machines are most profitable run-to-failure. Others require full condition assessment and condition-directed maintenance for highest profitability. For most, greatest profit requires a balanced mixture of preventive and condition-directed maintenance.
- Detailed, diagnostic mechanical condition assessment from vibration and fluid (lubricating and hydraulic oil) characteristics, electromechanical condition from static and dynamic electric (current) characteristics, internal wear, buildup and erosion from operating performance and efficiency and external conditions (leaks, loose fittings, unusual sound and smell) are all necessary for a complete, accurate picture of condition.
- Method of condition assessment, type and number of measurements and interval between successive measurements must be based on several factors. These include optimum methods to assess condition and provide earliest detection of change, current condition, rate of change, and the probability and consequences of failure. Anticipated time between discovery of a potential failure and requirement for corrective action as well as operating/production requirements and value are other important considerations.

- Condition assessment must be directed toward increasing commercial availability and reducing operating costs. Measures such as number of machines/points monitored and average vibration level are irrelevant and inconsistent with the principles of profit-centered maintenance.

Full, enthusiastic support from Operations/Production is a mandatory aspect of profit-centered maintenance. Contrasted with re-engineering, a one time radical change to organizational and administrative processes, profit-centered maintenance addresses all elements of the maintenance process: equipment, organizational, and administrative and is a process of continuous change. It is value-centered and has a broader scope than condition-directed maintenance.[4] Profit-centered maintenance demands proactive, preemptive maintenance, and requires continuous evaluation, refinement, and improvement.

Profit-centered maintenance is a powerful statement of commitment to the principal corporate objective. It is focused on outward, customer-oriented results. Profit-centered maintenance leads to optimum decisions and permanent solutions that maximize profitability.

Organization. Many leading visionaries are shifting to team-based, multifunctional organizational structures with overall responsibility for a logical, readily identifiable process or area.[5] This concept has numerous advantages over traditional "silo" organizations based on functional work groups. Characteristics of the team organization include single-person, end-to-end responsibility—including logistics and quality—and assignment of all skills necessary for normal operation. Benefits include greater awareness, ownership, involvement, responsibility, and improved teamwork. Counterproductive bickering and operating costs are diminished. Specialized, high skill and safety-related tasks and training remain within a central support organization. Examples include high voltage electrical testing, in-shop machine work, specialized repairs and training in areas such as instrument calibration, pump seal replacement and shaft alignment. As an example of how this concept is implemented, one facility allows cross-trained mechanics to tag out, disconnect, and remove a motor for repairs. Electrically reconnecting the motor requires a qualified electrician.

Shifting to this new form of organization requires a willingness and commitment to empower personnel and relinquish traditional hierarchical control.

Ingredients for success include:

- incentives that are entirely based on team success
- encouraging ownership, initiative, and responsibility; demanding accountability
- developing, encouraging, and supporting extraordinary performers
- promoting competency; penalizing mediocrity
- substituting initiative and flexibility for inefficient, counterproductive trade-and-craft mindsets

Some facilities may find the transition to a full-team organization too formidable to accomplish in a single step. Several in this position have used other measures to achieve comparable results. A number of companies reported outstanding results and greatly improved cooperation by simply having maintenance and operations/production managers exchange positions. One site found that transferring responsibility for the maintenance budget to operations proved exceptionally valuable toward establishing a cooperative relationship.

Recognize that massive change to a team-based organization may be opposed, both overtly and covertly by people who depend upon and feel more comfortable within a less challenging, traditional hierarchical organization. Recognize also that unless change of this scale is presented and implemented with a great deal of sensitivity, the very people needed most for success can be lost.

Information Systems.[6] These systems glue the fabric together. In production and manufacturing enterprises at least five layers of operating information can be identified—each with specific requirements:

- Information describing the current and projected condition of production equipment is vital. The threat and timing of potential problems and components affected must be communicated to production and maintenance planning. Recommended corrective action for optimized work scheduling and planning are vital elements of the asset management process. Requirements and completion records for safety and operational tests, equipment condition assessment, calendar or time-based maintenance, integrity tests for components such as safety and relief valves, corrosion thickness, cathodic and anodic protection voltage measurements and reliability records are examples of other information that must be readily available.

- Work process information is also quite necessary for optimized, value-directed maintenance. Some facilities have developed complete, verified instructions for every maintenance task. These instructions include safety procedures and precautions, parts and tools required, a step-by-step procedure to accomplish the task and unique task-specific considerations. In addition to greatly improving productivity, detailed instructions substantially reduce mistakes that result in post-repair failures.
- Supplying refined condition-assessment results to operations for display on the process control system is another much needed improvement. Better, more informative displays create a greater awareness of condition and the operating variables that influence condition. Confidence to initiate action if difficulties arise and the ability to contribute observations pertaining to variations are added value gained by ready access to easily interpreted condition information.
- Refined, easily interpreted, actionable information to operations/production planning and maintenance management systems is equally important. Contrasted with operators, these users need predictive condition-assessment information for medium and long-term planning. Will production assets be available to meet future contractual commitments? The information required includes equipment status, problem identification, classification, severity and rate of change, components affected, time to required action and recommendations for both operating and repair actions.[9] The timing and length of an optimized production outage, spare parts and personnel requirements are constructed from this information.
- Benchmark measures such as mean-time-between-repair (MTBR) and availability are valuable management information. Information required for executive and financial management includes cost-per-unit output, return on assets, life cycle costs, and operating profitability. These, and other measures, are needed to measure effectiveness, convey value, and justify the ongoing cost of the processes and people creating value. If performance measures trend opposite to require-

ments, information must be available to identify whether specific or enterprise-wide solutions must be implemented.

The ability to assemble vital management reports automatically is a crucial requirement of an information system. An expert should examine crucial information prior to transmittal and have the opportunity to add interpretation and editorial comments. However, the expert should not have to perform manual data gathering. The days when time was available to compile information from multiple sources and correlate it manually for management reports is long past. Reports must be self-generating, or at least all information must be available for report generation.

The traditional method of displaying measurements must be improved. Begin with a definition of information—it must be understood. Most process control professionals state that information conveyed to operators should be limited to that requiring action within a relatively short time period, typically a shift. Most also agree that conveying long-term threats and too much description to operators, for example, outer race bearing failure probable within a week or month, is irrelevant, distracting, and potentially counterproductive. A case can be made that differentiation between specific failure types is relevant to an operator only if it affects action required. Too much long-term information may create an indifference that ultimately results in missing a real requirement for immediate action.

Information must be displayed in a clear, understandable fashion. Individual measurements, such as vibration expressed in engineering units, and even measurement-versus-time trends do not meet requirements if too much skilled interpretation is required. Expert decision advisory systems will occupy a vastly expanded and important function in the data-to-information-conversion process. Expressing complex data as a single measure of machine life remaining, or as a normalized condition index has been suggested.[7] Information displayed in a friendly form such as a smiling face has far greater value and impact than values and even trends. Not surprisingly, there is a scientific basis for smiling faces and other imagery used to translate complex multidimensional measurements into an easily interpretable form.[8]

Information Exchange. This is a vital issue. There are three basic alternatives for information exchange. Many large corporations implement single-supplier, facility-wide information systems that include accounting, financial, personnel, operations, and maintenance. Others accomplish a custom integration to connect information components and practices currently in use within the enterprise. A third alternative employs self-integrating open systems that enable an enterprise to pick components that are best for their specific application with assurance of interoperability.

The single-supplier structure has the advantage of defined accountability. Disadvantages include total reliance on a single supplier and the difficulty of duplicating and maintaining levels of excellence equivalent to specialized, applications-specific information components within a single source system. The crucial question is whether an information system designed primarily for one purpose can be efficiently extended to accommodate the facilities and rich detail necessary to gain maximum value in specialized areas such as condition assessment, lifetime prediction, and condition-directed maintenance. And if not, how are these vital tasks incorporated into the overall architecture?

Integrating information components that are in use within an enterprise has the advantages of familiarity and presumably adequate performance for the tasks. Disadvantages include the hazards of institutionalizing current practice, which may not be best practice, and the high-cost, one-time, specialized nature of component integration. The necessity to redo all or part of a system integration in order to gain the benefits of advances in experience and technology is another potentially costly disadvantage.

Self integration, the so called "plug-and-play" open system has numerous advantages for enterprise information systems. Users can select components that are best for their specific application with assurance of full information exchange. System components and information can migrate to best-practice improvements at least cost as experience and technology increase. The personal computer model is instructive. Low-cost word processing, ready exchange of information between applications, and the current proliferation of CD-ROMs for multiple uses would not have occurred without a standard platform and open information exchange conventions.

Some areas, notably process control, are moving quickly in the direction of fully open systems. However, for asset management and maintenance the open systems solution has not yet developed. Why is this? Many maintenance professionals believe the challenge is insurmountable. Others believe there cannot be any departures from current work processes that may be unique to a single site. Suppliers may believe that maintaining absolute control over their portion of the information structure and all gateways in and out is to their commercial advantage. Many of these arguments appeared when process control transitioned from analog to digital systems twenty or so years ago. Ultimately, control system purchasers realized that open systems were the only way to achieve maximum performance at an affordable cost. Asset management and maintenance are not well served by going through the same process.

There is movement toward open systems in the maintenance and asset management areas. The activity accomplishing this valuable objective is called MIMOSA, the Machinery Information Management Open Systems Alliance.[10, 11]

In summary, gaining maximum value from process, production, and manufacturing equipment requires a comprehensive, value-oriented process that begins at design and extends through operation. Vital ingredients include continuing, well-planned machinery reliability enhancement, maintenance optimization, and life-cycle cost justification. Within this process, maintenance must be directed toward eliminating problems and safely reducing the need for maintenance. Perceptions must change. Improving equipment reliability at the very inception of a project, demanding quality during installation, and focusing on lifetime equipment management must be accepted and applied. Now is the time to drive the change process to your advantage.[12] The future is not very promising for enterprises that are significantly below competitive best. This completely revised edition focuses on some of the most important and highest return-on-investment methods, work processes, and techniques for your move toward competitive best. This book starts by showing the reader how to gain maximum value from manufacturing equipment. Not a bad place to start!

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Chapter 1

Requirements Specification

Long before machinery specifications can be prepared, the plant designers perform scoping studies encompassing a large number of options. Process and economic considerations are weighed, compared, and analyzed. Design philosophies are debated at the highest levels of management, business forecasts are studied, and thousands of questions are asked and answered before the machinery engineer is given his first opportunity to prepare an inquiry document for major machinery or detailed purchase specification packages for all the machinery in a process plant.

To the superficial observer, the job of specifying machinery would seem rather routine. But an experienced engineer knows that this is far from true. Ticking off a few check marks on a form sheet may define the extent of supply, but it certainly cannot pass as an adequate specification for major machinery. Then again, excessively bulky specifications may have the effect of frightening the bidder into adding significant extra charges for potential oversights, and badly worded specifications may prompt cost escalation to cover potential misunderstandings. At times a combination of bulk, cross referencing of many specification documents, and wording subject to misinterpretation has motivated vendors to decline to bid. As if this were not bad enough, an ill-conceived specification may burden process plants with the perennial "bad actor"—a piece of maintenance-intensive machinery not bad enough to replace with something new or different, but bad enough to drive up maintenance costs, sap maintenance manpower, and cause feelings of resignation or demotivation in personnel.

A good specification, therefore, is concise and precise. It will have to define your requirements in clear, understandable form. Yet it should encourage the vendor to offer more than the bare minimum requirements. Your next process plant should benefit from advances in the state-of-the-art of which the vendor may have knowledge if your joint, conscientious screening efforts can certify these changes to be safe, economic, and not prove to introduce downtime risks.

Industry Standards Available for Major Machinery in Process Plants

Table 1-1 represents a listing of presently available API (American Petroleum Institute) standards. These specifications were developed by panels of user engineers to define petrochemical process plant machinery in a professional fashion. Wherever possible, API standards should become the focal point document in machinery specifications for process plants.

Table 1-1
Principal API Standards for Mechanical Equipment (1997)

Standard	7B-11C	Specifications for Internal-Combustion Reciprocating Engines for Oil Field Service
Standard	541	Squirrel Cage Induction Motors—250 HP and Larger
Standard	546	Brushless Synchronous Machines—500kVA and Larger
Standard	610	Centrifugal Pumps for General Refinery Services
Standard	611	General-Purpose Steam Turbines for Refinery Services
Standard	612	Special-Purpose Steam Turbines for Refinery Services
Standard	613	Special-Purpose Gear Units for Refinery Services
Standard	614	Lubrication, Shaft-Sealing, and Control Oil Systems for Special-Purpose Applications
Standard	616	Gas Turbines for Refinery Services
Standard	617	Centrifugal Compressors for Petroleum, Chemical, and Gas Service Industries
Standard	618	Reciprocating Compressors for Petroleum, Chemical, and Gas Service Industries
Standard	619	Rotary-Type Positive Displacement Compressors for General Refinery Services
Standard	670	Vibration, Axial Position and Bearing-Temperature Monitoring Systems
Standard	671	Special-Purpose Couplings for Refinery Services
Standard	672	Packaged, Integrally Geared, Centrifugal Plant and Industrial Air Compressors for Petroleum, Chemical, and Gas Service Industries
Standard	674	Positive Displacement Pumps—Reciprocating
Standard	675	Positive Displacement Pumps—Controlled Volume
Standard	676	Positive Displacement Pumps—Rotary
Standard	677	General Purpose Gear Units for Refinery Services
Standard	678	Accelerometer-Based Vibration Monitoring System
Standard	681	Liquid Ring Vacuum Pumps and Compressors for Petroleum, Chemical, and Gas Industries Services
Standard	682	Shaft Sealing Systems for Centrifugal and Rotary Pumps
Standard	683	Rotor Dynamics and Balancing
Standard	686	Machinery Installation and Installation Design

Nevertheless, major U.S. petrochemical plants or contractors will rarely opt to use applicable API standards as the *only* procurement standard for machinery. Either the contractor or the plant owner, or both, makes use of specification supplements or design and construction standards reflecting particular experience, special needs, regional requirements, design and maintenance philosophies, and the like. All of these supplements serve a specific purpose. Compliance with government regulations, uniformity of training for operators or mechanics, common spare parts utilization among affiliated plants, utilization of locally available components, and a host of other purposes could be mentioned. The relative importance of these factors will change from plant to plant and from location to location. Unfortunately, so does the specification format—to the detriment of all parties involved.

How to Deal with the Typical API Data Sheet

API Standard 617, "Centrifugal Compressors for General Refinery Services,"* is the primary reference document for procuring a given centrifugal compressor. It con-

*API Standard 617, "Centrifugal Compressors for General Refinery Services," Sixth Edition, 1995. (*Courtesy of the American Petroleum Institute.*)

tains nine data sheets that are to be filled in by the purchaser—usually a major contracting or design firm acting on the owner's behalf—to the extent necessary to define the plant or process requirements. All remaining data must be provided by the vendor. Five of these data sheets merit closer examination.

Our review of the API centrifugal compressor data sheets will be representative of the structured approach recommended for all API-based machinery. It will be restricted to those items having an impact on successful startup, operation, reliability, and maintainability of centrifugal compressors.

On page 1 of the data sheets (Figure 1-1), we have drawn attention to the check point "as built." (The relevance of "as built" data is further highlighted in Figures 1-27 and 1-30.) Dimensional records are so important to turnaround maintenance, emergency repairs, and general troubleshooting that submission of these data should be made a contractual condition of sale.

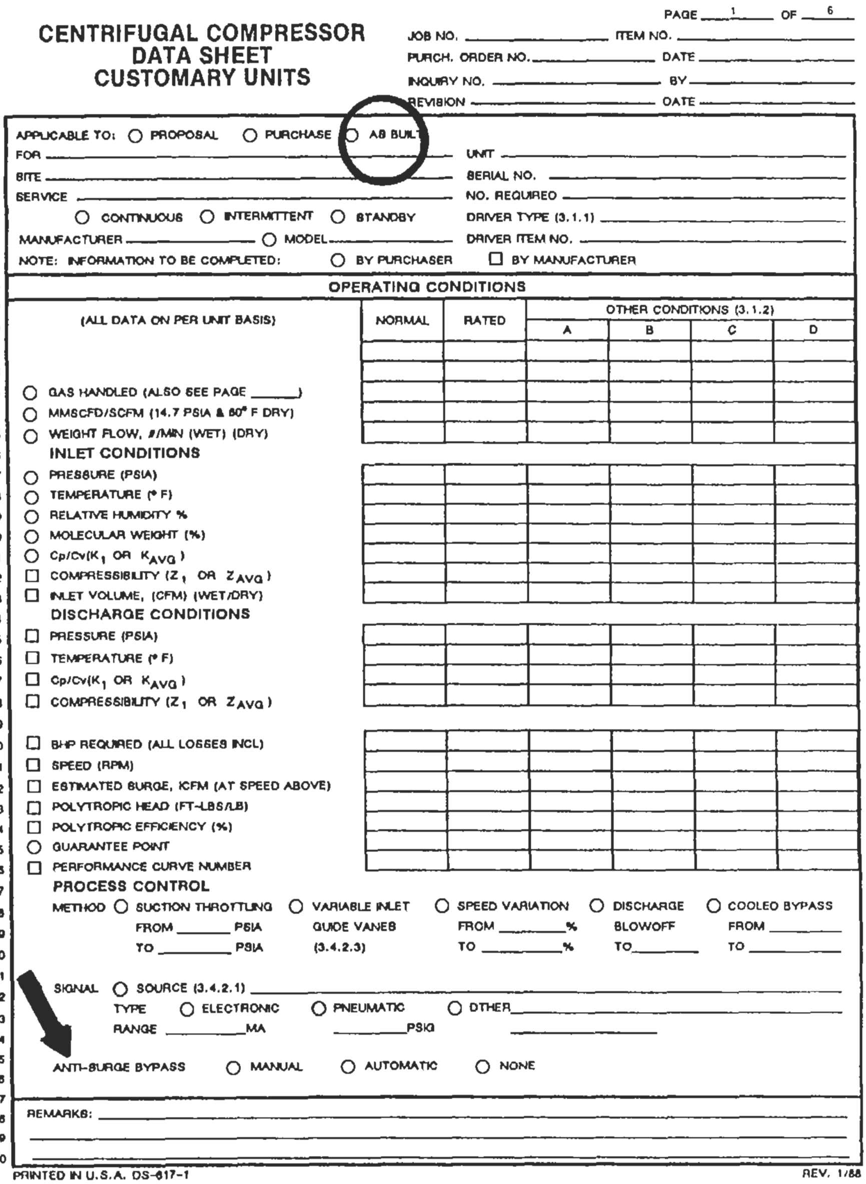
On the same page, we note the item "antisurge bypass." We highlight this item because proper and adequate surge-protection devices must be precisely specified in a narrative supplement. Although surge control via sophisticated process computers is possible, investigating commercially available, proven hardware devices is strongly recommended. These devices can do a creditable job as stand-alone devices or as analog emergency controls in case the digital process computer sampling rate, response time, or application should prove inadequate. Additional details are given in Chapter 2, "Applying and Reviewing Machinery Reliability Improvements Derived from Modern Electronics."

Data sheet page 2 (Figure 1-2) deals with, among other parameters, "other conditions." It would be prudent to give thought to a reasonable spectrum of alternative operating conditions to define the safe operating windows for centrifugal compressors from such points of view as performance curves, surge limits, efficiency, power demand, polymer fouling, etc. If the vendor is asked to generate all applicable performance predictions before issuance of a purchase order, the user will not suffer any unexpected setbacks later. Moreover, up-front costs are generally only a fraction of the cost of later analyses performed after the purchase order has been issued or the machine has been delivered.

Data sheet page 2 also points to the item "mezzanine." Mezzanine installations are largely motivated by the desire to use downward-oriented compressor nozzles only. This greatly facilitates compressor maintenance, since it permits horizontally split compressor top casing halves to be lifted without disturbing the piping.

The item "acoustic housing" is of interest for other reasons. Intended primarily for rotary-screw and high-speed centrifugal compressors, acoustic housings are generally constructed to direct the sound upward. Unfortunately, the reduction in noise pollution is often outweighed by such factors as decreased operator surveillance and housekeeping problems resulting from a reluctance to get near the "screamer," let alone get inside the enclosure. Alternative solutions are often available and should be explored.

(text continued on page 6)



PAGE 1 OF 6

CENTRIFUGAL COMPRESSOR DATA SHEET CUSTOMARY UNITS

JOB NO. ______ ITEM NO. ______
PURCH. ORDER NO. ______ DATE ______
INQUIRY NO. ______ BY ______
REVISION ______ DATE ______

APPLICABLE TO: ○ PROPOSAL ○ PURCHASE ○ AS BUILT
FOR ______ UNIT ______
SITE ______ SERIAL NO. ______
SERVICE ______ NO. REQUIRED ______
○ CONTINUOUS ○ INTERMITTENT ○ STANDBY DRIVER TYPE (3.1.1) ______
MANUFACTURER ______ ○ MODEL ______ DRIVER ITEM NO. ______
NOTE: INFORMATION TO BE COMPLETED: ○ BY PURCHASER □ BY MANUFACTURER

OPERATING CONDITIONS

(ALL DATA ON PER UNIT BASIS)	NORMAL	RATED	OTHER CONDITIONS (3.1.2) A	B	C	D
○ GAS HANDLED (ALSO SEE PAGE ____)						
○ MMSCFD/SCFM (14.7 PSIA & 60° F DRY)						
○ WEIGHT FLOW, #/MIN (WET) (DRY)						
INLET CONDITIONS						
○ PRESSURE (PSIA)						
○ TEMPERATURE (° F)						
○ RELATIVE HUMIDITY %						
○ MOLECULAR WEIGHT (%)						
○ Cp/Cv(K_1 OR K_{AVG})						
□ COMPRESSIBILITY (Z_1 OR Z_{AVG})						
□ INLET VOLUME, (CFM) (WET/DRY)						
DISCHARGE CONDITIONS						
□ PRESSURE (PSIA)						
□ TEMPERATURE (° F)						
□ Cp/Cv(K_1 OR K_{AVG})						
□ COMPRESSIBILITY (Z_1 OR Z_{AVG})						
□ BHP REQUIRED (ALL LOSSES INCL)						
□ SPEED (RPM)						
□ ESTIMATED SURGE, ICFM (AT SPEED ABOVE)						
□ POLYTROPIC HEAD (FT-LBS/LB)						
□ POLYTROPIC EFFICIENCY (%)						
○ GUARANTEE POINT						
□ PERFORMANCE CURVE NUMBER						

PROCESS CONTROL

METHOD ○ SUCTION THROTTLING FROM ____ PSIA TO ____ PSIA ○ VARIABLE INLET GUIDE VANES (3.4.2.3) ○ SPEED VARIATION FROM ____ % TO ____ % ○ DISCHARGE BLOWOFF TO ____ ○ COOLED BYPASS FROM ____ TO ____

SIGNAL ○ SOURCE (3.4.2.1) ______
TYPE ○ ELECTRONIC ○ PNEUMATIC ○ OTHER ______
RANGE ____ MA ____ PSIG ______

ANTI-SURGE BYPASS ○ MANUAL ○ AUTOMATIC ○ NONE

REMARKS: ______

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Figure 1-1. API 617, data sheet page 1. (*Courtesy American Petroleum Institute.*)

CENTRIFUGAL COMPRESSOR
DATA SHEET
CUSTOMARY UNITS

PAGE 2 OF 6
JOB NO. ______ ITEM NO. ______
REVISION ______ DATE ______
BY ______

OPERATING CONDITIONS (Continued) (3.1.2) (3.1.3)

GAS ANALYSIS: ○ MOL% ○		NORMAL	RATED	OTHER CONDITIONS				REMARKS: (2.11.1.3) (2.11.1.6)
				A	B	C	D	
	MW							
AIR	28.966							
OXYGEN	32.000							
NITROGEN	28.016							
WATER VAPOR	18.016							
CARBON MONOXIDE	28.010							
CARBON DIOXIDE	44.010							
HYDROGEN SULFIDE	34.076							(2.11.1.7)
HYDROGEN	2.016							
METHANE	16.042							
ETHYLENE	28.052							
ETHANE	30.068							
PROPYLENE	42.078							
PROPANE	44.094							
I-BUTANE	58.120							
n-BUTANE	58.120							
I-PENTANE	72.146							
n-PENTANE	72.146							
HEXANE PLUS								
TOTAL								
AVG. MOL. WT.								

LOCATION: (2.1.8)
○ INDOOR ○ HEATED ○ UNDER ROOF
○ OUTDOOR ○ UNHEATED ○ PARTIAL SIDES
○ GRADE ○ MEZZANINE ○ ______
○ ELEC. AREA CLASSIFICATION (2.1.14) CL ___ GR ___ DIV ___
○ WINTERIZATION REQD. (2.1.8) ○ TROPICALIZATION REQD. (3.4.6.6)

SITE DATA:
○ ELEVATION ______ FT BAROMETER ______ PSIA
○ RANGE OF AMBIENT TEMPS:

	DRY BULB	WET BULB
SITE RATED °F	______	______
NORMAL °F	______	______
MAXIMUM °F	______	______
MINIMUM °F	______	______

UNUSUAL CONDITIONS: ○ DUST ○ FUMES
○ OTHER (2.11.1.6) ______

REMARKS: ______

NOISE SPECIFICATIONS: (2.1.9)
○ APPLICABLE TO MACHINE:
SEE SPECIFICATION ______
○ APPLICABLE TO NEIGHBORHOOD:
SEE SPECIFICATION ______
ACOUSTIC HOUSING: ○ YES ○ NO

APPLICABLE SPECIFICATIONS:
API 617, CENTRIFUGAL COMPR. FOR GEN. REFINERY SERV.
○ VENDOR HAVING UNIT RESPONSIBILITY (2.9.1.7)
______
○ GOVERNING SPECIFICATION (IF DIFFERENT)
______

PAINTING:
○ MANUFACTURER'S STD.
○ OTHER ______

SHIPMENT: (4.4.1)
○ DOMESTIC ○ EXPORT ○ EXPORT BOXING REQD.
○ OUTDOOR STORAGE MORE THAN 6 MONTHS (4.4.1)
SPARE ROTOR ASSEMBLY PACKAGED FOR (4.4.3.10)
○ HORIZONTAL STORAGE ○ VERTICAL STORAGE

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Figure 1-2. API 617, data sheet page 2. (*Courtesy American Petroleum Institute.*)

(text continued from page 3)

On data sheet page 3 (Figure 1-3), the specifying engineer must determine whether the critical speed projections made by the vendor are based on proven analytical techniques. The purchaser would be well advised to acquire, also, an understanding of rotor sensitivity. How serious will be the response to rotor unbalance when there is midspan unbalance? What is the vibration response to coupling unbalance? Should the user engage a consultant to perform an independent study and submit a formal report on the findings?

The reader may be interested to know that from 1960 to 1980, the hot topic was when to use "at-speed" balancing. Fortunately, the increased availability and cost-effectiveness of modern balancing machinery and vacuum bunkers facilitates today's reliability professional's decision to use such methods. In recent times, a debate began on how to define whether a given rotor was "rigid" or "flexible." By ISO Standard 1925, a rotor is rigid if:

1. It could be corrected in any two arbitrarily selected planes and
2. After that correction, its unbalance did not significantly exceed the balancing tolerances—relative to the shaft axis—at any speed up to the maximum service speed. These conditions approximated very closely the final supporting system.

In laymen's terms, the rotor is rigid if its first lateral critical was above the maximum operating speed.

Thus, a rotor could be called rigid for one application (if it had a low service speed and/or liberal balancing tolerance); whereas, for another operation—demanding a higher speed and/or finer tolerance—it became flexible. By definition, a flexible rotor can operate above its first lateral critical speed.

Flexible rotors were primary candidates for high-speed balancing, assuming that the purchaser would pay the extra cost. However, high-speed balancing can be cost-effective. Assisted by Schenk-Trebel, a world-class manufacturer of balancing machinery, major machinery-repair and manufacturing facilities are pursuing self-sufficiency by acquiring at-speed balancing facilities. For example, Hickham Industries, Inc., LaPorte/Houston, Texas, began operating an "at-speed" facility in August 1996. Figure 1-4 illustrates the facility's huge size.

Turbomachinery rotors are installed and removed by an overhead crane. This method is safer, faster and enables more efficient floor space usage. The balancing bunker is a vacuum chamber at one millibar of vacuum, with a pump-down time of 15 minutes. It accepts rotors up to 280 in. long by 96 in. diameter, and capacities up to 50,000 lbs. These rotors can be spun at 16,000 RPM; whereas, rotors in the 2,750 lb.-league can be spun at 40,000 RPM.

CENTRIFUGAL COMPRESSOR DATA SHEET CUSTOMARY UNITS

PAGE 3 OF 6

JOB NO. ____ ITEM NO. ____

REVISION ____ DATE ____

BY ____

CONSTRUCTION FEATURES

☐ **SPEEDS:**

MAX. CONT. ____ RPM TRIP ____ RPM

MAX. TIP SPEEDS: ____ FPS @ RATED SPEED

____ FPS @ MAX. CONT. SPEED

☐ **LATERAL CRITICAL SPEEDS (DAMPED)**

FIRST CRITICAL ____ RPM ____ MODE

SECOND CRITICAL ____ RPM ____ MODE

THIRD CRITICAL ____ RPM ____ MODE

FOURTH CRITICAL ____ RPM ____ MODE

○ TRAIN LATERAL ANALYSIS REQUIRED (2.9.2.3)

○ UNDAMPED STIFFNESS MAP REQUIRED (2.9.2.4a)

○ TRAIN TORSIONAL ANALYSIS REQUIRED (TURBINE DRIVEN TRAIN) (2.9.4.5)

☐ **TORSIONAL CRITICAL SPEEDS:**

FIRST CRITICAL ____ RPM

SECOND CRITICAL ____ RPM

THIRD CRITICAL ____ RPM

FOURTH CRITICAL ____ RPM

☐ **VIBRATION:**

ALLOWABLE TEST LEVEL ____ MILS (PEAK TO PEAK)

☐ **ROTATION, VIEWED FROM DRIVEN END**

○ **MATERIALS INSPECTION REQUIREMENTS (4.2.3)**

○ SPECIAL CHARPY TESTING (2.11.3)

○ RADIOGRAPHY REQUIRED FOR ____

○ MAGNETIC PARTICLE REQUIRED FOR ____

○ LIQUID PENETRANT REQUIRED FOR ____

☐ **CASING:**

MODEL ____

CASING SPLIT ____

MATERIAL ____

THICKNESS (IN.) ____ CORR. ALLOW. (IN.) ____

MAX. WORKING PRESS ____ PSIG

MAX DESIGN PRESS ____ PSIG

TEST PRESS (PSIG): HELIUM ____ HYDRO ____

MAX. OPER. TEMP. ____ °F MIN. OPER. TEMP. ____ °F

MAX. NO. OF IMPELLERS FOR CASING ____

MAX. CASING CAPACITY (ICFM) ____

RADIOGRAPH QUALITY ○ YES ○ NO

CASING SPLIT SEALING ____

○ SYSTEM RELIEF VALVE SET PT. (2.2.3) ____ PSIG

☐ **DIAPHRAGMS:**

MATERIAL ____

☐ **IMPELLERS:**

NO. ____ DIAMETERS ____

NO. VANES EA. IMPELLER ____

TYPE (OPEN, ENCLOSED, ETC.) ____

TYPE FABRICATION ____

MATERIAL ____

MAX. YIELD STRENGTH (PSI) ____

BRINNEL HARDNESS: MAX ____ MIN. ____

SMALLEST TIP INTERNAL WIDTH (IN.) ____

MAX. MACH. NO. @ IMPELLER EYE ____

MAX. IMPELLER HEAD @ RATED SPD (FT-LBS/LB) ____

☐ **SHAFT:**

MATERIAL ____

DIA @ IMPELLERS (IN) ____ DIA @ COUPLING (IN.) ____

SHAFT END: TAPERED CYLINDRICAL

MAX. YIELD STRENGTH (PSI) ____

SHAFT HARDNESS (BNH) (Rc) ____

STRESS AT COUPLING (PSI) ____

☐ **BALANCE PISTON:**

MATERIAL ____ AREA ____ (IN^2)

FIXATION METHOD ____

☐ **SHAFT SLEEVES (2.8.2):**

AT INTERSTG. CLOSE CLEARANCE POINTS MATL ____

AT SHAFT SEALS ____ MATL ____

☐ **LABYRINTHS:**

INTERSTAGE

TYPE ____ MATERIAL ____

BALANCE PISTON

TYPE ____ MATERIAL ____

SHAFT SEALS:

○ SEAL TYPE (2.8.3) ____

○ SETTLING OUT PRESSURE (PSIG) ____

○ SPECIAL OPERATION (2.8.1) ____

○ SUPPLEMENTAL DEVICE REQUIRED FOR CONTACT SEALS (2.8.3.2) TYPE ____

○ BUFFER GAS SYSTEM REQUIRED (2.8.7)

○ TYPE BUFFER GAS ____

○ BUFFER GAS CONTROL SYSTEM SCHEMATIC BY VENDOR

○ PRESSURIZING GAS FOR SUBATMOSPHERIC SEALS (2.8.8)

☐ TYPE SEAL ____

☐ INNER OIL LEAKAGE GUAR. (GAL/DAY/SEAL) ____

☐ BUFFER GAS REQUIRED FOR:

☐ AIR RUN-IN ☐ OTHER ____

☐ BUFFER GAS FLOW (PER SEAL):

NORM: ____ LBS/MIN @ ____ PSI Δ P ____

MAX.: ____ LBS/MIN @ ____ PSI Δ P ____

☐ **BEARING HOUSING CONSTRUCTION:**

TYPE (SEPARATE, INTEGRAL) ____ SPLIT ____

MATERIAL ____

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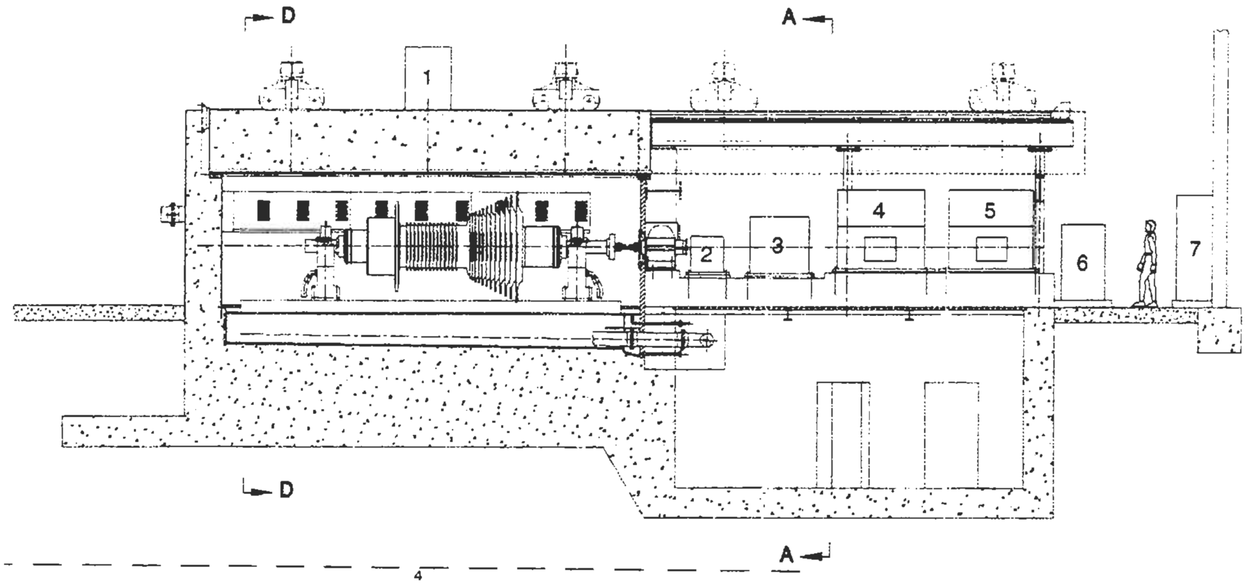
Figure 1-3. API 617, data sheet page 3. (*Courtesy American Petroleum Institute.*)



LEGEND

Pit Cover Hydraulic Power Unit 1
High Speed Gear 2
Main Gear Unit 3
DC Motor 4
DC Motor 5
Pedestal Flow and Jacking 6
SCR Controller 7

Figure 1-4. At-speed balance installation at Hickham Industries, LaPorte/Houston, Texas.

Using a state-of-the-art balancing facility for "at-speed" evaluations has these operational benefits:

- Complete confidence that all rotor shaft deflections reach only minimal amplitudes throughout the entire operating speed range
- Improved rotor reliability
- Full assurance in a smooth running rotor through its full-speed range
- Increased bearing and seal lifetime
- Extended operational life between scheduled maintenance turnarounds.

Needless to say, state-of-art "at-speed" balancing facilities can define rotor unbalance response with utmost precision and can achieve balance qualities that we could only dream of a few decades ago.

Returning to our data sheet topics, we note the entry "diaphragms." Compressor diaphragms are generally made of cast iron. With compressors becoming larger and larger, obtaining sound cast-iron diaphragms is becoming progressively more difficult. Uneven cooling of very large diaphragm castings can set up intolerably high residual stresses. A thorough experience check is needed.

The last arrow on data sheet page 3 points at the item "air run-in." Here, the owner's representative will have to ask himself whether speed, driver horsepower, discharge temperatures, and piping arrangement lend themselves to run-in on air. Will it be necessary to run-in on helium? What questions need to be resolved for a helium run? Cost? Availability? Leakage losses?

Data sheet page 4 (Figure 1-5) shows an arrow pointing at babbitt thickness. Certain babbitt types are stronger than others and permit higher loadings at the expense of being less forgiving if dirt particles should enter the bearing. Conversely, the softer babbitt may have less tolerance to high vibration or surge loading, but will pass slightly larger dirt particles without undue risk. Consequently, the babbitt type should be determined.

An up-to-date reliability professional may, at this point, explore the applicability of, and vendor experience with, flexure pivot™ and magnetic bearings. Flexure pivot bearings are produced by KMC, Inc., in West Greenwich, Rhode Island, and Bearings Plus, Inc., in Houston, Texas. Their development was prompted by the fact that with conventional tilting pad bearings, the high contact stresses between the pads and bearing shell can cause brinelling at the pivot location. This pivot wear increases the bearing clearance and reduces the bearing preload, thus altering the operating characteristics and increasing the susceptibility to vibration problems. Unloaded conventional tilting pads can also experience damage due to pad flutter. Flexure pivot radial pads are integral to the bearing shell and therefore experience no pivot wear. The relatively low rotational stiffness in the support webs is sufficient to eliminate pad flutter in the unloaded pads.

CENTRIFUGAL COMPRESSOR DATA SHEET CUSTOMARY UNITS

PAGE 4 OF 6
JOB NO. ______ ITEM NO. ______
REVISION ______ DATE ______
BY ______

CONSTRUCTION FEATURES (CONTINUED)

BEARINGS AND BEARING HOUSINGS

RADIAL	INLET	EXHAUST	THRUST	ACTIVE	INACTIVE
☐ TYPE			☐ TYPE		
☐ MANUFACTURER			☐ MANUFACTURER		
☐ LENGTH (IN)			☐ UNIT LOADING (MAX PSI)		
☐ SHAFT DIA (IN)			☐ UNIT LOAD (ULT.) (PSI)		
☐ UNIT LOAD (ACT/ALLOW)			☐ AREA (IN²)		
☐ BASE MATERIAL			☐ NO. PADS		
☐ BABBITT THICKNESS (IN)			☐ PIVOT: CENTER/OFFSET, %		
☐ No. PADS			☐ PAD BASE MATL		
☐ LOAD: B'TWN/ON PAD			LUBRICATION: ○ FLOODED ○ DIRECTED		
☐ PIVOT: CTR/OFFSET, %			THRUST COLLAR: ○ INTEGRAL ○ REPLACEABLE		
☐ ______			MATERIAL ______		
☐ ______					

BEARING TEMPERATURE DEVICES ○ SEE ATTACHED API-670 DATA SHT
○ THERMISTORS
○ TYPE ____ POS TEMP COEFF ____ NEG TEMP COEFF
○ TEMP SWITCH & INDICATOR BY: ____ PURCH ____ MFR
○ THERMOCOUPLES
○ SELECTOR SWITCH & IN BY: ____ PURCH ____ MFR
○ RESISTANCE TEMP DETECTORS
○ RESISTANCE MAT'L ______ ☐ ______ OHMS
○ SELECTOR SWITCH & INDICATOR BY: ____ PURCH ____ MFR
○ LOCATION–JOURNAL BRG
No. ____ EA PAD ____ EVERY OTH PAD ____ PER BRG
OTHER ______
○ LOCATION–THRUST BRG
No. ____ EA PAD ____ EVERY OTH PAD ____ PER BRG
OTHER ______
No. (INACT) ____ EA PAD ____ EVERY OTH PAD ____ PER BRG
OTHER ______
○ MONITOR SUPPLIED BY (3.4.7.3) ______
○ LOCATION ______ ENCLOSURE ______
○ MFR. ______ ☐ MODEL ______
☐ SCALE RGE ____ ALARM ☐ SET @ ______ °F
○ SHTDWN ☐ SET @ ____ °F ○ TIME DELAY ____ SEC

VIBRATION DETECTORS: ○ SEE ATTACHED API-670 DATA SHEET
○ TYPE ______ ☐ MODEL ______
○ MFR ______
○ No. AT EA SHAFT BEARING ______ TOTAL No. ______
○ OSCILLATOR–DETECTORS SUPPLIED BY ______
○ MFR ______ ☐ MODEL ______
MONITOR SUPPLIED BY (3.4.7.2) ______
○ LOCATION ______ ENCLOSURE ______
○ MFR. ______ ☐ MODEL ______
☐ SCALE RGE ____ ALARM ☐ SET @ ______ MILS
○ SHTDWN: ☐ SET @ ____ MILS ○ TIME DELAY ____ SEC

AXIAL POSITION DETECTOR: ○ SEE ATTACHED API-670 DATA SHEET
○ TYPE ______ ☐ MODEL ______
○ MFR ______ ○ NO. REQUIRED ______
○ OSCILLATOR-DEMODULATOR SUPPLIED BY ______
○ MFR ______ ☐ MODEL ______
MONITOR SUPPLIED BY (3.4.7.2) ______
○ LOCATION ______ ENCLOSURE ______
○ MFR. ______ ☐ MODEL ______
☐ SCALE RGE ____ ○ ALARM ☐ SET @ ______ MILS
○ SHTDWN: ☐ SET @ ____ MILS ○ TIME DELAY ____ SEC

CASING CONNECTIONS

CONNECTION	○ DESIGN APPROVAL REQ'D (2.11.2.11) 0	☐ SIZE	☐ FACING	○ POSITION (2.4.2.1)	☐ FLANGED OR STUDDED (2.4.2.1)	○ MATING FLG & GASKET BY VENDOR (2.4.2.3)	☐ GAS VELOCITY FT/SEC
INLET							
DISCHARGE							

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Figure 1-5. API 617, data sheet page 4. (*Courtesy American Petroleum Institute.*)

For magnetic bearings, scores of turbocompressors ranging in size to 15,000 hp and operating speeds around 10,000 RPM had either been manufactured or retrofitted with such bearings by 1997.

Magnetic bearings have several advantages and disadvantages.* Two primary advantages of magnetic bearings are the very low power consumption and very long life. Because there is no contact between the rotor and stator, there is no wear. Where fluid film bearings have high friction losses due to the oil shearing effects, magnetic-bearing losses are due to some low-level air drag, eddy currents, and hysteresis. Also, the losses associated with oil pumps, filters, and piping are much greater than the power associated with controls and power amplifiers. Overall, magnetic bearings normally have a lower power consumption than oil film bearings.

Magnetic bearings commonly have lower power consumption than rolling element bearings. Also, rolling element bearings have finite life and DN (diameter times RPM) limits. Because of the noncontact nature of magnetic bearings, they have much longer expected life and higher DN limitations.

Other advantages of magnetic bearings are related to reduced dependence on environmental conditions. Magnetic bearings do not require oil lubrication so they are well suited to applications such as canned pumps, turbomolecular vacuum pumps, turboexpanders, and centrifuges where oil cannot be employed. They can operate at much higher temperatures or at much lower temperatures than oil-lubricated bearings. A study of aircraft gas turbine engines indicates that the elimination of the oil supply and associated components with magnetic bearings could reduce the engine weight by approximately one fourth.

Among the disadvantages we find higher cost, larger size, and somewhat lower load capacity than in conventional bearings. Nevertheless, magnetic bearings have long left the prototype stage and may be real contenders for some equipment applications.

Getting back to our examination of Figure 1-5, we note the term "gas velocity." Gas velocities are relevant for future compressor uprates. Nozzle sizes must be chosen with future uprates in mind. This topic is further discussed later in this chapter.

The next circled item, on data sheet page 5 (Figure 1-6), deals with coupling selection. Two broad categories of couplings are available to the user: non-lubricated metallic disc and lubricated-gear-type couplings. Metallic disc or diaphragm couplings are engineered for maintenance-free infinite life, but proper alignment is critical; should the couplings ever fail, they may do so with little advance warning. Gear-type couplings are sensitive to lubrication deficiencies and can experience accelerated wear if operated with certain amounts of misalignment. Gear couplings require more maintenance than metallic disc-type couplings. On the other hand, they do give adequate warning of distress. So, which type should be specified? Should the coupling incorporate torque sensing and on-stream alignment monitoring devices? Should a promising new coupling type be specified, or would it be more prudent to

*Courtesy of S2M America, Roanoke, Virginia.

CENTRIFUGAL COMPRESSOR DATA SHEET CUSTOMARY UNITS

PAGE 5 OF 6
JOB NO. ______ ITEM NO. ______
REVISION ______ DATE ______
BY ______

☐ OTHER CONNECTIONS

SERVICE:	NO.	SIZE	TYPE
LUBE-OIL INLET			
LUBE-OIL OUTLET			
SEAL-OIL INLET			
SEAL-OIL OUTLET			
CASING DRAINS			
STAGE DRAINS			
VENTS			
COOLING WATER			
PRESSURE			
TEMPERATURE			
PURGE FOR:			
BRG. HOUSING			
BTWN BRG & SEAL			
BTWN SEAL & GAS			
SOLVENT INJECTION			

○ INDIVIDUAL STAGE DRAINS REQUIRED (2.4.3.2)
○ VALVED & BLINDED
○ VALVED & BLINDED & MANIFOLD

☐ ALLOWABLE PIPING FORCES AND MOMENTS:

	INLET		DISCHARGE			
	FORCE LB	MOMT FT.LB	FORCE LB	MOMT FT.LB	FORCE LB	MOMT FT.LB
AXIAL						
VERTICAL						
HORIZ. 90°						
	FORCE LB	MOMT FT.LB	FORCE LB	MOMT FT.LB	FORCE LB	MOMT FT.LB
AXIAL						
VERTICAL						
HORIZ. 90°						

○ ACCELEROMETER (3.4.7.4)
○ SEE ATTACHED API-678 DATA SHEET
○ TYPE______ ☐ MODEL______
○ MFR______ No. REQUIRED______
○ LOCATION______
○ OSCILATOR-DEMODULATORS SUPPLIED BY______
○ MFR______ ☐ MODEL______
○ MONITOR SUPPLIED BY (3.4.7.5) ______
○ LOCATION______ ENCLOSURE______
○ MFR______ ☐ MODEL______
☐ SCALE RANGE______ ○ ALARM: ☐ SET @ ____IN/SEC²
○ SHTDWN ☐ SET @ ____ IN/SEC² ○ TIME DELAY____SEC

ACCESSORIES

COUPLINGS AND GUARDS

NOTE: SEE ROTATING ELEMENTS - SHAFT ENDS
○ SEE ATTACHED API-671 DATA SHEET
COUPLING FURNISHED BY______
MANUFACTURER______ TYPE______ MODEL______
COUPLING GUARD FURNISHED BY: ______
TYPE: ○ FULLY ENCLOSED ○ SEMI-OPEN ○ OTHER

COUPLING DETAILS
☐ MAX O.D. ______IN
☐ HUB WEIGHT______LBS
☐ SPACER LENGTH______IN
☐ SPACER WEIGHT______LBS

○ VENDOR MOUNT HALF COUPLING
○ IDLING ADAPTER/SOLO PLATE REQ'D (3.2.4)
LUBRICATING REQUIREMENTS:
○ NON-LUBE ○ GREASE ○ CONT. OIL LUBE ○ OTHER
QUANTITY PER HUB______LBS OR GPM

MOUNTING PLATES

○ BASEPLATES: FURNISHED BY (3.3.1.1)______
○ COMPRESSOR ONLY (3.3.2.8) ○ DRIVER ○ GEAR
○ OTHER______
○ DRIP TRIM ○ LEVELING PADS (3.3.2.2)
○ COLUMN MOUNTING (3.3.2.3)
○ SUB-SOLE PLATES REQ'D (3.3.2.5)
☐ STAINLESS STEEL SHIM THICKNESS ______INCHES
PRIMER FOR EPOXY GROUT REQ'D (3.3.1.2.10)
TYPE______

○ SOLEPLATES: FURNISHED BY:______
☐ THICKNESS______IN
○ SUBSOLE PLATES REQ'D (3.3.3.2)
○ LEVELING (CHOCK) BLOCKS REQ'D
☐ STAINLESS STEEL SHIM THICKNESS
○ DRIVER ○ GEAR ○ COMPRESSOR
○ PRIMER FOR EPOXY GROUT REQ'D (3.3.1.2.10)
TYPE______

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Figure 1-6. API 617, data sheet page 5. (*Courtesy American Petroleum Institute.*)

purchase a well-proven "oldtimer," which will require preventive maintenance? Should we allow a short spacer length to be supplied with this coupling, or would not a longer spacer be far more tolerant of the anticipated misalignment between the driver and driven machine? A typical modern centrifugal compressor should be furnished with a generous spacer, preferably 20–30 in. (500mm–750mm) long. This spacer length ensures that driver-to-driven-machine angular misalignment stays within acceptable limits during temperature transients. However, compressor and driver vendors must be aware of the potential impact of this selection criterion and must design the machinery to remain insensitive to dynamic disturbances in spite of the increase in overhung weight. Couplings should preferably be sized for future uprate or maximum allowable shaft torque carrying capacity. This selection guideline will benefit the coupling hub engagement which, incidentally, should be a hydraulic dilation fit instead of the thermal heat-shrink method of yesteryear. Liberally sized shaft ends and coupling hubs will allow the safe use of interference fits on the order of 1.5–2 mils per inch (mm per m) of diameter *without* having to resort to key and keyway combinations.

Data sheet page 6 (Figure 1-7) is the last of the many API data sheets to contain material related to the centrifugal compressor proper.

In the "Shop Inspection and Tests" column, we have checked off some rather indispensable requirements. Also see the arrow that points to testing of compressor and driver combined, so-called string testing. Contrary to general belief, string testing in the vendor's shop is very rarely justified. Early field installation and testing of the entire train makes more sense, both technically and economically.

Although not specifically listed on the API data sheets, disassembling and reassembling the compressor to check bearings, seals, and internal condition is advocated for two reasons: first, it affords an excellent opportunity to develop a photographic record of these procedures for future reference by personnel engaged in turnaround maintenance and emergency repairs. Second, disassembling and subsequent reassembly is required for mechanical run testing of the spare rotor.

And that just about sums it up. Many of the rhetorical questions raised here were meant to alert the specifying user engineer to the need to know what to purchase and why to purchase it. The point is: the user must specify based on knowledge. The data sheets represent a summary checklist or tabulation of extent of supply rather than a specification. Therefore, a specification supplement or similar complementary instructions must accompany the API standard and the API data sheets.

Resourceful and forward-looking equipment purchasers or owner companies should require the manufacturer or vendor to develop and provide machinery installation instructions (Figure 1-8), equipment commissioning instructions (Figure 1-9), proposed instrument checkout guidelines or sequences (Figure 1-10), illustrated spare parts cross-reference tables (Figure 1-11), equipment startup instructions (see Figure 1-26), and other helpful documents. If these requirements are included in the bid request or invitation to submit cost proposals, they may be provided at very reasonable cost. On the other hand, attempting to acquire this important documentation at a later date may prove frustrating, expensive, or futile.

CENTRIFUGAL COMPRESSOR DATA SHEET CUSTOMARY UNITS

PAGE 6 OF 6
JOB NO. ______ ITEM NO. ______
REVISION ______ DATE ______
BY ______

UTILITIES

○ UTILITY CONDITIONS:

STEAM:	DRIVERS		HEATING	
INLET MIN	____ PSIG	____ °F	____ PSIG	____ °F
NORM	____ PSIG	____ °F	____ PSIG	____ °F
MAX	____ PSIG	____ °F	____ PSIG	____ °F
EXHAUST. MIN	____ PSIG	____ °F	____ PSIG	____ °F
NORM	____ PSIG	____ °F	____ PSIG	____ °F
MAX	____ PSIG	____ °F	____ PSIG	____ °F

ELECTRICITY: (3.4.6.1)

	DRIVERS	HEATING	CONTROL	SHUTDOWN
VOLTAGE	____	____	____	____
HERTZ	____	____	____	____
PHASE	____	____	____	____

COOLING WATER:
TEMP. INLET ____ °F MAX RETURN ____ °F
PRESS NORM ____ PSIG DESIGN ____ PSIG
MIN RETURN ____ PSIG MAX ALLOW ΔP ____ PSI
WATER SOURCE ______

INSTRUMENT AIR:
MAX PRESS ____ PSIG MIN PRESS ____ PSIG

SHOP INSPECTION AND TESTS: (4.1.5)

	REQ'D	WIT-NESSED	OBSER-VED
SHOP INSPECTION	✓		
CLEANLINESS (4.2.6)	✓		
QC PROGRAM REVIEW (4.2.8)	✓		
HYDROSTATIC	✓	○	○
IMPELLER OVERSPEED	✓	○	○
MECHANICAL RUN	●	○	○
□ CONTRACT COUPLING □ IDLING ADAPTOR(S)			
□ CONTRACT PROBES □ SHOP PROBES			
VARY LUBE & SEAL OIL PRESSURES AND TEMPERATURES (4.3.4.2.5)	○		
POLAR FORM VIB DATA (4.3.4.3.3)	○		
TAPE RECORD VIB DATA (4.3.4.3.6)	○		
TAPE DATA TO PURCHASER (4.3.4.3.7)	○		
SHAFT END SEAL INSP (4.3.4.4.1)	○		
GAS LEAK TEST DISCH PRESS (4.3.5.2)	○	○	○
○ BEFORE ○ AFTER MECH RUN			
PERFORMANCE TEST (GAS) (AIR) (4.3.6.1)	○	○	○
COMPLETE UNIT TEST (4.3.6.2)		○	○
TORSIONAL VIB MEAS (4.3.6.2)	○	○	○
TANDEM TEST (4.3.6.3)	○	○	○
GEAR TEST (4.3.6.4)	○	○	○
HELIUM LEAK TEST (4.3.6.5)	○	○	○
SOUND LEVEL TEST (4.3.6.6)	○	○	○
FULL LOAD/SPEED/PRESS TEST (4.3.6.9)	○	○	○
HYDRAULIC COUPLING INSP (4.3.6.10)	○	○	○
CERTIFIED COPIES OF TEST DATA (5.2.5.2 (b))	○	○	○

□ TOTAL UTILITY CONSUMPTION:
COOLING WATER ______ GPM
STEAM, NORMAL ______ LBS/HR
STEAM, MAX ______ LBS/HR
INSTRUMENT AIR ______ SCFM
HP (DRIVER) ______ HP
HP (AUXILIARIES) ______ HP
HEATERS ______ KW
PURGE (AIR OR N_2) ______ SCFM

MISCELLANEOUS:
□ RECOMMENDED STRAIGHT RUN OF PIPE DIAMETERS
○ BEFORE SUCTION ______
○ VENDOR'S REVIEW & COMMENTS ON PURCHASER'S PIPING & FOUNDATION (2.1.13)
○ COMPRESSOR TO BE SUITABLE FOR FIELD RUN IN ON AIR (2.1.16)
○ PROVISION FOR LIQUID INJECTION (2.1.10) ______
○ VENDOR'S REVIEW & COMMENTS ON PURCHASER'S CONTROL SYSTEMS (3.4.1.1)
○ EXTENT OF PROCESS PIPING BY VENDOR (3.5.3.1) ______
○ SHOP FITUP OF VENDOR PROCESS PIPING (4.4.3.11)
○ WELDING HARDNESS TESTING (4.2.7)
○ AUXILIARY EQUIPMENT MOTORS EXPLOSION PROOF (3.1.8)
○ ______

□ WEIGHTS (LB):
COMPR. ____ GEAR ____ DRIVER ____ BASE ____
ROTORS: COMPR. ____ DRIVER ____ GEAR ____
COMPR UPPER CASE ______
SOUR SEAL OIL TRAPS ______
LO. CONSOLE ____ S.O. CONSOLE ____
OVERHEAD SEAL OIL TANKS ______
MAX. FOR MAINTENANCE (IDENTIFY) ______
TOTAL SHIPPING WEIGHT ______

□ SPACE REQUIREMENTS (FT & IN.):
COMPLETE UNIT: L ____ W ____ H ____
LO. CONSOLE: L ____ W ____ H ____
S.O. CONSOLE: L ____ W ____ H ____
SOUR SEAL OIL TRAPS ______
OVERHEAD SEAL OIL TANKS ______

REMARKS:

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Figure 1-7. API 617, data sheet page 6. (*Courtesy American Petroleum Institute.*)

Narrative Specifications Lead to Better Machinery

By far the most effective method of specifying equipment is the single narrative document. Instead of using a series of disjointed individual specifications and stapling them into a stack of reference leaflet, the narrative document serves to blend all applicable references into a unified whole. In developing this single narrative, the responsible project engineer will use thought processes that tend to uncover oversights, weaknesses, and deficiencies in the procurement and design efforts for machinery installed in process plants.

Again, the single narrative pulls together only the truly relevant information, whereas the cross-referenced individual plant specification approach tends to be extremely bulky, leaving it to the vendor to find relevant specification clauses, and depriving the purchaser's project engineer from detecting oversights or other deficiencies. Experience confirms that additional cost incurred in developing single narrative specifications is often recovered before the plant starts to manufacture on-specification product at full capacity.

A single narrative supplement should spell out all requirements that add to, delete from, or provide explanatory details to the API focal point specification. A typical narrative specification for centrifugal compressors would state that "the following requirements are additions, deletions, modifications, and clarifications to API Standard 617, Sixth Edition, 1995." The narrative specification would not rewrite applicable, unchanged portions of API 617. Instead, it would identify supplementary or overriding requirements using the paragraph identification numbers used in the API standard. Here is a typical example:

2.12 Nameplates and Rotation Arrows

2.12.1 A nameplate should be securely attached at an easily accessible point on the equipment and on any other major piece of auxiliary equipment.

2.12.2 Rotation arrows shall be cast in or attached to each major item of rotating equipment. Nameplates and rotation arrows (if attached) shall be of AISI Standard Type 300 stainless steel or of nickel-copper alloy (Monel or its equivalent). Attachment pins shall be of the same material.

2.12.3 The purchaser's item number, the vendor's name, the machine serial number, and the machine size and type, as well as its minimum and maximum allowable design limits and rating data (including pressures, temperatures, speeds, and power), maximum allowable working pressures and temperatures, hydrostatic test pressures, and critical speeds, shall appear on the machine nameplate.

(text continued on page 20)

ZER	TURNAROUND AND MAINTENANCE INFORMATION ROTOR INSTALLATION MACHINE TAG NO. ZPT-04 A/B; ZPT-08 A/B/C	ZER

NOTE 3: "A" = ZERO SHOULD GIVE "J" = 0.010 TO 0.014 INCHES

C-03 - MEASURE BEARING CLEARANCES "C" AND "B" WITH PLASTIGAGE AND FEELERGAGE AND RECORD DATA IN SECTION 3254

C-04 - MEASURE ALL OTHER RADIAL CLEARANCES AS PER SECTION 3254 AND RECORD ON DATA SHEET

NOTE 4: ALL RADIAL CLEARANCES TO BE MEASURED FOUR TIMES AT 90° INTERVALS

D. MEASUREMENTS ZP-08 A/B

D-01 - ROTOR IN TIGHT POSITION TOWARDS COUPLING END ... LLAR, THUS DIMENSION "Q" EQUAL TO ZERO

D-02 - MEASURE ALL AXIAL CLEARANCES AS PER SEC... A, F TO N, Q, R, AND U AND RECORD D... DATA SHEET

NOTE 5: "Q" EQUAL TO ZERO ... 0.010 TO 0.014 INCHES

D-03 - MEASURE BEARING CL... AND "P" WITH PLASTIGAGE AND FEELERGAGE. ...ANCES ON DATA SHEET SECTION 3251

D-04 - MEASURE ... IAL CLEARANCES AS PER SECTION 3251 AND R... PER DATA SHEET

ALL RADIAL CLEARANCES TO BE MEASURED FOUR TIMES AT 90° INTERVALS

DETAILED INSTALLATION INSTRUCTIONS

...UREMENTS ZP-08C

E-01 - ROTOR IN TIGHT POSITION TOWARDS COUPLING END THRUST COLLAR, THUS "L" = ZERO

E-02 - MEASURE ALL AXIAL CLEARANCES AS PER SECTION 3252, PARA 3 AND RECORD DATA ON PROPER DATA SHEET

NOTE 7: "A" EQUAL TO ZERO SHOULD GIVE "J" 0.010 TO 0.014 INCHES

E-03 - MEASURE BEARING CLEARANCES "D" AND "E" WITH FEELERGAGE AND PLASTIGAGE. RECORD DATA ON DATA SHEET 3252

DATE		FILE REFERENCE
12/12/97	MACHINERY RELIABILITY PROGRAM	13-2-6.1 SECTION 3203

Figure 1-8. Typical manufacturer's installation instruction.

	PROCEDURE FOR COMPRESSOR RUN-IN LUBE AND SEAL OIL SYSTEM	
	MACHINE TAG NO. C-603 Comp.	

B-13 Bleed and fill active and inactive filters and coolers.

B-14 Increase speed of "A" pump to obtain 40 psig on VP599I downstream of coolers.

B-15 Adjust bearing oil pressure regulator, VP111CV to hold 18 psig at VP612I mounted on compressor deck instrument rack.

B-16 Increase speed of "A" pump to obtain 50 psig on VP599I downstream of coolers.

B-17 Adjust seal oil differential pressure regulators, VP11[illegible] and VP113CV, to obtain 35 psig on VP616 dI and VP614 d[illegible] on compressor deck instrument rack.

B-18 Increase speed of "A" pump to obtai[illegible] on V-P559I downstream of coolers.

B-19 Adjust turbine contro[illegible]ure regulator, V-P109CV, to maintain 100 psi[illegible] at the turbine.

B-20 Open b[illegible] around back pressure regulators VP112-CV and VP1[illegible] keep pump discharge pressure below 320 psig while [illegible]ng the speed of "A" pump to 3550 RPM.

B[illegible] Adjust V-P100-CV to maintain 250 psig on V-P609I on governor oil to VCT-01.

B-22 Adjust VP105-CV to VP106-CV to obtain 290 psig at VP599-I gauge downstream of filters.

B-23 Recheck settings of VP111-CV, Step C-14, and VP112-CV and VP113-CV, Step C-16.

B-24 Check sour oil drain trap level - should be half full with trap float controlling level.

TYPICAL COMMISSIONING INSTRUCTIONS

DATE		FILE REFERENCE
12/12/97	MACHINERY RELIABILITY PROGRAM	13-2-6.1 Section 201

Figure 1-9. Typical manufacturer's commissioning instructions.

HOCK

PROCEDURE FOR COMPRESSOR RUN-IN

LUBE AND SEAL OIL SYSTEM

FAST®

MACHINE TAG NO. VC-01

- Time for "C" to auto-start.
  ________ Sec.
- Minimum oil pressure on VP612-I on compressor deck Instrument Rack.
- Verify V-CT-01 motor control does not trip.
- Minimum level obtained in oil accumulator.

e. Bleed pressure off the lube oil alarm switches.

VP046-LA should annunciate at 15 psig. ________
Alarms at Local Panel ________
MAPS ________
VX020-CA alarms at Control Center ________

VP047LLA should annunciate at 13 psig. ________
Alarms at Local Panel ________
MAP ________
VX020-CA alarms at Control C ________

f. Bleed pressure off seal ... nes

VP041-LA should ... 30 psig. ________
Local Panel ________
MAPS ________
V... s at Control Center ________

... should annunciate at 25 psig. ________
Alarms at Local Panel ________
MAPS ________
VX021-CA alarms at Control Center ________

TYPICAL INSTRUMENT CHECKOUT

VP077-LA should annunciate at 15 psig. ________
Alarms at Local Panel ________
MAPS ________
VX021-CA alarms at Control Center ________

VP078LLA should annunciate at 15 psig. ________
Alarms at Local Panel ________
MAPS ________
VX021-CA alarms at Control Center ________

DATE		FILE REFERENCE
12/12/97	MACHINERY RELIABILITY PROGRAM	13-2-6.1 Section 201

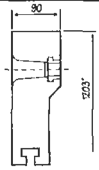
Figure 1-10. Typical instrument checkout guidelines.

LIST OF SPARES

WORKS ORDER	352586	PAGE	5/

PROCESS GAS LOW STG NUMBER 1 Set MACHINE NO. LCT01

MANUFACTURED BY: HIROSHIMA WORKS

SERIAL NO.	PARTICULARS	MATERIAL	NO. OF SUPPLY		DRAW-ING NO.	PART NO.	SKETCH	REMARKS
			WORKING	SPARE				
24	5th Stage Diaphragm and Nozzle	SUS403 SC46	1 Set	1 Set	765 -81962 765 -81974	2515 -11-16 21-26 2715 -11-14	 SIN 66904-825	US $ 17,200
25	6th Stage Diaphragm and Nozzle	SUS403 SC46	1 Set	1 Set	765 -81963	2516 -11-16 21-26		
2ᶜ	7th Stᵉ							

SPARE PARTS CROSS-REFERENCE LISTING

	...6415-14	EA	18015811
	... MFG. CO TURBINE		
	. IND. 6415-14		
	.. CENTER. FOR EXT. CONTROL VALVE	EA	
	PT.NO. 4814-13 ACME MFG. CO		
	ACME MFG. CO 4814-13		
	LCT01 LCT02		
90432-728	SPINDLE, REGULATING VALVE, PT. 4424-11	ST	N038G
	DWG. 765-82245 FOR ACME MFG. CO TURBINE		
	ACME MFG. CO HEAVY IND. 4424-11		
	LCT01		
90432-760	TUBE, COOLING FOR GLAND CONDENSERS PT.	EA	N036C
	6 DWG. 764-10391 FOR ACME MFG. CO		
	TURBINE		
	ACME MFG. CO HEAVY IND. 6		
	LCT01		
90432-763	TURN BUCKLE FOR GOVERNING LEVER	EA	
	PT.NO. 5641-47 ACME MFG. CO TURBINE		
	ACME MFG. CO 5841-47		
	LCT01 VCT01		
90432-764	TURN BUCKLE FOR GOVERNING LEVER PT.NO.	EA	
	5843-32 ACME MFG. CO TURBINE		
	ACME MFG. CO HEAVY IND. 4424-11		
	LCT01		

Figure 1-11. Spare parts cross-reference tables.

(text continued from page 15)

2.12.3 (Addition)

Serial numbers shall also be cast-in or steel-stamped on the casing. The Purchase Order number and yard number shall be included on the equipment nameplates. Compressor nameplates shall include rated capacity and normal capacity.

Here is another example:

4.3.3 Impeller Overspeed Test

Each impeller shall be subjected to an overspeed test at not less than 115 percent of maximum continuous speed for a minimum duration of 1 minute. Impeller dimensions identified by the manufacturer as critical (such as bore, eye seal, and outside diameter) shall be measured before and after each overspeed test. All such measurements and the test speeds shall be recorded and submitted for the purchaser's review following the test. Any permanent deformation of the bore or other critical dimension outside drawing tolerances might be cause for rejection.

4.3.3 (Substitution)

New impeller designs (without demonstrated operating experience) shall be subjected to an over-speed test of at least 120% . . . etc.

After continuing to add to, delete from, or substitute for the various requirements spelled out in API 617, the specifying engineer must further define such items or design elements as instrumentation, valves, auxiliary piping, allowable sound intensity, etc. Unless supplementary specifications for these items are entirely relevant to centrifugal compressors, the specifying engineer should extract only those portions that actually apply to the compressor manufacturer's scope of supply. Again: the specifying engineer should not resort to appending a series of general specifications from which the compressor manufacturer would have to pick an occasional applicable clause.

Instead of developing the narrative specification document, some users assemble many general plant standards or plant specifications into a thick folder which then becomes the procurement specification for a centrifugal compressor. In other words, a general specification describing winterizing of *all* machinery, and specifications on "Flush Oil Injection for Rotating Machinery," "Auxiliary Piping Fabrication and Installation," "Pressure Instruments," "Grouting," etc. are *all* handed to the equipment vendor without first culling the relevant information from the extraneous, or inapplicable, data. Leaving it to the equipment vendor to find relevant clauses hidden in many separate documents puts a burden on the vendor's personnel. Very often, this approach creates bulky specification packages that cause the vendor to add charges for potential oversights. In some cases vendors have refused to bid or have taken blanket exception to the entire specification by stating that their bid covers "Vendor's Standard"—no more, no less. Although there may be occasions when general "catch-all" specifications are appropriate, the user should apply these with discernment. By referring only to those paragraphs or clauses that really pertain to machinery and auxiliaries furnished by the vendor, the specifying engineer will reduce the probability of unexpected problems later in the job. Issuance of a *pertinent* specification package leads to more accurate cost proposals, generally lower prices, and higher quality machinery.

Considering Uprateability and Low Failure Risk

An early decision to provide for future capacity increases or power output uprates may prove highly advantageous in plant debottlenecking or future expansion situations. More often than not, the resulting pre-investment costs are surprisingly low, especially when unexpected mechanical reliability improvements result from the decision to pre-invest.

A process gas compressor for a specialty chemical plant will serve as an example. This compressor required a throughput of 9500 cfm (16,140 m^3/hr) to compress a medium molecular weight gas from about atmospheric pressure to approximately 120 psig (8.3 bar). The vendor's initial offer was for a compressor with a maximum throughput capability of 11000 cfm (18,660 m^3/hr). When encouraged to propose an alternative selection, the vendor submitted a marginally more expensive machine in the next larger casing size. Not only did this machine exhibit an uprate potential to 16000 cfm (27,180 m^3/hr) but it proved mechanically superior, a true workhorse of a compressor which, a good 20 years later, had weathered more abuse than the plant manager cares to remember.

The procurement of uprateable centrifugal compressors usually involves investigating the feasibility of removing the last impeller and moving all preceding impellers into the location previously occupied by the next higher stage. Only a new first-stage impeller would have to be bought later. Figure 1-12 illustrates this principle.

Another way of reducing the pre-investment cost difference would be to purchase the spare rotor (and one spare diaphragm and probably the coupling) to represent the most probable uprate case. An investigation of relevant process parameters would be required to determine whether the *present* plant requirements could be safely accom-

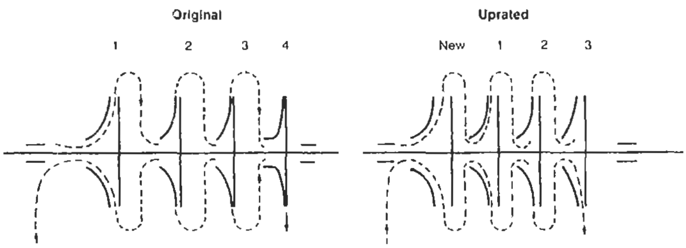


Figure 1-12. Compressor uprate through downward movement of stages.

modated by the "uprate spare" if insertion of the spare rotor is required before actual plant expansion.

In a similar vein, uprateability for reciprocating compressors might require procurement of stronger drive elements or a frame with blanked-off spaces for future connection of additional cylinders. Pumps would be purchased with one or more impeller locations "de-staged," i.e., spaces left blank for future installation of additional impellers. Steam turbines and large drive motors can be executed with through-shafts (double-ended shafts) for future addition of tandem drivers, etc.

Typical questions to ask or to consider are:

1. Power capability—will the driver (electric motor, gear-speed increaser, steam turbine, or gas turbine) handle the uprate requirements?
2. Capacity—will the casing be rated for the anticipated uprate pressures and will equipment nozzles be sized to pass the flow?
3. Speed—can the machine handle the uprate speed without exceeding critical speed and tip-speed criteria invoked by API or self-imposed by qualified manufacturers?

Screening studies may be conducted with the assumption that machine input power requirements increase in direct proportion to increased mass flow rates. Additionally, it is good engineering practice to add an overload contingency of roughly 10% to the uprate factor. Example: The uprate will be from a present 100 mass units per unit time to a future 130 mass units per unit time. The probable new power requirement will be 1.3 times the present requirement. The conservative approach would thus require driver sizing for (1.1) (1.3) = 1.43 times the present requirement.

Capacity uprate capabilities must take into account not only the manufacturer's casing design pressure but also the pressure ratings or relief-valve settings of downstream equipment. In the case of centrifugal compressor uprates, the desired uprate pressure ratio will result in a new polytropic head, H_p. Using the symbol n to denote polytropic exponents, Z for the compressibility factor, R for gas constant, T for absolute suction temperature, and r_p for compression ratio:

$$H_p = ZRT\,[n/(n-1)]\,[r_p^{(n-1)/n} - 1]$$

This calculation is needed to determine later the approximate uprate speed.

Uprate throughput limitations will usually be encountered if inlet nozzle velocities exceed 140 fps (42.6 m/sec) for air and lighter gases. For heavier-than-air-gases, maximum permissible inlet velocities may be significantly lower. Figure 1-13 gives a rule of thumb for permissible inlet velocities as a function of gas molecular weight and temperature.

Approximate uprate speeds can be calculated with the help of the H_p formula given earlier:

$$N \text{ uprate} = (N \text{ original}) \sqrt{H_p \text{ rerate} / H_p \text{ original}}$$

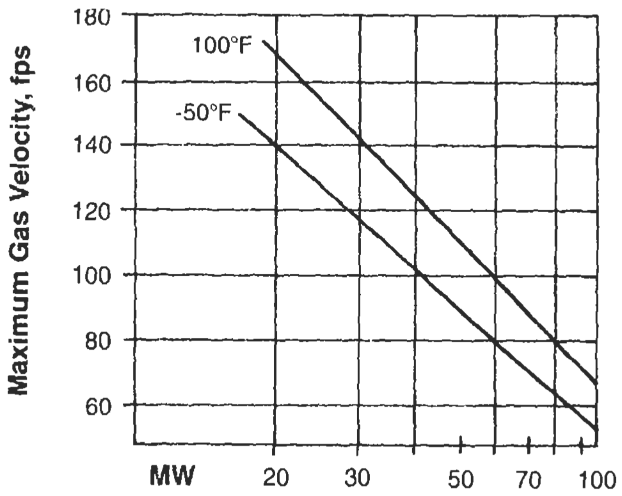


Figure 1-13. Permissible inlet velocity as function of mole weight and gas temperature.

The uprate speed thus calculated must not, however, exceed the mechanical limitations of impeller materials. This limit is typically reached at a peripheral speed of 950 fps (289 m/sec). If N uprate would exceed this limitation, an additional impeller would have to be added. Centrifugal compressors can usually be built with a blank stage, i.e., de-staged and ready to be modified at a later date.

Specifying shaft dimensions, couplings, and thrust bearings for the uprate case will inevitably reduce the risk of machine distress and premature failure. In fairness, it must be said that these measures will result in marginally higher friction losses at thrust bearings and couplings, thus causing a slight increase in horsepower requirements over base-case, normally sized machinery.

The most effective way of reducing failure risk at the specification stage is to examine vendor experience. Detailed experience lists should be submitted for the purchaser's review. These lists must clearly identify the locations where identical compressor impellers, shaft stresses, couplings, impeller or turbine blade stresses, machinery bearings, bearing spans, speeds, etc. have been successfully used in the past. A detailed review of the vendor's claimed component experience should be made *before* issuing purchase orders, and this review should include telephone contact with other users. In some cases, a plant visit a continent away will be well worth it. We know of instances where, in preparation for the procurement of diesel generators for South East Asia, the U.S.-based review engineer visited an installation in West Africa, and where a U.S. manufacturer's steam-turbine experience was reviewed by a site visit to a location halfway around the globe. It was well worth it, in both cases.

Auxiliary Systems for Turbomachinery: The Systematic Approach

Malfunction of auxiliary systems (e.g., speed governors, lube and seal-oil supply consoles, etc.) is responsible for a large portion of unscheduled downtime for turbomachinery. This is a fertile field for improvements in specifications, post-order reliability audits, pre-commissioning checkouts, and post-S/U maintenance.

To ensure that the specification is written for maximum equipment reliability, the specifying engineer must generally go beyond the industry's standard specification. He must know what it is he is specifying and how the system, subsystem, or even how a given component functions and performs. If he is not sufficiently qualified to make the decisions that necessarily lead to the procurement of highly reliable machinery, he should seek the advice of experienced plant engineers or consultants.

While it is beyond the scope of this text to rewrite or pre-define entire specifications for process plant machinery, it is important to make the preceding points as forcefully as possible. Using compressor lube and seal oil systems as an example, we want to see how the systematic examination of even a generally acceptable industry standard specification can lead to revisions and amendments that will make the equipment easier to operate and more maintainable, reliable, or accessible.

Specifying Lube and Seal Oil Auxiliaries

Lube oil or seal oil supply systems provide required quantities of lubricating or sealing oil to machinery bearings, gears, and/or seals. The oil has to be filtered, cooled (or preheated in some ambients), and pressurized. It has to be stored, purified, delivered, returned, metered, bypassed, degassed, switched through different headers, and blocked in. All of these functions require hardware, and while the purchaser may elect to leave the selection of hardware to the machinery manufacturer, the purchaser nevertheless must identify and specify the desired systems configuration.

API Standard 614,* "Lubrication, Shaft-Sealing, and Control Oil Systems for Special-Purpose Applications," can serve as a skeleton specification for lube and seal oil systems. However, to ensure reliable operation, a number of supplementary requirements should be specified by the purchaser. Referring to Figure 1-14, we would add to or modify the reservoir as follows:

- The filter-breather should be extended 6 or more feet (2 m) above the reservoir top to encourage oil vapors to condense inside the extension piece rather than escaping to the atmosphere. Furthermore, during periods of gas leakage past the compressor seals, we want the gas to escape well above grade.
- For better heat transfer and reduced corrosion risk, the steam-heater cavity at the reservoir bottom should be filled with a heat-transfer oil or perhaps discarded lube oil. A filler standpipe and breather cap should be provided.

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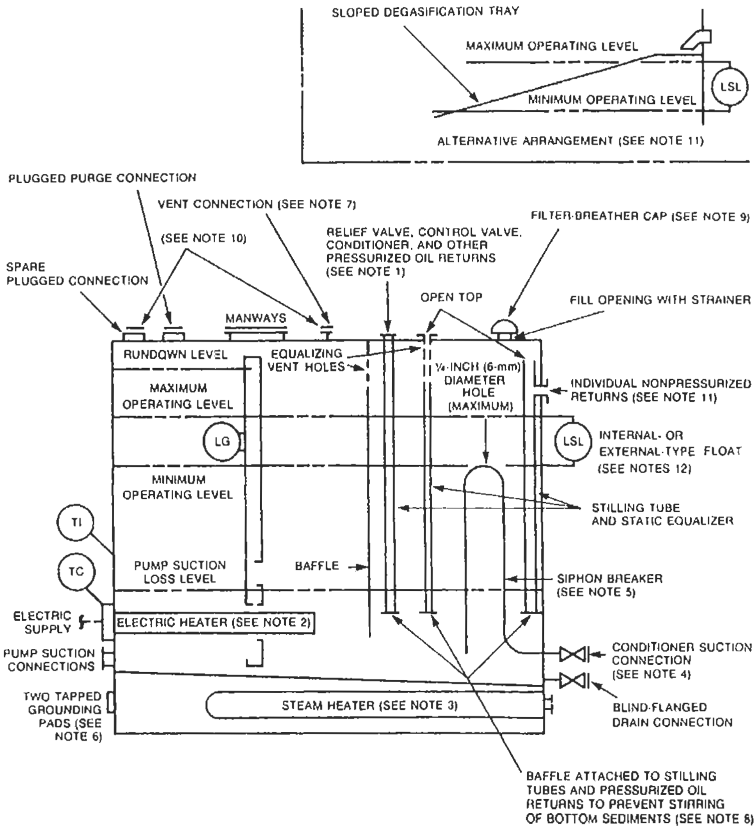
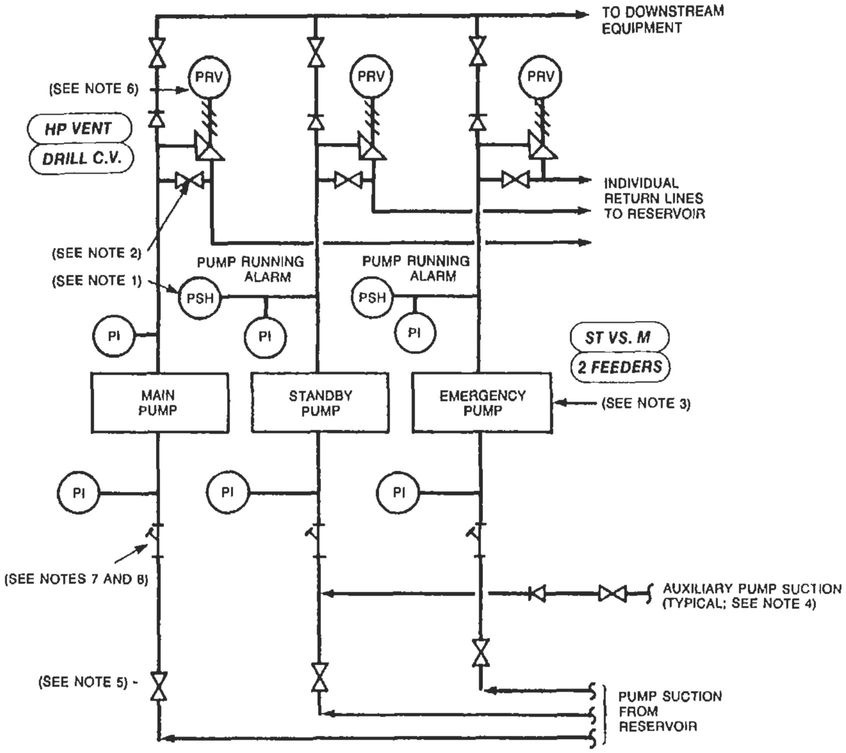
*API Standard 614, "Lubrication, Shaft-Sealing, and Control Oil Systems for Special-Purpose Applications," Third Edition, 1992, reprinted by courtesy of the American Petroleum Institute.



NOTES:
1. Option A-22a: The purchaser may specify a particular oil conditioner and other pressurized oil returns in addition to the spare top connection.
2. Option A-22b: The purchaser may specify an electric heater.
3. Option A-22c: The purchaser may specify a steam heater.
4. Option A-22d: The purchaser may specify an oil conditioner suction connection.
5. Option A-22e: The purchaser may specify a siphon breaker when an oil conditioner suction connection is specified.
6. Option A-22f: When specified, two tapped grounding pads positioned diagonally to each other shall be provided.
7. A blind flange shall be provided for venting the reservoir. For seal-oil reservoirs, this vent shall be piped to a safe location by the purchaser.
8. Individual oil returns shall be located away from the pump suction and arranged to provide the maximum residence time.
9. A filter-breather cap is not permitted on a reservoir containing seal oil.
10. Purge and vent connections shall enter the top of the reservoir. No extension tubes or seals are permitted.
11. For nonpressurized gravity oil return lines, a stilling tube or sloped degasification tray arranged to prevent splashing and provide free release of foam and gas is required for every return inlet and spare connection.
12. An internal-type float shall be protected by a static-conducting shield.

Figure 1-14. Oil reservoir, standard arrangement. *(Courtesy American Petroleum Institute.)*

- If an electric heater is used, it should be located in the steam-heater cavity at the reservoir bottom, as long as the cavity is filled with heat-transfer oil or discarded lube oil. This will reduce the risk of lube oil deterioration from prolonged contact with an excessively hot heater cartridge.



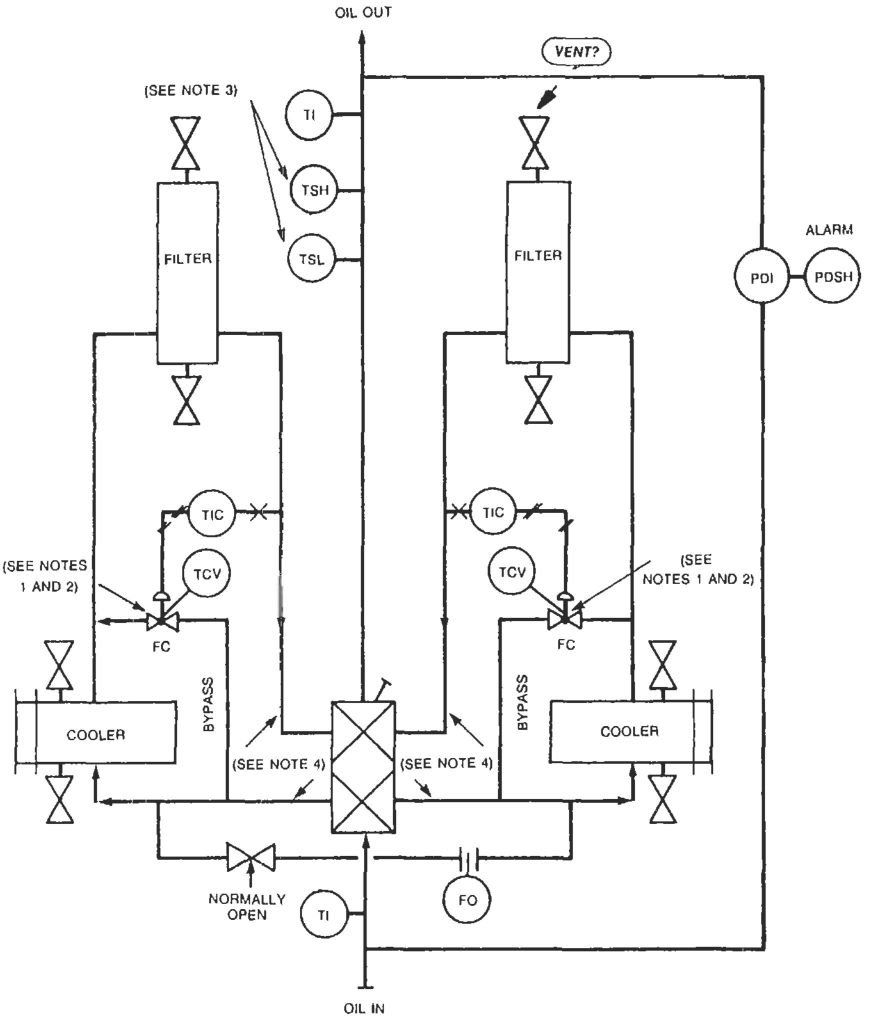
NOTES:
1. Option A-20a: Alarm switches are omitted if (a) the running signal is taken from the motor starter or (b) an alarm switch is on the turbine driver.
2. Option A-20b: The purchaser may specify a bypass valve to start.
3. Option A-20c: The purchaser may specify an emergency pump.
4. Option A-20d: For positive displacement pumps, the purchaser may specify an auxiliary emergency suction line from the reservoir to the main, standby, or emergency pump.
5. Suction valves are omitted for pumps submerged in the reservoir.
6. The pressure-regulating (relief) valve is omitted for centrifugal pumps.
7. For centrifugal pumps, the line strainers are omitted and temporary screens are provided.
8. A basket-type screen shall be used instead of a line strainer for the suction of pumps submerged in the reservoir.

Figure 1-15. Primary pump arrangement, centrifugal or positive displacement pumps. *(Courtesy American Petroleum Institute.)*

In Figure 1-15, primary pump arrangements (positive displacement pumps):

- Valved vents at high points should be required. This may be impractical, yet provision is required to prevent standby equipment from becoming vapor bound. Drilling a small hole into each discharge check-valve flapper will serve the same purpose.
- The main pump should preferably be steam-turbine driven. Motor-driven standby pumps will start faster and more reliably.
- Motor-driven spare and emergency pumps should be powered by separate feeders.

Figure 1-16 represents typical twin coolers and filters having the same continuous flow transfer valve. Note that in all but the most elementary cooler arrangements,

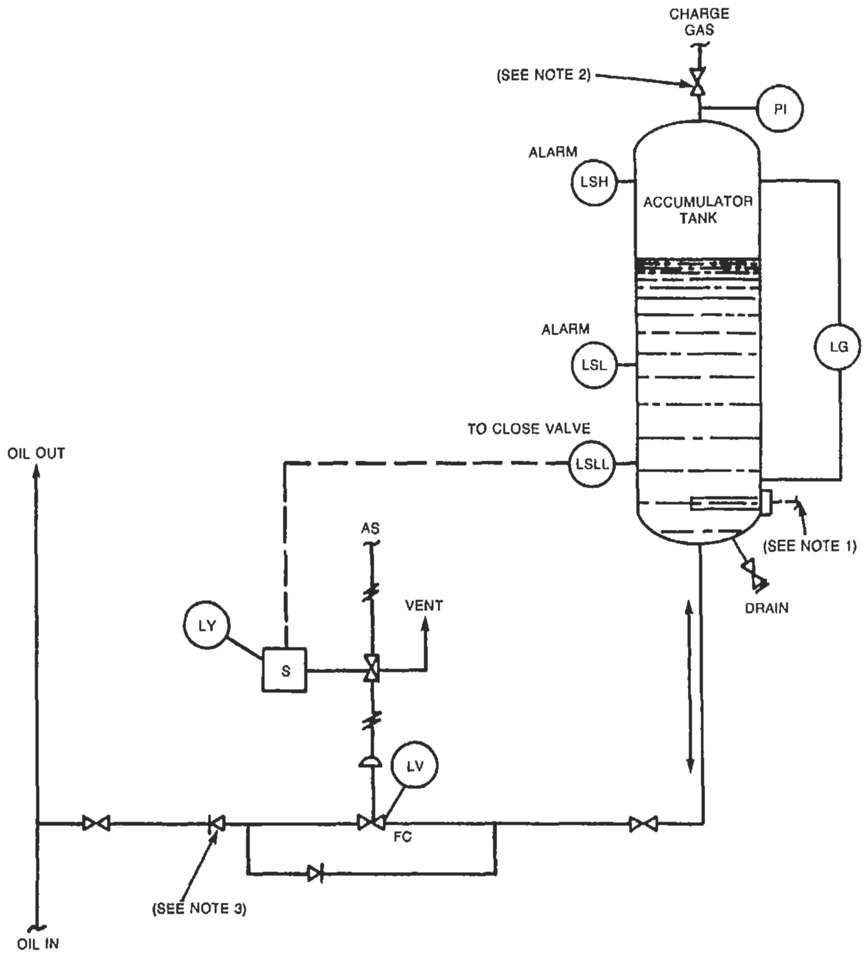


NOTES:
1. Option A-18a: The purchaser may specify a bypass oil line and a constant-temperature control valve.
2. Option A-18b: If the fail-closed (FC) feature of the direct-acting temperature control valve is not acceptable, the purchaser may specify a valve with a fail-locked (FL) feature.
3. Option A-18c: The purchaser may specify a high-temperature switch (TSH) and/or a low-temperature switch (TSL).
4. Option A-18d: The purchaser may specify tight shutoff requiring spectacle blinds.

Figure 1-16. Twin oil coolers and filters with a single continuous flow transfer valve. (*Courtesy American Petroleum Institute.*)

bypass oil lines and constant temperature control valves should be specified. Also, venting of filters and coolers to safe areas must be considered.

Figure 1-17 depicts a direct-contact-type accumulator. Allowing a trickle of oil to leave through an orificed bypass near the accumulator drain and returning this oil to

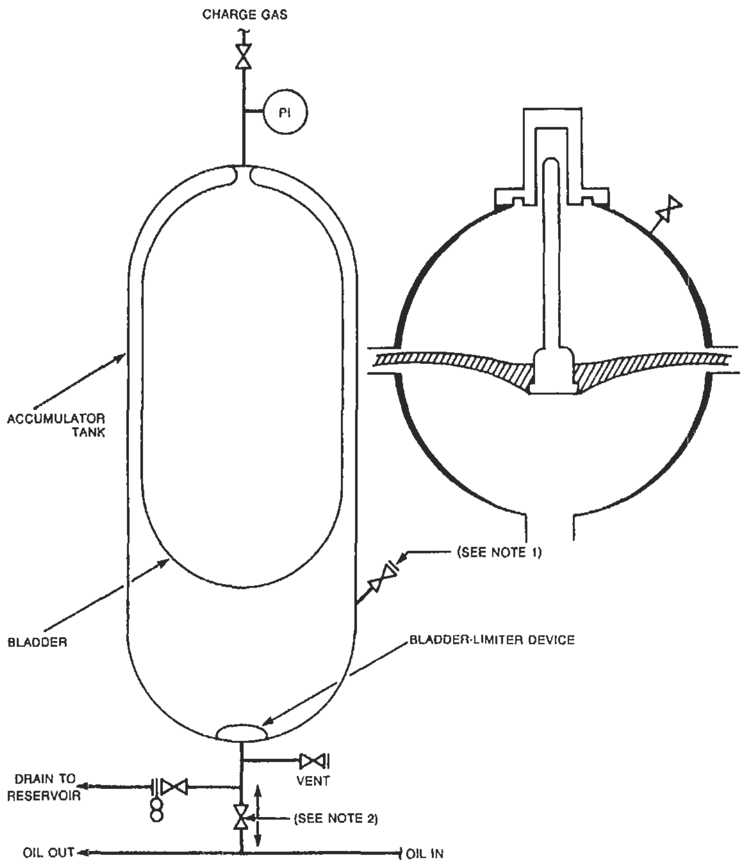


NOTES:
1. Option A-16a: The purchaser may specify an electric heater.
2. Option A-16b: The purchaser may specify a constant-pressure regulating system.
3. The seat or disk is drilled to reduce the recharging rate after an upset of the oil system.

Figure 1-17. Accumulator (direct-contact manual precharge type).

the supply reservoir would ensure having relatively fresh, "undegraded" oil in the accumulator at all times.

Figure 1-18 shows bladder and diaphragm-type accumlators. The specifying engineer should recognize that bladder failure or loss of charge gas will not register as an excursion of the pressure indicator. Checking for adequate charge-gas pressure will require the entire accumulator to be valved off and the trapped oil to be drained to the reservoir. Special diaphragm-type accumulators incorporating an indicator rod



NOTES:
1. Option A-15a: The purchaser may specify a valved vent if one is available.
2. The block valve is locked open.

Figure 1-18. Bladder and diaphragm-type accumulators.

and transparent dome have been used in some installations. The indicator rod is molded into the base of the synthetic rubber diaphragm. Its position can be observed through the transparent dome and will show the relative position of the diaphragm. This diaphragm accumulator (Figure 1-19) can be visually checked for operational readiness without valve manipulation.

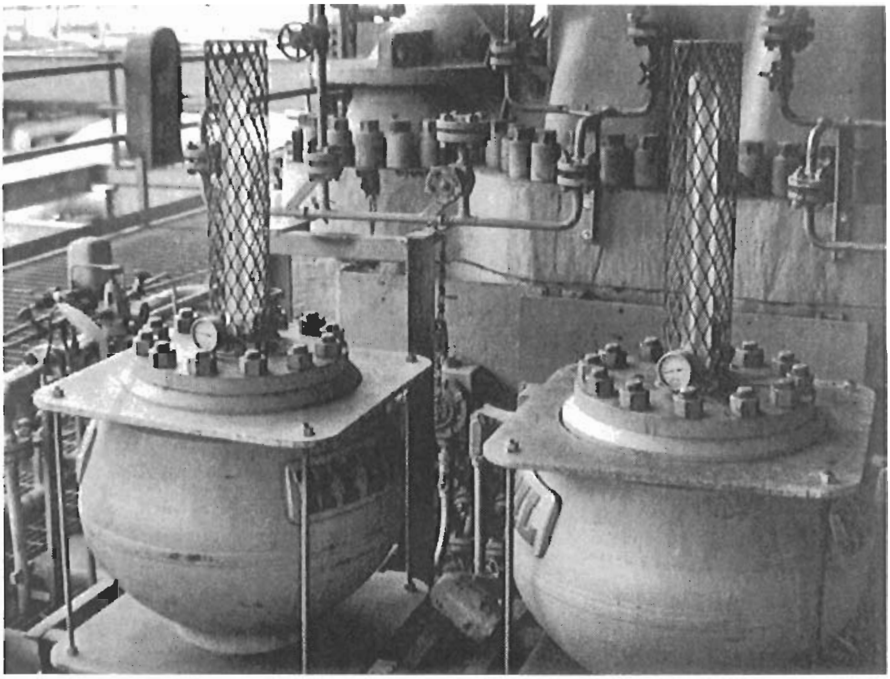
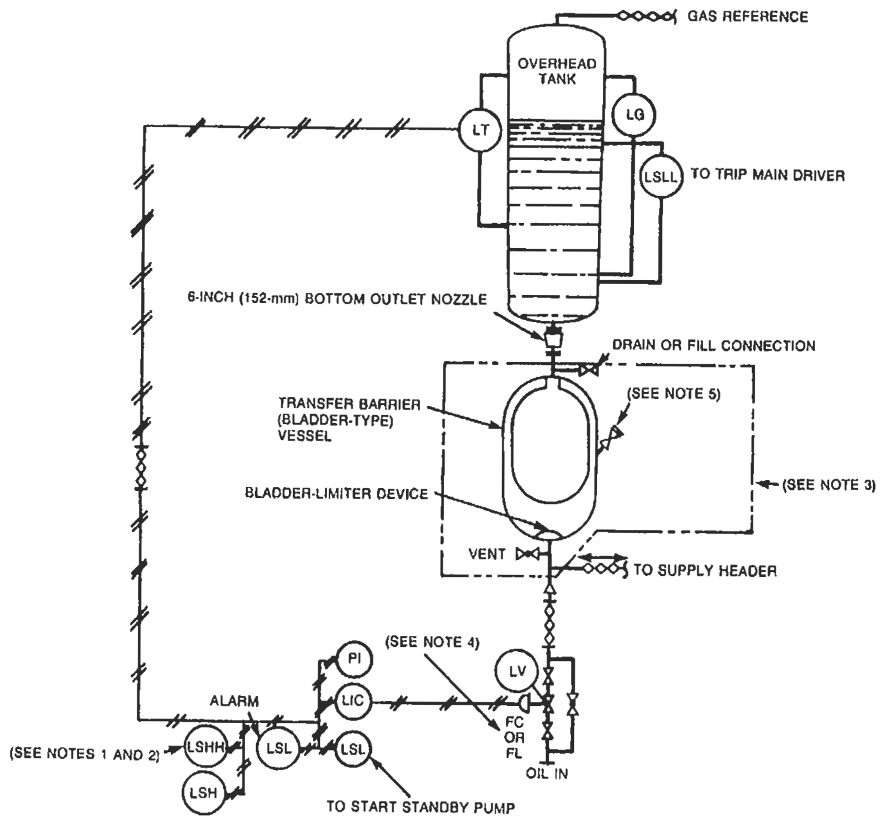


Figure 1-19. Diaphragm-type "surveillable" seal and control oil accumulators in use at a world-scale petrochemical plant since 1978.

Figures 1-20 and 1-21 show overhead tanks with and without transfer barriers. This type of seal oil supply system is sometimes used for compressor installations requiring separation of sour reference gas and seal oil contained in the accumulator vessel. However, the overall reliability of the installation could be improved by installing an orifice at the blowdown valve and continually routing a trickle flow of oil through the orifice and back to the reservoir. This would ensure that the oil volume will not be stagnant and subject to deterioration with the passage of time.

Figure 1-22 illustrates both inner seal drainers (sour seal oil traps) operating with simple float and expensive transmitter controls, respectively. Transmitter-controlled floats are generally able to accommodate a wider range of sour seal oil flows and are suitable for higher pressures.

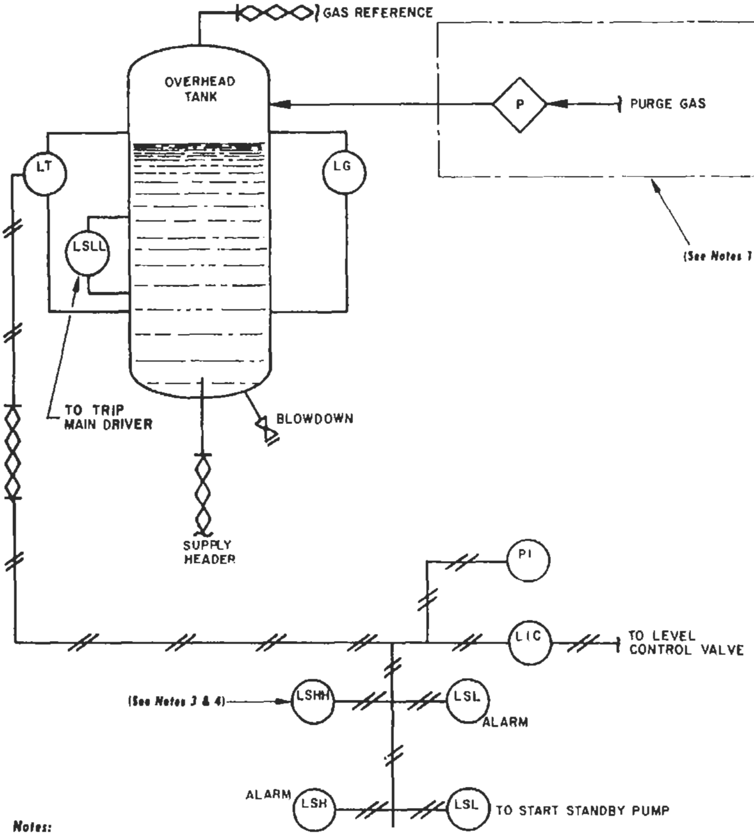


NOTES:
1. Option A-14a: If the pump is for seal oil only, the purchaser may specify a switch to stop the pump.
2. Option A-14b: If pumps are for a combined lube- and seal-oil system, the purchaser may specify an alarm switch.
3. Option A-14c: The purchaser may specify an accumulator with an isolation bladder.
4. Option A-14d: The purchaser may specify the desired failure action for the loop-actuated level control valve.
5. The purchaser may specify a valved vent if one is available.

Figure 1-20. Overhead tank with instrumentation and a bladder-type transfer vessel. (*Courtesy American Petroleum Institute.*)

Figure 1-23 shows an overhead lube oil coast-down tank. Modern compressor installations use overhead tanks for emergency oil supply in case of total failure of main and auxiliary lube oil pumps. The volume of oil supply should be sufficient to ensure bearing lubrication until the machine comes to a complete stop.

Typical lube oil rundown or coast-down tanks incorporate (1) an orifice, (2) check valve, and (3) rapid fill valve. During normal operation of the machinery unit, a trickle of lube oil flows through the orifice into the tank, and returns through a top-mounted nozzle back to the reservoir. When the static pressure in the elevated tank exceeds the pressure in (4) the supply header, lube oil flows down through the check



Notes:

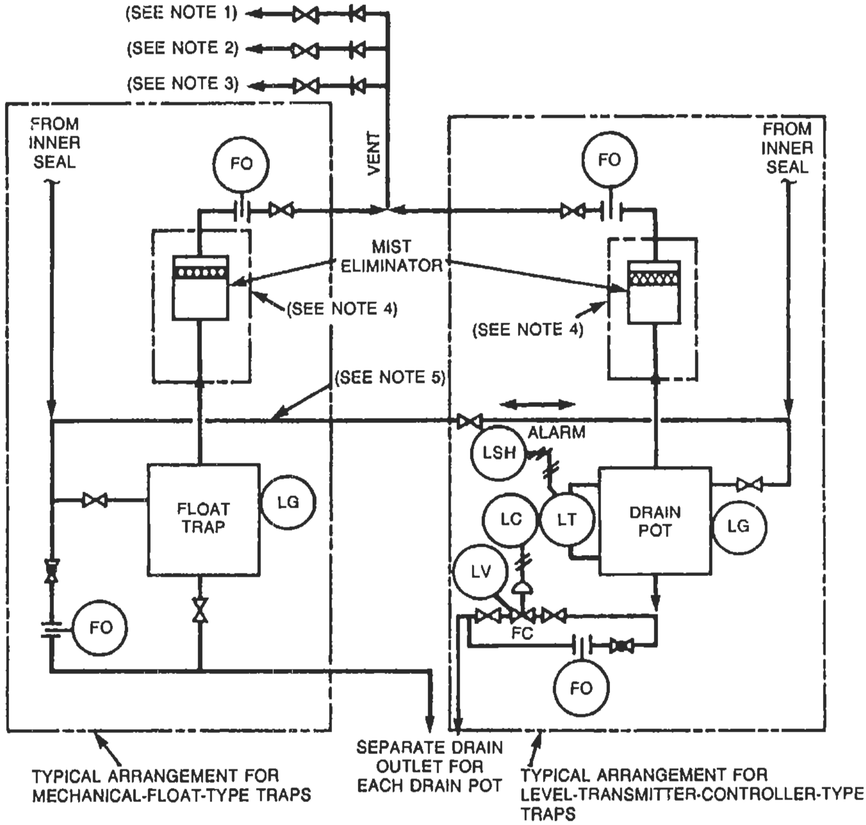
1. OPTION The purchaser may specify a connection for the use of purge gas
2. OPTION The purchaser shall specify the flow control when it is to be supplied by the vendor.
3. OPTION If pump is for seal oil only, the purchaser may specify a switch to stop the pump
4. OPTION The purchaser may specify an alarm switch if pumps are lube and seal combined.

Figure 1-21. Overhead tank (direct-contact type) with instrumentation. (*Courtesy American Petroleum Institute.*)

valve and provides bearing lubrication to the unit. The tank volume vacated by the lube oil is now occupied by air entering through (5) the vent check valve.

Vent check valves are however notorious leak paths. A superior rundown tank is used by European compressor manufacturers. Instead of a vent check valve, their designs employ an oversized return line to the reservoir. During rundown, the displaced oil volume is taken up by air or nitrogen flowing upward from the reservoir in the tank overflow line.

The preceding explanatory information was intended to demonstrate two main points:



NOTES:
1. Option A-12a: The purchaser may specify a vent to the flare.
2. Option A-12b: The purchaser may specify a vent to the gas system.
3. Option A-12c: The purchaser may specify a vent to the suction of a lower pressure casing.
4. Option A-12d: The purchaser may specify mist eliminators.
5. This line is omitted if the seals are not at the same pressure.

Figure 1-22. Inner seal drain traps. (*Courtesy American Petroleum Institute.*)

1. Vendor's standard execution or systems configuration is often not sufficient for maximum reliability of rotating machinery.
2. The specifying engineer should make a component review before analyzing the adequacy of an entire system. He must be familiar with the function of machinery components and interacting auxiliary systems before he can specify for reliability, maintainability, and efficiency.

To close the loop, we include Figure 1-24, a simplified lube and seal oil schematic for a turbine-driven centrifugal compressor.

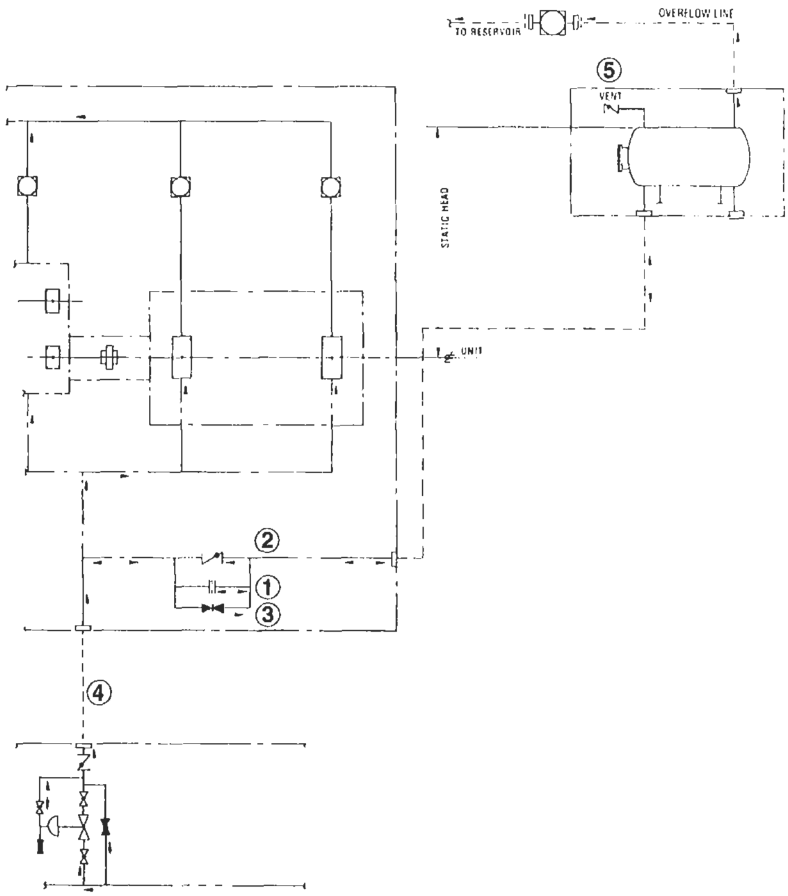


Figure 1-23. Overhead lube-oil rundown tanks. *(Courtesy of The Elliott Company, Jeannette, Pennsylvania.)*

Dealing with Deviations from the Specification

Even a very capable vendor will sometimes see fit to take exception to certain specification clauses. Whatever the reason he cannot or prefers not to comply, it will be to the purchaser's advantage to investigate both reason and significance of noncompliance.

Inquiry documents and purchase orders should make it clear that full compliance with the specification will be a contractually binding requirement unless the vendor has taken specific, written exception to a given clause. If a vendor takes exception, he should state the reasons. Does he believe the cost of the required feature exceeds its benefit? Or does he lack the sophistication to produce the component your specifica-

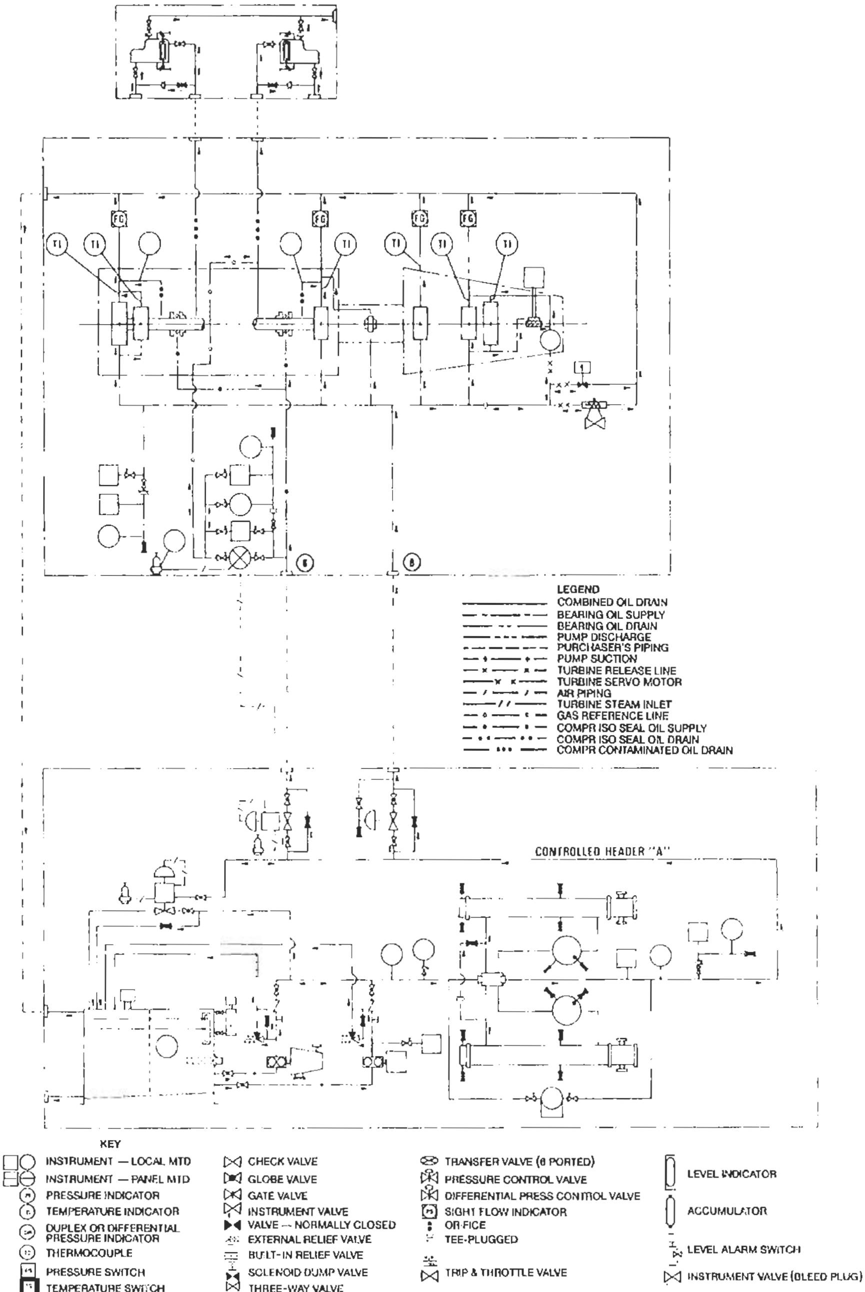


Figure 1-24. Simplified lube- and seal-oil schematic for a turbine-driven compressor. (*Courtesy of The Elliott Company, Jeannette, Pennsylvania.*)

tion asked for? Did you make the mistake of asking for a device that is technologically outdated or redundant? Whatever the reasons, they have to be weighed and evaluated. Rarely does a vendor's bid deserve to be discarded because of well-documented deviations from the purchaser's initial specifications. But extreme vigilance is still required because few vendors comprehend the reliability improvement-driven reasons that prompted you to specify upgraded components and configurations. Be prepared to hear "we've never supplied it that way," and "you really don't need that."

Deviations may deserve credits as well as debits. Procurement of the one machine that meets all specification clauses without regard for installed cost, future operating cost, and future maintenance cost is simply not warranted. A detailed review of user experience has proven helpful in many cases. Such reviews may require visits to other users' facilities. As a minimum, telephone contact with knowledgeable members of other users' maintenance and technical staff seems appropriate. Such contacts have sometimes proven quite revealing. We are reminded of the pump vendor whose reference list showed installations that had experienced such dramatic startup difficulties that the pumps were junked after a few weeks; or of the cooling-tower fan vendor who claimed hundreds of special blades had been commissioned over a year ago, when in fact the user had never run the fans and was disillusioned with the fan vendor's past performance. Again, a detailed experience check pays in most cases.

Conversely, purchasing a machine simply because it meets the basic specification and costs less than competitive equipment may not make much economic sense. The competing offer might be substantially more efficient or less maintenance-intensive at only marginally higher cost, and may be a better choice.

In the case of compressors or large steam turbines, the purchaser might require that critically important parameters, such as bearing spans, impellers, inlet Mach numbers, etc., fall within the vendor's proven range of experience. When deviations from the specifications are proposed in efforts to realize efficiency gains, or in efforts to lower the risk of compressor fouling, etc., the entire design or portions thereof may have to be subjected to computerized analysis. We recall a case where two competing vendors were asked to engage a world-renowned consulting company to perform analog studies of hyper-compressor valve behavior. The losing bidder was contractually assured payment for this work, whereas the winning bidder was to absorb the cost as part of the quoted price.

Cost of Inspection and Expediting

A properly conditioned cost comparison should assign inspection and expediting costs which may vary based on past vendor performance. An experienced purchaser should have data on typical man-hours expended in resolving quality control, delivery, or performance problems with vendor A versus vendor B. If these probable costs are known, there is no reason not to apply them to the bid sheet.

Economics of Efficiency and Extra Capacity

The values of extra efficiency and extra capacity are known and need no further explanation. However, the pitfalls merit further consideration. Efficiency improvements are often achieved at the expense of mechanical reliability. Specifically, tight labyrinth clearances may indicate high test-stand efficiency, but they are also prone to cause mechanical contact during operational upsets. Should this happen, the user is not only losing a fair measure of compressor efficiency, but is also running the risk of mechanical damage.

Extra capacity looks good in light of future uprate potential, but it may cost in terms of requiring large recycle streams to avoid compressor surge during initial operation of the compressor. Similar concerns exist for centrifugal pumps operating at flows substantially below their best efficiency point (BEP). This element is further explained in Chapter 2, "Selecting Major Machinery Vendors."

In today's energy-conscious production and procurement environment, strong contractual safeguards are recommended. In virtually all cases, the user must be prepared to pay the cost of certified and witnessed performance tests. If these tests are to be conducted in the field, it is important for vendor and purchaser to agree beforehand on test-equipment types, test-equipment location, and applicable test procedures to be followed.

Assessing First Cost Versus Maintenance Cost

Typical maintenance costs for major equipment installed and operated in the continental United States are given in Chapter 4, "Benchmarking Maintenance and Reliability." We have arranged the text in this sequence because maintenance costs generally tend to emerge into real focus after the plant has started up. However, from the point of view of the engineer who specifies machinery or selects machinery from among several competing offers, the *future* cost of maintenance should be of concern *now.* An example will illustrate why this is significant.

Let us assume a plant can satisfy a given compression service by using either a single electric-motor-driven centrifugal compressor sized for 100% capacity, or three gas-engine-driven reciprocating compressors sized for 50% capacity. The process engineers, cost estimators, and project engineers have developed data showing the foundation requirements, plot-plan arrangement, piping complexity, and utility costs for the two different approaches. Their balance sheet reflects the principal elements and is given in Table 1-2.

The initial judgment might favor integral gas-engine reciprocating compressors for this service. However, the picture may change when we apply current maintenance statistics for motor-driven centrifugal compressors and integral gas-engine-driven compressors, Table 1-3, and project these costs over the next few years. The importance of paying attention to future maintenance cost is further illustrated in Chapter 5, dealing with life-cycle costs studies.

Specifying Machinery Documentation Requirements

Safe operation, proper surveillance, and cost-optimized maintenance of machinery requires that a good deal of machinery-related data be available and made accessible

Table 1-2
Cost Analysis for Compression Systems

	Motor-Driven Centrifugal	Three Gas-Engine Driven Reciprocating Compressors
Horsepower input	9200	7400 (2 machines running)
Piping system	k$760	k$1384
Foundation	404	1390
3-year cost of power	14904	11888
Barrel compressor	2464	—
Reciprocating compressor	—	4440
Driver and gear	1582	Included/NA
Switchgear	400	—
Fuel-supply facilities	—	400
Total	k$20,514	k$19,502
Maintenance: See Table 1-3 or use plant data		
New Total		

to the plant's technical staff in advance of plant startup. Machinery data and documentation packages often arrive too late to contribute to a successful plant startup. Machinery then has to be commissioned, maintained, or repaired without proper guidance and with inadequate planning.

Startup planning also involves installation completeness reviews.* These reviews are critically important in the chain of pre-commissioning events leading to successful startup. They must be executed by following a checklist-type outline which allows the reviewer to ascertain that all installation-related items defined in the job specification have been furnished or executed. Unless the reviewer is following a written checklist, the installation completeness review may be hit-and-miss, and startup delays or costly failures could result.

Finally, there is documentation which is, indeed, needed only after the plant is on stream. However, this documentation, too, should be obtained before the equipment vendor is paid his final retention payment. Attempts to gather up the data sometime later have often proven to be expensive, frustrating, or unsuccessful. Based on actual experience, it is concluded that project teams should be given the responsibility of obtaining these data. Representing the owner of the plant, the project team may elect to have the design contractor put equipment vendors under contractual obligation to provide important data on time and in usable format.

What Each Machinery Data File Should Contain

Efficient and accurate repair and troubleshooting of machinery requires good documentation. Also, it is important that plant maintenance and technical service personnel are given ready access to this documentation, be it in paper or electronic format.

*For comprehensive information on this topic, see Bloch and Geitner, *Introduction to Machinery Reliability Assessment*, 2nd ed. Houston: Gulf Publishing Company.

Table 1-3
Estimated Average Maintenance Cost for U.S. Process Plants (1997)

Electric Motors • Mean time between failures: 50 operating months • Average overall maintenance cost for plants with 50,000–150,000 installed motor horsepower: \$2.75/hp/yr
Centrifugal Pumps • Mean time between failures: 56 operating months (good plant) • Average cost of shop materials and labor, exclusive of field, technical support, and overhead charges for plants with 50,000 to 150,000 installed motor horsepower: \$9.70/hp/yr • Probable actual cost of average pump repair, exclusive (inclusive) of burden, field labor, and technical support expenses: \$2000/event (\$9000/event)
Motor-Driven Reciprocating Compressors in Clean Service • Mean expenditure: \$9.20/hp/yr, in approx. 1000 hp size machines \$20.00/hp/yr, in approx. 300 hp size machines
Motor-Driven Reciprocating Compressors in Severe Service • Mean expenditure: \$25.00/hp/yr, over 1000 hp size machines \$37.90/hp/yr, up to 1000 hp size machines
Integral Gas-Engine Reciprocating Compressors For plants with installed total horsepower in the 10,000 to 30,000 hp range: \$45.00/hp/yr
Motor-Driven Centrifugal Compressor Trains (Clean Service) • Average overall maintenance cost for plants ranging from 60,000–180,000 installed hp: \$6.40/hp/yr
Steam Turbine-Driven Centrifugal Compressor Trains • Average overall turbomachinery train maintenance cost for plants ranging from 60,000–180,000 installed hp: \$9.20/hp/yr
Gas Turbines • Average overall maintenance cost: \$13.10/hp/yr, 4–10 gas turbines, gas-fired, 6,000–18,000 hp range \$10.00/hp/yr, 4–10 gas turbines, gas-fired, 25,000 hp and higher
Gas Turbine-Driven Centrifugal Compressor Trains • Average overall maintenance cost, trains in 40,000 hp range, natural gas-fired, clean compression service: \$35/hp/yr
Diesel Engines • Average overall maintenance cost, diesel engines only: \$42/hp/yr

Experience shows that major contractors and owner's project engineers orient their initial data-collection efforts primarily toward construction-related documentation. As the project progresses, more emphasis is placed on obtaining design-related machinery documentation as listed in the appendixes of API specifications for major machinery. These requirements are generally understood by contractors and machinery manufacturers. Before the project is closed out, this collection of data is assembled in many volumes of mechanical catalogs.

Unfortunately, data collection using API guidelines alone does not result in a complete data package or in an adequate data format. Moreover, data collection in itself does not automatically provide plant maintenance and technical personnel ready access to all pertinent machinery data. This is where machinery data-file folders and their electronic equivalents fit in.

Machinery data-file folders, manual, as shown in Figure 1-25, or computer-based as available from CMMS (Computerization Maintenance Management System) suppliers, should contain nine key items:

1. Installation, operating-surveillance, and maintenance instructions. These could be instruction manuals routinely available from equipment vendors, or instruction sheets and illustrations specially prepared for a given machine. All of this information is generally intended to become part of the plant's equipment reference data library. However, some specific instructions may also be required for field posting.

MACHINERY DATA FILES

( ) 1. Installation, Operating, & Maintenance Manual (Instructions)

( ) 2. Specification Sheet

( ) 3. Cross Sectional Drawings (if not included in Maintenance Manual) Outline (Dimensional), etc.

( ) 4. Bill of Material (<u>complete</u> parts list)

( ) 5. RPL (Required Parts List) - minimum to be stocked by stores or local vendor

( ) 6. Performance Curve (if applicable)

( ) 7. Mechanical Seal Information (Drawings, etc.)

( ) 8. Design Change Data

( ) 9. Computer Input Form

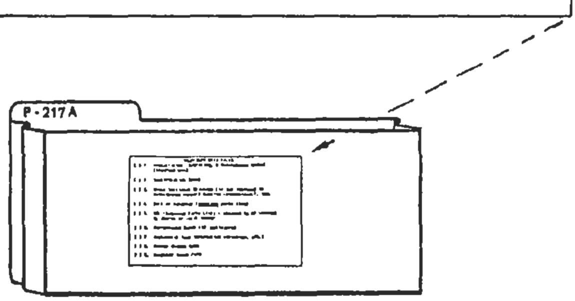


Figure 1-25. Machinery data files.

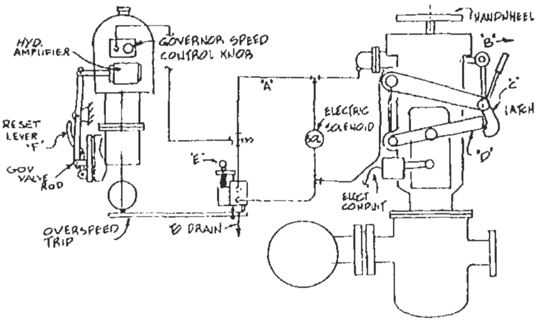
For example, Figure 1-26 represents one of many steam-turbine latching and startup instructions that was developed for both purposes, that is, reference information and field-posted instructions. The vendor should be contractually obligated to prepare these and similar illustrated instructions.

Figure 1-27 belongs in the surveillance data category. This tabulation provides key input for vibration spectrum analysis techniques that are an indispensable troubleshooting tool for modern process plants. Again, the data should be provided by the equipment vendor *before* plant startup. Efforts to acquire this information *after* the equipment vendor has received full payment for machinery and services are usually quite costly and frustrating.

2. Equipment specification sheets, such as API data sheets and supplementary data used at time of purchase.
3. Cross-section drawings showing the equipment assembly. These drawings must be dimensionally accurate. Dimensional outline drawings should also be included.
4. Bill of materials or complete parts list identifying components and materials of construction.

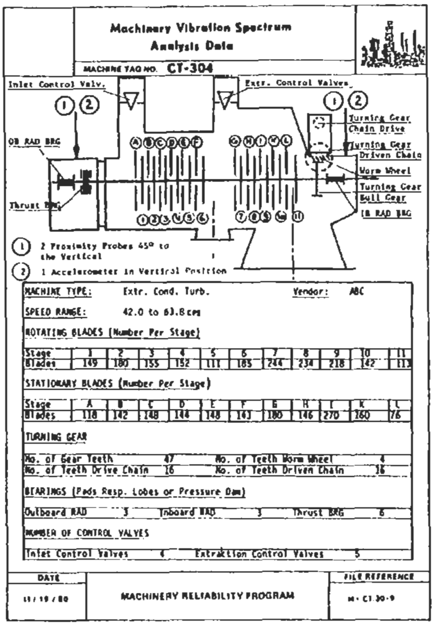
BT-IIB STEAM TURBINE, LATCHING AND STARTUP INSTRUCTIONS

1 Open all 6 drains (casing, inlet, exhaust drains)
2 Verify operation of lube oil system and all steam traps
3 ESTABLISH CONDENSER VACUUM, MOTIVE STEAM TO EJECTOR, COOLING WATER
4 Turbine exhaust verified open
5 Governor speed control knob turned "CCW" to minimum speed
6 Handwheel turned clockwise to completely close trip valve



7 Push S/O logic reset button on Local Panel
8 Be sure ball "E" is pushed in to relatch overspeed trip
9 If hydraulic pressure is established in line "A", rod "B" should have stroked in the direction of the arrow. This should allow latch "C" to engage lever "D"
10 Carefully open the handwheel and at the same time pull the reset lever "F" at the turbine governor valve to reset the governor hydraulic amplifier. Turbine should start. Slow roll for 15 min. at 500 RPM
11 If rod "B" does not stroke to the right, check if electric solenoid is operating properly. If solenoid is open, hydraulic pressure cannot be established in line "A"
12 Verify that governor is functioning properly by observing movement of turbine inlet steam valves
13 After governor has taken over, open handwheel fully. Verify turbine is operating at proper speed

Figure 1-26. General-purpose steam-turbine latching and startup instructions.



MACHINE TYPE: Extr. Cond. Turb. Vendor: ABC

SPEED RANGE: 42.0 to 63.8 cm

ROTATING BLADES (Number Per Stage)

Stage	1	2	3	4	5	6	7	8	9	10	11
Blades	149	180	155	152	111	185	244	234	218	142	[illegible]

STATIONARY BLADES (Number Per Stage)

Stage	A	B	C	D	E	F	G	H	I	K	L
Blades	118	142	148	144	148	143	180	146	270	160	76

TURNING GEAR

No. of Gear Teeth 47 No. of Teeth Worm Wheel 4
No. of Teeth Drive Chain 16 No. of Teeth Driven Chain 16

BEARINGS (Pads Resp. Lobes or Pressure Dam)

Outboard RAD 3 Inboard RAD 3 Thrust BRG 6

NUMBER OF CONTROL VALVES

Inlet Control Valves 4 Extraction Control Valves 5

DATE		FILE REFERENCE
11/19/80	MACHINERY RELIABILITY PROGRAM	M-CT 30-9

Figure 1-27. Machinery vibration spectrum analysis data.

5. Tabulation of minimum number of parts to be kept on hand by storehouse or local vendor.
6. Performance curve, if applicable.
7. Mechanical seal and seal-gland drawings, if applicable.
8. Design-change data. A typical design-change form is shown in Figure 1-28. It is used to document one of the many minor modifications which will inevitably be made after the machinery reaches the plant. This example shows a change which had to be implemented on oil-mist-lubricated motor bearings. After modifying the upper bearing retainer to provide oil mist flow *through* the bearing instead of *past* the bearing, the design-change form is placed in the file folder and a notation made on the front of the cover to let the user know how many design-change forms he should find inside.
9. Computer input forms. These forms should be given to the equipment vendor as part of the specification package for pumps, motors, and small steam turbines. Providing basic equipment data should be part of the vendor's contractual requirements. Some elements of a typical computer input form for pumps are highlighted in Figure 1-29. In addition to information given in the API data sheets, the equipment vendor must provide such important maintenance information as impeller and bushing clearances, manufacturing tolerances, and as-built internal dimensions. The computer input forms can serve as the nucleus of a computerized failure-report system for a given plant. As a minimum, properly

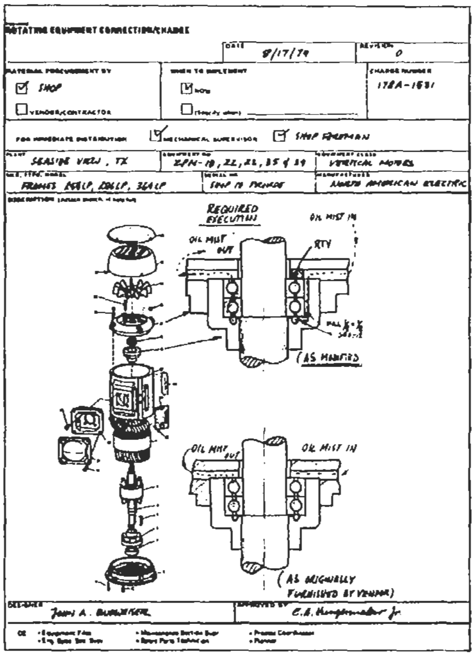


Figure 1-28. Rotating equipment correction/change.

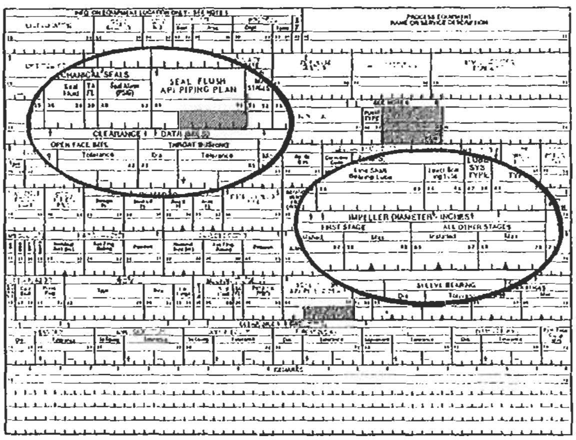


Figure 1-29. Process equipment computer input form.

filled-in forms represent a databank of valuable maintenance and interchangeability information. Large petrochemical companies may derive additional benefits from using these computerized data sheets for intra-affiliate analyses of commonality of spare parts, etc.

Major Machinery Turnaround Documentation

Efficient and effective future machinery maintenance should be planned at the inception of any given project. While this statement holds true in any industry, it takes on added importance in the petrochemical plant environment where downtime for unspared major machinery can cost staggering amounts of money.

Assembly and disassembly procedures for machinery or critical machine subassemblies and tabulations of critical machine dimensions will have to be acquired well before turnaround maintenance can be performed. The logical time for defining and assembling all these data is *prior* to equipment delivery.

The critical-dimension diagram shown in Figure 1-30 is typical of size and tolerance information which must be cataloged for axial as well as radial dimensions of turbomachinery rotors, bearings, seals, admission valves, governors, etc. Many of these data must be obtained during initial assembly of the equipment and cannot be retrieved once the machine leaves the manufacturer's facilities. This makes it again appropriate to require the equipment vendor to carefully record these dimensions at initial assembly and to furnish these data to the contractor and ultimate owner before the machinery is commissioned at the plant.

A photographic record of machinery assembly and disassembly is of similar importance to process plants. Figure 1-31 depicts a random page taken from the illustrated reassembly manual developed for a large mechanical-drive steam turbine. Each step is documented both pictorially and in writing. A margin column lists the time it takes a designated maintenance crew to perform the task. The number in parentheses gives the cumulative total crew-hours expended.

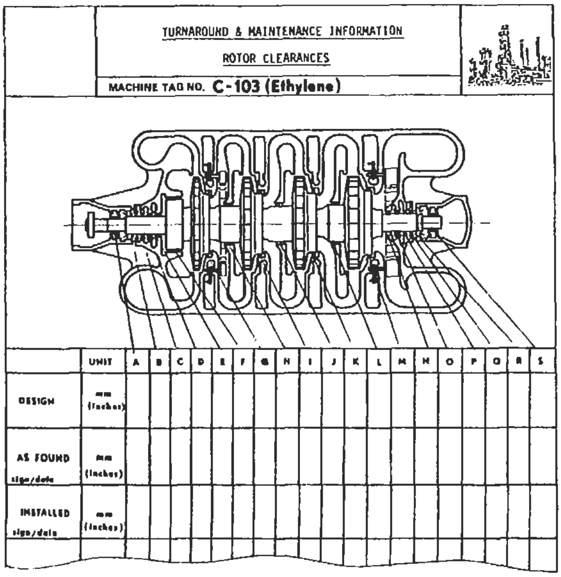


Figure 1-30. Turnaround and maintenance information: rotor clearances.

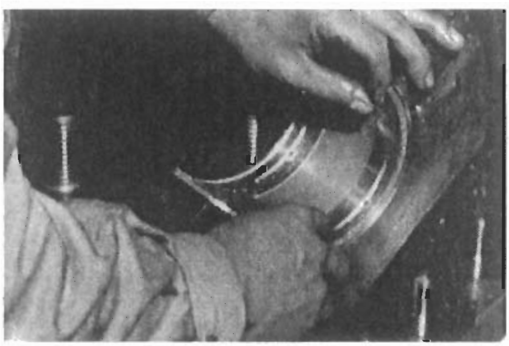
REASSEMBLY PROCEDURES

FOR MAJOR H.P. STEAM TURBINES

MACHINE TAG NO. **BT-201 Propane Refrigeration**

hrs

(35) The low pressure side casing bolts are tightened with a hammer or extension wrench until the specified torque values are reached. Exhaust casing connection bolting must be loose before the horizontal joint bolts are tightened. Only then can the attaching bolts be tightened. — 3 (62)



(36) Inspection before assembling the governor-side bearing. After cleaning the upper bearing housing, the bearing liner is fitted, and plastigage placed on the bearing liner. — 2 (65)

DATE		FILE REFERENCE
5/9/80	MACHINERY RELIABILITY PROGRAM	M-80.4-C201

Figure 1-31. Reassembly procedures for major HP steam turbines.

Illustrated instructions of this type also serve as training material for personnel involved in turnaround execution. A copy of the manual belongs on the compressor deck for the duration of the machinery turnaround. Its conscientious use cannot help but reduce the risk of oversights and delayed restarts of major machinery. This illustrated manual is also highly useful to planners whose detailed turnaround scheduling efforts will become more precise if actual times-to-perform-tasks are catalogued in this fashion.

Fully illustrated assembly and disassembly manuals are often routinely furnished by first-class overseas machinery manufacturers. There should be ample incentive for reputable vendors to express willingness to make similar instructions available to interested users. By the same token, project staff representing the ultimate owner of large, unspared turbomachinery should arrange for these illustrated manuals to be produced.

Spare Parts Identification Sheets

Most major contractors are experienced in advising their clients on recommended levels of spare parts procurement. These recommendations are generally reviewed and translated into a spare parts warehousing system that assigns to all spares such data as symbol numbers, reordering information, bin location, etc.

Among "etc.," we found quality-control information most helpful. Critical spare parts are identified with code letters indicating that the user's inspectors should check the parts at the point of origin or upon receipt at the user's warehouse. This procedure is bound to reduce the number of unpleasant surprises reported by petrochemical plants embarking on major machinery turnaround only to discover that spare parts errors were about to delay completion of machinery turnarounds.

Spare parts identification sheets differ from conventional spare parts documentation or traditional storehouse information in a number of ways. They are primarily intended as an aid to mechanics, machinists, and turnaround planners. These persons require that spare parts information be contained on a single sheet, not in separate catalogs or on computer printouts. In many cases, illustrations are required for positive identification of parts by personnel unfamiliar with either the machinery or the storehouse routine.

To satisfy all of these needs, process plants should require machinery spare parts information to be displayed as shown in the spare parts documentation sheet represented in Figure 1-32.

Major machinery spare parts documentation sheets must contain all the information needed by the mechanical work forces to locate the parts in the storehouse. These documentation sheets must allow mechanics, turnaround planners, and inspectors to verify stock levels, critical dimensions, and suitability of parts. Cross references and design-change and inspection information complete the sheet and make it a stand-alone, highly useful document.

Lubrication Data

Pertinent data on manual as well as automated equipment lubrication must be available well in advance of plant startup. Lubrication summaries are routinely prepared by the contractor and range in format from simply restating the original equipment manufacturer's recommended lubricants to an intelligent consolidation of

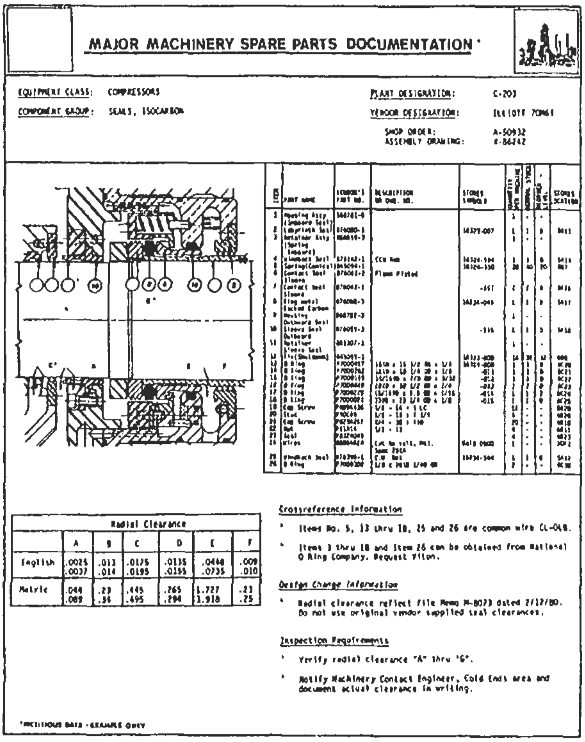


Figure 1-32. Major machinery spare parts documentation.

requirements allowing the process plant to standardize on a small number of premium-grade lubricants.

However for maximum usefulness and to reduce possible misunderstandings, additional documentation must be developed. The first of these supplementary listings should begin with a suitably reduced plot plan of a given area, block, or unit of the plant. Figure 1-33 illustrates this. All process machinery, regardless of its lubrication needs (periodic, life-time greased, circulating oil, etc.) is identified on this plot plan. The equipment shown on this "roadmap" is next transferred to a tabular listing and relevant parameters are added, as shown in Figure 1-34. Both Figures 1-33 and 1-34 can be utilized by process operators, maintenance personnel, or unskilled contract laborers to implement a conscientious lubrication program.

Finally, Figure 1-35 represents an oil-mist lubrication summary to be provided by the contractor or project team. This summary contains information that is essential to the proper sizing of components incorporated in oil-mist lubrication systems. In addition, the oil-mist lubrication summary allows contractor and user to determine the oil and air requirements of a given system.

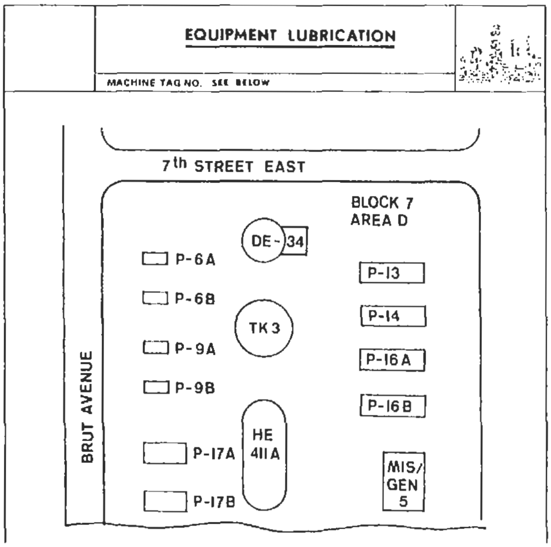


Figure 1-33. Equipment lubrication plot plan.

EQUIPMENT LUBRICATION SUMMARY

LOCATION DATA			LUBRICATION DATA					SCHEDULE				REMARKS
					OIL MIST							
BLOCK	AREA	EQUIPMENT ITEM NO.	LUBE TYPE/ FORMULATION	LUBE APPLICATION	WET SUMP	DRY SUMP	CABINET #	DAILY	WEEKLY	MONTHLY	QUARTERLY	(SPECIAL QUANTITY, DIFFERENT LUBE REQUIRED INBOARD VS. OUTBOARD, ETC.)
7	D	P-6A	EN-100	OM		x	MIS/GEN 5					
		PM-6A	UX-2	GG						x		
		P-6B	EN-100	OM		x						
		PM-6B	UX-2	GG						x		Add Grease While Running Only
		P-9A	EN-100	OM		x						
		PM-9A	EN-100	OM		x						
		P-9B	EN-100	OM		x						
		PT-9B	T-32	BO	x				x	x		Replace Oil Once/Month
		P-17A	T-32	CI	x							150 Gallon Capacity
		PT-17A	T-32	CI	x							150 Gallon Capacity

KEY:

LUBRICANTS	APPLICATION
EN-100 = ENMIST-100	OM = Oil Mist
UX-2 = UNIREX-2	GG = Grease Gun
T-32 = Teresstic-32	GC = Grease Central
G-90 = Gear Oil-90	BO = Bottle Oiler
SY-32 = Synesstic-32	CI = Circulating System

Figure 1-34. Equipment lubrication summary.

OIL MIST LUBRICATION SUMMARY

OIL MIST CONSOLE DESIGNATION: LOCATION:

NOTE 1: USE DIRECTED OIL IF BEARING PITCH LINE VELOCITY EXCEEDS 2000 FPM

NOTE 2: DRY SUMP OIL MIST LUBRICATION IS NOT SUITABLE FOR SLIDING (SLEEVE) BEARINGS. USE OIL MIST FOR PURGE

				ROLLING ELEMENT BRG CHARACTERISTICS										APPLICATION FITTINGS: INSTALLATION LOCATION AND DIAMETER, INCHES			
					INBOARD				OUTBOARD								
ITEM	MFR. AND MODEL	RPM	HP	V>2000FPM (SEE NOTE 1)	ID	OD	ROWS	BRG INCHES	ID	OD	ROWS	BRG INCHES	DIA. IF SLIDING BEARING (NOTE 2)	IN-BOARD	OUT-BOARD	COMMON TO INB & OUTB	OIL MIST FLOW TOTAL TO ITEM (SCFM)
P-6A	Goulds (Overhung)	3560	55		1.7	3.3	1	2	1.7	3.3	2	4				.060	.30
PM-6A	A-C 365TS (Horiz)	3560	75		1.9	3.5	1	2	1.9	3.5	1	4		.032	.032		.18
P-6B	Goulds (Overhung)	3560	55		1.7	3.3	1	2	1.7	3.3	2	4				.060	.30
PM-6B	A-C 365TS (Horiz)	3560	75		1.9	3.5	1	2	1.9	3.5	1	4		.032	.032		.18
P-9A	Bingham (Overhung)	1760	15		1.8	3.3	1	2	1.8	3.3	2	4				.047	.18
PM-9A	A-C 284T (Horiz)	1760	25		1.7	3.3	1	2	1.7	3.3	1	4		.032	.032		.18
P-9B	Bingham (Overhung)	1760	15		1.8	3.3	1	2	1.8	3.3	2	4				.06	.18
PT-9B	Elliott (Horiz)	1760	25										3	.084	.060		.48
P-17A	Pacific(Betw.Brg)	3570	2600										4	.073	.060		.75
PT-17A	Westinghouse	3570	3200										4.5	.073	.073		.90

Figure 1-35. Oil-mist lubrication summary.

Installation Completeness Checklists

Rotating equipment has to be thoroughly checked out before it is ready for startup. These completeness checks, customarily called "but-listing," will ensure that the actual installation complies with all applicable equipment specifications. For instance, on a motor-driven centrifugal pump, completeness checks would *include* installation-related items such as coupling alignment, baseplate grouting, seal flush piping, and lubrication, and *exclude* design-related items such as impeller diameters or seal material.

An installation completeness checklist must be reviewed and finally signed off by the engineer, technician, or field inspector responsible for a given checkout task. This list should take the place of more superficial hit-and-miss review efforts that have traditionally resulted from verbal instructions and "checkout from memory." The development of written installation completeness checklists should be handled by the engineering contractor or the owner's project engineer. This task should not prove difficult because virtually every checklist item can be picked off the detailed job specifications used for a given project. Essentially, the task requires that these job specifications be reviewed item by item and that requirements dealing with *field installation* of rotating machinery be transferred to the checklist in tabular format.

An example will serve to illustrate how installation completeness checklists were developed for a major grassroots petrochemical project. Without these checklists, it would not have been possible to entrust completeness reviews to contract millwrights unfamiliar with the owner's general practices and installation requirements.

For instance, the job specifications for centrifugal pumps were examined and construction elements relating to casings, seals, guards, strainers, piping, and pipe supports tabulated in checklist form. Whenever necessary for clarity, illustration, remarks, or sometimes cross references were added to the tabulations. These cross references would point to standardized small-bore (auxiliary) piping or standardized lubrication specifications, as shown in Figure 1-36.

Similar installation completeness checklists were developed for general-purpose steam turbines, centrifugal fans, and equipment baseplates. In each case, the end product was an installation or construction-related tabulation that reduced the contractor's job specification to 10% or 20% of its initial bulk and restated only those

	INSTALLATION COMPLETENESS REVIEW FOR ROTATING MACHINERY	Page No. 1 of 5 Rev./Date 0/3-2-79 Reference Standard: Plant Spec. 18-3.2

MACHINERY TYPE: All — Exclusions: P-202, P-323
TOPIC OF REVIEW: Lubrication and Bearing Types

The illustrations used in this section cover bearing arrangement and lubrication for electric motors, but the bearing types and principles apply to all oil or grease lubricated bearings.

LUBRICATION

1. Dry Sump Oil Mist Lubrication: Shall be applied to machinery with rolling element bearings only.
2. Wet Oil Sump or "Purge Mist": Shall be applied to machinery with sleeve bearings only.
3. Grease Lubricated Bearings: Where specified, shall have accessible grease fittings for standard grease guns. Fittings must be accessible without requiring removal of guards or covers.

ALL ROLLING ELEMENT BEARINGS

1. Double shielded bearings shall not be used with dry sump oil mist lubrication.
   A. Double shielded bearings are normally used with grease lubricated bearings.
   B. Double shielded bearings are indicated by the bearing identification code letters "PP" on electric motor nameplates.
2. Single shielded bearings are suitable for dry sump oil mist lubrication on Lightly Loaded Bearings. The Shielded Side of the bearing shall be turned away from the mist cavity.
   A. Lightly loaded motor bearings are indicated by the bearing identification code letters "BC"
   B. Single shielded bearings are indicated by the bearing identification code letter "P" on electric motor nameplates. Inlet and outlet connections shall be at the top and bottom housing cover.

THRUST BEARINGS ONLY

1. Non-Shield thrust bearings shall be used with dry sump oil mist lubrication.
   A. Antifriction thrust bearings are indicated by the bearing identification code letters BN, BA, BT, BS, BK, or BG on electric motor nameplates.
   B. No letters "P" are allowed in the thrust bearing identification code on electric motor nameplates.
2. Antifriction thrust bearings shall not be used as coupling end bearings on vertical electric motors driving pumps in hydrocarbon service.
   A. Oil mist inlet and vent connections on vertical electric motors: Shall be permanently marked with letter punches.

NOTE: If bearing shields are removed in order to bring equipment into compliance with any of the above guidelines, the letter(s) "P" must be obliterated from the bearing identification code on the motor nameplate.

Figure 1-36. Illustration completeness review for rotating machinery.

Table 1-4
Machinery Documentation Summary*

	Responsibility				Submit to Owner Before						
Document Title	**Contractor**	**Vendor**	**Project Team**	**S/U Staff**	**P.O. Issue**	**Coord. Meet.**	**Shop Test**	**Shipment**	**Field Erection**	**Field Test**	**Plant S/U**
Machinery data fill folder (All documents are to be submitted in both paper and electronic format)	√					√					
Installation, surveillance and maintenance instructions		√						√			
Special instructions and illustrations		√						√			
Commissioning instructions	√								√		
Machinery vibration analysis data		√								√	
Equipment specification sheets	√							√			
Dimensionally accurate cross sections		√							√		
Bill of materials		√					√				
Tabulation of minimum spares			√				√				
Performance curves		√						√			
Mechanical-seal tabulation	√								√		
Mechanical-seal and gland drawings	√								√		
Design change form				√							√
Computer input form		√						√			
Critical dimension sketch		√						√			
Assembly/disassembly manual		√									√
Spare parts identification sheet			√								√
Lubrication plot plan	√								√		
Lubrication summary	√								√		
Oil-mist lubrication summary	√				√						
Installation completeness checklist	√									√	

**This documentation shall be furnished by contractor and/or vendor and is in addition to documentation requirements indicated in the latest revision of applicable API Standards 610 through 686.*

areas which field personnel could reasonably be expected to review and ascertain. However, the tabulation contained enough information in narrative or illustrated form to convey the full meaning of all relevant checkout tasks.

Conclusion

Relevant documentation as compiled in Table 1-4 must be made available to engineers, maintenance work forces, and technical service personnel responsible for cost-effective, reliable operation of process plant machinery. The most appropriate time for acquiring this documentation is before the plant enters the startup phase. Documentation requirements written into the procurement specification are more likely fulfilled than documentation requests made at a later date.

In fact, "Best-of-Class" companies stipulate in their purchase orders that 10–15 percent of the negotiated cost of the entire order will be withheld until the vendor has provided *all* specified documents.

Finally, the job of machinery engineers, maintenance workforces, and technical service personnel will be further facilitated and cost savings realized if requisite documentation is presented in useful paper or electronic format. The foregoing examples were found to represent this format in paper form.

Chapter 2

Vendor Selection and Bid Conditioning

Selecting Major Machinery Vendors

It goes without saying that quality and reliability of rotating equipment for process plants are largely determined by the slate of vendors selected to submit cost proposals. Even the best specification may be lost on a vendor who lacks the skill, experience, resources, and follow-up organization needed to carry out the manifold detailed engineering, manufacturing, and quality-control steps that will culminate in a high-quality product.

Whenever possible, inquiry specifications should go only to vendors who meet these criteria. If a vendor can demonstrate compliance with the specified requirements and can stand the test of a rigorous bid comparison with his respective competitors, the purchaser should not be reluctant to place an order with this vendor. Subjective judgments should be left out of the selection process, and only tangible evidence of experience, capability, and past performance should be weighed. Of course, there will never be any substitute for plain old common sense and soundly conducted engineering reviews. This is especially so when prior experience simply doesn't exist or if there are compelling reasons to buy machinery incorporating prototype components, novel manufacturing techniques, etc.

We know of relatively few procurement situations in which capable vendors would not be able to demonstrate satisfactory experience for the service conditions imposed on the proposed machinery. Service conditions include parameters such as molecular weight, viscosity, volumetric capacity, power rating, speed, temperature, and pressure. Similarly, capable vendors can usually demonstrate mechanical-design experience for applicable critical parameters that could include, but not be limited to:

- Bearing span
- Bearing design, loading, size, and clearance
- Blade design
- Casing size and design, and exhaust orientation
- Casing-joint design configuration, gasketing, and bolting
- Coupling design and arrangement

- Extraction control—single and double extraction
- Impellers—structural design and performance
- Material selection
- Nozzle design
- Number of stages and staging arrangement
- Power-transmission components (design and arrangement)
- Rotor dynamics
- Sealing system

In summary, major machinery vendors should excel through past satisfactory experience in design, execution, and after-sale service of their product. To stand the most critical examination, a vendor should show cooperation with the purchaser's review engineer. The joint efforts of both parties are needed to demonstrate and verify the vendor's competence and to lead to reliable equipment selection.

Applying and Reviewing Machinery Reliability Improvements Derived from Modern Electronics

Although it is beyond the scope of this text to deal extensively with electronic auxiliaries, the topic of machinery reliability improvements cannot be adequately covered unless the importance of modern electronics is highlighted and review techniques are mentioned.

Compressor Surge Protection and Control

Improved efficiency and reliability of dynamic compression equipment will result if a soundly engineered electronic surge-control system is implemented. The benefits resulting from modern electronic-hardware-based surge-control systems are summarized in two well-written technical papers and listed below:[1,2]

- Less damage to clearances, seals, bearings, etc., which will help maintain the original design efficiency of the machine.
- Reduced recirculation or blow-off when compressors operate under partial loads.
- More stable pressure or flow control at partial loads.
- A more efficient machine design becomes possible, i.e., the machine design may be chosen for maximum energy efficiency rather than for surge safety, as long as effective surge control is assured.

The strategy and instrumentation for surge protection have seen very significant improvements since the late 1970s. Realization that surge constitutes very high speed flow oscillations or even flow reversals makes it imperative to discontinue the use of slow-speed transmitters and recorders. Modern plants should, therefore, utilize high-speed transmitters and antisurge controllers. Fast-action antisurge valves must be chosen to provide full-stroke opening in 1.5 seconds or less. These valves should be sized for full design flow and consideration be given to using two or more small valves in parallel where "false economy" would have perhaps dictated installing one large valve alone.

Selection of Speed Governors and Controls

Mechanical drive turbines, i.e., steam turbine drivers for large process plant compressors, have traditionally employed hydraulic speed-governing devices. The invention of the transistor made it possible to design electronic speed-control devices even more precise than the best hydraulic governors available. While this feature may be highly attractive for parallel operation of synchronous generators, it is of no great importance to most process plant compressors whose speed stability requirements are far less stringent and have generally been met by conventional, high-precision (NEMA-D) hydraulic governors.

The main reason for considering electronic governors seems to have been the desire to get away from failure and downtime due to contamination of hydraulic governor oil and/or governor drive failures. Hydraulic governors contain rotating (mechanical) parts that are driven via worm-gear engagement by the turbine rotor. These drives, which are commonly furnished by the turbine manufacturer, have not always proven totally reliable. The question is now whether it is wiser to stay with a perfectly satisfactory and reasonably simple NEMA-D hydraulic governor and to focus attention on reliability/design improvements for the governor drive arrangement, or get into the electronic governor business and hope for component reliability to meet all your expectations.

Again, the decision should hinge on a full review of the vendor's experience. Reliability analyses could be performed by third parties with experience in the probabilistic assessment of electronic component failures and redundancy design of electronic hardware. The failure or drifting of components could be detected by electronic self-checking features and annunciated well before a machine shut-down is imminent. Also, mechanical or hydromechanical redundancy features can be built into a governing system.

Digital Control Systems Represent State-of-Art*

Control systems offered for steam turbines driving process compressors now provide many more functions beyond speed control. These systems can be custom programmed to provide whatever extent of control desired by the end user of the machinery. Integrating the turbine and compressor control functions provides many benefits to the end user, including a more responsive and capable control system, increased reliability through redundancy, and a custom designed system capable of stabilizing the process during upset conditions. In addition, an integrated system reduces the number of components required for maintenance, training, digital control systems' communication, etc., resulting in economic savings. Among the functions offered in an integrated turbine and compressor control system are the following:

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*Contributed by Gary Bostick, Woodward Governor Company, Dallas, Texas, sales office.

- Speed control
- Extraction steam pressure/flow control
- Inlet/backpressure control
- Electronic overspeed trip
- Start-up sequencing (auto roll up)
- Critical speed avoidance
- Compressor anti-surge protection/control
- Compressor load/capacity control
- Compressor inlet temperature control (quench)
- Drum level control
- Lube oil monitoring
- Compressor seal monitoring
- Alarm and trip functions
- Compressor valve sequencing
- Emergency shutdown logic (ESD)

Digital technology also provides a more complete picture of the turbine and compressor performance to the engineers and operators through the use of PC computers and graphic display programs. Serial and Ethernet communications transmit and receive data from external devices that can perform logging and trending of data, event logging, sequence of event recording, and pictorial representations of the operating data specific to the turbine/compressor train and process. Even the compressor performance map can be displayed to the operator, along with the on-line performance data that pinpoint the operating point on the map. In this way, the operator literally has a picture of where the compressor is operating at all times.

Modern turbomachinery control systems integrate the control functions of the turbine, the compressor, and the process in order to make the machinery perform in such a way that it becomes an integral part of the overall process while at the same time providing the necessary protection for the equipment. Many plants today are required to operate from five to eight years between scheduled shutdowns, further justifying the need for reliable controls that can best interface with plant operations. Fault-tolerant, digital systems designed for high-speed control (Figure 2-1) are the logical choice for true state-of-the-art plants.

On-Stream Torque-Sensing and Hot-Alignment Monitoring Devices

Operating Principles. Most continuous torque-sensing devices for turbomachinery are based on the torsional windup of a shaft operating under load. This torsional windup results in angular deflection and strain, and both effects can be captured by sensors that are customarily built into or around the coupling spacer tube.[3]

However, several competing approaches can be used to sense deflection and/or spacer strain. One torque sensor consists of a four-arm strain gauge bridge laid to detect torsional strain. As a twisting load is applied to the shaft, a signal is created by the unbalancing of the strain gauge bridge. This signal is presented as voltage input to a rotary module located in a rotating collar, and is used to modulate a constant-

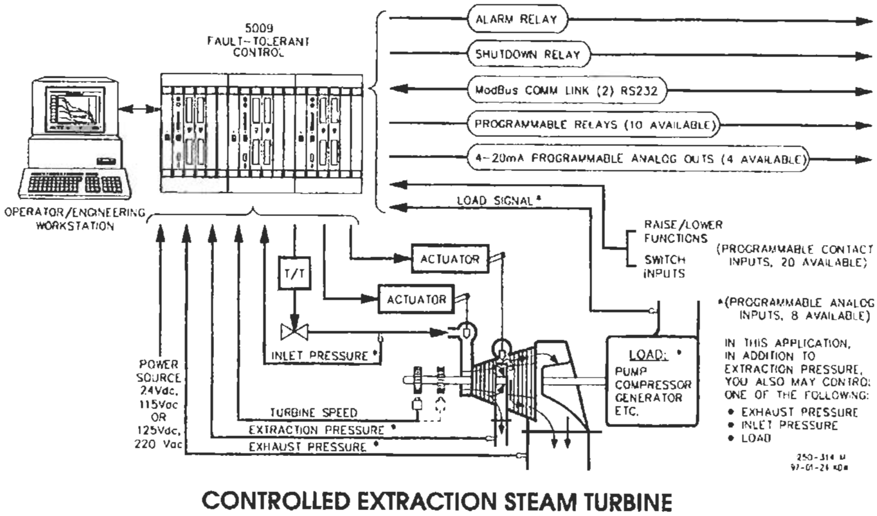

Figure 2-1. Fault-tolerant, digital control system for extraction steam turbine. ***(Courtesy of Woodward Governor Company, Loveland, Colorado.)***

amplitude square wave. The duty cycle of the square wave (i.e., pulse width) is directly proportional to the torsional load. A voltage-controlled oscillator is driven by the square wave. Its output, having been frequency modulated, is fed to a coupling loop imbedded in the rotating collar, and is then capacitance-coupled to a nearby stationary loop. The FM signal is carried via coaxial cable from the stationary coupling-loop base to a readout unit where it is demodulated, scaled, and displayed as torque in engineering units such as pound-inches, pound-feet, newton-meters, etc.

In another design, power to the strain gauge bridge and torque signals from it are transmitted through rotary transformers. AC power is fed to the shaft through one rotary transformer and then converted to DC to energize the strain gauge bridge and the shaft electronic circuits. A calibration control circuit periodically unbalances the strain gauge bridge to produce a precise calibration voltage that is fed through the same chopper and amplifier circuits as the strain gauge bridge torque signal and finally fed out from the shaft through a second rotary transformer.

Recognizing that shaft deflection is linearly related to applied torque, a third approach employs the phase-displacement technique to measure shaft twist, torque, rpm, and thus horsepower. Gears are attached to the coupling spacer tube and a slipped-on torque tube so that they displace when torque is applied. Variable reluctance sensors (proximity probes) are placed within 0.020 inch to 0.060 inch of the rotating gear teeth and produce electrical wave forms whose phase relationships are directly related to gear-tooth position and, consequently, shaft torque. The frequency of these signals is directly related to shaft rpm, which, in combination with shaft torque, provides a precise indication of shaft horsepower.

A typical installation illustrating this approach is shown in Figures 2-2 and 2-3.

Table 2-1
Comparison of Governors for Mechanical-Drive Turbines

Feature	Electronic Governor	Mechanical/Hydraulic Governor
Cost	At least two times base	Base
Sensitivity	Excellent	Very satisfactory
Dependability	Excellent. Malfunctions may develop with little or no warning. Failsafe devices are needed	Excellent. Malfunctions generally develop gradually.
Response to load changes	Excellent	Excellent
Complexity	Requires electronic engineer or first-class instrument-technician training	Will be understood by better operators and mechanics
Maintenance	By manufacturer's representative or resident electronic engineer/ instrument technician	By contract-maintenance specialist firms or resident mechanics
Spare parts investment	High, due to special nature of product	Low. Standard product available from affiliated process plants if urgent need should arise

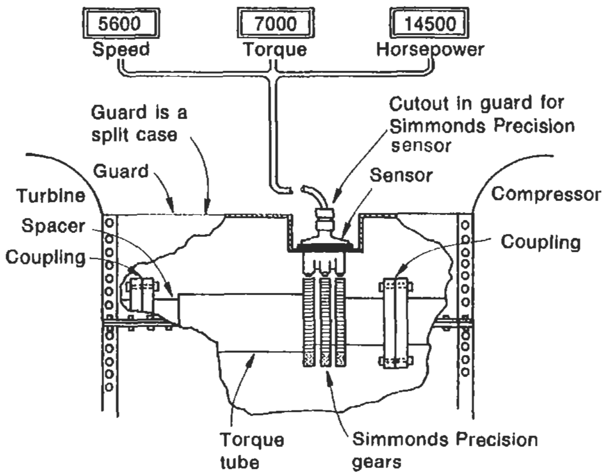

Figure 2-2. Typical installation schematic for phase-displacement-type torque meter.

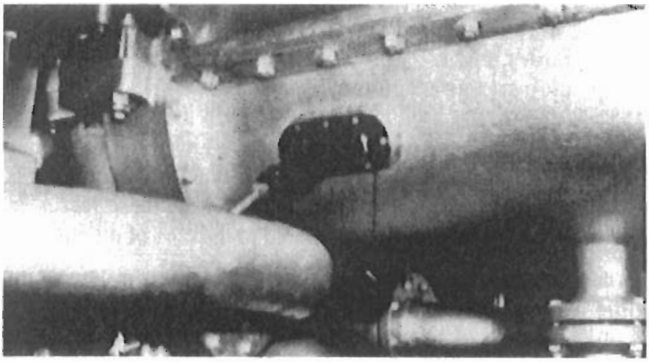

Figure 2-3. Phase-displacement-type torque-monitoring system installed on large turbomachinery train.

Torque Measurements Warn of Performance Deterioration. Performance deterioration of steam and gas turbines is often caused by fouling deposits on blading or nozzles. In steam turbines, evidence of these deposits is frequently not discovered until the steam flow increases to a point at which no additional steam can be passed through the unit and the turbine is no longer able to carry an assigned load. In gas turbine air compressors, fouling deposits will reduce the amount of air available for efficient combustion, and excessive fuel consumption or reduced load-carrying capacity may result. Similarly, process gas compressors are often subject to polymerization. The resulting flow impediment can seriously affect process operations and mechanical performance.

Turbomachinery performance can be restored by judicious application of on-stream cleaning methods.[4] Abrasion cleaning and solvent cleaning (water washing) have been identified as the two principal approaches. Details of suitable procedures and their relative merits have been amply documented in the literature[5,6] or can be obtained from the major equipment manufacturers. The problem to date has been to determine conveniently and accurately when to initiate the on-stream cleaning process. It is well understood that in pure economic terms, on-stream cleaning should commence when the cumulative cost of lost power due to fouling since the last cleaning cycle equals the cost of the cleaning procedure. This is where the on-stream torque-measuring system comes into play. Configured to sense torque, speed, and power at the load coupling, the device can be tied in with computer means accessing steam or fuel flow meters and heat-rate tables. For example, the average turbine efficiency could be calculated in one-minute intervals from:

$$\eta_e = \frac{N_{TK} \times 860}{\Delta i_{ad} \times D_o}$$

where

D_o = steam, kg/hr

N_{TK} = coupling power, kw

Δi_{ad} = enthalpy drop at constant entropy, kcal/kg

Knowing the "as-clean" efficiency of the turbine, we now also know the energy loss due to inefficiency in the one-minute interval. Since one HP minute equals 42.44 Btu, and one kw minute equals 14.34 kcal, the summation of these losses can be referenced to utility-cost debits and, finally, compared to the cost of on-stream cleaning.

Figure 2-4 can be used for screening studies showing the approximate cost of operating prime movers with an assumed efficiency decay for relatively short time periods. The potential cost of losing efficiency or output capability can be staggering. Howell and McConomy[7] documented a 7.2% capability loss that occurred on a 138 MW steam turbine within approximately 360 days. The same machine experienced a 3.6% loss in a 90-day period.

Axial-compressor fouling continues to be a common and persistent cause of reduced gas turbine efficiency. A 1% reduction in axial-compressor efficiency accounts for approximately 1½% increase in heat rate for a given power output. Even compressor stations that are not subjected to industrial pollutants or salty atmospheres are frequently prone to fouling. Typical of these conditions would be pipeline operations in the northern hemisphere where insects and fine dust are found to adhere to inlet guide vanes, compressor blades, and stator blading.[8]

Performance deterioration of gas turbines can be detected by combining turbine fuel flow rate with horsepower output. Display modules can thus incorporate readouts of horsepower produced per cfm of fuel consumed. Since the turbine manufac-

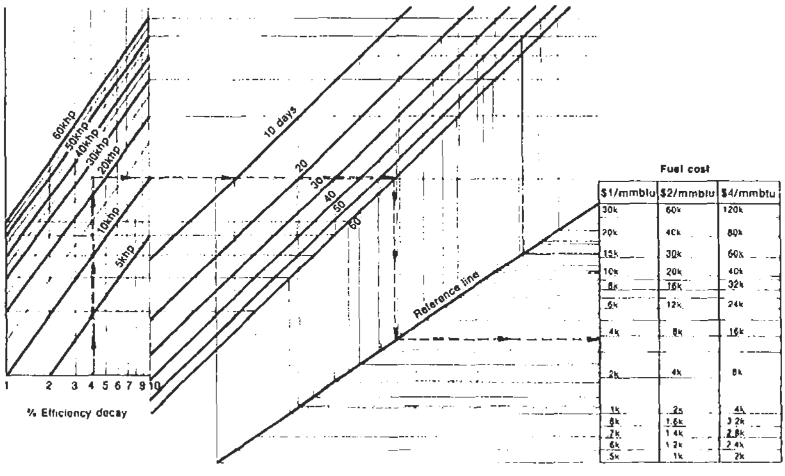

Figure 2-4. Cost of operating inefficient prime movers.

turers normally furnish test-stand verified performance curves (Figure 2-5), the degree of the performance deterioration is readily evident from a comparison of actual (fouled condition) and ideal (clean condition) fuel-consumption rates. As in the case of steam turbines, the average turbine efficiency, or average fuel wastage, could be calculated in convenient time intervals. The summation of energy losses can be referenced to fuel-cost debits and on-stream cleaning initiated when cumulative fuel costs outweigh cleaning costs.

In-depth performance analyses on pipeline drivers also established that exhaust-temperature measurement errors are virtually unavoidable on power turbines. These errors can be traced to slight changes in gas flow pattern and have been shown to cause considerable change in average temperature readings, even from a group of thermocouples. In turn, the errors can influence the power capability of a gas turbine if exhaust-temperature limits are observed or made one of the controlling parameters. With limit increases reported as high as 30°F (17°C), 7% increases in high ambient temperature power capabilities have been produced. The corresponding decrease in heat rates is 2%.[8]

The merits of performance monitoring via on-stream torque-sensing devices in combination with inlet-temperature and fuel-rate metering are thus evident.

Performance Optimization and Determination of Fault, Driver Versus Driven. Continuous torque-sensing devices can provide valuable information about other areas as well. While in the case of torque limitations on one or both of the coupled shafts the devices might serve as constraint controls, torque sensing may allow process optimization in computer-controlled compressor situations where several levels of refrigeration may be available.

In another application, turbocompressors may be configured to achieve desired flow and head by such methods as varying speed, varying stator-blade angle, and varying guide-vane angle. On large axial compressors, a combination of stator-blade

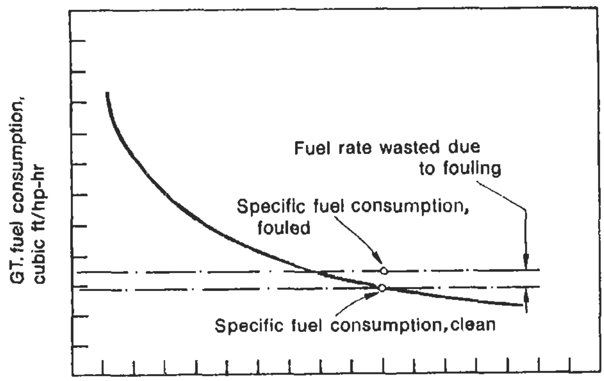

Figure 2-5. Gas-turbine fuel consumption versus output horsepower.

adjustment and speed variation of the order of 10%–15% not only improves the part-load efficiency of the compressor but increases the stable operating range.[9] Appreciable differences in energy consumption may result and savings of hundreds of horsepower realized if the optimum control mode is selected for axial equipment in the higher horsepower categories.

On-stream torque-measuring devices will also prove effective in determining whether increased energy input to the driver is caused by performance decay of the driver or the driven equipment. How the issue can be resolved by torque meters is best illustrated by the simple case of a gas turbine driving a centrifugal compressor. If high driver fuel consumption and high coupling horsepower are noted, the driven machine could be more highly loaded, mechanically deficient, or internally fouled. Diagnostic instrumentation or analytical procedures are available to determine which of these three possible causes is most probable. If high driver fuel consumption and normal coupling horsepower are noted, the most probable cause of the efficiency decay would be turbine fouling.

Hot-Alignment Monitoring Systems. Excessive misalignment between driver and driven machine can result in excessive vibration, high bearing loading, shaft breakage, and coupling failure. Proper alignment of turbomachinery requires knowledge of thermal growth in the radial direction of both machine casings. If diaphragm couplings are involved, changes in the axial position of the two machine shafts must be known as well.

One of the latest and potentially most advantageous methods of on-stream monitoring involves laser optics. The system shown in Figure 2-6 consists of a semiconductor laser (transducer) emitting a beam in the infrared range (wave-length 820 mm), a reflecting prism system, and a display system.[10] The laser beam is refracted through a prism and is caught by a receiver/detector. It is an intrinsically safe long-term monitoring system for continuous observation of alignment.

The system allows monitoring of critical machinery during operation so that changes such as foundation movements may be detected and dealt with in time.

It also facilitates measurement of changes in alignment due to thermal growth. This information may then be used to align the machines in the cold state with exactly the correct amount of compensating misalignment so that perfect alignment is reached when the machine warms up during operation. These measured values also allow users to compare and correct calculated growth figures for their machinery.

Each system permits two distinct alignment parameters to be observed at once. This means that two systems are mounted at 90° to one another to form a complete monitoring facility.

The system indicates information via a single display unit. Two distinct alignment parameters or other information, such as temperature, may be obtained from the display, whose display mode is switched by simply interrupting the laser beam (e.g., using one's hand). This display allows use of the monitoring system for general measurement tasks where minute displacements must be measured (e.g., civil engineering projects). (For more alignment information, see pages 463–477.)

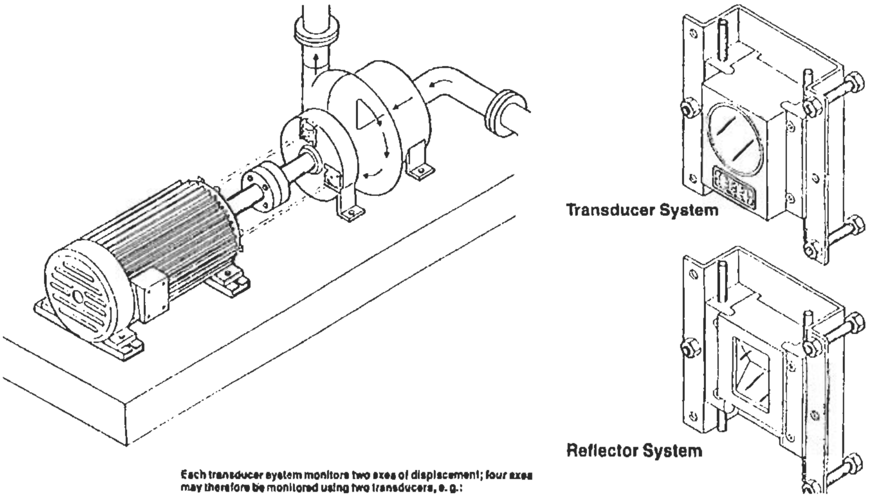

Figure 2-6. Hot alignment monitoring system. *(Courtesy Prueftechnik A.G., D-85730 Ismaning, Germany.)*

How to Review Critically Important Electronic Systems. As machinery systems become increasingly more complex and unforeseen downtime prohibitively expensive, reliability reviews must employ new approaches, techniques, and resources. Because defects in electronic governors, safety shutdown devices, and certain instruments can have a devasting effect on equipment reliability and plant profitability, the petrochemical industry will have to borrow its reliability review methods from the aerospace industry. One such review method is called "sneak analysis."

The Sneak Analysis Department of the Houston branch of the Boeing Aerospace Company defines sneak conditions as latent paths, timing, indication, or labels in electrical hardware or computer logic. The conditions generally exhibit unapparent cause-effect relations, and may inhibit a desired operation or initiate an unintended action. A sneak condition is not caused by component failure, but is a condition that has been inadvertently designed into a system. Some sneak conditions are evidenced as anomalies or spurious operative modes. Historically, sneak conditions have escaped rigid design screening efforts and caused commissioning delay or loss of equipment availability during operation or test.

Boeing's experience with sneak analysis in aerospace, military, commercial, and nuclear systems has shown that sneak conditions have distinct characteristics in all electrical and software systems. In *each* project (!) sneak conditions were found that had been missed by standard system verification procedures. They were found through application of a formalized, computer-aided, topological approach to detailed system analysis. Further, the analysis provided procedure checks, drawing and document error disclosures, and design concern conditions.

Sneak analysis identifies the proper and improper operation of a system's hardware and software. The analysis provides a systematic, consistent, and thorough review of the system's current and logic paths, down to the individual statement and component level. Sneak analysis is not restricted to critical functions (as are many other analyses) but analyzes the complete system, function by function. The analysis will identify the cause and recommend a solution to a sneak condition, design concern, or document error before the problem occurs. If left undetected, sneak conditions occurring during testing usually result in program delays to the project.

The sneak analysis process generates detailed functionally oriented patterns, called network trees, of the circuitry and software that can be reviewed individually or in groups to understand the system. These network trees not only make sneak analysis possible, but are a powerful tool in reducing the cost and improving the quality of other reliability and safety analyses, such as Failure Mode and Effects Analysis, Hazard Analysis, Fault Tree Analysis, Common Cause Failure Analysis, or Mean-Time-Between-Failure Analysis, which may be specified on a project.

The major benefits derived from the performance of sneak analysis are:

1. Savings of overall project dollars.
2. Increased confidence in system safety, reliability, and operability through independent design verification.
3. Fewer system development delays.

Identification of sneak conditions early in the project life cycle can provide cost savings as a result of changing a circuit or logic path on paper—rather than changing actual hardware or software. Empirical data obtained after performing a sneak analysis demonstrates that increased reliability and operability of the system occur when corrections for identified sneak conditions are made.

Selecting a Pump Vendor

The broad principles governing the selection of major machinery vendors were stated earlier in the chapter. How these principles can be applied most advantageously is again best illustrated in a typical example. Take pumps, for instance.

An up-to-date edition of *Thomas Register of American Manufacturers* will list dozens of pages of pump manufacturers. Their detailed product listing contains another 100 or so pages ranging from "Pumps, Acid," to "Pumps, Wine." Even after reducing the potential bidder's list to manufacturers of refinery-type centrifugal pumps, there remain some 20–30 vendors who could be invited to bid on a given project or for a given pumping service. Were all of them to be considered, much time and money would be spent on preparing bid specifications, providing the necessary vendor liaison, and finally evaluating the profusion of bids received.[11]

The need to limit bidding to a few capable vendors is quite evident. But what constitutes capable vendors? What criteria should be applied to narrow down the selection to manageable size? How many bids are manageable? This segment of our text attempts to give guidance in this regard. It explains selection procedures that have given satisfactory results in a large number of major refinery and chemical plant construction projects. More important, though, it shows how vendor selection and equipment selection criteria interact and must be given simultaneous consideration.

Standardization Within What Limits?

A petrochemical plant obviously would not find it practical to purchase equipment from too many manufacturers. Spare parts identification, procurement, and warehousing are expensive and leave margin for error. Also, it would be progressively more costly and difficult to train mechanical workforces for full proficiency in too many equipment types or models.

On the other hand, standardizing on too few manufacturers may deprive the user of optimally selected equipment. There is obviously no single manufacturer of centrifugal pumps who can lay claim to products that are consistently more efficient, easier to maintain, and more rugged than competitive equipment.

Experience shows that two or three vendors could adequately cover the on-site pumping services of a typical petrochemical plant. Five or six manufacturers should be invited to submit bids, and two or three of these subsequently selected for contract award. Off-site pumps, frequently of ANSI or ISO-type, could be selected from one additional vendor among those who had been invited to bid.

Highly specialized centrifugal pumps for critical services, such as high-pressure boiler feedwater or pipeline supply pumps, may have to be purchased from the most experienced source, regardless of whether the vendor is among those selected for on-site and off-site pumps. Purchase of these special pumps should be handled separately on an individual basis, rather than being lumped with other pumps.

Assessing Vendor Experience

Three principal characteristics identify a capable, experienced vendor:

1. He is in a position to provide extensive experience listings for equipment offered.
2. His marketing personnel are thoroughly supported by engineering departments. Both groups are willing to provide technical data beyond those that are customarily submitted with routine proposals.
3. His centrifugal pumps enjoy a reputation for sound design and infrequent maintenance requirements.

With a large project involving 200–300 or more pumps, it is often necessary to delegate to the contractor's equipment engineers the responsibility of verifying vendor experience with other users. For certain critical services (e.g., high-pressure boiler feedwater pumps, large cooling-water pumps, multistage pipeline or feed pumps, etc.), the owner's engineer would be well advised to check for himself. When procuring pumps that are required to comply with the standards of the American Petroleum Institute (i.e., API 610), a capable vendor will make a diligent effort to fill in all of the data requirements of the API specification sheet. However, the real depth of his technical know-how will show in the way he explains exceptions taken to API 610 or to supplementary user's specifications. Most users are willing to waive some specification requirements if the vendor is able to offer sound engineering reasons, but only the best qualified centrifugal pump vendors—those from whom you want to purchase—can state their reasons convincingly.

Concentrating on Problem Applications

In assessing vendor experience, the engineer responsible for vendor selection should concentrate on pumping services that have a history of being troublesome. The approaches proposed by the various bidders for solving typical problem applications may differ drastically and allow rapid separation of sound proposals from potentially troublesome ones.

One of the most common problems in pump application is insufficient net positive suction head available. Often this is not realized until vendors' proposals have been solicited and some vendors have failed to meet the conditions.

When NPSH is insufficient, the plant design should be reviewed to determine whether decreasing the length or increasing the size of the suction line is feasible. Raising suction-vessel elevations is another possibility, but economics may dictate selection of pumps designed especially for low NPSH. When NPSH availability is limited, the pump vendors, of course, will offer double suction pumps when possible. Also to be considered are inducer-type impellers, but it must be realized that inducers may have a limited flow range relative to a non-inducer-type pump. Also, inducers should never be used in erosive services. Pump performance deteriorates rapidly as the effectiveness of the inducer is reduced by erosion, and cavitation begins to take place.

Another solution to NPSH problems is to use a vertical deep-well pump. In order to minimize maintenance problems with this type of pump, each proposed pump should be checked for applicable service experience, with attention to exact model numbers, similar pumpage characteristics, and process conditions. High-head, low-capacity services also pose problems to engineers selecting pumps. In general, four types of pumps are available for this application: multistage horizontal centrifugal, multistage vertical centrifugal, reciprocating, and single-stage high-speed centrifugal. For process units in continuous duty, experience has shown that multistage vertical and reciprocating-type pumps require more maintenance than the other two types. If a pump must handle a wide range of specific gravities, or if the fluid is particularly viscous, a reciprocating pump may be the only answer. If there is limited NPSH available, a multistage vertical pump may be required. But for the bulk of high-head, low-capacity applications, serious consideration should be given to high-speed pumps, with the multi-stage horizontal centrifugal pump a good second choice.

Mechanical Seal Selection and Evaluation

Pump-application engineers will generally agree that seal and seal environmental system selection on many pumps is becoming more complex and time consuming than pump selection.[12] Still, an average of only 10%–35% of the allotted pump-engineering time is generally spent on the mechanical seal system.[13] Looking at the cost of seal failures and such consequences as major fires, release of toxic materials, and unit downtimes, there is reason to believe that more time should be spent on seal systems design. But before we decide *who* should spend this time, we should examine the various selection practices prevalent in the petrochemical industry.

Seal selections made entirely by the pump vendor have generally proven to be least reliable. The pump vendor is concerned that his competitor will underbid him, and thinks that the engineer selecting the pump will only look at the initial, installed cost without giving credit to the potential run-length extension and maintenance cost avoidance of superior seal components or seal system designs. Consequently, the least expensive seal is often selected, leaving plant operations or maintenance burdened with an inherently weak seal. Furthermore, pump manufacturers seldom receive experience feedback on seals furnished with their pumps. Seal selection by the pump vendor alone should thus be discouraged.

Contractor's or user's standards have generally been applied with somewhat higher success. Unfortunately, many of these standards are full of generalities and give little guidance on specific requirements. Very often the stated requirements do not separate barely acceptable from truly successful seal systems. Lack of specific guidance in an otherwise well-intended specification may significantly impair its usefulness and deprive the user of a low-risk sealing system.

Optimum seal selection practices should make extensive use of vendor experience. These practices must encourage the seal vendor to use his own gland design and to recommend *seal systems,* not just seals. To properly advise the user, seal vendors require full information on product composition, process conditions, crystallization temperatures, solids entrainment, and the like. All of these data are highly relevant if proper selection is to be ensured, and withholding data for "security" reasons may cost the user dearly. Optimum seal selection consists of the following steps:

- All relevant data must be disclosed to the seal vendor. If security is truly a valid concern, disclosure should be preceded by signing confidentiality agreements.
- At least three and preferably four major seal manufacturers with strong and capable representation in the user's geographic location should be invited to submit bids. The user should screen and verify vendor capability by such criteria as ability to furnish *engineered* seal components, e.g., special pumping screws instead of ineffective pumping rings, and by vendor's willingness to stock appropriate spares in the user's geographic area.
- The bid invitation should clearly state that the user is interested in buying a *seal system,* not just a seal.

- The bid invitation should instruct the vendor to propose a minimum of seal types. For example, better-than-necessary seal-face materials should be offered for some services, or bellows seals should be considered for an expanded range of applications in order to reduce the user's spare parts inventory.
- Proposals obtained from the various bidders must be examined critically for technical differences, and pertinent deviations should be noted for follow-up by the user's resident expert. These technical differences must be investigated by requesting that vendors submit experience data. These data should include the names of other users who may be contacted to obtain verification of satisfactory experience. The user's resident expert should make these contacts.
- Whenever possible, seal selection and evaluation procedures should make use of in-house feedback as to principal reasons for seal failures at that location. These data should be discussed with the vendor and solutions proposed.
- The user's priorities should be clearly established and transmitted to potential seal vendors. Safety, long life, standardization, and ease of maintenance all precede cost in order of importance.

These last topics merit further elaboration. Experience shows that the emphasis should clearly be towards standardization without sacrificing reliability. Seals should be cartridge seals that are simple to install and maintain. They should not be vulnerable to minor deviations in pump operation or properties of pumpage. Indeed, a given seal should suit the needs of many pumping services so as to reduce spare parts inventory, streamline training requirements and leave little room for maintenance errors.

These goals can be met with a well-defined selection strategy. Such a strategy may well result in the added bonus of greatly increased seal reliability and considerably lower maintenance expenditures for most petrochemical plants. Many major manufacturers of mechanical seals have indicated their support and willingness to cooperate in implementing our selection strategy. Specifically, this strategy requires the development of bid request information that asks the seal vendor to identify optimum seal configurations. Principal features of optimum seals are highlighted in the following pages; for additional details see Chapter 13.

Development of Bid Request Package

The development of bid request information is a key element in a selection strategy leading to the procurement of mechanical seals that comply with the user needs previously outlined. Bid requests must be forwarded to several experienced seal manufacturers and must clearly state that the manufacturer should submit cost proposals for only those services where his seal selection is backed by solid experience. His inability to furnish seals for some services should in no way disqualify him from submitting bids for those services where he is competent to provide a good product.

To begin with, the user should assemble API data sheets for the various pumps that need mechanical seals.

This pump tabulation must include and disclose all of the fluid properties and operating parameters known to the user. With disclosure thus going beyond the typical contents of API data sheets, the seal manufacturer can be requested to list the "operating windows" within which the proposed seal will function reliably in the pump tentatively selected. It is to be understood that the "operating window" refers to the *actual* seal with the seal materials, balance ratio, flush plan, stuffing box layout, etc., selected and disclosed by the vendor. Notice also that the *pumps* are designated "tentatively selected" because even a capable seal vendor may be unable to offer his optimum seal for a given pump. Should this be the case, it might be appropriate to consider selecting a more suitable pump model for the intended service.

As mentioned earlier, it is recommended to mail the bid request information to three or four capable mechanical seal manufacturers. Their response should be critically analyzed to ferret out significant deviations among the various proposals. These deviations may range from materials of construction to different API flush plans and from differences in basic configuration to differences in application philosophy of stationary vs. rotating seal members. Reconciling the deviations or differences will assist the engineer responsible for final selection in determining whose seal offer has the best chance of meeting the user criteria highlighted earlier.

Desirable Design Features Identified

Evaluation of the various bids is made easier by recognizing desirable design features incorporated in mechanical seals. Some of these merit closer consideration.

Cartridge Construction. Cartridge seals are designed for rapid installation on and removal from pump shafts. The cartridge seal is an arrangement of seal components on a shaft sleeve and in a seal gland constituting a single unit that is usually assembled and pre-set at the factory. Both bellows and spring-type seals can be cartridge-arranged if the pump stuffing box is large enough.

Cartridge seal units offer major maintenance advantages. Replacement is rapid and there is far less risk of assembly error and assembly damage than with conventional mechanical seal mounting. API 682 requires that cartridge seals be selected for typical pumps in petrochemical plants.

Silicon Carbide Hard Face Material. Heat generated at the seal faces must be rapidly conducted away if fluid vaporization and resulting problems are to be avoided. Depending on service conditions and pump design, either the rotating or stationary seal ring must be counted on to dissipate as much frictional heat as possible. High thermal conductivity and hardness make silicon carbide the preferred seal face material in many of the more severe applications.

Placement of O-Rings. An advantageous seal design recognizes that O-ring life can be reduced by close proximity to the heat source, by swelling due to chemical attack, and by operation in a dynamic mode, especially in the presence of erosive materials. Going to PTFE chevrons or wedges may allow operation at higher temperatures and reduced risk of chemical attack, but will lead to fretting of metal surfaces in contact with it. Bellows seals eliminate many of these problems by using static secondary sealing. Seals with spring loaded running faces are forced to use dynamic means of secondary sealing which could, in some instances, be more prone to failure. This potential problem can be overcome by selecting "stationary" and/or gas seal designs. In a stationary seal, the spring-loaded face is not rotating.

Mechanical Design Considerations. Important differences can exist in the mechanical designs of competing vendors. For instance, execution E of Figure 2-7 shows the method of clamping rotating hard face (1) against shaft sleeve (2) of a stationary bellows seal assembly. This clamping method ensures perpendicularity between shaft centerline and rotating seal faces. However, this clamping method also invites distortion at the running faces. Execution F tends to avoid distortion by mounting the rotating hard face in a resilient backing ring (3) and by pulling mounting ring (4) against collar (5).

Two different clamping methods are shown in Figure 2-8. Both of these methods were devised to eliminate distortion of running faces. However, in execution G the collar is set-screwed to the shaft or shaft sleeve, whereas in execution H the mating ring carrier is set-screwed to the shaft or shaft sleeve. Experience shows that perpen-

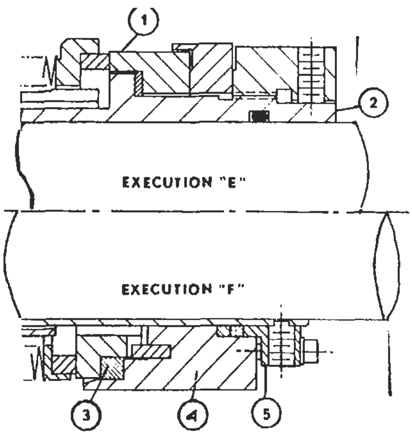

Figure 2-7. Two different mounting methods for rotating hard faces in mechanical seals.

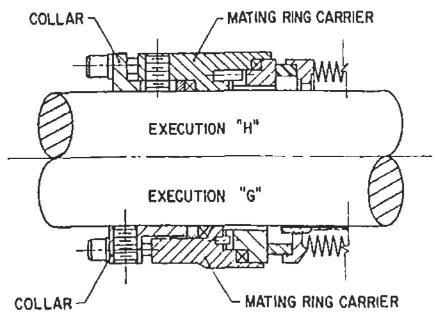

Figure 2-8. Mechanical seal face clamping methods devised to reduce risk of distorting rotating face.

dicularity and seal setting accuracy are more difficult to achieve with the clamping method indicated as execution G in Figure 2-8, which requires very careful adjustment of cap screws inserted through the collar.

Another interesting difference exists in the carbon holders of the more conventional mechanical seals. The upper half of Figure 2-9 shows a lock ring (1) executed with a set screw (2) that engages a slot in carbon holder (3). Under certain service conditions, contact between set screw and slot may cause a wear pattern that may prevent proper seal operation. Moreover, tightening of the set screw can distort the relatively thin lock ring and cause contact or interference between lock ring OD and carbon holder ID. The construction features shown in the lower half of Figure 2-9 would tend to eliminate both of these potential problems by providing an axially oriented drive pin (4) and a considerably heavier lock ring (1). Both designs shown in Figure 2-9 deserve credit for presenting relatively smooth, low turbulence surfaces to their respective fluid environment. This is largely accomplished by locating the springs on the atmospheric side of the seal.

Whenever possible, seals should avoid having the spring (or springs) immersed in the fluid. Figure 2-10 shows two of several seals that probably give excellent service in many services and applications. These seals use one or more stationary springs and incorporate several desirable features: a cartridge arrangement for ease of installation; the single non-rotating spring shown with the design in the top half of Figure 2-10 is arranged to operate away from the product; a bronze spring retainer (11) serves as a throttle bushing; and the relatively clean profile inside the stuffing box

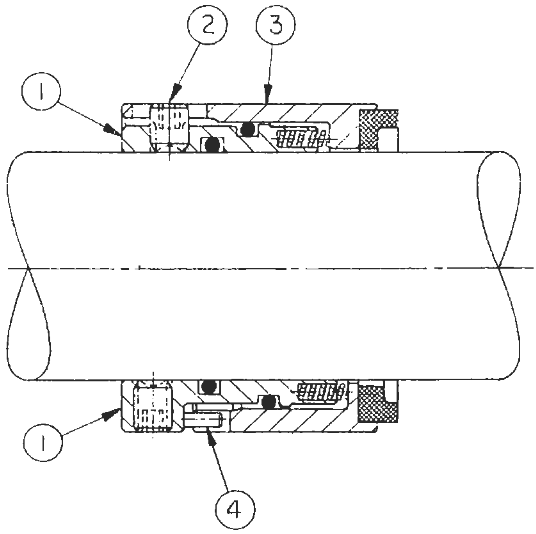

Figure 2-9. Two conventional mechanical seals with carbon holder driven by modified set screw (upper half) and horizontal pin (lower half).

reduces seal drag. (Note that rotating components are identified with even, and stationary components with odd numbers.)

Except for using several springs, the seal design shown in the bottom half of Figure 2-10 appears to be quite similar to the design shown in the top half. Both seals incorporate the desirable features of many similar stationary seal designs:

- Self-squaring faces. This feature may result in appreciably better seal life for pumps with excessive shaft deflection or pumps operating with nominal shaft deflection at high speeds.
- Non-flexing springs. Spring life extension and long-term, uniform pressure can be expected.
- Pre-assembled cartridge construction. These seals can be shipped with the gland plate in place. No field measurements or settings will be required.

However, a closer look will show functional differences in the arrangement of the O-ring (3). It could be argued that progressive wear of the seal faces shown in the upper half of Figure 2-10 will cause the O-ring to make sliding contact with a clean portion of part (9), whereas advancing the stationary seal face in the lower half of Figure 2-10 will cause the O-ring (3) to slide over a wetted and potentially contaminated portion of the gland plate (15).

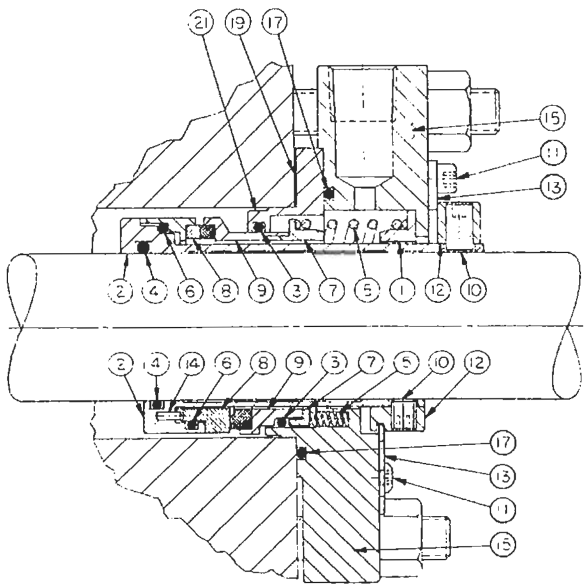

Figure 2-10. Two different stationary seals with non-rotating springs located away from pumped liquid.

Functionally similar features can also be found in an intermediate range of bellows seals as shown in Figures 2-11 and 2-12. Rotating bellows seals tend to be self-cleaning by virtue of centrifugal action. As mentioned earlier, they do not incorporate sliding (dynamic) elastomers. Instead, they use static secondary sealing means.

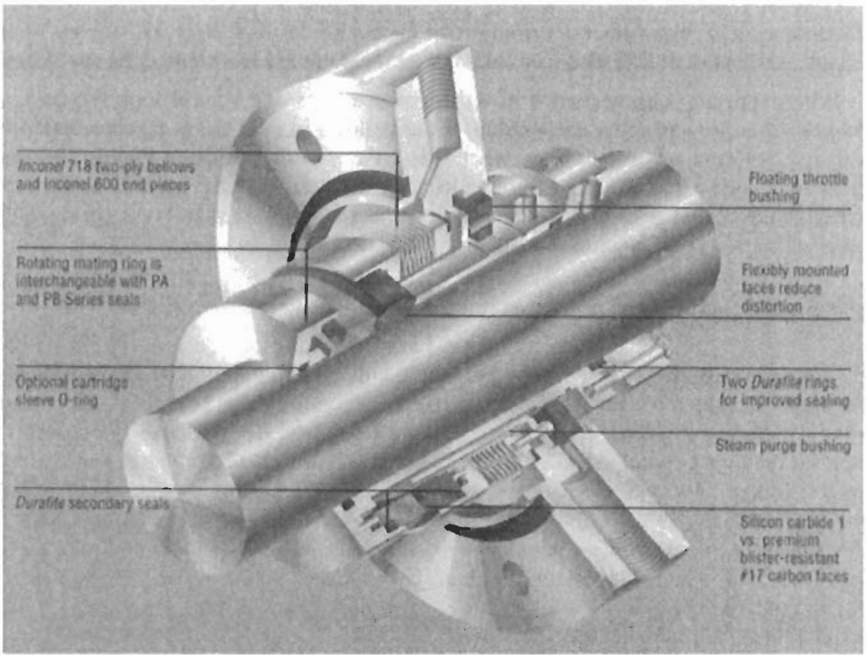

Figure 2-11. Bellows seal per API 682. (*Courtesy Flowserve Corporation.*)

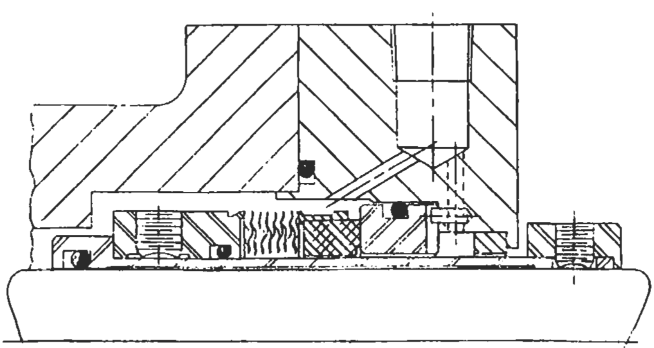

Figure 2-12. Cartridge-mounted bellows-type mechanical seal. (*Courtesy Borg-Warner Seals.*)

Properly designed stationary seals have given outstanding service in abrasive slurry applications. Figure 2-13 shows one such seal installed in a cone-shaped environment. This short "tapered stuffing box" promotes the outward flow of abrasive particles and at all times allows new, fresh fluid to contact the balanced single-acting, dead-ended seal. It should be noted that *long* tapers will require vortex-breaker ribs; these are usually cast or welded into the housing bore.

Generally, stationary-type seals or gas seals should be preferred in applications encountering face peripheral velocities in excess of 4,500 feet (1,372 meters) per minute, serious shaft deflection, or coking after the pumpage has crossed the seal faces.

When applying bellows seals in light hydrocarbons, we should look for design features that prevent torsional windup of the bellows in case the seal faces undergo slip-stick motion relative to each other. Also, the seal face balance line of bellows seals may shift when applied without adequate forethought in light hydrocarbon services. If the design does not alleviate these concerns, the user may favor spring, and especially gas-type mechanical seals for light hydrocarbon services.

Bid Comparison

After the bids are received, they must be tabulated and compared. The reviewing engineer should look for significant differences among the competing bids and should determine which offer incorporates most of the desirable design features. Special features beyond those specified by the purchaser may have been proposed by some bidders and would deserve extra credit for reducing the risk of catastrophic failure incidents. Each special feature must be given a separate assessment of value. Alternatively, the purchaser may decide that bidder X's offer is less expensive than, but nevertheless technically equal to the offer made by bidder Y. He may now wish to upgrade his selection by asking X to furnish the seals with optional, although not previously specified features.

Using optimized seal selection procedures as outlined requires close cooperation between seal user and seal vendor. This should not be too difficult to achieve and the benefits to both parties should be quite evident. In this cooperative effort, the user has the responsibility of disclosing to the vendor fluid properties, application parameters, permissible leakage rates, and even maintenance practices. This disclosure will allow the seal vendor to obtain a better feel for user know-how and sophistication. The vendor can then effectively plan ways to assist the user. He can remind the

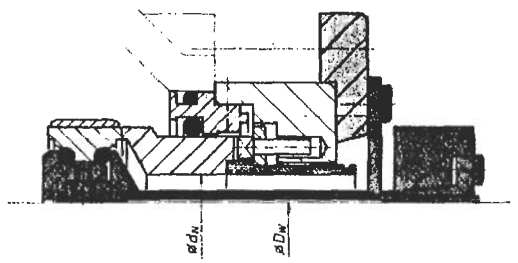

Figure 2-13. Stationary seal with successful experience record in abrasive-containing services. *(Courtesy Burgmann Seals America, Houston, Texas.)*

user that there are perhaps elements of pump design which will influence the success of even the best available seal system. In some cases, the seal manufacturer will have to deal directly with the pump manufacturer in efforts to define such pump modifications and might be necessary to accommodate optimum seal configurations.

The results of these efforts are shown in Reference 13, which documents significant increases in seal mean time between failures at three British oil refineries. This is plotted later in Figure 5-4.

Gas-Lubricated Mechanical Seals for Pumps. The gas sealing technology used in gas compressors has been successfully applied to the emission-free sealing of liquid pumps since the early 1990s. Gas-lubricated seals are thus an option to be weighed against both conventional seals and such higher-cost alternatives as canned motor and/or magnetically driven sealless pumps. While sealless pumps have their place, the potential purchaser should carefully study anticipated installation and operating and maintenance costs before making an informed choice.

Gas-lubricated seals employ a pressurized gas supply system and are available as single or double seals. Figure 2-14 shows a double mechanical gas seal in the unvented seal chamber of a conventional centrifugal pump. The seal or barrier gas, typically nitrogen, is introduced at a pressure of 0.2 Mpa (3psi) higher than the product pressure. Since many single-type gas seals find application as safety seals, the reader is referred to Chapter 13, pages 550–558, which describe these seals in greater detail.

Good Proposal Data Lead to Comprehensive Bid Tabulation

As mentioned earlier, it is essential that completed API data sheets be submitted with each proposed pump. The proposal package must also include performance curves and typical pump cross-sectional drawings. In addition to these, the vendor must state minimum allowable flow and NPSH required ($NPSH_R$) for the entire capacity range. Because minimum allowable flow could be governed either by thermal or mechanical considerations, the vendor should be asked to specify his basis.

With these data and any notable exceptions given by the various bidders, a comprehensive bid tabulation can now be constructed. Careful review will narrow this bid tabulation to two or three principal manufacturers, as illustrated in Figure 2-15. These would be the manufacturers whose equipment more consistently offered high performance by demonstrating such features as low risk suction specific speeds and operation near the best efficiency point (BEP). (For a more detailed treatment of this topic, refer to Chapter 3, "Reliability Review for Centrifugal Pumps.")

Engineers involved in pump-failure analysis have long suspected that cavitation erosion can plague even pumps in pumping circuits where the available net positive suction head ($NPSH_A$) exceeds the manufacturer's certified required value, $NPSH_R$. $NPSH_R$ tests by vendors are principally concerned with determining the performance drop-off point. A drop of 3% in total dynamic head is usually considered indicative of cavitation, and whatever NPSH is available at that point is though to be the NPSH required by the pump. However, it should be realized that significant mechanical damage may be encountered by long-term operation of some pumps with inadequate

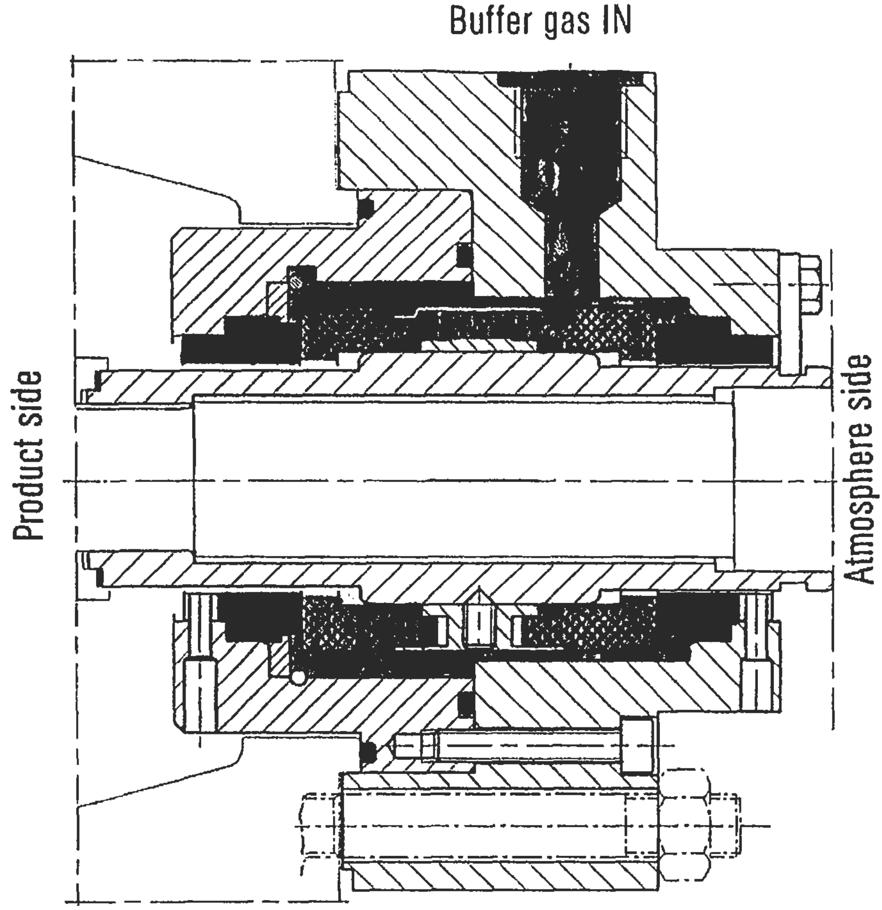

Figure 2-14. Gas-lubricated mechanical seal (double seal). (*Courtesy of Burgmann Seals America, Houston, Texas.*)

margin between $NPSH_A$ and $NPSH_R$. These findings have prompted many users to develop and use tentative guidelines that are largely empirical in nature.[14] The publication of quantitative correlations of suction recirculation as a function of proximity to best-efficiency-point (BEP) flow for pumps with different construction and performance characteristics had to wait until late 1981. Details are given in Chapter 3.

Bid Tabulation and Bid Conditioning: An Overview

Bid *tabulations* represent the culmination of the engineer's specification efforts. Proposals are submitted by the various vendors and pertinent items are written up side by side to facilitate easy comparison. To the experienced engineer, the bid tabu-

BID TABULATION

PAGE ___ OF ___

APPROPRIATION NO.

SPECIFICATION NO.

TITLE CLARIFIER SLUDGE REMOVAL PUMP ZP-02 A/B

FLOW RATE: 285GPM	DISCH. PRESS.: 75 PSIG	SPECIFIC GRAV.: 1.05	PRESS. DIFF.: 75PSI
TOTAL HEAD: 182 FT	PUMPING TEMP.: 93°F	NPSH AVAIL: 25 FT	SUCT. PRESS.: 5 PSIG

MANUFACTURER					
	I	II	III	IV	V
• PUMP					
SIZE & TYPE	18-4	12L-4	10AK	8"GH	HC/6
ORIENTATION	VERTICAL	VERTICAL	VERTICAL	VERTICAL	VERTICAL
STAGES/VOLUTE	4/	3/	4/	7/	6/
MATERIAL CASE/TRIM	CAST IRON	NI-RESIST/STEEL	NI-RESIST/STEEL	NI-RESIST/STEEL	NI-RESIST STEEL
SEAL TYPE	BF-IDI	BF-IDI	XF-IDI	BF-IDI	BF-IDI
API FLUSH PLAN					
API COOLING PLAN					
CONNECTIONS					
SUCT. SIZE/RATING	OPEN W/ STRAINER	OPEN/WITH STRAINER	OPEN W/ STRAINER	OPEN W/ STRAINER	OPEN W/ STRAINER
DISCH. SIZE/RATING	4"/125#	4"/125#	4"/300#	6"/125#	6"/
LOCAT. SUCT./DISCH.	BOTTOM/SIDE	BOTTOM/SIDE	BOTTOM/SIDE	BOTTOM/SIDE	BOTTOM/SIDE
IMPELLER BID DIA./MAX. DIA.	3.72/3.84	7.75/8.75	7.5/7.875	5.06/5.31	5.75/6.12
NPSHR, FT.	14FT	7FT	4FT	6FT	6FT
RPM	3550	1760	1760	1760	1760
EFFICIENCY	77%	79%	79%	78%	77%
BHP DESIGN	20	19.0	19.1	19.6	21
• DRIVER					
TYPE	INDUCT	INDUCT	INDUCT	INDUCT	INDUCT
MFR	SIEM/ALLIS	SIEM/ALLIS	SIEM/ALLIS	GEN. ELECT.	SIEM/ALLIS
POWER, HP	25	25	25	25	25
ENCLOSURE	TEFC	TEFC	TEFC	TEFC	TEFC
FRAME	284LPH	284LPH	256LP	284LP	284LP
• PRICING					
PERF. TEST		NOT REQ. 500.-			NOT REQ. 179.-
HYDRO					NOT REQ. 326.-
DISASS'Y AFT. TEST					
SEAL FLUSH PIPE					
CW PIPING	N.A.	N.A.	N.A.	N.A.	N.A.
CLOSE SIGHT FLOW INDICATOR	N.A.	N.A.	N.A.	N.A.	N.A.
VENT & DRAIN PIPE	N.A.	N.A.	N.A.	N.A.	N.A.
PUMP BASIC		* 14073.-	15880.-	10245.-	15032.-
DRIVER		1474.-	1610.-	1342.-	1053.-
FREIGHT		(EST) 250.-	205.-	(EST) 250.-	(EST.) 260.-
TOTAL		15797.-	17695.-	11837.-	16345.-
HP DEBIT	N.A.	BASE	BASE	(EST) 600.-	(EST) 2000.-
BASEPLATE DEB	N.A.	N.A.	N.A		N.A.
EXPEDITING		2000.-	BASE	BASE	BASE
DELIVERY PROM.		24 WKS	28-30 WKS	26 WKS	26-28 WKS
		* EX DISCHARGE HEAD & MOT. MT			
ADJUSTED TOTAL		17797.-	17695.-	12437.-	18345.-

RECOMMENDATION: PURCHASE PUMPS OFFERED IN COLUMN IV. PROVIDE DIMENSIONAL DATA SHOWING LOCATION OF PIT OPENING RELATIVE TO DISCHARGE PIPE FLANGE AND REQUEST VENDOR TO FABRICATE APPROPRIATE TRANSITION PIECE

NOTE: THE TWO PUMPS LISTED IN COLUMNS II & III WOULD REQUIRE ENLARGEMENT OF PIT OPENING

ESTIMATED COST	QUOTED COST	NOTED BY	PREPARED BY	DATE
$20000 EACH	$11837.-			1/23/80

Figure 2-15. Bid tabulation for pumps.

lation is far more than a mere comparison: it is a master checklist of relevant items, some of which can have a profound influence on equipment safety, reliability, maintainability, and availability. Bid tabulations highlight the strengths and weaknesses of certain proposals. They indicate which items are acceptable or unacceptable, which items require follow-up, or which items require that the vendor be challenged outright to explain inconsistencies, design extrapolations, or oversights.

Bid *conditioning* refers to the assignment of dollar credits and debits for strengths and weaknesses, pluses and minuses, in a vendor's offer. Credits and debits may be assigned to reflect differences in contractually defined operating efficiencies, component strength differences, uprate capability, tolerance for occasional overloading, better or inferior maintainability, likelihood of delivery delays, extra expediting and inspection efforts, field service capabilities, spare parts requirements, etc.

Bid-conditioning efforts for large unspared turbomachinery can be extensive and time consuming. Inevitably, though, the exercise will be worth the effort. Figure 2-16 shows a fictitious bid tabulation sheet for centrifugal compressors. Figure 2-17 explains some of the many entries for which credits and debits must be assigned, or which must at least be considered before a final "conditioned" value can be calculated for each of the three offers.

We have alluded to cost considerations other than pure bids prices. With progressively higher energy costs, pump efficiency becomes a major factor. In some locations, the yearly cost (in 1998) of one horsepower now often exceeds $500, and one might have justifiable concern that quoted efficiencies could be falsely inflated. Mere consideration of quoted efficiencies should be replaced by certified test-stand efficiencies, field feedback, and perhaps contractual penalty clauses. Credits and debits for efficiency deviations must compare the future value of money and the anticipated operating life of pumps. Depending on the rate of return acceptable for energy conservation on the project, the value of one horsepower saved may justify an incremental investment of several times $500—perhaps $4,000 in 1998 dollars.

Credits can be assigned also to recognize lower maintenance costs. If it is possible to make such an assessment based on comparisons of repair costs, the credit or debit number can be used outright. If only repair frequencies are available for comparison, it should be remembered that average pump repairs often cost in excess of $7,000 per event (including shop labor, materials, field labor, and burden) in 1998.

There is value also in demonstrably better, heavier, more easily groutable baseplates for large horizontal centrifugal pumps. Again, a rule of thumb: $1,200.

Unlike a bid tabulation sheet listing only API data and cost, bid conditioning serves to bring all offers on the same common denominator by assessing all relevant parameters. The bid tabulation, shown in Figure 2-15, illustrates this supplementary input. It can be seen that more often than not, the best *centrifugal pump* is neither the most expensive nor the least expensive one on the bid slate. The best pump *manufacturer* is one who realizes that addressing a user's concern allows him to outflank the competition. And the most capable *reliability engineer* is one who heeds the advice given by John Ruskin: "There is hardly anything in the world that some man cannot make a little worse and sell a little cheaper, and the man who buys on price alone is this man's lawful prey."

BID TABULATION FOR CENTRIFUGAL COMPRESSORS

		Vendor		
Item #	Description	A	B	C
1	Compressor Model	X-100	Y-101	Z-102
2	Type and Split	Hor	Hor	Barrel
3	No. of Impellers	8	7	8
4	Total No. of Impellers per Section	4/4	4/3	4/4
5	Diameter of Impellers	D(A)	D(B)	D(C)
6	Speed at Normal Point (rpm)	8217	8305	8180
7	Max. Continuous Speed (rpm)	9039	8720	8589
8	Tip Speed at Normal Point (ft/s)	920	900	870
9	Tip Speed at Max. Cont. Speed (ft/s)	1012	945	914
10	1st and 2nd Critical Speeds (rpm)	6217/18010	6847/19807	6617/21530
11	Absorbed Power at Normal PT (BHP Compressor)	12750	12813	12955
12	Driver Output Power @ Normal PT (BHP Driver)	12802	12900	13005
13	Max. Disch. Temp. @ Max Cont. Speed (°F)	292	318	298
14	Z Inlet/Average/Outlet	.983/.969	.991/.983	979/.972
15	Max. Capacity of Casing (acfm @ inlet)	17000	16200	18250
16	Suction Nozzle Size/Rating/Facing	D-F1	D-F3	D-F4
17	Suction Nozzle Gas Velocity (ft/s)	1D2	108	123
18	Discharge Nozzle Size/Rating/Facing	D-G1	D-G3	D-G4
19	Discharge Nozzle Gas Velocity (ft/s)	650	680	670
20	Material of Casing	A-111	A-121	A-811
21	Material of Shaft	S-111	S-111	S-1550
22	Material of Impellers	IMP-X	IMP-Y	IMP-X
23	Material of Diaphragms	CI	CI	CS
24	Material of Labyrinths	ALU	ALU-TEF	ALU
25	Weight of Compressor (lb)	13280	14000	12860
26	Heaviest Maintenance Load (lb)	6380	6000	4280
27	Base Plate Common for Turbine & Compressors	Yes	Yes	No
28	Common Turb. & Comp. Lube System Mfd by	Vendor	Vendor	Subcont
29	Separate/Common Lube & Seal System	Common	Common	Common
30	Seal Type	Film	Mech.Cont.	Film
31	Overhead Bladder Tank/Capacity (Gallons)	58	62	123
32	Overhead Seal Rundown Tank/Capacity	88	None	72
33	Lube Oil Rundown Tank/Capacity (Gallons)	1000	2000	2000
34	Method of Fabrication of Impellers	Welded	Riveted	Welded
35	Thrust Bearing Mfr./Type	KTB	Michell	Own
36	Thrust Bearing Loading/Rating (PSI)	125/500	182/600	220/300
37	Journal Bearings Mfr./Type	Orion	Self	Waukesha
38	Journal B earings Max. Load/Rating	140/200	140/180	85/500
39	Lube Oil Clarifier	Vac.Dehyd.	Coalescer	Centrif.
40	Coupling Mfr./Type	Bendix 416	Bendix 316	Zurn/G
41	Local Control Panel	Yes	Yes	Yes
42	Axial Movement Indicator/Type	Bently	Self	Airfix
43	Vibration Monitoring Equip./Mfr.	Bently	Dymac	Metrix
44	Torsional Responsibility	None	None	None
45	Performance Curves	Attached	Attached	Later
46	Pressure Ratio Rise to Surge Flow (%)	7%	7%	3%
47	Spare Parts List, Priced	Attached	Later	Later
48	Spare Rotor	Included		Included
49	Mechanical Running Tests	Included	Inluded	
50	Closed Loop Performance Test	None	None	None
51	Winterization	Included	None	None
52	Place of Manufacture	City H	City J	Overseas
53	Experience Qualification	Submitted		Later
54	API Data Sheet	Attached	Attached	Later
55	Wheel Experience Record	Proprietary	Later	Attached
56	Bearing Span, inches	83	90	80
57	Prior Experience	Yes	Yes	Yes
58	Spare Rotor Fitup	Yes	Yes	Yes
59	Testing and Inspection	Per Spec	Per Spec	Per Spec
60	Impeller Lineup	CAABCFGH	FFCDABA	GMMPJFSS
61	Inlet Mach Number (each wheel)	.72/.78...	.82/.91...	.78/.81...
62	Diffuser Width (each stage)	2.5/1.8...	2.7/1.9...	2.6/2.0...
63	Impeller Width (each stage)	2.25/1.5...	2.6/2.38...	2.3/1.86...
64	Shaft Diam. at Impellers (nominal, inches)	P	Q	R
65	Shaft Stress at Coupling, psi	6380	7880	11380
66	Coupling Bore, inches	CB-A	CB-B	CB-C
67	Coupling Rating (HP/100 rpm) & S.F.	K/2.8	L/2.9	M/3.2

Figure 2-16. Bid tabulation for centrifugal compressors.

CONSIDERATIONS FOR EVALUATING CENTRIFUGAL COMPRESSOR BIDS

Item #	Questions, Considerations and/or Action Plans
1	Have all compressor models been successfully operated elsewhere?
2	Are there any advantages/disadvantages to barrel construction for this service?
3)	Does the machine with fewer impellers produce too much head in the second section?
4)	
5	Compare quoted diameters with tip speeds. Suitable for impeller fabrication method?
6)	Consistent with tip speeds? Maximum allowed continuous speed?
7)	
8)	Are these values consistent with the related data, above? Does
9)	the vendor have experience with impellers at these tip speeds?
10	How calculated? Sufficiently far from anticipated operating speed range?
11)	Are apparent efficiencies and losses realistic? How determined?
12)	
13	Are higher anticipated discharge temperatures (Vendor B) detrimental to process? Costly?
14	Explain significant difference in gas data for Vendor B. Reconcile with others.
15	Are capacity differences significant in light of future uprate requirements?
16)	Nozzle ratings may reflect design conservatism and uprate capabilities. Have you
17)	made a comparison? Do vendor proposed velocities accurately reflect stated gas
18)	volumes divided by nozzle area? Reconcile discrepencies.
19)	
20)	Are quoted materials consistent with specified materials? Industry practices?
21)	Vendor experience with quoted materials?
22)	
23	Should credit be given to cast steel construction?
24	Why is Vendor B offering PTFE-coated labyrinth material?
25)	Does lower maintenance weight allow installation of less expensive overhead
26)	crane? Is credit due?
27	Investigate possible effects of Vendor C not offering common baseplate.
28	Who is responsible for the design of the subcontracted baseplate proposed by Vendor C?
29	Would separate lube and seal oil systems have been advantageous?
30	Why does Vendor B propose mechanical contact seals? Are there any advantages?
31)	What is this overhead vessel accomplishing? Vendor A lists the rundown capacity
32)	to be in excess of the vessel capacity. Is this logical?
33	What is the calculation basis used to arrive at 1000 gallons vs. 2000 gallons rundown capacity?
34	Any adverse experience with riveted impellers in this service?
35	Why does Vendor C propose to manufacture his own thrust bearings?
36	Could a more suitable balance piston design be used by Vendor C for thrust reduction?
37	Why does Vendor B propose to manufacture his own journal bearings?
38	Investigate rating basis used by each vendor. Why the difference?
39	Is the clarifier intended to remove only water? Water and light hydrocarbons?
40	Coupling offer A allows for higher torque, but less misalignment than offer B. Have you evaluated the pros and cons?
41	Is each control panel fully equipped with all specified instruments? Any extra ones?
42	What do we know about offers B and C? Long-term experience?
43	Any service problems anticipated for the installation destination?
44	Important only if geared train configuration.
45	Is Vendor C offering an untested prototype? Why no performance curves?
46	Low pressure rise to surge would mandate implementation of flow control scheme. Is this possible? Investigate process considerations.
47	Insist on early submittal of spare parts data from Vendors B and C.
48	Clarify status of rotor via discussion with Vendor B.
49	Clarify status of mechanical run tests via discussion with Vendor C.
50	Verify no closed loop performance test was specified. Is it desirable?
51	Why did Vendor A include winterization? What does it consist of?
52	Shipping costs, potential delays, extra expediting costs, service arrangements must be understood - especially for overseas point of origin.
53	Discuss experience qualifications with Vendors B and C as soon as possible.
54	Vendor C must submit full API data before further consideration can be given. Does he have something to hide?
55	Position taken by Vendor A is not acceptable. Expedite Vendor B.
56	Verify that bearing spans are covered by vendors' previous experience.
57)	Are vendors familiar with data submission requirements imposed by purchaser?
58)	Will they contractually agree to submit such data as rotor design dimensions
59)	and as-built dimensions?
60	Is each impeller of proven design? Will impeller performance curves be provided?
61	Are these Mach numbers sufficiently low for the gas being compressed?
62)	Are impeller and diffuser dimensions large enough for potentially fouling
63)	services?
64	Verify experience to ensure against rotor dynamics problems.
65	Do stress calculations make use of stress concentration factors or only simplified calculations? Reconcile high stress quoted by Vendor C.
66	Are quoted coupling bores near maximum permissible diameters? How are hubs engaged to shafts? Hydraulic fit? Who will supply special mounting tools?
67	Are couplings suitable for future uprate requirements?

Figure 2-17. Considerations for evaluating centrifugal compressor bids.

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Chapter 3

Machinery Reliability Audits and Reviews

Audits Versus Reviews

For the purpose of this textbook, *machinery reliability audit* is defined as any rigorous analysis of a vendor's overall design after issuance of the purchase order and before commencement of equipment fabrication. *Reliability review* is defined as a less formal, on-going assessment of component or subsystem selection, design, execution, or testing.

Reliability *audits* tend to use outside resources for brief, concentrated efforts during the first two months after issuance of the purchase order.

Reliability *reviews* are assigned to one or more experienced machinery engineers who would be involved in a project from the time specifications are written until the machinery leaves the vendor's shop for shipment to the plant site.

The primary purpose of the audit is to flush out deep-seated or fundamental design problems on major compressors and drivers. A secondary purpose is to determine which design parameters should be subjected to non-routine computer analysis, and whether follow-up reviews should employ other than routine approaches.

Machinery reliability reviews are aimed at ensuring compliance with all applicable specifications. These reviews also judge the acceptability of certain deviations from applicable specifications. Moreover, an experienced reliability review engineer will provide guidance on a host of items which either *could* not, or simply *had* not, been specified in writing.

Where to Concentrate Audit and Review Efforts

Reliable and efficient machinery is probably the most important factor ensuring profitable operation of process plants. This contention becomes law in the petrochemical industry where economic considerations often mandate the use of single, unspared machinery trains to support the entire operation of steam crackers producing as much as 800,000 metric tons (~1.76 billion lbs) of ethylene per year. When plants in this size range experience emergency shutdowns of a few hours' duration, flare losses alone can amount to $400,000 or more. Evidently, the incentives to build reliability into the machinery installation are very high. This is generally recognized by contractors and plant owners who allocate funds and personnel to conduct reliability reviews before taking delivery of the machinery, during its installation, or even after the plant goes on stream.

Of course, reliability assurance efforts made *before* delivery of the machinery are more cost effective than post-delivery or post-startup endeavors aimed toward the same goals. However, the questions remain how to optimally conduct these efforts, how to man them, and which components or systems to subject to close scrutiny. This is where an analysis of available failure statistics will prove helpful. A review of the failure statistics of rotating machinery used in modern process plants will help determine where the company's money should be spent for highest probable returns. Moreover, failure statistics can often be used to determine the value of and justification for these efforts.

Experience shows that a petrochemical project in the $800,000,000 range would optimally staff machinery reliability audits with four engineers for a four-month period, and machinery reliability reviews with two engineers for a period of 2–3 years. The total cost of these efforts would be in the league of $800,000–$950,000. If this sounds like a lot of money, the reader may wish to contrast it with the value of a single startup delay day, say $550,000, or the cost of two unforeseen days of downtime—perhaps accompanied by the thunder of two tall flare stacks for the better portion of two days.

Machinery reliability audits and reviews can be a tremendously worthwhile investment as long as they are performed by experienced engineers. Of course, this presupposes that a perceptive project manager will see to it that the resulting recommendations are, in fact, implemented.

Rotordynamic Design Audits*

By far the most prevalent and also most important design audit effort is focused on turbomachinery rotordynamics. It is in this area that design weaknesses can often be spotted and appropriate changes implemented before the equipment leaves the manufacturer's shop. Large multinational petrochemical companies are sometimes staffed to handle these audits. However, in most instances this audit task is entrusted to independent consulting companies with the experience and technical resources to perform this critically important task soon after a purchase order has been issued.

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*Source: J. C. Wachel, Engineering Dynamics Inc., San Antonio, Texas, USA. Originally presented at the 15th Turbomachinery Symposium, Texas A&M University, Corpus Christi, Texas, 1986. Reprinted by permission.

Of course, rotordynamics design audits may be equally valuable for existing equipment with inherent design defects. These defects may manifest themselves in a number of ways; they include sensitivity to unbalance and sensitivity to misalignment, and range all the way to frequent, unexplained downtime. Moreover, retroactive rotordynamic design audits represent an excellent means of determining the merits of component upgrading in existing turbomachinery.

On new equipment, the decision to perform a rotordynamic design audit is generally based on the type of machine, the manufacturer's experience with similar sizes, speeds, etc., and the assessment of the benefits versus the cost of the analysis. If it could be assumed that nothing would go wrong, then the audit would not be needed. However, statistics show that design and manufacturing problems do occur that result in considerable delay to projects. Cook[1] indicates that over half of the major projects in the 1974–1984 time frame encountered a critical speed design problem and/or high vibration near rated speed. This study indicated that the delay time to correct design equipment error could be as high as 100 weeks. For some performance-related problems, up to four years were needed to correct the difficulties.

Exxon Chemical Company statistics for the late 1970s and early 1980s indicated that approximately 22% of the unscheduled downtime events for major turbocompressors in process plants were caused by the rotor/shaft systems.[2] Considering all the unscheduled downtime causes which could be vibration-related, the percentage would be greater than 50%. A study by an insurance company found that failures expected each year were about one out of every 186 for steam turbines, and one out of every 26 for gas turbines.

Data such as this and the author's experience in troubleshooting vibration and failure problems indicate that design audits can help prevent many of the problems causing unscheduled downtime, project delays, and/or failures by identifying potential problem areas before manufacture.

Another reason for performing an independent audit is the fact that the system may consist of used equipment. In order to avoid any contractual liabilities, the manufacturer may not want to perform the rotordynamic calculations on the new system or the changes that are being made.

The following are major types of rotordynamic design audits that can be performed, and they are discussed in the following sections.

1. Lateral Critical Speed Analyses
    - Critical speed map
    - Undamped natural frequencies
    - Undamped mode shapes
    - Bearing and seal stiffnesses and damping
    - Rotor response to unbalance
    - Pedestal and foundation effects on response
    - Stability
2. Torsional Critical Speed Analyses
    - Natural frequencies
    - Mode shapes

- Interference diagram
- Coupling dynamic torques
- Dynamic gear loads
- Harmonic torque loads for reciprocating machinery
- Torsional vibrations
- Shaft stresses

3. Transient Torsional Analyses
   - Start-up time
   - Stress versus time
   - Cumulative fatigue
   - Allowable number of starts
4. Impeller and Blade Analyses
   - Natural frequencies
   - Mode shapes
   - Interference diagram
   - Experimental shaker tests or modal analysis
5. Pulsation Analyses
   - Acoustic resonances
   - Mode shapes
   - Shaking forces
   - Surge effects

Lateral Critical Speed Analysis

The most common design audits are the lateral and torsional critical speed audits since they potentially offer the most benefits. Experience indicates that many systems have been installed with critical speeds in the running speed range and have run successfully for years before troubles are encountered. This sometimes is difficult to understand, but a design audit that considers the entire range of possible values for the shaft unbalances and bearing and seal parameters will usually indicate the possibility of a problem.

A lateral critical speed audit should include these calculations:

1. Critical speed map
2. Undamped natural frequencies and mode shapes
3. Bearing stiffness and damping properties
4. Seal stiffness and damping properties
5. Rotor response to unbalance
6. Pedestal and foundation effects on response
7. Rotor stability

The first step in performing a lateral critical speed analysis is to model the shaft with sufficient detail and number of masses to accurately simulate the rotor responses through its speed range. An accurate shaft drawing giving the dimensions, weights, and centers of gravity of all added masses is needed to develop the model.

Generally, each significant shaft diameter change is represented by one or more stations. A station is generally located at each added mass or inertia, at each bearing and seal location, and at each potential unbalance location. A typical rotor shaft drawing and the computer model are given in Figure 3-1.

Rotating elements such as wheels and impellers are modeled as added masses and inertias at the appropriate locations on the shaft. The polar and transverse mass moments of inertia are included in the analyses to simulate the gyroscopic effects on the rotor. The gyroscopic effects are particularly significant on overhung rotors where the impeller or disk produces a restoring moment when whirling in a deflected position.

Couplings are simulated as concentrated added weights and inertias. Normally the half coupling weight is placed at the center of gravity of the half coupling. When necessary, the entire train, including the driver and driven equipment, can be modeled by using programs that can simulate the shear loading across the coupling without transferring the moments. Once the shaft model is completed, the critical speed map can be calculated.

Critical Speed Map. This is a logarithmic plot of the undamped lateral critical speeds versus the combined support stiffness, consisting of the bearing and support structure as springs in series. The critical speed map for a seven-stage compressor rotor is given in Figure 3-2. The critical speed map provides the information needed to understand the basic response behavior of rotors; therefore, it is important to understand how the map is developed.

For large values of support stiffness, the rotor critical speeds are called the rigid bearing critical speeds. If the bearing stiffness is infinity, the vibrations are zero at the bearings, and the first natural frequency for shafts that do not have overhung impellers or disks is analogous to a simply supported beam.

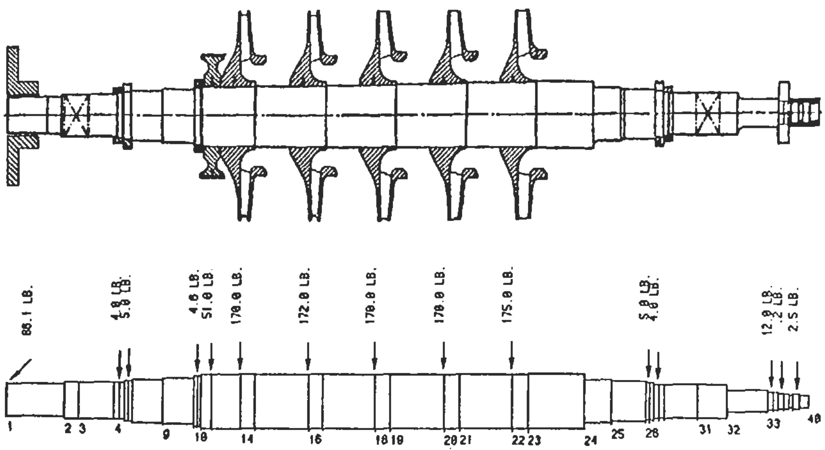


Figure 3-1. Typical shaft drawing and computer model.

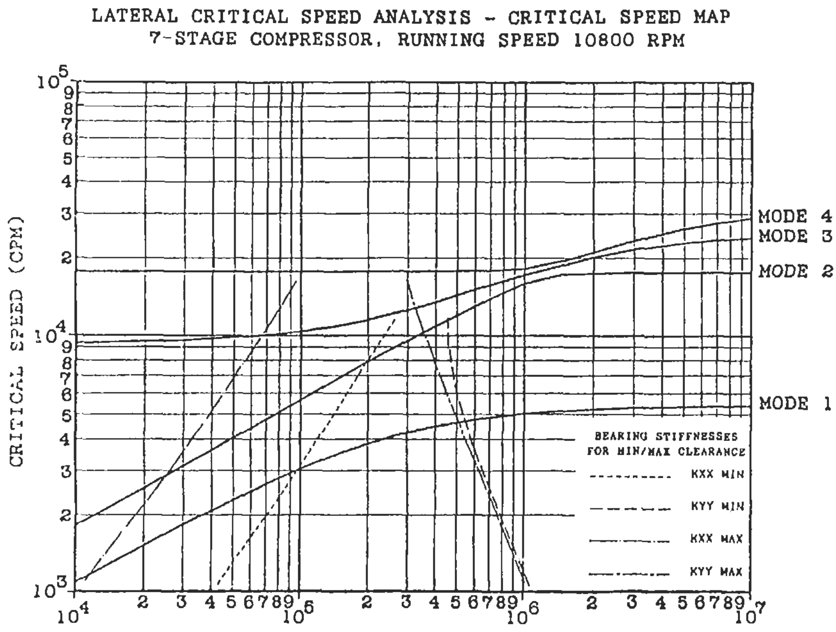


Figure 3-2. Critical speed map for seven-stage compressor.

A normalized critical speed map is given in Figure 3-3 to illustrate the ratios for the various criticals for low and high support stiffness values and to illustrate the mode shapes that the rotor will have at different bearing and support stiffness values. For the rigid bearing critical speeds, the mode shape for the first mode would be a half-sine wave (one loop), the second critical speed would be a two-loop mode and would occur at four times the first mode critical, the third critical speed would be a three-loop mode and would be nine times the first critical, etc. For most rotors, the bearing stiffnesses are less than infinity and the second critical will be less than four times the first critical and is typically two to three times the first critical.

For low values of support stiffness (shaft stiffness is large compared to support stiffness), the first critical speed is a function of the total rotor weight and the sum of the two support spring stiffnesses. For an ideal long slender beam, the second mode is similar to the rocking of a shaft on two springs and is equal to 1.73 times the first critical speed. Since both the first and second modes are a function of the support stiffness, the slope of the frequency lines for the first and second critical speeds versus support stiffness is proportional to the square root of the stiffness for low values of support stiffness compared to the shaft stiffness.

For a support stiffness of zero, the third and fourth modes would be analogous to the first and second free-free modes of a beam. For an ideal uniform beam, the ratio

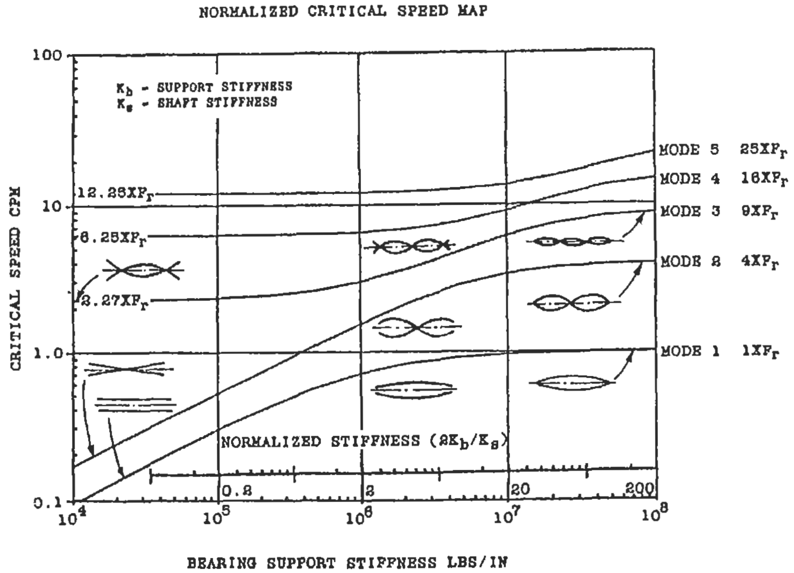


Figure 3-3. Normalized critical speed map.

of the frequencies for these modes compared to the first critical speed for rigid bearings is 2.27 and 6.25.

Bearing Stiffness and Damping. The dynamic stiffness and damping coefficients of bearings can be adequately simulated using eight linear coefficients (Kxx, Kyy, Kxy, Kyx, Cxx, Cyy, Cxy, Cyx) (Figure 3-4). This information along with the lubricant minimum film thickness, flow, power loss, and temperature rise at operating conditions are needed to evaluate the bearing design. The bearing stiffness and damping coefficients are calculated as functions of the bearing type, length, diameter, viscosity, load, speed, clearance and the Sommerfeld number which is defined as:

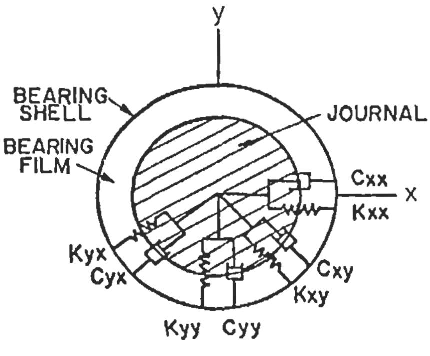


Figure 3-4. Bearing stiffness and damping coefficients.

$$S = \frac{\mu NDL}{W}\left(\frac{R}{C}\right)^2 \tag{3-1}$$

where μ = lubricant viscosity, lb-sec/in.^{2}
N = rotor speed, Hz
D = bearing diameter, in.
L = bearing length, in.
R = bearing radius, in.
W = bearing load, lbs
C = radial machined clearance, in.

Several papers[3–5] discuss various types of bearings and their stiffness and damping characteristics. Table 3-1 gives a typical set of bearing calculations for a 5 shoe, load on the pad, tilting pad bearing. The dimensionless coefficients and the actual stiffness and damping values are shown versus speed and eccentricity ratio.

The normal procedure in a design audit is to calculate the bearing characteristics for the range of expected clearances, preload and oil temperatures. The maximum

Table 3-1
Bearing Coefficients for Tilting Pad Bearing

5-Shoe Tilting Pad Bearing

Load on pad	**L/D = .42**	**Preload = .444**
Arc length = 52.00°	**Pivot ang. = 54.00°**	**Offset = .540**

Dimensionless Tilting Pad Bearing Coefficients

ECC. (dim)	Sommerfeld (dim)	Kxx (dim)	Kyy (dim)	Cxx (dim)	Cyy (dim)	Torque (dim)
.05	3.96E+00	31.352	31.641	34.475	34.677	25.157
.10	1.96E+00	15.657	16.264	17.144	17.564	25.246
.15	1.27E+00	10.398	11.355	11.311	11.966	25.397
.20	9.25E-01	7.740	9.084	8.350	9.257	25.613
.25	7.09E-01	6.113	7.889	6.530	7.707	25.900
.30	5.60E-01	4.996	7.254	5.227	6.744	26.265
.35	4.49E-01	4.162	6.964	4.343	6.120	26.720
.40	3.62E-01	3.503	6.918	3.607	5.714	27.279
.45	2.92E-01	2.956	7.072	3.002	5.461	27.963
.50	2.34E-01	2.486	7.409	2.489	5.320	28.803
.55	1.85E-01	2.072	7.935	2.045	5.269	29.842
.60	1.44E-01	1.701	8.678	1.653	5.291	31.144
.65	1.10E-01	1.366	9.695	1.307	5.375	32.813
.70	8.04E-02	1.062	11.089	1.000	5.502	35.020
.75	5.63E-02	.789	13.046	.731	5.659	38.078
.80	3.68E-02	.548	15.901	.500	5.875	42.625
.85	2.17E-02	.344	20.595	.310	6.233	50.248
.90	1.08E-02	.183	30.649	.162	7.020	66.733

(table continued on next page)

Table 3-1 (continued)
Bearing Coefficients for Tilting Pad Bearing

Dimensional Tilting Pad Bearing Coefficients

L = 1.875 in. D = 4.500 in. C = .00450 in.
Load = 438.0 lbs μ = 2.50E − 06 REYNS

ECC. (dim)	Speed (rpm)	Kxx (lb/in.)	Kyy (lb/in.)	Cxx (lb-s/in.)	Cyy (lb-s/in.)	HP loss (hp)
.05	19748.2	3.05E+06	3.08E+06	1.62E+03	1.63E+03	6.16E+01
.10	9744.4	1.52E+06	1.58E+06	1.64E+03	1.68E+03	1.50E+01
.15	6350.6	1.01E+06	1.11E+06	1.66E+03	1.75E+03	6.43E+00
.20	4611.7	7.53E+05	8.84E+05	1.68E+03	1.87E+03	3.42E+00
.25	3535.0	5.95E+05	7.68E+05	1.72E+03	2.03E+03	2.03E+00
.30	2790.6	4.86E+05	7.06E+05	1.76E+03	2.25E+03	1.28E+00
.35	2237.0	4.05E+05	6.78E+05	1.80E+03	2.54E+03	8.39E−01
.40	1804.7	3.41E+05	6.73E+05	1.86E+03	2.94E+03	5.58E−01
.45	1455.3	2.88E+05	6.88E+05	1.92E+03	3.49E+03	3.72E−01
.50	1166.3	2.42E+05	7.21E+05	1.98E+03	4.24E+03	2.46E−01
.55	923.9	2.02E+05	7.72E+05	2.06E+03	5.30E+03	1.60E−01
.60	719.1	1.66E+05	8.45E+05	2.14E+03	6.84E+03	1.01E−01
.65	546.0	1.33E+05	9.44E+05	2.22E+03	9.15E+03	6.14E−02
.70	400.7	1.03E+05	1.08E+06	2.32E+03	1.28E+04	3.53E−02
.75	280.4	7.68E+04	1.27E+06	2.42E+03	1.88E+04	1.88E−02
.80	183.2	5.33E+04	1.55E+06	2.54E+03	2.98E+04	8.98E−03
.85	107.9	3.35E+04	2.00E+06	2.67E+03	5.37E+04	3.67E−03
.90	53.7	1.78E+04	2.98E+06	2.81E+03	1.22E+05	1.21E−03

clearance, minimum preload and highest oil temperature usually define the minimum stiffness. The other extreme is obtained from the minimum clearance, maximum preload and the coldest oil temperature. This will typically define the range of expected stiffness and damping coefficients for the bearings. Table 3-2 gives an example of the bearing clearances and preloads that can be obtained by considering the range of dimensions on the shaft and bearing.

The anticipated range of rotor response should be calculated with the range of bearing values and various combinations of unbalance. This is important since one part of the API specifications 617 (2.8.1.9) states: "If the lateral critical speed as calculated or revealed during mechanical testing falls within the specified operating speed range or fails to meet the separation margin requirements after practical design efforts are exhausted, the unit vendor shall demonstrate an insensitive rotor design. This insensitivity must be proven by operation on the test stand at the critical speed in question with the rotor unbalanced." The significance of this can be appreciated from the results (Figure 3-5) of the balanced mechanical test and the sensitivity test for the compressor whose critical speed map was given in Figure 3-2. It can be seen that the balanced run data showed no troublesome resonances. When the out-of-phase sensitivity tests were made, high amplitude critical speeds were excited.

Table 3-2
Clearance Variations With Tolerances

Dimension	Minimum	Maximum
Bearing Bore, in.	3.0040	3.0070
Journal Diameter, in.	2.9990	3.0000
Pad Curvature (Diametrical), in.	3.0070	3.0090

CB mils	CP mils	Preload M (dim)	Brg. Bore in.	Pad Bore in.	Journal OD in.
2.50	4.00	0.3750	3.0040	3.0070	2.9990
2.50	5.00	0.5000	3.0040	3.0090	2.9990
4.00	4.00	0.0000	3.0070	3.0070	2.9990
4.00	5.00	0.2000	3.0070	3.0090	2.9990
2.00	3.50	0.4286	3.0040	3.0070	3.0000
2.00	4.50	0.5556	3.0040	3.0090	3.0000
3.50	3.50	0.0000	3.0070	3.0070	3.0000
3.50	4.50	0.2222	3.0070	3.0098	3.0000

CB is the assembled radial clearance.
CP is the machined radial clearance.
M is the bearing preload factor.

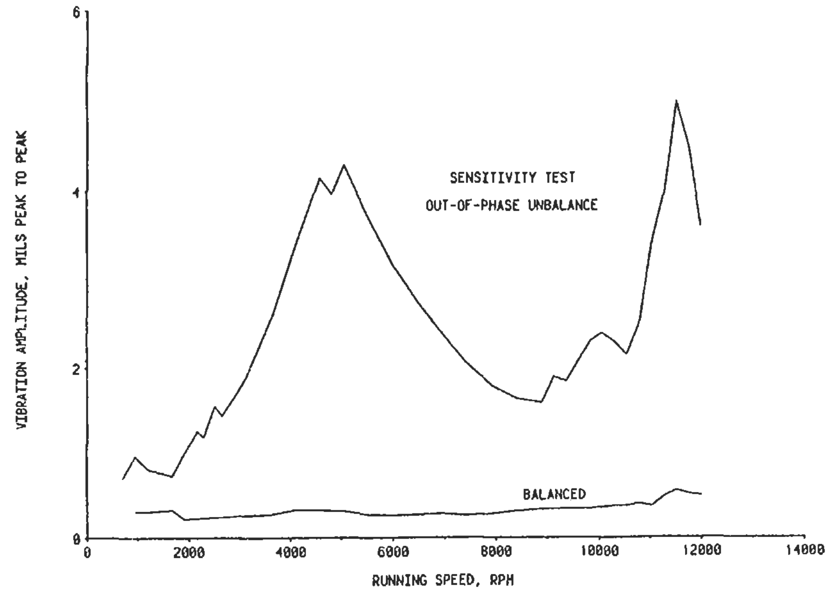


Figure 3-5. Comparison of balanced mechanical run and out-of-phase unbalance sensitivity test.

In the design stage, it is not possible to know the exact installed configuration with regard to bearings (clearance, preload) and balance (location of unbalance). Usually a mechanical test will be limited to one configuration (clearance, preload, unbalance), which may not show any problem. Changes introduced later by spare parts during turnarounds may change sensitive dimensions, which may result in a higher response. For this reason, some satisfactorily operating machines change vibration characteristics after an overhaul.

Undamped Natural Frequencies. If the principal bearing stiffnesses are plotted on the critical speed map (Figure 3-2), the location of the undamped natural frequencies (critical speeds) are identified. By calculating the bearing stiffness and damping coefficients over the expected range of bearing clearances, preload, and viscosity variation, the anticipated range of criticals can be estimated. For rotor systems with tilting-pad bearings, which do not have cross-coupling stiffnesses, the measured criticals will be near these undamped intersections. For bearings with significant cross-coupling stiffness and damping values, the damped critical speeds are usually higher than the undamped critical speeds.

The vertical and horizontal bearing stiffnesses calculated for minimum and maximum clearances are plotted on the critical speed map in Figure 3-2. In this example the vertical stiffnesses did not change significantly with the change in clearance; however, the horizontal stiffness changed by a factor of 4 to 1. This change illustrates the importance of considering the various combinations of clearances in the calculations. The intersection of these stiffness curves defines the undamped horizontal and vertical critical speeds.

Undamped Mode Shapes. An undamped mode shape is associated with each undamped natural frequency (critical speed) and can be used to describe the rotor vibration characteristics. For a vertical stiffness of 464,000 lbs/in., the mode shapes for the first and second undamped natural frequencies are shown in Figures 3-6 and 3-7.

The plotted mode shapes were calculated assuming no damping. The actual vibration mode shapes and response frequencies during operation can vary depending upon the unbalance distribution and damping. The shaft vibrations at any shaft running speed can be calculated for different unbalances using a rotor response program.

Evaluation of Critical Speed Map Calculations. To summarize, in the evaluation of the adequacy of the rotor from the critical speed map and the mode shapes, the following items should be examined.

1. The proximity of the critical speed to running speed or speed range. The undamped lateral speeds should not coincide with the running speed. In order to determine if the actual critical speed will cause excessive vibrations, it is necessary to perform a rotor response to unbalance analysis.
2. The location of the critical speed relative to the support stiffness. If the critical speed is near the rigid bearing criticals (flexible shaft region), increasing the bearing stiffness will not increase the critical speed. Also vibration amplitudes will be low at the bearings and therefore low damping will be available. This

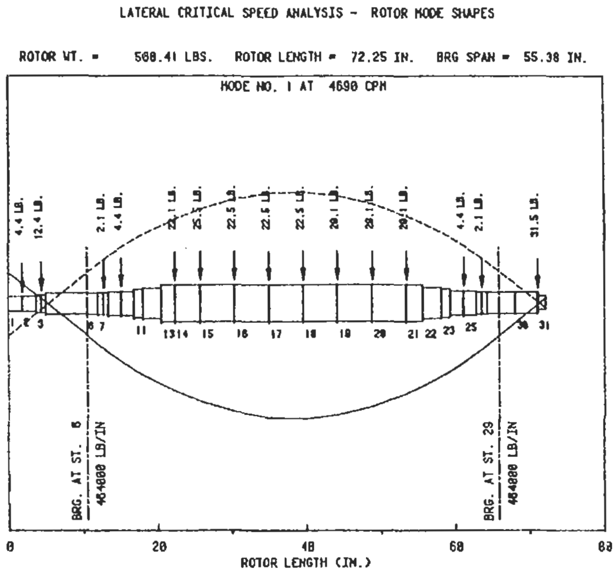


Figure 3-6. Undamped Mode Shape for first critical speed.

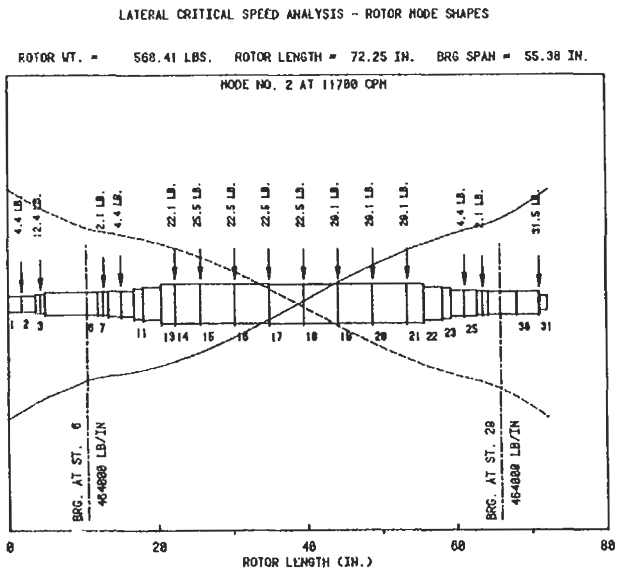


Figure 3-7. Undamped Mode Shape for second critical speed.

can contribute to rotordynamic instabilities, which will be discussed later. If the critical speeds are in the area of low support stiffness (stiff shaft region), the critical speeds are strongly dependent upon the bearing stiffness and damping parameters and the critical speeds can shift considerably.

3. The mode shape of the critical speed. The mode shapes are used to assess the response of the rotor to potential unbalances. For example, a rotor that has a conical whirl mode (second critical) would be sensitive to coupling unbalance, but not strongly influenced by midspan unbalance.

Seal Stiffness and Damping Coefficients. In addition to the bearing stiffness and damping effects, the seals and labyrinths can influence the rotor critical speeds and response. Generally, oil ring seals are designed to float with the shaft since they are held in place by frictional forces dependent upon the pressure balance force and the coefficient of friction. Lubrication and seal oil systems are discussed elsewhere.[6] If the seals do not float with the shaft and lock up, they can add additional stiffness and damping. In such cases, they are treated as additional bearings in the rotordynamic calculations. The seal stiffness and damping coefficients are calculated by assuming that the seals are locked at some eccentricity ratio and that the seals are non-cavitating. Typical values of seal stiffness and damping for centrifugal compressors will be less significant than the bearings; however, in some designs they can change the rotor response characteristics.

Rotor Response to Unbalance. Computer programs are available today that can calculate the elliptical shaft orbit at any location along the length of a rotor for various types of bearings, pedestal stiffnesses, pedestal masses, seals, labyrinths, unbalance combinations, etc. These programs are used to determine the installed rotor's response to unbalance and accurately predict the critical speeds over the entire range of variables. The actual critical speed locations as determined from response peaks caused by unbalance are strongly influenced by the following factors[7,8]:

1. Bearing direct stiffness and damping values
2. Bearing cross-coupled stiffness and damping values
3. Location of the unbalance
4. Location of measurement point
5. Bearing support flexibility

To illustrate the sensitivity of the peak response critical speeds for the compressor whose critical speed map is given in Figure 3-2, the responses due to coupling unbalance and midspan impeller unbalance were calculated. The allowable vibration amplitude (API 617) for this compressor was 1.03 mils peak-to-peak since its maximum continuous speed was 11,300 rpm.

The normal unbalance used in an analysis produces a force equal to 10% of the rotor weight. Usually, rotor response to unbalance calculations are made for midspan unbalance, coupling unbalance, and moment type unbalance. An unbalance equal to a force of 5% rotor weight is usually applied at the coupling to excite the rotor. For moment unbalances, an unbalance equal to the 5% rotor weight is used at the coupling and another equal unbalance is used out-of-phase on the impeller or wheel farthest from the coupling or on the other coupling if it is a drive-through machine. This type of unbalance was used in the mechanical run sensitivity test shown in Figure 3-5.

Sensitivity to Bearing Clearance. The computer response analyses for coupling unbalance are given in Figures 3-8, 3-9, and 3-10 for minimum, nominal, and maximum clearances. In order to compare the computer results with the test data, the com-

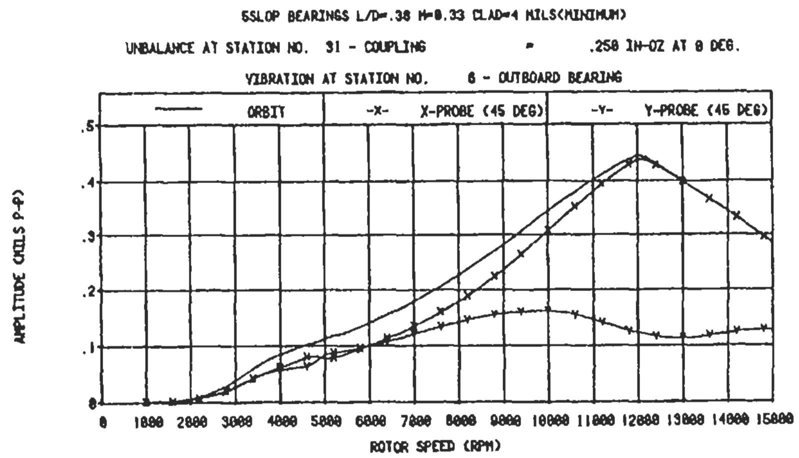


Figure 3-8. Rotor response calculations for coupling unbalance and minimum clearance.

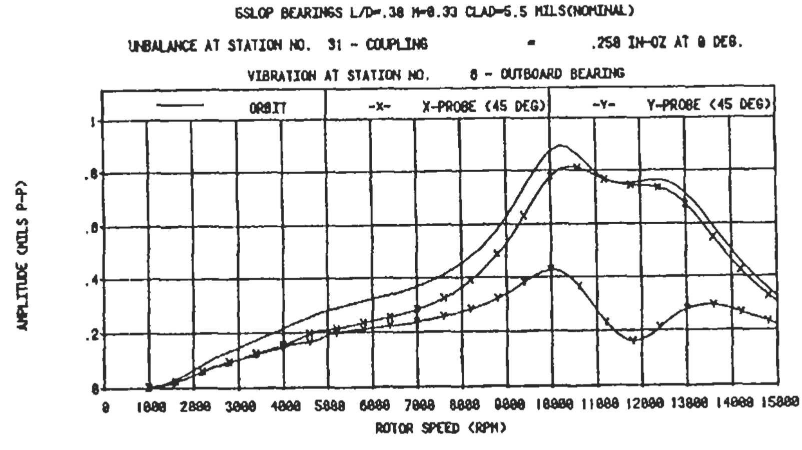


Figure 3-9. Rotor response calculations for coupling unbalance and nominal clearance.

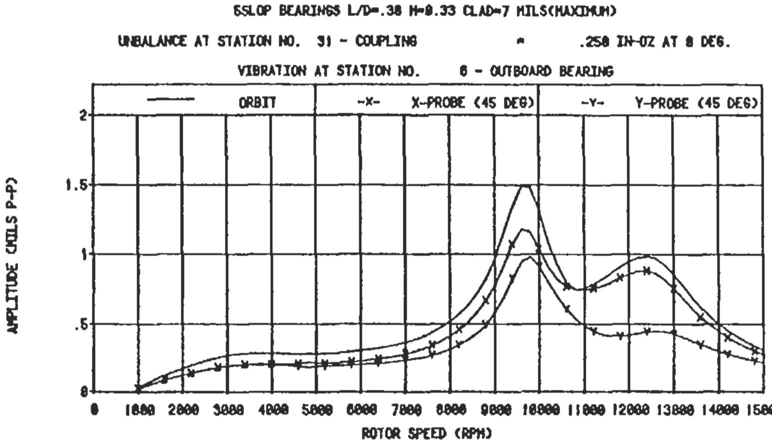


Figure 3-10. Rotor response calculations for coupling unbalance and maximum clearance.

pressor vibrations were predicted at the two 45° probe locations. The vibrations in the horizontal and vertical directions or any other direction can also be predicted. Critical speed responses were predicted at 4,800 cpm (well damped) and another critical at 12,000 cpm (11% above rated speed of 10,800 rpm) for the minimum clearance.

The predicted amplitudes were less than the API limit for the minimum and nominal clearances. However, when the maximum bearing clearance was used, the responses became more pronounced and the predicted amplitudes exceeded the API limits of 1.03 mils. The increased clearance lowered the predicted peak response to 9,700 rpm, which is below the rated speed of 10,800 rpm.

Figures 3-11 and 3-12 show the predicted unbalance response for midspan unbalance for minimum and maximum clearances. For minimum clearance, response is noted at the first critical speed near 4,800 rpm with very little response at the second critical speed near 12,000 rpm. However, for the maximum clearance, the second critical speed at 9,700 rpm becomes the predominant response. This again shows the sensitivity of a rotor to bearing clearance changes.

Sensitivity to Unbalance Location. A design audit response analysis was performed on a power turbine with an overhung disk that had a speed range of approximately 2,500–5,000 rpm. Figure 3-13 shows the results of the response calculations for unbalance at the coupling. Two peak response critical speeds were excited at 3,000 and 5,000 rpm. For this unit, very little difference in predicted responses was noted as the bearing parameters were changed. However, when the unbalance was moved to the disk (Figure 3-14), there were considerable differences in the predicted responses. Note that peak responses at 800, 2,500, and 5,000 rpm are predicted. The two low frequency critical speeds are considerably different from the 3,000 rpm mode excited by coupling unbalance.

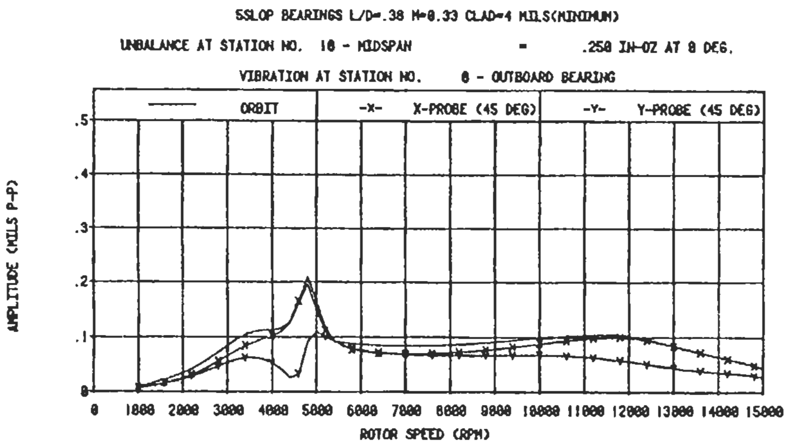


Figure 3-11. Rotor response calculations for midspan unbalance and minimum clearance.

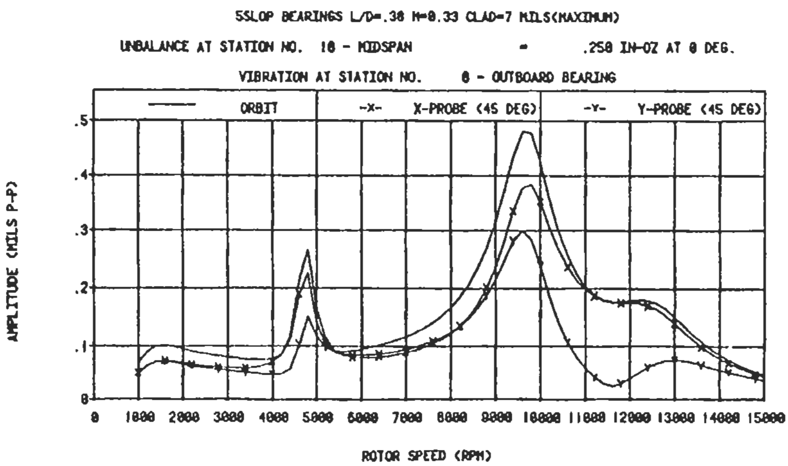


Figure 3-12. Rotor response calculations for midspan unbalance and maximum clearance.

Sensitivity to Pedestal and Foundation Flexibility. The stiffness, mass, and damping of the bearing support structure should be considered in a rotordynamic audit. Basically, the bearing and pedestal stiffnesses combine as springs in series. If the bearing stiffnesses are very low compared to the pedestal stiffnesses, the critical speeds and rotor response will not be greatly affected by the pedestal and support flexibility. Computer programs for dual level rotors are available that can consider the pedestal mass, stiffness, and damping values in the rotor response to unbalance calculations.

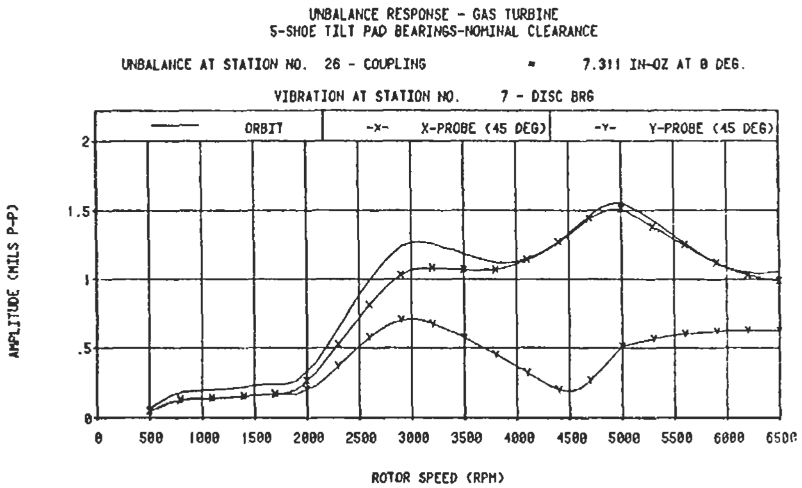


Figure 3-13. Rotor response calculations for coupling unbalance on gas turbine.

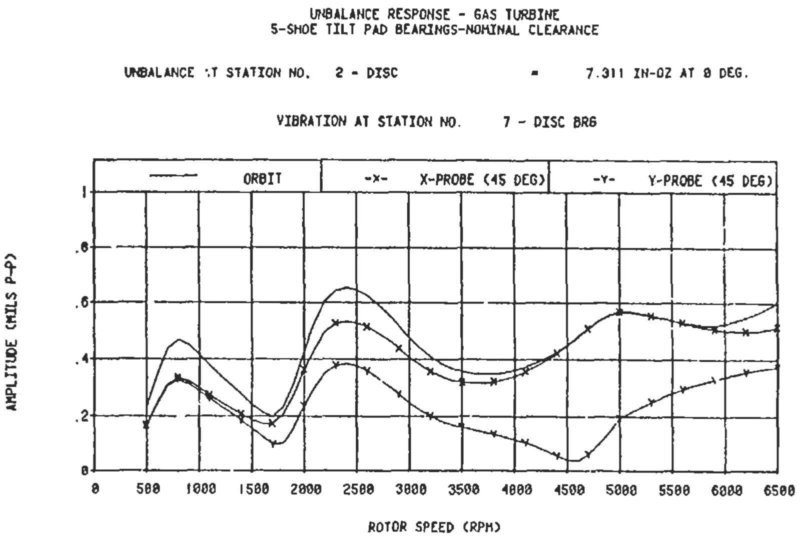


Figure 3-14. Rotor response calculations for disc unbalance on gas turbine.

In the design stage the determination of accurate values for the pedestal and supports can be difficult due to the complex shapes and uncertainties in the bolted joints, grout, and other factors. Finite element programs can be used to determine the needed stiffness and damping values; however, this increases the cost and time of the analysis.

To determine if a complex finite element analysis of the support structure is needed, parametric runs can be made varying the support stiffness from large values to values comparable to the bearing stiffnesses. This information will show how sensitive the rotor critical speeds and responses are to pedestal stiffnesses. Generally, pedestal stiffnesses may vary from approximately 1 million to 20 million lb/in. The horizontal stiffnesses are usually less than the vertical stiffnesses; therefore, the horizontal critical speeds will be lower than the vertical. Pedestal stiffnesses are important in the analysis of large rotors such as induced draft fans where the pedestal and foundation stiffnesses will be lower than typical bearing stiffnesses.

Special Lateral Response Analyses

Liquid Pump Lateral Response Analyses. Pump rotordynamics are dependent on a greater number of design variables than are many other types of rotating equipment. Besides the journal bearing and shaft characteristics, the dynamic characteristics of the seals and the impeller-diffuser interaction can have significant effects on the critical speed location, rotor unbalance sensitivity, and rotor stability.[9,10]

For modeling purposes, seals can be treated as bearings in the sense that direct and cross-coupled stiffness and damping properties can be calculated based on the seal's hydrostatic and hydrodynamic properties. Seal clearances, geometry, pressure drop, fluid properties, inlet swirl, surface roughness, and shaft speed are all important in these calculations. Since the pressure drop across seals increases approximately with the square of the pump speed, the seal stiffness also increases with the square of the speed. This increasing stiffness effect is often thought of as a "negative" mass effect, which is usually referred to as the "Lomakin effect" or the "Lomakin mass."[11] In some cases the theoretical Lomakin mass or stiffness effect can be of sufficient magnitude to prevent the critical speed of the rotor from ever being coincident with the synchronous speed.

The accurate prediction of the stiffness and damping properties of seals for different geometries and operating conditions is a subject of ongoing research.[12,13] The basic theories presented by Black[14] have been modified to account for finite length seals, inlet swirl, surface roughness, and other important parameters. However, a universally accepted procedure to accurately predict seal properties is not available for all the types of seals in use today. This is particularly true for grooved seals. Unless seal effects are correctly modeled, calculated critical speeds can be significantly different from actual critical speeds.

Grooved seal designs used in commercial pumps have been tested recently, and techniques have been developed whereby the seal geometry can be specified and the characteristics calculated for specific assumptions with regard to inlet swirl, groove design, etc.[12,13]

The rotordynamic analysis of an eight-stage centrifugal pump using serrated (grooved) seals was discussed by Tison[9] and will be used to illustrate a design audit of a pump.[9] The first step in a rotordynamic analysis of a pump is to model the basic rotor, using the lumped parameter techniques. A sketch of the rotor with the location of the seals and bearings is given in Figure 3-15. Note that the pump shaft is analyzed as a rotor with 11 bearings (2 cylindrical bearings, 8 impellers, and the balance piston).

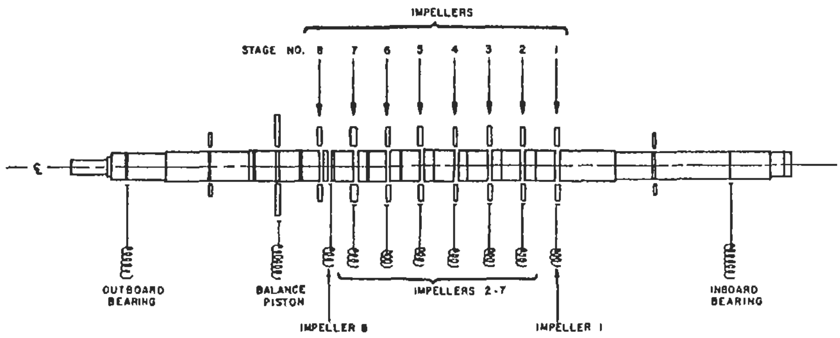


Figure 3-15. Pump rotor model showing locations of all 11 effective bearings.

Rotor unbalance response calculations are the key analysis in the design stage for determining if a pump rotor is acceptable from a dynamics standpoint. In order to bracket the expected range of critical speeds, the unbalance response of a pump should be analyzed for three cases: The first case with no seal effects and maximum bearing clearances which represents the overall minimum expected support stiffness for the rotor (lowest critical speed); the second case would be with minimum seal and bearing clearances which represents the maximum expected support stiffness and therefore, the highest critical speed; and the third case should be considered with maximum bearing clearances and seal clearances of twice the design clearance to simulate worn seals which represents the pump condition after long periods of service.

The minimum calculated critical speed for the 8-stage pump for the maximum bearing clearance, no seal case was 1,700 cpm as shown in Figure 3-16. The American Petroleum Institute (API) Standard 610 allowable unbalanced was applied at the rotor midspan to excite the first mode. The results of the intermediate (worn seals) case are presented in Figure 3-17. The worn seals increase the predicted response peak to approximately 1,800 cpm. With minimum clearances at the bearings and seals, the frequency increases to 2,200 cpm, as shown in Figure 3-18.

Gear Shaft Lateral Response Analysis. When performing a lateral critical speed analysis of gear shafts, the effect of the transmitted torque must be considered in the bearing load analysis. API 613 for Special-Purpose Gear Units For Refinery Services (1995) specifies that the critical speeds should not be less than 20% above the operating speed. The critical speed at approximately 10%, 50%, and 100% load and maximum continuous speed should be considered in the calculations.

Figure 3-19 shows the relationship of the various forces acting on the gear shaft that has helical gear teeth. Since the bearing forces will depend upon the gear weight and the transmitted horsepower, calculations should be made over the range of 10% to

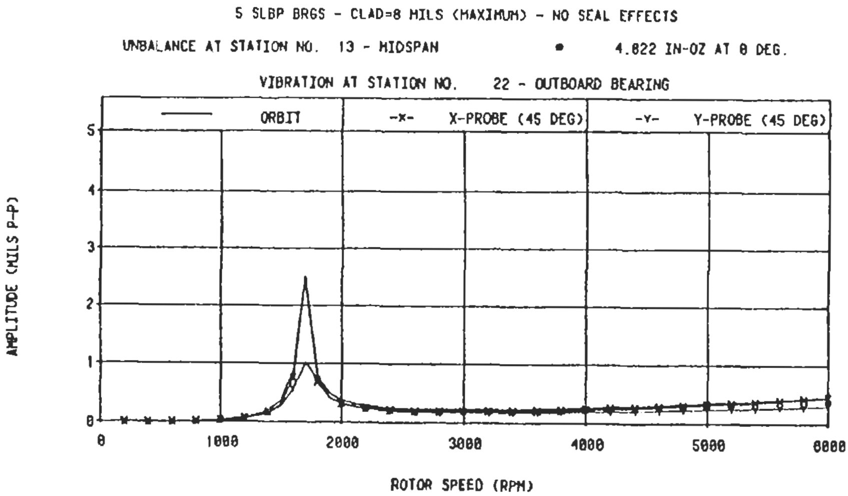


Figure 3-16. Calculated unbalance response at outboard bearing for API unbalance at midspan (no seal effects, maximum bearing clearances).

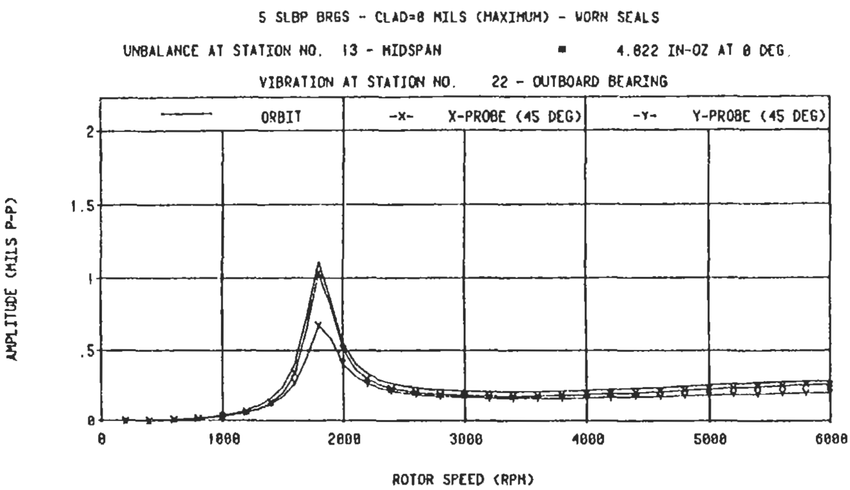


Figure 3-17. Calculated unbalance response at outboard bearing for API unbalance at midspan (maximum (worn) seal clearances, maximum bearing clearance).

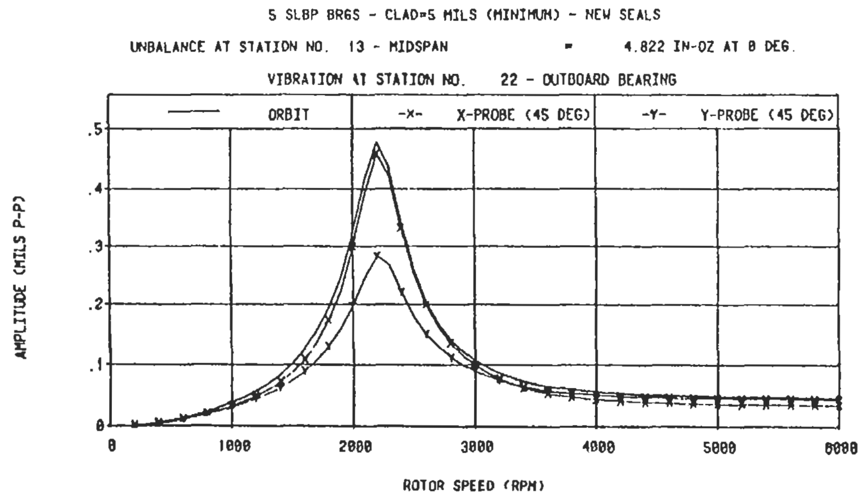


Figure 3-18. Calculated unbalance response at outboard bearing for API unbalance at midspan (minimum (new) seal clearances, minimum bearing clearance).

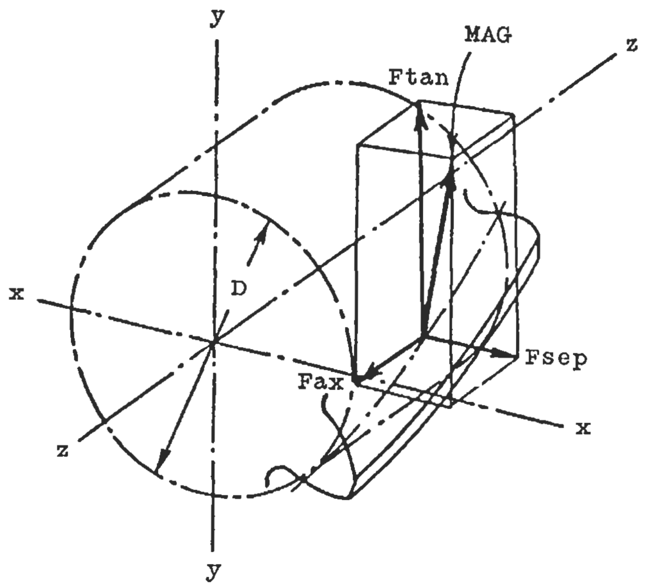


Figure 3-19. Transmitted gear load on helical gear teeth.

100% load. Calculations made on a recent audit of a gear are given in Table 3-3. The gear shafts used four lobed bearings and the bearings stiffnesses and damping changed considerably as the load changed (Table 3-4). To evaluate the changes in fixed-element bearing properties, rotor response to unbalance calculations must be made.

Table 3-3
Bearing Loads as a Function of Transmitted Horsepower

Horsepower = 31,046 bhp
Driver Speed = 5,670 rpm
Driven Speed = 10,742 rpm
PD Driver = 23.190 in.
PD Driven = 12.240 in.
Press. Ang. = 20°
Helix Ang. = 14°

Wt Loads:		Mesh dist:	
A = −778.0 lb			A = 14.17 in.
B = −1,349.0 lb			B = 14.17 in.
C = −527.0 lb			C = 14.17 in.
D = −396.0 lb			D = 14.17 in.

Right handed helix on driver
Upmesh—speed increaser

Bull Gear (Driver) Bearing Forces—lb

%Load	BRG	|Fsep|	|Ftan|	Fwt	Fxnet	Fynet	Mag	Deg	Fax
100.00	A	5,568.	14,875.	− 778.	−2,646.	−15,653.	15,875.	260.	7,142.
100.00	B	5,568.	14,875.	−1,349.	−8,490.	−16,224.	18,311.	242.	
75.00	A	4,176.	11,156.	− 778.	−1,984.	−11,934.	12,098.	261.	5,357.
75.00	B	4,176.	11,156.	−1,349.	−6,368.	−12,505.	14,033.	243.	
50.00	A	2,784.	7,438.	− 778.	−1,323.	− 8,216.	8,321.	261.	3,571.
50.00	B	2,784.	7,438.	−1,349.	−4,245.	− 8,787.	9,758.	244.	
25.00	A	1,392.	3,719.	− 778.	− 661.	− 4,497.	4,545.	262.	1,786.
25.00	B	1,392.	3,719.	−1,349.	−2,123.	− 5,068.	5,494.	247.	
10.00	A	557.	1,488.	− 778.	− 265.	− 2,266.	2,281.	263.	714.
10.00	B	557.	1,488.	−1,349.	− 849.	− 2,837.	2,961.	253.	

Pinion (Driven) Bearing Forces—lb

% Load	BRG	|Fsep|	|Ftan|	Fwt	Fxnet	Fynet	Mag	Deg	Fax
100.00	C	5,568.	14,875.	−527.	7,110.	14,348.	16,013.	64.	−7,142.
100.00	D	5,568.	14,875.	−396.	4,026.	14,479.	15,028.	74.	
75.00	C	4,176.	11,156.	−527.	5,333.	10,629.	11,892.	63.	−5,357.
75.00	D	4,176.	11,156.	−396.	3,019.	10,760.	11,176.	74.	
50.00	C	2,784.	7,438.	−527.	3,555.	6,911.	7,771.	63.	−3,571.
50.00	D	2,784.	7,438.	−396.	2,013.	7,042.	7,324.	74.	
25.00	C	1,392.	3,719	− 527.	1,778.	3,192.	3,653.	61.	−1,786.
25.00	D	1,392.	3,719.	− 396.	1,006.	3,323.	3,472.	73.	
10.00	C	557.	1,488.	− 527.	711.	961.	1,195.	53.	− 714.
10.00	D	557.	1,488.	−396.	403.	1,092.	1,163.	70.	

Table 3-4
Hydrodynamic Bearing Coefficients for Four Lobed Bearing

Low Speed Gear
L/D = 0.867 D = 7.086 in. L = 6.14 in.
CL = 6.03 mils (nominal) RPM = 5,670

Load	Kxx $\times 10^6$	Kxy $\times 10^6$	Kyx $\times 10^6$	Kyy $\times 10^6$	Cxx	Cxy	Cyx	Cyy
10%	0.075	0.564	−1.110	1.080	1,870	361	315	3,780
50%	1.040	0.559	−3.460	5.130	2,360	−322	297	12,000
100%	2.500	0.972	−7.760	12.400	3,880	−1,960	−1,890	28,900

High Speed Pinion
L/D = 1 D = 6.75 in. L = 6.75 in.
CL = 5.38 mils (nominal) RPM = 10,742

Load	Kxx $\times 10^6$	Kxy $\times 10^6$	Kyx $\times 10^6$	Kyy $\times 10^6$	Cxx	Cxy	Cyx	Cyy
10%	0.464	−0.175	0.708	0.976	656	14	422	1,720
50%	2.090	1.440	3.410	7.310	868	811	2,250	8,960
100%	5.390	5.090	9.190	19.600	1,256	2,569	4,859	24,270

The effect of load variation for the rotor response can be seen in Figure 3-20, which gives the response to midspan unbalance for 10, 50, and 100% load. It can be seen that the critical speed would coincide with rated speed between 50% and 100% load.

It is often difficult, if not impossible, to meet the required separation margin of API 613 over the entire load range. Generally, the gear shaft and bearing designs are changed to move the critical speeds from the disallowed range for the normal loads. The response analysis is then made to verify that the vibrations will be below acceptable values even if on resonance at the lower loads.

Rotor Stability Analyses. Rotor stability continues to be of major concern, especially for high-pressure compressors.[15–17] Rotor instability occurs when the rotor destabilizing forces are greater than the rotor stabilizing forces. The destabilizing forces can be caused by: the bearings, oil seals, rotor unbalances, friction in shrink fits, or by aerodynamic loading effects such as rotating stall in either impellers or diffusers, impeller blade loading edge incidence, jets and wakes at impeller tips, diffuser stall, pressure pulsations and acoustical resonances, surge, and labyrinth seals.

Instabilities in rotors can cause high vibrations with several different characteristics. They generally can be classified as bearing related, self-excited, and forced nonsynchronous instabilities. Oil whirl and half-speed whirl are bearing related instabilities and are caused by the cross coupling from the bearing stiffness and damping in fixed geometry bearings. Half-speed whirl will result in rotor vibrations at approximately one-half of the running speed frequency. Oil whirl describes a special type of

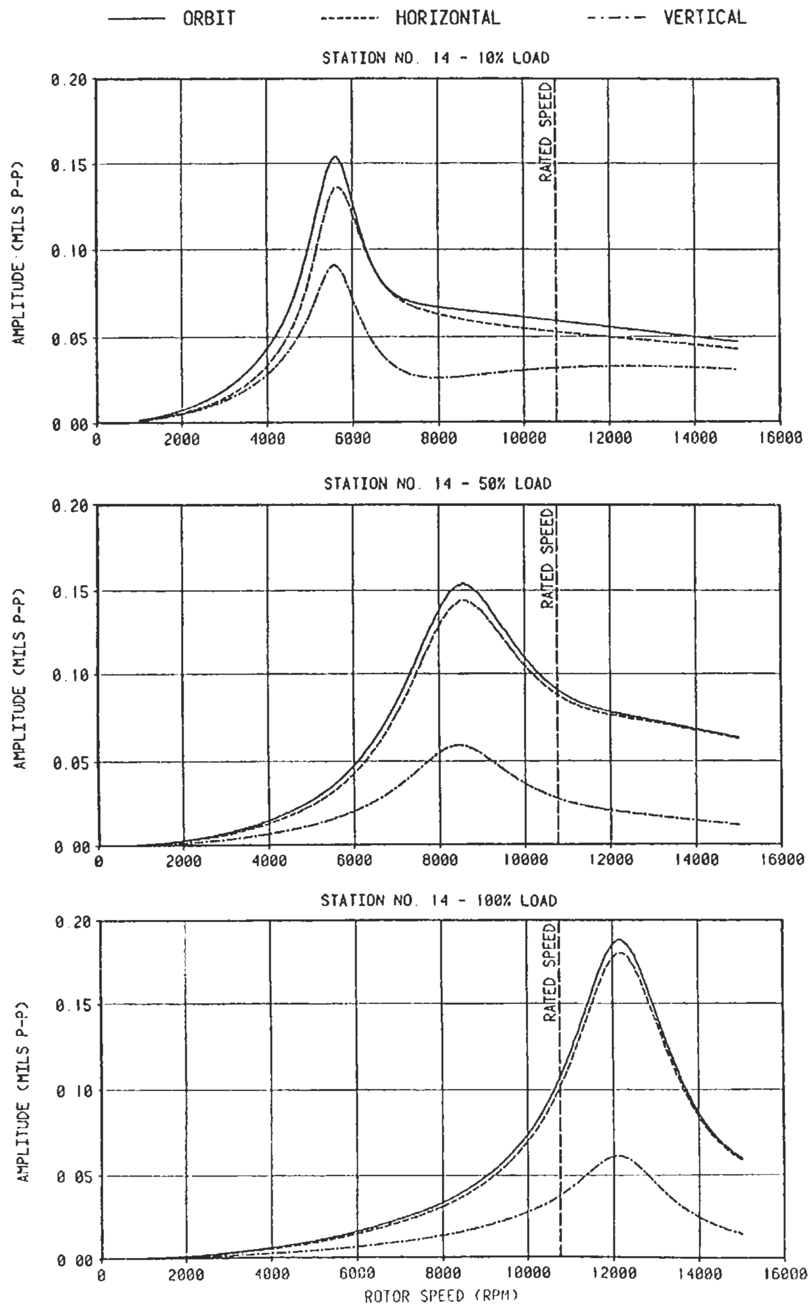


Figure 3-20. Calculated pinion gear response for 10%, 50%, and 100% load for midspan unbalance.

subsynchronous vibration which tracks approximately half-speed up to the point where the speed is two times the first critical. As the speed increases, the subynchronous vibration will remain near the first critical speed. These types of instabilities can generally be solved by changing the bearing design to a pressure dam, elliptical, or offset-half bearing or changing to a tilting pad bearing.

A second type of instability vibrations can occur on any rotor, including those with tilted pad bearings. The vibrations will usually occur near the rotor first critical speed or may track running speed at some fractional speed. These types of instability vibrations are sometimes called self-excited vibrations since the motion of the rotor creates the forcing mechanism that causes the instability.[18]

A third type of instability is called a forced non-synchronous instability and can be caused by stage stall in the compressor last stages or by acoustical resonances in the system.[19,20] This type of instability usually occurs at 10%–20% of running speed as dictated by the acoustical response characteristics of the diffuser and passage geometry.

The predominant method used in performing a stability analysis is to calculate the damped (complex) eigenvalues and logarithmic decrement (log dec) of rotor, bearing, and seal assembly.[21] A positive log dec indicates that a rotor system is stable, whereas a negative log dec indicates an unstable system. Experience has shown that due to uncertainties in the calculations, the calculated log dec should be greater than 0.3 to ensure stability. The damped eigenvalue and log dec are sometimes plotted in a synchronous stability map. Damped eigenvalues generally occur near the shaft critical speeds; however, in some heavily damped rotors they can be significantly different from the responses due to unbalance.

Rotor stability programs are available that can model the rotor stability for most of the destabilizing mechanisms; however, some of the mechanisms that influence it are not clearly understood.[22] It has been well documented that increased horsepower, speed, discharge pressure, molecular weight, and pressure ratio can cause a decrease in the rotor stability. Many units that are stable at low speeds and pressure become unstable at higher values. To predict the stability of a rotor at the design operating conditions, the rotor shaft, bearings, and seals are modeled and the log dec is calculated as a function of aerodynamic loading. An equation, based on experience from several instability problems, includes many of the factors that have shown to be important in rotor stability such as horsepower, speed, diameter of impeller, density ratio across the compressor, impeller and diffuser restrictive dimensions, and molecular weight of fluid.[15] This equation can be used to predict the approximate aerodynamic loading that the unit should be able to withstand. The aerodynamic loading is a cross coupling term, usually applied near the center of the rotor. Conceptually, it can be thought of as a component that detracts from the stabilizing forces in the system.

In the normal audit procedures, the stability is calculated as a function of aerodynamic loading with a computer model of the rotor, bearings, and seals. In the evaluation of the stability, it is desirable to have a log dec at zero aerodynamic loading greater than 0.3 and still greater than 0.1 at the calculated aerodynamic loading. The log dec should be calculated for the range of bearing and seal properties expected as shown in Figure 3-21, which gives a plot for the lowest forward mode and indicates the estimated aerodynamic loading.

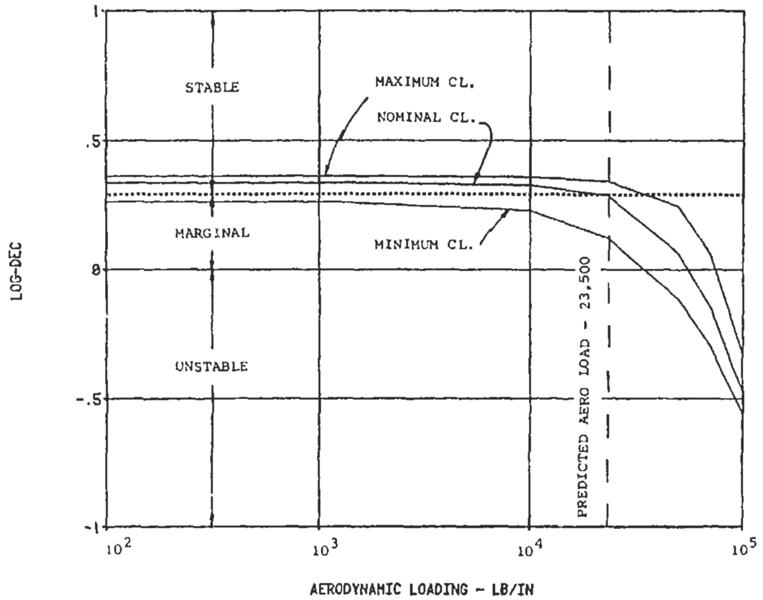


Figure 3-21. Calculated log dec versus aerodynamics cross coupling for stability analysis of centrifugal compressor.

Kirk[17] has developed stability criteria that use the final pressure times the differential pressure across the machine versus the ratio of running speed to the first critical speed. Stroh[23] suggested a work/energy relationship to determine the aerodynamic loading.

Torsional Critical Speed Analyses

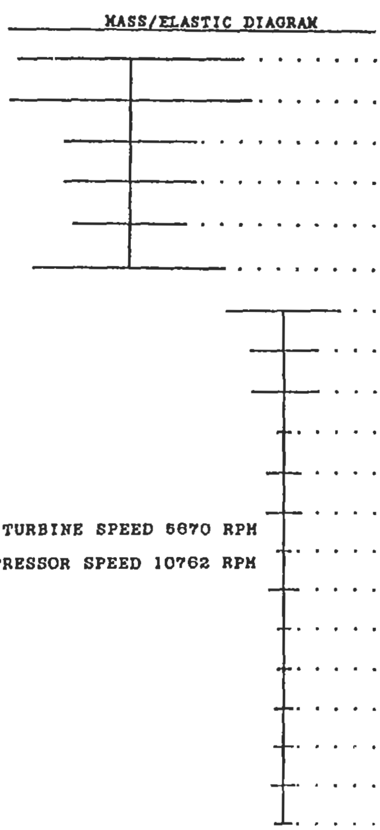
All rotating and reciprocating systems have torsional vibrations. Operation on a torsional natural frequency can cause shaft failures without an obvious increase in the lateral vibrations. Therefore, it is important to ensure that all torsional natural frequencies are sufficiently removed from excitation frequencies.

A torsional audit should include the following:

1. Calculation of the torsional natural frequencies and associated mode shapes.
2. An interference diagram that shows the torsional natural frequencies and the excitation components as a function of speed.
3. Calculation of the coupling torques to ensure that the coupling can handle the dynamic loads.
4. Calculation of shaft stresses, even if allowable margins are satisfied.
5. Calculation of transient torsional stresses and allowable number of starts for synchronous motor drives.

Torsional natural frequencies are a function of the torsional masses and the torsional stiffnesses between the masses. The natural frequencies and mode shapes are generally calculated by the Holzer method or by eigenvalue-eigenvector procedures. Either of the methods can give accurate results. It is desired that the torsional natural frequencies have a 10% margin away from all potential excitation mechanisms.

An example of the mass-elastic diagram of a torsional system is given in Figure 3-22. The natural frequencies and mode shapes associated with the first four natural frequencies are given in Figure 3-23. The mode shapes can be used to determine the most influential springs and masses in the system. This information is important if encroachment is calculated and system changes must be made to detune the systems. Parametric analyses should be made of the coupling stiffness if changes are necessary, since most torsional problems can be solved by coupling changes.



MASS NO.	WR2 in-lb-s2	K(1E-6) in-lb/rad	STATION DESCRIPTION
1	591.48	5051.65	STG 1 WHL
2	948.73	506.31	STG 2 WHL
3	43.20	237.39	THRUST DSK
4	42.97	17.50	RM-604 HUB
5	26.12	165.80	RM-604 HUB
6	238.03	1000.00	BULL GEAR
7	25.71	106.48	PINION
8	6.61	13.80	RM-454 HUB
9	6.42	37.24	RM-454 HUB
10	.38	370.56	SLEEVE
11	2.74	180.53	STG 1 IMP
12	2.75	220.02	STG 2 IMP
13	.33	220.02	LABYRINTH
14	2.41	352.03	STG 3 IMP
15	.21	270.79	DIV LABY
16	.21	352.03	DIV LABY
17	1.71	220.02	STG 6 IMP
18	1.75	163.74	STG 5 IMP
19	2.02	57.87	STG 4 IMP
20	1.69	.00	BAL PISTON

Figure 3-22. Torsional mass-elastic data for gas turbine-compressor train.

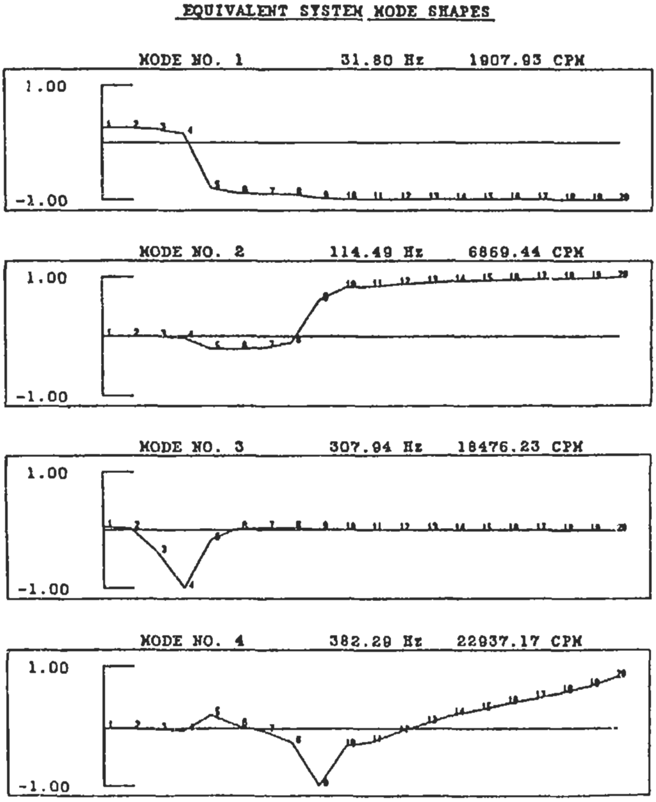


Figure 3-23. Torsional natural frequencies and mode shapes.

An interference diagram for the turbine-driven compressor with a gear box is given in Figure 3-24. The rated speeds are 5,670 rpm for the gas turbine and 10,762 rpm for the compressor. In this system, excitation at 1X and 2X the gas turbine and compressor speeds are possible. The 1X excitation of gas turbine speed excites the first critical speed at 1,907 rpm; however, it will not reach the second natural frequency at 6,869 rpm since maximum speed would be less than 6,000 rpm. The compressor speed (1X) excitation would excite the first torsional natural frequency at 1,005 rpm and the second natural frequency at 3,619 rpm on the gas turbine.

Once the system has been modeled and the natural frequencies have been determined, the forcing functions should be applied. The forcing functions represent dynamic torques applied at locations in the system that are likely to generate torque variations. Identification of all possible sources of vibration is an important step in diagnosing an existing vibration problem or avoiding problems at the design stage.

The most likely sources of dynamic torques include reciprocating engines, gears, fans, turbines, compressors, pumps, motors (synchronous and induction), couplings, fluid interaction (pulsations), and load variations.

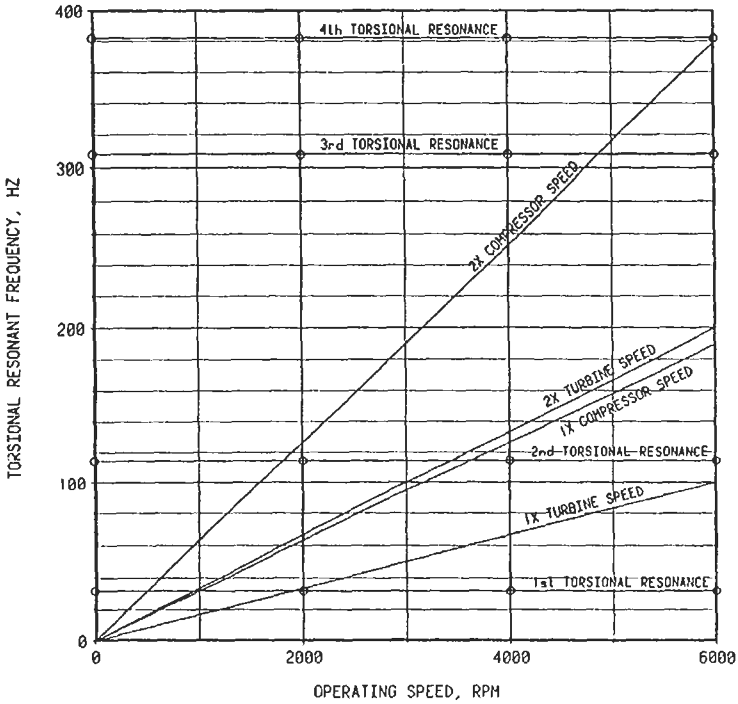


Figure 3-24. Interference diagram for gas turbine-compressor train.

To evaluate the stresses at resonance, the torsional excitation must be applied to the system. For systems with gear boxes, a torque modulation of 1%, zero-peak is a representative torque value that has proven to be appropriate for most cases. As a rule of thumb, excitations at the higher orders for gears are inversely proportional to the order numbers: the second order excitation is 0.5%, the third is 0.33%, etc.

The torque excitation should be applied at the appropriate location and the torsional stresses calculated on the resonant frequency and at the running speed. An example of the stress calculations of the second natural frequency resonance is given in Table 3-5. It shows that a 1% torque excitation on the bull gear would cause a maximum torsional stress of 4,179 psi p-p in shaft 9, which is the compressor shaft between the coupling and the first-stage impeller. The dynamic torque modulation across the couplings is calculated for the applied input modulation. For this mode, the maximum torsional vibrations occur across the compressor coupling and the dynamic torque modulation was 2,626 ft-lb.

Variable Speed Drives. Units that use a variable speed drive in conjunction with an electric motor will have excitation torques at the running speed frequencies, and at several multiples, depending upon the design of the variable speed drive.[24] Figure 3-25 gives an interference diagram for one such system. It is difficult to remove all coinci-

Table 3-5
Torsional Stress Calculations at the Second Torsional Natural Frequency for 1% Excitation at the Bull Gear

Dynamic torques (1% zero-peak) applied at the bull gear
Maximum resultant torsional stresses at the 2nd torsional resonance 6,869.44 cpm

Shaft	Stress psi P-P	scf	Stress psi P-P
1	3.92	2.00	7.83
2	34.47	1.50	51.71
3	117.23	3.00	351.68
4*	Dynamic torque variation		468.82 ft-lbs
5	21.76	3.00	65.27
6*	Gear mesh		
7	1,334.81	3.00	4,004.43
8*	Dynamic torque variation		2,626.80 ft-lbs
9	1,393.16	3.00	4,179.49
10	859.02	1.50	1,288.52
11	723.70	1.50	1,085.55
12	582.18	1.50	873.27
13	564.75	1.50	847.12
14	434.23	1.50	651.35
15	422.73	1.50	634.09
16	411.05	1.50	616.57
17	314.94	1.50	472.41
18	215.28	1.50	322.93
19	98.88	1.50	148.32

*Note: *Values are dynamic torque variation across coupling or gear, ft-lbs*

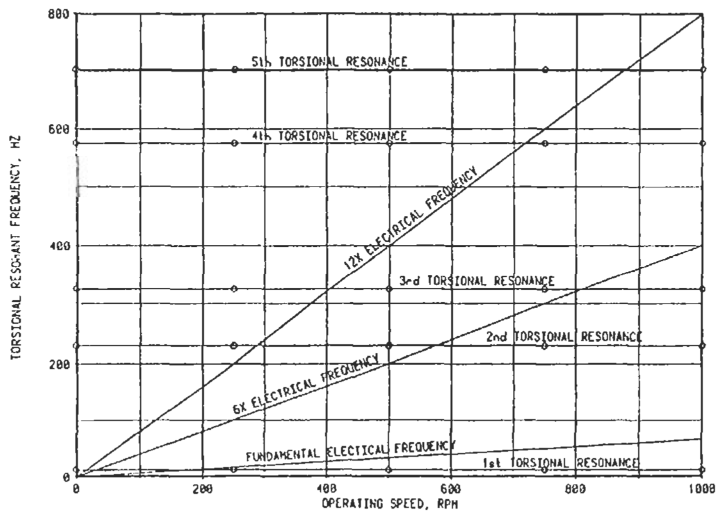


Figure 3-25. Interference diagram for induction motor with variable speed drive.

dence of resonances with the excitation sources over a wide speed range; therefore, stress calculations must be made to evaluate the adequacy of the system response.

Reciprocating Machinery. For reciprocating units such as compressors, pumps, or engines, the harmonic excitation torques must be calculated and applied at the appropriate shaft location to calculate the stresses.[25]

Allowable Torsional Stresses. The calculated torsional stresses must be compared to applicable criteria. The allowable values given by Military Standard 164 are appropriate for most rotating equipment. The allowable zero-peak endurance limit is equal to the ultimate tensile strength divided by 25. The MIL Spec uses this as a global derating factor rather than calculating on the basis of individual factors accounting for keyways, surface roughness effects, and the like. When comparing calculated stresses to this value, the appropriate stress concentration factor and a safety factor must be used. Generally, a safety factor of 2 is used for fatigue analysis. When these factors are used, it can be shown that fairly low levels of torsional stress can cause failures, especially when it is observed that the standard keyway (USA Standard ANSI B17.1) has a stress concentration factor of 3.

We should not lose sight of the fact that process machinery is expected to live much longer than military hardware, and that our process machinery manufacturer has, perhaps:

1. no S-N curves
2. no intention of applying individual derating factors for either known stress raisers or unknown superimposed stresses
3. no interest in determining coupling and misalignment-induced stress adders, etc.,

It would thus be reasonable to use a global derating value of 75, and, indeed, world-class turbomachinery manufacturers such as Elliott, Dresser-Rand, Mannesmann-Demag, Sulzer, Mitsubishi, and surely many others, have both the experience and analytical capability to virtually guarantee unlimited life of turbomachinery shafts operating at relatively much higher mean torsional stresses. A typical example would be steam turbine shafts with tensile strengths of 105,000 psi (ult.) and steady-state torsional stresses of 10,900 psi, where this latter stress simply uses the standard calculation formula

$$\tau_{(mean)} = 16T/\pi d^3.$$

Nevertheless, $\tau_{(ult)}/75$ is not at all unreasonable for machines built by the "other" manufacturers. A midwestern U.S. plant uses rotary blowers direct-coupled to 200 hp, 1800 rpm motors. The blowers came with 2½ inch diameter shafts that had an ultimate strength in tension of 80,000 psi. Although nominal stresses are thus only 2281 psi, the plant experienced *many* shaft failures with derating values as high as 80,000/2281 = *35*. A typical torsional stress allowable thus becomes the ultimate tensile strength divided by 75.

Transient Torsional Analysis

After the steady state analysis is made, a transient analysis should be made to evaluate the startup stresses and allowable number of startups for synchronous motor systems.[22,26] The transient analysis refers to the conditions on startup, which are continually changing because of the increasing torque and speed of the system. When a synchronous motor starts, an excitation is imposed upon the torsional system due to field slippage. As the motor increases in speed, the torsional excitation frequency decreases from twice power line frequency (typically 120 Hz) linearly with speed toward zero. During this startup, the torsional system will be excited at several of its resonant frequencies if they are between 0 and 120 Hz, as shown in Figure 3-26. The response amplitudes and shaft stresses depend upon the resonant frequencies, the average and pulsating torque when the system passes through these resonant frequencies, the damping in the system, and the load torques. The startup analyses can be made for loaded or unloaded operation. The transient response is also affected by the starting acceleration rate of the motor. For slow motor startups, the system will stay at a resonant frequency for a longer period of time, allowing stresses to be amplified. If acceleration is rapid, passing through the resonance quickly will minimize the amplitude increase at resonant frequencies.

Synchronous motors develop a strong oscillating starting torque because of slippage between the rotor and stator fields. Although this is only a transient excitation, the pulsating torque can be strong enough to exceed the torsional endurance limit of

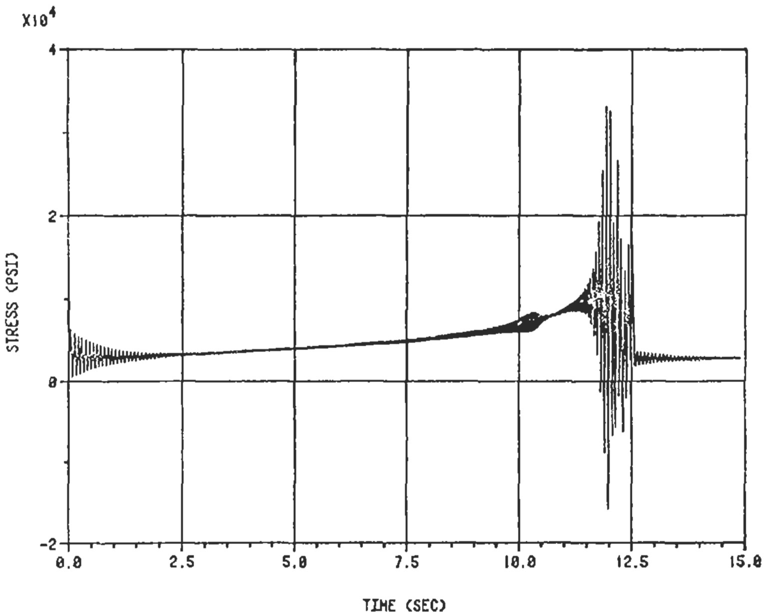


Figure 3-26. Torsional stresses introduced into motor shaft during synchronous motor startup.

the shaft. For this reason the transient stresses must be calculated and compared to the endurance limit stress. It is not necessary that the transient stresses be less than the endurance limit stress; however, the stresses must be sufficiently low to allow an acceptable number of starts. If the transient stresses exceed the endurance limit, the cumulative fatigue concept is applied to the stresses in excess of the endurance limit stress to determine how many starts can be allowed for the system.

Cumulative fatigue theory is used to estimate the number of cycles a certain stress level can be endured before shaft failure would occur. This is based upon a plot of stress versus number of cycles (S-N curve), which defines the stress conditions at which a failure should occur. The S-N curve is based upon actual tests of specimens of a particular type of metal and defines the stress levels at which failures have occurred in these test specimens. These S-N curves are available for most types of shafting materials. Using the appropriate curve, the allowed number of cycles for a particular stress can be determined. It is possible to calculate the number of total startups that can be made with the system before a shaft failure is predicted. Since the stress levels vary both in amplitude and frequency, a more complex calculation must be made to determine the fraction of the total fatigue that has occurred. The stress levels for each cycle are analyzed to determine the percentage of cumulative fatigue and the allowable number of startups can then be determined.

The calculation of the allowable number of starts is strongly dependent upon the stress versus cycles to failure curve and whether torsional stresses higher than the torsional yield are allowed. In the design stage it is preferable to design the system such that the introduced torsional stresses do not exceed yield. This can usually be accomplished through appropriate coupling changes.

Impeller and Blade Responses

A design audit should also include an assessment of the potential excitation of blade or impeller natural frequencies. Several papers document such problems.[27–30] The impeller and blade response analysis should include:

1. The blade and impeller natural frequencies
2. The mode shapes
3. Interference diagram indicating potential excitation mechanisms and the natural frequencies.

The interference diagram, which gives the blade and impeller natural frequencies and the various potential excitation mechanisms, is the key to prevention of failures. The resonances should be sufficiently removed from the major excitations in the operating speed range.

In the design stage, it is possible to calculate the natural frequencies and mode shapes using finite element method [FEM] computer programs. However, the accuracy of predictions depends to a great extent upon the experience of the analyst and the complexity of the system.

Since the blades and impellers will usually be available in advance of the rotor assembly, the most accurate natural frequency and mode shape data can be obtained from shaker tests or by modal analysis methods. The modal analysis technique uses a two-channel analyzer and an impact hammer and accelerometer to determine the natural frequencies and mode shapes. For example, the natural frequencies and mode shapes of a centrifugal impeller were measured using modal analysis techniques (Figure 3-27). When these frequencies were compared to values determined from a shaker study, good correlation was obtained. The mode shape for the two-diameter mode is given in Figure 3-28.

An interference diagram for this impeller is given in Figure 3-29. Note that potential excitation mechanisms include vane passage frequency (15X) and two times vane passage frequency (30X).

It is sometimes impossible to completely avoid all interferences over a wide speed range, since there are so many natural frequencies. For most systems, in order for a failure to occur, several things usually occur together. First, there must be a mechanical natural frequency. Second, there must be a definite excitation frequency, such as vane passing or diffuser vane frequency. Third, there must be some acoustical resonant frequency that amplifies the energy generated; and fourth, there must be the appropriate phase relationship that causes the pulsation to cause a shaking force on the impeller or blade. The best way to avoid such problems is to avoid coincidence of the resonances with the excitation mechanisms.

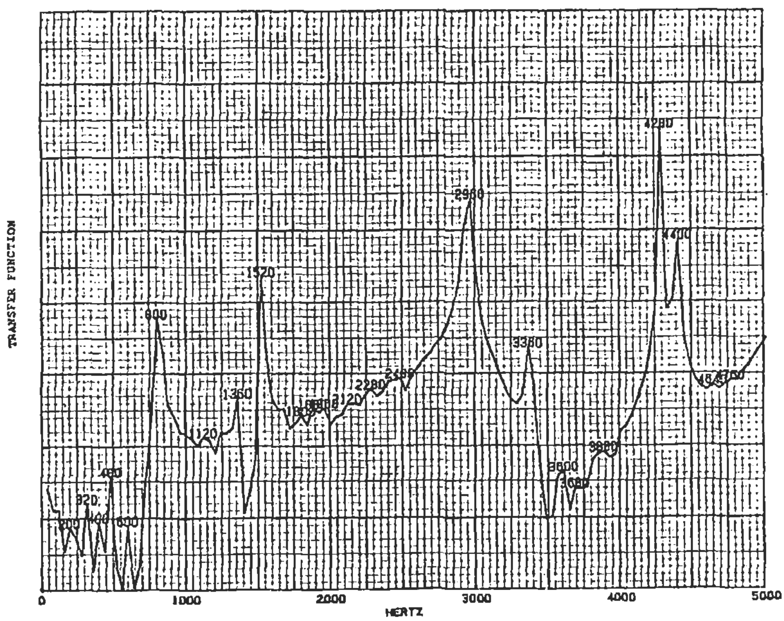


Figure 3-27. Natural frequencies of centrifugal impeller.

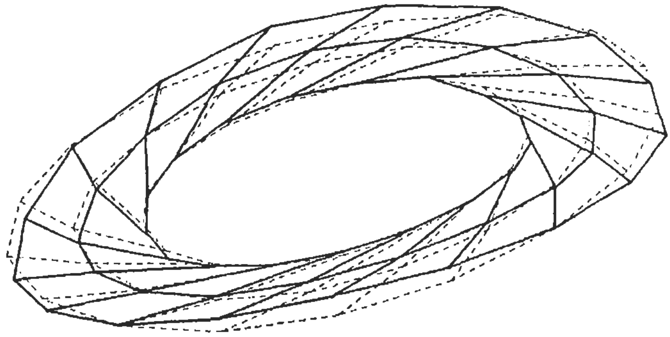


Figure 3-28. Two-diameter mode shape for centrifugal impeller at 1,360 Hz determined by modal analysis tests.

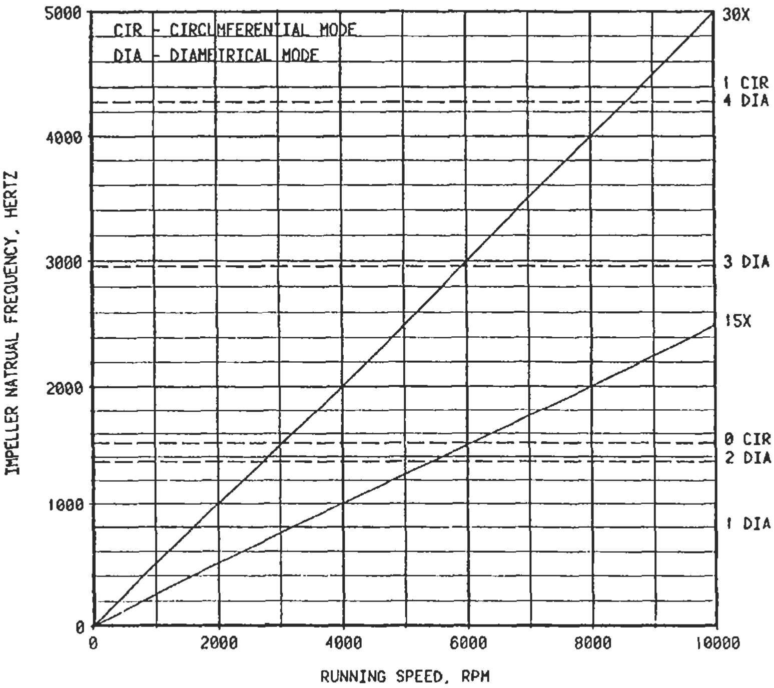


Figure 3-29. Interference diagram for centrifugal impeller.

Pulsations

Pulsations can cause problems in rotating equipment as well as reciprocating machinery.[31,32] Pulsation resonances occur in piping systems and are a function of the fluid properties and the piping, compressor, or pump geometry.

Pulsations can cause premature surge in centrifugal compressors and pumps if the generated pulses, such as from stage stall,[16] match one of the pulsation resonances of the system. The potential pulsating excitation mechanisms for piping systems are the running speed component and its multiples, vane, and blade passing frequency and those caused by flow excited (Strouhal frequency) phenomena.[19]

In the design stage, the acoustical natural frequencies of piping systems can be calculated using either digital[25] or analog modeling procedures.[31] A model of a piping system analyzed on a digital computer is given in Figure 3-30. The predicted

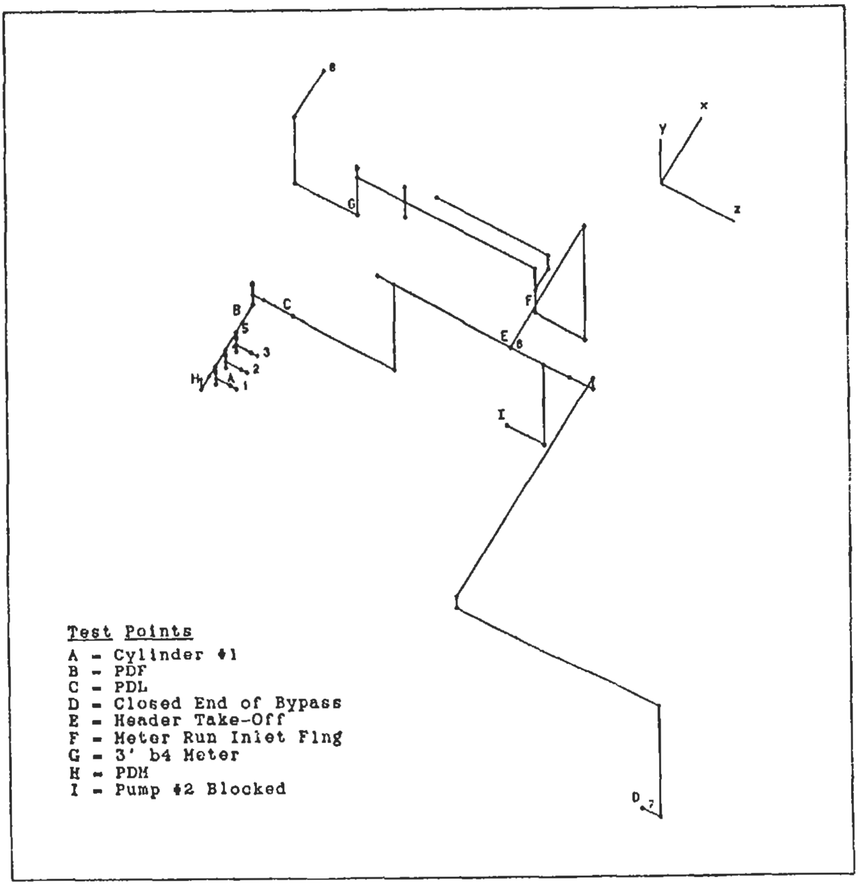


Figure 3-30. Digital computer model of pump system for pulsation analysis.

pulsations in the reciprocating pump system at selected locations are given in Figure 3-31. These pulsation levels define the energy in the pump and the piping shaking forces, and can be used to define the necessary piping supports and span lengths to achieve acceptable vibration levels. The program can be used to redesign the piping to reduce the pulsations to acceptable levels.

Summary and Conclusions

Rigorous rotordynamic analyses must be thought of as additional insurance that the machinery will run without major problems. Many of the analysis procedures and computer programs that have been developed are being used by both the manufacturer and by consultants who offer these design audit services. As with many computer programs, the interpretation of the computer results is dependent upon the skill and

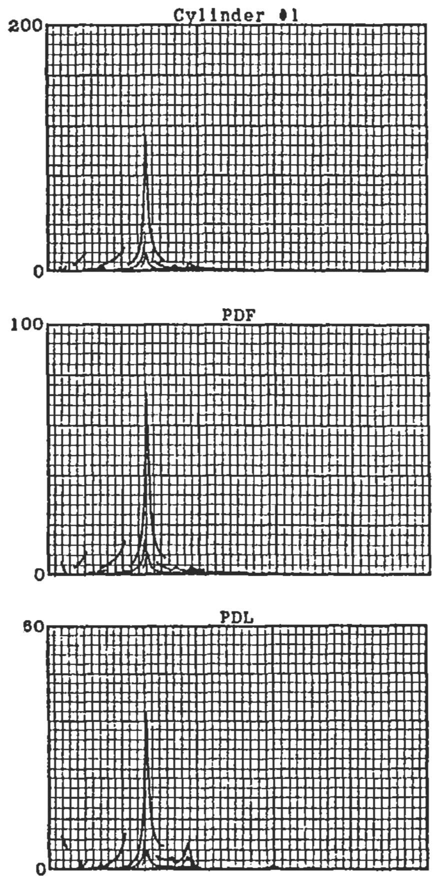


Figure 3-31. Predicted pulsations in pump and piping system for pump speed range of 170–260 rpm.

experience of the analyst. The manufacturer produces many machines that have no problems and thus has the confidence that the machinery will run successfully. The independent consultant usually is asked to look only at machinery with some kind of problem. Therefore, consultants probably look at the analysis from a slightly different viewpoint. The goal of the manufacturer, the user and the independent consultant who make audits is the same: everyone wants a trouble-free machine.

The independent audit generally occurs after the manufacturer has finalized his drawings. In many cases where problems were found during the audit, it turned out that although the manufacturer had made an analysis in the initial stages, some dimensions were changed during the manufacturing phase, causing significant change in the calculated responses. If the independent audit is made, any differences between the manufacturer's and the consultant's calculations can be resolved before installation.

Guidelines as to when an independent audit should be obtained are influenced by many factors. Generally audits should be performed on:

1. New prototype machines that are extrapolations in horsepower, pressure, number of impellers, bearing span, or incorporate new concepts.
2. Machines which, if unreliable, would cause costly downtime.
3. Machines that are not spared (no backups).
4. Expensive machines and installations in which the cost of the audit is insignificant compared to total cost.

Typical costs for analyses are difficult to specify since the scope of the work depends upon the adequacy of the supplied information, the complexity of the machinery, and the number of parametric variation analyses required. If accurate drawings and system information can be supplied to independent consultants who perform these studies, then accurate cost estimates can be given. Exxon's experience may, however, give the most powerful incentive to studies of this type: It has been estimated that for every dollar spent on machinery reliability verification *before* machinery commissioning, ten dollars were returned at plant startup, and one hundred dollars were saved during the life of the plant.

Failure Statistics for Centrifugal Compressors

For any type of machinery considered, knowledge of failure causes and downtime statistics allows us to determine which components merit closer review. Also, properly kept records could alert the review engineer to equipment types or models which should be avoided. In some cases, failure statistics might provide key input to a definitive specification. In other words, "you learn from the mistakes of others." All of this presupposes that "others" saw fit to record their experiences. If an engineer has these data available he will no doubt use them before selecting machinery.

Whenever specific data are lacking, the review engineer has to resort to general statistics. These statistics are typically available in the form given for centrifugal compressors, Table 3-6 or in the form of availability tables (see Chapter 4). Probable

Table 3-6
Typical Distribution of Unscheduled Downtime Events for Major Turbocompressors in Process Plants

Approximate number of shutdowns per train per year: 2

Cause of Problem	Estimated Frequency	Estimated Average Downtime Hrs/Event	Events/Yr	Hrs/Yr
Rotor/shaft	22%	122	.44	54
Instrumentation	21%	4	.42	2
Radial bearings	13%	28	.26	7
Blades/impellers	8%	110	.16	18
Thrust bearings	6%	22	.12	3
Compressor seals	6%	48	.12	6
Motor windings	3%	200	.06	12
Diaphragms	1%	350	.02	7
Miscellaneous causes	20%	70	.40	28
All causes	100%		2.00	137 hours

problem-cause distribution and duration are shown in Table 3-6. The table also shows where detailed design reviews might prove most profitable.

Experienced process plants using a conscientious program of mechanical and instrument-condition surveillance could expect to achieve compressor train availabilities exceeding 98.5%. Unscheduled downtime events in state-of-the-art facilities occur fewer than once per year per train.

Table 3-6 shows that rotor and shaft distress rank highest in downtime hours per year per train. Blade or impeller problems rank next, followed by motor failures. Obviously, centrifugal compressor reliability audits and follow-up reviews should concentrate on these areas first.

Failure Statistics for Steam Turbines

Failure statistics for special-purpose steam turbines are often separated into those for impulse turbines and those for reaction turbines. One such important statistic is represented in Figure 3-32. Of practical interest are primarily those failures which have a reasonable probability of occurring within 80,000 operating hours. The author's experience would indicate that each turbine type is acceptable as long as rigorous selection criteria are applied. First and foremost of these would be an investigation of vendor experience, blade stresses, and blade vibration behavior.

Operating data accrued after 1970 support the belief that rotor blades furnished with impulse turbines rank about even with reaction-turbine blading. A prerequisite to operation would be that they were subjected to user design audits and pre-commissioning reliability reviews.

Interesting statistics have also been quoted by a large North American industrial equipment insurance company. Over a number of years, this company has insured an average of 6,353 steam turbines. During this time period, one out of every 186 steam

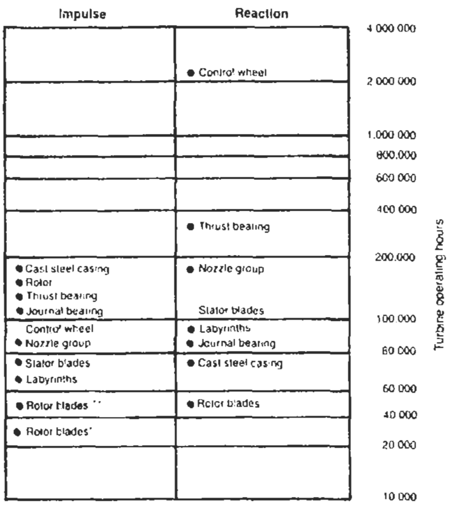


Figure 3-32. Average probable turbine operating hours for individual components without damage. (From *Der Maschinenschaden,* Volume 43 (1970), No. 1, pp. 1–40 Allianz Versicherungs-AG, Munich, Germany. Transcription courtesy of Siemens-America.)

turbines experienced failures which were serious enough to require disbursements from the insurance carrier.

These failure figures appear to cover only the most serious events. At 0.03 events per year per steam-turbine driver, the insurer's statistics might lead us to believe that we could relax our audit and review efforts. However, we must keep in mind that our efforts are aimed not only at eliminating major wrecks but nuisance trips, excessive downtime, startup delays, and frequent maintenance as well.

It should also be recognized that an awareness of failure causes is necessary for the effective implementation of machinery reliability audits. For instance, bearing distress in steam turbines, large electric motors, and associated connected equipment is often caused by the action of stray electric currents. This type of damage is best eliminated by the up-front installation of current leak-off brushes, Figure 3-33. A well-designed leak-off system can be monitored for effectiveness and serviced with the equipment operating.

Failure Statistics for Gas Turbines

In May, 1980, Allianz Versicherungs-AG of Munich, Germany, released a report on the failures of modern industrial gas turbines.* Figure 3-34 gives the distribution

*Leopold, J., "Erfahrungen Mit Stationären Gasturbinen Moderner Bauart," *Der Maschinenschaden,* Vol. 53, No. 5, 1980.

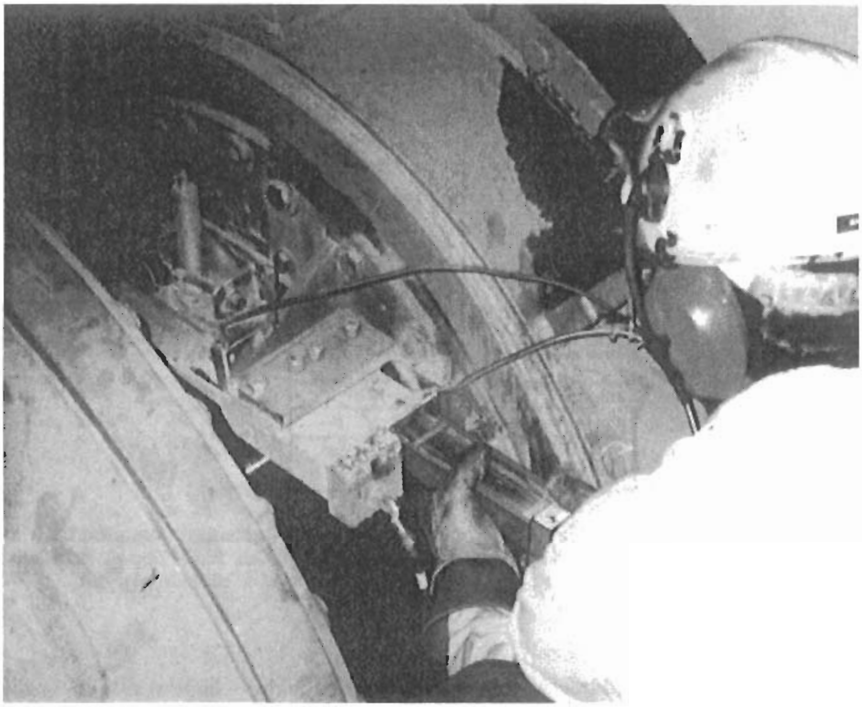


Figure 3-33. Shaft riding brushes in turbine-generator application. *(Courtesy of Sohre Turbomachinery Inc., Ware, Massachusetts.)*

of primary failure causes for industrial gas turbines from 1970 to 1979. Figure 3-35 shows the component damage distribution for the same machines.

Gas turbines have been found to experience more frequent failures than steam turbines. Quoting again the insurance company mentioned earlier, we would expect one serious failure per 26 gas turbines per year. (They reported 20 failures per year out of an average population of 520 gas turbines.) For gas-turbine-driven generators, their statistics show 4.3 driver failures for every failure of the driven equipment.

Typical primary failure causes are reported in Table 3-7.

It should be noted that the failure cause distribution given by U.S. insurance carriers differs from that reported by Allianz in Figure 3-35. It is very difficult to weigh the significance of this observation—especially when we are being told that an identical series of failure reports submitted to both insurance carriers was coded quite differently by the two companies.

Failure Statistics for Centrifugal Pumps

Many petrochemical plants assemble *some* data on pump failure causes, but few of them take the time to make sure that pertinent findings reflect in the next issue of

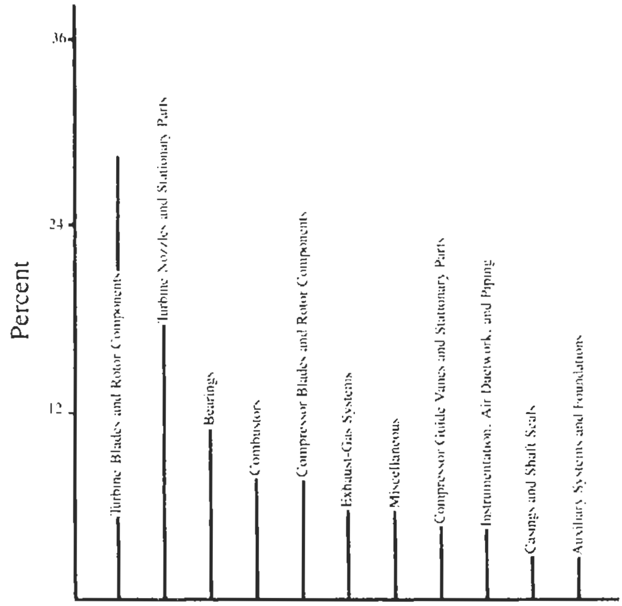


Figure 3-34. Distribution of gas turbine component damage. (*Courtesy Der Maschinenschaden, No. 153, 1980.*)

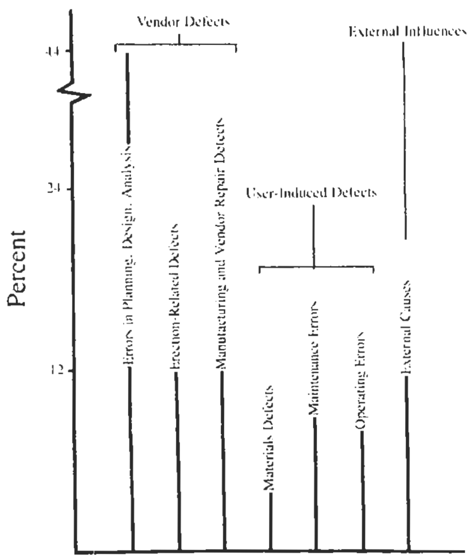


Figure 3-35. Distribution of primary causes of failure for industrial gas turbines. (*Courtesy Der Maschinenschaden, No. 153, 1980.*)

Table 3-7
Primary Causes of Gas Turbine Failures in the USA

Primary Failure Cause, Gas Turbines Driving:	Faulty Maintenance	Design Problems	Operating Problems	Bad Repairs	Other
Generators (utility units)	42%	19%	14%	4%	21%
Compressors, pumps, etc. (mechanical-drive units)	60%	14%	18%	6%	2%

relevant specifications. For instance, the typical yearly repair summary given in Table 3-8 would prove informative to administrative personnel, but would not yield much material of technical relevance.

Once the failure causes are broken down into such elements as O-ring failures, bearing overload failures, incorrect metallurgy, etc., it may become possible to translate failure analysis results into better specifications. For instance, swelling of Viton O-rings in services containing certain chemicals may lead to the specification of Buna or PTFE elastomers; thorough analysis of bearing overload failures could point the way toward requiring pump vendors to provide bearing load values at off-design conditions; and frequent erosion damage in a given pumping service could lead to specifying hard-metal overlays.

Nevertheless, Table 3-8 does represent fairly accurately the pump failure picture of large U.S. refineries. As can be seen, mechanical-seal failures seem to predominate. Similarly, the tabulation summary given in Table 3-9 points to mechanical-seal distress as an area of concern at four overseas plants, also.

Table 3-8
Typical Yearly Repair Summary, Centrifugal Pumps

Centrifugal pumps installed: 2560
Centrifugal pumps operating at any given time: 1252 (average)
Total pumps repaired in 1979: 768
Pumps repaired at site location: 382
Pumps repaired at own shops: 267
Pumps repaired at outside shops: 119

Failure Causes	Repaired on Site	Repaired in Shop
Mechanical seals	112	153
Bearing distress	72	83
Vibration events	8	13
Packing leakage	103	22
Shaft problems/couplings	21	60
Case failure/auxiliary lines	35	2
Stuck	20	13
Bad performance	4	18
Other causes	17	22
Total	382	386

Table 3-9
Primary Failure Causes, Centrifugal Pumps
(Four Overseas Plants)

	Repair Frequency	(Percent)
Part	**1976**	**1977**
Mechanical Seals	702 (40)	708 (40)
Antifriction Bearings	122 (7)	111 (6)
Impellers	80 (7)	93 (5)
Shafts	72 (4)	86 (5)
Casings and Diaphragms	49 (3)	24 (1)
Sleeve Bearings	18 (1)	22 (1)
Piping	64 (4)	21 (1)
Conventional Packing	10 (1)	4
Lip-type Seals	3	2
Couplings	44 (3)	2
Suction Strainers	39 (2)	127 (7)
Others	533 (30)	569 (34)
	1,736 (100)	1,769 (100)

Auditing and Reviewing Centrifugal Compressors

There is little difference in how an experienced engineer approaches audit and review tasks for centrifugal compressors as opposed to those for turbines, gears, and other machinery. In each case he must obtain drawings and other technical data from equipment vendors. He will then review all pertinent documentation for consistency, safety, compliance with specifications, etc., and document all areas requiring follow-up. Of course, he will also initiate and review the rotordynamic design audits described earlier in this chapter.

Centrifugal compressor documentation requirements probably exceed those of most other machinery with the possible exception of large mechanical drive steam turbines. A listing of relevant documentation is contained in API Standard 617. Using the API tabulation facilitates outlining the items recommended for review:

1. Certified dimensional outline drawing, including:
   a. Journal-bearing clearances
   b. Rotor float
   c. Labyrinth, packing, and seal clearances
   d. Axial position of impellers relative to guide vanes
   e. List of connections

Journal-bearing clearances may be required for rotor sensitivity studies. Bearing dimensions allow rapid calculation of bearing loading and serve to screen for potential oil whirl.

Labyrinth, packing, and seal clearances may be too tight for normal process operation. The vendor may attempt to show good efficiency (low recirculation) during shop performance tests.

Axial position of impellers relative to guide vanes needs to be reviewed in conjunction with rotor float dimension. Is rubbing likely to occur?

List of connections may uncover dimensional mismatching with purchaser's lines, excessive flow velocities, omission of specified injection points, etc.

2. Cross-sectional drawing and bill of materials

   These documents are primarily used for verification of impeller dimensions and internal porting, visualization of maintenance access, materials selection, and assessment of number of spare parts needed, etc. A copy of this drawing and the bill of materials should also be forwarded to responsible maintenance personnel.

3. Rotor assembly drawing, including:
   a. Axial position from active thrust-collar face to:
      1. Each impeller
      2. Each radial probe
      3. Each journal-bearing centerline
      4. High-pressure side of balance drum
   b. Thrust-collar assembly details, including:
      1. Collar-shaft fit with tolerance
      2. Concentricity (or run-out) tolerance
      3. Required torque for locknut
      4. Surface-finish requirements for collar faces
      5. Preheat method and temperature requirements for "shrunk-on" collar installation
   c. Balance drum details, including:
      1. Length of drum
      2. Diameter of drum
      3. Labyrinth details
   d. Dimensioned shaft end(s) for coupling mounting(s)
   e. Bill of materials

   Axial position data are required for rotor dynamics analyses and maintenance records. Accurate rotor dynamics studies would further require submitting weight or mass moment of inertia data for impellers and balance drum.

   Thrust-collar assembly details are to be analyzed for non-fretting engagement and field maintenance feasibility. Hydraulic fit is preferred.

   Balance-drum details are needed for rotor dynamics analyses and maintenance reviews.

   Dimensioned shaft ends for coupling mountings allow calculation of stress levels, margins of safety, uprateability, and coupling maintenance.

   The bill of materials is used to compare component designs and materials being released for fabrication. Again, the bill of materials will allow definition of spare parts requirements.

4. Thrust-bearing assembly drawing and bill of materials

   These are used to verify thrust-bearing size and capacity. They are important if directed oil-spray lubrication has been specified, and are essential maintenance information.

5. Journal-bearing assembly drawing and bill of materials

   Bearing dimensions are required to calculate bearing loading and rotor dynamic behavior, and for maintenance records.
6. Seal assembly drawing and bill of materials

   These are required to compare seal dimensions, clearances, and tolerances with similar data from seals operating properly under essentially identical operating conditions.
7. Coupling assembly drawing and bill of materials

   These are used for calculations verifying load-carrying capacity, mass moment of inertia, overhung weight, shaft-fit criteria, dimensional compatibility between driver and driven equipment, material selection, match-marking, assembly and disassembly provisions, and spare parts availability.
8. Seal oil or seal gas schematic, including:
   a. Steady-state and transient gas or oil flows and pressures
   b. Control, alarm, and trip settings
   c. Heat loads
   d. Utility requirements including electrical, water, and air
   e. Pipe and valve sizes
   f. Bill of materials

   Oil or gas flows and pressures must change as a function of gas-pressure and compressor-speed changes. The review must verify that the seal gas or seal oil supply can accommodate all anticipated requirements for a given compressor. This would include operation during run-in on air.

   Control, alarm, and trip settings are required for operating and maintenance manuals, as well as for initial field implementation by the contractor.

   Heat loads are required for capacity checks on oil coolers.

   Utilities requirements are required for proper sizing of switchgear, steam lines, etc.

   Pipe and valve sizes are employed in calculations verifying that maximum acceptable flow velocities are not needed.
9. Seal oil or seal gas assembly drawings and list of connections

   These are required for contractor's (purchaser's) connecting design.
10. Seal oil or seal gas component drawings and data, including:
    a. Pumps and drivers, or motive gas suppliers:
       1. Certified dimensional outline drawing
       2. Cross section and bill of materials
       3. Mechanical seal drawing and bill of materials
       4. Priced spare parts list and recommendations
       5. Instruction and operating manuals
       6. Completed data forms for pumps and drivers

    b. Overhead tank, reservoir, and drain tanks:
       1. Fabrication drawings
       2. Maximum, minimum, and normal liquid levels
       3. Design calculation

c. Coolers and filters:
    1. Fabrication drawings
    2. Priced spare parts list and recommendations
    3. Completed data form for cooler(s)

d. Instrumentation:
    1. Controllers
    2. Switches
    3. Control valves
    4. Gauges

Pumps and drivers are reviewed for accessibility, coupling arrangements, baseplate mounting method, proximity of discharge and suction pipe, etc. Motive gas suppliers are given applicable scrutiny.

Overhead tank, main reservoir, and drain tanks (degassing tank, sour seal oil reservoir) must comply with specifications. Should overhead tanks be given thermal insulation?

Coolers must be suitable for *heating* the seal oil during oil flushing operations. Are they sized to cool the oil flow resulting from more than one pump operation? Can filters be fully drained? Do they have vent provisions? What is their collapsing pressure? What kind of cartridges do they accept? Specification compliance must be ascertained.

Is instrumentation accessible? Can it be checked, calibrated, or replaced without causing a shutdown? Is it properly identified? Are controllers and transmitters located at optimum locations for rapid sensing and control? Are switches of sound design and are they manufactured by a reputable company? Control valves sized right? Gauges made of acceptable metallurgy? Proper ranges?

11. Lube oil schematic, including:
    a. Steady-state and transient oil flows and pressures
    b. Control, alarm, and trip settings
    c. Heat loads
    d. Utility requirements including electrical, water, air, steam, and nitrogen
    e. Pipe and valve sizes
    f. Bill of materials

Are steady-state and transient flows within capability of pumps and accumulator? Will pumps and accumulators satisfy turbine hydraulic transients? Accumulator maintainable?

Are control, alarm, and trip settings tabulated?

Do heat loads have to be accommodated by fouled coolers?

Utility requirements are needed to allow plant design to proceed in such areas as electrical protective devices, water supply lines, and nitrogen supply for blanketing of reservoir. Steam requirements must be identified for turbine-driven pumps.

Pipe and valve sizes need to be checked to determine acceptable flow velocity.

The bill of materials should be reviewed to identify both inexpensive and hard-to-obtain components. It should be reviewed also by maintenance personnel. Are O-rings, rolling element bearings, etc., identified so as to allow purchase from the actual manufacturers of these components?

12. Lube oil assembly drawing and list of connections

    These are required for contractor's (purchaser's) connecting design.

13. Lube oil component drawings and data, including:
    a. Pumps and drivers:
       1. Certified dimensional outline drawing
       2. Cross section and bill of materials
       3. Mechanical seal drawing and bill of materials
       4. Performance curves for centrifugal pumps
       5. Priced spare parts list and recommendations
       6. Instruction and operating manuals
       7. Completed data forms for pumps and drivers

    b. Coolers, filters, and reservoir:
       1. Fabrication drawings
       2. Maximum, minimum, and normal liquid levels in reservoir
       3. Completed data form for cooler(s)
       4. Priced spare parts list and recommendations

    c. Instrumentation:
       1. Controllers
       2. Switches
       3. Control valves
       4. Gauges

    Refer back to item 10. The same reviews are necessary here. Note here that performance curves are required whenever pumps are involved, regardless of whether they are of the centrifugal or positive-displacement (screw) type. Positive-displacement pumps undergo "slippage" which varies with the viscosity of pumpage.

    Instruction and operating manuals are intended for future incorporation in owner's "Mechanical Procedures Manual" and conventional plant operating manuals.

14. Electrical and instrumentation schematics and bill of materials

    The machinery review engineer should be given responsibility for obtaining these data and forwarding them to his electrical/instrument engineering counterparts for review and comment.

15. Electrical and instrumentation arrangement drawing and list of connections

    Same as item 14. At the completion of reviews by electrical/instrument engineering personnel, the final arrangement will be implemented by the contractor.

16. Polytropic head and polytropic efficiency versus ICFM curves for each section or casing on multiple section or casing units in addition to composite curves at 80%, 90%, 100%, and 105% of rated speed

    Note: Request information on probable location of surge line for various molecular weight gases, as required.

    These are important data for future uprate and general performance verification studies, and can be used for purchaser's check on vendor's predicted performance.

17. Discharge pressure and brake horsepower versus ICFM curves at rated conditions for each section or casing on multiple section or casing units in addition to composite curves at 80%, 90%, 100%, and 105% of rated speed

    For variable molecular weight (MW) gases, curves also shall be furnished at maximum and minimum MW. For air compressors, curves also shall be furnished at three additional specified inlet temperatures.
18. "Pressure above suction pressure behind the balance drum" versus "unit loading of the thrust shoes," both in pounds per square inch (bar), using rated conditions as the curve basis

    The curve shall extend from a pressure equal to suction pressure behind the drum to a pressure corresponding to at least 500 pounds per square inch (35 ata) unit loading on the thrust shoes. Balance drum OD, effective balance drum area, and expected and maximum recommended allowable pressure behind the balance drum shall be shown on the curve sheet.

    Will balance drum labyrinth wear cause overloading of the thrust bearing? What happens when fouling (polymerization) occurs in the balance line? Is the design safe for a wide range of suction pressures?
19. Speed versus starting torque curve

    Will the motor be designed to safely start the compressor? Even more important for gas turbine drivers!
20. Vibration analysis data, including:
    a. Number of vanes—each impeller
    b. Number of vanes—each guide vane
    c. Number of teeth—gear-type couplings

    These are required for machine signature "real time" on line diagnostic or spectrum analysis. They will allow identification of relevant frequencies, and possibly be useful in determining which component has undergone deterioration. Refer also to the illustrative example in Chapter 1, "How to Deal with the Typical API Data Sheet."
21. Lateral critical analysis, including:
    a. Method used
    b. Graphical display of bearing and support stiffness and its effect on critical speeds
    c. Graphical display of rotor response to unbalance
    d. Graphical display of overhung moment and its effect on critical speed

    Reviews will identify whether there is risk of operating too close to critical speed, or whether rotor is likely to vibrate at the slightest sign of unbalance. If gear couplings are used, the effective (instantaneous) overhung moment may change as a function of tooth loading or tooth friction. The probability of encountering critical speed problems as a function of gear-coupling deterioration can be investigated by examining graphical displays of effective overhung moment versus critical speed.
22. Torsional critical speed analysis for all motor and gear units, including:
    a. Method used
    b. Graphical display of mass-elastic system
    c. Tabulation identifying the mass-moment torsional stiffness for each component in the mass-elastic system

d. Graphical display of exciting sources (revolutions per minute of any gear in the train, etc.)
e. Graphical display of torsional critical speeds and deflections (mode-shape diagrams)

Torsional critical speeds coinciding with the running speeds of rotating elements in a turbine-gear-compressor or motor-gear-compressor train can cause oscillatory forces of such magnitude as to drastically shorten component life. The data listed are required to determine the probability of speed coincidence, and should coincidence exist, they will allow calculation of resulting stresses. The purchaser may opt to duplicate the manufacturer's torsional analysis with in-house or outside resources. Alternatively, he may arrange for a field test of actual torsional stresses.

23. Transient torsional analysis for all synchronous motor-driven units

Transient, momentary torsional stresses on synchronous motors can be extremely severe and have been responsible for a number of catastrophic failures. The vendor should submit his analysis for review by the purchaser or his consultants.

24. Allowable flange loading (not to be exceeded by piping forces and moments)

These forces and moments can be readily calculated by computers, and virtually all contractors now employ this analysis tool. Correctly used, it will ensure that equipment flange loadings remain within acceptable limits not only under all foreseeable operating conditions but also while spare equipment connected to the same piping system is temporarily removed for maintenance.

25. An alignment diagram, including recommended limits during operation

Cold-alignment offset calculations are to be reviewed for accuracy and appropriateness of manufacturer's assumptions. These data are then used for initial cold alignments (via reverse indicator readings).

26. Weld procedure

These procedures are commonly reviewed by the purchaser's metallurgy specialists. Improper procedures have been responsible for commissioning delays and serious failures. A review of weld procedures can encompass piping, vessels, machinery casings, and even fan-blade spares.

27. Hydrostatic test logs

Together with weld procedures, hydrostatic test logs should become part of the inspection record system of modern process plants.

28. Mechanical run test logs, including:

a. Oil flows and temperatures
b. Vibration
c. Bearing-metal temperatures
d. Actual critical speeds

These test logs should provide verification for all predicted values. If audits and reviews have been properly conducted, the mechanical run tests will, at best, uncover vendor quality-control errors. Deep-seated design errors should not surface at this stage in the job execution. The mechanical run test can provide typical target values for comparison with initial field operation of major machinery. These logs should be retained for future reference.

29. Rotor-balance logs

    Rotor-balance target values given by the manufacturer can be compared with minimum requirements quoted in the literature. Figure 3-36 shows a typical comparison chart. Rotor-balance logs should also be retained in the purchaser's equipment records.

30. Rotor mechanical and electrical run-out

    Maximum acceptable mechanical run-out values are specified in the API standards.

31. "As-built" data sheets

    "As-built" data sheets as shown in Chapter 1 are the key ingredient of a machinery-turnaround records system. The merits of cataloging these essential data are self-evident. Observation and determination of wear is important for failure analysis, and "as-built" data sheets provide a record of materials used in equipment fabrication. Furthermore, these sheets allow both determination and restoration of worn components.

32. "As-built" dimensions and data

    a. Shaft or sleeve diameters at:
        1. Thrust collar
        2. Each seal component
        3. Each impeller
        4. Each interstage labyrinth
        5. Each journal bearing

    b. Each impeller bore
    c. Each labyrinth bore
    d. Each bushing-seal component

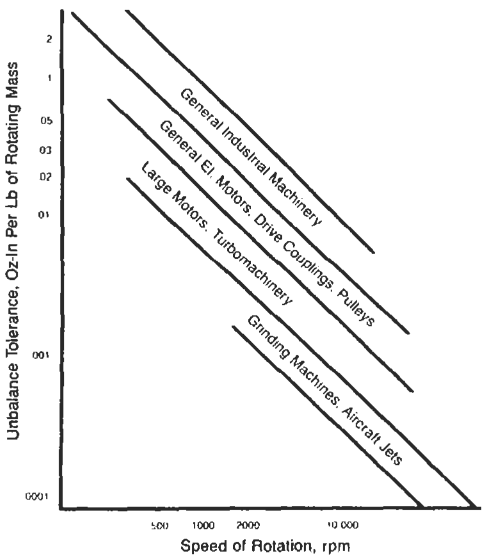


Figure 3-36. Unbalance tolerance versus speed. ***(Courtesy VDI Standards.)***

e. Each journal-bearing ID
f. Thrust-bearing concentricity
g. Metallurgy and heat treatment for:
 1. Shafts
 2. Impellers
 3. Thrust collars
 4. Hardness readings when H_2S is present

Typical wear parts, or parts typically destroyed by massive equipment failures, are listed as items a through f. Metallurgy and heat treatment of highly stressed parts coming in contact with H_2S are important if failure due to stress-corrosion cracking is to be avoided (g).

33. Operating and maintenance manuals

Manuals shall be furnished describing installation, operation, and maintenance procedures. Each manual shall include the following sections:

Section 1—Installation:

a. Storage instructions
b. Foundation
c. Grouting
d. Setting equipment, rigging procedures, and component weights
e. Alignment
f. Piping recommendations
g. Composite outline drawing for compressor train including anchor-bolt locations

While primarily used for installation guidance, Section 1 contains information which should go into the purchaser's construction record system.

Section 2—Operation:

a. Startup
b. Normal shutdown
c. Emergency shutdown
d. Operating limits
e. Routine operational procedures
f. Lube and seal oil recommendations

Section 2 contains information that the purchaser's machinery engineers should utilize in developing such comprehensive machinery commissioning instructions as lube and seal oil flushing and checkout procedures, compressor air, helium, or vacuum run-in instructions, compressor process runs, and compressor field-performance runs.

Section 3—Disassembly and reassembly instructions:

a. Rotor in casing
b. Rotor unstacking and restacking procedures
c. Journal bearings
d. Thrust bearings
e. Seals
f. Thrust collars

These instructions are indispensable for field maintenance. They should preferably go into a "mechanical procedures," or "turnaround" manual. Close screening of these instructions may reveal special tooling or shop-facilities requirements.

Section 4—Performance curves:
a. Polytropic head and polytropic efficiency versus ICFM
b. Discharge pressure and brake horsepower versus ICFM
c. Balance-drum pressure versus thrust loading
d. Speed versus starting torque

Performance curves allow us to review probable versus actual surge limits, probable uprate capabilities of major machinery, and maximum permissible balance-line fouling before onset of thrust-bearing distress. The speed versus starting torque characteristic curves are important for driver sizing.

Section 5—Vibration data:
a. Vibration analysis data
b. Lateral critical speed analysis
c. Torsional critical speed analysis
d. Transient torsional analysis.

Vibration data previously submitted by the vendor are updated, supplemented, or simply included in their original form for the purpose of having official data in a single manual.

Section 6—"As-built" data:
a. "As-built" data sheets
b. "As-built" dimensions and data
c. Hydrostatic test logs
d. Mechanical run test logs
e. Rotor-balance logs
f. Rotor mechanical and electrical run-out

As in Section 5, these provide a comprehensive, final, updated issue.

Section 7—Drawing and data requirements:
a. Certified dimensional outline drawing and list of connections
b. Cross-sectional drawing and bill of materials
c. Rotor drawing and bill of materials
d. Thrust-bearing assembly drawing and bill of materials
e. Journal-bearing assembly drawing and bill of materials
f. Seal assembly drawing and bill of materials
g. Seal oil or seal gas schematic and bill of materials
h. Seal oil or seal gas arrangement drawing and list of connections
i. Seal oil or seal gas component drawings and data
j. Lube oil schematic and bill of materials
k. Lube oil arrangement drawing and list of connections
l. Lube oil component drawings and data
m. Electrical and instrumentation schematics and bill of materials
n. Electrical and instrumentation arrangement drawing and list of connections

Again, this provides a final wrapup of certified drawings for the purchaser's permanent records.

The vendor will usually issue official operating and maintenance manuals around the time the completed machine leaves his factory. While this may be acceptable in view of the fact that all of these data had previously been made available to the purchaser or his contractor, any delay in the *originally* scheduled transmission of essential documents may have serious impact on the completion of construction and, therefore, machinery startup targets. To forestall any such delays, project managers would be well advised to link progress payments to the timely and diligent execution of all data-transmittal requirements. Once full payment has been made while certain review documents are yet outstanding, the purchaser loses virtually all his leverage and may not be able to proceed with critically important review or audit efforts.

For additional discussion of machinery-related documentation, refer to Chapter 1, "Specifying Machinery Documentation Requirements."

Auditing and Reviewing Steam Turbines

Reliability audits and reviews for steam turbines differ little from those described earlier for centrifugal compressors. While in-depth audits are rarely justified for small, general purpose steam turbines, such audits are well worth pursuing for the majority of special purpose machines, some of which are now driving process plant compressors in the 60,000-KW range. Conventional, on-going reliability reviews are appropriate for both special- and general purpose steam turbines

API Standard 612 lists most of the documentation helpful in this job. Virtually all of the documents tabulated must be reviewed for general purpose *and* special purpose turbines. In addition, audits are recommended for special purpose turbine documentation identified with an asterisk (*).

1. Certified dimensional outline drawing, including list of connections
2. Cross-sectional drawings and bill of materials, including:
   *a. Journal-bearing clearances and tolerance
   b. Rotor float (axial)
   *c. Seal clearances (shaft end and internal labyrinth) and tolerance
   *d. Axial position of wheel(s) relative to inlet nozzle or diaphragms and tolerance
   e. OD of all wheel(s) at blade tip
3. Rotor assembly drawing, including:
   a. Axial position from active thrust-collar face to:
      1. Each wheel (inlet side)
      2. Each radial probe
      3. Each journal-bearing centerline
      4. Key phasor notch
   b. Thrust-collar assembly details, including:
      *1. Collar-shaft fit with tolerance
      *2. Concentricity (or axial run-out) tolerance

3. Required torque for locknut
4. Surface-finish requirements for collar faces
*5. Preheat method and temperature requirements for "shrunk-on" collar installation

c. Dimensioned shaft end(s) for coupling mounting(s)
d. Bill of materials

4. *Thrust-bearing assembly drawing and bill of materials
5. *Journal-bearing assembly drawing and bill of materials
6. *Packing and/or labyrinth drawings and bill of materials
7. Coupling assembly drawing and bill of materials
8. Gland sealing and leakoff schematic including:
   a. Steady-state and transient steam and air flows and pressures
   b. Relief and control valve settings
   c. Utility requirements including electrical, water, steam and air
   d. Pipe and valve sizes
   e. Bill of materials
9. *Gland sealing and leakoff arrangement drawing and list of connections
10. Gland Sealing and leakoff component outline and sectional drawings and data including:
    a. Gland condenser fabrication drawing and bill of materials, also completed API data form for condenser
    b. Air or water ejector drawing and performance curves
    c. Control valves, relief valves, and instrumentation
    d. Vacuum pump schematic, performance curves, cross section, outline drawing, and utility requirements (if pump is furnished)
11. Lube oil schematic, including:
    *a. Steady-state and transient oil flows and pressures at each use point
    b. Control, alarm, and trip settings (pressure and recommended temperature)
    c. Heat loads at each use point and maximum load
    d. Pipe and valve sizes
    e. Utility requirements including electrical, water, and air
    f. Bill of materials
12. Lube oil system arrangement drawing and list of connections
13. Lube oil component drawings and data, including:
    a. Pumps and drivers:
       1. Certified dimensional outline drawing
       2. Cross section and bill of materials
       3. Mechanical seal drawing and bill of materials
       4. Performance curves for centrifugal pumps
       5. Instruction and operating manuals with completed API data forms for drivers
    b. Coolers, filters, and reservoir
       1. Fabrication drawings
       2. Maximum, minimum, and normal liquid levels in reservoir
       3. Completed TEMA data sheets for coolers
    c. Instrumentation:
       1. Controllers
       2. Switches

3. Control valves
4. Gauges

d. Priced spare parts list(s) and recommendations

14. Electrical and instrumentation schematics and bill of materials
15. Electrical and instrumentation arrangement drawing(s) and lists of connections
16. Control and trip system data
    a. Valve lift sequence on multivalves and final settings
    b. Pin-setting diagram
    *c. Control and trip system drawings
    d. Control setting instructions
17. *Governor cross-section and setting instructions
18. Steam flow versus horsepower curves at normal and rated loads and speeds
19. Steam flow versus first-stage pressure curve for multistage machines or nozzle bowl pressure for single-valve single-stage machines at normal and rated speed with normal steam conditions
20. Steam flow versus speed and efficiency curves at normal steam conditions
21. *Steam flow versus thrust-bearing load curve
22. Extraction performance curves
23. *Steam condition correction factors:
    a. Inlet pressure to maximum and minimum values listed on data sheet in increments agreed at order
    b. Inlet temperature to maximum and minimum values listed on data sheet in increments agreed at order
    c. Speed (80%–105%—5% increments)
    d. Exhaust pressure to maximum and minimum values listed on data sheet in increments agreed at order
24. Vibration analysis data, including:
    a. Number of blades—each wheel
    b. Number of blades—each diaphragm
    c. Number of nozzles—nozzle block, single valve
    —nozzle section, multivalve
    d. Campbell diagram
    e. Goodman diagram
25. *Lateral critical analysis report, including:
    a. Method used
    b. Graphical display of bearing and support stiffness and its effect on critical speeds
    c. Graphical display of rotor response to unbalance
    d. Graphical display of overhung moment and its effect on critical speed
26. Allowable flange loading(s)
27. *A coupling alignment diagram, including recommended limits during operation Note: All shaft end position changes and support growths from 70°F (21°C) ambient reference
28. *Weld procedures
29. Hydrostatic test logs
30. Mechanical run test logs including:
    a. Oil flows, pressures, and temperatures

b. Vibration including X-Y plot of amplitude versus rpm during startup and shutdown
c. Bearing-metal temperatures
d. Observed critical speeds

31. Rotor-balance logs
32. Rotor mechanical and electrical run-out
33. "As-built" API data sheets
34. "As-built" dimensions (including design tolerances) and/or data for:
    a. Shaft or sleeve diameters at:
        1. Thrust collar (for separate collars)
        2. Each seal component
        3. Each wheel (for "stacked" rotors)
        4. Each interstage labyrinth
        5. Each journal bearing
    b. Each wheel bore (for "stacked" rotors) and OD
    c. Each labyrinth or seal ring bore
    d. Thrust-collar bore (for separable collars)
    e. Each journal-bearing ID
    f. Thrust-bearing concentricity (axial run-out)
    g. Metallurgy and heat treatment for:
        1. Shaft
        2. Wheels
        3. Thrust collar
        4. Blades (buckets)
35. Operating and maintenance manuals

    Manuals shall be furnished describing installation, operation, and maintenance procedures. Each manual shall include the following sections:

    Section 1—Installation:

    a. Storage
    b. Foundation
    c. Grouting
    d. Setting equipment, rigging procedures, and component weights
    e. Alignment
    f. Piping recommendations including allowable flange loads
    g. Composite outline drawing for driven/driver train, including anchor-bolt locations

    Section 2—Operation:

    a. Startup
    b. Normal shutdown
    c. Emergency shutdown
    d. Operating limits
    e. Lube oil recommendations

    Section 3—Disassembly and reassembly instructions:

    a. Rotor in casing
    b. Rotor unstacking and restacking procedures
    c. Journal bearings

d. Thrust bearing
e. Seals
f. Thrust collar
g. Wheel reblading procedures

Section 4—Performance curves:
a. Steam flow versus horsepower
b. Steam flow versus first-stage pressure
c. Steam flow versus speed and efficiency
d. Steam flow versus thrust-bearing load
e. Extraction curves
f. Steam condition correction factors (prefer nomograph)

Section 5—Vibration data:
a. Vibration analysis data
b. Lateral critical speed analysis

Section 6—"As-built" data:
a. "As-built" API data sheets
b. "As-built" dimensions and/or data (refer to item 34)
c. Hydrostatic test logs
d. Mechanical run test logs
e. Rotor balance logs
f. Rotor mechanical and electrical run-out log

Section 7—Drawing and data requirements:
a. Certified dimensional outline drawing and list of connections
b. Cross-sectional drawing and bill of materials
c. Rotor drawing and bill of materials
d. Thrust-bearing assembly drawing and bill of materials
e. Journal-bearing assembly drawing and bill of materials
f. Seal component drawing and bill of materials
g. Lube oil schematic and bill of materials
h. Lube oil arrangement drawing and list of connections
i. Lube oil component drawings and data
j. Electrical and instrumentation schematics and bill of materials
k. Electrical and instrumentation arrangement drawing and list of connections
l. Control and trip system drawings and data

Note: Items 11, 12, 13, and 35, Section 7, g through i inclusive, required only for turbine manufacturer's scope of supply

Evaluating Major Reciprocating Compressors

Occasionally, reliability reviews extend into the field of reciprocating compressors. Their inherent efficiency and ability to achieve high compression ratios have led to the design of reciprocating compressors absorbing in excess of 20,000 HP (~15,000 kw) and discharge pressures in excess of 50,000 psi (3,447 bar). Very often, compressor manufacturers are asked to propose larger and larger machines. This leads to certain extrapolations and occasional risks for the purchaser who may end up serving as the vendor's test bed or research auxiliary.

Evidently, careful review and selection procedures may be in order. Presented are sample component analysis procedures which serve to identify components that are potentially limiting in uprate or extrapolation situations. In turn, these analyses will lead to a more objective evaluation of competitive bids for major reciprocating compressors from several manufacturers.

Manufacturers' Scale-Up Philosophies Differ. Generally speaking, compressor-frame ratings are fixed by component sizing criteria used in the initial design of the machine. With few exceptions, the first several compressor frames of a given model or frame designation are sold for, or operating with, throughput rates which keep component loading conservatively low. In an uprate situation, the manufacturer will have to investigate the suitability of a given frame for higher throughput rates than are indicated by prior experience. If higher throughput is to be achieved merely by pressure increases while maintaining the actual inlet volume relatively unchanged, the effect on component loading can be readily assessed and the adequacy of the components in question analyzed.

However, increased throughput must often be achieved without changes in pressure levels. In this case, the manufacturer has one or more design options to choose from. Among these options are:

- Increasing the compressor rpm.
- Increasing the piston or plunger diameter.
- Increasing the piston or plunger stroke.
- Adding more cylinders.

The purchaser is now faced with the task of analyzing the potential risks associated with the scale-up approach used by a given manufacturer. The machinery engineer will immediately recognize that:

- Increasing the compressor rotative speed will result in higher linear piston velocities, higher inertia loading of compressor components, and more valve cycles over a given time period.
- Increasing the piston or plunger diameter will result in higher component loading and may raise some questions as to proven manufacturing experience with unusually large piston or plunger diameters. In some cases, the heat dissipation achievable with large-diameter pistons or plungers may become a point of contention.
- Increasing the piston or plunger stroke will result in higher linear piston velocities, crankshaft redesign, and different component loading.
- Extending the compressor frame length to add more crank throws (cylinders) requires primarily a statistical assessment of the added downtime risk resulting from more parts, and also an analysis of increased costs of foundation and piping.

Detailed Component Analysis Recommended. The potential purchaser of extended-size reciprocating compressors should familiarize himself with the experience of any manufacturer under close consideration. Examination of a manufacturer's experience should serve to identify design extrapolations, or scale-ups. In examining manufacturers' experience, the purchaser's engineer *cannot,* in many cases, look at data such as maximum horsepower compressor built to date. The following example will illustrate this point.

Let us assume a machine has been furnished with 10 equally loaded cylinders absorbing a total of 14,000 HP. The manufacturer may proudly point to his 14,000 HP experience when bidding, say, an 8-cylinder machine absorbing "only" 13,000 HP. He is not likely to emphasize that his 13,000 HP machine may subject many of the compressor components to forces that are approximately 16% higher than those seen by his well-proven 14,000 HP machine.

The value of component strength comparisons between competing offers is also evident from a review of manufacturers' frame-rating terminology. How does one compare one vendor's "instantaneous overload capacity" with another vendor's "continuous overload capacity," "rated capacity," or "design capacity?" Component strength analysis is one way to obtain a grasp of design conservatism and, thus, vendor standing. This analysis should include the load-carrying capability of the various compressor bearings, cylinder design factors of safety, and combined gas and inertia loading of compressor components.

Reciprocating-compressor component strength analysis requires summing up the gas and inertia forces acting on opposing pistons or plungers. Figures 3-37 and 3-38 are graphical representations of the resultant forces on first- and second-stage crank bearings of a large hyper compressor used for ethylene compression. Instantaneous gas pressures acting in hyper-compressor cylinders can be calculated with the method shown later. The same section shows a typical sample calculation for inertia forces acting at a particular crank angle. These calculations lead to a tabulation of

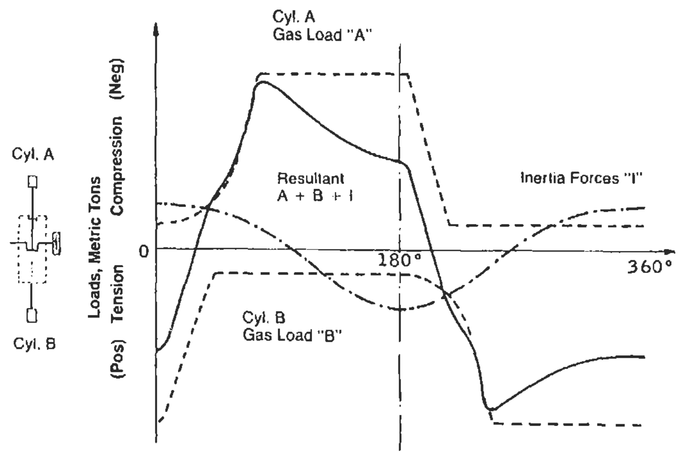


Figure 3-37. Forces acting on first stage of typical hyper compressor.

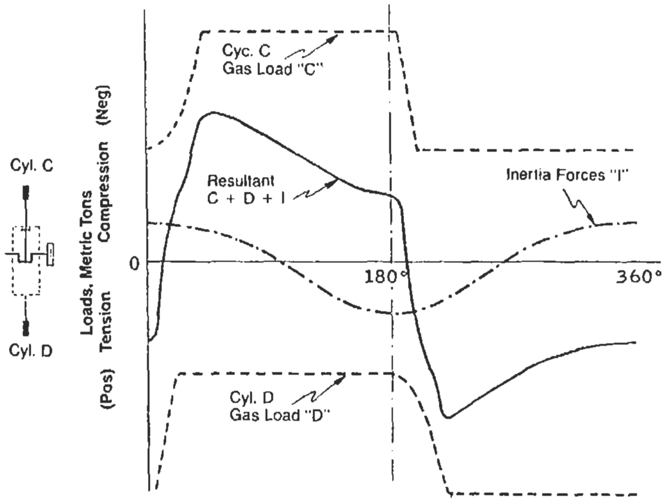


Figure 3-38. Forces acting on a second stage of typical hyper compressor.

design conservatism for selected components. The actual factor of safety built into a given component may be the subject of debate; however, comparing the ratios of maximum allowable loading versus actual loading in the designs of manufacturers X and Y does indeed give a relevant indication of comparative strengths.

Other Factors to Be Considered. In addition to performing detailed analyses of component strength, the potential purchaser should review the safety and maintenance requirements of a given design.

Safety reviews must include an assessment of the likelihood of piston or plunger failures occurring during machine operation. The consequences of such failures must be studied and understood. Failure likelihood and failure consequences are influenced by elements such as cylinder misalignment; provisions to effectively vent leakage flow from packing and frame areas; provisions to permit cylinder head bolts to lift off, or stretch, well within the material elastic limit during potential decomposition events; and so forth.

The relative ease and projected frequency of reciprocating-compressor maintenance may prove to be a significant cost factor. The review should focus on such potential needs as cylinder support, cylinder alignment, bearing and slide replacements, valve maintenance, and hydraulic tensionsing provisions. Feedback from user installations may be helpful to a degree, but experience shows that variations in catalysts or lubricants used by different installations often represent enough variables to question the validity of direct comparisons of, say, maintenance frequency for valves.

Vendor Standings Can Be Compared. Vendor selection based purely on installed cost of compressors may be quite misleading because overall operating costs are obviously affected by many of the factors mentioned earlier. However, an effort can be made to compare vendor standings by using a tabular rating similar to

Table 3-10
Comparison of Vendor Standings (Illustrative Example Only)

		Rank		Value × Rank	
	Value	Vendor X	Vendor Y	Vendor X	Vendor Y
Technical backup	3	4	8	12	24
Frame Rating	2	4	5	8	10
Packing cup experience	2	4	4	8	8
Piston or plunger failure	3	8	8	24	24
Leakage disposal	3	7	7	21	21
Spare parts supply	2	7	5	14	10
Erection coverage	1	5	5	5	5
Turnaround time	1	6	6	6	6
Auxiliary systems	1	6	6	6	6
Hydraulic tensioning	1	6	7	6	7
Piston speed conservatism	2	4	5	8	10
Cylinder alignment ease	2	6	8	12	16
Valve experience	2	6	6	12	12
Winterizing experience	1	8	5	8	5
Bearing material and design	2	7	9	14	18
Inspection and quality control	2	6	9	12	18
Cooling oil system	1	4	8	4	8
Total				180	208

Value scale: 3 = very important
2 = average importance
1 = below average

Ranking scale: 10 = perfect, no improvement possible
5 = average quality
1 = barely acceptable

the one shown in Table 3-10. Factors to be considered in compressor selection are assigned values ranging from 3 (very important) to 1 (below average). These values are then multiplied by ranking numbers from 10 (perfect, no improvement possible) to 1 (barely acceptable) and the resulting products added for the various offers under consideration. The total summation can be used as an indication of relative standing among competitive offers.

Typical Component Strength Analysis for Hyper Compressors

Assume a 214 rpm machine, with moving components weighing 4,000 kg:

1. Determination of instantaneous gas pressure in cylinder, P.

Where

P_S = suction pressure in cylinder
P_D = discharge pressure in cylinder
ϵ = clearance volume in cylinder
S' = fraction of total stroke (at instant of calculation)
S = total maximum stroke length
k = polytropic exponent at compression
k' = polytropic exponent at expansion

Approximation for S′:

$S' = R\,[1 - \cos\Theta + (R\sin^2\Theta)/2\ell]$, where

$R = 1/2$ stroke, and $R/2\ell = \text{ratio}\ \dfrac{\text{stroke}/2}{\text{connecting rod length}}$

Θ = crank angle, starting at inboard zero reference

$$P = P_S\left(\frac{1+\epsilon}{1+\epsilon-S'/S}\right)^k \quad \text{during compression stroke}$$

$$P = P_D\left(\frac{\epsilon}{\epsilon+S'/S}\right)^{k'} \quad \text{during expansion stroke}$$

2. Determination of maximum gas force in a representative first stage.
   $P_D - P_S = 2{,}000 - 300 \quad = 1{,}700\ \text{kg/cm}^2$ (= net effective pressure)
   Plunger diameter = 9 cm
   Plunger area $\dfrac{(\pi)(81)}{4} = 63.8\ \text{cm}^2$
   Net effective plunger force = (1,700) (63.8) = 108,200 kg
3. Determination of maximum gas force in a representative second stage.
   $P_D - P_S = 3{,}600 - 1{,}750 \quad = 1{,}850\ \text{kg/cm}^2$ (= net effective pressure)
   Plunger diameter = 8 cm
   Plunger area $\dfrac{(\pi)(64)}{4} = 50.3\ \text{cm}^2$
   Net effective plunger force = (1,850) (50.3) = 93,000 kg
4. Inertia load contribution during time of maximum gas load. Here we are using crank angle $\Theta = 260°$, 80° + 180°—a typical arbitrary example. In Figure 3-39:

$$X = C_oA - CA,\ CA = CD + DA$$
$$C_oA = \ell + R,\ CA = \ell\cos\varnothing + R\cos\Theta$$
$$X = \ell + R - \ell\cos\varnothing - R\cos\Theta$$
$$h = \ell\sin\varnothing = R\sin\Theta$$

$$\sin\varnothing = (R\sin\Theta)/\ell,\ \cos\varnothing = \sqrt{1-\sin^2\varnothing}$$
$$= \sqrt{1-(R^2\sin^2\Theta/\ell^2}$$
$$X = \ell + R - \ell\sqrt{1-(R^2\sin^2\Theta/\ell^2} - R\cos\Theta$$

Using binomial theorem expansion

$$\text{i.e. } (a \pm b)^n = a^n \pm na^{n-1}b + \frac{n\,(n-1)}{2!}a^{n-2}b^2 \pm \ldots.$$

$$(\ell^2 - R^2\sin^2\Theta)^{1/2} = \ell - \frac{R^2\sin^2\Theta}{2\ell} - \frac{R^4\sin^4\Theta}{8\ell^3} + \ldots.$$
$$\approx [1 - (R^2\sin^2\Theta)/2\,\ell^2]\,\ell$$

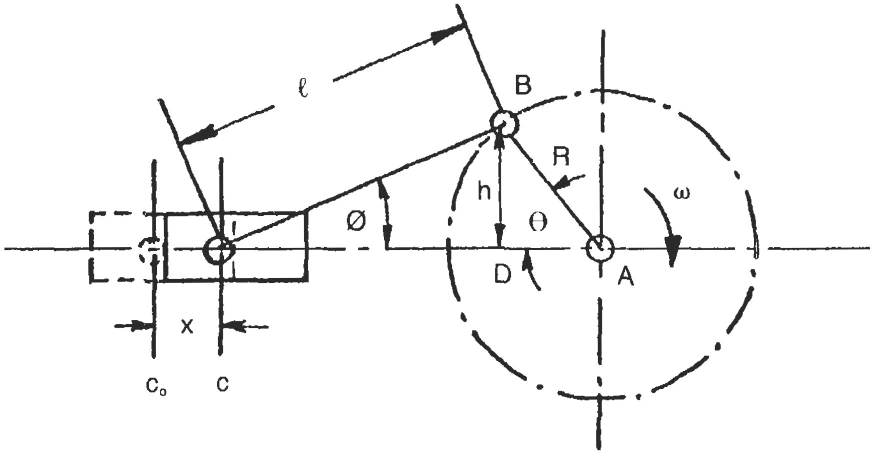


Figure 3-39. Diagram for calculating inertia load contribution during time of maximum gas load on reciprocating compressors.

where

$X = \ell + R - \ell + (R^2 \sin^2 \Theta)/2\ell - R \cos \Theta = R (1 - \cos \Theta) + (R^2 \sin^2 \Theta)/2\ell$

$v = dx/dt = R \sin \Theta\, d\Theta/dt + (R^2/2\ell^2) \sin \Theta \cos \Theta\, d\Theta/dt$

$= R\, \omega \sin \Theta + (R^2/2\ell)\, \omega \sin 2\, \Theta = R\, \omega\, [\sin \Theta + (R/2\ell) \sin 2\, \Theta]$

$a = dv/dt = R\, \omega\, [\cos \Theta\, d\Theta/dt + (R/2\ell)\, 2 \cos 2\, \Theta d\Theta/dt]$

$= R\, \omega^2\, [\cos \Theta + (R/\ell)\cos 2\, \Theta]$

$$\omega = \frac{214 \text{ rev}}{60 \text{ sec}}\ \frac{2\ \pi \text{ rad}}{\text{rev}} = 22.4 \text{ rad / sec}$$

$R = .1875$ m, $R / \ell = .163$ Next, calculate inertia load:

$$F = ma = \left(\frac{4000 \text{ kg}}{9.81 \text{ m / sec}^2}\right)(.1875\text{m})\left(500\ \frac{\text{rad}^2}{\text{sec}^2}\right)[-.174 + (.163 \times .94)]$$

$$= 12{,}500 \text{ kg}$$

5. Combined maximum load, gas, and inertia.
   108,200 + 12,500 = 120,700 kg
6. Bearing manufacturer's data indicating maximum allowable pressures.
   A typical graph of the type shown in Figure 3-40 should be requested from either the bearing or compressor manufacturer. Data derived from these graphs will be used in the next analysis step, determination of design conservatism for bearings, Table 3-11. This step will identify the bearing overload capacity built into competing offers.

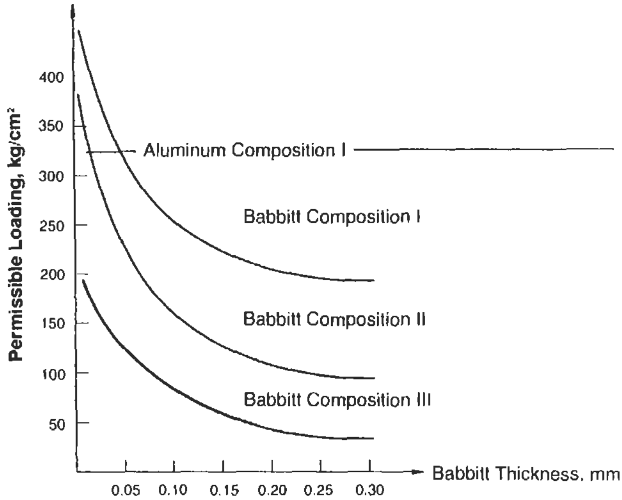


Figure 3-40. Maximum allowable pressures for babbitted bearings.

Reliability Review for Centrifugal Pumps

Significant design differences can exist among centrifugal pumps in spite of their compliance with such accepted industry standards as API 610 or ANSI B73.1 and B73.2. A number of reliability improvement or verification steps can be recommended, and three typical examples of design weaknesses are illustrated later.

Potential design weaknesses can be discovered in the course of examining dimensionally accurate cross-sectional drawings. There are two compelling reasons to conduct this drawing review during the bid-evaluation phase of a project: first, some pump manufacturers may *not* be able to respond to user requests for accurate drawings after the order is placed; and second, the design weakness could be significant enough to require extensive redesign. In the latter case, the purchaser may be better off selecting a different pump model.

Looking at pump components, we note that rolling-element bearings are not always selected for satisfactory operation over a wide range of operating conditions. This is where user experience and better selection guidelines can be of value. Improvements can come from better engineering and better dissemination or presentation of pertinent data. Bearing-related improvements are described later.

Service Life is Influenced by NPSH and Suction-Specific Speed Criteria

It is beyond the scope of this text to examine pump hydraulic performance in detail. Hence, a brief overview will suffice.

Many of the major contractors and experienced machinery engineers in the petrochemical industry are aware that centrifugal pumps with high suction-specific speeds

Table 3-11
Determination of Design Conservatism for Bearings (Arbitrary Example Values)

			Bearing Data				
	Projected Area cm^2	Actual Bearing Pressure kg/cm^2	Composition	Thickness mm	Maximum Allowable Load, kg/cm^2	Ratio of Maximum Permissible/Actual Bearing Pressure	Overload Capacity
Vendor X							
Main Bearing	1,900	35	II	0.20	45	1.29	29%
Crank Bearing	1,700	78	II	0.30	100	1.28	28%
Crosshead Bearing	1,400	95	II	0.30	100	1.05	5%
Vendor Y							
Main Bearing	1,800	40	II	0.20	115	2.88	188%
Crank Bearing	1,800	80	II	0.20	115	1.44	44%
Crosshead Bearing	1,200	120	I	0.10	260	2.16	116%

Result: Overload Capacity Vendor X = 5%. Crosshead bearings are the weakest link
Overload Capacity Vendor Y = 44%. Crank bearings are the weakest link

(N_{SSS})* can often be expected to give satisfactory operation only over a narrow operating range.[33,34] For these pumps, vendor's quoted net positive suction head values ($NPSH_R$) away from the best efficiency point (BEP) are rightly questioned. Also, reasonable explanations have lately been offered in the technical literature linking high specific-speed impellers with recirculation phenomena which, in turn, are influenced by pump geometry. Intolerable recirculation appears to exist in high head pumps using large impeller inlet-eye areas in efforts to achieve low $NPSH_R$. See Figure 3-41 for details.

In a 1977 technical paper,[33] Irving Taylor sought to quantify probable $NPSH_R$ for zero cavitation erosion of pumps in various services.

Providing a series of approximate curves for pumps with different suction-specific speeds (N_{SSS}), he alerted users to the fact that NPSH requirements for long-term satisfactory operation of pumps away from the best efficiency point (BEP) may be sub-

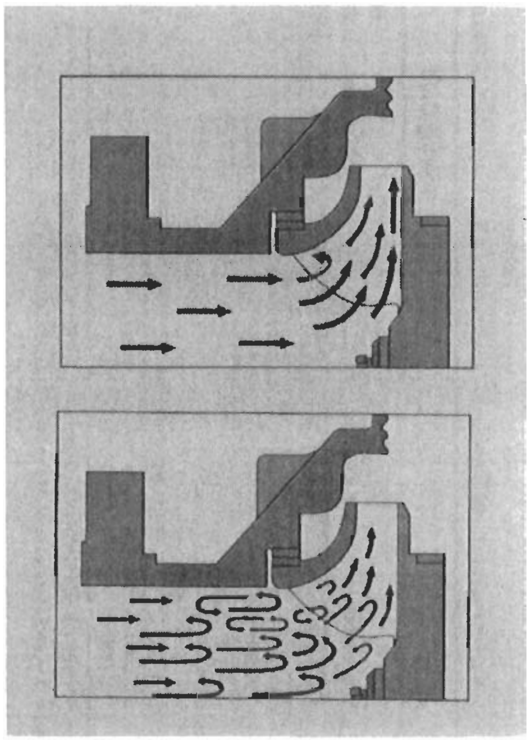


Figure 3-41. Running a single-suction impeller at normal flow (top) avoids the internal recirculation that occurs at low flow conditions (bottom).

$$*N_{SSS} = \frac{N \times Q^{0.5}}{(NPSH_R)^{.75}},$$

where N = pump speed, rpm
Q = pump flow, gpm, at BEP

stantially greater than claimed by the pump vendor. This is in part due to the fact that Hydraulic Institute Standards permit pump manufactures to plot as $NPSH_R$ values those test points where a 3% drop in discharge pressure was noted.[35] Unfortunately, cavitation may already be quite severe at these points. It has also been suspected that vendors have occasionally extrapolated a few test points into a smooth curve that did not represent the real situation.

Then again, user demand for high efficiency, lower cost, and low $NPSH_R$ pumps drove suction-specific speeds into the range above 12,000, where certain deviations from BEP flow could cause internal recirculation phenomena which render pump operation hydraulically unstable. Turbulent flow causes impeller erosion and significant fluctuations in mechanical seal action, bearing loading, and shaft deflection. It is worthwhile to note that for many pumps the manufacturer's published minimum acceptable flow values were based on flow necessary to prevent excessive heat buildup. These values may not even approach the minimum permissible flow for erosion avoidance in high N_{SSS} pumps; they have been found off by very substantial margins on numerous occasions.

In view of this experience, process plants occasionally have used procurement guidelines and "rule-of-thumb" conservatism aimed at achieving higher pump reliability. Table 3-12 represents one of these experience-based guidelines.

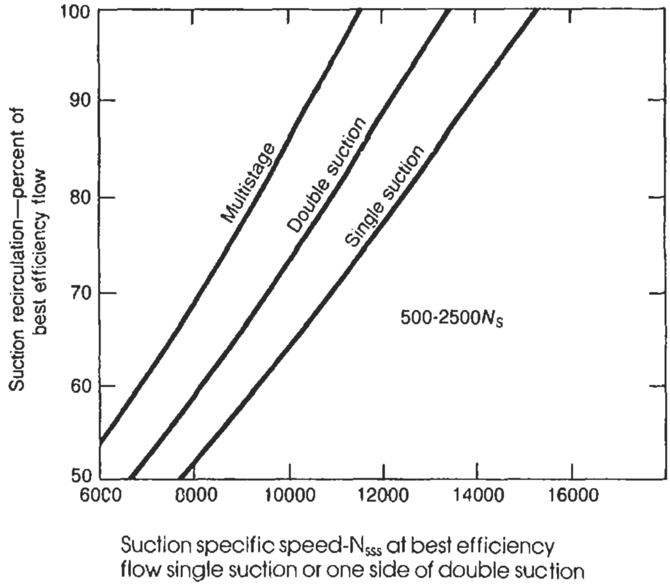
Since about 1980, pump manufacturers have occasionally published data that can be used to better establish the allowable minimum flow for pumps with given designs and operating characteristics. While earlier representations illustrated typical observations but tended to avoid giving numerical values, Figure 3-42 can be used to avoid the selection of pumps which would give poor service at operation away from BEP. The top part of Figure 3-42 should be used with pumps whose specific speed (N_S)* ranges from 500–2,500. The lower part of Figure 3-42 applies to pumps with N_S-values from 2,500 to 10,000.

The following example illustrates how to use the screening values shown in Figure 3-42. Assume we have a 3,560 hp pump with a BEP capacity of 2,600 gpm. A dou-

Table 3-12
Minimum Acceptable Flow for Centrifugal Pumps

Suction Specific Speed, N_{SSS}	Minimum Flow as a Percentage of BEP Flow
<9,000	45%
9,000–9,999	50%
10,000–10,999	60%
11,000–11,999	70%
12,000–12,999	75%
13,000–13,999	80%
14,000–15,999	85%
16,000 and up	90%

*$N_S = N\,(Q)^{0.5}/(H)^{0.75}$ where N = pump speed, rpm;
Q = BEP capacity in gpm; H = head in ft, at BEP capacity of largest diameter impeller



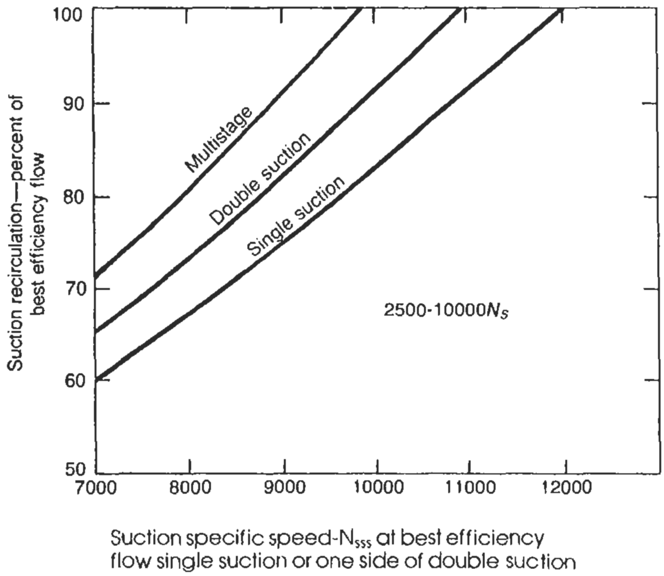


Figure 3-42. Suction recirculation versus suction specific speed N_{SSS}[4]

ble-flow impeller is used, developing 280 ft of head at BEP capacity, and the manufacturer gives an $NPSH_R$ of 23 ft. For this pump the specific speed, N_S, would be

$$N(Q)^{0.5}/(H)^{0.75} = (3{,}560)(1{,}300)^{0.5}/(280)^{0.75} = 1{,}875$$

This indicates that the upper portion of Figure 3-42 should be used. We calculate

$$N_{SSS} = N(Q)^{0.5}/(NPSH_R)^{0.75} = (3{,}560)(1{,}300)^{0.5}/(23)^{0.75} = 12{,}225$$

Entering the horizontal scale at 12,225, we find that for double suction pumps in this specific range, we should expect operation without damaging impeller-internal fluid recirculation, as long as the flow exceeds 88% of the flow at BEP, or $0.88 \times 2{,}600 = 2{,}288$ gpm.

Recognizing that pumps in water service with a flow of 2,500 gpm or less and heads of 150 ft or less operate at only moderate energy levels, Reference 36 suggests that their minimum flow values can be set at 50% of the indicated recirculation values for continuous operation and 25% of the indicated recirculation values for intermittent operation. For pumps of *all* sizes in hydrocarbon service, the minimum flow values can be set at 60% of the indicated recirculation flow if continuous and 25% of the indicated recirculation flow if intermittent operation is anticipated.

A similar graphic representation of minimum continuous stable flow was proposed by C. C. Heald in 1983. His approach is to separate pumps into three size categories: A, B, and C:

A—Single stage, overhung, 2-in. discharge and smaller multistage, 2-in. discharge and smaller
B—Single stage, overhung, 3–4-in. discharge (3,600 rpm), 6-in. discharge and larger (1,800 rpm and lower) single stage, double suction, 4-in. discharge and smaller multistage, 3 discharge and larger
C—Single stage, overhung, 6-in. discharge and larger (3,600 rpm) single stage, double suction, 6-in. discharge and larger

Next, the reviewer would calculate the pump suction specific speed, N_{SSS}, and enter Figure 3-43 at the calculated value. The N_{SSS}-line intersection with the applicable curve would show, on the horizontal scale, the percent-of-the-best-efficiency point capacity for the maximum diameter impeller fitting in this particular pump. Following this step, the reviewer would enter Figure 3-44 at the point indicating ratio $NPSH_A/NPSH_R$, and read off a "minimum flow correction factor" K_m on the horizontal scale. The two values thus found from Figure 3-43 and 3-44 would be multiplied by the flow at best efficiency (BEP flow) of the maximum diameter impeller and the result considered a reasonable approximation for minimum continuous stable flow (MCSF).

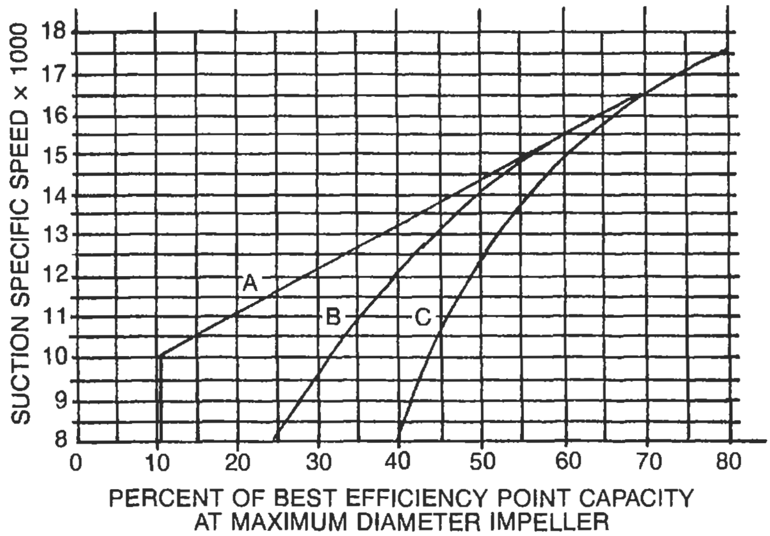


Figure 3-43. Minimum continuous stable flow versus suction specific speed of centrifugal pumps for general refinery services. (*Courtesy C. C. Heald and R. Palgrave, Form 70941, Ingersoll-Rand Company, 1985.*)

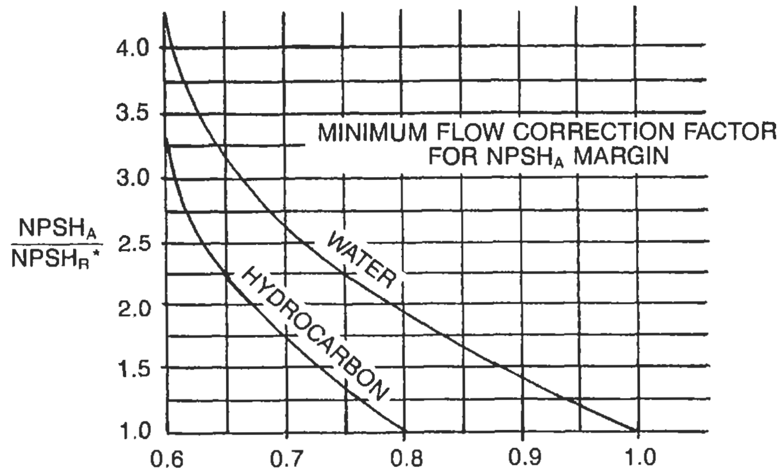


Figure 3-44. Minimum flow correction factor for NPSHA margin. (*Courtesy C. C. Heald and R. Palgrave, Form 70941, Ingersoll-Rand, 1985.*)

The following sample problem will illustrate this particular method:

Example: Pump—6 × 8 × 10 overhung process pump
Speed—3,560 rpm
Design conditions—1,600 gpm
300 ft TDH
31 ft $NPSH_A$
0.85 S.G. hydrocarbon
Pump design data—22 ft $NPSH_R$ at BEP
14,000 N_{SSS} at 1,600 gpm (BEP)
From MCSF chart: MCSF = 56% of BEP
$0.56 \times 1{,}600 = 896$ gpm
From NPSH margin chart:

$$\frac{NPSH_A}{NPSH_R\ *} = \frac{31}{22} = 1.41$$

(* at BEP)

$K = 0.735$

MCSF Corrected $= 0.735 \times 896 = 659$ gpm

Rounding off, we would specify 660 gpm minimum continuous stable flow.

There are five points to remember:

1. N_{SSS} is always defined at the pump best efficiency point at maximum design impeller diameter.
2. $N_{SSS} = (rpm)(gpm)^{0.5}/(NPSH_R)^{0.75}$
3. gpm is *per eye;* for double suction pumps use one-half the total pump flow at BEP.
4. When determining NPSH margin, use $NPSH_R$ at the *BEP FLOW.*
5. The determination of minimum continuous stable flow by using Figures 3-40 through 3-42 is approximate and occasionally tends to be conservative, i.e., lower flows are sometimes acceptable.

However, while it may be possible for some pumps to operate at flows away from BEP without experiencing hydraulic problems, it is important to recognize that unacceptably high radial forces could act on an impeller. Figure 3-45 compares the effect of three casing designs on radial force.[37] Unless a given pump is designed for sustained low flow operation, excessive shaft deflection, high vibration, and problems with bearings and seals may result.

These observations strongly suggest that the life expectancy of centrifugal-pump components is influenced by tradeoffs or interaction of pump suction-specific speed (N_{SSS}), throughput percentage referring to flow at BEP (Q actual/Q @ BEP), NPSH margin ($NPSH_A/NPSH_R$ or $NPSH_A$ minus $NPSH_R$), head rise per stage, and casing design. A closer examination of References 33 through 37 will support this writer's

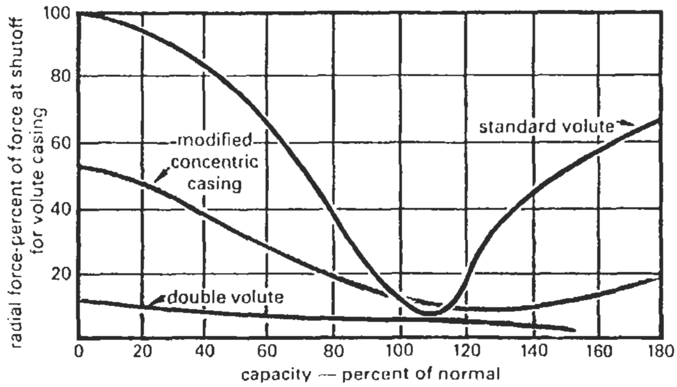


Figure 3-45. Comparison of the effect of three pump casing designs on radial force. *(Courtesy Pump World, Vol. 2, No. 1, 1976.)*

contention that competent correlation research on these parameters is highly desirable. Consideration of an additional parameter—shaft deflection—is intuitively evident to the mechanical engineer who visualizes overhung impeller construction of centrifugal pumps as essentially a cantilevered beam with the concentrated load (i.e., unbalanced fluid force) P acting at the free end and producing a maximum deflection of $(PL^3)/(3EI)$. With Young's modulus E a constant, shaft deflection can be reduced by selecting pumps with a low ratio of L^3/I, where L = overhung distance impeller to nearest bearing and I = shaft area moment of inertia, a function of the shaft diameter D raised to the fourth power. All of this can be translated to read: For a given casing design, overhung impeller centrifugal pumps with low L^3/D^4 values will undergo less shaft deflection than pumps with high L^3/D^4 values.

Choosing ANSI, API, Back-Pullout, or Between-Bearing Pumps. To the best of our knowledge, there are no firm rules or regulations defining when it would be either appropriate or necessary to use a particular pump geometry or style. However, on the issue of ANSI vs. API pumps, we can at least offer rules-of-thumb that are shared by many HP reliability professionals.

For toxic, flammable, or explosion-prone services, especially at onsite locations in close proximity to furnaces and boilers, a large number of reliability-minded machinery engineers would use API-610 8th Edition pumps if one or more of the following were to exist:

1. Head exceeds 350 ft (106.6m)
2. Temperature of pumpage exceeds 300°F (149°C) on pumps with discharge flange sizes larger than 4 in., or 350°F (177°C) on pumps with 4-in. discharge flange size or less
3. Driver horsepower exceeds 100 hp (74 kW)
4. Suction pressure in excess of 75 psig (516 kpa)
5. Rated flow exceeds flow at best efficiency point (BEP)
6. Pump speed in excess of 3,600 rpm.

We have seen exceptions made when deviations from the rule-of-thumb were judged minor, or in situations where the pump manufacturer was able to demonstrate considerable experience with ANSI pumps under the same, or even more adverse conditions.

Cost Justification. If your company is interested in seeing the cost justification for purchasing the generally stronger API pumps instead of normally satisfactory ANSI pumps, consider statistical approximations.

Suppose that under average conditions of maintenance effectiveness and installation care (foundation, baseplate stiffness, grouting, piping configuration, etc.), ASNI pumps in hydrocarbon services required service every 18 months vs. three years for API-610 8th Edition pumps. Assume further that each maintenance event requires the rather conservative expenditure of \$6,000, including burden and overhead costs. Knowing that you are likely to have a \$600,000 fire for every 1,000 pmp failures means that each failure event will incur a \$600 cost adder. All of this translates to:

ANSI	Pump repair cost, per year	\$4,000
	Fire event adder	400
	Probable cost per year:	**\$4,000**
API	Pump repair cost, per year	\$2,000
	Fire event adder	200
	Probable cost per year:	**\$2,200**

This would yield a yearly maintenance cost advantage of \$2,200 in favor of API pumps under the stated average maintenance effectiveness conditions. Add to it production loss credits, and the number increases. Also, how many service interventions per year equal one maintenance worker, or one reliability/technical support person, or one planner/supervisor/purchasing specialist? Or just how much would it be worth to your plant if your reliability engineers could spend more of their time on predictive maintenance and failure avoidance rather than having to struggle with equipment failures? Certainly a subject worth pondering.

Between-Bearing Pumps. There is some evidence that overhung impeller construction is occasionally more prone to maintenance or downtime than impeller-between-bearing construction. In view of this, some contractors and pump purchasers have, in the past, applied rather arbitrary selection guidelines.

One such guideline limits impeller diameters for overhung pumps to 15 in. (381mm). Another guideline calls for in-between-bearing geometries whenever the product of horsepower and rpm (rotational speed) exceeds 900,000. For example, at 3,600 rpm, motor ratings in excess of 250 hp would favor selecting between-bearing pumps.

How rigidly should these rules be applied, if at all? We would consider them appropriate for screening purposes. Allow deviations if the pump is of double-volute design and if calculated shaft deflections do not exceed 0.002 in. over the entire operating flow range, from zero to full BEP flow. Alternatively, allow deviations if the pump manufacturer can demonstrate long-term, satisfactory experience under identical (or more severe) operating conditions.

Significant Differences In Bearings and Bearing Housings

The eighth edition of API 610 continues to require axially preloaded 40° angular contact thrust bearings. This requirement stems from three observations that pump users have made over the decades: Thrust bearings fail relatively often; 40° angular contact bearings have higher allowable thrust load ratings than bearings with 15° or 29° contact angles; and many failed bearings exhibit ball skid marks in the race areas.

Axial preloading greatly reduces the risk of incurring bearing distress due to skidding of rolling elements. Details on this failure mode can be found in many publications from major bearing manufacturers. A typical analogy to skidding can be seen in aircraft landings. Upon initial touchdown, the wheels will skid until their peripheral speed has caught up with the forward speed of the plane. Just as skidding would cause accelerated wear of tires, it would result in potentially severe metal-to-metal contact in a rolling element bearing.

Axial preloading can ensure that the bearing will always be loaded. With pairs of angular contact bearings, axial preloading may be necessary.

Preloading or flush-grinding of thrust bearing sets will also prevent axial oscillatory movement of pump rotors. This motion is quite prevalent in pumps that experience cavitation or low-flow induced internal recirculation. The resulting instantaneous acceleration forces must be absorbed by the rolling element bearings. Again, bearing defects are much more likely to develop with bearings that operate with axial looseness than with preloaded or flush-ground bearings that operate without looseness in the axial direction.

In most applications, properly installed and lubricated axial preload bearings have extended the mean time between pump repairs. This is why many pump manufacturers, and especially the overseas manufactures of centrifugal pumps, do not take issue with the API requirement. However, there are instances where preloaded 40° angular contact bearings have been unable to solve problems or have made a problem worse. While we might have assumed that bearings with larger contact angles create more frictional heat, research by the FAG Bearing Corporation demonstrated 40° angular contact bearings generate *less* heat than thrust bearings with less angularity.[38] Nevertheless, by its very nature, preloading adds to the heat load, and using an interference fit between shaft and inner ring compounds the problem.

Most ball bearing manufacturers consider bearings to be preloaded if an offset, or predetermined gap, exists between the inner ring and outer ring faces while a light nominal load, often called a gauge load, is applied to the inner and outer ring thrust faces (see Figure 3-46). When axially clamped in a back-to-back or face-to-face fashion with a second equivalent bearing, the balls and races are forced to deflect, thus creating an internal load or preload. Preload can also be created by interference fits between the shaft and bearing bore, or the housing and bearing outside diameter, and by temperature differentials between inner and outer rings—most often due to the inner ring running warmer than the outer ring. Interference fits and temperature differentials decrease internal clearance; this will either create or cause an increase in preload. It is important to realize that shaft interference fits for back-to-back mounted angular contact bearings must be near the minimum of the range normally

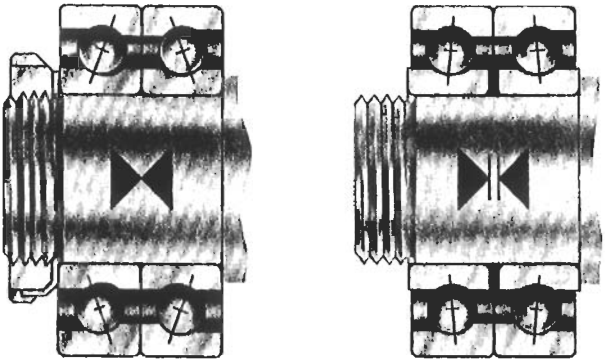


Figure 3-46. Application of gage load to axially pre-loaded bearings will show gap between inner races. (*Courtesy MRC Bearings.*)

allowed for radial or Conrad bearings. An interference fit near the typical maximum allowable is almost certain to result in greatly reduced life of these bearing sets.

In this regard, Figure 3-47 will prove very enlightening. It shows that for a given bearing (FAG 7314, 70 mm bore diameter) a shaft interference fit of 0.0003 in. will

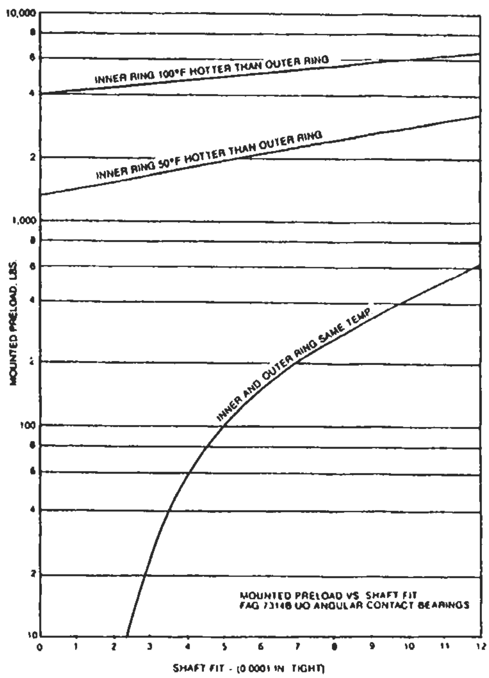


Figure 3-47. Mounted preload versus shaft fit for a specific 70-mm bore diameter ball bearing. (*Courtesy FAG Bearing Corp.*)

produce an almost insignificant preload of approximately 22 lbs, whereas an interference fit of 0.0007 in. would result in a mounted preload of 200 lbs. A much more significant preload would result from temperature differences between inner and outer bearing rings. Such differences could exist in pumps if heat migrated from high temperature pumpage along the shaft, or if the pump design incorporated cooling provisions which might artificially cool the outer ring and would thus prevent it from expanding. By far the worst scenario would be for a pump operator to apply a stream of cooling water to the bearing housing from a firehose. It is very discouraging to see this done even today, but old habits are difficult to break, regardless how counterproductive they may be.

Several ball bearing manufacturers now recommend "flush ground" bearings or bearings with different contact angles (e.g., 40° on the active and 15° on the inactive side) for back-to-back installation in centrifugal pumps. During operation, these bearings either become preloaded or remain preloaded. This prevents (or minimizes) ball skidding problems discussed on page 172. Moreover, flush-ground bearings will operate at a lower temperature than the initially preloaded pair. Lower temperatures can improve lube oil oxidation stability and life. Lowering the total load (i.e., thrust load plus radial load plus preload) would allow us to anticipate greater oil film thickness and less metal-to-metal contact.

Specification Amendments Covering Pump Bearings. The "best-of-competition" plants in the processing industry use definitive component specifications whenever reliability engineering concepts are actively pursued. They somehow realize that buying on price alone is rather naive and that a better product will have to command a better price.

A Dutch refinery with 11,000 pumps amends API 610 by asking that:

- All rolling elements should have metal rolling element retainers.
- Parallel roller bearings are preferred as pure radial bearings.
- Roller bearings should have the roller retaining rim on the inner race.
- Shielded or sealed bearings should not be used except for vertical in-line pumps up to 22 kW.

A major refinery in Texas tacks a machinery component parts specification to both inquiry and purchase documents (Table 3-13).

When the API 610 Seventh Edition became available in February of 1989, we advised a large-scale user of centrifugal pump bearings to consider replacing paragraph 2.9.1.5 with the replacement wording shown in Table 3-14.

Take your pick, or combine elements of the various specification amendments into a document that suits your particular needs. Seek out the world's leading manufacturers of rolling element bearings and ask them if they have a business team, or application engineering group dedicated to fluid machinery. Get their advice, request their relevant literature—you may be pleasantly surprised to see the value of sound, experience-based recommendations. But don't expect to get better bearings, or any other precision components, without making an effort to assemble a pertinent specification document.

Table 3-13
Machinery Component Specification

Radial and Angular Contact Ball Bearings

1.0 Scope

1.1 This specification covers the mandatory requirements for radial and angular contact ball bearings used in general purpose process machinery.

1.2 Bearings shall comply with the specification requirements of the AFBMA for ball bearings. Bearing dimensional tolerances shall meet or surpass the tolerances defined by ABEC Class 1.

1.3 All bearings supplied per this specification shall be obtained directly from the identified bearing manufacturer or authorized distributor.

1.4 The bearings shall be packaged per AFBMA Standards, Section 6. The bearing box shall be marked to identify the original bearing manufacturer and the alphanumeric bearing identification code or designation system.

2.0 Ball bearing design

2.1 Radial and angular ball bearings shall have the following design and features:

Bearing type	ABEC clearances	Ball retainer	Other features	Acceptable manufacturers
Single row, deep groove ball bearings	C-3	Riveted steel	Conrad type, no filling slot allowed	SKF/MRC, FAG Torrington/ Fafnir, NSK, NTN, KOYO
Double row, deep groove ball bearing	C-3	Stamped steel	No filling slot allowed	SKF/MRC, NTN, Torrington/ Fafnir, KOYO
40° angular contact ball bearing	Standard	Machined bronze	Land riding, if available; duplex mountable, *nil* prel.	SKF/MRC,FAG, Torrington/ Fafnir, KOYO, Rollway

2.2 Bearing shields, seals, snap rings, etc., shall have the configuration specified by purchaser's applicable spare parts symbol number or specified description for the stipulated application.

2.3 No substitutions are allowed for the bearing manufacturers or the bearing features shown without the approval of the purchaser's rotating equipment reliability engineer.

Regreasable vs. Non-Regreasable Rolling Element Bearings

At the inception of a project, the specifying engineeer will often be confronted with the question of when sealed, non-regreasable bearings present advantages over the traditional, regreasable variety. It is intuitively evident that fully sealed rolling element bearings would be the right choice for a wide range of appliances, such as a vacuum cleaner or household cooling fan. Conversely, experienced engineers have known for years that regreasable bearings are the correct choice for larger size or higher speed bearings. There was, however, no clear-cut guideline as to when to request one or the other, or which of the two modes—regreasable or non-regreasable—would make more economic sense when the labor cost of regreasing and the cost of rectifying greasing-related errors were considered.

Some time ago, a large international bearing manufacturer provided a rule of thumb that can be used under typical circumstances. The rule requires calculating the

Table 3-14
Replacement Wording Proposed for API-610 7th Edition, Paragraph 2.9.1.5

Specification Amendment for Heavy-Duty Centrifugal Pumps

	(This S.A. is appropriate for attachment to inquiry or purchase document.)
2.9	Bearings and bearing housings
2.9.1	Bearings for horizontal pumps
2.9.1.5	(Exception)
	a. If ball-type thrust bearings are used, the vendor shall verify the adequacy of duplex, single-row, 40° (0.7 radian) angular-contact type (7000 series) bearings, installed back-to-back (DB). This verification shall include selection of proper preload values to prevent skidding of the unloaded bearing over the entire operating range of the pump.
	b. If a., above, points to inadequacies or potential problems, the vendor shall offer suitably preloaded sets of ball-type thrust bearings with dissimilar contact angles (e.g., 40°/15°).
	c. Sets of properly preloaded, back-to-back mounted 15° or 29° angular contact bearings are acceptable for pumps with double-flow impellers located between two bearing housings.
	d. In each case, the need for and extent of preload shall be determined by the vendor to suit the application and meet the bearing life requirements of 2.9.1.1.

DN value of the bearing, with *D* being the bearing bore (in mm), and *N* representing the shaft speed (in rpm). For *DN* values below 80,000, we agreed to a preference for sealed, lifetime lubricated bearings. The range from 80,000 to 108,000 we labeled the gray area, and from 108,000 to 300,000 we defined a clear advantage for regreasable bearings. (It has been a long-standing practice for bearing manufacturers to recommend oil lubrication for bearings with *DN* values greater than 300,000.)

With *DN* values of 80,000 obtained, for example, by a 40 mm bore diameter radial ball bearing operating at 2,000 rpm, Figure 3-48 recommends regreasing intervals of approximately 9,000 hours—about one year. If such a bearing is provided with seals and is therefore lifetime lubricated, we anticipate it to become unserviceable after roughly two and a half times the stipulated regreasing period. These experience figures have been validated by some users and are reasonably correct. For a radial ball bearing with a *DN* of 200,000, say 100 mm and 2,000 rpm, the recommended regreasing interval shown in Figure 3-48 would be slightly less than 3,000 hours, or four months of continuous operation. Using the two and a half times rule, we might expect frequent failures after about ten months of operation—clearly not acceptable for most process plants. Thus, pay attention to *DN* values when deciding how and when grease lubrication is applied or reapplied.

Marginal Lubrication: A Factor in Pump Failures

As indicated earlier, a pump manufacturer will occasionally supply pumps with inherent design vulnerabilities. In fact, we must assume that the designs of at least some pump manufacturers who give bearing temperature concerns as the reason for disliking lightly preloaded or high-contact angle bearings are really suffering from marginal lubrication. In one instance, we found a manufacturer supplying pumps with excessive bearing inner race interference fit. Had he chosen to furnish preloaded bearings as well, early bearing distress would have been virtually certain.

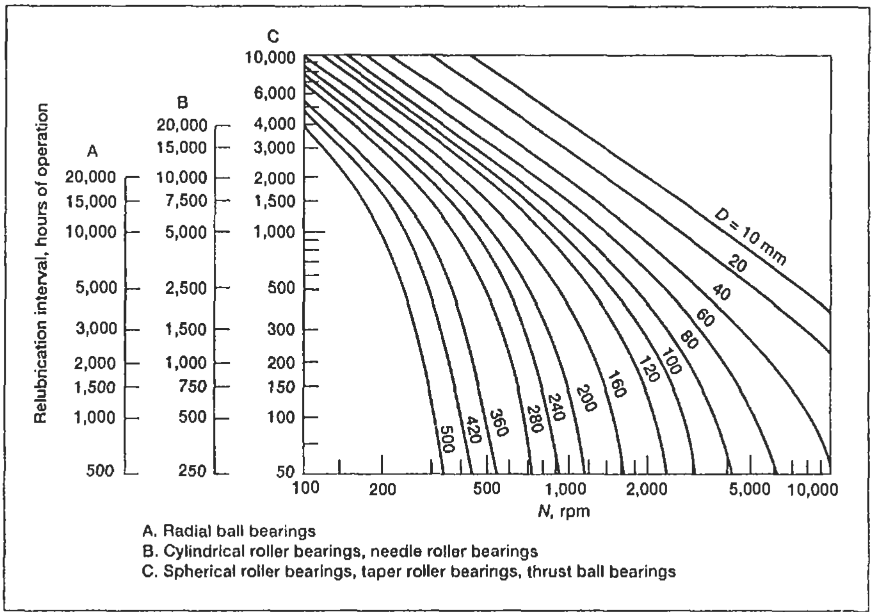


Figure 3-48. Relubrication interval. *(Courtesy SKF Industries, King of Prussia, PA.)*

A close comparison of the bearing housings of problem pumps with those of pumps with low failure frequency can be quite revealing. Low bearing failure rates are reported for the execution shown in Figure 3-49. These axially preloaded bearings are routinely used by a German manufacturer. As shown, they elected to achieve the proper preloading by selectively different dimensioning the width of spacers "A" and "B." The pump manufacturer can thus control the preload by making appropriate adjustments. Flinger disc "C" tosses lube oil onto the surrounding surfaces and from there it flows into trough "D" and on towards both inboard and outboard bearing locations.

The periphery of flinger "C" dips into the lube oil level; however, the lube oil level is generally maintained well below the center of the lowermost ball. This reduces oil churning and friction-induced heat-up of lube oil and bearings that would be more likely to occur in the design shown in Figure 3-50.

There is a good reason to introduce the lube oil between the two back-to-back oriented angular contact bearings. In this design, the cage inclination promotes through-flow of lubricant, whereas many other designs attempt to introduce the lube oil at points that oppose through-flows. Figure 3-51a allows us to see how the cage inclination of back-to-back mounted angular contact bearings with steeper angles promotes a centrifugal outward-oriented flinging action from side "a" to side "b." Conversely, if conventionally lubricated angular contact ball bearings are back-to-back mounted as shown in Figure 3-51b, lubricant flow may become marginal or insufficient. Subject to proper selection and utilization of proper installation procedures,

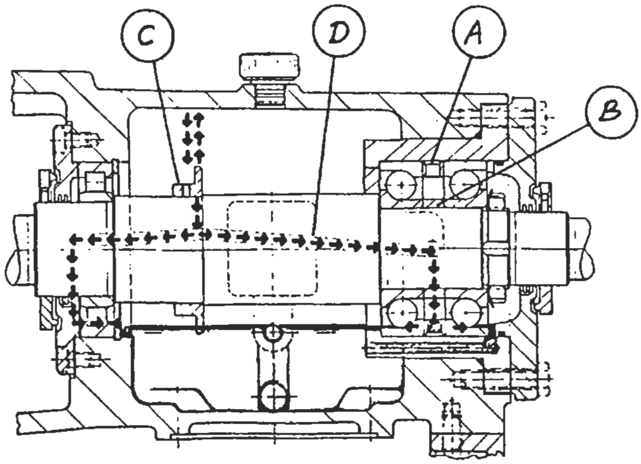


Figure 3-49. Successful pump bearing housing providing optimum lubrication. (*Courtesy KSB Pump Co.*)

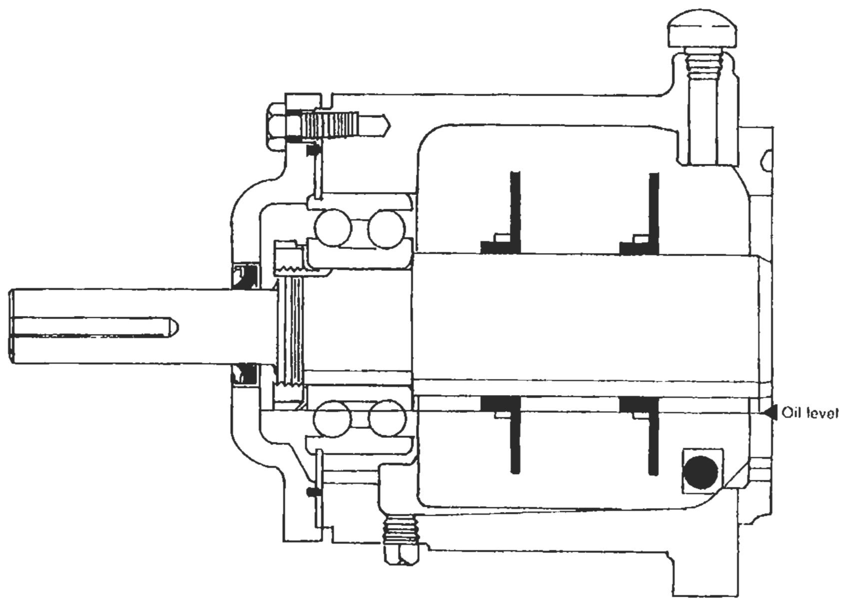
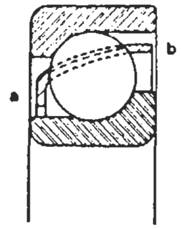
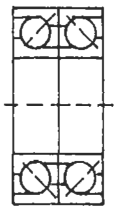
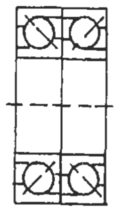


Figure 3-50. Lube oil covering major portion of lowermost ball promotes generation of frictional heat.



a): Skew of ball separator promotes lubricant flow from side "a" to side "b"





b): Back-to-Back Mounted Angular Contact Bearings

c): Face-to-Face Mounted Angular Contact Bearings

Figure 3-51. Angular contact bearings used for thrust takeup in centrifugal pumps. (*Courtesy SKF Bearing Co.*)

face-to-face mounting (Figure 3-51c) may be advantageous in those instances where lubricant flow needs to be improved. It clearly promotes through-flow of lube oil and is one of the reasons why the manufacturer of the heavy duty pump shown in Figure 3-52 opted for face-to-face orientation. Oil spray generated by the flinger disc will flow in the preferred direction through the adjacent bearing and on to the next one. Note, however, that this presupposes that the temperature difference between inner and outer races is minimal. If the temperature were substantial, growth of the inner ring would force the bearing into a condition of high axial preload.

Two additional illustrations, Figure 3-53 and 3-54, show oil rings inserted in trapezoidal ramps, which allow lube oil to move towards the bearing internals. It should be noted that both of these illustrations violate API-610 8th Edition (August 1995). Paragraph 2.9.1.3, which requires bearings to be directly mounted on the shaft and disallows bearing carriers. Also, snap rings and spring-type washers are not permitted. Figures 3-55 and 3-57 show executions that facilitate lube oil flow toward the bearings.

However, oil rings as shown in Figures 3-53, 3-54, and 3-57 to 3-59 have a tendency to malfunction unless the equipment centerline is aligned and installed truly parallel and horizontal. This true alignment is difficult to achieve, since the machinist or millwright will usually place shims under either the outboard or inboard legs of the equipment while performing driver-to-driven alignment in the field. It is for this reason that flinger disks represent the preferred configuration.

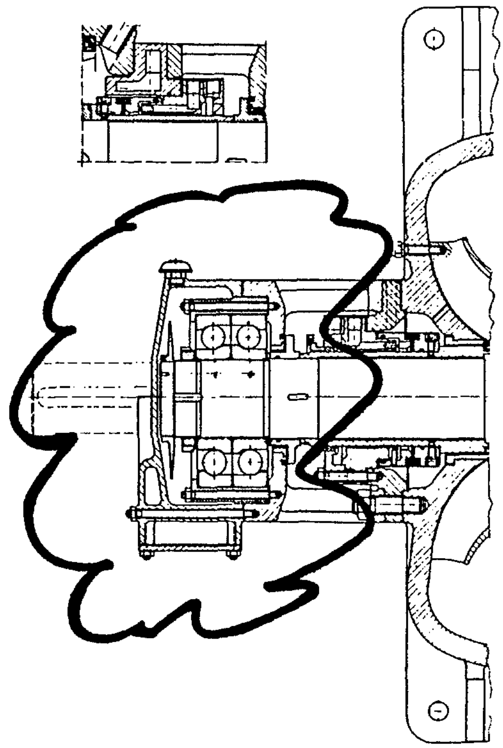


Figure 3-52. Oil spray generated by the flinger disc will easily flow through this face-to-face oriented bearing set. (*Courtesy Ochsner Kreiselpumpen Co. Linz, Austria.*)

Figure 3-56 shows how the oil delivery capability of oil rings varies with shaft speed. Grooved rings are demonstrably superior to the traditional flat rings and should be applied whenever the less vulnerable flinger disk, shown in Figures 3-50 and 3-55, is not available.

The Goulds Pump Company has experimented with different executions and found that keeping the oil level below the rotating elements results in lower bearing temperatures than would be achieved if lube oil were to reach to the center of the lowermost ball. Goulds explained that their optimum design, Figure 3-59, evolved from the one shown in Figure 3-58, which incorporated a cupped oil flinger design. The throw-off action of the cupped disks did not differ significantly from that obtainable with plain disks. Moreover, using two separate disks occasionally resulted in incorrect installation by inexperienced repair shops. The manufacturer corrected this situation by using the single spool or spacer piece in Figure 3-59 for oil ring containment.

Goulds established a satisfactory working window for the design using ISO Grade 68 lube oils and shaft diameters in the 2.5-in. range. Adequate oil flow existed with

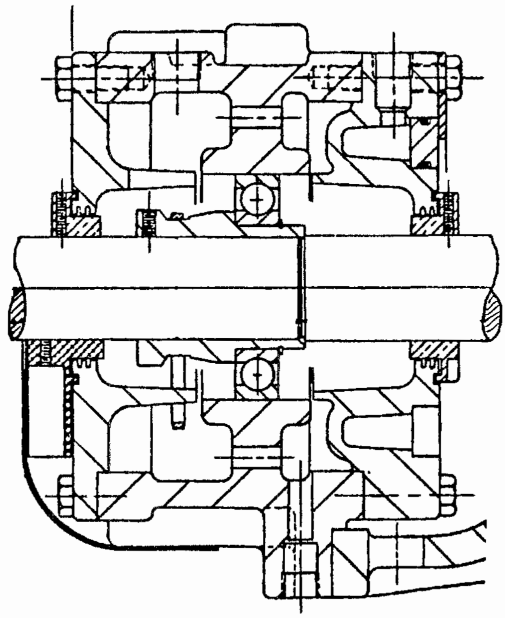


Figure 3-53. Oil rings inserted in trapezoidal ramps allow lube oil to move freely toward the bearing internals. (*Courtesy Byron Jackson Pump Co.*)

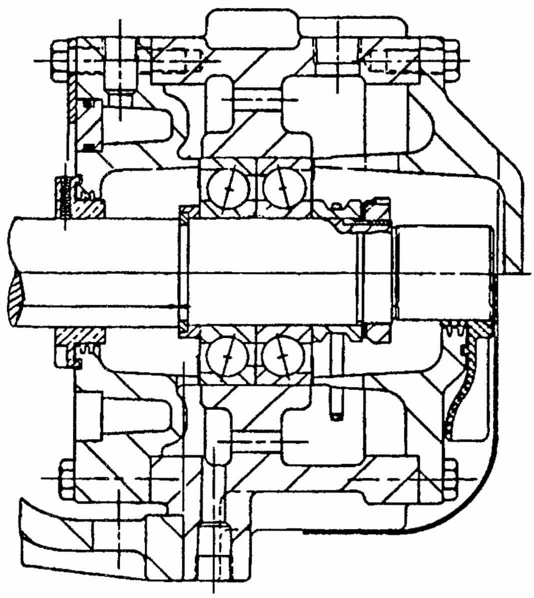


Figure 3-54. Oil rings guided in trapezoidal slots facilitate lubricant migration toward bearing. (*Courtesy Byron Jackson Pump Co.*)

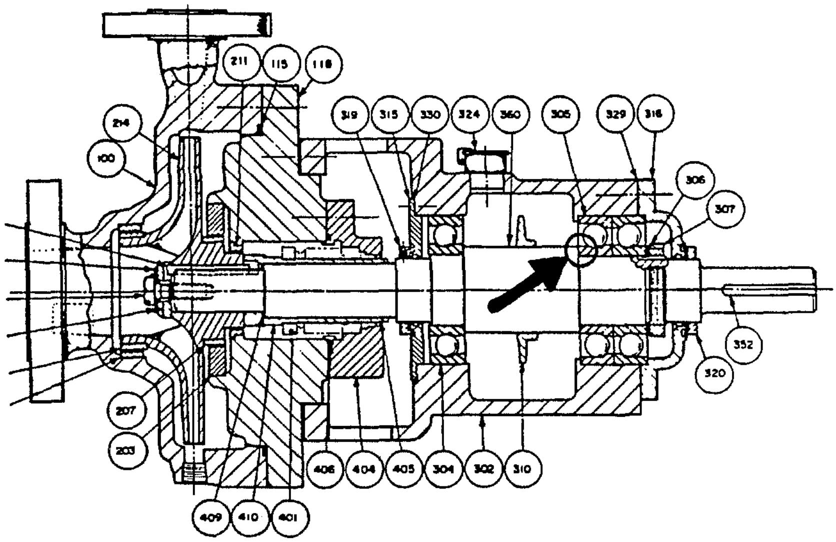


Figure 3-55. Oil droplets flung against the top of a bearing housing have a chance of falling back on the shaft and migrating toward the bearing internals.

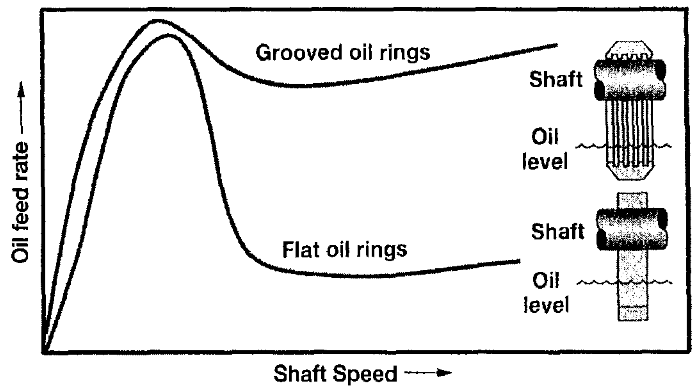


Figure 3-56. Oil delivery vs. shaft speed for different oil rings.

oil temperatures purposely induced to range from 65°F to 220°F at speeds from 880 rpm to 3,550 rpm.

This company also experimented with redesigned power ends for their line of pumps and found that a three-fold increase in oil sump capacity resulted in significantly reduced oil temperatures. The value of the attendant increase in bearing life

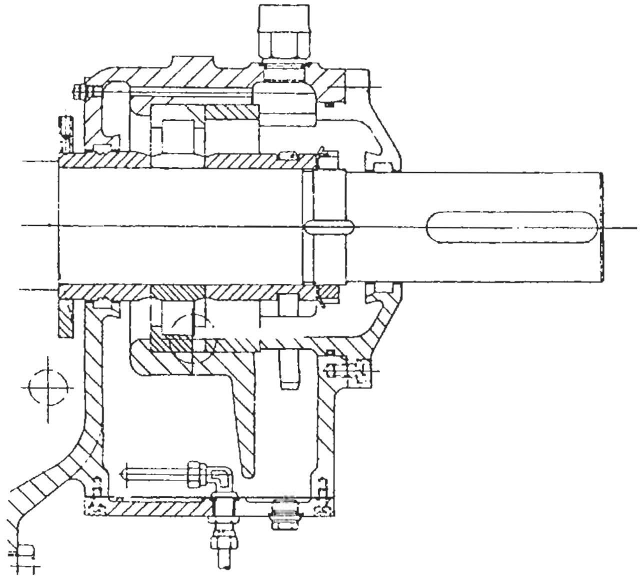


Figure 3-57. Oil ring and shaft sleeve outer contours are in line with bearing inner ring surface. This facilitates oil flow toward bearing interior.

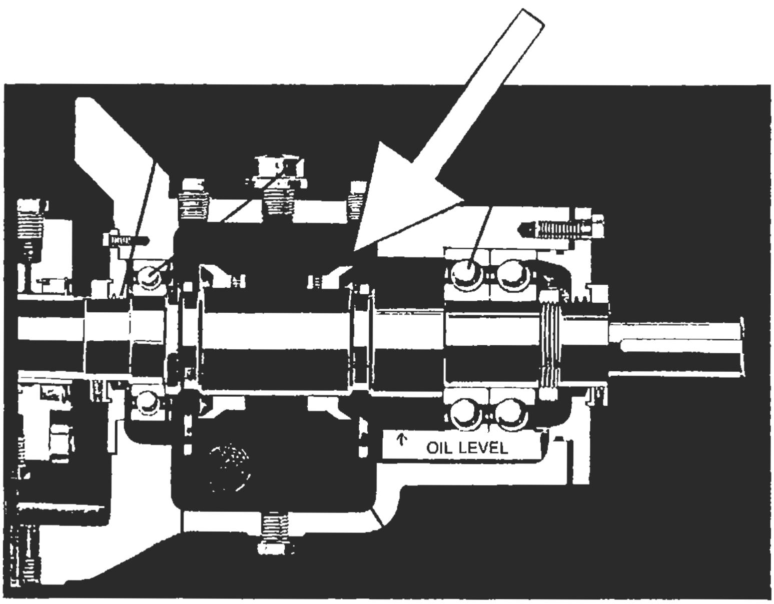


Figure 3-58. Cupped flinger design found in some older pumps. (*Courtesy Goulds Pumps.*)

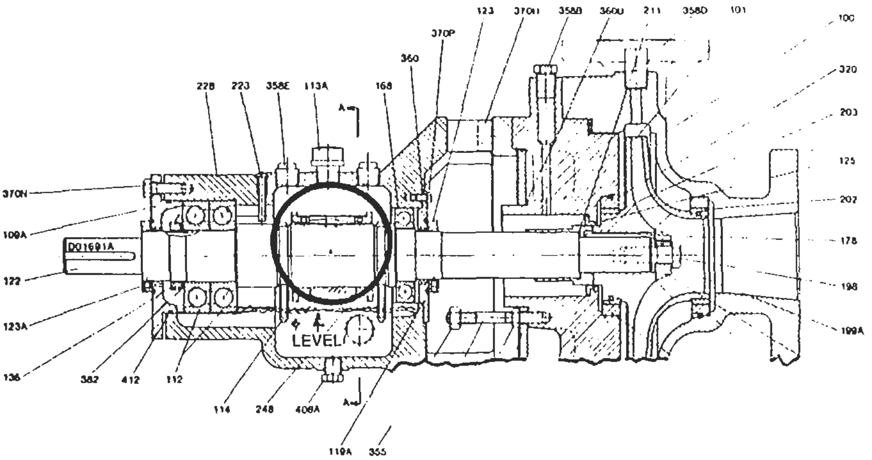


Figure 3-59. Optimum design for Goulds pumps. A single flinger spool serves to retain two oil rings. Oil pumped up by the oil rings contacts the flinger and is sprayed into the housing walls.

will be discussed in Chapter 5, Life-Cycle Costing. Figure 3-60 gives a comparison between bearing submergence in oil, operating temperature, and bearing life for the old and new designs.

Applying Roller Bearings in Centrifugal Pumps

Figure 3-61 shows a cylindrical roller bearing on the inboard side of the bearing housing. A tapered roller bearing is used on the coupling side of a fan-equipped pump, Figure 3-62.

It is certainly possible to extend bearing life by using cylindrical and/or tapered roller bearings in many pump models. Figures 3-63 and 3-64 illustrate this for bearings that can be used in the same bearing bracket. Note that this manufacturer uses the oil ring (Figure 3-61) only to maintain a uniform oil temperature throughout the entire sump. The oil ring is not expected to pump oil into the bearings, although it would probably do so in the stepless shaft-to-bearing-inner-ring geometry shown in Figure 3-62. Adequate lubrication is assured by maintaining oil levels to the center of the lowermost rolling element. At a speed of 1,750 rpm and an axial load of 12,000 Newtons, the angular contact bearing (Figure 3-61) will have a projected L-10 life of 18,500 hours versus 90,000 hours for the tapered roller bearing shown in Figure 3-60. Many of the latter bearings are rated for long life, as indicated in the pump bearing life vs. axial thrust vs. speed graphs illustrated in Figures 3-63 and 3-64. Needless to say, these extended life figures are meaningless unless bearings are properly installed and unless clean, adequate lube oil flow is assured at all times. The overwhelming majority of bearings fail due to lube oil deficiencies, including contamination, or due to skidding (see page 172), caused by light loads.

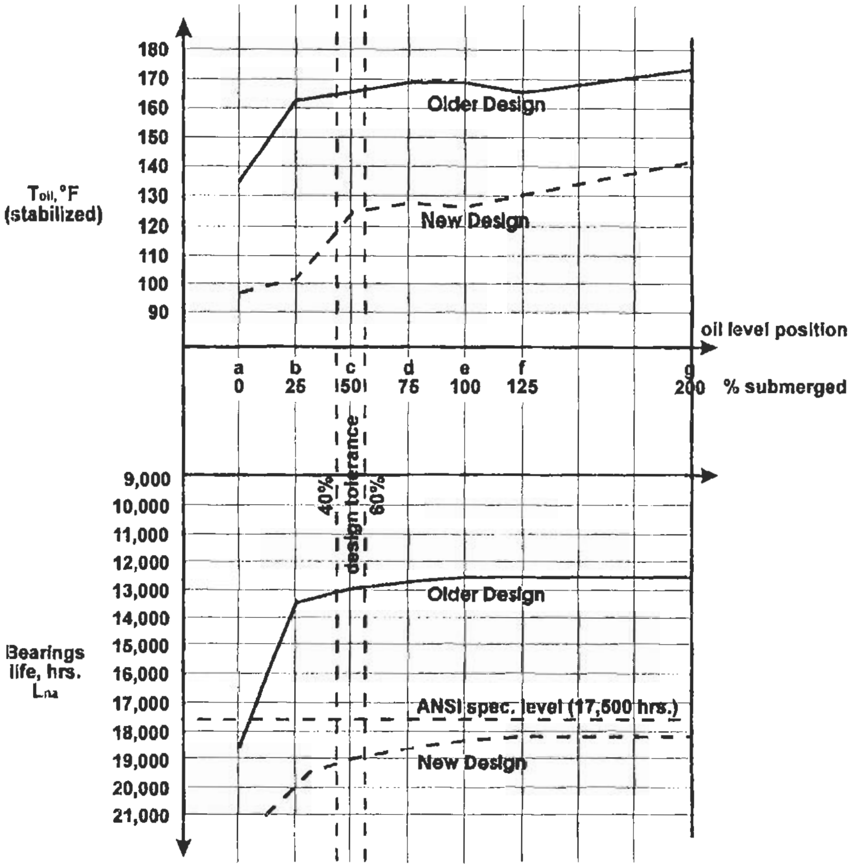


Figure 3-60. Comparison between bearing submergence in oil, operating temperature, and bearing life for the old and new designs. (*Courtesy of Goulds Pump, Seneca Falls, New York.*)

However, while roller and tapered roller bearings may present a viable alternative in selected instances, they do have potential drawbacks when used in centrifugal pumps: First, proper alignment of the shaft is much more critical to ensure proper roller-to-race contact and minimize stresses. Spacing or shimming between housing end plate and housing must be quite accurate. This assembly method is used to ascertain proper roller-to-race contact (i.e., not too much end-play in the shaft) without preload. Tapered roller bearings also have much higher internal friction and thus will not run as fast and as cool as comparable ball bearings. Therefore, proper lubrication, i.e., proper type of lubricant, proper amount, and proper application method will be even more important with roller and tapered roller bearings than with ball bearings.

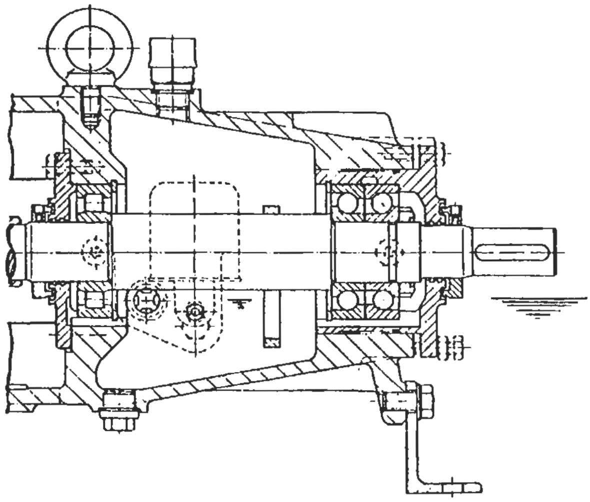


Figure 3-61. Roller bearing on inboard side of a centrifugal pump. (*Courtesy Sulzer-Weise, Bruchsal, Germany.*)

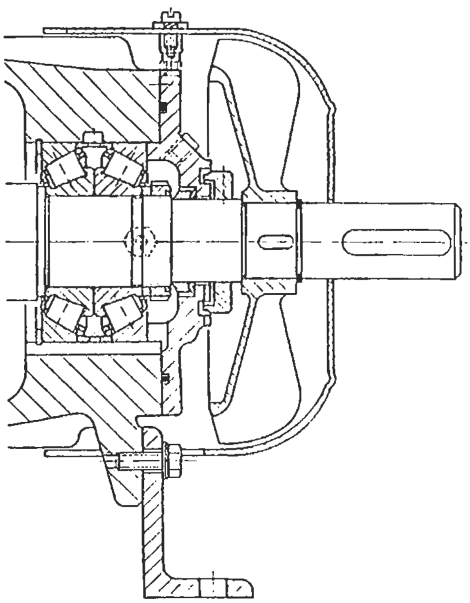


Figure 3-62. Fan and tapered roller bearing on thrust end. (*Courtesy Sulzer-Weise, Bruchsal, Germany.*)

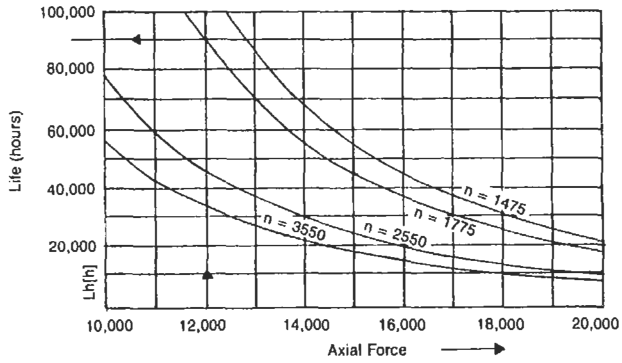


Figure 3-63. Tapered roller bearing life at various speeds plotted against axial force in a centrifugal pump.

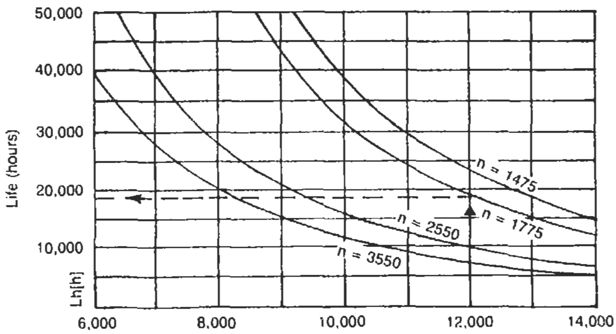


Figure 3-64. Ball bearing of comparable size. Life at various speeds plotted against axial force in a centrifugal pump. (*Courtesy Sulzer-Weise, Bruchsal, West Germany.*)

How Much Oil Is Enough?

The MRC Bearing Division of SKF USA calculates the theoretical oil flow required for cooling from the expression:

$$Q = \frac{.000673 \times Fr \times P \times PD \times rpm}{H_S \times (T_o - T_i)} \text{ lb / minute}$$

where Fr = coefficient of friction referred to PD

.00076 = cylindrical roller bearings

.00089 = pure thrust ball bearings

.00103 = radial ball bearings

.00152 = angular contact ball bearings

.002–005 = tapered roller bearings

P = imposed equivalent load, lb
PD = pitch diameter, in.
rpm = operating speed
T_o = outlet oil temperature, °F
T_i = inlet oil temperature, °F ($T_o - T_i$, generally about 50°F)
H_s = specific heat of oil in Btu/lb/°F (usually .46–.48)
= .195 + .000478 (460 + T_i)

Conversion: (lbs of oil/min) × (0.135) = gal/min

From this equation it would seem that larger quantities of oil are needed than could reasonably be expected from either oil ring or flinger methods. This discrepancy takes on several orders of magnitude if we realize that with oil-mist lubrication three rows of bearings with a 3-in. bore diameter use 3 grams of lube oil per hour and can operate for years.

Observing the bearing manufacturer's coefficient of friction numbers we note that the value for angular contact bearings exceeds the one for radial ball bearings by a factor of 1.5. It stands to reason that angular contact thrust will either require more cooling oil than pure radial bearings, or will run warmer than pure radial bearings. Also, getting the right amount of oil to an angular contact thrust bearing will both be more critical and more difficult than getting the right amount to a typical radial bearing.

Bearing Selection Can Make a Difference

API-610, 8th Edition, requires ball-type thrust bearings, if used, to be dual single row, 40° (0.7 radian), light preload, angular contact type (7000 series), installed back-to-back. A lot of controversy revolves around this specification clause, with some bearing manufacturers expressing concern that the term "light preload" does not adequately quantify the desirable preload. Also, users occasionally report more failures with preloaded bearings than with conventional bearings. However, neither observation has presented a dilemma to this writer. Here is why.

The overall intent of providing preload is to prevent axial shuttling of the rotor and skidding of the rolling elements in a bearing. Skidding can be extremely detrimental to rolling element bearings and we have often observed the *unloaded* half of a duplex-mounted bearing generate more heat and fail before the loaded half showed any distress. However, using a preloaded bearing may mandate lowering the customary interference fit between shaft and bearing inner ring. Unless this is done, additional heat may not be carried away by certain oil ring and/or flinger arrangements often found in centrifugal pumps. Also, if marginal lubrication was provided to begin with, preloading may be "the straw that breaks the camel's back." It is in those instances that flush ground bearings may offer adequate operating preload to assure quiet operation, minimize ball skidding, and reduce the risk of thermal runaway.

The introduction of Pumpac® thrust bearings by the MRC Company has allowed many users to extend pump mean-time between failure (MTBF). MRC's thrust bearing system eliminates skidding by mounting a 40° angular contact bearing back-to-back with a 15° angular contact bearing.

Air Cooling Provisions for Bearing Housings—How Good?

As explained later in this text (see pages 434–440), it can be demonstrated that cooling water can be deleted from virtually all centrifugal pumps with rolling element bearings. It was shown that sizeable maintenance cost credits resulted from this deletion and that these cost credits could be attributed to several factors:

- Cooling water pipes did not have to be maintained.
- Utilities requirements and the attendent operating costs were reduced.
- Bearing failure incidents decreased substantially.

The last observation was most striking because it was perhaps least expected.[39,40] A decrease in bearing failure incidents can be explained by reduced water contamination and by lowered risk of incurring distortion of bearing races. Contamination originates with cold cooling water, which promotes condensation of water vapors contained in the oil/air mixture inside the bearing housing. Distortion comes from non-uniform cooling through water jackets. These are sometimes partially surrounding the bearing outer race and can force the bearing to assume an out-of-round shape.

Several pump manufacturers have implemented air cooled bearing housings in efforts to provide a suitable temperature environment. They often execute the bearing housings with cast-in cooling fins and advertise that no fan is required. For high-temperature pumpage, typically 500°F and higher, a shaft-mounted fan is often used. We found some of these fans thoroughly engineered for low noise and high efficiency. Figure 3-62 depicts a typical example of an intelligently engineered fan. In contrast, the "fan" offered by another manufacturer (Figure 3-65, item 375) leaves much to be desired.

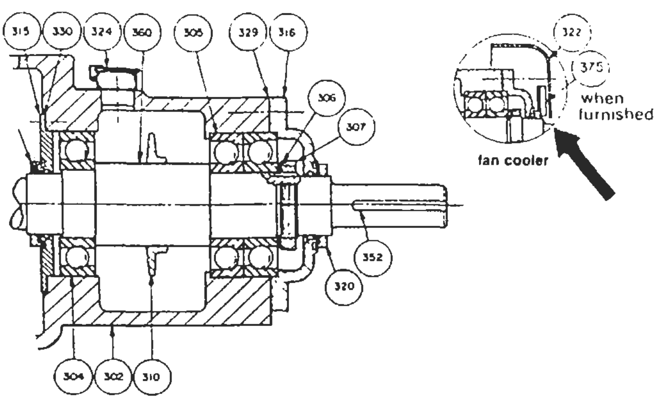


Figure 3-65. "Bearing cooling fan" on a centrifugal pump.

However, we are certainly not ready to give a blanket endorsement to all fan-cooled bearing housings. Air-cooled bearing housings can cause bearing distress by allowing uneven cooling of bearing rings. The inner ring is mounted on the shaft and the combined assembly represents a poor heat sink. In comparison, the bearing housing is a relatively effective heat sink, especially if cooled externally. The heat transfer rate is influenced by the properties of the housing material, by housing geometry and by the temperature difference between pumpage and external ambient conditions. Fan cooling is forced convection and thus increases heat transfer. If material properties and housing geometry are assumed constant, cooling the housing by any of several possible methods will increase the rate of heat transfer and thus cools the bearing *outer* rings. Very little heat is transferred through rotating elements from the inner ring. The inner ring, therefore, runs hotter than the outer ring.

The result, of course, is differential thermal growth, with the inner ring expanding more than the outer ring. If bearings are initially flush ground (i.e., no preload and no end-play) or ground for preload, cooling the housing creates or, respectively, increases radial and axial preload and negative clearance exists. The resulting temperature excursion may or may not be self-limiting. In any event, there would now be increased demands on the lubricant to effectively prevent metal-to-metal contact. (See also Figure 3-47).

Stuffing Box Cooling Is Not Usually Effective

Many pumping services require that the mechanical seal environment be kept at moderate temperatures. This is generally not difficult to achieve if external flush injection is used. In this case, a flush cooler can perform the task, but, of course, at some utility expense (fan power in air-cooled systems, cooling water in conventional heat exchanger circuits). The cost of recirculating the flush fluid may have to be added as well.

Another option for achieving a moderate seal environment is stuffing box cooling. In conventional pumps, Figure 3-66, the stuffing box cavity ("A") is rather remote from the seal faces ("B") that we wish to cool. An experiment conducted around 1970 showed a disappointing 1° to 2°F decrease at the seal faces when cooling water was introduced into a previously empty stuffing box jacket.

A superior design, from the point of view of effective cooling, is shown in Figure 3-67. The manufacturer recognized that heat migration from the casing is primarily responsible for elevated stuffing box temperatures. He, therefore, designed the pump with an air gap "A" ahead of the cooling water cavity "B." Equally important is the fact that the throat bushing "C" is made extremely long and that cavity "B" contacts the throat bushing over a good portion of this length. It should be intuitively evident, however, that this configuration will lead to directionally higher L/D ratios than competing designs. Accordingly, shaft deflections must be compensated by controlling the forces acting radially on pump impellers and by more careful design of component clearances.

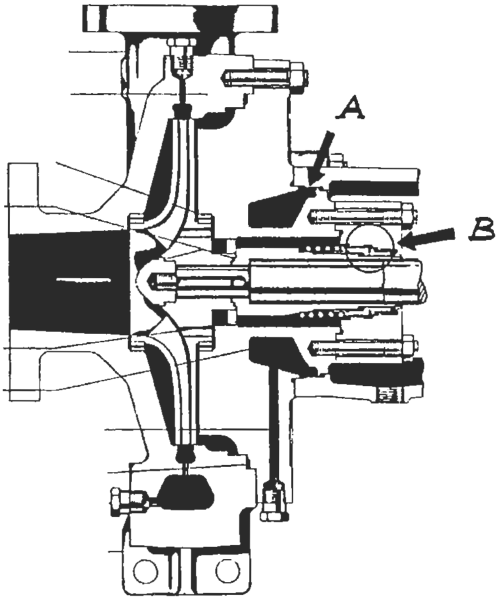


Figure 3-66. Conventional stuffing box cooling is quite ineffective for control of the seal environment.

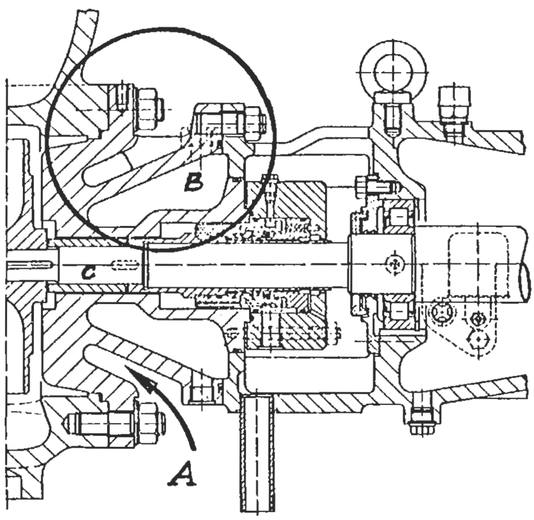


Figure 3-67. Effective control of mechanical seal environment is achieved in pumps with this cooling jacket execution. (*Courtesy Sulzer-Weise, Bruchsal, Germany.*)

Pumps for Handling Entrained Gases

Gas entrainment can occur on virtually any horizontal centrifugal pump. After a seal replacement, the entire seal cavity is filled with air. Unless special vent provisions are installed and venting procedures followed, liquid being admitted into the pump will fill all but the uppermost region of the seal cavity. Unless a small vent hole has been drilled near the 12 o'clock position indicated in Figure 3-68, air will remain trapped for often a considerable length of time. Since air is a very poor conductor of heat, the seal will now tend to warp, simply because the liquid-wetted parts are likely to experience a much cooler temperature profile.

Depending on pump construction features and configurational details, this drilled vent may be beneficial even in pumps specifically designed for inherently greater tolerance of gas entrainment. Such pumps would typically incorporate semi-open impellers, i.e., impellers without covers.

However, maintaining pumping efficiency requires close spacing between the open end of the impeller and the pump casing. This makes pumps with wear plates and good adjustment features more attractive to users with long-term reliability and maintainability goals. Excellent results are also obtained with certain inducer-equipped pumps (Figure 3-69), especially if the inducer geometry is custom-designed for a given application and can thus accommodate liquid and gas properties and conditions.

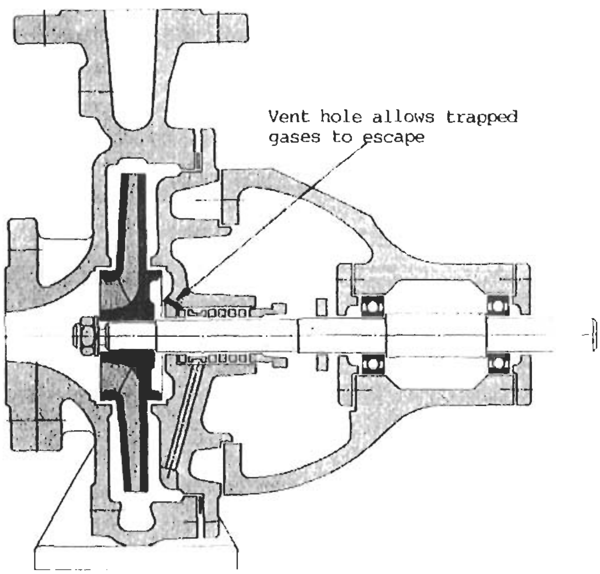


Figure 3-68. Vent hole allowing trapped gas to escape from seal cavity will protect against warpage of seal face.

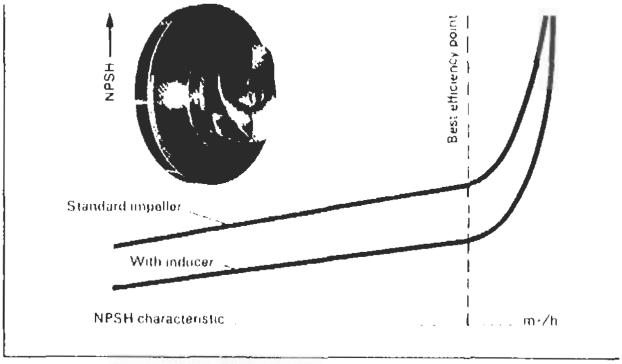


Figure 3-69. Properly designed inducers can lower NPSH requirements of pump impellers and increase the allowable gas entrained in the pumped liquid.

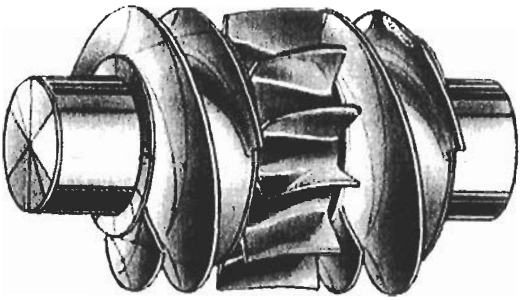


Figure 3-70. Hydraulics of a multiphase pump developed by Sulzer at the initiative of the Institut Francais du Petrole and the oil companies Total and Statoil, with the support of coaxially arranged stationary diffuser.

For extremely demanding applications, Sulzer "Poseidon" pumps may be of greatest value. A single pump is capable of delivering, for example, unprocessed crude oil containing oil, gas, water, and solid constituents to the processing plant. Such multiphase pumps handle mixtures of liquid and gas that may contain from 0% to 95% gas.

The multiphase pump is a turbomachine, similar to the injection pump used in the oil industry but differing mainly in its hydraulics. Each pump stage (so-called hydraulics) consists of an impeller mounted on a rotating shaft and a fixed diffuser (see Figure 3-70). The rotary part is an open impeller with helicoidal blades. The sections of the impeller blades and the arrangement of the diffuser blades are designed to minimize separation of the gas and liquid mixture during compression. In addition, the hydraulics are designed so that solids present (e.g. sand) remain in suspension and can pass through unhindered. Furthermore, precautions are taken to prevent deposition of solids in the pump casing.

Whereas the stages of normal multistage liquid pumps are all identical, those of multiphase pumps have different geometries because the gas flow volume diminishes in the course of compression. All these design innovations add up to an extremely wide pumping flexibility covering a wide operating range. Yet, if absolutely new operating parameters have to be accommodated (such as wells with gradually declining productivity), a new hydraulics block could then be fitted to anticipate the conditions expected.

Selection Criteria For Zero Emission Pumps

Zero emission pumps are making inroads in user plants worldwide. There is an understandable desire to use pumps that are designed to be completely sealed and vapor tight. But there is also considerable confusion due to the various claims and counterclaims from respective manufacturers of the two principal configurations of zero emission, or sealless pumps.

First, the two broad categories of zero emission pumps are magnet drive units (Figure 3-71) and canned motor pumps (Figure 3-72). Each achieves hermetic sealing of the pumpage through use of a containment shell. The magnet drive unit uses an external, or outer, rotating magnet ring, whereas the canned motor pump has the containment shell surrounded by the stator windings of an induction motor.

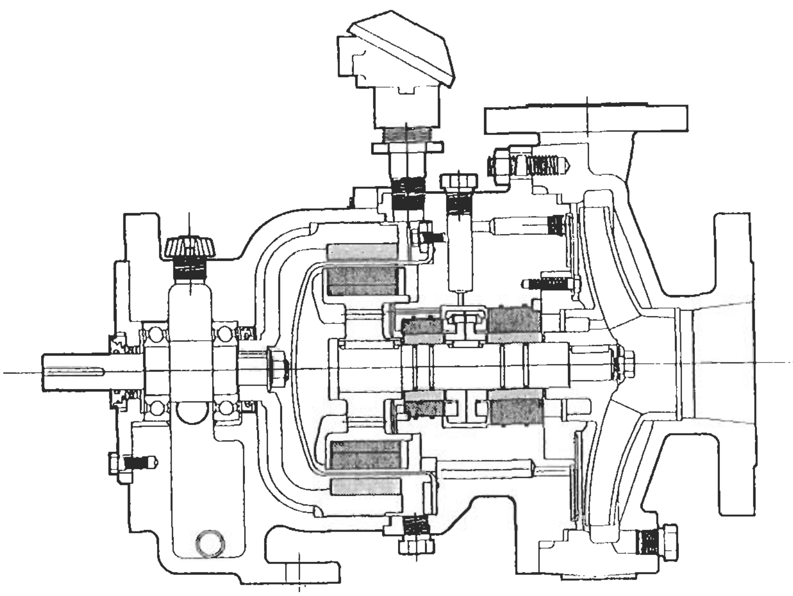


Figure 3-71. Magnetic drive pump. ***(Courtesy of Goulds Pumps, Seneca Falls, New York.)***

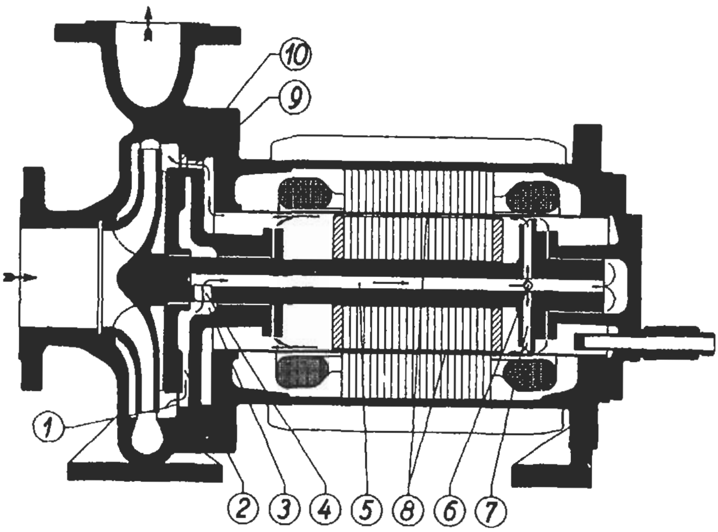


Figure 3-72. Canned motor pump. *(Courtesy of Hermetic-Pumpen GmbH; D-7803 Gundelfingen, Germany.)*

In the particular version of a canned motor pump depicted in Figure 3-72, a self-cleaning filter (1) is fitted in the pump section at the periphery of the impeller. A part of the partial flow taken from the main transport flow is forwarded through this filter (2), through passageways and a chamber (3) to perform motor cooling and slide bearing lubrication. This "partial flow" enters the hollow shaft (5) via transverse passages (4) and travels via radial holes (6) to an auxiliary impeller (7) mounted in the rotor chamber that transports it back to the pump pressure chamber via the clearance (8) between stator and rotor as well as through passages (9). The pressure level at the withdrawal point is almost equal to the transport pressure of the pump. The entire rotor chamber is thus superimposed with this pressure. Vaporization is impossible, as the static pressure is always higher than the vapor pressure of the heated cooling/lubrication flow. The auxiliary impeller (7) that generates the required pressure increase above the rotor is designed in such a way that it can take up the partial flow directly from passages in the shaft. This has the advantage of compact design; in other words, it can be accommodated in the rotor chamber. It also renders additional sealing gaps unnecessary, and this has a favorable effect on the operating characteristics of the motor.

Most zero emission pumps use product-lubricated bearings. Hence, if the product is highly abrasive or has low lubricity, it is prudent to consider zero emission pumps that use a separate bearing lube circuit. This separate loop can be filled with a pumpage-compatible fluid that can be cooled, if necessary. And, while this may add to both cost and complexity, it may be the only prudent configuration for a particular service.

A good zero emission pump should be designed to balance axial thrust. It should incorporate supervisory instrumentation and have a guaranteed efficiency. This efficiency should be referenced to the pumpage-to-motor input power ratio, not input power to the pump rotor, which might not accurately account for eddy current or other magnetic losses.

Look for experience. Some zero emission pump manufacturers are trying to make inroads on price alone. Be sure to buy pumps that come with protective instrumentation. Be aware that reliability and rock-bottom price are often mutually exclusive. Write a specification or acquire a specification, but by all means *invoke* a specification.

Pay particular attention to the type of containment shell proposed by a particular vendor (See Table 3-15 for a comparison of typical materials.) Verify vendor experience and suitability for fluid conditions.

Table 3-15
Comparison of Containment Shell Materials for Zero Emission Pumps

Material	Magnet Losses	Dimensional Stability	Cost	Chemical Resistance Acids	Chemical Resistance Solvents	Safety	Tensile* Strength	Abrasion Resistance
316SS	High	Excellent	5	Fair	Excellent	Good	82	Good
Hastelloy C276	Medium	Excellent	7	Excellent	Excellent	Excellent	111	Good
Peek	Low	Fair	6	Good	Poor	Fair	23	Poor
FiberGlass Re. Vinyl Est	Low	Good	3	Good	Poor	Fair	39	Poor
FiberGlass Re. Epoxy	Low	Good	3	Good	Fair	Fair	30	Poor
PVDF	Low	Fair	5	Good	Poor	Poor	8	Good
Ceramic	Low	Excellent	10	Excellent	Good	Good	3–10	Excellent

**ksi*
Courtesy of Goulds Pumps, Seneca Falls, N.Y.

Develop operating procedures, commissioning procedures, and repair procedures. And never forget: If you allow zero emission pumps to be abused the way process plants sometimes abuse conventional pumps, neither magnetic drive nor canned motor pumps will give long-term reliable service.

Study our attempt at tabulating the pros and cons of the two principal types of zero emission pumps by assigning a maximum of three points to positive attributes and one point to somewhat less advantageous features. (See Table 3-16.) Take issue if you wish, or assign your own numbers based on in-house experience, but study it nevertheless.

Finally, resist the inclination to simply total up the above numbers. Assign special value to vendor experience in a given size or application. Remember to summarize your requirements in a well laid out specification. Look at the vendors' response and make an experience check. Then choose wisely.

Table 3-16
Ranking of Features for Magnetic Drive and Canned Motor Pumps

	Mag Drive	Canned Motor
Safety in case of containment can penetration	2	3
Range of pumping temperatures	2	3
Range of feasible pressures	1	3
NPSH characteristics	3	1
Ability to cope with abrasive and low lubricity pumpage	1	2
Vulnerability to starting difficulties	1	2
Operating noise	1	3
External dimensions (3 = small, 2 = larger)	2	3
Ability to be arranged vertically	1	2
Ease of repair	3	1
Average cost of procurement	1	2
Foundation or support requirements	2	3
Mech. auxiliaries (extra bearings, coupling)	1	2

Design Appraisals for Special-Purpose Gearing

Speed-increaser gears are frequently required between electric motor drivers and centrifugal compressors, or on compressor strings with several casings. Efficient sizing and speed optimization would require the use of gearing on multi-body strings.

Most large gear units use helical tooth design. Most of the gears used by U.S.-based process plants are double helical. This allows for efficient operation with virtually no axial-thrust generation. It has the disadvantage of relatively high cost and requires complex manufacturing setups to ensure that the two mirror-image tooth profiles are dimensional duplicates of each other. Finish grinding is not easily accomplished on double-helical gears. Also, these gears are sensitive to coupling lockup if gear-type couplings are used.

Single-helical gears are primarily manufactured by European companies. They generate high axial thrust which must be taken up by appropriately sized thrust bearings. Single-helical gears are less expensive, somewhat less efficient, but frequently more precise than double-helical gears. As a final manufacturing step, precision grinding methods can be applied. Again, on the debit side, the larger face width of single-helical gears as opposed to equivalently rated double-helical gears would tend to produce approximately twice the tooth deflection and stress value. Finally, torsional vibration could cause potentially more serious axial vibration in single-helical gears than in double-helical gears.

Selection parameters and design criteria include a wide range of possible hardnesses for mating gears. High hardness values may be necessary to obtain a desired horsepower carrying capacity within the limitations of acceptable pitch-line velocity. The drawback would be a tendency toward scoring due to high load intensity and high sliding velocity. Softer gear teeth would comply better with certain misalignment or tooth inaccuracies. Of course, they would not have the same basic strength as harder gears.

Procurement of gears designed for acceptable long-term operation with medium-hard teeth is advantageous because future uprates may be possible by simply purchasing replacement gears with greater tooth hardness.

Design Appraisals Shortcuts

Vendor experience with the design and fabrication of special-purpose gearing is no less important than vendor experience with any other critically important machinery category. Questions to be asked relate to pitch-line velocities, gear-blank (web) construction, horsepower levels, bearing design, gear-speed ratios, etc. When vendor experience has been established to the review engineer's satisfaction, he is ready to proceed with a comparison of competing bids. This comparison is aimed at determining which of the various offers may represent a stronger, potentially less failure-prone gear.

Design appraisals can be complex and time consuming if efforts are made to use the full complement of AGMA (American Gear Manufacturers Association) rating formulas. Moreover, cycles to failures calculated with some of these rating formulas can be drastically influenced by minor changes in the assumed or anticipated surface roughness, tooth spacing, etc. A sensible approach to gear design appraisals would not, therefore, use calculated probable cycles to failure in an absolute way. The review engineer would utilize the data only to make a comparison of competing offers and to assign a ranking order.

In the late 1960s, Robert H. Pearson,[41] then chief engineer of the Sier-Bath Gear Company, equated the mathematical expression for estimated gear-tooth compressive stress to that for allowable fatigue stress. Compressive stress is a measure of surface durability and pitting. Pearson's work, summarized in an article published by *Machine Design* magazine in 1968 determined

$$N_c = 3.8 \times 10^{-10} \left[\frac{dFI(H + 150)^2}{C_d W_t} \right]^{8.77}$$

In this expression

N_c = life in cycles to failure
d = pinion operating pitch diameter, in
F = face width, in
H = hardness, bhn
I = geometry factor
W_t = tangential driving force, lbs

The geometry factor I is obtained by dividing durability factor C_3 (obtained from Figure 3-73) by a materials factor $(s_{sc}/C_p)^2$ (obtained from Figure 3-74). W_t is readily calculated by dividing the pinion output torque by the pinion pitch radius. C_d, however, is a good deal more difficult to obtain. Five factors make up C_d and are defined as follows:

AGMA STANDARD PRACTICE FOR
HIGH SPEED HELICAL AND HERRINGBONE GEAR UNITS

SUGGESTED GEAR AND PINION HARDNESS COMBINATIONS

	MINIMUM BRINELL HARDNESS													
GEAR	180	210	225	245	255	270	285	300	315	335	350	51 R_C	55 R_C	58 R_C
PINION	210	245	265	285	300	315	335	350	365	385	400	51 R_C	55 R_C	58 R_C

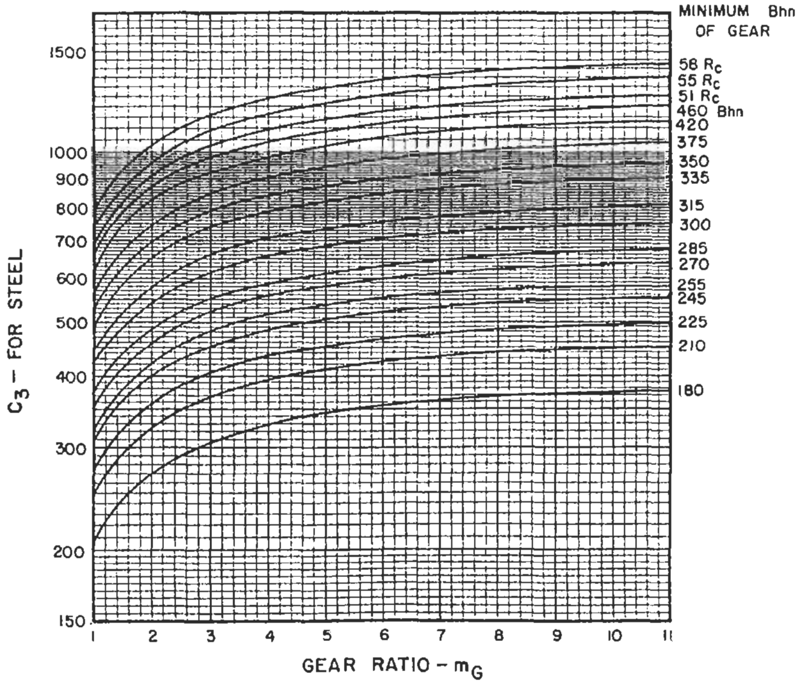
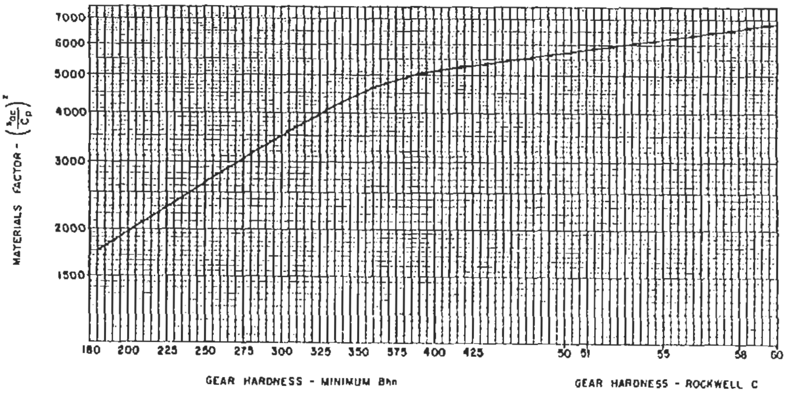


Figure 3-73. Strength factors for high-speed gear units. (*Courtesy AGMA, No. 421.06.*)

C_0: The overload factor allows for momentary torques and overloads imposed by the driver or driven load. Though best established by field experience, it can be estimated from Table 3-17. Since this factor can derate gear life by about 2,000 times, care should be taken in selecting C_0. Uniformity and frequency of torque fluctuation, particularly as related to the tooth mesh frequency, are important. If the rate is low and not a specific harmonic, a low value of C_0 can be used safely.

C_V: The dynamic factor has to do with the action between mating teeth. Its magnitude depends on spacing accuracy, pitch-line velocity, inertia, and stiffness of connected masses. Lubricant viscosity comes into play at pitch-line velocities of about 20,000 fpm and higher. Until being questioned in the mid 1970s,



AGMA STANDARD PRACTICE FOR
HIGH SPEED HELICAL AND HERRINGBONE GEAR UNITS

VALUES ARE TO BE TAKEN FROM THE ABOVE CURVE FOR THE MINIMUM HARDNESS SPECIFIED FOR THE GEAR
VALUES FOR SUGGESTED GEAR AND PINION HARDNESS COMBINATIONS ARE TABULATED BELOW FOR CONVENIENCE
PINION AND GEAR MAY BE THE SAME HARDNESS WHEN THE GEAR RATIO IS 1.5 OR LESS.

	MINIMUM BRINELL HARDNESS													
GEAR	180	210	225	245	255	270	285	300	315	335	350	51 R_c	55 R_c	58 R_c
PINION	210	245	265	285	300	315	335	350	365	385	400	51 R_c	55 R_c	58 R_c
$\left(\frac{s_{ac}}{C_p}\right)^2$	1750	2100	2300	2560	2700	2950	3200	3460	3750	4200	4460	5800	6200	6600

MATERIALS FACTOR FOR STEEL - $\left(\frac{s_{ac}}{C_p}\right)^2$

Figure 3-74. Materials factors for surface durability rating of helical and herringbone gears for high-speed gear units. (*Courtesy AGMA, No. 421.06.*)

Table 3-17
Overload Factor, C_o*

Characteristic of Driver	Characteristic of Driven Load: Uniform	Moderate Shock	Heavy Shock
Uniform load	1.00	1.25	1.75 and up
Light shock	1.25	1.50	2.00 and up
Medium shock	1.50	1.75	2.25 and up

**Source: AGMA. By permission.*

commonly used values of C_v ranged from 1.0 to 0.5 and lower. The minimum value of C_v would derate gear life by about 440 times.

Tooth-spacing accuracy exerts the greatest influence on C_v. For AGMA Quality 12 and very low stiffness and inertia, C_v could approach 1.0.

C_s: The size factor allows for gear and gear-tooth size, tooth-contact pattern, and reliability of material heat treatment. Generally, this factor is taken as 1, although it may reach about 1.25 when all influencing conditions are unfavorable. A value of 1.25 would derate gear life about 7 times.

C_f: The surface-condition factor takes into account tooth surface finish and residual stress. Generally, it is taken as 1. However, it may go to about 1.25 for rough finishes that would cause localized contacts, or when residual stresses are expected to be high or unpredictable. Both conditions can warrant a value as high as 1.50; this would derate gear life about 35 times.

C_m: The load distribution factor allows for anything that might prevent 100% gear-tooth contact: lead and profile errors, stiffness of gears and mountings, deviations from true alignment of the gear axis, and uneven thermal expansion during operation. If all such conditions were ideal, C_m would be 1.00; but practical conditions may drive this value to 3.00 or more. For a value of 3.00, gear life would be derated 1,530 times.

Experience shows that wide-face gears require special considerations to offset net misalignment and to obtain good load distribution at full torque. Generally, a face width equal to the pinion diameter is the best compromise.

It should be remembered that misalignment cannot be readily absorbed by the gear teeth except through plastic deformation, which is difficult to predict in terms of life. Extreme profile modifications in a helical mesh can reduce instantaneous lines of contact to nearly point contacts; this is as bad as severe misalignment. Usually a few ten thousandths addendum relief is enough to assure a smooth load transfer from tooth to tooth.

The combined derating factor C_d could reasonably vary from 1.00 to 20.25, with the latter reducing life about 2.87×10^{11} times. AGMA Standard 411.02 is very helpful in determining limits for C_d. Table 3-18, taken from AGMA Standard 411.02, gives derating factors for aircraft-quality gears: these may be used as guides in establishing minimum limits for C_d for any quality level. From a practical point of view, quality speed-increasing gears supplied to the petrochemical industry for turbomachinery drives most often exhibit life-cycle characteristics relating to a C_d of approximately 2.0. For double-helical gears using turbine oil as a mesh lubricant, N_c should be calculated on the basis of this derating factor.

Table 3-18
Derating Factor C_d for Aircraft and Helical Gears*

	AGMA Gear Quality Number			
Application	**9**	**10**	**11**	**12**
Main propulsion drive gears				
Continuous	—	1.8	1.5	1.2
Takeoff	—	1.5	1.2	1.0
Power-takeoff gears	2.4	2.1	1.8	1.5
Auxiliary power supply units	—	2.1	1.8	1.5

**Source: AGMA. By permission.*

Example: You receive two proposals for a gear-speed increaser with an input speed of 1792 rpm, an output speed of 8088 rpm, and a rated power output of 13,200 HP. Given the following data, which gear should you buy?

Pinion Data	Vendor A	Vendor B
Pinion diameter d	9.48 in	8.92 in
Face width F (total)	18.00 in	18.25 in
Minimum Brinell hardness	388	412

For shortcut design appraisal, use $C_d = 2.0$. Next, for Vendor A, calculate

$$W_{tA} = \frac{T}{d/2} = \frac{(63{,}025)\,(13{,}200)\,(2)}{(8{,}088)\,(9.48)} = 21{,}700 \text{ lbs}$$

Similarly, for B:

$$W_{tB} = \frac{T}{d/2} = \frac{(63{,}025)\,(13{,}200)\,(2)}{(8{,}088)\,(8.92)} = 23{,}063 \text{ lbs}$$

then

$$\begin{aligned} N_c &= 3.8 \times 10^{-10} \left[\frac{(9.48)\,(18)\,(0.194)\,(538)^2}{(2.0)\,(21{,}700)} \right]^{8.77} \\ &= 13.68 \times 10^{20} \text{ cycles} = 32 \text{ years (Vendor A)} \end{aligned}$$

and

$$\begin{aligned} N_c &= 3.8 \times 10^{-10} \left[\frac{(8.92)\,(18.25)\,(0.190)\,(562)^2}{(2.0)\,(23{,}063)} \right]^{8.77} \\ &= 9.51 \times 10^{20} \text{ cycles} = 22 \text{ years (Vendor B)} \end{aligned}$$

This example calculation establishes that Vendor A's offer ranks ahead of Vendor B's offer.

Alternative Gear Strength Comparison. An alternative method of comparing the *relative* strength of several gears is available to the engineer. Based on work performed by Lufkin Gear Company's James Partridge, we have often found this approach quite helpful for rapid screening studies. It consists of five steps:

Step 1: Collect data

P_{SC} = transmitted horsepower
n_p = pinion rpm
d = pinion pitch diameter, in.
F = net face width of the narrowest of the mating gears or the sum of the face widths of each helix of double helical, in.
m_G = gear ratio
A = pressure angle
H_g = gear hardness
L = net face width and gap
Ψ = helix angle
J = geometry factor (If not available from manufacturer, use Table 3-19)
SF = service factor (Table 3-20)
P_n = normal diametrical pitch = $P_d/\cos \Psi = N/(d\cos \Psi)$
N = number of teeth in piston

Step 2: Calculate *allowable* tooth pitting index, K_o

$$\text{Allowable } K_o = \frac{\text{Material Index Number}}{\text{Service Factor}}$$

a. Use material index number from Table 3-21 or Figure 3-75.
b. Use service factor from Table 3-20.
c. Verify that maximum pinion length/pitch diameter ratio (L/D) given in Table 3-21 is not exceeded on proposed gears.

Table 3-19
Gear Tooth Strength "J" Factor

No. of Teeth	Ground or Shaved 14.5° P.A.	Ground or Shaved 20° P.A.	Full Round Bottom 14.5° P.A.	Full Round Bottom 20° P.A.
Single Helical				
20	.39	.47	.44	.51
30	.43	.50	.49	.55
60	.47	.55	.56	.60
150	.50	.56	.58	.61
500	.51	.58	.60	.63
Double Helical				
20	.37	.44	.43	.49
30	.39	.47	.46	.52
60	.42	.50	.50	.55
150	.43	.51	.52	.56
500	.45	.52	.53	.57

Note: Values are estimate for full depth tooth and P.A. (Pressure Angle) is measured in the normal plane.

Table 3-20
Minimum Gear Service Factor (SF)

	Prime Mover		
Driven Equipment	**Motor**	**Turbine**	**Internal Combustion Engine**
Blowers	1.4	1.6	1.7
Centrifugal			
Compressors		1.6	1.7
Centrifugal	1.4	1.6	1.7
Axial	1.4	1.6	1.7
Rotary Lobe (radial, axial, screw, etc.)	1.7	1.7	1.7
Reciprocating	2.0	2.0	2.3
Fans			
Centrifugal	1.4	1.6	1.7
Forced Draft	1.4	1.6	1.7
Induced Draft	1.7	2.0	2.2
Generators and exciters			
Base Load or Continuous	1.1	1.1	1.3
Peak Duty Cycle	1.3	1.3	1.7
Pumps			
Centrifugal (All service except as listed below)	1.3	1.5	1.7
Centrifugal—Boiler feed	1.7	2.0	—
Centrifugal—Hot oil	1.7	2.0	—
High-speed Centrifugal (Over 3,600 rpm)	1.7	2.0	—
Centrifugal—Water supply	1.5	1.7	2.0
Rotary-Axial Flow—All types	1.5	1.5	1.8
Rotary—Gear	1.5	1.5	1.8
Reciprocating	2.0	2.0	2.3

Step 3: Calculate *actual* tooth pitting index, K′

$$K' = \frac{126{,}000 \times P_{sc}}{n_p \times d^2 \times F} \times \frac{m_G + 1}{m_G}$$

Verify that actual K′ does not exceed allowable K_o-value: K_o/K' must be ≧ 1.0.

Step 4: Calculate maximum *actual* bending stress

$$S_t = \frac{P_{sc} \times 126{,}000 \times P_n}{n_p \times d \times F} \times (SF)\frac{1.8 \cos \Psi}{J}$$

Step 5: Determine maximum *allowable* bending stress—from Figure 3-76, enter gear hardness value.

Verify that maximum actual bending stress does not exceed maximum allowable bending stress.

Table 3-21
Material Index Numbers and Maximum L/d's

Gear Hardness Minimum	Pinion Hardness Minimum	Material Index Number	Maximum Pinion L/d Double Helical	Single Helical
223 Bhn	265 Bhn	130	2.4	1.7
262 Bhn	302 Bhn	160	2.3	1.6
302 Bhn	341 Bhn	200	2.2	1.5
352 Bhn	$50R_c$ (Nitrided)	260	2.0	1.45
$50R_c$ (Nitrided)	$50R_c$ (Nitrided)	300	1.9	1.4
$55R_c$ (Carburized)	$55R_c$ (Carburized)	410	1.7	1.35
$58R_c$ (Carburized)	$58R_c$ (Carburized)	440	1.6	1.3

Note:
L = Net face width plus gap
d = Pinion pitch diameter

The sample calculation on page 190 indicates that *both* gears are satisfactory, since the maximum *actual* bending stresses do not exceed the maximum *allowable* bending stresses. As long as the final ratio calculated in Step 5 equals or exceeds 1, the gear incorporates an adequate safety margin for the intended service. On the other hand, the higher this ratio, the stronger the gear. Everything else being equal, Vendor B made the better offer.

Checking for Scoring Susceptibility. The susceptibility of a given design to scoring damage in service can be checked by calculating a so-called scoring index:

$$\text{S.I.} = \left(\frac{W_t}{F}\right)^{.75} \frac{(N_p)^{.5}}{(d)^{.25}}$$

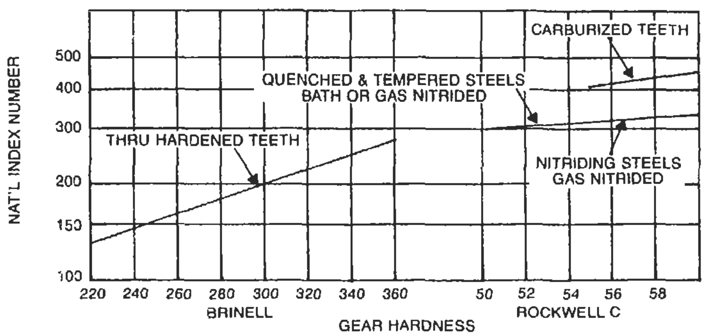


Figure 3-75. Material index numbers.

Example Problem

Step 1:

Vendor A Proposes:

$P_{sc} = 13{,}200$ hp
$n_p = 8{,}088$ rpm
$d = 9.48$ in.
$F = 18$ in.
$H_G = 338$ Bhn minimum

$$m_G = \frac{8{,}088}{1{,}792} = 4.513$$

$\Psi = 30.02°$
$N = 41$ pinion teeth
$A = 14.5°$, ground teeth
$L = 19$ in.

Step 2: (From Figure 3-75)

$$K_o = \frac{246}{1.4} = 175.7$$

$L/d = 19/9.48 = 2$

Step 3:

$$K' = \frac{126{,}000 \times 13{,}200 \times 5.513}{8{,}088 \times 89.87 \times 18 \times 4.513}$$

$= 155.3$
Ratio: $175.7/155.3 = 1.13$

Step 4:

$P_n = N/(d/\cos\Psi)$
$= 41/9.48 \times 0.8658) = 5$

$$S_t = \frac{13{,}200 \times 126{,}000 \times 5}{8{,}088 \times 9.48 \times 18} \times 1.4$$

$$\times \frac{1.8 \times 0.8658}{0.50}$$

$= 26{,}293$ psi

Step 5: (From Figure 3-76)

For $H_G = 338$ Bhn
$S_t = 29{,}000$ psi

$$\text{Ratio} = \frac{29{,}000}{26{,}293} = 1.103$$

Vendor B Proposes:

$P_{sc} = 13{,}200$ hp
$n_p = 8{,}087$ rpm
$d = 8.77$ in.
$F = 18.25$ in.
$H_G = 362$ Bhn minimum

$$m_G = \frac{8{,}087}{1{,}792} = 4.512$$

$\Psi = 27.2°$
$N = 39$ pinion teeth
$A = 20°$, shaved teeth
$L = 19.5$ in.

$$K_o = \frac{280}{1.4} = 200$$

$L/d = 19.5/8.77 = 2.22$

$$K' = \frac{126{,}000 \times 13{,}200 \times 5.512}{8{,}087 \times 76.91 \times 18.25 \times 4.512}$$

$= 179$
Ratio: $200/179 = 1.12$

$P_n = N(d/\cos\Psi)$
$= 39/(8.77 \times 0.8894) = 5$

$$S_t = \frac{13{,}200 \times 126{,}000 \times 5}{8{,}087 \times 8.77 \times 18.25} \times 1.4$$

$$\times \frac{1.8 \times 0.8894}{0.56}$$

$= 25{,}714$ psi

For $H_G = 362$ Bhn
$S_t = 31{,}500$ psi

$$\text{Ratio} = \frac{31{,}500}{25{,}714} = 1.225$$

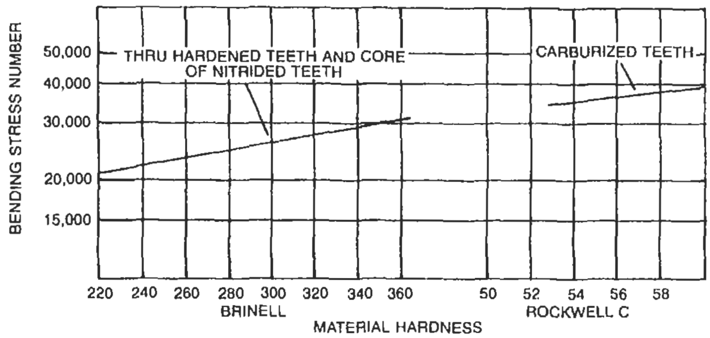


Figure 3-76. Bending stress numbers.

In this expression, the previously defined parameters are joined by N_p, the pinion speed (rpm).

The following may be assumed:

S.I. < 14,000: slight probability of scoring

14,000 < S.I. < 18,000: moderate probability

S.I. > 18,000: high probability of scoring

Reviewing our example shows

$$S.I._A = \left(\frac{21,700}{18}\right)^{.75} \frac{(8,088)^{.5}}{(9.48)^{.25}} = 10,486$$

and

$$S.I._B = \left(\frac{23,700}{18.25}\right)^{.75} \frac{(8,088)^{.5}}{(8.92)^{.25}} = 11,029$$

Both offers are completely acceptable from the point of view of scoring.

Gear-Tooth Backlash Review

Many gear manufacturers consider the backlash values found in Table 3-22 appropriate for solid or forged-steel gears.

These backlash values have sometimes been viewed as relatively tight. However, even these seemingly close backlash tolerances take into account a 50°F temperature rise and gear growth due to centrifugal force at a pitch-line velocity of 25,000 fpm.

Table 3-22
Backlash Values for Solid or Forged-Steel Gears

Center Distance, Inches	Normal Diametral Pitch					
	1–2	2–3	3–4	4–8	8–12	12–20
to 6	min/max		.010/.016	.008.014	.008.014	.006/.012
6–10		.014/.022	.012/.018	.010/.016	.010/.016	.008/.014
10–15		.016/.024	.014/.022	.012/.018	.012/.018	
15–20	.020/.030	.018/.028	.016/.024	.014/.022	.012/.020	
20–25	.022/.032	.020/.030	.018/.026	.016/.024	.014/.022	
25–30	.024/.034	.022/.032	.020/.030	.018/.028	.016/.026	
30–35	.026/.036	.024/.034	.022/.032	.020/.030		
35–40	.026/.036	.026/.036	.024/.034	.022/.032		

Notes: 1. Includes gear blank temperature 50°F above housing.
2. Includes centrifugal growth up to 25,000 fpm.
3. Backlash is a minimum of 1½ times that required by (1) and (2).

When backlash figures do appear too tight, we can perform the following simplified calculations to verify the adequacy of a vendor's backlash values.

Radial stresses in solid rotating discs, psi:

$$S_r = \frac{\rho v^2}{g}\left(\frac{\mu + 3}{8}\right)\left[1 - \left(\frac{r}{r_o}\right)^2\right]$$

where
ρ = weight density, lb/in^3 (.280 for steel)
g = 386 in/sec^2
μ = Poisson's ratio (0.30 for steel)
r_o = outside radius of disc, in
r = variable radius at which stress is to be found
v = peripheral velocity, in/sec

From Hooke's law, for uniaxial loading:

$$\text{Strain } \epsilon = \left(\frac{S_r}{E}\right); \quad \frac{S_r}{30 \times 10^6} = \frac{\text{in}}{\text{in}}$$

Maximum growth in radial direction:

$$\delta = (\epsilon)\ (\text{center distance, in})$$

Decrease in backlash due to maximum growth in radial direction:

$$\Delta x_1 = (2)\ (\delta \tan \Theta)$$

where

Θ = pressure angle

Decrease in backlash due to thermal expansion:

$$\Delta x_2 = (2\alpha)(\Delta T)(C \tan \Theta)$$

where

α = temperature coefficient of expansion (6.5×10^{-6} in/in for steel)
ΔT = maximum anticipated temperature rise, °F
C = center distance, gear-to-pinion, in

Total decrease in backlash = $\Delta x_1 + \Delta x_2$

Mesh Lubrication

Industry practices indicate that some flexibility can be conceded on nozzle placement. As an example, for pitch-line velocities between 5,000 and 15,000 fpm, the oil spray could be directed to either the incoming or outgoing side of the mesh. However, the most reliable means of mesh lubrication would be to spray oil into the outgoing mesh side because it would allow maximum cooling time for the gear set, and would apply cooling oil at areas of highest temperature. Above 15,000 fpm, about 90% of the oil should be sprayed into the outgoing, and only about 10% into the incoming mesh. This assures the application of *cooling* oil in the high-temperature areas, and *lubricating* oil in the high-contact stress areas. As for total oil quantity sprayed into the gear teeth, we could use one of two rules of thumb:

1. For every inch of face width, 5 gpm of flow would be required. This guideline does not consider the effects of speed on horsepower ratings and heat loss, but appears to be valid for a wide range of gears.
2. Oil flow, gpm = $\dfrac{0.42 \times \text{gear HP lost}}{1.4}$

Application of this formula should limit gear temperature rise to approximately 40°F, even on less-than-average-efficiency gears. While the highest flow value may not necessarily be optimum flow for a given mesh, it should nevertheless govern the sizing of the lube system.

Why "Property Rated" Gears Still Fail

The design of the gear teeth for desired strength and durability is generally given close attention by both the purchaser and the manufacturer, and service factors are applied according to AGMA 421.06 to take care of expected overload conditions.

However, gear-tooth failures still occur. One basic but frequently overlooked cause of gear failures is tooth overload due to axial-thrust transmission to the mesh. The most common source of externally imposed axial thrust arises from gear couplings used to connect the gear box to the driver and driven equipment. Gear-unit thrust bearings are usually rated to absorb the maximum anticipated axial thrust. However, due to the transient nature of coupling-transmitted axial thrust, it is rarely included in the tooth strength and durability ratings.

This section shows how to assess the overload effects of axial-thrust transmission on double-helical gears as an aid to failure investigation. It also outlines a number of options to keep coupling-transmitted excessive axial thrust from contributing to a gearing problem.

Thrust Transmitted Through Gear-Tooth Couplings. Friction between the teeth of gear-type couplings is responsible for the transmission of axial thrust between two coupled machines. This friction resists the normal relative movement between coupled shafts that occurs due to thermal expansion or hydraulic forces and results in transmission of axial forces from one rotor to another through the coupling. The maximum axial thrust transmitted is $F = 2\ T\mu/D_p \cos \Theta$, where T is the torque at the coupling (in pound-inches), μ is the coefficient of friction between the coupling teeth, D_p is the coupling tooth pitch diameter, and Θ the coupling tooth pressure angle. The coefficient of friction, μ, is assumed to be 0.15 (API 613), although some petrochemical industries report values as high as 0.30. Higher values are simply a reflection of conditions frequently encountered in petrochemical plants with less than optimum coupling design and/or lubrication.

The coupling friction forces are generally transient in nature. However, relatively high friction factors have actually been observed to persist for long periods of time on poorly lubricated gear couplings. The actual value depends on such variables as tooth interference due to thermal and sometimes centrifugal growth, viscosity and adhesion of lubricant, or lubricant film interruption because of either severe coupling misalignment *or* long-term operation with virtually perfect alignment.

Location of Thrust Bearing Must Be Considered. The location of the gear-box thrust bearing will determine whether thrust transmission will occur for a given machinery train arrangement. For the train configuration shown in Figure 3-77, the thrust bearing is generally part of the high-speed pinion shaft assembly. Using a coupling friction coefficient of 0.3, the pinion thrust bearing could be exposed to a maximum axial thrust of

$$F_1 = \frac{(2)(0.3)(T_{HSS})}{D_{PHS} \cos \Theta}$$

(where T_{HSS} = torque acting on high-speed shaft, and D_{PHS} = tooth pitch diameter of high speed coupling) plus M_A, the magnetic centering force of the motor.

However, the maximum possible axial force imposed on the gear *mesh* on Figure 3-77 is not related to F_1, but rather, as will be seen later, to the tangential driving load F_T and the magnetic centering force M_A of the motor.

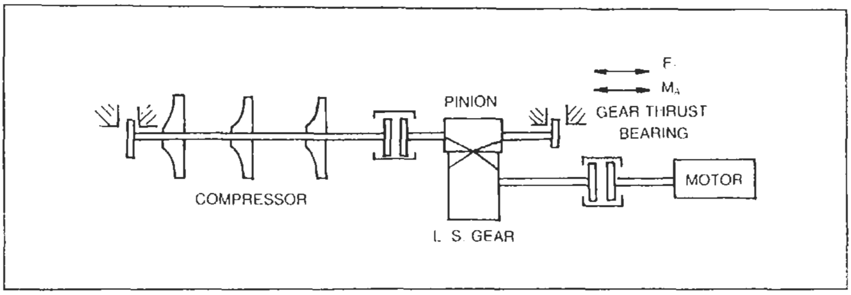


Figure 3-77. Forces acting on thrust bearing mounted on gear unit pinion shaft.

If the thrust bearing is mounted on the low-speed gear shaft, as shown in Figure 3-78, it could be exposed to the same maximum thrust as the bearing shown in Figure 3-77. However, the thrust force is now transmitted across the gear mesh. The maximum possible axial force imposed on the mesh is now not only a function of F_T but of F_t as well.

The analysis can, of course, be extended to drivers other than motors. Unlike motors, which are generally furnished with axially free-floating rotors, these other drivers will probably incorporate thrust bearings.

Axial Thrust Transmitted Through Gear Mesh Can Overload One Helix. This condition can best be shown by sample calculations.

Let us assume we are dealing with a drive arrangement as shown in Figure 3-77. The compressor absorbs 5,000 HP at 6,400 rpm, and the step-up gear has been designed with a 1.5 service factor and this equates to torque.

$$T_c = \frac{(63{,}000)(\text{HP})}{\text{rpm (Compressor)}}$$

$$T_c = \frac{(63{,}000)(5{,}000)}{6{,}400}$$

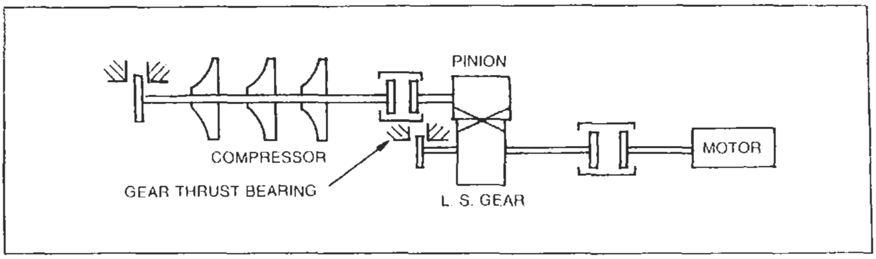


Figure 3-78. Forces acting on thrust bearing mounted on low-speed gear shaft.

or 49,200 lb-in. Using a pinion pitch diameter of 12 in, the tangential driving load would be

$$F_T = \frac{(2)(49,200)}{12}$$

or 8,200 lbs. This tangential driving load can be resolved into axial-thrust components by multiplication with the tangent of the helix angle. For $\Psi = 30°$, the axial thrust per helix will equal (8,200/2) tan 30°, or 2,365 lbs. In addition, it is readily evident that the gear mesh can be exposed to an external force, F_A. The magnitude of F_A is the lesser of the two values, F_2 or M_A typically around 1,200 lbs for a 5,000 HP induction motor turning at 1,780 rpm, and

$$F_2 = \frac{(2)\,(0.3)\,(T_{LSS})}{D_{PLS} \cos \Theta}$$

Since the motor torque T_{LSS} is (63,000) (5,000)/1,780 = 177,000 lb-in., and low-speed coupling pitch diameters D_{PLS} are probably in the 10 in to 12 in range, F_2 will no doubt be significantly larger than M_A. Accordingly, the magnetic centering force of the motor constitutes the external force imposed on the gear mesh, i.e. it may create an axial push or pull acting through coupling teeth that refuse to slide.

A free-body diagram can now be constructed to investigate whether the various forces can overload one helix of a typical double-helical gear. Using trigonometric relationships and summation of forces (see Figure 3-79), we have

$A = C \tan \Psi$, $C = A/\tan \Psi = 1.72A$
$B = D \tan \Psi$, $D = B/\tan \Psi = 1.72\,B$
$C + D = F_T = 8,200$ lbs
$A + B = F_T \tan \Psi = 4,730$ lbs
$A - B + F_A = 0$
$A - B + 1,200 = 0$
$2A = 3,530$ lbs, $A = 1,765$ lbs
$B = 2,965$ lbs, $C = 3,060$ lbs, $D = 5,140$ lbs

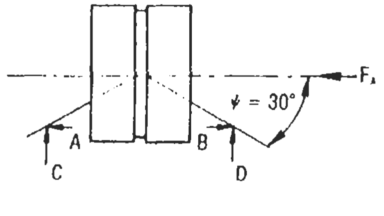


Figure 3-79. Free-body diagram for double-helical gears arranged per Figure 3-72.

Rated tangential load per helix:

$$\frac{8{,}200 \times 1.5}{2} = 6{,}150 \text{ lbs}$$

Tooth loading with maximum possible thrust:

$$\frac{(100)\,(5{,}140)}{6{,}150} = 84\% \text{ of rated tangential load}$$

This analysis shows that even with the maximum anticipated external force, the gear teeth are safe for the particular train configuration shown.

For the drive arrangement shown in Figure 3-78, the external force F_A acting on the gear mesh can be as much as

$$F_1 = \frac{(2)(0.3)(T_{HSS})}{D_{PHS} \cos \Theta}$$

This condition can exist because the low-speed shaft thrust bearing constrains the axial movement of the input gear. Thus

$$F_1 = F_A = \frac{(2)(0.3)(49{,}200)}{(7)(0.965)\,*} = 4{,}380 \text{ lbs}$$

We now have the following force equations (see Figure 3-79):

$A = C \tan \Psi$, $C = A/\tan \Psi = 1.72A$
$B = D \tan \Psi$, $D = B/\tan \Psi = 1.72B$
$C + D = F_T = 8{,}200$ lbs
$A + B = F_T \tan \Psi = 4{,}730$ lbs
$A - B + F_A = 0$
$A - B + 4{,}380 = 0$
$2A = 350$ lbs, $A = 175$ lbs
$B = 4{,}555$ lbs, $C = 300$ lbs
$D = 7{,}900$ lbs

Rated tangential load per helix:

$$\frac{8{,}200 \times 1.5}{2} = 6{,}150 \text{ lbs}$$

---

*Assuming a coupling-tooth pitch diameter of 7 in and a pressure angle of 14.5°. Note that some gear couplings have pressure angles as high as 40°, which may cause a very significant increase in F_A.

Tooth loading with maximum possible thrust:

$$\frac{(100)\,(7{,}900)}{6{,}150} = 128\% \text{ of rated tangential load}$$

This analysis demonstrated that a gear-type coupling with an anticipated maximum coefficient of friction of 0.3 may allow the transmission of excessive axial thrust into the gear mesh. In other words, if a gear required all of its designed-in service factor initially, the additional thrust transmitted through gear-type couplings may further reduce the service factor to a serious degree. While not usually causing immediate tooth failures, the increased loading may result in gear failure after a relatively short time of operation should the axial thrust be continuously present.

If the failure analysis concludes a gear mesh failure due to axial thrust, there are a number of options, some of which are included here.

- The coupling and coupling lubrication system can be examined to determine the cause of excessive frictional forces.
- A thrust bearing with lower load-carrying capacity can be installed purposely. This, in effect, makes the thrust bearing the weak link, or sacrificial component. Using adequate safeguards, such as temperature thermocouples embedded in the thrust-bearing material and connected to a shutdown system, potential failures thus confined to the thrust-bearing replacements are certainly less costly and less time consuming than gear-tooth failures.
- Replace couplings with large tooth pitch diameter gear couplings.
- The coupling can be replaced with an axially soft diaphragm coupling which transmits little or no axial thrust. Diaphragm couplings accommodate misalignment while transmitting power by material flexure. However, the purchaser should recognize that high axial movement cannot be accommodated by diaphragm couplings. This requires that the initial system axial alignment take into account the thermal expansion of both coupled shafts.
- Replacement gear elements can be purchased with a larger service factor than listed in AGMA 421.06.

The calculations illustrated in this section can actually be performed to define the service factor necessary to prevent gear mesh failures resulting from axial-thrust transmission.

Determination of Effect of High-Speed Coupling Lockup Transmitted Through Mesh. See Figure 3-80 ($A - B + T_A = 0$).

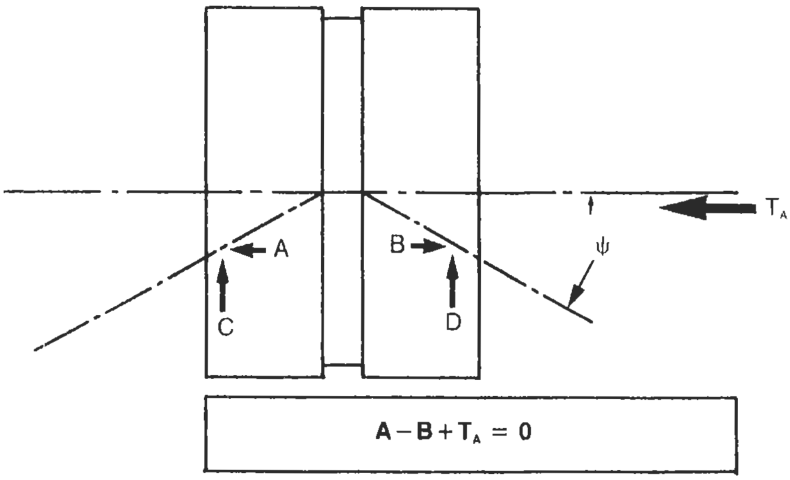


Figure 3-80. Free-body diagram for double-helical gear arranged per Figure 3-78.

$$T = \text{torque} = \frac{63{,}000 \times HP}{rpm} \qquad = \underline{\qquad\quad} \text{ in - lb}$$

$$T_A = \text{axial thrust} = \frac{.3 \times \text{torque}}{\text{coupling radius}} \qquad = \underline{\qquad\quad} \text{ lbs}$$

$$F = \begin{matrix}\text{tangential}\\ \text{driving load}\end{matrix} = \frac{2 \times \text{torque}}{\text{pinion pitch diameter}} \qquad = \underline{\qquad\quad} \text{ lbs}$$

$$\begin{matrix}\text{thrust per}\\ \text{helix}\end{matrix} = \frac{\text{tangential driving load} \times \tan \Psi}{2} \qquad = \underline{\qquad\quad} \text{ lbs}$$

$$A = C \tan \Psi \qquad C = \frac{A}{\tan \Psi}$$

$$B = D \tan \Psi \qquad D = \frac{B}{\tan \Psi}$$

$C + D = F$
$A + B = F \tan \Psi$
$A - B = -T_A$

$2A =$ ________ lbs $\qquad A =$ ________ lbs
$B =$ ________ lbs
$C =$ ________ lbs
$D =$ ________ lbs

Rated tangential load per helix $= * \dfrac{F^1}{2} =$ ________ lbs

Service factor with maximum thrust $= \dfrac{F^1}{2}$ largest force = ________

Evaluating Cooling Tower Fans and Their Drive Systems

Over the past decades, cooling tower fans have been designed and put into service with diameters exceeding 30 ft. More often than not, the vendor is simply extrapolating his past design by going from 26 ft to 28 ft, 30 ft, or even larger diameters. Extrapolation is generally synonymous with simple scaleup of representative dimensions. At other times, only the blade length is increased and the blade hub or blade internals are left untouched.

Many of these extrapolations have resulted in costly failures risking extended downtime or injury to personnel. Detailed design reviews are appropriate and the following items represent a cross section of topics.

1. The dynamic natural frequency of cooling fan blades should be at least 20% away from the fan rpm and its multiples.

   Background: Force amplification resulting from coincidence or near-coincidence of blade natural frequency and forcing frequency has caused catastrophic blade failures in many cooling tower installations. Figure 3-81 shows one such event.
2. If urethane-foam filter material is used in constructing fan blades, the vendor should submit data showing dynamic natural frequency of blades after urethane filler material loses intimate bonding, i.e., delaminates or separates from blade-skin interior surfaces. Alternatively, the vendor should submit proof that delamination or separation between filler and blade-skin interior will not occur with his design.

   Background: Loss of bonding has been experienced on a large number of blades. The resultant lowering of the blade dynamic natural frequency may cause coincidence or near-coincidence of blade natural frequency and forcing frequency.

$^*F^1$ = actual tangential driving load times lowest calculated service factor (service factor possibly per manufacturer's nameplate).

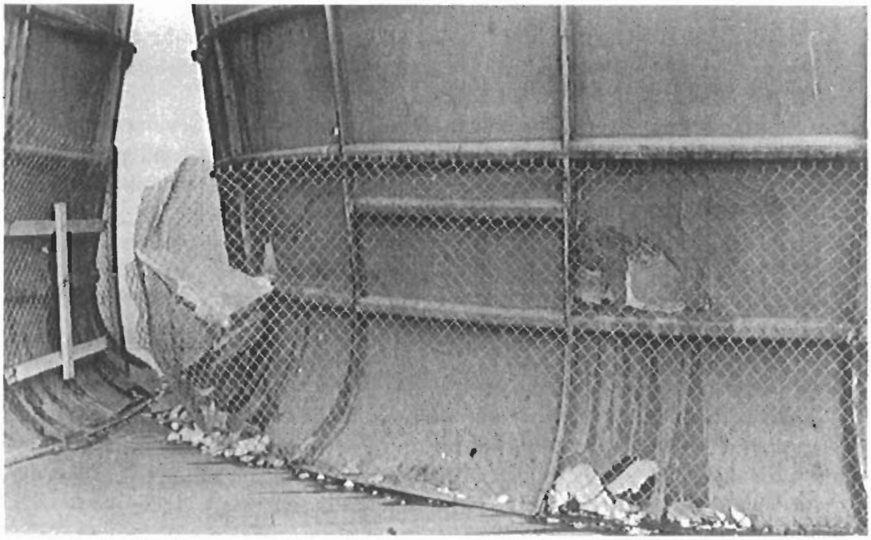


Figure 3-81. Cooling tower fan blade failure. The broken blade pieces were caught in a safety fence.

3. Vendor should show proof that the proposed fan-drive system is torsionally equivalent to at least six fan drives operating at the same speed. These fan drives should have been in actual operation at design load and speed for at least two years. Alternatively, the vendor may submit certified results of torsiograph testing or torsional critical speed calculations conducted by at least two independent torsional consultants.

   Background:

   a. Torsional critical speed problems have been responsible for costly gear failures. In one recent installation, the vendor calculated the first torsional critical speed at 908 cpm. This would have meant operating only 2% away from two times intermediate shaft speed of the right-angle reducing gear (2 × 445 = 890 rpm), and potentially excessive stresses might have been generated by torsionally resonant conditions. The user commissioned an independent consultant to perform analytical and field torsiograph analyses. The measured natural frequencies of 9 cps and 82 cps (540 and 4920 cpm) compared well with the consultant's calculated torsional criticals of 9.2 cps and 80.8 cps (552 cpm and 4848 cpm), but proved far from the vendor's calculated values.
   b. A detailed check of torsionally equivalent fan-drive systems previously delivered to other users showed that none of them was actually operating.
   c. Unfortunately, data pertaining to torsional natural frequencies and hunting tooth combinations are not usually part of the specifications package for cooling towers. It is recommended that relevant portions of special-purpose gear specifications (API, AGMA) be invoked for cooling tower gears as well.
4. The vendor's welding procedures and blade-stress calculations should be submitted for purchaser's review no later than four weeks after placement of the order.

Background: Serious failures of fan spokes (i.e. central hub arms) and fan blades have been attributed to inadequate weld procedures, insufficient design margins of safety, or both.

5. Vibration cutout devices should incorporate velocity- or acceleration-based solid-state circuitry, a built-in electronic delay circuit, and analog outputs to allow continuous monitoring and trend analysis. The automatic shutdown feature of the vibration monitoring device should energize to trip.

   Background: Some vibration cutout switches furnished by cooling tower vendors are simple devices which have clearly demonstrated unsatisfactory service life and have failed to actuate under emergency conditions. Fires and severe mechanical damage at several locations have been attributed to faulty vibration cutout devices.
6. At one process plant, cyclone fences were place around the fan stacks after the first blade failure sprayed debris over a wide area of the unit. These fences entrapped virtually all significant pieces of subsequent blade failures. Similar fences should be placed around the fiberglass stacks of cooling tower installations when using extra-large or unproven blade designs.

Cooling Tower Fan Mechanical Test Example. With a low-frequency accelerometer temporarily clamped to a point mid-span on the airfoil skin, each blade is struck and the resulting frequency displayed on a digital frequency analyzer. Observed values represent the blade static frequency f_{ns}. These values are recorded for later comparison with a "safe design" criterion.

Knowledgeable vendors define "safe design" as

$$\frac{(N \times \text{rpm}) - f_{nd}}{\text{rpm}} \geqq .2 \tag{3-1}$$

In this expression, N is any integer from 1 to perhaps two times the number of blades utilized in the fan. Rpm is the fan rpm, and f_{nd} is the blade dynamic frequency. This dynamic frequency f_{nd} is related to the static frequency f_{ns} by the expression

$$f_{nd} = \sqrt{f_{ns}^{\,2} + k\,(\text{rpm})^2} \tag{3-2}$$

The factor k is experimentally determined by the fan vendor and relates the dynamic frequency to the static frequency.

In our example, k is given to be 1.5; the fan speed is 117 rpm. When we plug the values for N and rpm into Equation (3-1), we find a "safe" value of $f_{nd} \leq 445$ cpm. The highest permissible static frequency f_{ns} should therefore not exceed

$$\begin{aligned} f_{ns} &= \sqrt{(445)^2 - 1.5\,(117)^2} \\ &= 421.3 \text{ cpm or } 7.02 \text{ cps} \end{aligned} \tag{3-3}$$

Where these values are exceeded, the user should experimentally verify the effect of adding tip weights on blade static natural frequency. In one such test, identical weights were clamped to the tips of two blades, one having an unweighted f_{ns} of 7.48 cps, and the other having an f_{ns} of 7.12 cps. The addition of this weight lowered the f_{ns} values of the two blade to 6.52 cps and 6.32 cps, respectively. This test proved that the *permanent* bonding of equal weights into the blade tips could be considered a viable fix for these blades, and would shift f_{nd} into the "safe" range.

Blade stress investigations would follow. These experimental tests would require that strain gauges be bonded to the most highly stressed portions of blade spar and hub arms. Wires would have to connect with telemetry instrumentation located in the center of the fan. This would allow the recording of alternating stresses of fan components exposed to vibration frequencies and vibration amplitudes of blades fitted with weighted tips.

Reliability Reviews in Uprate Situations

In principle, uprate situations require at least the same diligent review of a manufacturer's design as would the original review of rotating machinery being built from the ground up. In addition to the thermodynamic and rotor-dynamics analyses, much emphasis must be devoted to strength-of-materials criteria.

However, well-structured stress reviews can be rewarding, and have resulted in very significant cost savings to process plants. The actual example of a large mechanical-drive steam turbine in an overseas installation illustrates how one such uprate review task was approached.

Overview

Steam turbines and centrifugal compressors are generally provided with standardized shaft dimensions at their respective coupling ends. While this standardization approach will, of course, result in stress levels within manufacturer's allowable limits for initially rated conditions of the equipment, the equipment owner may find the maximum safe (or allowable) stress levels exceeded at some proposed future uprate conditions.

At first glance, then, the equipment uprate would appear to require time-consuming and often costly shaft replacements. However, closer examination of how the equipment vendor arrived at his maximum allowable stress levels may show that such shaft replacements can often be avoided without undue risk if the coupling selection is optimized. This conclusion is based on the fact that gear-type couplings have the potential of inducing in a shaft both torsional stresses *and* bending stresses, whereas diaphragm couplings tend to primarily induce torsional stresses and insignificant bending stresses at best. Bending moments caused by couplings transmitting torque while misaligned can be quite high and possibly contribute to bearing distress, seal wear, shaft-fatigue stresses, shaft lateral vibrations, deflections, and whirl.

The economic incentives of finding ways of salvaging major rotating equipment shafts are illustrated on a steam-turbine shaft originally rated to transmit 17,600 HP maximum at 6,400 rpm. Several years ago, a turbine uprate to 19,600 HP was authorized. It was determined that the required change-out of stationary steam-path components would cost around \$60,000, but a combined replacement cost of about \$500,000 was quoted for the main and spare rotor shafts. A rigorous calculation of shaft stresses showed the shaft *factor of safety to be greater at 19,600 HP using a diaphragm coupling than at 17,600 HP using a conventional gear coupling!*

Maximum Shaft Stress Calculated

Efforts to determine whether or not rotating equipment power uprates require shaft replacements should be preceded by shaft stress calculations. The coupling end of the steam-turbine shaft used in our example had the dimensions shown in Figure 3-82.

For ASTM A-293 shaft material, heat treatment and stabilization at 1,000°F resulted in the following properties:

Ultimate strength in tension	σ_{ut} = 105,000 psi
Ultimate strength in shear	τ_{ut} = 60,600 psi
Endurance limit in shear	τ_E = 30,300 psi
Minimum yield strength in tension	σ_{yp} = 80,000 psi
Minimum yield strength in shear	τ_{yp} = 40,000 psi
Endurance limit in tension	σ_E = 52,500 psi

Most of these properties are used in a Soderberg diagram[42] similar to Figure 3-83.

In addition to some of the nomenclature given earlier for the shaft properties, the diagram uses σ_m (steady tensile stress component), σ_a (alternating tensile stress component), and k_f (the stress concentration factor for the particular keyway dimensions shown in Figure 3-82). The stress concentration factor for our sample keyway is 2.9 (obtained from Figure 3-84).[43]

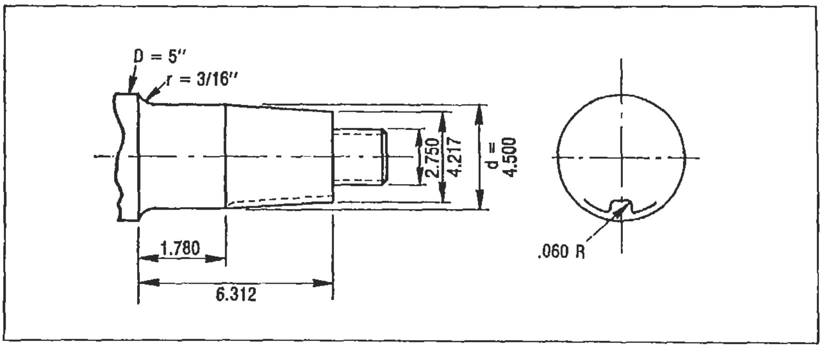


Figure 3-82. Steam-turbine shaft dimensions.

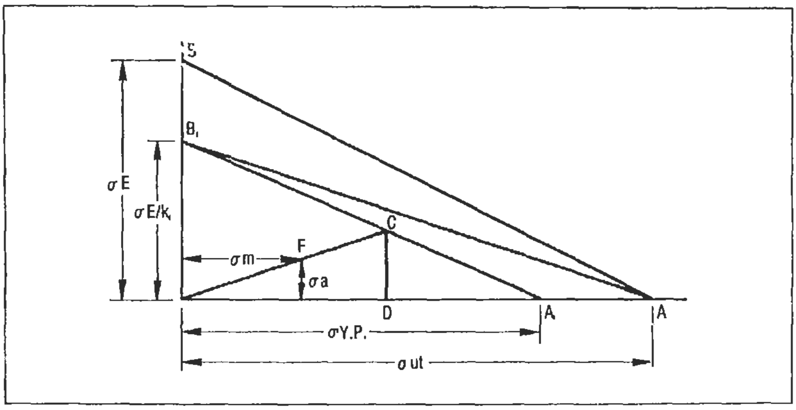


Figure 3-83. Soderberg diagram for steam-turbine shaft.

The line B_1A_1 is considered to define the limiting stress condition for a specimen with stress concentrations. If the steady stress on the specimen is given by the abscissa OD, then the limiting amplitude of the alternating stress is given by the ordinate DC, and point C will represent the limiting stress condition. The corresponding safe condition will be represented by point F with coordinates σ_m and σ_a (or τ_m and τ_a). These coordinates are obtained by dividing the coordinates of point C by the factor of safety n. From the similarity of triangles we have:

$$OB_1A_1 = DCA_1$$

$$\frac{CD}{\sigma_E / k_f} = \frac{DA}{\sigma_{yp}} = \frac{\sigma_{yp} - OD}{\sigma_{yp}}$$

or

$$\frac{CD}{\sigma_E / k_f} = \frac{OD}{\sigma_{yp}} = 1$$

Dividing this equation by n, we obtain for the safe stress condition (point F):

$$\frac{\sigma_a}{\sigma_E / k_f} + \frac{\sigma_m}{\sigma_{yp}} = \frac{1}{n} \tag{3-4}$$

where

$$n = \frac{1}{k_f \dfrac{\sigma_a}{\sigma_E} + \dfrac{\sigma_m}{\sigma_{yp}}}$$

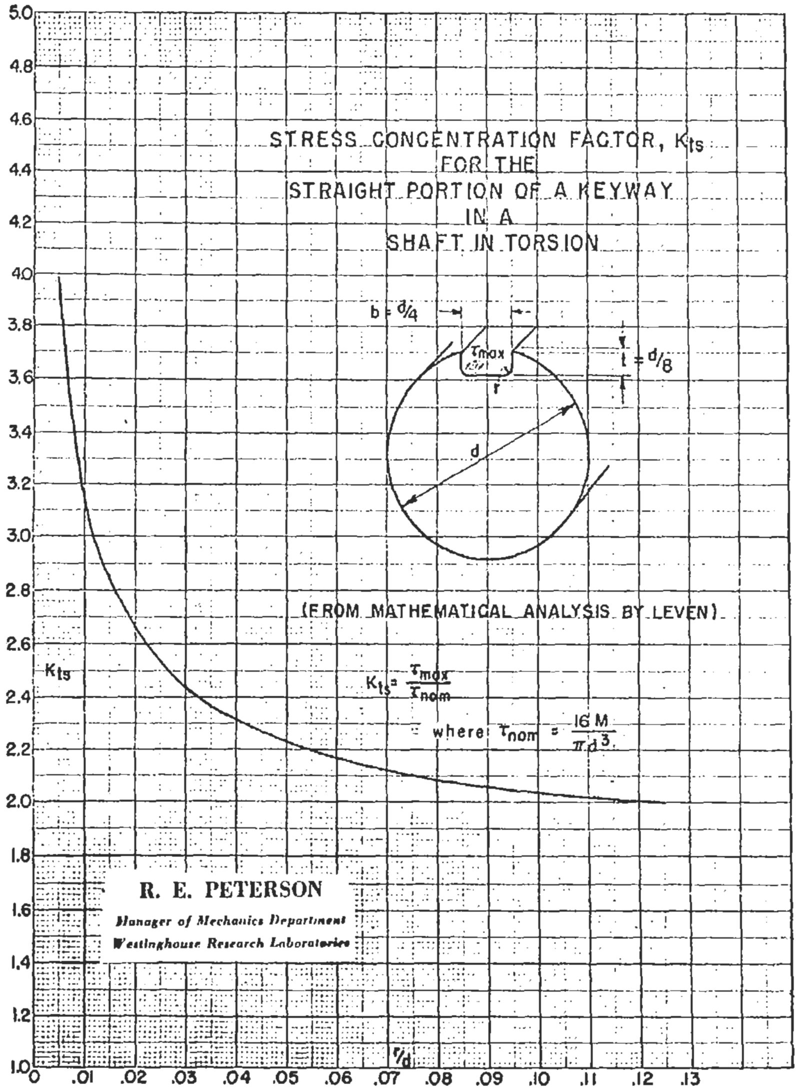


Figure 3-84. Stress concentration factors for keyed shafts.

This is the expression for the factor of safety in case of *uniaxial stresses.*

In the case of *combined stresses,* the equivalent stresses for σ_a and σ_m should be substituted in Equation 3-4, and Reference 42 shows how the equation for torsion reduces to Equation 3-5. Using $k_{f'}$, instead of k_f (obtained from Figure 3-85):

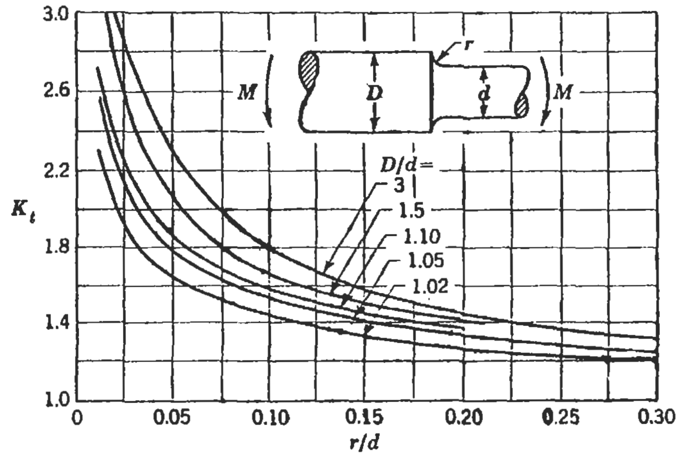


Figure 3-85. *Stress concentration factors for shafts in bending (Courtesy of Shigley, J. E., and Mischke, C. R.; Machine Design, McGraw-Hill, Inc., New York, 1986, p. 12.12.)*

$$n = \frac{1}{\sqrt{3}\left(k_{f'}\ \frac{\tau_a}{\sigma_E} + \frac{\tau_m}{\sigma_{yp}}\right)} \tag{3-5}$$

For *combined bending and torsion,* the factor of safety is obtained from

$$n = \frac{1}{\sqrt{\left(k_f\ \frac{\sigma_a}{\sigma_E} + \frac{\sigma_m}{\sigma_{yp}}\right)^2 + 3\left(k_{f'}\ \frac{\tau_a}{\sigma_E} + \frac{\tau_m}{\sigma_{yp}}\right)^2}} \tag{3-6}$$

If bending acts alone, τ_a and τ_m vanish and the expression for the factor of safety in the case of unilateral stresses results. When σ_a and σ_m vanish, we have the equation for n in torsion.

Torsional Stresses Can Be Readily Calculated

For the desired uprate conditions given earlier, we calculate shaft torque output as

$$T = \frac{(63{,}000)\ (19{,}600\ \text{HP})}{6{,}400\ \text{rpm}} = 193{,}000\ \text{lb - in}$$

The steady torsional stress is, therefore,

$$\tau_m = \frac{16T}{\pi d^3} = \frac{(16)(193{,}000)}{\pi(4.5)^3} = 10{,}900 \text{ psi}$$

Making the generally accepted assumption that the alternating torsional stress will not exceed 20% of the steady stress, we obtain

$$\tau_a = (0.2)(10{,}900) = 2{,}180 \text{ psi}$$

Bending Moments Assessed for Gear Coupling

There are three relevant bending moments* caused by a gear coupling when transmitting torque with angular or parallel misalignment:

- Moment caused by shift of contact point. This moment acts in the plane of angular misalignment and tends to straighten the coupling. It can be expressed as

  $$M_c = \left(\frac{T}{D_p/2}\right)\left(\frac{X}{2}\right)$$

  where T is the shaft torque, D_p is the gear-coupling pitch diameter and X is the length of the gear tooth face (see Figure 3-86).

- Moment caused by coupling friction. This moment acts in a plane at a right angle relative to the angular misalignment. It has the magnitude

  $$M_F = T\mu$$

  where μ is the coefficient of friction.
- Moment caused by turning torque through a misalignment angle. It acts in the same direction as the friction moment M_F and can be expressed as

  $$M_T = T \sin \alpha$$

The total moment is the vector sum of the individual moments:

$$M_{total} = \sqrt{M_c^2 + (M_F + M_T)^2}$$

A gear coupling suitable to transmit 19,600 HP at 6,400 rpm is assumed to have a pitch diameter $D_p = 9$ in and a face width of 1.3 in. Using a coefficient of friction $\mu = 0.3$

*An additional moment, due to overhung weight of coupling, is usually negligibly small and is omitted from the bending moment analysis for both gear and diaphragm couplings.

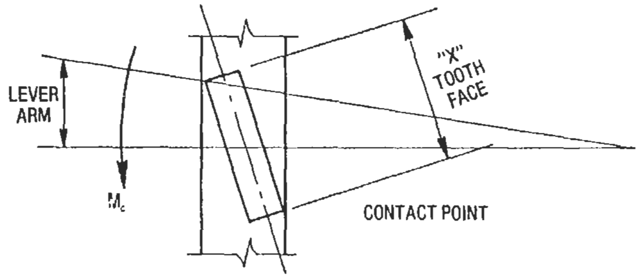


Figure 3-86. Shift in contact point experienced by gear-type coupling.

and a misalignment angle $\alpha = 0.057°$ (approximately 0.001 in/in parallel offset), we calculate

$$M_c = \left(\frac{193,000}{4.5}\right)\left(\frac{1.3}{2}\right) = 27,878 \text{ lb - in}$$

$$M_F = (193,000)\,(0.3) = 57,900 \text{ lb - in}$$

$$M_T = (193,000)\,(\sin 0.057) = 193 \text{ lb - in}$$

$$M_{Total} = \sqrt{(27,878)^2 + (57,900 + 193)^2} = 64,435 \text{ lb - in}$$

This moment is fixed relative to the actual angular misalignment α, and is seen by the shaft as a rotating bending moment at a once-per-revolution frequency.

The contoured diaphragm coupling causes two bending moments:

- Moment caused by angular misalignment which results in bending the diaphragm

  $$M_B = k_B\alpha$$

  In this expression, k_B equals the angular spring rate of the diaphragm (lb-in/degree); α is the misalignment angle. This moment acts in the plane of angular misalignment, as did M_c in the gear-coupling analysis.
- Moment caused by turning torque through a misalignment angle α. It can be expressed as

  $$M_T = T \sin \alpha$$

  The total moment is now

  $$M_{Total} = \sqrt{{M_B}^2 + {M_T}^2}$$

A suitable contoured diaphragm coupling has a diaphragm diameter of 16.5 in and an angular spring rate k_B = 18,800 lb-in/degree. Thus, for the misalignment angle used earlier,

$$M_B = (18{,}800)\,(0.057) = 1{,}080 \text{ lb - in}$$

$$M_T = (193{,}000)\,(\sin 0.057) = 193 \text{ lb - in}$$

$$M_{TOTAL} = \sqrt{(1{,}080)^2 + (193)^2} = 1{,}095 \text{ lb - in}$$

Alternating Stresses Compared

Comparing the bending moments caused by gear couplings with those resulting from contoured diaphragm couplings shows the former to be significant and the latter virtually negligible in comparison.

The cyclic bending stress imposed on a gear-coupling-equipped shaft can be computed from

$$\sigma_a = \frac{M_{total}\ C}{I}$$

where C and I are the shaft radius and shaft area moments of inertia, respectively. Thus

$$\sigma_a = \frac{(64{,}435)\,(2.25)}{\pi\, d^4 / 64}$$

$$= \frac{(64{,}435)\,(2.25)\,(64)}{\pi\,(4.5)^4} = 7{,}206 \text{ psi}$$

In addition, there is a mean tensile stress acting on the shaft cross-sectional area. This mean stress equates to

$$\sigma_m = \frac{T\mu}{(D_p / 2)\,(\pi C^2)\cos\Theta}$$

$$= \frac{(193{,}000)\,(0.3)}{(4.5)\,\pi\,(2.25)^2\,(0.94)} = 860 \text{ psi}$$

where $\Theta = 20°$, the pressure angle assumed for the gear teeth.

The cyclic bending stress seen by the diaphragm-coupling-equipped shaft can be obtained by a rapid ratio calculation:

$$\frac{\sigma_a \text{ (diaphragm coupling)}}{\sigma_a \text{(gear coupling)}} = \frac{M_{total} \text{ (diaphragm coupling)}}{M_{total} \text{ (gear coupling)}}$$

$$\sigma_a \text{ (diaphragm coupling)} = \frac{(1{,}095)\,(7{,}026)}{64{,}435} = 122 \text{ psi}$$

The mean tensile stress acting on the cross-sectional area of a diaphragm-coupling-equipped shaft depends on how far the diaphragm is displaced axially from its neutral rest position, and on the axial spring rate of the diaphragm. Assuming the diaphragm of this sample case was displaced by its maximum permissible distance of 0.100 in., it would exert a force of 1,950 lbs on the shaft cross-sectional area.[44] This would cause a mean stress

$$\sigma_m = \frac{1{,}950}{\pi C^2} = \frac{1{,}950}{\pi(2.25)^2} = 125 \text{ psi}$$

Shaft Factor of Safety Evaluated

Before actually calculating the shaft factors of safety for torque transmission with either coupling type, one must determine the stress concentration factors k_f and $k_{f'}$. Using values of r/D, D/d, and R/d from Figure 3-82, Reference 45 gives stress concentration factors $k_f = 1.95$, and Reference 43 gives $k_{f'} = 2.9$. The stress concentration factor k_f' results from the keyway and must be used in torsional stress calculations. Factor k_f takes into account the shaft step going from 4.5 in. to 5.0 in. diameters. It must be used in the bending stress calculation. Note also Figure 3-87,

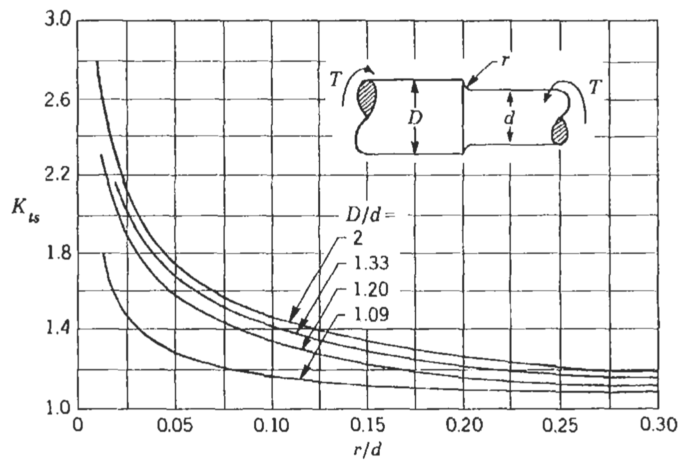


Figure 3-87. Stress concentration factors for shafts in torsion. (*Courtesy Shigley, J. E., and Mischke, C. R.; Machine Design, McGraw-Hill, Inc., New York, 1986, p. 12.12*)

which would have to be used for keyless shafts, and which is reproduced here for the sake of completeness.[45]

Since the gear-coupling-equipped shaft is loaded in torsion *and* potentially bending, we employ the expression for combined torsion and bending and obtain the factor of safety (from Equation 3-6):

$$n = \frac{1}{\sqrt{\left(k_f \frac{\sigma_a}{\sigma_E} + \frac{\sigma_m}{\sigma_{yp}}\right)^2 + 3\left(k_{f'} \frac{\tau_a}{\sigma_E} + \frac{\tau_m}{\sigma_{yp}}\right)^2}} \text{ Thus,}$$

$$n = \frac{1}{\sqrt{\left(1.95 \frac{7{,}206}{52{,}500} + \frac{860}{80{,}000}\right)^2 + 3\left(2.9 \frac{2{,}180}{52{,}500} + \frac{10{,}900}{80{,}000}\right)^2}} = 1.90$$

Using the same mathematical expression to calculate the factor of safety for the shaft equipped with the contoured diaphragm coupling:

$$n = \frac{1}{\sqrt{\left(1.95 \frac{122}{52{,}500} + \frac{125}{80{,}000}\right)^2 + 3\left(2.9 \frac{2{,}180}{52{,}500} + \frac{10{,}900}{80{,}000}\right)^2}} = 2.25$$

The value

$$\left(1.95 \frac{122}{52{,}500} + \frac{125}{80{,}000}\right)^2$$

in the previous expression is so small that applying the equation for pure torsion (Equation 3-5) would have given the same factor of safety, n = 2.25.

Shaft Still Adequate Under Uprate Horsepower Conditions

At this point, we may want to recall that the preceding analysis was made to determine whether the steam-turbine shaft would require replacement should the output horsepower be increased beyond the manufacturer's maximum permissible rating of 17,600 HP. Rather than engaging in a debate as to safe maximum absolute values of stress in turbine shafts, we merely investigated what factor of safety the shaft design embodied while equipped with a suitable gear coupling transmitting 17,600 HP at 6,400 rpm:

$$\text{Shaft torque } T = \frac{(63{,}000)\,(17{,}600)}{6{,}400} = 173{,}250 \text{ lb - in.}$$

$$\text{Steady torsional stress } \tau_m = \frac{16T}{\pi_d^{\,3}} = \frac{(16)\,(173{,}250)}{\pi(4.5)^3} = 9{,}680 \text{ psi}$$

Alternating torsional stress $\tau_a = (0.2)\,(9{,}680) = 1{,}940$ psi

$$M_C = \left(\frac{173{,}250}{4.5}\right)\left(\frac{1.3}{2}\right) = 25{,}023 \text{ lb - in.}$$

$$M_F = (173{,}250)\,(0.3) = 52{,}000 \text{ lb-in.}$$

$$M_T = (173{,}250)\,(\sin 0.057) = 173 \text{ lb-in.}$$

$$M_{total} = \sqrt{(25{,}025)^2 + (52{,}000 + 173)^2}$$

$$M_{total} = 57{,}864 \text{ lb-in.}$$

$$\sigma_a = \frac{(57{,}864)\,(2.25)\,64}{(\pi)\,(4.5)^4} = 6{,}472 \text{ psi}$$

$$\sigma_m = \frac{(173{,}250)\,(0.03)}{(4.5)\,\pi\,(2.25)^2\,0.94} = 770 \text{ psi}$$

$$n = \frac{1}{\sqrt{\left(1.95\,\dfrac{6{,}472}{52{,}500} + \dfrac{770}{80{,}000}\right)^2 + 3\left(2.9\,\dfrac{1{,}940}{52{,}500} + \dfrac{9{,}680}{80{,}000}\right)^2}}$$

$$= 2.14$$

The shaft factor of safety, while certainly adequate for operation at 17,600 hp with gear-type couplings, was thus shown to be only 2.14 in the original design case with conventional gear couplings. Equipping the same shaft with contoured diaphragm couplings made operation at 19,600 hp not only possible, but actually increased the factor of safety to 2.25.[46]

Reliable Shaft-Hub Connections for Turbomachinery Couplings

Coupling hubs for turbocompressors must fit very tightly on compressor shafts. Not only does the interference fit have to be high enough to prevent slipping at maximum applied torque, but potentially serious weakening of shafts due to fretting action must also be avoided. The term "fretting" describes component damage that occurs at the interface between contacting, highly loaded metal surfaces when subjected to extremely small relative (or vibratory) motion. On the other hand, the coupling hub must be designed for easy removal in case rapid access to compressor-shaft seals should become necessary.

Satisfying both of these requirements does not generally present any serious problems for equipment shafts up to approximately three inches nominal diameter. It does, however, require progressively more attention and design sophistication as shaft sizes reach eight or more inches on cracked gas and propylene compressors in modern ethylene plants.

Keyless coupling-hub engagement is a logical choice for many of these applications. Hubs up to about three inches nominal bore can be effectively mounted by conventional heat-shrink methods, while hubs with larger bore diameters are probably best suited for hydraulic dilation fit-up.

Torsional Holding Requirement Must Be Defined

There is no uniformly accepted design practice governing either the fit-up integrity or torsional holding requirement of coupling hubs on equipment shafts. One school of thought opts for interference dimensions which will ensure slip-free transmission of rated turbocompressor torque. Others believe that the interference-fit dimensions and, in some cases, the entire coupling design should allow safe transmission of the maximum allowable torque value for a given shaft material and nominal shaft diameter. The two approaches are illustrated in Figure 3-88.

Graphically representing the mean torsional stress τ_m as a function of torque T,

$$\tau_m = \frac{16T}{\pi d^3} \tag{3-7}$$

Figure 3-88 also shows typically accepted maximum allowable torque values for various shaft diameters made of the more common turbomachinery shaft steels, AISI 1040, 4140, and 4340. Assuming a turbocompressor absorbing 7,000 hp at 4,500 rpm (155 hp/100 rpm) were to incorporate a four-inch diameter, AISI 4140 shaft, its rated torque would be

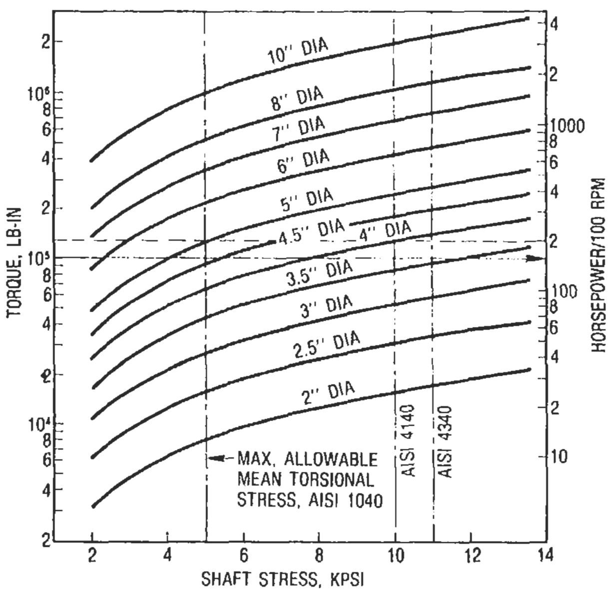


Figure 3-88. Shaft torque versus stress for typical shaft sizes.

$$T = \frac{63{,}000 \times 7{,}000}{4{,}500} = 98{,}000 \text{ lb-in.}$$

while its maximum allowable torque value would be

$$T = \frac{\tau_m \pi d^3}{16} = 125{,}664 \text{ lb-in.}$$

or about 199 hp/100 rpm. The maximum allowable torque expression in this case uses τ_m = 10,000 psi, the generally accepted "safe allowable" torsional mean stress for AISI 4140.

The more conservative approach would be to select a coupling for the higher of the two torque values. The hub fit-up dimensions would then logically be chosen to allow slip-free transmission of the higher torque.

Finally, it should be recognized that there is a certain degree of design conservatism in calculating maximum allowable torque from Equation 3-7 and using 5,000 psi, 10,000 psi, and 11,000 psi as maximum allowable torsional mean stresses for AISI 1040, 4140, and 4340 steels, respectively. This conservatism may vanish if the combined action of stress concentrations at fillets, superimposed alternating torsional stresses, or alternating bending stresses from inadequate coupling designs should cause the combined stresses to go 30% + over the maximum allowable values. In such cases a Soderberg analysis may be used to more closely establish shaft factors of safety.[42,46]

Torsional Holding Ability Can Be Calculated

The torque required to cause complete slippage of a press fit is given by

$$T = \left(\frac{\pi\mu}{2} p\right) L\, d^2 \qquad (3\text{-}8)$$

where μ is the coefficient of friction, p the unit press-fit pressure between shaft and hub, L the length of the hub bore, and d the nominal shaft diameter.[47]

Using the widely accepted average value of 0.12 for the coefficient of friction,[47,48] Figure 3-89 was plotted to show the relationship between torsional holding ability (or requirement), press-fit pressures, and coupling-bore dimensions. The press-fit pressures refer to ratios of shaft diameter over hub diameter d/D, and interference fits. Figure 3-90 illustrates this relationship which is based on the mathematical expression for press-fit contact pressures of steel hubs on solid steel shafts:

$$P = \frac{eE\,(D^2 - d^2)}{2dD^2} \qquad (3\text{-}9)$$

In this formula, e represents the *total* diametral interference and E is the modulus of elasticity for steel.[49] Alternatively, contact pressure could be expressed as a function of the interference rate i. This rate equals the diametral interference divided by the shaft diameter; e.g., in./in. or mm/mm:

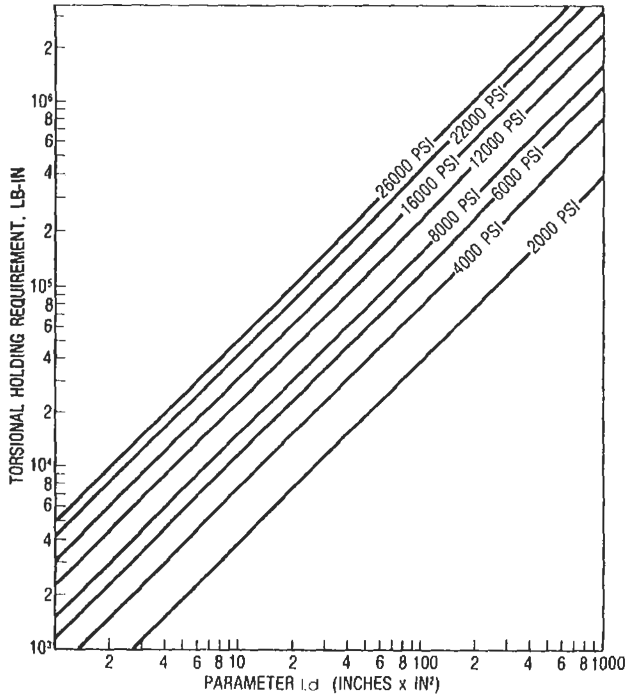


Figure 3-89. Torsional holding ability at various interference-fit pressures with a given Ld2 factor.

$$P = \frac{iE\,(D^2 - d^2)}{2D^2} \qquad (3\text{-}10)$$

The deformations of the hub will remain elastic until the maximum equivalent tensile stress (equal to twice the maximum shear stress at the bore of the hub) becomes greater than the yield strength of the hub material in tension. This maximum equivalent tensile stress in the hub simply equates to modulus of elasticity times interference fit per inch of shaft diameter. In other words, maximum equivalent tensile stress is 30,000 psi if the interference fit is 0.001 inch per inch, 60,000 psi if the fit is 0.002 inch per inch, etc. Experimentation by Werth[50] has shown that for steel hubs of essentially plain cylindrical configuration, both the mounting pressure and holding force continue to increase up to a limiting fit value of 0.003 inch per inch of shaft diameter. Limiting the maximum equivalent tensile stress to the yield strength of the hub material is thus considered to be very conservative for plain cylindrical hubs with heavy wall thickness.

The required interference fit is generally achieved by expanding the coupling hub and then sliding it over the shaft. An alternative to heating the hub would be to cool the shaft by means of a coolant such as dry ice. However, since the shaft cooling method is rarely used in field situations, we can confine our analysis to the three prominent practical methods: thermal expansion, hydraulic dilation, and friction fit.

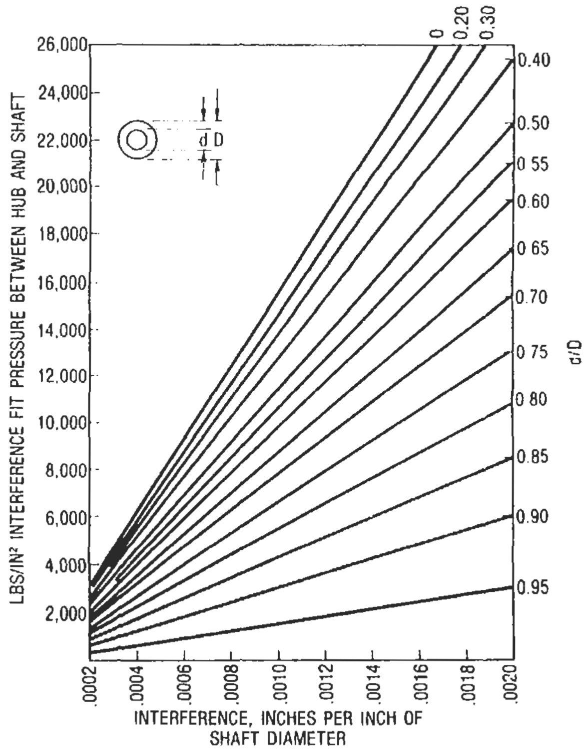


Figure 3-90. Interference fits required for various shaft/hub diameter ratios to reach required interference-fit pressures.

Maximum Combined Stresses

Experience shows that highly satisfactory interference fits are generally reached when the material at the hub bore reaches about 80% of the yield stress. The maximum combined stress is composed of compressive and tensile components and, for a conventional coupling hub, occurs at the bore. Its value can be expressed as:

$$\sigma = \frac{P\sqrt{3 + c^4}}{1 - c^2} \text{ lb / in.}^2 \qquad (3\text{-}11)$$

where c = d/D. Studies have indicated that coupling hubs with c values of 0.4 or lower will operate quite well even if the full yield stress value is reached by the maximum combined stress at the coupling bore.[49]

Hub Mounting by Thermal Expansion

Thermal expansion of hubs is accomplished through application of heat. Immersion in high flashpoint heat-transfer oil, heating in an oven atmosphere, or soaking in steam are customary methods. The temperature required to achieve a given amount of expansion can be calculated from:

$$\Delta T = \frac{e}{\alpha d} \tag{3-12}$$

where e = diametral expansion (interference) in inches; a = coefficient of linear expansion, inch per inch per °F; and d = initial diameter of the bore (before expansion) in inches. Figure 3-86 represents this relationship graphically, with α assumed to be 6.3×10^{-6} inch per inch. It should be noted that the ΔT values of Figure 3-91 must be added to the ambient temperature of the shaft when fitting a hub. Also, ease of assembly may make it desirable to thermally expand the hub beyond the theoretically required expansion value (solid line) to an "easy assembly" value (dotted line). Our earlier example might illustrate this point.

Here, a four-inch diameter AISI shaft transmits a maximum permissible torque of 125,664 lb-in. This torsional holding requirement is to be achieved with a coupling having a $Ld^2 = 56$. According to Figure 3-89, the required press-fit pressure is 12,000 psi. Assuming further that the coupling d/D ratio is 0.68, Figure 3-90 shows a

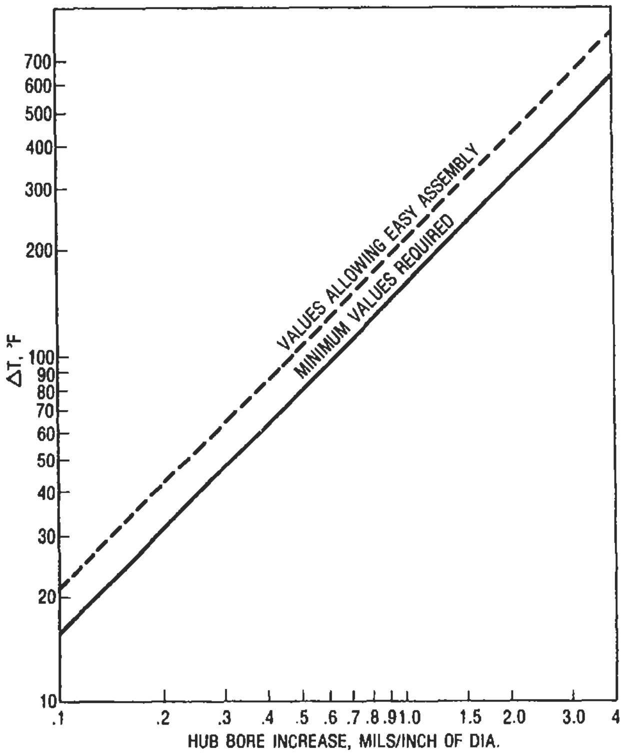


Figure 3-91. Temperature change above ambient required to increase hub bore.

required interference of 0.0015 inch per inch of shaft diameter. While the solid line in Figure 3-91 would now tell us that the hub could expand this required amount by heating it to a temperature 238°F above ambient, we may actually want to facilitate the hub attachment operation by expanding it another 2 mils, or alternatively, say 0.002 inch. Thus the hub should expand 0.008 mil on a 4-inch shaft, or 2 mils per inch of diameter. We would therefore, choose to enter the abscissa of Figure 3-91 at 2.0 mils/inch and find that the hub should be heated to a temperature 318°F above ambient. The same result can be obtained by entering the abscissa at 1.5 mils/inch and reading off the ΔT value at the dotted-line intersection.

Disassembly Requirements for Thermally Mounted Hubs

The maximum axial force F required to disassemble shrink fits varies directly as the thickness of the outer member, or hub, the length of the outer member, the difference in diameters of the mating members, and the coefficient of friction.[48] This force in pounds may be approximated by:

$$F = \mu \pi dLp \tag{3-13}$$

and is graphically shown on Figure 3-92 as a function of press-fit pressure p and parameter dL. A considerably higher coefficient of friction, $\mu = 0.24$, is assumed to ensure that pull-off devices are designed or selected for maximum anticipated forces. Experience shows that coupling hubs up to perhaps three inches nominal bore can be

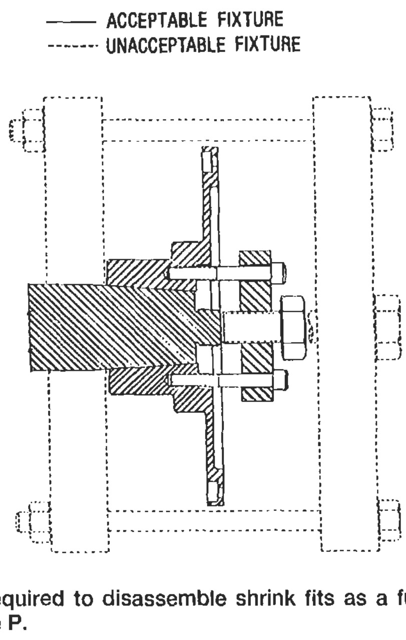


Figure 3-92. Force required to disassemble shrink fits as a function of parameter dL and press-fit pressure P.

removed using mechanical pull-off devices similar to the one shown in Figure 3-93, without requiring the application of heat in most cases.

However, it is critical to relate the force requirement F to the proof stresses of the pull-off bolts connecting fixture and coupling hub. At least four alloy bolts are required in most applications. The large conventional center bolt in this fixture should actually be used to remove the coupling hub from the shaft. Note also that only the solid outline in Figure 3-93 represents an acceptable mechanical pull-off fixture. Efforts to shortcut this procedure by making a simple fixture similar to the one shown in dotted outline form incur serious risk of skewing. Skewing, in turn, not only tends to overstress bolts, but can lead to galling of shaft and bore surfaces.

The hydraulic pull-off fixture of Figure 3-94 accomplishes hub removal under controlled conditions of pressure, and thus force application. The applied force can easily be matched or limited to the combined proof strength of the alloy steel pull-off bolts. Figure 3-95, representing a hydraulic mounting fixture, is the logical companion device to the hydraulic pull-off fixture of the preceding figure. Hydraulic mounting is rapid, easily controllable, and quite appropriate at installations which do not have the hub heating facilities outlined earlier.

Hydraulic Hub Dilation. By far the most sophisticated and appropriate method of mounting and removing large coupling hubs from turbomachinery shafts involve hub dilation by introducing suitable hydraulic fluids into the interface between shaft sur-

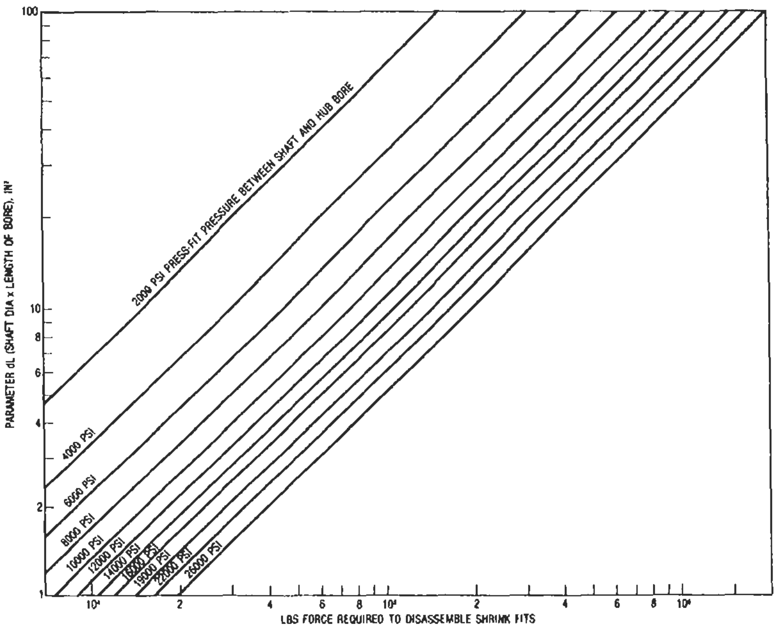
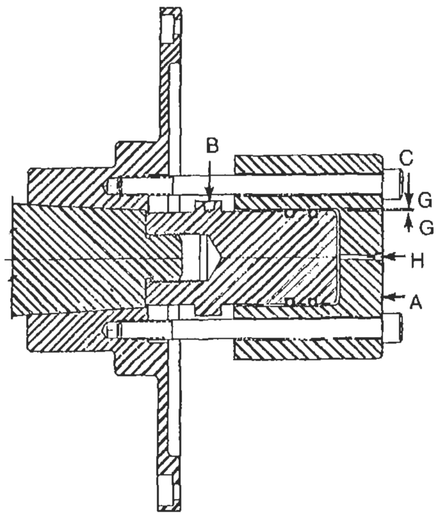


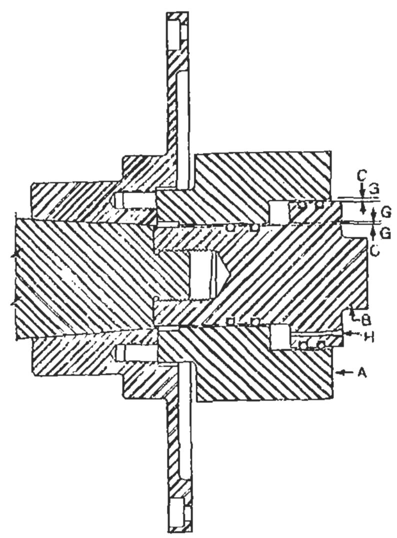
Figure 3-93. Mechanical pull-off for tapered-bore coupling hubs.[40]



Notes:

1. Surfaces marked G to be ground and polished.
2. Diametral clearance C to be 0.001 ± 0.0005 inch.
3. Location H to accept 30,000 psi hydraulic fitting.
4. O-rings to be Buna-N 90 durometer.
5. Drill 4 spanner wrench holes on circumference B.
6. Outer cup A to have wall thickness consistent with 30,000 psi pressure retention requirement.
7. Outer cup A to be AISI 4340 steel.

Figure 3-94. Hydraulic pull-off fixture for tapered-bore coupling hubs.



Notes:

1. Surfaces marked G to be ground and polished.
2. Diametral clearance C to be 0.001 ± 0.0005 inch.
3. Location H to accept 30,000 psi hydraulic fitting.
4. O-rings to be Buna-N 90 durometer.
5. Machine,wrench flat at B.
6. Outer cup A to have wall thickness consistent with 30,000 psi pressure retention requirement.
7. Outer cup A to be AISI 4340 steel.

Figure 3-95. Hydraulic mounting fixture for tapered-bore coupling hubs.

face and coupling-hub bore. This can be accomplished by introducing the pressurized fluid through a transverse passage drilled into the coupling hub.

Either a combination assembly/disassembly fixture, as shown in Figure 3-96, or a pusher assembly in conjunction with hydraulic fluid injection through the shaft center (Figure 3-97) can be used to mount and dismount the coupling hub.

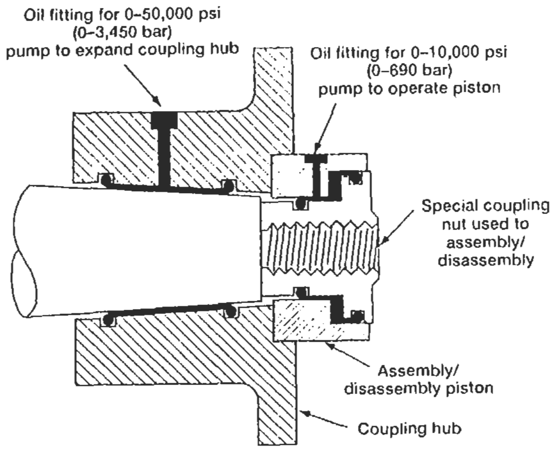


Figure 3-96. Hydraulic hub dilation by direct injection.

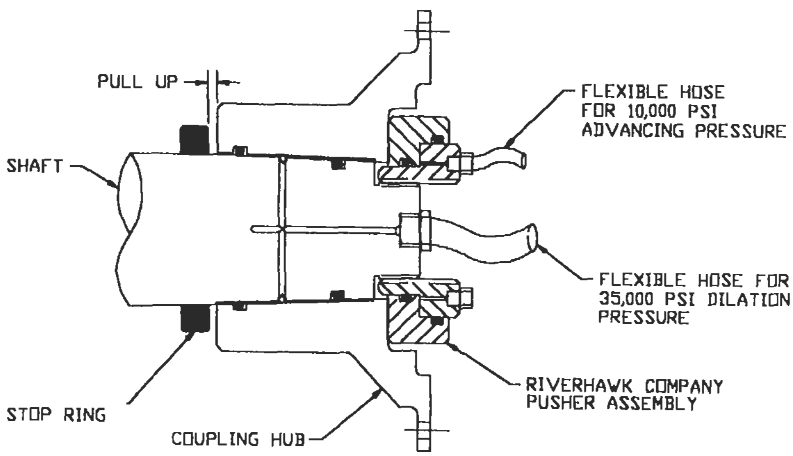


Figure 3-97. Hydraulic hub dilation by injection through shaft. (*Courtesy of Riverhawk Company, Utica, New York.*)

Two hand-operated, high-pressure rams are required for the mounting operation. One of these applies a high axial force to the assembly sleeve, while the second one forces its separate supply of hydraulic fluid into the shaft-coupling bore interface.

In the early 1970s, coupling hubs and shaft ends intended for hydraulic dilation were often furnished with O-rings. More recently, it has been shown that accurately machined mating tapers no longer require O-rings for satisfactory mounting and dismounting of most commonly applied turbomachinery couplings.

For cylindrical shaft ends, which are still occasionally found on older equipment, the reliability professional might consider Coupling Corporation of America's "Anderson Hub Clamp." This patented device (Figure 3-98) is designed to accommodate most shafts, especially those with somewhat questionable surface smoothness. On retrofits, users have had particular success on straight, tapered, keyed and keyless shafts.

The system is based on an asymmetrical profile thread into the outside diameter of clamp hub flange (1) and the bore of clamp hub flange (2). When the installer torques the loading screws (3), a predetermined uniform force pushes the sleeve away from the flange. The special asymmetrical threads cause this axial force to exert a clamping action radially inward on the split sleeve portion of the clamp hub flange. It can be shown that the gripping action is sufficient to prevent slippage on virtually any machine shaft.

Conclusion

The design approaches outlined here represent proven methods of reliable shaft-hub connections. Although very tight fit is essential if the risk of potentially dangerous fretting action is to be avoided, it has been shown that this requirement does not have to be incompatible with easy hub removal in case rapid access to turbomachinery shaft seals should become necessary.

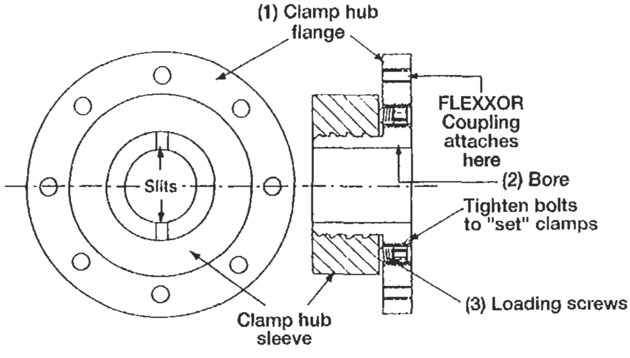


Figure 3-98. Anderson hub clamp for attaching coupling hubs to cylindrical shaft ends.

How to Keep Track of Reliability Review Tasks

As mentioned earlier, machinery reliability reviews are aimed at ensuring compliance with all of the specification clauses invoked by the purchaser. In addition, the reviewing engineer uses his own background and experience to determine whether the proposed design, layout, construction features, etc. meet the long-term reliability criteria established by the purchaser.

To assist him in executing these tasks, the reviewing engineer generally instructs the contractor or equipment manufacturer to submit drawings and other data for his review. Many of the vendor data and drawing requirements are tabulated in the appendixes of applicable API standards and can be adapted to serve as checklists for the task at hand. Other checklists may have to be derived from the reviewing engineer's experience.

Keeping track of the status of your documentation reviews is best accomplished by first listing the vendor drawing and data requirements for a given machinery category. Figures 3-99 through 3-103 represent these listings for the most prominent machinery categories encountered in typical petrochemical process plants. Whenever possible, the listings should contain the data requirements of available API standards. For instance, Figure 3-99 is derived from API Standard 617; the review engineer may wish to use the versions found in the various current and applicable editions of API Standards.

Each "tracking sheet" is supplemented by three columns in which the reviewer can enter the appraisal status (e.g., "Preliminary Review Completed," "Comments Forwarded," and "Final (Corrected) Copy Reviewed"). Also, the "tracking sheet" shows seven columns which indicate a particular job phase during which the vendor or contractor is expected to submit certain drawings or analytical data for the equipment owner's review. The decision as to when (i.e., during which job phase) data are to be submitted is best made by mutual agreement of all parties involved.

We have reproduced reliability review checklists listing generalized rotating machinery data review requirements for centrifugal compressors (Figure 3-99), special-purpose steam turbines (Figure 3-100), centrifugal pumps (Figure 3-101), cooling tower fan systems (Figure 3-102), and forced and induced draft fans (Figure 3-103).

Machinery Reliability Audits for Existing Plants

Experience shows that even a mature operating plant will benefit from periodic reliability audits. As can be expected, machinery reliability audits look for factors that *could have,* or in some cases already *have had,* an adverse impact on machinery reliability, and therefore plant profitability. Machinery reliability audits differ from conventional safety and plant operability audits by emphasizing the machinery-related aspects of plant operation, organization, maintenance, troubleshooting, planning, training, and other related functions. For example, individuals conducting a conventional plant audit might simply review material control procedures by verifying the existence of a parts and materials coding system; whereas, in a machinery reliability audit, specific designation of critical spare parts would be recommended by, perhaps,

Project Title ____________ Date Ordered ____________
Project No ____________ Item No. ____________
Purchase Order No. ____________
Location ____________ Unit ____________ Inquiry No. ____________
Service ____________ Date ____________

Submit to Owner Before:

Preliminary Review	Comments	Final Copy	Description	● P.O. Issue	● Coordination Meeting	● Shop Test	● Shipment	● Field Erection	● Field Test	● Plant Startup		
			1. Certified dimensional outline drawing and list of connections.									
			2. Cross-sectional drawing and bill of materials.									
			3. Rotor assembly drawing and bill of materials.									
			4. Thrust-bearing assembly drawing and bill of materials.									
			5. Journal-bearing assembly drawing and bill of materials.									
			6 Seal assembly drawing and bill of materials.									
			7. Coupling assembly drawing and bill of materials.									
			8. Seal-oil schematic and bill of materials.									
			9 Seal-oil assembly drawing and list of connections.									
			10. Seal-oil component drawing and data.									
			11. Lube-oil schematic and bill of materials.									
			12. Lube-oil assembly drawing and list of connections.									
			13. Lube-oil component drawing and data.									
			14. Electrical and instrumentation schematics and bill of materials.									
			15 Electrical and instrumentation arrangement drawing and list of connections.									
			16. Polytropic head and efficiency versus icfm.									
			17 Discharge pressure and bhp versus icfm.									
			18. Balance line p versus thrust load.									
			19. Speed versus starting torque.									
			20. Vibration analysis data.									
			21. Lateral critical analysis.									
			22. Torsional critical analysis.									
			23. Transient torsional analysis.									
			24. Allowable flange loading.									
			25. Alignment diagram.									
			26. Weld procedures.									
			27. Hydrostatic test logs.									
			28. Mechanical run test logs.									
			29. Rotor balance logs.									
			30. Rotor mechanical and electrical runout.									
			31. "As-built" data sheets.									
			32. "As-built" dimensions and data.									
			33. Operating and maintenance manuals									
			34 Spare parts recommendation and price list.									

Figure 3-99. Centrifugal compressor vendor drawing and data requirements.

assigning the parts identifying code letters. This would assure that whenever a coded part were to be ordered, both the purchasing department and warehouse receiving personnel would be notified of the special inspection requirements needed to assure machine reliability.

Machinery reliability audits should be conducted by mechanical engineers with extensive design and plant machinery experience. These audits are generally structured to:

Project Title ______ Date Ordered ______
Project No ______ Item No ______
Purchase Order No. ______
Location ______ Unit ______ Inquiry No. ______
Service ______ Date ______

Submit to Owner Before:

Preliminary Review	Comments	Final Copy	Description	● P.O. Issue	● Coordination Meeting	● Shop Test	● Shipment	● Field Erection	● Field Test	● Plant Startup		
			1. Certified dimensional outline drawing and list of connections.									
			2. Cross-sectional drawing and bill of materials.									
			3. Rotor assembly drawing and bill of materials.									
			4. Thrust-bearing assembly drawing and bill of materials.									
			5. Journal-bearing assembly drawing and bill of materials.									
			6. Packing and labyrinth drawings and bill of materials.									
			7. Coupling assembly drawing and bill of materials.									
			8. Gland sealing and leakoff schematic and bill of materials.									
			9. Gland sealing and leakoff arrangement drawing and list of connections.									
			10. Gland sealing and leakoff component drawings and data.									
			11. Lube-oil schematic and bill of materials.									
			12. Lube-oil arrangement drawing and list of connections.									
			13. Lube-oil component drawings and data.									
			14. Electrical and instrumentation schematics and bill of materials.									
			15. Electrical and instrumentation arrangement drawing and list of connections.									
			16. Control and trip system.									
			17. Governor details.									
			18. Steam flow versus horsepower.									
			19. Steam flow versus first-stage pressure.									
			20. Steam flow versus speed and efficiency.									
			21. Steam flow versus thrust-bearing load.									
			22. Extraction performance curves.									
			23. Steam correction charts.									
			24. Vibration analysis data.									
			25. Lateral critical analysis.									
			26. Allowable flange loading.									
			27. Alignment diagram									
			28. Weld procedures.									
			29. Hydrostatic test logs									
			30. Mechanical run test logs.									
			31. Rotor balance logs.									
			32. Rotor mechanical and electrical runout.									
			33. "As-built" data sheets.									
			34. "As-built" dimensions.									
			35. Operating and maintenance manuals.									
			36. Spare parts recommendation and price list.									

Figure 3-100. Special purpose steam turbine vendor drawing and data requirements.

- Review an installation for existing or potential deficiencies in equipment and auxiliaries in order to point out solutions or steps that will lead to definite improvement.
- Survey the adequacy of operating and training procedures at a given plant site.

Project Title ______ Date Ordered ______
Project No ______ Item No ______
Purchase Order No ______
Location ______ Unit ______ Inquiry No ______
Service ______ Date ______

Submit to Owner Before:

Preliminary Review	Comments	Final Copy	Description	P.O. Issue	Coordination Meeting	Shop Test	Shipment	Field Erection	Field Test	Plant Startup		
			1. Certified dimensional outline drawing and list of connections									
			2. Dimensionally accurate cross-sectional drawing and bill of materials.									
			3. Rotor assembly drawing and bill of materials.									
			4. Thrust-bearing assembly drawing with axial float and rated capacity indication.									
			5. Dimensions on computerized records data input form.									
			6. API spec sheet.									
			7. Performance curve.									
			8. Mechanical-seal drawing									
			9. Tandem seal auxiliary documentation									
			10. Installation, operating, and maintenance manual.									
			11. Journal-bearing assembly drawing and bill of materials									
			12. Coupling assembly drawing and bill of materials									
			13. Lube-oil arrangement drawing and list of connections.									
			14. Oil-mist component drawing and data.									
			15. Lube-oil schematics and bill of materials if circulating or pressure lube system.									
			16. Electrical and instrumentation schematics and bill of materials									
			17. Electrical and instrumentation arrangement drawing and list of connections, including tandem-seal auxiliaries									
			18. Control and trip system, if auto start stop.									
			19. Governor details, if steam-turbine driven.									
			20. Vibration analysis data, if multistage pump									
			21. Critical speed data, if vertical-column pump.									
			22. Suction specific speed calculation.									
			23. Diagram of shaft deflection versus flow									
			24. Diagram of axial thrust versus flow.									
			25. Allowable flange loading, if other than API 610.									
			26. Alignment recommendations.									
			27. Hydrostatic test logs.									
			29. Rotor-balance logs									
			30. Rotor mechanical and electrical runout									
			31. "As built" data sheets									
			32. "As built" dimensions.									
			33. Spare parts recommendation and price list									

Figure 3-101. Centrifugal pump vendor drawing and data requirements.

- Examine spare parts purchase, storage, and quality control procedures; including OEM vs. NON-OEM procurement practices.
- Outline steps, merits, and cost of resolving problems and deficiencies that could, if unaddressed, have an adverse impact upon plant operation.
- Appraise upgrade opportunities and define cost vs. short- and long-term benefits.
- Review the extent and sufficiency of root-cause failure analysis practiced at the plant.

Project Title ______ Date Ordered ______
Project No ______ Item No ______
Purchase Order No ______
Location ______ Unit ______ Inquiry No ______
Service ______ Date ______

Submit to Owner Before:

Preliminary Review	Comments	Final Copy	Description	P.O. Issue	Coordination Meeting	Shop Test	Shipment	Field Erection	Field Test	Plant Startup		
			1. Certified dimensional outline drawing and list of connections for gear-speed reducer.									
			2 Same, for motor									
			3. Gear cross-sectional drawing and bill of materials									
			4 Gear-strength rating and service-factor data.									
			5. Fan hub cross-sectional drawing									
			6. Fan blade cross-sectional drawing									
			7. Fan blade core weld procedure.									
			8 Fan blade static resonant frequency									
			9. Fan blade dynamic resonant frequency									
			10. Fan blade angle versus absorbed power plot.									
			11. Fan balancing procedure.									
			12. Fan drive system torsional critical speed data									
			13. Gear, motor, and line-shaft bearing lube data.									
			14. Coupling rating, service factor, material									
			15. Lube-oil arrangement drawing and list of connections, including oil-mist application.									
			16 Vibration cutout switch technical data weatherproofing, calibration, mounting testing, schematics									
			17. Alignment instructions									
			18 Motor electrical schematics and bill of materials.									
			19 Operating and maintenance manuals									
			20 Vibration spectrum identification data.									
			21. Spare parts recommendation and parts list									

Figure 3-102. Cooling tower fan system vendor drawing and data requirements.

- Recommend the most cost-effective methods and frequencies of machinery condition monitoring.
- Perform any other investigations, as determined during the audit, aimed at uncovering existing or potential impediments to achieving reliable equipment performance.

Methods and Scope*

Methods, scope, and typical details of a thorough machinery reliability audit involve looking closely at the various hardware systems, work processes, procedures, and organizational lineups in place at a given facility. To allow efficient transfer of machinery technology, plant management should designate one of their engineers to work with the audit team.

At well-managed, modern plants, the emphasis should be on identifying practices and procedures that could have an adverse future impact on short- and long-term

*Adapted from Bloch/Muller NPRA paper MC-89-74, National Petroleum Refiners Association Maintenance Conference, San Antonio, Texas, 1989.

Project Title ______________ Date Ordered ______________
Project No ______________ Item No ______________
Purchase Order No ______________
Location ______________ Unit ______________ Inquiry No ______________
Service ______________ Date ______________

Submit to Owner Before:

Preliminary Review	Comments	Final Copy	Description	P.O. Issue	Coordination Meeting	Shop Test	Shipment	Field Erection	Field Test	Plant Startup		
			1 Certified dimensional outline drawing and list of connections									
			2 Cross-sectional drawing and bill of materials									
			3 Rotor assembly drawing and bill of materials									
			4. Thrust-bearing assembly drawing and bill of materials.									
			5 Journal-bearing assembly drawing and bill of materials									
			6 Packing and labyrinth drawing and bill of materials									
			7. Coupling assembly drawing and bill of materials									
			8 Lube-oil schematic and bill of materials.									
			9 Lube-oil arrangement drawing and list of connections.									
			10 Lube-oil component drawings and data									
			11. Vibration analysis data (spectrum identification)									
			12 Lateral critical analysis									
			13 Allowable flange loading									
			14 Alignment diagram									
			15 Weld procedures									
			16 Slow-roll gear cross-sectional data and bill of materials									
			17. Gear-strength rating and service-factor data									
			18 Overrunning-clutch technical data									
			19. Damper linkage and actuation details									
			20. Damper-linkage lubrication data									
			21 Performance curve									
			22 Mechanical run test logs									
			23 Rotor balance logs									
			24. Rotor mechanical and electrical runout.									
			25 "As built" data sheets									
			26 "As built" dimensions, indicated on a diagram									
			27. Operating and maintenance manuals									
			28 Spare parts recommendation and price list									

Figure 3-103. Forced/induced draft fans vendor drawing and data requirements.

reliability. While an audit charter does not normally include the troubleshooting of specific problems, a cross-section of existing problems normally is investigated in order to evaluate the diagnostic procedures used by personnel at a given complex.

The reliability audit typically concentrates on the following:

- Site surveys are performed of all critical machines, their auxiliary equipment and instrumentation.
- Reviews of operating procedures are conducted.
- Organizational relationships and their impact on equipment reliability are reviewed.
- Discussions are being held with various levels of plant personnel involved in maintenance, operations, and technical support functions.
- Reviews are performed of the following:
  -Protective system testing methods
  -Shop and stores facilities
  -Shop repair procedures
  -Lube oil testing routines

-Machinery surveillance practices
-Turnaround, incident, and failure reports
-Log sheets

While some problems may be specific to certain process units, the emphasis is often on identifying reliability factors common to all sites that are part of a plant or complex.

To assure that the observations are consistent and accurate, meetings should be held at the conclusion of the audit at each site to enable the various participants to hear and question the assessments of the audit team. This is deemed by plant personnel to be an important feature of the audit process.

Major Machinery Best Reviewed by Using Checklists

The field audit of a given machine explores relevant deviations and existing, as well as potential, vulnerabilities in great detail. Consequently, the audit effort is greatly facilitated by checklists that are structured to uncover problems in many areas. Problem areas often include poor machine-to-operator interface, instrumentation and surveillance-related deficiencies, insufficient instructions and procedures, or perhaps components and machine auxiliaries that do not represent state-of-the-art technology and mandate frequent precautionary shutdowns. An example of a detailed machinery review checklist is shown in Figure 3-104.

Interviews Prove Informative

Many of the audit findings and recommendations will be the result of responses to questions raised by the audit team. It will always be helpful to schedule interviews with a "diagonal slice" through the organization chart, i.e., personnel representing various job levels in the operations, maintenance, and technical workforce. Their descriptions of their actual or perceived roles, observations, concerns, preferences, etc., hold important clues that are evaluated by the audit team. In all cases, the interview process will provide a strong indication of the degree of knowledge of the person being interviewed and the extent of interdepartmental cooperation practiced at the plant.

The author's experience in performing many audits indicates that a good "diagonal slice" through a plant organization should include process operators, operations supervision, machinists or mechanics, planners, maintenance supervision, process and mechanical/machinery engineers, technical services supervision, and warehouse or stores supervision.

The actual audit interview process generally follows the items described in Figure 3-105. By the time the interview process is complete, the audit team generally has a very good idea as to who is responsible for what, how the various groups cooperate with each other, how well the plant pursues true root-cause analysis of machinery failures and other interpersonal factors. A great deal of insight is also gained in reviewing problem response time, resource utilization, technical training, and plant safety. Again, a number of checklists are used by the audit team to ascertain that none of the many important factors contributing to machinery reliability are being overlooked. Figure 3-106 represents one of these checklists.

Figure 3-104. Simplified audit checklist for steam turbine driven centrifugal compressor.

Unit: C-1-22
Date: 8/15/98
Reviewer: ______________

Location and Activity

Control Room

1. Determine deviations in operating data: Review log sheets and compare trends.
2. Do operators know of problems or have specific concerns in areas of machine performance or mechanical condition?
3. Check all compressor instrumentation in the control house and log all data relating to compressors. Compare to previous data. If changes are significant, discuss with process personnel.
   (a) Panel alarms—Note and question operators on any compressor alarms.
   (b) Record and compare:
      (1) Vibration levels
      (2) Alert and alarm set points: Are these set at the correct concern levels?
      (3) Monitor O.K. lights
      (4) Gap voltage
      (5) Flows
      (6) Pressures
      (7) Temperatures

On the compressor platform, check the following:

4. Panel alarms. Note and discuss with operators if alarms are activated. Verify if alarms are real or false.
5. Record and analyze
   (a) Vibration levels
   (b) Alert and alarm set
   (c) O.K. lights on monitors
   (d) Gap voltages for bearing problems
6. Local panel
   (a) Record and compare turbine speed, steam flow, pressures, temperatures, exhaust temperature and pressure.
   (b) Temperature recorder
      (1) Is recorder working? If not, request that work order be initiated.
      (2) Record and analyze all available temperatures (thrust bearings, journal bearings, etc.)
7. Walk around the turbine and compressor and do the following:
   (a) Log all local temperatures and steam pressures.
   (b) Record and analyze all temperatures and pressures on local gauges for lube and seal systems and for bearings and seals on turbine and compressor.
   (c) Check governor for proper oil level.
   (d) Is governor controlling at steady speed?
   (e) Any unusual vibration in trip throttle valve?
   (f) Check inlet steam valves. Any unusual position on cams?
   (g) Check linkage, extraction valves, etc., to make sure no pins are backing out, cotter keys broken or missing.
   (h) Visually check for loose nuts and bolts.
   (i) Check for steam leaks around piping and casing.
   (j) Note any changes from normal exhaust vacuum. (Could indicate condenser or exhaust leaks or problems.)
   (k) Look at all lube, seal, and coupling drains for proper oil flow.
   (l) Listen for unusual noises in turbine or compressor.
   (m) If compressor has wheel wash facilities, investigate flow rate and timing. Are the flow meters working? Flush should be <3% of weight with ½ to suction line and ½ to impellers.
   (n) Check for oil leaks around bearings, coupling covers, lube and seal oil piping.
   (o) Check overhead seal oil pots for normal level or check DP where pressure regulators are used.
   (p) Check sour oil pots for normal level. Flooded pots indicate oil going into compressor.
   (q) Sweet and sour seal oil drains: Does flow appear normal? (Excessive flow could mean seal damage or other problems.)
   (r) Are vent gas recovery facilities installed? In operation? If in operation, have the operators experienced any problems?
   (s) Are sour oil pot vent lines tagged to show orifice sizes? Vent gas recovery?
   (t) Check atmosphere vents. Any unusual amounts of hydrocarbon vapors?
8. Lube and seal oil consoles
   (a) Check oil reservoir levels.
   (b) Check lube and seal oil cooler inlet and outlet temperatures. Should be 110–120°F. Higher than normal could indicate plugged coolers, high cooling water temperature.

Location and Activity (Continued)

(c) Check filter DP's
(d) Feel pumps for unusual vibration.
(e) Is steam pump spare idling? Note condition of carbon seals.
(f) Check piping, conduit, boxes, etc., for leaks, inadequate supports, plates missing, exposed wires, etc., that might cause problems or false trips.
(g) Check lube oil accumulators for level and pressure.

9. Note any bad instruments. This includes unreadable instruments because of paint, grease, etc.
10. Visually check compressor house area for scaffold boards, loose materials, or work being done that could cause machine problems by accident or ignorance.
11. Pick up a copy of operator log sheets for this compressor train. Review for adequacy of machinery data logged.
    Frequency of logging? Recommend revision, as applicable.
12. Weekly log—Do operators have a weekly machinery log? Pick up a sample of each. Prepare weekly log sheets for each unit.

Figure 3-105. Audit Interview Questions

- Tell us about yourself:
  - -Education
  - -Prior Experience
  - \- Current Responsibilities
- Describe your job function:
  - -Area of plant coverage
  - -Organizational responsibility
  - -How many people report to you?
  - -Whom do you report to?
  - -What is your yearly budget?
- How much of your job is organizational, supervisory, technical?
  - -Which part do you like best?
  - -For which part would you need more time to do a better job?
- Is there a clear statement of your job function?
  - -Is your performance measured against it?
- What would you describe as your major problem areas?
  - -Technical?
  - -Operational?
  - -Organizational?
- Where do you see areas of known or potential impact on the reliability of your plant operations?
- Where do you need help most to deal with current problems?
- Would additional training help? If so, what type of training?
- What areas do you see as future problem areas?
- How are machinery problems handled?
  - -Whom do you contact?
  - -Do you resolve problems as a team or are solutions individually offered by the designated expert?
  - -What happens when there are disagreements?
  - -Do you work them out internally?
  - -Do problems get bumped to management to resolve?

Typical audit interview questions. (Continued)

- What major problems have occurred in recent years?
- How is machinery monitored?
  -Operations
  -Technical
  -Maintenance
- Are data collected for long-term records and future reference?
  -Data distribution routine: How done, where sent?
- Do you have a means of rapidly relating operational problems with their impact on process machinery? If so, how is it done?
- What would you like to see implemented in order to improve the reliability of the plant?

Figure 3-106. Machinery Reliability Audits Functional Checklist

The comprehensive evaluation of machinery reliability not only requires a detailed evaluation of the history and operation of the equipment, but also an evaluation of the plant support and operation functions. To evaluate to what extent these functions impact machinery reliability, an evaluation procedure, as outlined below, is used to identify areas that are contributing to or preventing the achievement of maximum potential machinery reliability.

I. Process Operations Function
  A. Organization
    1. Structure
      a) Plant-wide
      b) MTS/Technical/Maintenance Relationship
    2. Manning
      a) Number/Position
      b) Dedicated for Machinery
      c) Skill/Experience
      d) Multi-skilled (Scheduled Rotation)
      e) In-house and Contracted Resource
    3. Responsibility/Authority (Boundaries of Jurisdiction)
      a) Mission/Role Statements
      b) Priority Setting
      c) Horizontal (Across Functions)
      d) Vertical (Hierarchy)
      e) Understanding (Communicated)
    4. Training—Initial/Continuing
      a) Scope (Classroom, On-The-Job)
      b) Facilities/Resources
      c) Rehearsals and Simulations
  B. Standard Operating Procedures
    1. Routine Tasks and Actions
      a) Verification of Instrument Indications
      b) Visual Checks, Including Local Auxiliary Systems
      c) Data Logging and Observations
      d) Testing of Auxiliary/Backup and Positive Systems
    2. Normal Startup and Shutdown
    3. Emergencies
    4. Monitoring Hardware/Software Systems of Process Variables and Mechanical Conditions
    5. Computer Systems
    6. Control House Systems
    7. Local Compressor/Generator Facility
    8. Others
II. Maintenance Function
  A. Organization
    1. Structure
      a) Plant-wide
      b) MTS/Process Operations/Technical Operations
      c) Craft Inter-relationships (Instr., Elect., Mach.)
    2. Manning
      a) Number
      b) Dedicated for Machinery
      c) Skill/Experience
      d) Multi-skilled Personnel (Scheduled Rotation)
      e) In-house/Contracted Resources

Machinery audit functional checklist. (Continued)

3. Responsibility/Authority (Boundaries of Jurisdiction)
   a) Mission/Role Statements
   b) Horizontal (Across Functions)
   c) Vertical (Hierarchy)
   d) Understanding (Communicated)
4. Training—Initial/Continuing
   a) Scope (Classroom, On-The-Job)
   b. Facilites/Resources
   c) Contractor Qualifications/Training

B. Standard Maintenance Procedures
1. Routine Tasks and Actions
   a) Testing of Auxiliary/Backup and Protective Systems
   b) Parts Availability Verification and Quality Control
   c) Other, e.g., Vibration Signature Analysis, Oil Condition, Seal Oil Leakage Rates
2. Turnaround Planning
   a) Organization
   b) Procedure for Development
   c) Machinery System Coordination
   d) Backlog Review
3. Emergency

C. Systems in Place
1. Inspection Tools/Techniques Available and Used
2. Preventive/Predictive
3. New, Small Expansion Project Responsibilities

IV. Spare Parts Availability/Quality Control Function

A. Organization
1. Structure
   a) Plant-wide
   b) Relationships: MTS, Process, Technical, Maintenance, Purchasing, Stores
2. Manning for Routine & Special Spares, i.e., New and Rebuilds
   a) In-house/Contract Inspection and Expediting
3. Responsibility/Authority
   a) Mission/Role Statements (Materials Management Policy)
   b) Definition of Parts Needs, Initial and Restocking
   c) Inventory Control

B. Procedures and Systems
1. Purchase Order Preparation
   a) Routine Spares—Automatic Reordering
   b) Special Request Spares—Rebuilds, T/A Preordering
2. Inspection and Expediting Procedures
   a) Schedule, Quality, Deviation from P.O.
3. Material Received Acknowledgement
4. Material Received Inspection
5. Special Storage Requirements
6. Intra- and Inter-plant Interchangeability
7. Inventory Control

Typical Findings are Discussed with Plant Personnel

As was indicated earlier, the findings and recommendations of a rigorous machinery reliability audit are always discussed with local management and/or other designated personnel before the audit team leaves the site. These discussions enable all attendees to hear and question the team's assessment. Better yet, the discussion provides an excellent forum for debate and discourse, allowing all concerned parties an opportunity to "buy into" the conclusions and recommendations.

The results of a typical machinery audit are often presented with three areas of emphasis. These areas deal with *operational practices and procedures influencing reliability, machinery impact on reliability,* and *organizational means for improving reliability.* Follow-up material compiled and forwarded by the audit team generally includes a series of supporting documents designed to aid in implementing better control of machine reliability through organizational, mechanical, and procedural means. This "appendix package" typically includes documents such as a specific role statement for important machinery-related engineering positions, sample procedures for the performance of critical instrument on-stream inspection, improved startup procedures for vulnerable turbomachinery trains, and guidelines for upgrading reciprocating compressor components.

To appreciate the scope of a well thought out machinery audit effort, a sample of observations and recommendations dealing with operational practices and procedures influencing reliability is given below:

- Operators, at times, may not have sufficient opportunity to review procedures when under pressure to restart a compressor train; consequently, errors in procedure might occur. It is thus often recommended that critical start up procedures be provided on laminated plastic cards sized to fit in the operator's shirt pocket (Figure 3-107).
- For steam turbines, there may well be an awareness that special procedures have to be employed to avoid component damage from "rotor bows" when restarting after a short shutdown, but specific steps are often not readily available in written form. Startup diagrams for normal ("cold start") and special ("hot restart") sequence may have to be developed for field-posting in painted sign or engraved plaque format.
- Protective systems are sometimes not being tested "on the run" for fear that they might cause an inadvertent shutdown. To deal with this and other related problems, the audit team may leave written recommendation and relevant documents, such as:
  - A sample instrument checkout procedure that describes typical on-stream verification routines for critical instruments such as compressor suction drum high-level shutdowns.

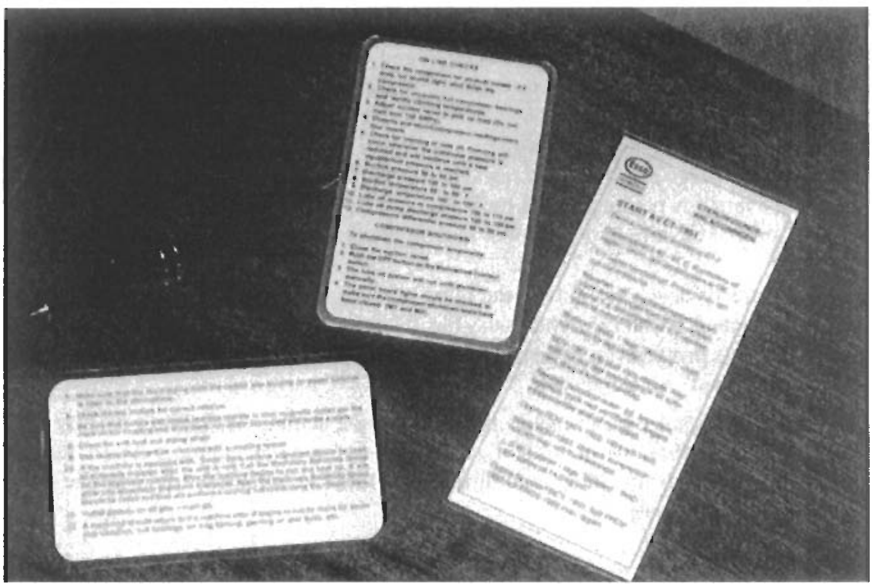


Figure 3-107. Shirt-pocket checklists can prevent costly operating errors.

-On steam turbine driven compressors the "trip oil" dump valve should be modified to allow isolation in order to permit on-the-run testing of all trips and alarms.
-On motor driven compressors the "tripping relay" should have isolation provisions to achieve the same testing function as on steam turbine driven compressors.
-On steam turbines, following any shutdown in which alarm or tripping circuit instrumentation could have been affected or after compressor repairs or turnarounds, the turbine should be tripped once on "slow roll" to assure the operation of the tripping circuit.

- Lube oil analysis procedures might generally be adequate, but not up to the standards used by leading plants. In these plants, in addition to testing for H_2O, flash point, viscosity and appearance, testing includes oxidation inhibitor and acid number in order to help assure longer life of bearings and seals.

Similarly, a few examples of "machinery impact" observations and recommendations typically documented by an audit team are highlighted next:

- Disc-pack couplings provided on pumps may not have a means for on-line inspection. Provisions for hinged covers or the retrofitting of expanded sheet metal windows is recommended for the coupling guards of disc-pack couplings. Hinged or expanded sheet metal covers facilitate visual inspection of couplings under operation through the use of a strobe light to "freeze" the coupling allowing observation of disc "bowing" or other malfunctions that manifest themselves at speed and load.
- Compressor trains are occasionally found with gear-type couplings that can limit extended plant operation. A systematic program of planning for future conversion to low-maintenance, high reliability dry-type couplings is sometimes advocated.
- Pump failure experience at petrochemical plants always merits close review. Measures to upgrade mechanical seal selection and rolling element bearing procurement are usually explained and an implementation schedule mapped out for the plant.
- Based on field experience elsewhere, an audit team may consider hydro-pneumatic governors installed on large steam turbines a threat to long-term reliability of major machinery. The technical reasons would have to be explained and electronic governor retrofits recommended for future consideration. The cost basis for retrofit application of electronic governors must be provided as part of the audit follow-up.

As a means of determining how the plant pursues chronic problems, an audit team should use actual problems, when possible. This not only provides immediate assistance, but also provides a means of demonstrating failure analysis concepts to an audience "primed" for the results.

Addressing organizational means for improving machinery reliability usually constitutes the third phase of an audit. To further re-enforce the technical capability of a client's staff, audit teams often recommend and adapt to the needs of a specific facility three key techniques:

- A detailed *role statement* typically is developed for the position of Equipment Reliability Engineer. Experience shows that without this statement, the perceived role is going to differ widely from person to person. While the individual slated for the job may have a clear idea of what's expected, others may not have the same understanding. Publishing this role statement, which by the way includes not only what the job *is*, but also what it is *not*, avoids the frequent misunderstandings that often occur in positions of this type.

- A *Service Factor Stewardship Committee* charter is sometimes developed. The thought behind this activity is based on a simple premise: The successful identification and resolution of plant problems requires that:
  - -All departments (operations, technical, maintenance) recognize that a problem exists.
  - -All departments participate in the resolution of the plant problem since, in the end, they are all affected by the problem.
  - -One individual is generally given the responsibility for resolving the entire problem.
  - -There is an organizational means for the individual or group to efficiently use the skills of the entire organization.

In addition to developing means for resolving problems, successful plant organizations look beyond this effort and consider problem-solving sessions as part of a consistently applied cooperative effort to improve service factor. Once problems are seen as but one of the elements that have an impact on the service factor, a more cohesive and effective approach can be used to identify the root causes, eliminate them, and establish procedures to prevent their recurrence.

The implementation of an organization such as the Service Factor Stewardship Committee has proved to be an effective means for bringing together diverse elements in a plant organization for the purpose of improving plant service factor.

- A technical training approach emphasizing state-of-the-art technology needs to be defined for the mechanical/machinery workforce at many plants. This phased approach starts out with two deceptively simple, but highly effective self-training methods.
  - -The plant should subscribe to several important trade and engineering journals. These should be routed to key recipients—usually engineers or technicians for review and screening. Within five working days from receipt, the key readers should be obligated to identify and pass on to selected coworkers, write-ups, advertisements, editorials, etc., that are of potential importance to the plant or to a given individual on the machinery/technical staff.
  - -Each machinery/technical person should periodically be called upon to present informal, 10–20-minute "shirt-sleeve seminars" to mechanics and machinists assigned to both shop and field. This would compel the presenter to do a bit of research, educating himself and others in the process, and contributing to the development of team spirit, mutual respect, and cooperation among different groups of the organization.

From here, the phased approach to training moves toward in-plant courses in specific technology areas ranging from maintenance to machinery performance analysis.

Attendance at well-defined known-to-be-relevant outside seminars or symposia is the next step. Again, the audit team must be fully familiar with the content and relevance of these training opportunities in order to properly assess their applicability to the needs of a given plant.

Machinery Reliability Audits Have Wide Application

It has been said that a properly executed machinery reliability audit compares a given plant with the best of competition. Audits can be performed on virtually any process plant with heavy dependence on machinery. Refineries and large chemical plants are obviously among the most likely beneficiaries of well-structured audits; however, utilities, paper and pulp, mining, and other industries will find machinery reliability audits a suitable means for appraising vulnerabilities.

An experienced audit team should be careful to keep the focus on existing as well as potential reliability-related observations and recommendations. Machinery observations that are only cost-related, e.g., periodic lube oil replacement vs. periodic on-stream purification of turbomachinery lube oil would become an issue only if it were uncovered that periodic replacement is not practiced, or if lube oil contamination were found to exist.

The machinery reliability audit process is clearly a means for the transfer of knowledge and technology. It has been the author's experience that this type of audit differs from other, more generalized plant reliability assessments by being far more specific. Properly manned and structured, machinery reliability audits very often offer solutions on the spot, making their value both tangible and immediate. Equally important, machinery reliability audits provide state-of-the-art updates, technology transfer, and a greatly reduced risk of unexpected plant outages attributable to machinery distress.

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Chapter 4

Maintenance and Benchmarking Reliability

Maintenance Measurement*

Enterprise assets are used to produce the products and services that enable an organization to achieve its ultimate goal—to expand and sustain the business. The effective use of these assets is the responsibility of everyone in the organization.

Measurements, or metrics, are used to ascertain how efficiently the assets are being used. They identify where and when assets are being used inefficiently and require change. They also reveal if applied changes have positively or negatively affected the enterprise. Without measurements, the company has no perspective of the impact or direction of a change. Measurements help set and adjust enterprise direction.

Many elements affect the direction of an enterprise and therefore its asset utilization. Customers, competition, and change itself drive the need for assets to be maintained and continually improved.

- Continually changing customer product and service demands determine the speed and direction of asset utilization.
- To meet demand, competitors change and improve their assets. The asset utilization of a business should exceed its competitors' capabilities or the enterprise will not grow and sustain itself.
- Technology changes require continuous measurement of asset utilization to determine when improvements are required. Both the timing and complexity of asset changes are critical to the success of an enterprise. Measurements guide and direct that change.

Plant and equipment maintenance is an important enterprise function. Maintenance affects all enterprise assets, whether they are production equipment on the

*Adapted, by permission, from Bewig, Lou, "Maintenance Measurement," *Maintenance Technology*, December, 1966.

shop floor or the office facilities that support human resources. Maintenance represents from 5% to 15% of the enterprise's total cost of goods sold (or production costs) and is one area in which costs are steadily increasing.

To manage any asset, those responsible for making asset decisions must measure its operation. Without measurements, they do not know where they stand. They do not know when to make changes. They do not know if changes made are moving the operation in the desired direction. They do not know whether the distance they have moved is toward or away from the desired goals. Without measurements, decision makers have no perception of the possible improvement potential. These statements seem obvious, but if they are so obvious, why isn't more being accomplished through the use of measurements?

- The measurement management theory is not understood
- Measurement time and costs are not justified
- The importance of the functions needing measurement is not known
- There is lack of knowledge of how and what to measure
- The potential improvement is unknown.

Enterprise information is as much an asset as any piece of equipment or worker. The information collected through a measurement program is one of the enterprise's most valuable assets, but it is significantly underused. Information, like all other enterprise assets, has an initial acquisition cost and a maintenance cost. As an asset, it must be used and maintained to be of value. In most cases, collected data are not being used efficiently.

Goals and Metrics

The starting place for any measurement program is to establish enterprise goals. These goals establish enterprise direction. Fewer focused goals are better than many broad goals; the ultimate enterprise goal is to expand and sustain the business (the associated metric is the existence of the business).

A critical element of a goal is how it is to be measured. A goal without a measurement has little value. Enterprise goals and their associated metrics are developed at three levels: the strategic level, the tactical level, and the operational level.

Goals developed at the strategic level have associated elements known as critical business issues (CBI). CBIs are the elements necessary to achieve a goal. For example, the goal to expand and sustain the business has a related CBI of having an efficient and effective maintenance operation.

When enterprise goals, metrics, and CBIs are being developed, several elemental interrelationships must be considered. These relationships have varying impacts. Relationship matrices can be established to assist in developing and assessing the goals and CBI interrelationships, Figure 4-1.

Enterprise goals and metrics should be developed from the top down. Strategic level CBIs relating to an efficient and effective maintenance operation become a maintenance operation's goals. These goals and their associated metrics are at the

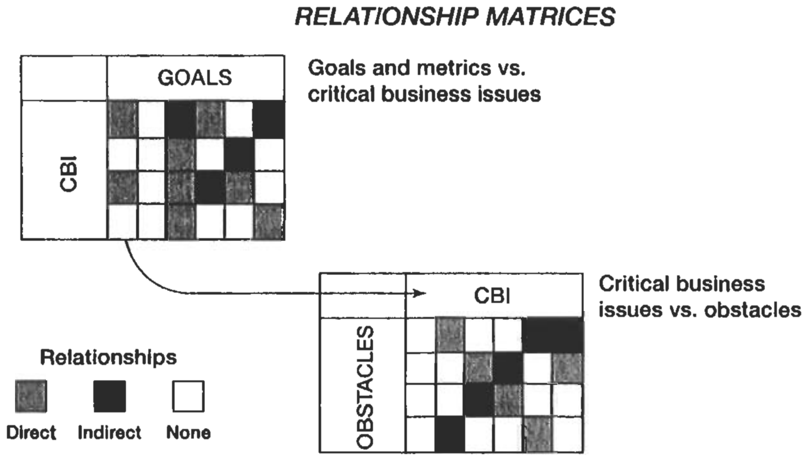

Figure 4-1. Relationship matrices.

tactical level. Tactical level goals and metrics also have associated CBIs. These CBIs become the operational level goals with their associated metrics. This top-down approach ensures that operational level goals and metrics relate to tactical and strategic goals and metrics, Figure 4-2. This goal focus sets enterprise direction.

Metric data are used to determine the impact of change on an operation. These data are a valuable enterprise asset that must be maximized. Metrics have three important aspects:

- Determining what metric data to gather and use
- Gathering the metric data
- Using the metric data.

To be effective, metrics must be carefully selected to measure what is important. What is measured will improve. Selecting what is most important to the operation, and measuring it, will focus improvement efforts. After what is to be measured is determined, appropriate data must be collected, stored, and made accessible.

Metrics are like other tools: If they are not used, they have little value. The value of metric data lies in the skills of those using the tools or metrics. To provide value, the metric asset must be used within the operation. For the value of the metric asset to be maximized, the data must be shared with other organizations.

If common metrics are established for each enterprise operation, the data can be shared among many operations and results can be compared. Comparison of metric data benefits all organizations sharing the information. Value is realized by adapting (not adopting) what is learned from the information. The competitive advantage is not in the metric data, but in how the information is used, just as it is with any other tool.

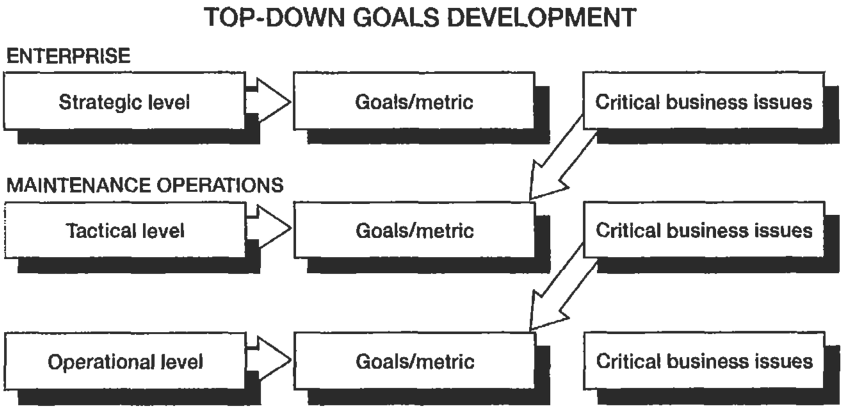

Figure 4-2. A top-down approach ensures that operational level goals and metrics relate to tactical and strategic goals and metrics.

Measurement Attributes

To be efficient and effective, metrics must have the following attributes. Measurements must be:

- Appropriate. They must accurately measure the specific operational aspect that requires measurement.
- Acceptable. They must be considered, by all concerned, to measure the desired operational aspect.
- Simple. They must be easy to understand, easy to gather, and easy to interpret.
- Unambiguous. Metric results must communicate a clear message relating to the operation being measured.
- Comparable. Metric data must be capable of being analyzed in relationship to previously gathered internal and external data.

Strategic Level Maintenance Measures

The following strategic level metrics are suggested as a common set of maintenance measurements:

- Maintenance costs as a percent of cost of goods sold (or total production costs)
- Maintenance costs as a percent of machinery and equipment replacement value
- Number of service maintenance employees as a percent of direct labor employees
- Spare parts inventory as a percent of machinery and equipment replacement value
- Spare parts inventory turns
- Percent of routine scheduled maintenance hours

- Certifiable training costs per employee
- Maintenance-related downtime

To assure accurate comparisons, it is extremely important that measurements be calculated in the same manner from consistent data. Calculations also must be clearly understood for meaningful comparisons. For example, the exact elements that generate "cost of goods sold" or "total production cost" vary from business to business and must be calculated consistently for an accurate comparison.

Another example is in the use of the number of service maintenance employees. The appropriateness of this measure depends on the level of facility automation. More maintenance workers may be required to support the automated operation, although fewer direct labor employees may be needed. The formulas for calculating the suggested maintenance measurements are shown in Table 4-1.

No one measurement can paint the entire picture of an operation. These suggested strategic level maintenance measures attempt to cover all the major maintenance elements: human resources; materials; machinery, facilities, and production equipment; and the maintenance processes.

Table 4-1
Formulas for Calculating Strategic Level Measurements

Measurement	Formula
Maintenance costs as percent of cost of goods sold	$\dfrac{\textit{Maintenance costs}}{\textit{Cost of goods sold}} \times 100$
Maintenance costs as a percent of machinery and equipment replacement value	$\dfrac{\text{Service maintenance costs}}{\textit{Machinery and equipment replacement value}} \times 100$
Number of service maintenance employees as a percent of direct labor employees	$\dfrac{\textit{Number of service maintenance employees}}{\textit{Number of direct labor employees}} \times 100$
Spare parts inventory as a percent of machinery and equipment replacement value	$\dfrac{\textit{Spare parts inventory value}}{\textit{Machinery and equipment replacement value}} \times 100$
Spare parts inventory turns	$\dfrac{\textit{Value of spare parts issued in past 12 months}}{\textit{Average spare parts inventory value for past 12 months}}$
Routine scheduled maintenance hours as a percent of total maintenance hours	$\dfrac{\textit{Routine scheduled maintenance hours}}{\textit{Total maintenance hours}} \times 100$
Certifiable training costs per employee	$\dfrac{\textit{Certified training costs}}{\textit{Total number of service maintenance employees}}$
Maintenance-related downtime as a percent of total downtime	$\dfrac{\textit{Maintenance — related downtime}}{\textit{Total downtime}} \times 100$

Tactical Level Maintenance Measures

Succeeding with strategic level maintenance measurement comparison will generate significant improvements. Tactical level maintenance measures should be developed to support the enterprise's associated tactical goals. Although these measurements are important, no specific measurements are suggested here. These measurements can be developed in relationship to specific enterprise and maintenance operation situations.

Operational Level Maintenance Measures

The operational level is the lowest level of maintenance measurement. Comparisons at this level are not as important as those at the strategic level. Operational level comparisons are most beneficial in the investigation of specific improvement situations. Operational level metrics are used primarily by a maintenance department to better manage its operation. Some suggested operational level metrics follow:

- Total minimum maintenance cost
- Life-cycle cost of asset ownership
- Mean time between failures
- Mean time to restore
- Overall equipment effectiveness (a total productive maintenance, or TPM measurement)
- Average response time to unscheduled machine failure
- Percent of time machine is available to run versus scheduled run time
- Periodic customer satisfaction surveys
- Periodic skilled trades work constraint analysis

These measurements, like the strategic level measurements, must be checked consistently and used to be effective. It is recommended that a maintenance engineering function be established to manage and use maintenance data, among other responsibilities.

Benchmarking

A benchmarking industry has emerged in recent years, along with a broad range of measurements (including maintenance measurements) that theoretically define "world class." For the most part, these are bottom-line-type measurements that have value when internal performance is compared. Table 4-2 lists some world-class value examples for strategic level measurements.*

Various sources contacted for these values indicated that they would not validate the numbers. There are too many associated variables and there was generally too much risk associated with publishing the values. Commercial sources felt that their values were proprietary and that publishing the values divulges information from which they derive revenue. These are just some of the problems with benchmarking.

*Numerical values have been adjusted to reflect H. P. Bloch/F. K. Geitner's experience.

Table 4-2
Sample Values for Strategic Level Measurements

Measurement	Value Best	Value Average
Maintenance costs as a percent of cost of goods sold	1.9	4.0
Maintenance costs as a percent of machinery and equipment replacement value	2.0	5.0
Number of service maintenance employees as a percent of direct labor employees	3.0	7.0
Spare parts inventory as a percent of machinery and equipment replacement value	1.0	3.0
Spare parts inventory turns	1.4	0.5
Routine scheduled maintenance hours as percent of total maintenance hours	80.0	35.0
Certified training costs per employee	$1,200	$400
Maintenance-related downtime as a percent of total downtime	4.0	10.0

As a result, some words of caution are necessary regarding benchmarking and the information on the world-class characteristics just presented. The information is provided because everyone seems interested. The use of this information is, however, somewhat questionable. Questions such as the following arise:

- What is the world-class environment (for example, industry and manufacturing type)?
- Where does the measured operation fit?
- If, in a specific area, an operation's measurement exceeds world class, is there no need for improvement?
- How are the measurements calculated? Is our operation calculating the components consistently?
- Is world class too expensive for our operation?

These questions reflect just some of the issues that render world-class information questionable.

Benchmarking and data comparison are good tools if understood and used correctly. A lot of good can come from comparing an operation with others that are similar and dissimilar. New ideas and concepts from other operations can be identified and adapted. The most productive use of benchmarking and value comparison is to identify where improvements are needed and determine the impact of change.

The productive way to use external benchmarking is to determine what is internally important (goals and CBIs) and use benchmarking to measure those operational aspects. In the true benchmarking concept, the internal information can then be compared with information from other organizations. The comparison must be controlled and focused on the operational aspects being studied. This approach ensures that apples are compared with apples. Sharing internal benchmark information serves only to optimize the asset. Again, the benchmark data are the tool, and how they are used is what makes the data productive.

The biggest problem with measurements and benchmarks is high development and utilization costs. Everyone in the organization must be educated concerning the costs of measurement collection and utilization and their associated values. Value is not received unless data are collected and used. The tools and time used to input, store, and retrieve the measurement data are expensive. Everyone involved must be motivated to accurately and consistently collect data, and then to use the information.

Using the information can involve the cost of establishing a data utilization function with dedicated people. Measurement values and costs are high, and management must accept the associated values and costs. Everyone is looking for the quick and inexpensive answer. Therefore world-class measurements are attractive. The process is simple and inexpensive, but the results are questionable.

True benchmarking is even more involved and costly. Once an operation knows what it wants to measure, it needs to collect internal data and compare like data with its benchmarking partners. There are collection costs and costs to locate benchmark partners, as well as costs of travel to the benchmark partners' operations to collect the required data. Data must then be analyzed and new approaches determined. All these procedures take time and money—more than management usually understands and wants to expend.

In spite of the perceived high measurement costs, the resulting benefits are high and the payback period is short. Wise enterprises seek out benchmarking partners and share data. Without the measurement road map, the business direction is not known until it is too late to adjust.

Organize to Manage Reliability*

An analysis of maintenance costs in hydrocarbon processing industry (HPI) plants has revealed that attitudes and practices of personnel are the major single "bottom line" factor. In reaching this conclusion a world-renowned benchmarking organization examined comparative analyses of plant records over the past decade. They learned that there was a wide range of performance independent of refinery age, capacity, processing complexity, and location. Facilities of all extremes in these attributes are included in both high-cost and low-cost categories. Those in the lowest quartile of performance posted twice the resource consumption as the best quartile. Furthermore, there was almost no similarity between refineries within a single company.

The Record. Figure 4-3 illustrates rising profitability during 1986–1988 because of market conditions. As the militaries mobilized into the Arabian Gulf in 1990, profits climbed sharply. By 1992, margins relaxed somewhat to pre-war levels. But if we look at trend data of a constant trend group of 68 refineries, the picture is different. From the top curve in Figure 4-3, we note that the difference in profitability between the highest and lowest quartiles was about 5% in 1986. But by 1992, the gap had widened to 12%. This divergence is not merely an industry average phenomenon. It is clearly a difference in performance of two groups, which is not related to the industry average performance.

Similar performance differences can be found in industry maintenance data. Figure 4-4 portrays a six-year trend of maintenance cost for the same trend group of refineries. The data represent total annual refinery maintenance costs per unit of

*Adapted by permission of R. Ricketts, Solomon Associates, Inc. Dallas, Texas. From a paper presented at the 1994 NPRA Maintenance Conference, May 24–27, 1994, New Orleans, LA.

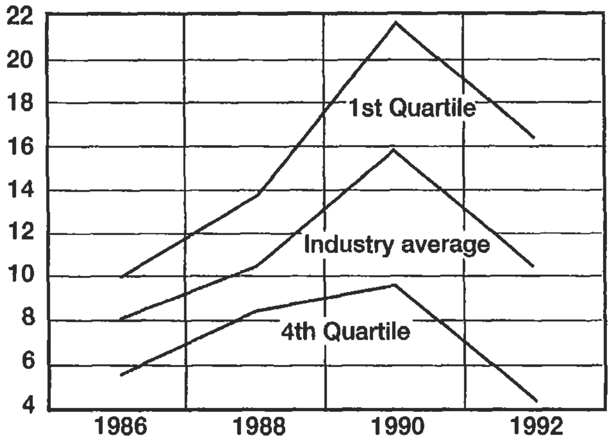

Figure 4-3. Cash basis, ROI, in percent.

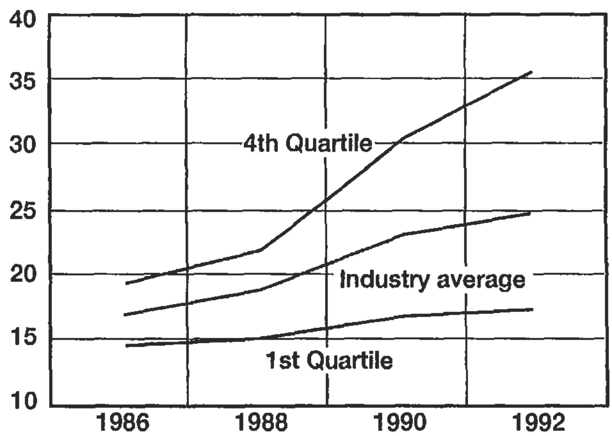

Figure 4-4. Maintenance index, U.S. $/EDC.

refinery capacity and complexity ("EDC," or equivalent daily capacity). The mid-range curve is the industry average, revealing an increasing expenditure of about 6% per year over the six-year period. This increase will surprise no one. It is characteristic of inflationary pressures and increasing emphasis on control of refinery emissions. But when the data are viewed in terms of the performance spectrum, a very different relationship unfolds. Those refineries represented by the lower curve are the lowest-cost quartile. They posted increases of less than 1% annually. On the other hand, the highest-cost quartile's spending (upper curve) doubled during the same period.

The message is that management practices are widely divergent. They represent some very different approaches to managing resource consumption in a competitive environment. Those in the high-cost group may find it hard to remain viable.

Cost vs. Availability. A concern closely related to maintenance cost is the impact of maintenance activities on availability of processing facilities. Figure 4-5 charts the performance of eleven U.S. refineries that recorded the greatest increase in costs during 1986–1992. Note that accompanying the rise in costs was a major decline in mechanical reliability. Nine refineries with the greatest improvement in mechanical reliability are represented in Figure 4-6. They climbed dramatically from average into the best quartile of mechanical reliability. So it seems that, at least in refining, improved mechanical reliability isn't related to the amount of maintenance effort.

Maintenance Spending. About 35% of the average U.S. refinery maintenance budget is for higher-profit processes as shown in Figure 4-7. All other types of processes combined account for 35%. Utilities, marine, and offsites consume the remainder of the budget at 30%. These statistics raise a question of the primary focal point of the maintenance budget. Figure 4-8 provides the answer. Fifty-four percent of the average refiner's budget is earmarked for equipment repair and programmed replacement, while energy conservation, environment and safety, and other requirements consume 38% of the budget. Reliability improvement programs account for a mere 8% of the expenditure.

Analysis of high-cost and low-cost segments of the industry reveals a wide variation in performance and trend data. The performance gap is getting wider. These differences are not related to age, size, or location as it would be tempting to believe. Furthermore, our benchmarking organization, Solomon Associates, has not observed differences in craft competence between the high- and low-cost performers. The reason

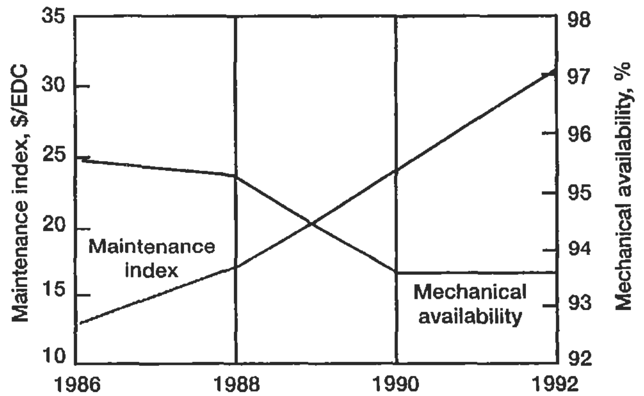

Figure 4-5. U.S. maintenance results, eleven refineries that had largest cost increases, 1986–1992.

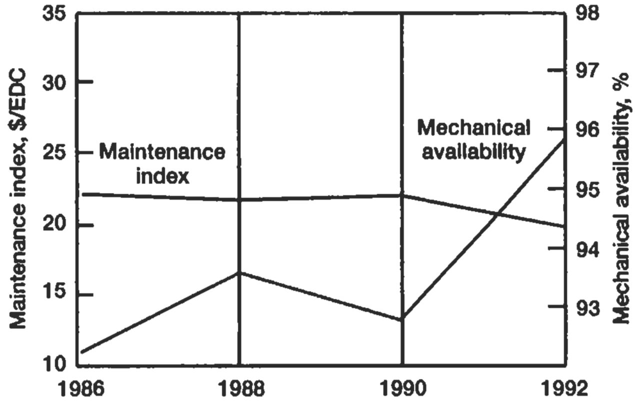

Figure 4-6. U.S. maintenance results, encompassing nine refineries that had the greatest improvement in mechanical reliability, 1986–1992.

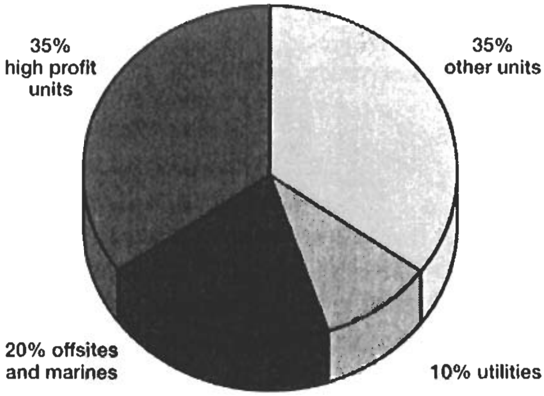

Figure 4-7. Refinery maintenance expense.

for the differences is simply that the lowest-quartile cost group has less demand for repair maintenance and thus does less work in this area.

Table 4-3 is taken from a recent worldwide maintenance management study by the same analysts. The lowest quartile's craftsmen have four times more pieces of rotating equipment per person to maintain than the highest-cost quartile. Those in the highest-cost quartile are kept busy repairing failures. They have no opportunity to examine the causes of these failures. They thus can't formulate actions to make permanent repairs or to devise preventive and predictive remedies.

Organization

There are two types of organizational approaches: repair focused and reliability focused.

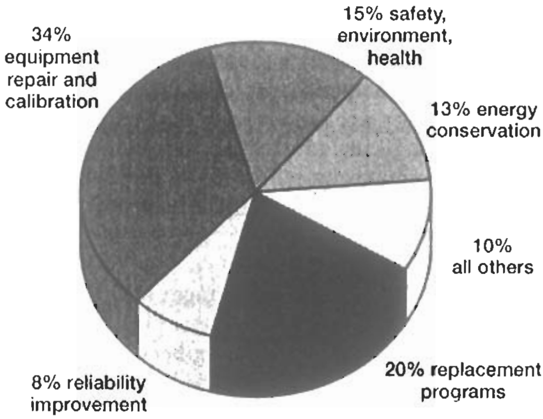

Figure 4-8. Refinery maintenance benefits.

Table 4-3
Equipment density

	(Per million units of capacity and complexity)			
	Lowest cost quartile	Items per craftsmen	Highest cost quartile	Items per craftsmen
Pumps	630	4.8	700	1.1
Compressors	35	0.3	40	0.1
Pressure vessels	490	3.7	500	0.8
Heat exchangers	390	2.9	525	0.8
Safety valves	840	6.3	1,150	1.8

Repair-Focused Organization. This organization style embraces the philosophy that equipment will fail and that the mission of the maintenance force is to respond quickly to equipment in distress. Failures are expected because they are the norm. Management and craftsmen stay occupied in repair activity and have no opportunity to examine failure cause. Staffing is designed to accommodate rapid repair, often including sizeable maintenance crews on non-day shifts. When failures do not fully occupy the workforce, the organization focuses on lower priority (frequently unnecessary) minor projects to "stay busy."

Reliability-Focused Organization. Maintenance repairs in this style are viewed differently. They are not expected to happen. They are viewed as an exceptional event and a result of a flawed aspect of maintenance policy and management focus. The specter of a recurring failure and its incumbent cost is unacceptable. The organization is sized to manage a condition-based monitoring system and assigns high priority to the elimination of failure. Unnecessary work is not performed regardless of the current work load.

Success in Managing Reliability. Achieving high mechanical reliability does not require simply more maintenance spending and more overhauls and other repairs. The best performers require less expenditures for higher mechanical reliability. What is required is a management approach reflected in the practices of superior performers, which is purposeful management of reliability for results, making repairs permanent when it counts, uncompromising pursuit of equipment condition assessment, and ongoing analysis of plant data. What practices and policies are associated with low-cost, high reliability performance?

- **Organizational purpose.** A prime factor distinguishing the better reliability and maintenance performers is that they recognize that plant reliability is not simply a result of repair effort. Not only that, they have been convinced that eliminating failure is the organization's prime mission. Consequently, they have designed their organizations to achieve results. Ease of managing the maintenance work assignments is given lower importance.
- **Information.** Refineries generate large quantities of information that describe physical needs of the equipment. Repair history, man-hour requirements for work tasks, costs, and equipment operating performance are familiar examples. The better performers recognize that their operating information is a company asset that can be saved for re-use, and that analysis of these data provides information to help in decision making. Whether they use paper or electronic systems, they don't accept excuses for not recording data or employing it to plan future work efforts.
- **Use of available technology.** It is evident that there is a lot of reliability know-how present in refineries. It includes reliability-centered maintenance, sneak analysis, furnace tube creep rate capsules, and a wide array of recent technology advances to help engineer reliability improvement. Disappointingly, recent survey data reveal that there is not much effort expended in putting these tools to work. Limited manpower, time, and funds are cited as barriers to progress in adapting new reliability technology. Yet these same refineries may pay for thousands of hours of permanent contract labor for which work may be created to keep them occupied.

  The new technologies may be reviewed by persons with special interests, but often without regard to how they can best be applied in other functional areas. For example, reliability-centered maintenance approaches define how resources for preventive maintenance can be best assigned to preserve system function. These same principles could also be useful in optimizing process operator resources.
- **Management review of reliability activities.** Most refineries report that some reliability improvement activities are being carried out in areas of the organization that have distinct engineering disciplines, such as control systems and process engineering. But at the same time they acknowledge that these specialty groups are acting in isolation from each other, without structured review of their aggregate effort by management. Priorities, expenditures, and results are not being evaluated from a view of what is best for the refinery as a whole. The better performers do not allow this to occur. They typically conduct formal, scheduled reliability program reviews by the refinery management team, or by a reliability management organizational element (see page 237) accountable for such activities.

- **Accountability for reliability.** Refiners who have achieved control of their facilities' performance and reliability have assigned individuals to be accountable for success of the operating units. This accountability is assigned in writing and may typically be included in three-to-five year operating plans. Such plans include the maintenance and reliability performance targets, objectives, budgets, reliability improvement spending for problem equipment, and the job performance expectations for each position in their organizations. Where accountability is absent, cost effective, organized problem solving and results are seldom observed.

Organizational Style and Structure

Recent worldwide maintenance and reliability management analyses have provided insight into organizational issues that affect maintenance efficiency and reliability effectiveness:

- **Maintenance cost—a stewardship role for process operations.** Processing is the reason for the refinery to exist. It must have support from all other departments to carry out plans formulated by the company's management. In this sense, it is not difficult to envision Process Operations as the custodian or "owner" of the processing machinery and equipment in addition to the hydrocarbon streams they control and direct. Furthermore, because operating assets are the sole source of revenue generation, it is similarly plausible to consider Process Operations as responsible for the equipment's integrity, with maintenance and engineering in subsidiary roles. This concept is gaining acceptance and has been seen to strengthen the review of actual need of specific maintenance work and to stimulate operating personnel into increased awareness of their roles as the eyes and ears of the operating equipment.
- **Maintenance organization structure.** Solomon Associates has identified eight structural models that classify most existing organizations. These models are based on how the maintenance group is departmentalized and where in the organization resource consumption is prioritized: centrally or distributed to refinery subdivisions. The Management Task model is characteristic of the best performers. As opposed to an organization by craft lines or by maintenance task type (mechanical, civil, electrical), the Management Task model provides planning and systems, central shops and stores services, and maintenance engineering departments to support a work-execution group. The better performers are represented equally by area assignment and central assignment approaches to controlling the workforce, but all employ centralized control of maintenance policy and work priority determination. No organizations employing an operating area team or "business-unit" approach of distributed maintenance are represented in the better performers group.
- **Use of process operators.** Several refineries have tapped the resourcefulness of their process operators to contribute to reliability improvement and maintenance. Recognizing that the always-present operating forces can be the first line of defense against equipment failures, there refineries have provided training and motivation to increase the operators' awareness of equipment difficulties and to perform light maintenance tasks which remove both the consumption of maintenance man-hours and the administrative burden of processing the associated work orders.

The most recent studies reveal that about a third of the field process operators' shift time is unstructured and available for tasks other than attending to the processes.

- **Reliability-oriented repairs.** There have been successes in improving reliability through corrective actions proposed on a regular basis as part of routine maintenance repair work. The mechanism depends on solid technical knowledge in the craftsmen and supervisors in the routine maintenance corps and a readily available maintenance engineering group for support as required. The aim is to make most repairs permanent if feasible. The results of such a process in one particularly successful refinery have eliminated recurring failures to an extent that far less repair maintenance is required than in most refineries, and the routine maintenance costs are among the lowest in the world.
- **The crew concept—a future solution for productivity?** Several refineries in the world have decided to staff their facilities with a cadre of workers capable of both operating and maintaining the equipment. This type of craftsman-operator has typically been developed by hiring persons with technical or craft skills and training them to operate the processing units. Proficiency in maintenance and in operation is established and maintained by a system of rotation, and multi-craft skill development is stressed. The refineries are willing to invest significantly in the development of such persons. One such refiner has measured an average of over two maintenance skill levels per person in addition to the operator skills. The efficiency and effectiveness afforded by their success is reflected in the significant fact that they pay their crew 35% more than other refiners in the region, yet their total maintenance outlay is 25% less than the next best refinery. There are several other significant rewards:

The crews carry out all preventive maintenance and condition monitoring schedules while on operating shift assignment.

Rather than assigning two persons to a maintenance job, those assigned to day-shift process operations are scheduled to provide short-term assistance.

The crew on operating shift performs all preparations for equipment scheduled for maintenance, including purging and draining, disassembly, and electrical lock-out and disconnection.

This versatility has enabled those using the dual-function crew concept to record up to 12% of the refinery's total maintenance demand and reliability improvement as being satisfied by the crew members while on operating assignment. This approach to effectiveness cannot be universal. The attitudes of both the workforce and management must be synchronized to the same standards of expectations, performance, and job enjoyment. But it may be possible for more refineries to approach crew benefits if existing work-rule demarcations can be relaxed or modified. Solomon Associates has observed a more willing acceptance for efficiency improvement in the younger workers. They appear willing to have a try at learning more than one skill, but may be confused by attitudes of their first-line supervisors who matured in a single-skill era and may be unwittingly holding on to concepts that do not support development of versatility.

Maintenance Cost Vs. Replacement Asset Value: Another Maintenance Spending Benchmark*

Maintenance costs compared to replacement asset value (RAV) has become a widely accepted macro measure of performance, especially in North America. The consensus of a number of respected maintenance and reliability professionals is that the amount of money that should be spent on maintenance labor, materials, and outside services is directly related to the amount it would cost to rebuild the plant, in today's dollars. Only the costs of noncapitalized maintenance work are included. Costs of turnarounds or major rebuilds that are expensed every so many years should be allocated evenly over the years between such events. All maintenance expenditures, including salaries for management supervision and staff, and other maintenance overhead, should be included. The cost of land, roads, underground utilities, etc., is not included in the RAV.

Companies typically calculate RAV by adding a yearly inflation value to the original cost of the plant. They increase RAV by the value of expansions and modifications and decrease it by the value of decommissioned equipment. This figure should be validated against the financial basis the company uses for property insurance.

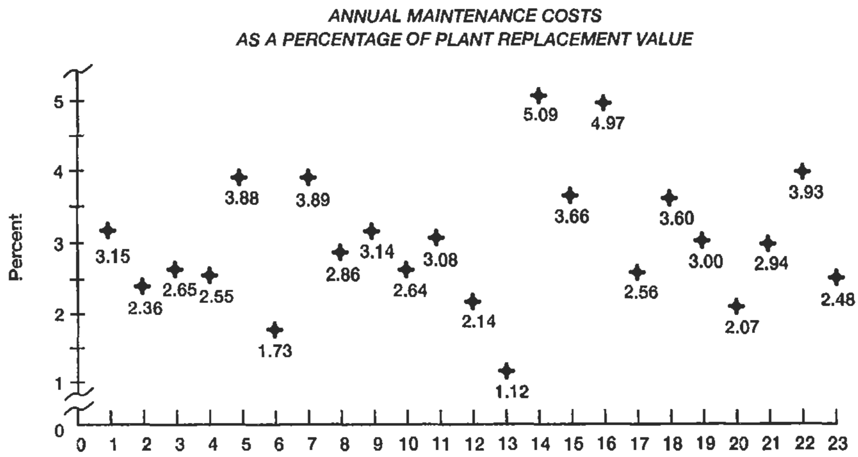

Figure 4-9. Annual maintenance costs as a percentage of plant replacement value. (*Courtesy of HSB Reliability Technologies, as published in Maintenance Technology, July–August 1996.*)

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*Raymond J. Oliverson, HSB Reliability Technologies, Kingwood, Texas, as reported in *Maintenance Technology,* July–August 1996. Adapted by permission.

Maintenance excellence begins when maintenance expense is in the range of 2% of RAV. It is believed that this indicator is not industry specific. One major caution must be kept in mind: a plant could be spending less than 2%, but not be maintaining the facilities and equipment at an appropriate level. Other indicators, such as maintenance cost per unit of output, mean time between failure, and reactive vs. planned maintenance, should be used to verify that the maintenance service level is proper.

The accompanying chart, Figure 4-9, shows results from 25 plants in the oil and gas industry recently benchmarked by HSB Reliability Technologies. Managers can gain insight into the relative performance and improvement potential of their plants easily and quickly by comparing maintenance costs to RAV.

Chapter 5

Life Cycle Cost Studies

Virtually every process plant manager subscribes to the goal of extending equipment life, availability, and reliability. Achieving these objectives usually requires up-front effort and money, and both seem to be scarce resources.

But even the realistic manager who knows that reliability comes at a price may not want to authorize these expenditures on the basis of intuition alone. Instead, he may ask for cost justification linked to a payback period, a cost-benefit calculation, a life cycle improvement multiplier, or some other tangible factor.

It is usually at this point in the chain of consideration that the reliability engineer decides he has no data and the issue is closed. Back to status quo—business as usual!

But it doesn't have to be that way. There are many means to determine with reasonable accuracy the monetary incentives or justification for equipment and component upgrading. This chapter illustrates how some experienced technical people are accomplishing this task, and highlights other avenues available to reliability professionals who see merit in de-emphasizing the purely intuitive approach and wish to use appropriate numerical alternatives instead.

We start out by explaining a variety of greatly simplified, but nevertheless, valid approaches. The interested reliability professional may then wish to direct his attention to the second part of this chapter, dealing with the more rigorous, classical life cycle cost (LCC) methods.

Simplified Life Cycle Cost Estimating

Life cycle cost estimating is rapidly becoming one of the reliability engineer's most effective improvement tools. This estimating approach takes into account the initial purchase and installation costs of equipment, auxiliaries, and software systems; to these it adds the cost of failure events, including, of course, lost production.[1,2]

Electronic Governor Example

Consider the sample case of an electronic governor system installed on a mechanical drive steam turbine coupled to a process gas compressor in the late 1980s. Let's assume the plant expected to operate for at least 30 more years, but had to consider three options. The first choice was to keep the existing hydro-mechanical governor—a zero initial cost option.

Purchasing a new, nonredundant electronic governor might be the second option, and installing a new, fault-resistant (fully redundant) electronic governing system might represent the third possible course of action.

The yearly repair costs would be calculated by multiplying the average frequency of failure by the cost of each failure event. The mean time between governor failure (MTBF) and mean time to repair or replace a governor (MTTR) are typically needed to perform a reliability analysis:

$$C_Y = (C_G)\,(8760)/(\text{MTBF} + \text{MTTR}) \tag{5-1}$$

where C_Y = annual cost of failures for a governor or associated (governed) system
C_G = cost per failure event
MTBF = mean time between failure, hours
MTTR = mean time to repair or replace, hours

The following information is known about the three options:

The 28 hydromechanical governors at this plant have failed a total of 33 times in the last seven years, requiring an average repair or replacement time of 18 hours. Thus, MTTR is 18 hours, and

$$\text{MTBF} = \frac{(28)\,(7\text{ yrs})\,(8{,}760\text{ hrs / yr}) - (33\text{ failures})\,(18\text{ hrs})}{33\text{ failures}}$$
$$= 52{,}011\text{ hours or }5.94\text{ years}$$

In this example, the cost to repair or replace a hydromechanical governor is \$12,370. Production losses are primarily influenced by the need to flare huge amounts of hydrocarbon feed for approximately 1½ hours per outage event. This costs the plant \$72,820, plus \$5,120 in lost profits and \$4,960 in restarting, overtime, and associated costs. Adding \$20,000 for environmental fines, which will likely be assessed against the plant, the total now stands at \$115,270/5.9 years, or \$19,537/year.

For the nonredundant electronic governor alternative, the plant has to depend on outside sources for projected failure data. It was found that others experienced one failure every 80,270 hours, or 9.16 years. The projected MTTR for nonredundant electronic governors is only 4 hours and will have little influence on the MTBF expression.

Replacing the hydromechanical governor with an electronic alternative will cost \$49,700 in acquisition and conversion costs. In case of failure, troubleshooting and component replacement costs are estimated at \$3,700. Since the projected MTTR of 4 hours exceeds the necessary furnace tube cool-down period of 1½ hours, it will again be necessary to flare for 1½ hours. Accordingly, pursuit of this option will again incur the \$72,820; \$5,120; \$4,960 and \$20,000 components and the cumulative total will be \$106,600/9.16 years, or \$11,638/year level annual costs.

Finally, we look at option 3, the redundant electronic governor system. Its acquisition and conversion costs are \$71,870. In case of failure of one of the two modules, troubleshooting and component replacement costs are again estimated at $3,700. Although the projected MTTR is 4 hours, the probable MTBR of this active redundant system is 9.16 + (9.16/2) = 13.74 years.* Prorated expenditures attributable to flare losses, lost profits, restart expenses, overtime, and environmental fines will be again $106,600 resulting in annual costs of 106,600/13.74, or $7,758.

The total life cycle cost for each option can now be obtained by adding the initial acquisition cost, the initial installation cost, and the recurring yearly costs. A present value conversion accounts for the time value of money and allows future operations (OC), maintenance (MC), lost production (LP), and even decommissioning costs (DC) to be added to present acquisition and installation costs. The total life cycle cost (LCC Total) is thus:

= AC + IC + present value of (OC + MC + LP + DC) Reference 3

Present value is cost multiplied by the cumulative present worth factor

$$\left[\frac{(1+i)^n - 1}{i(1+i)^n}\right] \tag{5-2}$$

where: i = real annual interest rates in percent, and
n = number of years

The factor may be obtained from tables as a function of interest rate and time (years), but it is also readily available on most computer spreadsheet programs as a present value (PV) function.

For the various options, using annual interest rates of 6%, and projecting a 30-year plant life, we would obtain present values of $267,024; $209,881; and $178,659, respectively.

Total life cycle costs would be $267,024 for option 1; $209,881 for option 2; and $178,659 for option 3. The redundant electronic governor option would thus be favored.

How to Obtain Data on Failure Frequencies

Anticipated failure frequencies and life expectancies of machinery and components are not always readily available. Nevertheless, an experienced reliability professional will not be deterred in his search for data. He or she may resort to a telephone survey of known users, communicate with the service departments of original manufacturers and repair shops, engage in literature search in a technical library, or

*MTBF of a randomly failing multiple component active redundant system may be evaluated $MTBF = \frac{1}{\lambda}\left[1 + \frac{1}{2} + \ldots + \frac{1}{c}\right]$, where the failure rate $\lambda = \frac{1}{MTBF}$, and c = number of parallel components.

LCC Calculation for Governor Options

Interest Rate:	**6.00%**
Project Life:	**30 Years**

Cost Contributor	Existing Governor	Single El. Governor	Redundant Electronic Governors
MTBF (Years)	5.94	9.16	13.74
MTTR (Hours)	18	4	4
Cost Per Component Failure Event ($)	12,370	3,700	3,700
Associated Costs Per Failure Event ($)	102,900	102,900	102,900
Acquisition and Installation ($)	0	49,700	71,870
Cost of Component Failures Per Year ($)	2,082	404	269
Associated Costs Per Year ($)	17,317	11,233	7,489
Annual Costs	19,399	11,637	7,758
Present Value of Level Annual Costs	267,024	160,181	106,789
LCC Total:	267,024	209,881	178,659

dig in his own files for technical papers and magazine articles that could shed light on the matter.

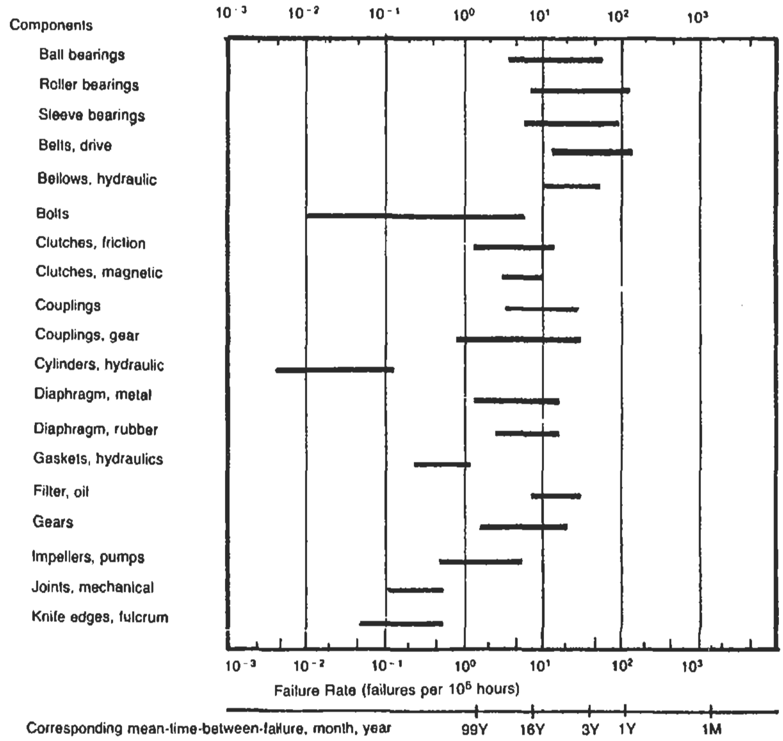
Or, the reader could simply review Appendix B of this text, which deals with common-sense reliability models. Under "Rotational Alignment Effects on Cost and Reliability," one would discover that "good" alignment practices are likely to yield MTBF multipliers of around 0.65, while "better" and "best" alignment practices are expected to result in multipliers of 0.92 and 0.98, respectively. Similarly, grouting effects or the effects of different piping practices on component life, and thus overall cost and reliability, can be discerned from this useful Appendix.

Bloch and Geitner[4] present the life spans of selected machinery components and equipment in their book. They are reproduced for the reader's convenience as Tables 5-1 through 5-3. It will be immediately evident that for some components there is a wide range of probable life expectancies.

Take ball bearings, for example. Table 5-1 shows them to last anywhere from 1.9 to 19 years—not a bad guess for grease-lubricated electric motor bearings in the average chemical plant. One can, indeed, expect about two years continuous operation from sealed bearings in a 10 HP electric motor; whereas, open bearings—periodically relubricated using both proper grease type and application procedure—will often last 20 years or more.

A reliability engineer might use the data contained in Tables 5-1 through 5-3 as a model for compiling his own statistical component life expectancy database. He might further subdivide the various component categories and assign life expectancies as shown in Table 5-4. Or, he might find merit in the approach taken by a large ethylene plant in the mid-Western United States, Table 5-5. Their efforts to thus quantify anticipated mean times between equipment failures have improved the accuracy of their life cycle cost computations. This, in turn, has led to greater visibility and enhanced respect for the diligent contributions of reliability professionals at their plant site.[5]

Table 5-1
Life Spans of Selected Machinery Components and Equipment



Component Upgrading and Its Influence on Equipment MTBF

Equipment mean time between failure (MTBF) can be calculated with reasonable accuracy from the expression

$$MTBF = \frac{1}{\left[\left(\frac{1}{L_1}\right)^2 + \left(\frac{1}{L_2}\right)^2 + \left(\frac{1}{L_3}\right)^2 + \left(\frac{1}{L_4}\right)^2\right]^{0.5}} \tag{5-3}$$

Here, L = estimated life, in years, of the component subject to failure.[6]

It is often possible to identify wear parts of the most failure-prone components of a machine and assign experience-based or otherwise known values of L_1, L_2, etc. to these components. The influence or effect of individual component upgrading on

(text continued on page 266)

Table 5-2
Life Spans of Selected Machinery Components and Equipment

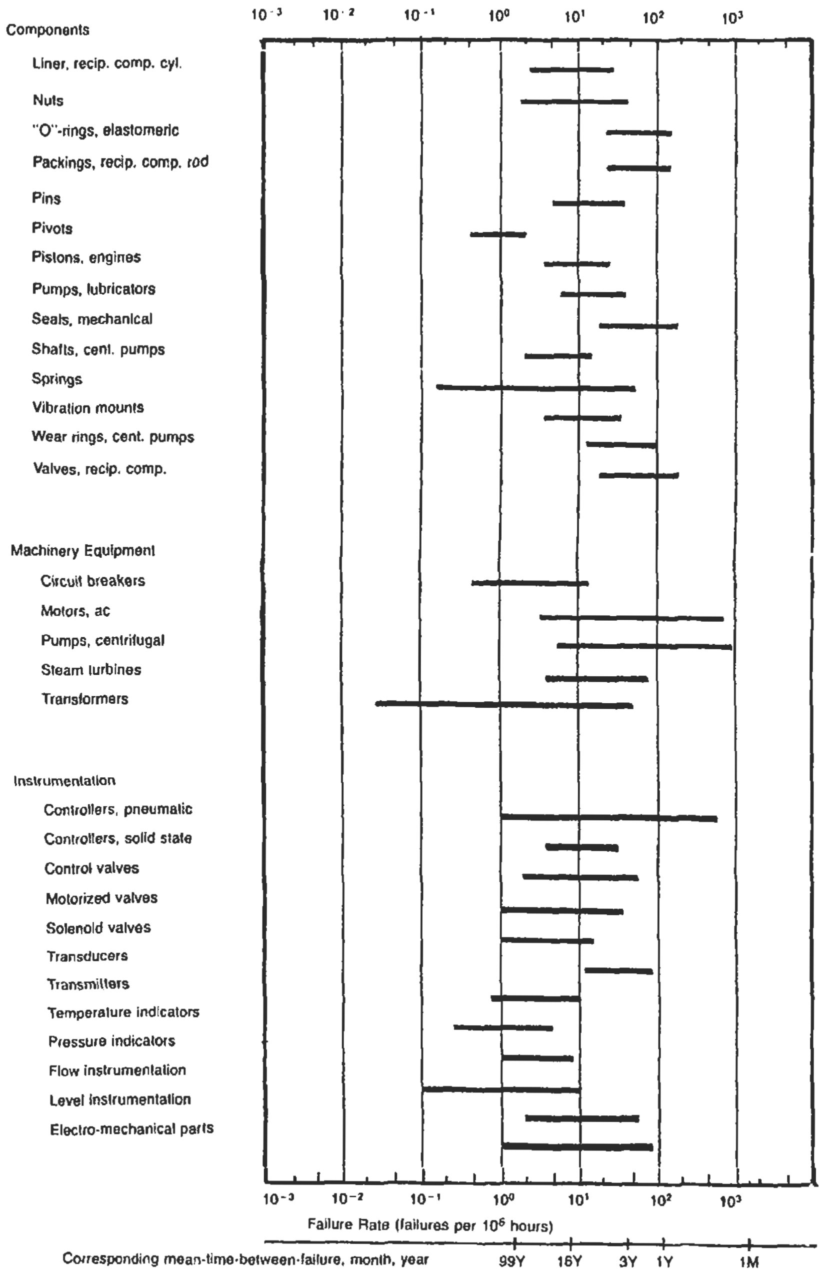


Table 5-3
Life Spans of Selected Machinery Components and Equipment

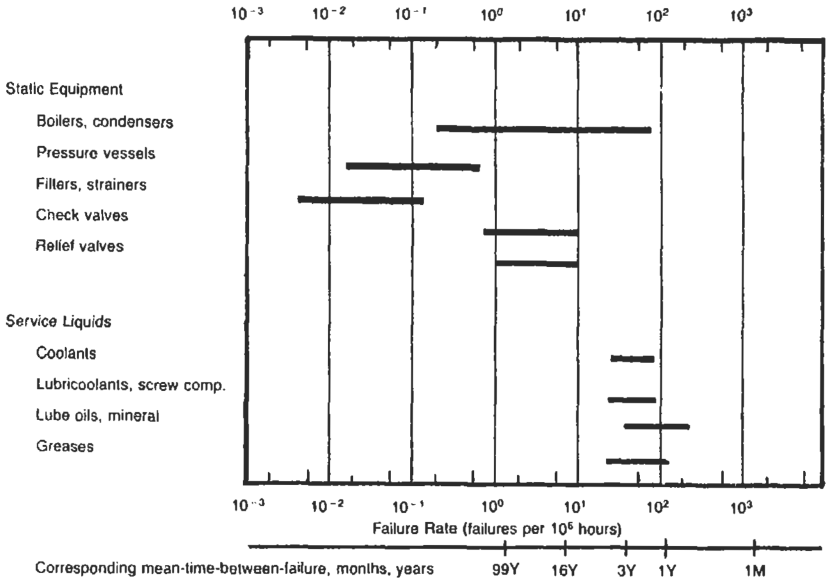


Table 5-4
Component Life Expectancies Further Subdivided to Reflect the Experience of *Your* Facility (Example)

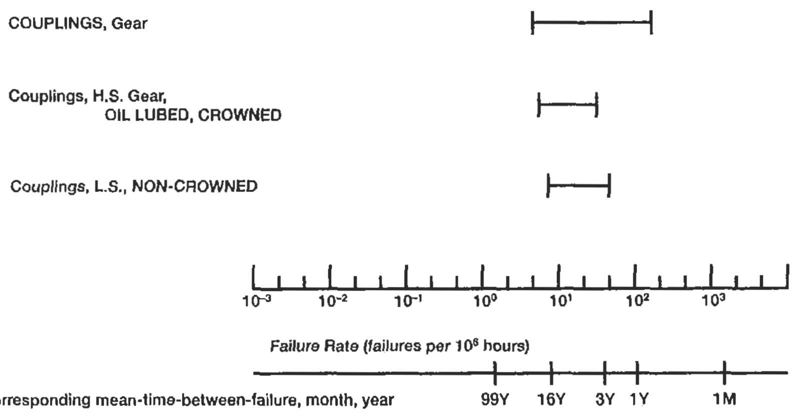


Table 5-5
MTBF Comparison for Direct-Drive Vs. Belt-Drive Blowers
(Based on Component Failure Rate Data from Table 5-2)
Direct-Drive Shows a 20% Higher Average MTBF

Belt-drive	best	worst
	X /1e6 hr	X /1e6 hr
oil seal	8.00	10.00
bearing	5.00	50.00
windings	10.00	20.00
bearing	5.00	50.00
oil seal	8.00	10.00
shaft	0.10	0.50
v-belt	20.00	80.00
shaft	0.10	0.50
oil seal	8.00	10.00
bearing	3.00	10.00
oil seal	8.00	10.00
oil seal	8.00	10.00
bearing	5.00	50.00
gearset	8.00	50.00
bearing	5.00	50.00
oil seal	8.00	10.00
oil seal	8.00	10.00
bearing	5.00	50.00
Total	122.20	481.00
est. MTBF	11.2 months	2.8 months

ave. MTBF 7 months

Direct-drive	best	worst
	X /1e6 hr	X /1e6 hr
oil seal	8.00	10.00
bearing	5.00	50.00
windings	10.00	20.00
bearing	5.00	50.00
oil seal	8.00	10.00
shaft	0.02	0.10
coupling	0.01	0.10
shaft	0.02	0.10
oil seal	8.00	10.00
bearing	3.00	10.00
oil seal	8.00	10.00
oil seal	8.00	10.00
bearing	5.00	50.00
gearset	8.00	50.00
bearing	5.00	50.00
oil seal	8.00	10.00
oil seal	8.00	10.00
bearing	5.00	50.00
	102.05	400.30
	13.4 months	3.4 months

8.4 months

(*text continued from page 263*)

machine MTBF can thus be visualized and appropriate life cycle cost calculations initiated.

Suppose we had a series of centrifugal pumps with typically estimated lives of 2.5 years for mechanical seals, 5 years for ball bearings, 7 years for couplings, and 15 years for shafts. In that case, the anticipated overall MTBF of these pumps will probably be reasonably close to the inverse of

$$\left[\left(\frac{1}{2.5}\right)^2+\left(\frac{1}{5}\right)^2+\left(\frac{1}{7}\right)+\left(\frac{1}{15}\right)^2\right]^{0.5}, \text{ or } 2.11 \text{ years}$$

If bearing housing seals (Figure 5-1 and pages 447–449) were to extend seal and bearing lives to an estimated 3.5 and 10 years, respectively, the anticipated pump MTBF could reasonably be expected to reach the inverse of

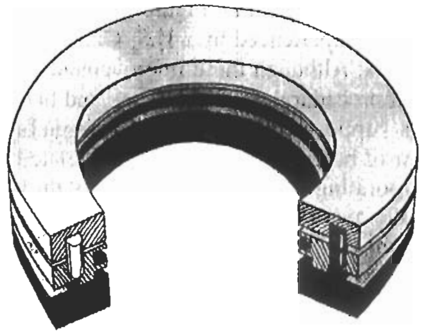


Figure 5-1. INPRO RMS-700 magnetic seal.

$$\left[\left(\frac{1}{3.5}\right)^2+\left(\frac{1}{10}\right)^2+\left(\frac{1}{7}\right)+\left(\frac{1}{15}\right)^2\right]^{0.5}, \text{ or 2.93 years.}$$

The monetary value of this improvement could again be determined from a life cycle cost computation, or it could be stipulated in even simpler terms as a benefit-to-cost calculation.

Benefit-to-Cost Calculations

Perhaps the oldest form of cost justification practiced on a wide scale consists of comparing the cost of an upgrade option with the direct yearly value of maintenance cost avoidance. If we assume that a particular upgrade option, say a set of magnetic bearing housing seals, would cost us $800 and result in shifting the pump MTBF from previously 2.11 years to now 2.93 years, and assuming further that a pump repair costs $7,000 by the time materials, labor, overhead, benefits, spare parts procurement, shop supervision, planning, vibration monitoring, and reliability engineering involvement have all been factored in, our yearly pump repair cost will have dropped from $3,318 ($7,000/2.11) to $2,389 ($7,000/2.93). The ensuing cost savings of $929/year will go on for years, while the one-time outlay of $800 will have a payback of (800/929) 12, or 10.3 months. There is a very important additional benefit to the systematic extension of equipment life that we have not even considered. Instead of getting bogged down in frequent breakdown maintenance tasks, reliability engineers will be able to devote their attention to other reliability improvement opportunities, making money for their employer.

Making Better Use of Published Failure Data

More often than we might think, there have been and will continue to be published, important reliability-related data that we could apply in cost justification and projected life cycle cost studies for our own plant.

One of many such examples would be Figure 5-2, which illustrates the reduction in bearing failures actually experienced by a U.S. Gulf Coast petrochemical company in the span of 4½ years. Although these improvements are undoubtedly attributable to a combination of procedural, organizational, and hardware-specific improvement measures, let us suppose for the sake of reasonable illustration that this downturn in the number of bearing replacements was related to pumps. We will further assume that incorporating improved components during repair events would typically add $500 to the average pump repair cost of $6,700. However, the $500 add-on applies only if implemented when pumps are in the shop for repairs of any kind. It has been estimated that implementation on the basis of *purposely* taking a pump to the shop to effect these improvements would cost $3,470 per pump.

At issue is whether "Unit C" at this facility should implement option 1; i.e., to have its 232 pumps retrofitted with the requisite component upgrades at $3,470 per pump or option 2, "upgrading whenever in the shop for other reasons," or whether option 3, "leaving everything as is" (business as usual) is financially more attractive. This is how we might proceed.

From Figure 5-2, note that the mean failure rate in February 1990 was 17.7 bearing failures per 1,000 pieces of rotating equipment. By March 1994, this mean rate of failure had been reduced to 6.7. Option 1, conversion/upgrading of 232 pumps during the next shutdown would cost $(232) (3470) = $805,040. This one-time expenditure would likely result in yearly savings of (17.7-6.7) (0.232) (12 months) ($6,700) = $205,181. The resulting payback period would be 805,040/205,181 = 3.9 years and savings over a 4½-year period would amount to ($205,181) (4.5) = $923,315.

Next, we'll examine option 2 with the assumption that we could expect to duplicate the published experience of the U.S. Gulf Coast petrochemical company men-

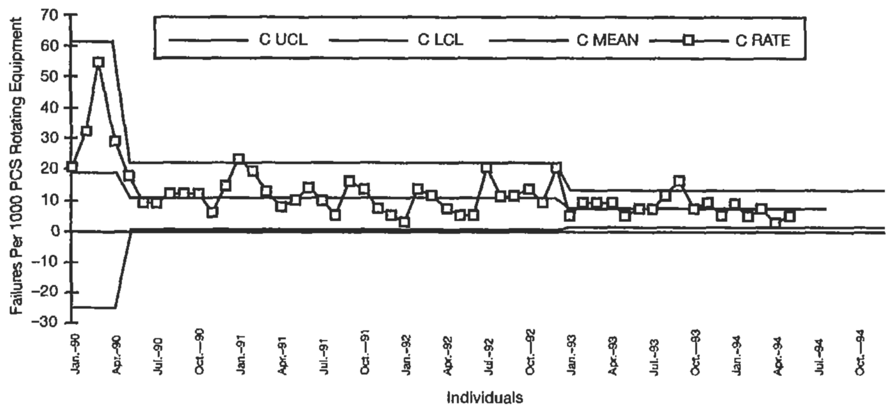


Figure 5-2. Bearing failure rate per 1,000 machines at a U.S. chemical plant.

tioned earlier.[8] This assumption is graphically represented in Figure 5-3. Over a period of 4½ years (54 months), we would anticipate having to repair a total of pumps equal to the shaded area of the diagram:

[(17.7 - 6.7) (54) (232)/(2) (1,000)] + [(6.7) (54) (232)/1,000] = 153 pumps.

Since all of of these repairs would incorporate upgrade components at $500 per pump, our repair expenditures in a 4½-year period would total (153) (7,200) = $1,101,600.

Calculating the cost of option 3, "business as usual," is easiest. Failures would continue at a rate of 17.7 per 1,000 machines per month. In a 54-month period, the cost would be ($6,700) (17.7) (54) (232)/1,000 = $1,485,695. Clearly then, "business as usual" is your most expensive option.

Determining the Value of a Component Upgrading Project

Earlier in this discussion, we had encouraged reliability professionals to extend their horizons by reviewing peer data published worldwide.

In 1992, a British reliability engineer published the results of failure reduction programs at three refineries.[9] As indicated in Figure 5-4, refinery A documented an MTBF increase from 29 months at the end of year 2, and to 71 months at the end of year 7. Accordingly, their pump run lengths experienced an increase of 42 months in the span of 5 years. Since these increases are attributable to upgrade efforts that went beyond seal improvements, we will temporarily put them aside and will focus instead on refineries B and C. The latter two refineries documented seal-related

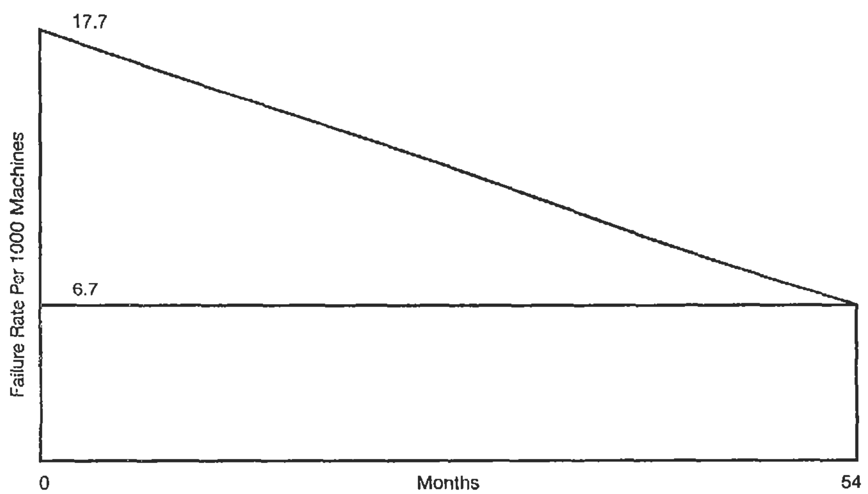


Figure 5-3. Reasonably anticipated pump failure rate reduction due to upgrade efforts.

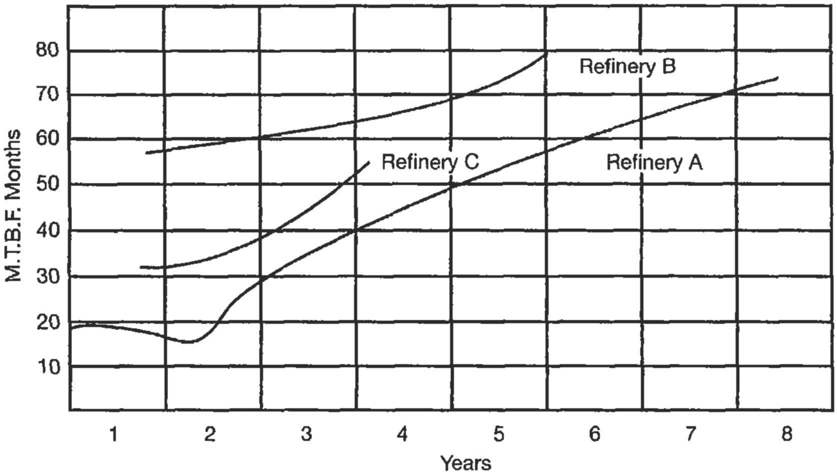


Figure 5-4. Improved mean-time-between-failure (MTBF) at three British oil refineries were attributed to seal upgrade and selection strategy.[9]

MTBF increases of (80-57)/57 = 40% in 4 years (refinery B) and (50-33) 33 = 51% in 2 years (refinery C).

It makes good sense to see more substantial improvement possibilities for the refinery that has the lower MTBF to start with. We note that refinery C started with 33 months seal MFBF and that *our* refinery is presently at 28 months MFBF. Returning to refinery A and their overall pump MTBF, which had increased from 30 months at the end of year 2 to about 71 months at the end of year 7, we calculate an MTBF increase of (71-30)/30 = 36% in 5 years. If we take into account the observation that refineries starting with MTBF figures of 30 months have experienced MTBF increases around 25% per year, we should feel little reluctance assuming that our own plant could go from an MTBF of 28 months to one of 56 months in the span of five years.

Let us, therefore, assume that a refinery wanted to embark on such a mechanical seal MTBF Improvement Program and management requested an appropriately documented and referenced cost and benefit projection.

We have 1,474 centrifugal pumps at our plant site. Our seal MTBF was originally calculated from (1,474 pumps installed) (12 months/yr)/632 seal failures/year = 28 months. Furthermore, it is known that upgrading to superior seal configurations and improved seal materials would add $1,700 to each pump repair and that typical pump repairs, using traditional grade seals, would cost approximately $5,000.

Assuming a linear MTBF increase from 28 months today to 56 months five years from now, we might now opt to calculate our yearly repair cost outlay in the most straight-forward manner and list our results in Table 5-6.

Table 5-7 highlights a similar approach that can be used to estimate the economic justification for retrofitting small pumps with new casing covers.[10] These "back-pull-

Table 5-6
Assessing the Monetary Value of Pump Upgrading

Pump Population: 1474
Interest Rate: 6.00%

	Now	PV	Year 1	Year 2	Year 3	Year 4	Year 5
Projected MTBF, (1474)(12)/x	28.0		33.6	39.2	44.8	50.4	56.0
Projected Repairs, x	632		526	451	394	350	315
Repair Costs—Option 1, (5000)(x)	3,160,000	13,311,070	3,160,000	3,160,000	3,160,000	3,160,000	3,160,000
Repair Costs—Option 2, (6700)(x)		11,664,994	3,524,200	3,021,700	2,639,800	2,345,000	2,110,500

5-Year Total w/o Upgrading: $(3,160,000)(5) = $15,800,000

5-Year Total with Upgrading: = $13,641,200

Straight Savings over 5 Years: = $2,158,700

Savings Based on Present Value: $1, 646,075

Table 5-7
Estimating the Economic Justification for ANSI-PLUS Retrofits

Estimating The Economic Justification For ANSI-Plus Retrofits		
	Example Plant	Your Plant
1. Total number of ANSI pumps installed	417	
2. Number of ANSI pumps previously repaired each year, all causes	212	
3. Average cost of each repair (direct labor, materials, associated costs, overhead)	$3,122	
4. Number of pumps failing, Month #1, and being retrofitted with ANSI-PLUS parts	18	
5. Failure projections of remainder of 2-year conversion period: 17 repairs Month #2 16 repairs Month #3 16 repairs Month #4 16 repairs Month #5 15 repairs Month #6 15 repairs Month #7 14 repairs Month #8 14 repairs Month #9 13 repairs Month #10 13 repairs Month #11 12 repairs Month #12 12 repairs Month #13		
12 repairs Month #14 12 repairs Month #15 11 repairs Month #16 11 repairs Month #17 11 repairs Month #18 11 repairs Month #19 10 repairs Month #20 10 repairs Month #21 10 repairs Month #22 10 repairs Month #23 9 repairs Month #24 and every month thereafter. Total pump repairs in a 2 year period:	308	
6. Cost difference, average, for repairing with ANSI-PLUS retrofit instead of conventional repair parts	$1,120	
7. Additional maintenance cost outlay for installing ANSI-PLUS retrofit parts instead of conventional repairs: (Item 5 x Item 6)	$344,960	
8. Avoided repairs in 24-month time period [(2 x Item 2) – Item 5]	116	
9. Value of avoided repairs in 24-month time period. (Item 3 x Item 8)	$362,152	

out" pumps were originally furnished with conventional stuffing boxes that could accommodate only relatively small diameter mechanical seals. Whenever one of the plant's 417 pumps undergoes a shop repair, new covers with considerably more favorable seal housing dimensions are fitted to the casing. The bottom line results show that after two years of routinely upgrading in this manner, the value of avoided repairs ($362,152) exceeds the additional maintenance cost outlay ($344,960) for installing the pump enhancements.

Many Different Cost Justification Methods are Available to the Reliability Professional

We have attempted to show how a number of straightforward calculation approaches can be, and are being applied, to determine life cycle costs, cost-benefit ratios, or payback periods for reliability improvements in process plants. A resourceful reliability professional will, of course, diligently collect and compile failure statistics for equipment and components at his or her plant site. However, this diligent professional would continue to reach out for other data sources to augment and validate in-house data. It should be obvious that much of these "other" data could be used for the purpose of setting goals and would allow comparisons among plants or industry segments.

Finally, the occasional *lack* of data need not be a deterrent to making judicious assumptions and well-explained "educated guesses." While our estimates may sometimes be a bit off the mark, pursuing the various cost justification options will always be better than the status quo, or a complacent "business-as-usual" approach to equipment reliability improvement.

Life Cycle Cost Assessment: The Rigorous Method*

Life cycle costs (LCC) are summations of cost estimates from inception to disposal for both equipment and projects as determined by an analytical study and estimate of total costs experienced during the lifetime of the equipment. The objective of LCC analysis is to choose the most cost-effective approach from a series of alternatives so the least long-term cost of ownership is achieved.

LCC analysis helps engineers justify equipment and process selection based on total costs rather than the initial purchase price. Usually the cost of operation, maintenance, and disposal costs exceed all other costs many times over. Life cycle costs are the total costs estimated to be incurred in the design, development, production, operation, maintenance, support, and final disposition of a major system over its anticipated useful life span. The best balance among cost elements is achieved when the total LCC is minimized. As with most engineering tools, LCC provides best results when both art and science are merged with good judgment.

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*Contributed by H. Paul Barringer, P. E., Barringer & Associates, Inc., Humble, TX and David P. Weber, D. Weber Systems, Inc., Maineville OH.

Introduction

Procurement costs are widely used as the primary (and sometimes only) criteria for equipment or system selection. This single purpose criterion is simple to use but often results in bad financial decisions. Procurement costs tell only one part of the story—most frequently the story is so simple, the results may be damaging to the financial well being of the business enterprise. Often the initial procurement costs, based on simple rules, are so cheap they are not affordable. Simple tools (meaning composed of only one thing) usually give simple results (meaning insubstantial, superficial, and not to be taken seriously). Remember the adage: "It's unwise to pay too much, but it's foolish to spend too little." This is the operating principle of LCC. End users and suppliers of equipment can use life cycle costs for:

- **Affordability studies**—measure the impact of a system or project's LCC on long-term budgets and operating results.
- **Source selection studies**—compare estimated LCC among competing systems or suppliers of goods and services.
- **Design trade-offs**—influence design aspects of plants and equipment that directly impact LCC.
- **Repair level analysis**—quantify maintenance demands and costs rather than using rules of thumb such as ". . . maintenance costs ought to be less than "x" % of the capital cost of the equipment."
- **Warranty and repair costs**—suppliers of goods and services along with end-users need to understand the cost of early failures in equipment selection and use.
- **Suppliers' sales strategies**—can merge specific equipment grades with general operating experience and end-user failure rates using LCC to sell for best benefits rather than just selling on the attributes of low, first cost.

This chapter is directed toward making LCC understandable and usable by the average engineer. Usually the only value in the life cycle cost equation that is well known and clearly identified is procurement cost—but that's only the tip of the iceberg. Seeing the tip of an iceberg (similar to the obviousness of procurement cost) does not guarantee clear and safe passage around an iceberg. Hidden, underlying, substructures of an iceberg (similar to the bulk of other costs associated with life cycle costing for equipment and systems) contain the hazards.

Life cycle cost studies gained prominence in the mid-1960s when LCC was the subject of considerable interest and publications.

Technical societies such as the Society of Automotive Engineers include life cycle costs in the *RMS Guidebook* (SAE 1995) with a convenient summary of the principles. Also, the Institute of Industrial Engineers includes a short section on life cycles and how they relate to life cycle costs in the *Handbook of Industrial Engineering* (IEE 1992).

The limitations of LCC are accepted in the same manner as are normal restrictions on other engineering tools. Usefulness has been demonstrated by passing the test of time with practitioners who have learned how to minimize LCC limitations. As with all cost techniques (and typical of all engineering tools), the limitations can result in substantial setbacks when judgment is not used. Here are some of the most often cited LCC limitations, some true, some imagined:

- LCC is not an exact science. Everyone gets different answers and the answers are neither wrong nor right—only reasonable or unreasonable. LCC experts do not exist because the subjects are too broad and too deep.
- LCC outputs are only estimates and can never be more accurate than the inputs and the intervals used for the estimates. This is particularly true for cost-risk analysis.
- LCC estimates lack accuracy. Errors in accuracy are difficult to measure as the variances obtained by statistical methods are often large.
- LCC models operate with limited cost databases and the cost of acquiring data in the operating and support areas is both difficult to obtain and expensive to acquire.
- LCC cost models must be calibrated to be highly useful.
- LCC models require volumes of data and often only a few handfuls of data exist, and most of the available data is suspect.
- LCC requires a scenario for how the money expenditure model will be constructed for acquisition of equipment, how the model will age with use, how damage will occur, how learning curves for repairs and replacements will occur, how cost processors will function (design costs, labor costs, material costs, parts consumption, spare parts costs, shipping costs, scheduled and unscheduled maintenance costs) for each time period, how many years the model will survive, how many units will be produced/sold, and similar details required for building cost scenarios. Most details require extensive extrapolations and obtaining facts is difficult.
- LCC models (by sellers) and cost-of-ownership (COO) models (by end users) have credibility gaps caused by using different values in each model. Often credibility issues center on which is right and which is wrong (a win-lose issue) rather than harmonizing both models (for a win-win effort) using available data.
- LCC results are not good budgeting tools. They're effective only as comparison/trade-off tools. Producing good LCC results requires a project team approach because specialized expertise is needed.
- LCC should be an integral part of the design and support process to design for the lowest long-term cost of ownership. End users can use LCC for affordability studies, source selection studies of competing systems, warranty pricing and cost-effectiveness studies. Suppliers find LCC useful for identifying costs drivers and ranking the comparison of competing designs and support approaches.
- LCC, unfortunately, is only useful for Department of Defense (DoD) projects and is seldom applied to commercial areas because few practitioners exist for preparing LCC.

Remember this adage when considering LCC limitations: In the land of the blind, a one-eyed man is king! LCC can help improve our blinded sight. We don't need the most wonderful sight in the world, it just needs to be more acute than our fiercest competitor so that we have an improvement in the cost of operating our plants. DoD tools and techniques are frequently used effectively in commercial areas and this is true of life cycle costing.

Why Use LCC?

LCC helps change provincial perspectives for business issues with emphasis on enhancing economic competitiveness by working for the lowest long-term cost of ownership. Too often parochial views result in ineffective actions best characterized by short-term cost advantages (but long-term costly decisions). Consider these typical events observed in most companies:

- **Engineering** wants to meet capital budgets. Hence, the engineering function avoids specifying cost effective, redundant equipment needed to accommodate expected costly failures.
- **Purchasing** buys lower grade equipment to get favorable purchase prices.
- **Project engineering** builds plants with a view towards successfully running the plant only during startup and a few months beyond rather than taking the long-term view of low-cost operation.
- **Process engineering** employs the philosophy that all equipment is capable of operating at 150% of its rated condition without failure and other departments will be responsible for remedying equipment abuse.
- **Maintenance** defers required corrective/preventive actions to reduce budgets. Long term costs increase because of neglect and for the sake of meeting short-term management gains.
- **Reliability engineering** is assigned improvement tasks with no budgets for accomplishing the goals.

Management is responsible for harmonizing these potential conflicts under the banner of operating for the lowest long-term cost of ownership. The glue binding these conflicts together is a teamwork approach for minimizing LCC. When properly used with good engineering judgment, LCC provides a rich set of information for making cost-effective, long-term decisions. LCC can be used as a management decision tool for:

- **Costing discipline**—concerned with operating and support cost estimates.
- **Procurement technique**—used as a tool to determine cost per usage.
- **Acquisition tool**—concerned with balancing acquisition and ownership costs.
- **Design trade-off**—integrates effects of availability, reliability, maintainability, capability, and system effectiveness into x-y charts that are understandable for cost-effective screening methods.

Be aware that financial performance measures are as numerous as engineering measures. What really counts for owners and shareholders is return on capital employed and economic profit derived from the enterprise. If too much is spent for capital, achieving appropriate returns on the capital is more challenging and likewise economic profit is too low for generating adequate cash returns.

Every business has a minimum rate of return for projects and this rate of return should be substantially higher than borrowing rates for money. If the minimum attractive rate of return is set too high, then many reasonably good projects get disqualified. If the minimum rate of return is set too low, then too many marginal projects get accepted and the business becomes a "bank." Projects are typically screened against a minimum rate that also changes with time and conditions. For example, if the cost of money to a corporation is 9%, then the minimum attractive rate of return may be at least 12% to merit consideration for a successful project. It's unwise to champion projects with rates of return less than the minimum attractive rate, and sometimes the best project may involve the alternative of doing nothing rather than buying into a poorly performing project. The objective of successful projects is to find opportunities that are worth much more than they cost over time. Projects must exceed the minimum attractive rates of return so wealth is created for the stockholders.

Economic calculations are well defined but the most difficult financial question is what discount rate should be used. Accounting and finance organizations set internal discount rates to make economic decisions easy for engineers (remember, the discount rate is always changing). Discount factors reflect a host of relationships and considerations that include very low risk investment returns such as U.S. government T-bills, factors for projects such as estimated uncertainties, internal rates of returns, and so forth. Discount factors vary by company and over time. In general, consider a typical discount value of 12%, which is neither very low nor very high for the calculations that will follow. Using the discount rate of 12%, consider the results for two questions using $FV = PV^* (1 + i)^n$, where FV is future value, PV is present value, i is discount rate, and n is number of years into the future:

1) What is the present value (PV) of US$1.00 today over time?
2) What is the future value (FV) of US$1.00 received over time?

Cash flows into and out of a business according to cash outlays and receipts of business transactions. The discounting method is used to summarize transactions over the life of the investment in terms of present or future dollars. Discount rates in Table 5-8 are used as multipliers or dividers to put financial transactions into the present value of money to answer the two questions posed above.

Net present value (NPV) is an important economic measure for projects or equipment taking into account discount factors and cash flow. The present value (PV) of an investment is the maximum amount a firm could pay for the opportunity of making the investment without being financially handicapped. The net present value (NPV) is the present value of proceeds minus present value of outlays. Net present value calculations start with a discount rate, followed by finding the present value of the cash proceeds expected from the investment, then followed by finding the present value of the outlays. The net of this calculation is the net present value. High NPV projects and processes provide wealth for the shareholders. Cash availability and strategies aside, when competing projects are judged for acceptance, the project with the greatest NPV is usually the winner.

Cash flow is very important to any enterprise. Positive cash flow into the company assures its continuing existence. The concept is simple: no cash, no company! One project can't borrow cash from another project; hence, all cash generating actions are usually judged by themselves. The term cash flow is generalized and refers to the flow of money. Cash flow is not the same as the accounting terms "profit" and "income." For projects, the general view is that cash flows out in one or more years

Table 5-8
Present Value and Future Value

Discount Rate = 12%																					
Years hence	0	1	2	3	4	5	6	7	8	9	10	11	12	13	14	16	16	17	18	19	20
Present value of US$1.00	1.00	0.89	0.80	0.71	0.64	0.57	0.51	0.45	0.40	0.36	0.32	0.29	0.26	0.23	0.20	0.18	0.16	0.15	0.13	0.12	0.10
Future value of US$1.00	1.00	1.12	1.25	1.40	1.57	1.76	1.97	2.21	2.48	2.77	3.11	3.48	3.90	4.36	4.89	5.47	6.13	6.87	7.69	8.61	9.65

and cash begins to flow in for a series of many years. The amount of cash "thrown off" by a project is an important consideration and a helpful criterion for evaluating projects. For many projects cash flow results from cost savings, depreciation, and taxes. Of course, depreciation schedules change as accounting departments select the schedule that legally results in the greatest profits. Most accounting departments use the same general form for calculating relative changes in cash flow, although specific details are highly variable. For example, the straight line depreciation schedule may be used for accounting profit purposes and accelerated depreciation for tax purposes. Depreciation is a non-cash cost and must be excluded or added back to determine actual cash flow. Cash flows (after taxes to get the real flow of cash) in each period are adjusted by a discount factor to calculate present value for each year. The net present value is the sum of all present values for the allowed time periods.

Most fixed assets and other projects have a limited useful life. Accounting practices gradually change fixed assets into expense with a process called depreciation over the accepted long life of an asset. All equipment has a finite life based on both deterioration and obsolescence. Judgment is required in estimating and setting actual service life of assets. Two common methods are used for calculating depreciation based on acquisition cost less salvage:

1. The straight line method is based on consumption of a fixed percentage of the equipment cost. Often straight-line depreciation is used for internal accounting reports of profit/loss.
2. The accelerated method is based on the amount of service provided when a larger amount of depreciation is consumed in the early years and the depreciation for each year is found by applying a rate to the book value of the asset at the beginning of that year rather than to the original cost of the asset. Book value is cost less total depreciation accumulated up to that time. Accelerated depreciation is often used for tax and cash reporting purposes. Depreciation methods are different for accounting and tax considerations. Remember that depreciation is non-cash and is only a process of allocation to future periods. For the calculations below, the straight-line depreciation schedule will be used, and Table 5-9 shows the contrast between straight-line and double-declining-balance depreciation.

Income tax rates vary and may require inclusion of state as well as federal taxes. For calculation purposes, consider the tax rate is 38% based on the profit before tax numbers. Profit before taxes may be positive or negative. When profit before tax is negative, the company receives a tax credit either a carry-back or carry-forward.

Table 5-9
Straight Line and Double Declining Depreciation

Depreciation Schedule For A US$1 Investment																						Depreciation
Years hence	0	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	Total
Straight line	0.00	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	1.00
Double declining	0.00	0.10	0.09	0.08	0.07	0.07	0.06	0.05	0.05	0.04	0.04	0.03	0.03	0.03	0.03	0.02	0.02	0.02	0.02	0.02	0.01	0.88

When profit before tax is positive, the company pays taxes. For a project or process, tax numbers are used to calculate cash flows. After the tax is included, the cash flow is discounted to get present value, and the sum of all present values gives the NPV.

When the net present values are known for the project life, then the discounted cash flow (DCF) rate can be calculated to arrive at a profitability index. The discounted cash flow rate is the return that forces the NPV to zero. It's the maximum cost of capital that can be paid just to break even on the project. The DCF index defines an economical quality value for the project and is useful for comparing projects of different sizes. Large DCFs are desirable but small investments and big savings result in numbers that are often questioned. The DCF index sometimes has problems in ranking project desirability so base final decisions on the NPV. Of course, this assumes a common time period for the life of the items.

If equipment life among alternatives is not the same, then a more complicated analysis is required to divide the NPV by an annuity factor. The annuity factor depends on equipment lifetime, discount rates, and equivalent annual cash flow to correct for unequal equipment life. This calculation puts NPV alternatives on an equivalent annual basis using an annuity factor (AF) where $AF = [((1 + i)^N - 1)/i*(1 + i)^N]$ and i = discount rate, N = equipment life. The annual equivalent NPV, (AENPV) is found by AENPV = NPV/AF. For example, if two competing projects each project a NPV = $150,000 using a DCF = 12%, and case 1 has equipment life of five years while case 2 has a life of ten years, by common sense, case 1 is preferable. But here is how it works out: $AF_{N=5} = 3.605$ and $AF_{N=10} = 5.650$ so that $AENPV_{N=5}$ = $41,611 and $AENPV_{N=10}$ = $26,547. This shows the five-year life case is 1.6 fold more attractive.

Engineers must be concerned with life cycle costs for making important economic decisions through engineering actions. Management deplores engineers who are engineering smart but economics stupid. Engineers must get the equation balanced to create wealth for shareholders. Often this means: **stop** doing some things the old way, and **start** doing new things in smarter ways.

Example 1 shown below illustrates the above ideas and concepts for a financial analysis. This example is **not** a LCC model but it is typical of how equipment is justified:

Example: Two alternatives are being considered for installing an on-line spare pump in parallel with an existing ANSI grade pump to avoid outages that have plagued a chemical plant. The parallel pump (which will be operated every other week on a rotation schedule and whenever pump failure occurs—this is an incremental investment/operation) will save, on the average, US$12,000 per year in out-of-pocket production losses for products that cannot be shipped during the outages. Pumps under consideration, with expected 20 year lives, are 1) another ANSI pump at US$8,000 installed cost or 2) an ANSI enhanced pump at US$18,000 installed cost. Another choice is 3) do nothing. Which course of action should we recommend using the concepts described above? Consider alternatives given in Table 5-10; note year 0 is now, and year 1 is next year. By this financial analysis, installing an ANSI pump results in the largest NPV.

These spreadsheets are the usual justifications prepared by accounting departments; however, this analysis **does not** take into account life cycle costs. Accounting departments will use this technique unless engineers provide details about how equipment survives or dies in operating environments. Adding expected failure rates and renewals makes the accounting analysis smarter and gets the analysis closer to real world conditions.

Should the ANSI pump be installed based on the favorable NPV? That depends upon the company's demand for cash and other details to be described below.

What Goes Into Life Cycle Costs?

LCC includes every cost that is appropriate, and appropriateness changes with each specific case that is tailored to fit the situation. LCC follows the process shown in Figure 5-5. The basic tree for LCC starts with a very simple tree based on the costs for acquisition and the costs for sustaining the acquisition during its life is shown in Figure 5-6.

Acquisition and sustaining costs are not mutually exclusive. Whenever equipment or processes are acquired, it must be understood that they always require extra costs to sustain the acquisition. Acquisition and sustaining costs are found by gathering the correct inputs, building the input database, evaluating the LCC, and conducting a sensitivity analysis to identify cost drivers.

Frequently the cost of sustaining equipment is two to twenty times the acquisition cost. Consider the cost for a simple ANSI pump. The power cost for driving the pump during its lifetime is many times larger than the acquisition cost of the pump. Are ANSI pumps bought with an emphasis on energy efficient drivers and energy efficient rotating parts, or is the acquisition simply based on the lowest purchase price?

The often-cited rule of thumb is 65% of the total LCC is set when the equipment is specified! This means do not consider the specification process lightly. Realize the first obvious cost (hardware acquisition) is usually the smallest amount of cash that will be spent during the life of the acquisition and most sustaining expenses are not obvious. Every example has its own unique set of costs and problems to solve for minimizing LCC. Minimizing LCC pushes up NPV and creates wealth for shareholders. Finding LCC requires finding details for both acquisition and sustaining costs with many details involved in the effort.

(text continued on page 282)

Table 5-10
Financial NPV Without LCC Content

	Year																				
	0	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
Alternative #1-ANSI Pump																					
Capital Cost	8000																				
Savings		12000	12000	12000	12000	12000	12000	12000	12000	12000	12000	12000	12000	12000	12000	12000	12000	12000	12000	12000	12000
Depreciation		400	400	400	400	400	400	400	400	400	400	400	400	400	400	400	400	400	400	400	400
Profit b/4 taxes		11600	11600	11600	11600	11600	11600	11600	11600	11600	11600	11600	11600	11600	11600	11600	11600	11600	11600	11600	11600
Tax Provision		-4408	-4408	-4408	-4408	-4408	-4408	-4408	-4408	-4408	-4408	-4408	-4408	-4408	-4408	-4408	-4408	-4408	-4408	-4408	-4408
Net Income		7192	7192	7192	7192	7192	7192	7192	7192	7192	7192	7192	7192	7192	7192	7192	7192	7192	7192	7192	7192
Add Back Depreciation		400	400	400	400	400	400	400	400	400	400	400	400	400	400	400	400	400	400	400	400
Cash Flow	-8000	7592	7592	7592	7592	7592	7592	7592	7592	7592	7592	7592	7592	7592	7592	7592	7592	7592	7592	7592	7592
Discount Factors	1.00	1.12	1.25	1.40	1.57	1.76	1.97	2.21	2.48	2.77	3.11	3.48	3.90	4.36	4.89	5.47	6.13	6.87	7.69	8.61	9.65
Present Value	-8000	6779	6052	5404	4825	4308	3846	3434	3066	2738	2444	2183	1949	1740	1553	1387	1238	1106	987	881	787
Net Present Value	$48,708	using a 12% discount rate																			
Alternative #2-ANSI Enhanced Pump																					
Capital Cost	18000																				
Savings		12000	12000	12000	12000	12000	12000	12000	12000	12000	12000	12000	12000	12000	12000	12000	12000	12000	12000	12000	12000
Depreciation		900	900	900	900	900	900	900	900	900	900	900	900	900	900	900	900	900	900	900	900
Profit b/4 taxes		11100	11100	11100	11100	11100	11100	11100	11100	11100	11100	11100	11100	11100	11100	11100	11100	11100	11100	11100	11100
Tax Provision		-4218	-4218	-4218	-4218	-4218	-4218	-4218	-4218	-4218	-4218	-4218	-4218	-4218	-4218	-4218	-4218	-4218	-4218	-4218	-4218
Net Income		6882	6882	6882	6882	6882	6882	6882	6882	6882	6882	6882	6882	6882	6882	6882	6882	6882	6882	6882	6882
Add Back Depreciation		900	900	900	900	900	900	900	900	900	900	900	900	900	900	900	900	900	900	900	900
Cash Flow	-18000	7782	7782	7782	7782	7782	7782	7782	7782	7782	7782	7782	7782	7782	7782	7782	7782	7782	7782	7782	7782
Discount Factors	1.00	1.12	1.25	1.40	1.57	1.76	1.97	2.21	2.48	2.77	3.11	3.48	3.90	4.36	4.89	5.47	6.13	6.87	7.69	8.61	9.65
Present Value	-18000	6948	6204	5539	4946	4416	3943	3520	3143	2806	2506	2237	1997	1783	1592	1422	1269	1133	1012	904	807
Net Present Value	$40,127	using a 12% discount rate																			
Alternative #3-Do Nothing																					
Capital Cost	0																				
Savings		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Depreciation		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Profit b/4 taxes		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Tax Provision		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Net Income		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Add Back Depreciation		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Cash Flow	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Discount Factors	1.00	1.12	1.25	1.40	1.57	1.76	1.97	2.21	2.48	2.77	3.11	3.48	3.90	4.36	4.89	5.47	6.13	6.87	7.69	8.61	9.65
Present Value	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Net Present Value	$ -	using a 12% discount rate																			

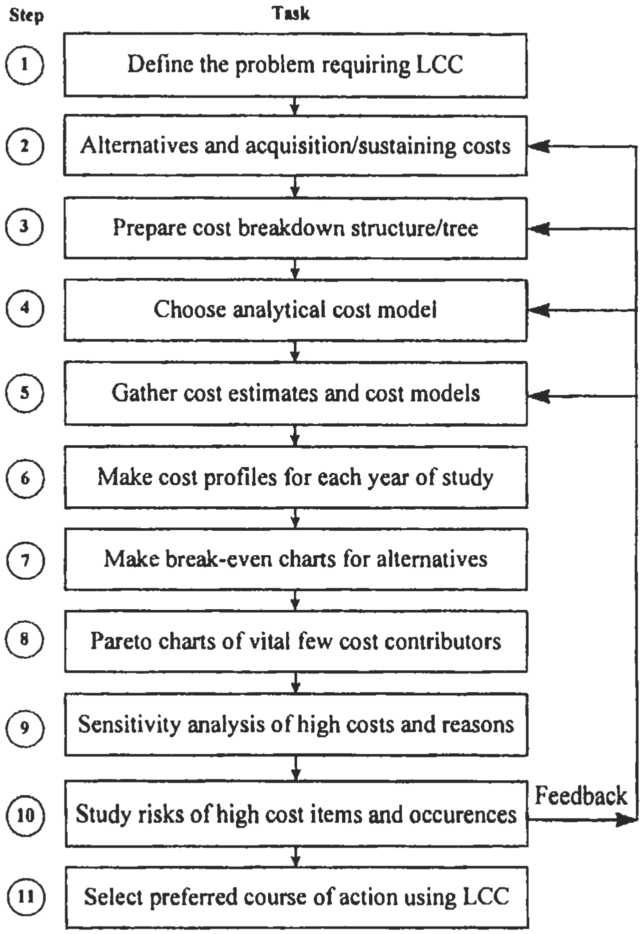


Figure 5-5. Life cycle costing process.

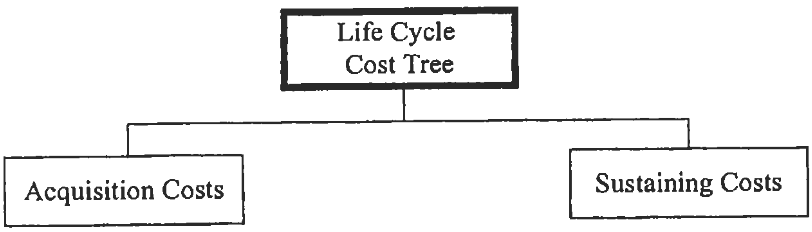


Figure 5-6. Top levels of LCC tree.

(text continued from page 279)

Acquisition costs have several branches for the tree as shown in Figure 5-7.

Each branch of the acquisition tree also has other branches that are described in detail in such references as SAE (1993) and Fabrycky (1991).

Sustaining costs have several branches for the tree as shown in Figure 5-8.

What cost goes into each branch of the acquisition and sustaining branches? It all depends on the specific case and is generally driven by common sense. Consider the details under each category.

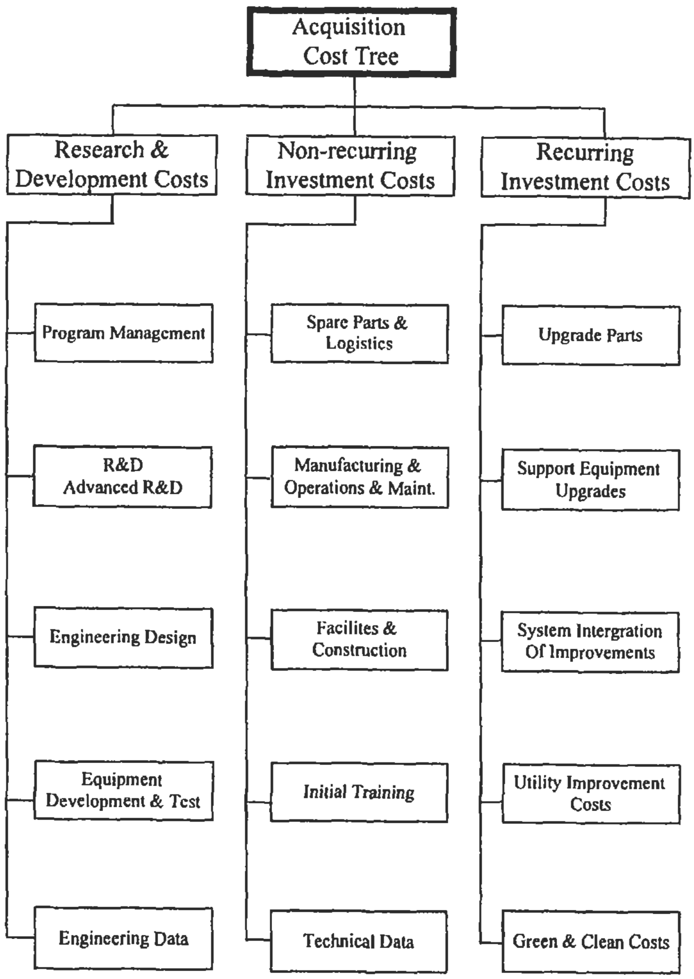


Figure 5-7. Acquisition cost tree.

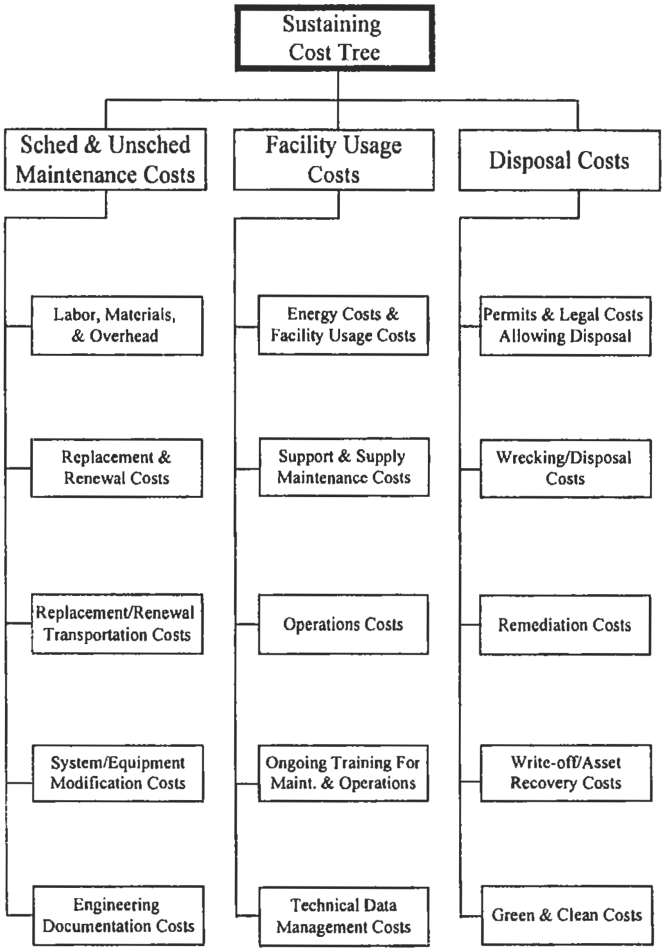


Figure 5-8. Sustaining cost tree.

There probably are special categories under each item of acquisition cost and sustaining cost. Building a pulp and paper mill or modifying coker drums at a refinery to prevent characteristic overstress that occurs during quench cycles would have different cost structures than those used for building a nuclear reactor. Include the appropriate cost elements and discard the elements that do not substantially influence LCC. Consider these alternative LCC models as descried by Raheja (1991):

1. LCC = nonrecurring costs + recurring costs
2. LCC = initial price + warranty costs + repair, maintenance, and operating costs to end users

3. LCC = manufacturer's cost + maintenance costs and downtime costs to end users

SAE (1993) also has a LCC model directed toward a manufacturing environment:

4. LCC = acquisition costs + operating costs + scheduled maintenance + unscheduled maintenance + conversion/decommission

The SAE model breaks down the costs as shown in Figure 5-9.

The LCC models above, and much more complicated models described in the British Standards BS-5760 (BSI 1983), include costs to suppliers, end users, and "innocent bystanders"—in short, the costs are viewed from a total systems perspective. LCC vary with events, time, and conditions. Many cost variables are not deterministic but are truly probabilistic. This usually requires starting with arithmetic values for cost and then growing the cost numbers into the more accurate, but more complicated, probabilistic values.

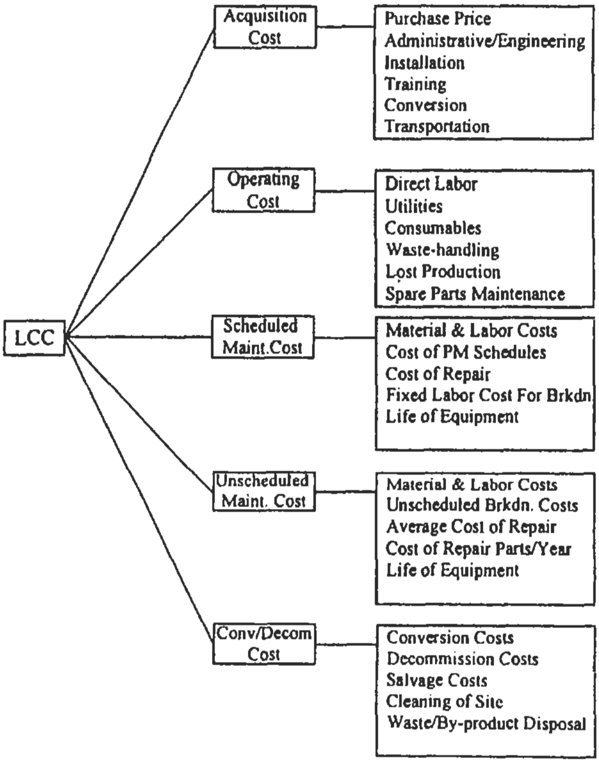


Figure 5-9. Life cycle cost breakdown per SAE model.

Trade-Off Tools For LCC

One helpful tool for easing LCC calculations involving probabilities is the effectiveness equation that gives a figure of merit for judging the chances of producing the intended results. The effectiveness equation is described in several different formats in which each element varies as a probability and the issue is finding a system effectiveness value that gives lowest long-term cost of ownership:

System effectiveness = effectiveness/LCC

Cost is a measure of resource usage (cost estimates can never include all possible elements but hopefully include the most important elements). Effectiveness is a measure of value received (effectiveness rarely includes all value elements as many are too difficult to quantify) and effectiveness varies from 0 to 1:

Effectiveness = availability * reliability * maintainability * capability
= availability * reliability * performance (maintainability * capability)
= availability * dependability (reliability * maintainability) * capability

In plain English, the effectiveness equation is the product of the chance the equipment or system will be available to perform its duty, will operate for a given time without failure, can be repaired without excessive maintenance loss time, and can perform its intended production activity according to the standard. Each element of the effectiveness equation is premised on a firm datum that changes with name plate ratings to obtain a true value that lies between 0 and 1:

- Availability deals with the duration of uptime for operations and is a measure of *how often* the system is alive and well. It is often expressed as (uptime)/(uptime + downtime) with many different variants. Also availability may be the product of many different terms such as

  $$A = A_{\text{hardware}} * A_{\text{software}} * A_{\text{humans}} * A_{\text{interfaces}} * A_{\text{process}}$$

  and similar configurations. Availability issues deal with at least three main factors (Davidson 1988) for 1) increasing time to failure, 2) decreasing downtime due to repairs or scheduled maintenance, and 3) accomplishing items 1 and 2 in a cost-effective manner as the higher the availability, the greater is the capacity for making money because the equipment has a higher in-service life.
- Reliability deals with reducing the frequency of failures over a time interval and is a measure of *the odds for failure-free operation* during a given interval, i.e., it is a measure of success for a failure free operation. It is often expressed as

  $$R(t) = \exp(-t/\text{MTBF}) = \exp(-\lambda t)$$

  where λ is failure rate and MTBF is mean time between failure. MTBF measures how often the system will fail. MTBF is a basic figure of merit for reliability (or failure rate that is the reciprocal of MTBF) for exponential failure modes. Also reliability may be the product of many different terms such as

$$R = R_{utilities} * R_{feed\ plant} * R_{processing} * R_{packaging} * R_{shipping}$$

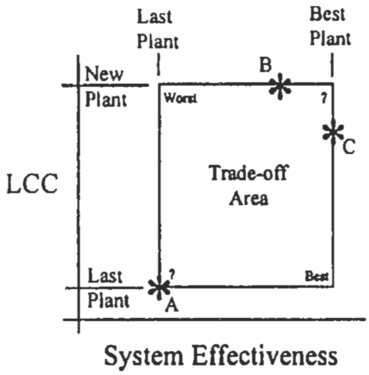
and similar configurations. To the user of a product, reliability is measured by problem-free operation (resulting in increased productive capability while requiring fewer spare parts and less manpower for maintenance activities, which results in lower costs). To the supplier of a product, reliability is measured by completing a failure-free warranty period under specified operating conditions. Improving reliability occurs at an increased capital cost but brings with it the expectation for improving availability, decreasing downtime and associated maintenance costs, improved secondary failure costs, and results in a better chance for making money because the equipment is free from failures for longer periods of time.

- Maintainability deals with duration of maintenance outages or *how long* it takes to achieve the maintenance actions compared to a datum. The key figure of merit is often the mean time to repair (MTTR), which measures the ease of maintenance upon failure. On a qualitative basis, it refers to the ease with which hardware or software is restored to a functioning state. On a quantitative basis, it has probabilities as described for availability and is measured based on the total downtime that includes all diagnosis, troubleshooting, teardown, removal/replacement, active repair time, verification that the repair is adequate, time delays for logistic movements, and administrative maintenance delays. Maintainability is different than repairability which is the probability a failed item is restored to operable condition within a specified active repair time.
- Capability deals with productive output compared to inherent productive output, which is a measure of *how well* the production activity is performed compared to the datum. This index measures the systems capability to perform the intended function on a system basis. Often the term is synonymous with productivity, which is the product of efficiency times utilization. Efficiency measures the productive work output versus the work input; whereas, utilization is the ratio of time spent on productive efforts to the total time consumed.
- Dependability is the product of reliability and maintainability. It measures *how long* things perform. Related issues about nonoperational influences are not included..

System effectiveness equations are helpful for understanding benchmarks, past, present, and future status as shown in Figure 5-10 for understanding trade-off information.

The lower right-hand corner of Figure 5-10 brings much joy and happiness often described as "bang for the buck." The upper left-hand corner brings much grief. The remaining two corners raise questions about worth and value. The system effectiveness equation is useful for trade-off studies as shown in the attached outcomes in Figure 5-11.

System effectiveness equations have great impact on the LCC because so many decisions made in the early periods of a project carve the value of LCC into stone. About two thirds of the total LCC are fixed during project conception even though



Parameter	A Last Plant	B New Plant	C Best Plant
Availability	0.95	0.95	0.98
Reliability	0.3	0.4	0.6
Maintainability	0.7	0.7	0.7
Capability	0.7	0.8	0.6
Sys. Effect.	0.14	0.22	0.25
LCC	80	100	95

Figure 5-10. Benchmark data shown in trade-off format.

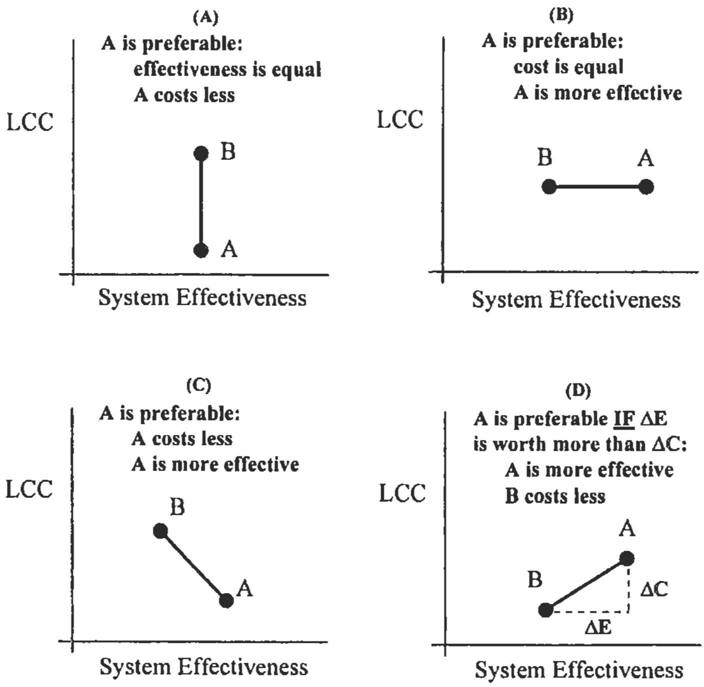


Figure 5-11. Some possible outcomes from trade-off studies.

expenditure of funds will flow at a later time, and the chance to influence LCC cost reductions grows smaller as shown in Figure 5-12.

Engineering sizes and aims the cost funnel, and production/maintenance pours money into the funnel. Consider LCC early in the game when the final outcome can be influenced for better business results. Making major changes in LCC when the project is turned over to production is not possible because the die has been cast. Breaking poverty cycles of building cheap plants and repairing them often—at great expense—can be accomplished in at least two ways: 1) use LCC techniques, or 2) make the capital project team indentured servants for at least 8 years to operate the plant so that new projects are designed for the least long-term costs of ownership so it builds wealth for stockholders. Either method is effective at producing wealth because thoughtful, value judgments are used rather than minimizing first cost only to get high long-term cost of ownership. However, facts are required for getting the action started for improvements!

Engineering Facts

LCC requires facts that are driven by data. Most engineers are of the opinion they lack data. In fact, data is widely available as a starting point for LCC. Often data

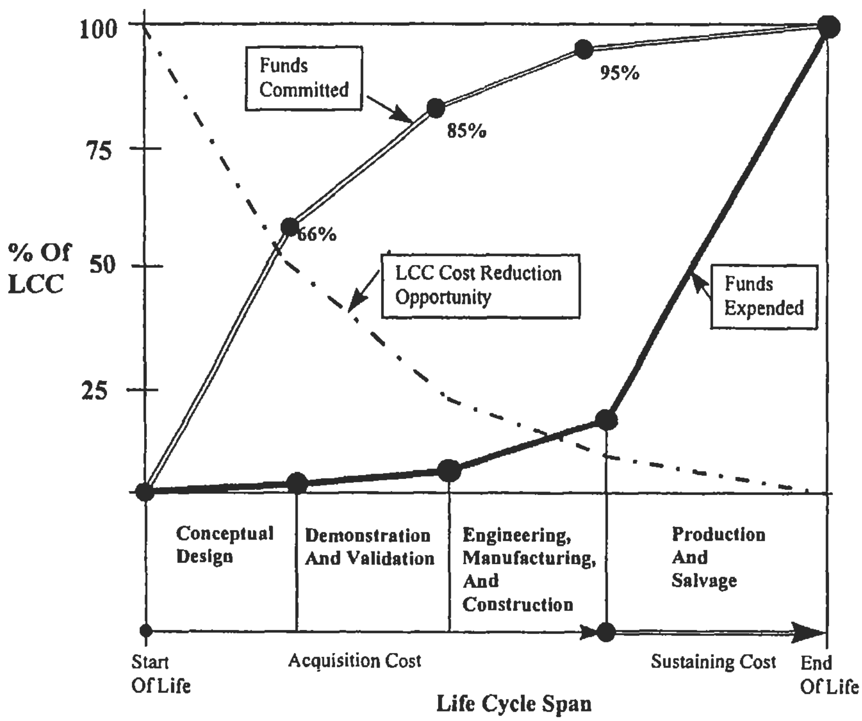


Figure 5-12. Funding trends by commitment and expenditure.

resides in local computer files, but it has not been analyzed or put to effective use. Analysis can start with arithmetic analysis and grow to more complicated statistical analysis. Follow the guidelines for each step listed to work out a typical engineering problem. (Remember, a single right or wrong method/solution does not exist. Many methods and routes can be used to find LCC). If you disagree with the cost or life data, substitute your own values determined by local operating conditions, local costs, and local grades of equipment.

Step 1: Define the Problem. A pump is operating without an on-line spare. At pump failure, the process shuts down and financial losses are incurred as each hour of down-time results in a gross margin loss of US$4,000/hour of outage. Find an effective LCC alternative as the plant has an estimated 10 years of remaining life and is expected to be sold out during this interval.

Step 2: Alternatives and acquisitions/sustaining costs. Consider three obvious alternatives for LCC (other alternatives exist for solving this problem, however, the list is pared for brevity):

1. Do nothing. Continue solo ANSI pump operations with a 100 horsepower, 1750 RPM, 250 psi, 500 gpm, 70% hydraulic efficiency, while pumping fluid with a specific gravity of 1.
2. Add a new, second ANSI pump in parallel (literally in redundant standby) that can be started immediately without the loss of production upon failure of the running pump. Alternate running of the parallel unit every other week to avoid typical failures incurred by nonoperating equipment. The capital costs for the second pump are $8,000 plus $3,000 for check/isolation valves, plus $2,500 for installation.
3. Remove the existing solo ANSI pump and replace it with a new solo API pump with the same performance as for the ANSI model. The API pump cost $18,000 plus $3,500 for installation and the installation will incur a four-hour loss of production for connecting the new pump.

Step 3: Prepare cost breakdown structure/tree. For the do-nothing case, the cost breakdown structure will incur cost in the categories given in Figure 5-13; the cost breakdown structure depicted in Figure 5-14 refers to the redundant ANSI case. For the API pump case, the cost breakdown structure will incur cost in the categories given in Figure 5-15. The individual details for each case will become obvious in step 5.

Step 4: Choose analytical cost model. The model used for this case is explained in an engineering spreadsheet. The spreadsheet merges cost details and failure details to prepare the NPV calculations. Failure costs are prorated into each year because the specific time for failure, because of chance events, is not known.

(text continued on page 292)

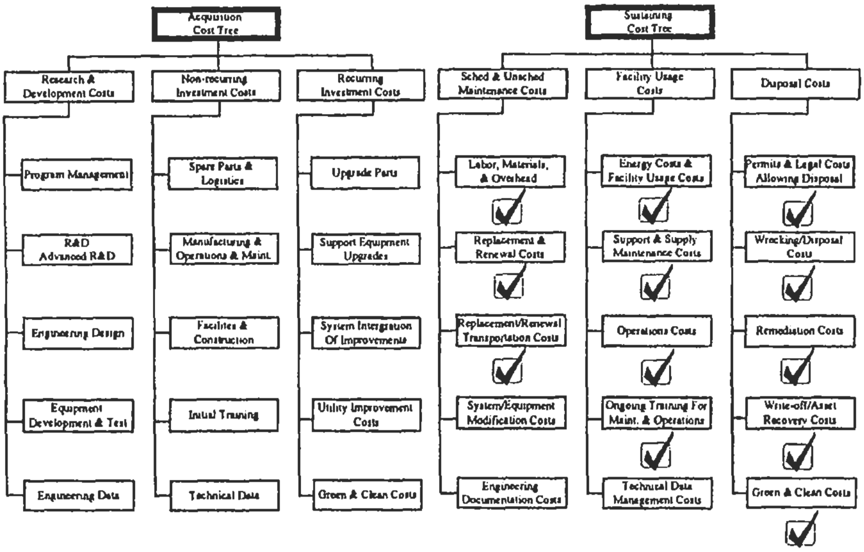


Figure 5-13. Cost components for solo ANSI pump.

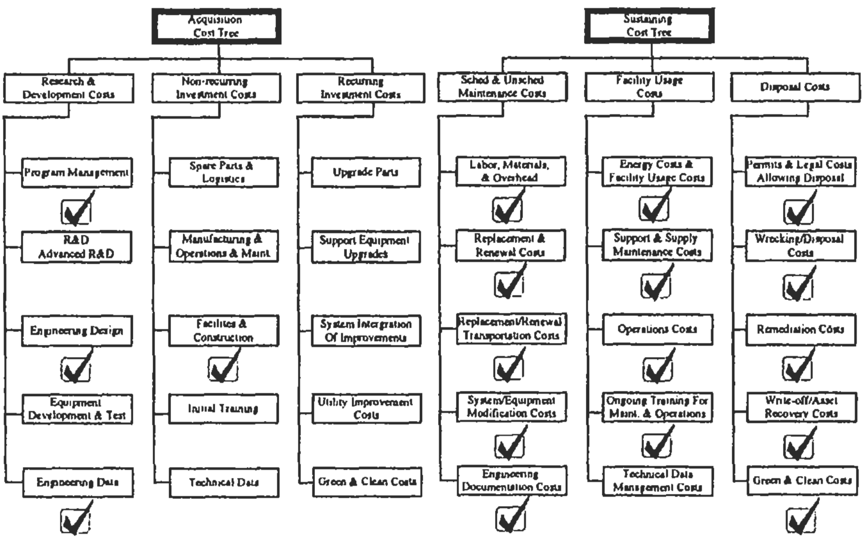


Figure 5-14. Cost components for parallel/redundant ANSI pumps.

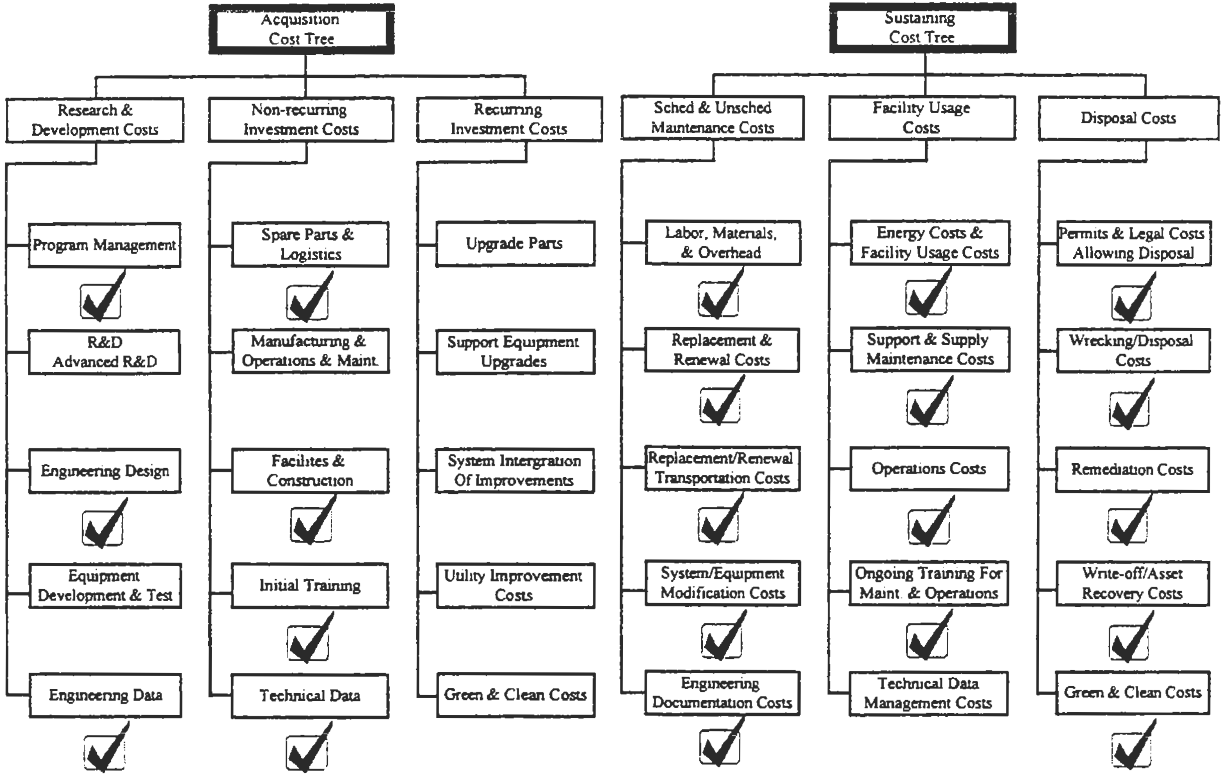


Figure 5-15. Cost components for solo API pump.

(*text continued from page 289*)

The same spreadsheet will be used with more details when statistical uncertainty is added in a section that follows.

Step 5: Gather cost estimates and cost models. This is the complicated section where all the details are assembled. Of course, the more thorough the collection process, the better the LCC model. For this text, the details have been shortened with just enough information described to show the trends.

Alternative #1: Do-nothing case—the datum. Use the following details from plant experience.

Assume all the equipment follows the exponential distribution for reliability with constant failure rates. Note the reciprocal of failure rate is the mean time to failure. Since failure rates are constant, use one year time buckets to collect the cost of failures per year as the literal failure date is unknown. Use the following assumptions based on an accounting principle that costs will follow activity—in this case it will follow failure activity.

Capital cost are zero as the solo ANSI pump is currently a sunk cost and will not change.

Lost gross margin occurs at US$4,000/hour when the process is down for repairs.

Annual power cost for running the pump is US$165/year per horsepower. The plant incurs 1.6 power outages each year for an average downtime of 0.5 hours, and this cost is charged into plant overhead rather than to individual pieces of equipment.

Annual power costs are (US$165/hp-yr) * (100 hp) = US$16,500.

Pump seals have a mean time to failure of three years. When seal failure occurs, eight hours of downtime is also lost production time. Maintenance crew costs for labor, incidental materials, and expense are US$100/hr. Seal replacement costs are US$1,500/seal plus US$300/incident for bearing replacements that occur as good maintenance practice while the pump is disassembled. Seal and bearing transportation costs are usually expedited and cost US$150 per incident.

Annual seal costs are (1 yr/3 years/failure) * {US$(1500 + 300 + 150) + (US$100/hr) * 8 hours + (US$4,000/hour * 8 hours)} = US$11,583

Pump shafts have a mean time to failure of 18 years. When shaft failure occurs, ten hours of downtime is also lost production time. Maintenance crew costs for labor, incidental materials, and expense are US$100/hour. Shaft replacement costs are US$2,500/shaft plus US$1,800/incident for seal and bearing replacements that occur as good maintenance practice while the pump is disassembled. Shaft, seal, and bearing transportation costs are usually expedited and cost US$450 per incident.

Annual shaft costs are (1 year/18 years/failure) * {US$(2,500 + 1,800 + 450) + (US$100/hour) * 10 hours + (US$4,000/hour * 10 hours)} = US$2,542

Pump impellers have a mean time to failure of 12 years. When impeller failure occurs, 8 hours of downtime is also lost production time. Maintenance crew costs are US$100/hr. Impeller replacement costs are US$3,000/impeller plus US$1,800/incident for seal and bearing replacements that occur as good maintenance practice while the pump is disassembled. Impeller, seal, and bearing transportation costs are expedited and cost US$750 per incident.

Annual impeller costs are (1 year/12 years/failure) * {US$(3,000 + 1,800 + 750) + (US$100/hour) * 8 hours + (US$4,000/hour * 8 hours)} = US$3,171

Pump housings (scroll end) have a mean time to failure of 18 years. When housing failures occur, 14 hours of downtime is also lost production time. Maintenance crew costs are US$100/hour. Housing replacement costs are US$3,000/housing plus US$1,800/incident for seal and bearing that occur as good maintenance practice while the pump is disassembled. Housing, seal, and bearing transportation costs are expedited and cost US$1,150 per incident.

Annual housing costs are (1 year/18 years/failure) * {US$(3,000 + 1,800 + 1,150) + (US$100/hour) * 14 hours + (US$4,000/hour * 14 hours)} = US$3,519

Pump bearing sets (a set = two bearings) have a mean time to failure of four years. When bearing failure occurs, eight hours of downtime is also lost production time. Maintenance crew costs are US$100/hour. Bearing replacement costs are US$300/bearing plus US$1,500/incident for seal replacement that occurs as good maintenance practice while the pump is disassembled. Bearing and seal transportation costs are usually expedited and cost US$300 per incident.

Annual bearing costs are (1 year/4 years/failure) * {US$(300 + 1,500 + 300) + (US$100/hour) * 8 hours + (US$4,000/hour * 8 hours)} = US$8,688

Motors have a mean time to failure of 12 years considering all causes. (Motors have many parts and can fail for many reasons. A thorough analysis would be more accurate than this overview approach taken by lumping all details into one MTBF number.) When motor failure occurs, eight hours of downtime is also lost production time as the motor is swapped for a similar unit in stores. Maintenance crew costs are US$100/hour. Motor replacement costs are US$3,000/motor. Motor transportation costs for expedited delivery use US$500.

Annual motor costs are (1 year/12 years/failure) * {US$(3,000 + 500) + (US$100/hour) * 8 hours + (US$4,000/hour * 8 hours)} = US$3,025

Couplings have a mean time to failure of eight years considering all causes. When coupling failure occurs, eight hours of downtime is also lost production time. Maintenance crew costs for labor, incidental materials, and expense are US$100/hour. Coupling replacement costs are US$400. Coupling transportation costs for expedited delivery are US$300.

Annual coupling costs are (1 year/8 years/failure) * {US$(400 + 300) + (US$100/hr) * 8 hours + (US$4,000/hour * 8 hours)} = US$4,188

Maintenance personnel visit the pump monthly for routine PM inspection, lube oil addition/change out, and emissions tests. Maintenance cost is US$50/hour for labor, incidental materials, and expense with 1 hour on the average charged per visit. No failure times are incurred during this activity.

Annual maintenance PM costs are (12 visits * 1 hour/visit) * US$50/hour = US$600

Operations visits the pump once per week for routine PM inspection and vibration logging. Operations cost is US$35/hour for labor and expense, with 0.2 hours charged for each visit.

Annual operations PM costs are (52 visits * 0.2 hour/visit) * US$35/hour = US$364

The Reliability Group receives vibration data from operations by e-mail and scans the data weekly for abnormalities. Surveillance cost is US$50/hour for labor and expense, and on the average, 0.2 hours is charged for each weekly visit.

Annual vibrations PM costs are (52 visits * 0.2 hour/visit) * US$50/hour = US$520

Maintenance and operations conduct a joint tailgate training session on good maintenance and operation practices for this pump once per year. Three people from maintenance attend at US$50/hour-person and three people from operations attend at US$35/hour-person. The training session consumes and elapsed time of 0.5 hours.

Annual training costs are (0.5 hour * (3 people * US$50 + 3 people * US$35)) = US$128

Disposal costs will occur as a lump sum at the end of the ten-year remaining life are expected to be US$500 for permits and legal costs associated with disposition, US$500 for wrecking/disposal costs, US$1,000 for remediation costs, US$0 for write-off/recovery costs, and US$1,000 estimated green/clean costs associated with disposal of the asset. These costs will occur in the final year. Table 5-11 shows non-annualized acquisition and sustaining costs for the existing solo ANSI pump, while Table 5-12 shows the annualized recurring costs.

A quick cost review of the single ANSI pump shows lost gross margin from outages is the biggest annual cost problem as shown in Table 5-12 for a sustaining cost of US$54,827/year. The ANSI pump will consume 16.7 corrective and 35.8 preventive man-hours each year.

Use of MTBFs and expected failures are based on the exponential distribution that is an acceptable first cut for costs, but this technique is not an accurate predictor of failures for wear-out phenomena expected for many of these components. An improved accuracy method will be described later using Weibull distributions for failures.

Table 5-11
Non-annualized Acquisition and Sustaining Costs for Solo ANSI Pump

ANSI Pump:

Cost Element	Year 0	Year 1	Year 2	Year 3	Year 4	Year 5	Year 6	Year 7	Year 8	Year 9	Year 10
Acquisition Costs:											
Program Management	0										
Engineering Design	0										
Engineering Data	0										
Spare parts & Logistics	0										
Facilities & Construction	0										
Initial Training	0										
Technical Data	0										
Capital Equipment	0										
Sustaining Costs:											
Documentation Costs		0									
Disposal Costs											3000
Total =	0	0	0	0	0	0	0	0	0	0	3000

Alternative #2: Add redundant ANSI pump. Use the following details from plant experience.

This case results in pumps installed in parallel but operated as a standby redundant system as the redundant components are not energized but are literally standing by waiting to be used when failure of the operating system is detected. Of course, the detection/switching device is very important for calculating overall system reliability, and for this case the reliability is assumed to be 100%. Also for simplicity, the reliability of the system is calculated as if the redundant pumps are operating in parallel. Furthermore, experience in most chemical plants and refineries shows impending failure is usually detected and redundant systems are usually started in a timely manner to avoid lost production from the failing device; therefore, assume no loss of production by use of redundant pumps.

Capital costs for the redundant ANSI pumps are $8,000 plus $3,000 for check/isolation valves, plus $2,500 for construction and installation along with US$1,000 for program management, US$1,500 for engineering design, and US$1,000 for documentation. Likewise, the plant maintenance organization will incur US$1,000 for engineering documentation costs to put the equipment into the paperwork system.

Lost gross margin occurs at US$4,000/hour when the process is down for repairs.

Annual power cost for running the pump is US$165/year per horsepower. Remember, either the old pump runs or the new pump (not both at the same time). The plant incurs 1.6 power outages each year for an average downtime of 0.5 hours, and this cost is charged into plant overhead rather than to individual pieces of equipment.

Annual power costs are (US$165/hp-year) * (100 hp) = US$16,500

Assume no lost production time by use of the redundant pumps. Keep all other costs as described for the single ANSI pump and depreciate the assets over the ten year project life.

Table 5-12
Annual Sustaining Cost for Single ANSI Pump

ANSI Pump:

Cost Element	MTBF, years	Failures per year or activity per yr	Elapsed Repair or Activity hours	Activity Cost US$/hr	Cost For Lab.,Exp., & Matl US$	Part Cost US$	Logistics Cost US$ per incident	Lost Gross Margin US$	Electrical Power Costs US$	Total Cost US$/yr
Electricity	–	–	–					–	$ 16,500	$ 16,500
Seal	3	0.3333	8	100	$ 267	$ 600	$ 50	$ 10,667		$ 11,583
Shaft	18	0.0556	10	100	$ 56	$ 239	$ 25	$ 2,222		$ 2,542
Impeller	12	0.0833	8	100	$ 67	$ 400	$ 38	$ 2,667		$ 3,171
Housing	18	0.0556	14	100	$ 78	$ 267	$ 64	$ 3,111		$ 3,519
Pump Bearings	4	0.2500	8	100	$ 200	$ 450	$ 38	$ 8,000		$ 8,688
Motors	12	0.0833	8	100	$ 67	$ 250	$ 42	$ 2,667		$ 3,025
Coupling	8	0.1250	8	100	$ 100	$ 50	$ 38	$ 4,000		$ 4,188
Maintenance PM visits			12	50	$ 600					$ 600
Operations PM visits			10.4	35	$ 364					$ 364
Vibration Dept			10.4	50	$ 520					$ 520
Training costs			0.5	255	$ 128					$ 128
Total					$ 2,445	$ 2,256	$ 293	$ 33,333	$ 16,600	$ 54,827

	Item Cost US$	Logistics Cost US$
Seal cost=	$ 1,500	$ 75
Bearings cost=	$ 300	$ 75
Shaft cost=	$ 2,500	$ 300
Impeller cost=	$ 3,000	$ 300
Pump housing=	$ 3,000	$ 1,000
Motor cost=	$ 3,000	$ 500
Coupling cost=	$ 400	$ 300
Lost gross margin US$/hr =	$ 4,000	
Power cost(US$165/hp-yr)=	$ 165	
Motor size(hp)=	100	

System failure rate= 0.9861
System MTBF= 1.01408

1 yr reliability, R= 37%
1 yr unreliability, UR= 63%
1 yr Availability, A= 0.999

Good maintenance practice:

Seal replacement = seals+bearings
Bearing replacement=seals+bearings
Shaft replacement=shaft+seals+bearings
Impeller replacement=impeller+seals+bearings
Housing replacement=housing+seals+bearings

Disposal costs occur as a lump sum at the end of the ten-year remaining life are expected to be US$500 for permits and legal costs associated with disposition, US$500 for wrecking/disposal costs, US$1,000 for remediation costs, US$0 for write-off/recovery costs, and US$1,000 estimated green/clean costs associated with disposal of the asset. These costs will occur in the final year. Table 5-13 shows non-annualized acquisition and sustaining costs for the parallel ANSI pumps; Table 5-14 shows the annualized recurring costs for parallel/redundant ANSI pumps.

A quick cost review of the redundant ANSI pump shows electrical power costs are the biggest annual cost problem shown in Table 5-14 for a sustaining cost of US$21,493/year.

Alternative #3: Replace Solo ANSI Pump with Solo API Pump. Use the following details from plant experience.

Capital costs are $18,000 for a solo API pump plus $3,500 for construction and installation along with US$1,000 for program management, US$1,500 for engineering design, US$1,000 for documentation, and US$500 for technical data. Likewise, the plant maintenance organization will incur US$1,000 for engineering documentation costs to put the equipment into the paperwork system and US$1,500 for training costs associated with the new class of equipment. Spare parts for the new equipment will be increased by $2,900 for a new set of seals and bearings.

Lost gross margin occurs at US$4,000/hour when the process is down for repairs. Costs will be charged for each specific case using the accounting principle that cost follows activity.

Annual power cost for running the pump is US$165/year per horsepower. The plant incurs 1.6 power outages each year for an average downtime of 0.5 hours, and this cost is charged into plant overhead rather than to individual pieces of equipment.

Annual power costs are (US$165/hp-year) * (100 hp) = US$16,500

See Table 5-13 for other failure details, and plan to depreciate the assets over the 10-year project life.

Table 5-13
Non-annualized Acquisition and Sustaining Costs for Parallel ANSI Pumps

Parallel/Redundant ANSI Pumps:

Cost Element	Year 0	Year 1	Year 2	Year 3	Year 4	Year 5	Year 6	Year 7	Year 8	Year 9	Year 10
Acquisition Costs:											
Program Management	1000										
Engineering Design	1500										
Engineering Data	0										
Spare parts & Logistics	0										
Facilities & Construction	0										
Initial Training	0										
Technical Data	0										
Capital Equipment	13500										
Sustaining Costs:											
Documentation Costs	1000	0									
Disposal Costs											3000
Total =	17000	0	0	0	0	0	0	0	0	0	3000

Table 5-14
Annual Costs For Parallel/Redundant ANSI Pumps

Parallel/Redundant ANSI Pumps:

Cost Element	MTBF, years	Failures per year or activity per yr	Elapsed Repair or Activity hours	Activity Cost US$/hr	Cost For Lab.,Exp., & Matl US$	Part Cost US$	Logistics Cost US$ per incident	Lost Gross Margin US$	Electrical Power Costs US$	Total Cost US$/yr
Electricity	–	–	–						$ 16,500	$ 16,500
Seal	3	0.3333	8	100	$ 267	$ 600	$ 50			$ 917
Shaft	18	0.0556	10	100	$ 56	$ 239	$ 25			$ 319
Impeller	12	0.0833	8	100	$ 67	$ 400	$ 38			$ 504
Housing	18	0.0556	14	100	$ 78	$ 267	$ 64			$ 408
Pump Bearings	4	0.2500	8	100	$ 200	$ 450	$ 38			$ 688
Motors	12	0.0833	8	100	$ 67	$ 250	$ 42			$ 358
Coupling	8	0.1250	8	100	$ 100	$ 50	$ 38			$ 188
Maintenance PM visits			12	50	$ 600					$ 600
Operations PM visits			10.4	35	$ 364					$ 364
Vibration Dept			10.4	50	$ 520					$ 520
Training costs			0.5	255	$ 128					$ 128
Total					$ 2,445	$ 2,256	$ 293	$ -	$ 16,500	$ 21,493

	Item Cost US$	Logistics Cost US$
Seal cost=	$ 1,500	$ 75
Bearings cost=	$ 300	$ 75
Shaft cost=	$ 2,500	$ 300
Impeller cost=	$ 3,000	$ 300
Pump housing=	$ 3,000	$ 1,000
Motor cost=	$ 3,000	$ 500
Coupling cost=	$ 400	$ 300
Lost gross margin US$/hr =	$ 4,000	
Power cost(US$165/hp-yr)=	$ 165	
Motor size(hp)=	100	

System failure rate= 0.9861
System MTBF= 1.01408

1 yr reliability, R= 61%
1 yr unreliability, UR= 39%
1 yr Availability, A= 0.9999+

Good maintenance practice:

Seal replacement = seals+bearings
Bearing replacement=seals+bearings
Shaft replacement=shaft+seals+bearings
Impeller replacement=impeller+seals+bearings
Housing replacement=housing+seals+bearings

Disposal costs will occur as a lump sum cost at the end of the ten-year remaining life are expected to be US$500 for permits and legal costs associated with disposition, US$500 for wrecking/disposal costs, US$1,000 for remediation costs, US$0 for write-off/recovery costs, and US$1,000 estimated green/clean costs associated with disposal of the asset. These disposal costs will occur in the final year. Table 5-15 shows non-annualized acquisition and sustaining costs for a new solo API pump; the annualized recurring costs for parallel/redundant ANSI Pumps are given in Table 5-16.

A quick cost review for the solo API pump shows lost gross margin from outages is still the biggest annual cost problem (just as it was for the ANSI pump) as shown in Table 5-16 for a sustaining cost of US$44,444/year. The API pump will consume 11.5 corrective (5.2 hours less than an ANSI pump) and 35.8 preventive (no difference from ANSI) man-hours each year.

Step 6: Make Cost Profiles for Each Year of Study. This step will take into account the annualized charges shown above in Tables 5-11, 5-12, and 5-13 plus the lumped charges at the front and rear end of the project as shown in Table 5-17.

From an examination of these alternatives, adding the ANSI pump in parallel looks more attractive based on the NPV at the 12% discount rate using straight-line depreciation. No revenue stream is included in these calculations, so the case with the smallest loss will be the most attractive case.

Remember, each company will have its favorite discount rate, depreciation schedule, and method for making capital decisions. That means local conditions may prevail in making decisions.

Step 7: Make Break-Even Charts for Alternatives. Break-even charts are useful tools for showing effects of fixed and variable costs. Results for the three alterna-

Table 5-15
Non-annualized Acquisition and Sustaining Costs For A New Solo API Pump

API Pump:

Cost Element	Year 0	Year 1	Year 2	Year 3	Year 4	Year 5	Year 6	Year 7	Year 8	Year 9	Year 10
Acquisition Costs:											
Program Management	1000										
Engineering Dasign	1500										
Engineering Data	1000										
Spare parts & Logistics	2900										
Facilities & Construction	3500										
Initial Training	1500										
Technical Data	500										
Capital Equipment	18000										
Sustaining Costs:											
Documentation Costs	1000	0									
Disposal Costs											3000
Total =	30900	0	0	0	0	0	0	0	0	0	3000

Table 5-16
Annual Sustaining Costs For API Pump

API Pump:

Cost Element	MTBF, years	Failures per year or activity per yr	Elapsed Repair or Activity hours	Activity Cost US$/hr	Cost For Lab., Exp., & Matl US$	Part Cost US$	Logistics Cost US$ per incident	Lost Gross Margin US$	Electrical Power Costs US$	Total Cost US$/yr
Electricity	–	–	–					–	$ 16,500	$ 16,500
Seal	4.5	0.2222	8	100	$ 178	$ 644	$ 33	$ 7,111		$ 7,967
Shaft	22	0.0455	10	100	$ 45	$ 291	$ 20	$ 1,818		$ 2,175
Impeller	16	0.0625	8	100	$ 50	$ 400	$ 28	$ 2,000		$ 2,478
Housing	22	0.0455	14	100	$ 64	$ 336	$ 52	$ 2,545		$ 2,998
Pump Bearings	6	0.1667	8	100	$ 133	$ 483	$ 25	$ 5,333		$ 5,975
Motors	12	0.0833	8	100	$ 67	$ 250	$ 42	$ 2,667		$ 3,025
Coupling	20	0.0500	8	100	$ 40	$ 60	$ 15	$ 1,600		$ 1,715
Maintenance PM visits			12	50	$ 600					$ 600
Operations PM visits			10.4	35	$ 364					$ 364
Vibration Dept			10.4	50	$ 520					$ 520
Training costs			0.5	255	$ 128					$ 128
Total					**$ 2,188**	**$ 2,466**	**$ 216**	**$ 23,075**	**$ 16,500**	**$ 44,444**

	Item Cost US$	Logistics Cost US$
Seal cost=	$ 2,500	$ 75
Bearings cost=	$ 400	$ 75
Shaft cost=	$ 3,500	$ 300
Impeller cost=	$ 3,500	$ 300
Pump housing=	$ 4,500	$ 1,000
Motor cost=	$ 3,000	$ 500
Coupling cost=	$ 1,200	$ 300
Lost gross margin US$/hr =	$ 4,000	
Power cost(US$165/hp-yr)=	$ 165	
Motor size(hp)=	100	

System failure rate= 0.6756
System MTBF= 1.4801

1 yr reliability, R= 51%
1 yr unreliability, UR= 49%
1 yr Availability, A= 0.999

Good maintenance practice:
Seal replacement = seals+bearings
Bearing replacement=seals+bearings
Shaft replacement=shaft+seals+bearings
Impeller replacement=impeller+seals+bearings
Housing replacement=housing+seals+bearings

Table 5-17
Summary of Cost Profiles for Each Alternative

	Year										
	0	1	2	3	4	5	6	7	8	9	10
Alternative #1-Existing Solo ANSI Pump											
Capital	0										
Cost		57827	57827	57827	57827	57827	57827	57827	57827	57827	60827
Savings		0	0	0	0	0	0	0	0	0	0
Depreciation	0	0	0	0	0	0	0	0	0	0	0
Profit b/4 taxes		-57827	-57827	-57827	-57827	-57827	-57827	-57827	-57827	-57827	-60827
Tax Provision		21974	21974	21974	21974	21974	21974	21974	21974	21974	23114
Net Income		-35853	-35853	-35853	-35853	-35853	-35853	-35853	-35853	-35853	-37713
Add Back Depreciation		0	0	0	0	0	0	0	0	0	0
Cash Flow	0	-35853	-35853	-35853	-35853	-35853	-35853	-35853	-35853	-35853	-37713
Discount Factors	1.00	1.12	1.25	1.40	1.57	1.76	1.97	2.21	2.48	2.77	3.11
Present Value	0	-32011	-28582	-25519	-22785	-20344	-18164	-16218	-14480	-12929	-12142
Net Present Value	$ (203,175)	using a 12% discount rate									
Alternative #2-Add Parallel/Redundant ANSI Pump											
Capital	13500										
Cost	3500	21493	21493	21493	21493	21493	21493	21493	21493	21493	24493
Savings		0	0	0	0	0	0	0	0	0	0
Depreciation		1350	1350	1350	1350	1350	1350	1350	1350	1350	1350
Profit b/4 taxes		-22843	-22843	-22843	-22843	-22843	-22843	-22843	-22843	-22843	-25843
Tax Provision		8680	8680	8680	8680	8680	8680	8680	8680	8680	9820
Net Income		-14163	-14163	-14163	-14163	-14163	-14163	-14163	-14163	-14163	-16023
Add Back Depreciation		1350	1350	1350	1350	1350	1350	1350	1350	1350	1350
Cash Flow	-17000	-12813	-12813	-12813	-12813	-12813	-12813	-12813	-12813	-12813	-14673
Discount Factors	1.00	1.12	1.25	1.40	1.57	1.76	1.97	2.21	2.48	2.77	3.11
Present Value	-17000	-11440	-10214	-9120	-8143	-7270	-6491	-5796	-5175	-4620	-4724
Net Present Value	$ (89,993)	using a 12% discount rate									
Alternative #3-Replace ANSI Pump With Solo API Pump											
Capital	18000										
Cost	12900	44444	44444	44444	44444	44444	44444	44444	44444	44444	47444
Savings		0	0	0	0	0	0	0	0	0	0
Depreciation		1800	1800	1800	1800	1800	1800	1800	1800	1800	1800
Profit b/4 taxes		-46244	-46244	-46244	-46244	-46244	-46244	-46244	-46244	-46244	-49244
Tax Provision		17573	17573	17573	17573	17573	17573	17573	17573	17573	18713
Net Income		-28671	-28671	-28671	-28671	-28671	-28671	-28671	-28671	-28671	-30531
Add Back Depreciation		1800	1800	1800	1800	1800	1800	1800	1800	1800	1800
Cash Flow	-30900	-26871	-26871	-26871	-26871	-26871	-26871	-26871	-26871	-26871	-28731
Discount Factors	1.00	1.12	1.25	1.40	1.57	1.76	1.97	2.21	2.48	2.77	3.11
Present Value	-30900	-23992	-21422	-19126	-17077	-15247	-13614	-12155	-10853	-9690	-9251
Net Present Value	$ (183,328)	using a 12% discount rate									

tives are shown in Figure 5-16 for a quick grasp of how the break-even points compare to the base case.

Cumulative present values are shown on the y-axis to combine cost of money with time and show how the effects of expenditures and cost reductions play together. Of course, the issue is to choose alternatives that payback quickly and payback big returns!

The parallel ANSI pump cost line crosses the datum line for the solo ANSI pump in ~1 year; therefore, the costs are less for the redundant system after passing the one-year mark. The solo API pump crosses the datum line in ~5 years and the cost are less than the solo ANSI pump, but the redundant ANSI pump system continues to have a lower cost and thus is more desirable.

Step 8: Pareto charts of vital few cost contributors. The purpose of Pareto charts is to identify the vital few cost contributors so the details can be itemized for sensitivity analysis and ignore the trivial many issues. Pareto rules say that 10% to 20%

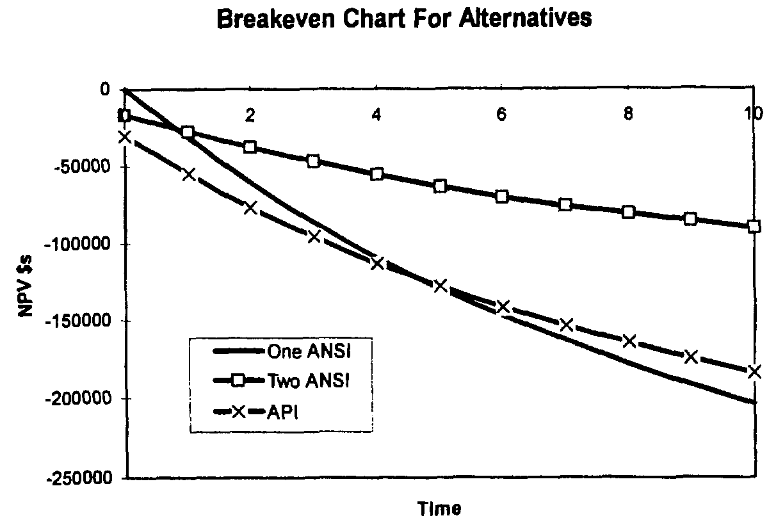


Figure 5-16. Breakeven chart.

of the elements of a cost analysis will identify 60% to 80% of the total cost. These items are the vital few items of concern and need to be carefully considered.

The cost elements for the solo ANSI pump are shown in Figure 5-17 with the high cost of lost gross margins more than twice the cost of the next item. Compare the absolute magnitude of the costs with the cost elements for Figures 5-17, 5-18, and 5-19.

When redundant ANSI pumps are installed, the Pareto chart looks substantially different. This is shown in Figure 5-18, where electrical power becomes the most significant cost item.

When an API pump is substituted for the ANSI pump, the Pareto costs look similar to Figure 5-17, but the magnitudes are different. This is shown in Figure 5-19.

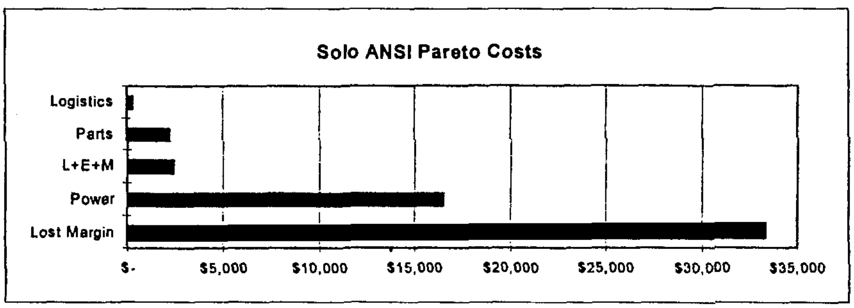


Figure 5-17. Pareto cost chart for solo ANSI pump.

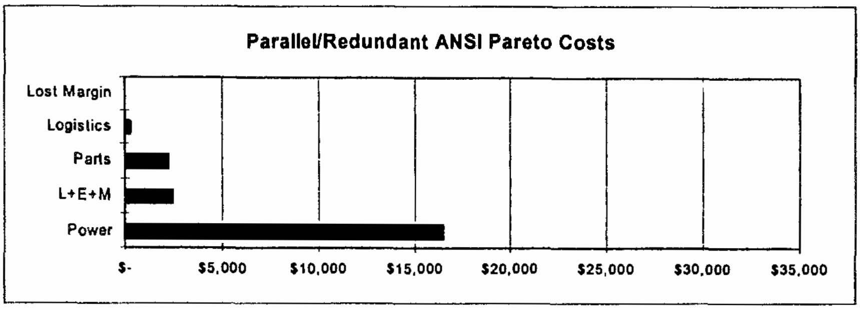


Figure 5-18. Pareto cost chart for parallel-redundant pumps.

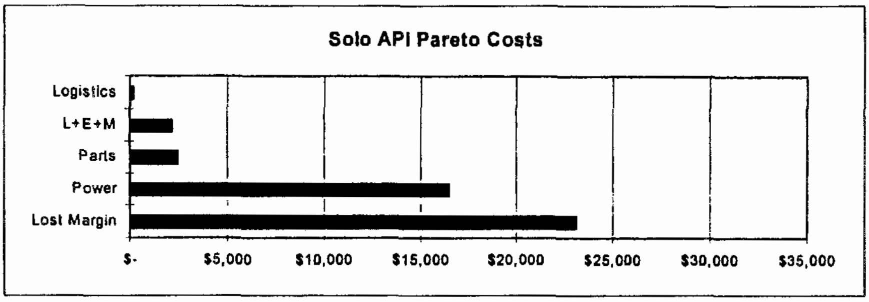


Figure 5-19. Pareto cost chart for solo API pump.

Step 9: Prepare Sensitivity Analysis of High Costs and Reasons for High Cost. A sensitivity analysis allows us to study key parameters affecting LCC. In Table 5-12, the analysis begins with mean time between failures that drives the failure rate. Because all of the components are in series, the failure rates for the exponential distribution can be added to obtain an overall failure rate for the system. Figure 5-17 shows the key for controlling cost is to avoid the downtime that results in lost gross margin caused by *un*reliability.

Unreliability can be reduced by using a higher grade pump as shown in Figure 5-19, or the penalty of lost gross margin is avoided by using a redundant pump as shown in Figure 5-18. Of course, small incremental reductions in lost margin can be achieved by performing the repair work faster. This is frequently the spur rammed into the side of the maintenance organization. Unfortunately, the incremental gain achieved by the faster repairs is very small compared to using a redundancy strategy that leapfrogs the problem and makes major reductions in lost margins, as shown in Figure 5-18. Many industrial organizations concentrate on small incremental gains of working faster (feels good, but isn't too effective) rather than using a smarter reliability strategy to avoid the breakdowns (preventing the problem rather than providing efficient first aid responses) that are the root cause for loss of gross margins.

One issue that is hidden in Figures 5-17 and 5-19 raises its head in Figure 5-18. It's the cost of electrical power to drive the pump. Power consumed is a direct result of work performed, energy lost in inefficient motors/bearing, and energy lost in pump dynamics. Energy savings by use of high efficiency motors can save 2%–5% of the total power cost, and choosing high-efficiency internals for the pumps can save 5%–10% of the total power cost. In short, purchase high-efficiency motors and high-efficiency internals carefully matched to the task to achieve a short payback period. If pump internals were selected for 80% pump efficiency rather than the 70% efficiency used for the calculations, the lower power consumed would be US$16,500 * (70%/80%) = $14,438, which results in a savings of US$2,062 each year, or about equal to all maintenance labor efforts spent to correct failures! The point is this: Examining cost-reduction possibilities by use of LCC details can be productive for discovering real savings opportunities rather than following the old recipes. In short, creating wealth for shareholders often means **stop** doing some things the old way and **start** doing new things in smarter ways.

Using the overall ANSI pump failure rates and a mission time of one year, the reliability at the end of one year is calculated as 37% (which is about the same as saying one pump in three will operate for a one-year interval without some type of failure). The chance for failure-free intervals is low. Much of this poor reliability is driven by how the pump is operated. Optimum conditions are rarely achieved in production plants because of variations in operating conditions and operating styles.

Figure 5-20 illustrates the sensitivity of pump reliability to pump curves and other well-known problems. The shape of the reliability curve is dependent upon many pump features and operating conditions. Figure 5-21 shows other possible sensitivity studies that combine multiple features.

Of course, the effectiveness equation offers good information because the largest single variable is reliability. The other components of the effectiveness equation in Table 5-18 have minor variations.

The life cycle cost shown in Figure 5-22 is the NPV result of the alternatives to put LCC into business terms. The shape of the curve is decided by selection of alternatives and cost drivers.

Step 10: Study Risks of High Cost Items and Occurrences. Failure data is available from many sources to test whether the assumptions made in the analysis are valid or if unusual risks have been taken with numbers used in the study. Consider the failure rate values given in Table 5-19.

An example of the conversion from failure rates to mean time between failures is:

MTBF = 1/((4E-06 failures/hour) * (8760 hours/year)) = 28.5 years

Compare Table 5-19 to Table 5-12 and Table 5-14 for ANSI pumps and the data look comparable except that the failure rate for impellers may have been selected too high and thus the MTBF is lower than shown in Table 5-19. Let local operating conditions and experience decide the correct value. When comparing Table 5-19 to Table 5-16 for the API pump, the results look okay.

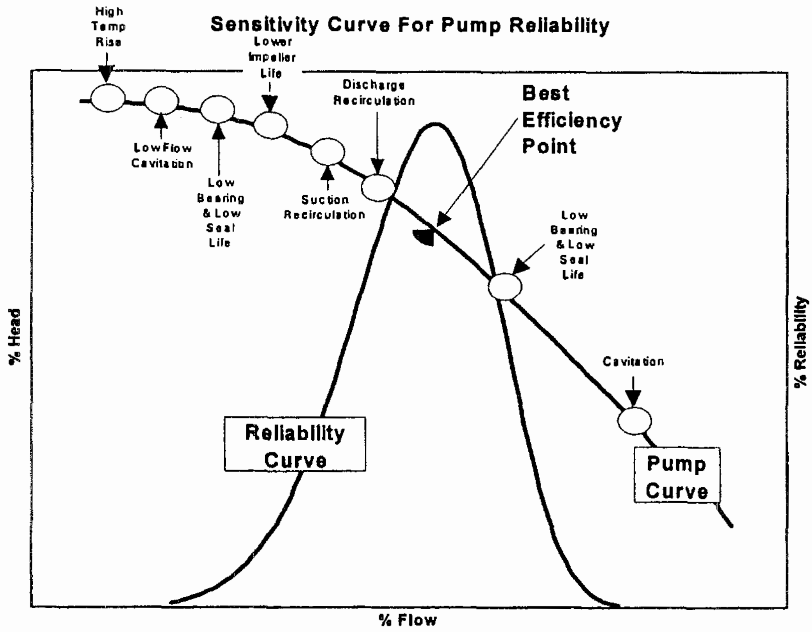


Figure 5-20. Pump reliability vs pump performance curve.

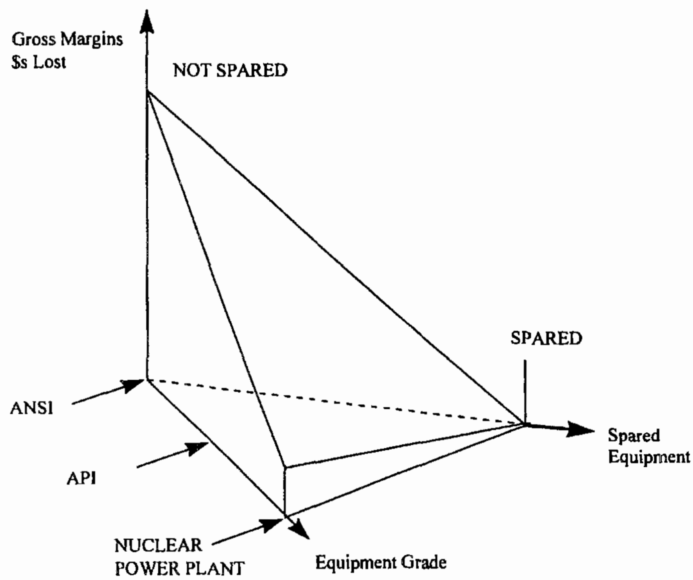


Figure 5-21. Various inputs vs lost gross margin dollars.

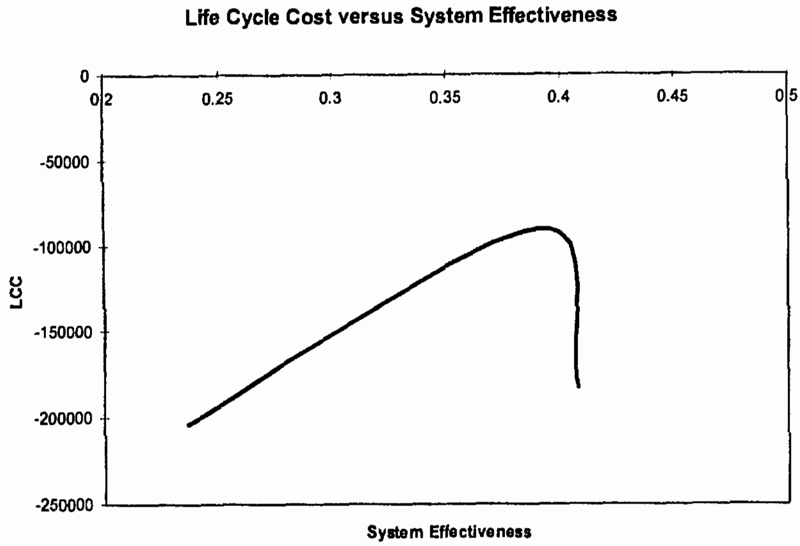


Figure 5-22. LCC and system effectiveness.

Table 5-18
Alternatives Versus Effectiveness and LCC = NPV

Parameter	Solo ANSI Pump	Parallel ANSI Pump	Solo API Pump
Availability	0.999	.9999	.9993
Reliability	0.37	0.61	0.51
Maintainability	0.8 estimated	0.8 estimated	0.8 estimated
Capability	0.8 estimated	0.8 estimated	0.8 estimated
System Effectiveness	0.2366	0.3904	0.4077
Life Cycle Cost	-$203,175	-$ 89,993	-$183,328

Table 5-19
Failure Data Converted to Mean Time Between Failures

	Failure rate (failures/10^6 hours)		MTBF (years)	
Item	Low	High	Low	High
Ball Bearings	4	70	1.6	28.5
Couplings	3	40	2.9	38.0
Housing				
Impeller	0.7	8	14.3	163
Motors	5	900	0.1	22.8
Seals	20	30	3.8	5.7
Shafts	3	20	5.7	38.1

So, where did the failure rate for the pump housing come from? Use experience or other sources. One-stop shopping for failure rates is not possible!

Select cost data from local plant experiences or proposed cost structures for new plants.

Step 11: Select Preferred Course of Action Using LCC. The selection of a parallel/redundant strategy using ANSI pumps is the most attractive alternative out of the three proposed because it avoids process failure and thus reduces the high cost of *un*reliability. Buy equipment that is electrical-power-efficient, yet reliable and correctly sized. Ascertain high hydraulic efficiency to make substantial reductions in electrical power consumption, which is usually a hidden cost item but clearly identified by LCC as a vital element.

Adding Uncertainty to the LCC Results

Each element in the above LCC computation is uncertain. Nothing fails on schedule. Nothing is repaired in exactly the same time interval. Seldom are costs for acquiring goods and services the same price each time. Furthermore, experience tells us that knowledge of failure modes for equipment is required to make best use of reliability-centered-maintenance (RCM) strategies. Uncertainty requires the use of statistical distributions in addition to the usual arithmetic.

Most engineers know about normal (Gaussian) statistical distributions that employ a mean value, x-bar, to describe central tendencies and a standard deviation, σ, to describe scatter in the data. A better statistical distribution for explaining the life and repair times for equipment are Weibull distributions with a shape factor, β (similar to σ), and a characteristic life, η.

Statistical distributions give a different value every time data is drawn for solving spreadsheet problems because of chance selections. Thus Monte Carlo simulation techniques are used to join probability distributions and economic data to solve problems of uncertainty using spreadsheet techniques. Monte Carlo simulation techniques use random numbers to generate failure data and cost data considering the statistical distributions. Monte Carlo results are similar to real life because the results have variations around a given theme.

Monte Carlo results are used with common spreadsheet programs such as Excel™ or Lotus™. Specialized add-in programs such as @Risk™ can add uncertainty to the calculations. Instead of producing a single answer, the Monte Carlo results provide a central trend while providing an idea about the expected variations that may result from many interactions. Ideas about the variations in results are obtained by repeating the Monte Carlo trials many times and studying the end results. With fast PCs on almost every engineers desk, it is possible to conduct 10,000 iterations of a complicated spreadsheet in only a few minutes at a very low cost.

A flag was raised in the Alternative #1, Do-nothing Case, section about exponential failure distributions. With the exponential distribution, the chance for failure is uniform for each period and this does not conform to equipment expectations where wear-out failure modes may predominate with their increasing failure rates as equipment ages. Weibull failure database information is available to supplement the failure data given earlier in this chapter. A partial listing of the Weibull database is shown in Table 5-20. Recent papers describe how to put the Weibull database information to work.

Here's how the Weibull database and Monte Carlo simulations work using the coupling data as an example. Given $\beta = 2.0$ and $\eta = 75{,}000$ hours, what is a Monte Carlo age-to-failure? Solving the Weibull equation for time,

$$t = \eta * \{\ln (1/(1 - CDF))\} = (1/\beta)$$

where CDF is the cumulative distribution function that always varies between 0 and 1. The CDF range is convenient because spreadsheets also have a random number function that varies between 0 and 1. This means if the CDF = (chosen by a number between 0 and 1) = 0.3756, then the Weibull age to failure is 51,470 hours (or 5.9 years) as driven by the random choice of the number 0.3756. Contrast the Weibull results for age-to-failure with results from the exponential distribution, ($\beta = 1$) age-to-failure that produces 35,322 hours or (4.0 years) using the same random number. When the random numbers are used over and over, specific ages-to-failure are selected as representative of specific ages-to-failure.

Table 5-21 shows how Monte Carlo simulation works for the unspared ANSI pump.

- In segment A, the Weibull values are used with random numbers to draw a random age-to-failure. Other ages-to-failure are propagated across the ten-year study period showing how many failures are expected for each year of the study (and multiple failures for an item can occur in a period). The reader has the opportunity to modify the scenario and accompanying logic statements to build more complex failure propagation tables taking into account how good maintenance practices will reduce the number of failures occurring each period.
- In segment B, the numbers of failures are added for cumulative failure results.

Table 5-20
Typical Weibull Failure Data

	Beta Values (Weibull Shape Factors)			Eta Values (Weibull Characteristic Life—hrs)		
	Low	Typical	High	Low	Typical	High
Item						
Ball Bearings	0.7	1.3	3.5	14000	40000	250000
Couplings	0.8	2	6	25000	75000	333000
Housing						
Impeller	0.5	2.5	6	125000	150000	1400000
Motors	0.5	1.2	3	1000	100000	200000
Seals	0.8	1.4	4	3000	25000	50000
Shafts	0.8	1.2	3	50000	50000	300000

- In segment C, the cumulative failures are normalized to an annual basis.
- In segment D, costs are added to the failures and the costs are slightly overstated because costs for good maintenance practices are included even with the slightly elevated failure rates noted above for section (a). Compare annual costs of Table 5-21 with the results from Table 5-12.

Why spend the time and effort building such complicated analysis schemes? The time required to construct Table 5-21 was ~10 hours and run time was 2.6 hours for 10,000 iterations on a 486/50MHz computer, and 17 minutes for a Pentium Pro 200MHz. Monte Carlo simulations are more correct than the generalized and simplified data. The Excel spreadsheet from Table 5-21 is available for download from the World Wide Web (Barringer 1996) and it can serve as a guide for building more complicated analyses to obtain more accurate LCC information.

Table 5-21
Monte Carlo Simulation

Unspared ANSI Pump Life-Cycle Cost Simulation In An Excel Spreadsheet

a) Individual Iteration

				Project Year Of Replacement And Number Of Replacements Required									
Cost Element	η	β	1st Age To Failure	1	2	3	4	5	6	7	8	9	10
Electricity													
Seal	3	1.4	0.23	2	0	0	2	1	0	0	1	0	2
Shaft	18	1.2	33.96	0	0	0	0	0	0	0	0	0	0
Impeller	12	2.5	9.19	0	0	0	0	0	0	0	0	0	1
Housing	18	1.3	12.71	0	0	0	0	0	0	0	0	0	0
Pump Bearings	4	1.3	1.97	0	1	0	0	0	0	1	1	0	0
Motors	12	1.2	5.26	0	0	0	0	0	1	0	0	0	0
Coupling	8	2.0	8.11	0	0	0	0	0	0	0	0	1	0
			Hours Down Time =	16.00	8.00	0.00	16.00	8.00	8.00	8.00	16.00	8.00	24.00
			Number Of Failures=	2	1	0	2	1	1	1	2	1	3

b) Cum. Iterations—> 10002

				Project Year Of Replacement And Cumulative Number Of Replacements Required									
Cost Element	η	β		1	2	3	4	5	6	7	8	9	10
Electricity													
Seal	3	1.4		2038	3018	3462	3644	3644	3703	3653	3613	3643	3738
Shaft	18	1.2		313	405	436	443	465	495	533	541	540	584
Impeller	12	2.5		9	80	190	318	450	586	681	795	876	992
Housing	18	1.3		247	342	378	425	482	477	499	546	552	526
Pump Bearings	4	1.3		1676	2193	2507	2582	2610	2591	2711	2658	2754	2708
Motors	12	1.2		508	631	724	733	756	785	788	897	810	890
Coupling	8	2.0		141	431	757	959	1153	1345	1338	1445	1458	1438
			Cumulative Hours Down Time =	41564	59662	70744	76268	80022	83708	85684	88318	89456	91332
			Cumulative Number of Failures=	4932	7100	8452	9104	9540	9982	10203	10495	10633	10876

c) Annual Failures Expected

				Project Year And Average Number Of Failures Required Each Year									
Cost Element	η	β		1	2	3	4	5	6	7	8	9	10
Electricity													
Seal	3	1.4		0.204	0.302	0.346	0.364	0.364	0.370	0.365	0.361	0.364	0.374
Shaft	18	1.2		0.031	0.040	0.044	0.044	0.046	0.049	0.053	0.054	0.054	0.058
Impeller	12	2.5		0.001	0.008	0.019	0.032	0.045	0.059	0.068	0.079	0.088	0.099
Housing	18	1.3		0.025	0.034	0.038	0.042	0.046	0.048	0.050	0.055	0.055	0.053
Pump Bearings	4	1.3		0.168	0.219	0.251	0.258	0.261	0.259	0.271	0.266	0.275	0.271
Motors	12	1.2		0.051	0.063	0.072	0.073	0.076	0.078	0.079	0.090	0.081	0.089
Coupling	8	2.0		0.014	0.043	0.076	0.096	0.115	0.134	0.134	0.144	0.146	0.144
			Average Down Time Hours =	4.16	5.97	7.07	7.63	8.00	8.37	8.57	8.83	8.94	9.13
			Average Number Of Failures=	0.49	0.71	0.85	0.91	0.95	1.00	1.02	1.05	1.06	1.09

d) Annual Cost Expected For Each Time Interval

				Project Year And Annual Costs Expected From Simulation									
Cost Element	η	β		1	2	3	4	5	6	7	8	9	10
Electricity	–	–		16500	16500	16500	16500	16500	16500	16500	16500	16500	16500
Seal	3	1.4		7081	10485	12028	12660	12660	12865	12692	12553	12657	12987
Shaft	18	1.2		1432	1853	1994	2028	2127	2264	2438	2475	2470	2671
Impeller	12	2.5		34	304	723	1210	1712	2229	2591	3024	3333	3774
Housing	18	1.3		1564	2166	2381	2692	2926	3021	3161	3458	3496	3332
Pump Bearings	4	1.3		5823	7619	8710	8971	9068	9002	9419	9235	9568	9408
Motors	12	1.2		1844	2290	2628	2660	2744	2849	2860	3255	2940	3230
Coupling	8	2.0		546	1670	2933	3715	4467	5211	5184	5598	5649	5571
Maintenance PM visits				$ 600	$ 600	$ 600	$ 600	$ 600	$ 600	$ 600	$ 600	$ 600	$ 600
Operations PM visits				$ 364	$ 364	$ 364	$ 364	$ 364	$ 364	$ 364	$ 364	$ 364	$ 364
Vibration Dept				$ 520	$ 520	$ 520	$ 520	$ 520	$ 520	$ 520	$ 520	$ 520	$ 520
Training costs				$ 128	$ 128	$ 128	$ 128	$ 128	$ 128	$ 128	$ 128	$ 128	$ 128
Total				$36,436	$44,499	$49,609	$52,046	$53,816	$55,553	$56,466	$57,710	$58,224	$59,085
			Approximate system failure rate (failures/yr)=	0.493	0.710	0.845	0.910	0.954	0.998	1.020	1.049	1.083	1.087
			Approximate system MTBF(years/failure) =	2.03	1.41	1.18	1.10	1.05	1.00	0.98	0.95	0.94	0.92
			Theoretical 1 yr Reliability =	60.5%									
			1 yr reliability, R=	61.1%	49.2%	43.0%	40.2%	38.5%	36.9%	36.1%	35.0%	34.5%	33.7%
			1 yr Availability, A=	99.95%	99.93%	99.92%	99.91%	99.91%	99.90%	99.90%	99.90%	99.90%	99.90%

Summary

Life cycle costs include cradle-to-grave costs. When failure costs are included, the quantity of manpower required can be engineered to avoid the use of antique rules of thumb about how maintenance budgets are established.

LCC techniques provide methods to consider trade-off ideas with visualization techniques as described above, which are helpful for engineers. Likewise, LCC analysis provides NPV techniques of importance for financial organizations, and LCC details give both groups common ground for communication. With LCC details, the financial organizations can complete DCF calculations.

Some chemical plants have cost values and failure data for ANSI pumps that are different from that shown above. As examples, coupling costs are around US$100 and the associated logistics costs are perhaps US$75 for couplings with a MTBF of ~3 years, seal life is ~1.5 years, shaft life is ~4.5 years, impeller life is ~3.5 years, pump housing life is ~6 years, and the cost of bearings is ~US$140. Of course, using these "not so commendable" values for a chemical plant will result in higher maintenance costs and greater maintenance expenditures.

Each of the examples described above can be made more accurate by using more complicated models. For one example, in the Monte Carlo model, the time for repairs can be changed from a fixed interval to a statistical interval by simply using a log-normal distribution. This will provide a more realistic idea of the time expended and costs incurred. Spare part quantities can also be calculated.

Good alternatives for LCC require creative ideas. It is the role of the engineer to suggest and recommend cost effective alternatives. Much lower LCC's are obtained when creative efforts are employed in the design area. Making changes downstream in the operating plants has smaller chances for improvements because they come too late in the improvement cycle. Design engineers are the most important link in devising cost-effective plants, and naturally, the burden of LCC falls on their shoulders, but design engineers can't perform an effective analysis unless they have reasonable failure data from operations. Therefore, there is a need for plant and industry databases of failure characteristics. Remember, to obtain good failure data, both failure and success data must be identified. If only the failure information is considered, the failure database will be too pessimistic; no one will believe it and few people will use overly pessimistic data.

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Chapter 6

Extending Motor Life in the Process Plant Environment*

The extensive use of polyphase AC, squirrel-cage, induction motors in process industries to drive pumps, fans, compressors, conveyors and other equipment emphasizes the importance of preventing motor failures and maximizing motor life and dependability. When the high cost of downtime can approach $100,000 an hour, the most effective motor is not necessarily the most efficient motor; it is the one that runs consistently, with maximum uptime.

While it is commonly assumed that motors fail because of electrical malfunction, 75% of motor failures are initiated by mechanical problems, problems which ultimately cause the insulation and/or bearing systems to fail. Insulation and bearing system failures are evident in about 90% of all motor failures.

Learning how to extend motor life requires understanding motor designs, materials, and operating requirements.

Motors are designed to deliver differing speed and torque characteristics that satisfy different loads. Some loads may be hard to start and relatively easy to keep running once initial inertia is overcome. Other loads may be easy to start but require more power as speed increases. Common motor designs, as defined by the National Electrical Manufacturers Association (NEMA), with different speed and torque curves may be selected for different loads (see Figure 6-1).

Some process industry motors such as oil well pumping motors require high slip characteristics to withstand the cyclical load. Others require explosion-proof enclosures. Most are severe-duty applications, requiring protection against the effects of corrosives and other contaminants.

Knowing what kind of motor is presently installed and evaluating the specific motor application are the first steps to extending motor life. A misapplied replacement motor, whether designed for conventional or high efficiency, is doomed to early failure or wasteful operation.

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*Source: Jean J. Revelt, The Lincoln Electric Company, St. Louis, Missouri. Based on a presentation at the 5th International Process Plant Reliability Conference, Houston, Texas, October 1996.

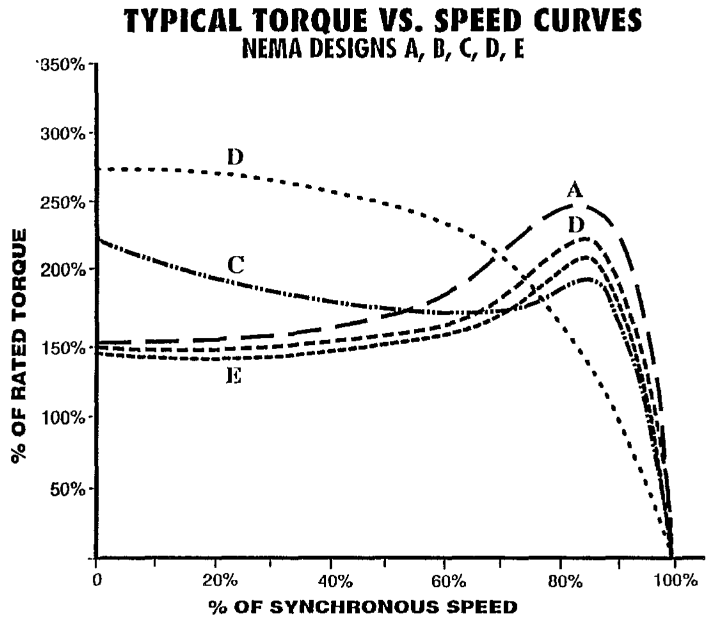


Figure 6-1. Typical torque vs. speed curves for NEMA designs A, B, C, D, E.

Squirrel Cage Motors Are Most Prevalent

The stator of a squirrel cage polyphase motor is comprised of a group of coils of insulated wire that create a rotating magnetic field as the polyphase (three-phase) alternating current rises and falls within the coils. The rotating magnetic field also induces currents and magnetic fields in the rotor, which is mounted on bearings and is free to turn. The attraction and repulsion of the magnetic fields in the rotor and stator create the torque that causes the rotor to turn.

While most of the current is converted to mechanical energy needed to drive the connected equipment, some of it is converted to heat and referred to as "watt losses." This heat must be removed by the motor ventilation system. Heat is the enemy of the motor insulation and can also curtail bearing life. The condition of both insulation and bearings essentially governs the life of the motor.

Motor life can be maximized by protecting the insulation through reducing watt losses and/or improving the ventilation system. Similarly, bearing life can be extended by maintaining an adequate amount of uncontaminated lubrication.

Motor Insulation Systems

In addition to the polymer coating that insulates the coiled magnet wire turns from each other, insulation materials are applied to prevent the current from flowing into the frame as it passes through the stator slots and from short circuiting to adjacent coils. Adjacent coils are connected to different phases and are at different potentials at any given time. Very large motors have heavy, formed coil wires instead of wound coils. These are individually wrapped with insulating material before insertion into the stator slots.

After the coils, slot insulation and phase insulation are all in place, the entire stator assembly is varnished to stabilize and adhere the windings to each other. The varnish is *not* part of the insulation system. Its purpose is to prevent turn-to-turn movement from induced magnetic force, and possible abrasion of the windings when the motor is under load. On motors intended for use in extreme-moisture environments, the windings may be totally encapsulated in polyester resin to both stabilize and protect them.

Insulation Classification

Some insulation materials are better able to withstand heat, i.e., have greater thermal capacity, than others. And some motors, because of high watt losses and the relative limitations of their ventilation systems, will experience higher temperature rises above ambient than others during operation. Classifications based on thermal capacity have been established for insulation systems. Classifications, Table 6-1, define the maximum allowable temperature rise the system materials may withstand without premature failure. In general, the cost of insulation materials and systems increases in proportion to their thermal capacity.

Class A insulation systems for small and medium single-phase and three-phase motors include materials that will deliver suitable life when the motor is operated up to the limiting (maximum) Class A temperature of 105°C. Class B systems are designed for a limiting temperature of 130°C; Class F for 155°C; Class H for 180°C.

Ambient Plus

The operating temperature of the motor insulation includes the ambient temperature plus any temperature rise during operation that the ventilation system is unable to prevent.

Table 6-1
Maximum Allowable Temperature Rise

		Insulation Class	
Service Factor	**Temperature**	**B**	**F**
1.0/1.15	Ambient Temperature	40°C	40°C
1.0	Temperature Rise	80°C	105°C
1.15	Temperature Rise	90°C	115°C
1.0	Total Temperature	120°C	145°C
1.15	Total Temperature	130°C	155°C

**Source: Lincoln Bulletin E7, p. 6.*

For NEMA performance ratings, ambient temperature is assumed to be 40°C. The ambient temperature of actual applications may be higher or lower, affecting the amount of temperature rise that the insulation system may withstand without exceeding its class limits.

Most of the temperature rise results from watt losses, the power required to magnetize the stator and rotor and convert electrical energy to mechanical energy. Nevertheless, frictional heat from poor bearing lubrication may be a contributing factor.

$$\text{Watt Losses} = \text{electrical energy}_{IN} - \text{mechanical energy}_{OUT}$$

The rate of heat generation (dH/dt), as the current passes through the windings and rotor, increases in proportion to the square of the current (I^2) and the resistance of the wire and rotor bars (R).

$$dH/dt = I^2R$$

While the resistances of the windings and the rotor remain relatively constant, the amount of current increases along with the load. Hence, the motor runs progressively hotter as the load increases.

Motor ventilation systems are designed to achieve "watt loss equilibrium" and dissipate enough heat to keep the insulation within its temperature limits—as long as the motor remains within its rated load capacity (rated horsepower) and service factor.

The capacity of the ventilation system is determined by the thermal conductivity of the materials (aluminum versus cast iron and steel), degree of mechanical contact with radiation surfaces, available radiation surface area (stagger-stacked laminations, fins, etc.), and amount of air flow over the radiating surfaces. Air flow is affected by fans, shrouds, baffles, etc.

If the motor is loaded beyond its rated capacity and service factor, the added heat from greater watt loss can exceed the capacity of the ventilation system, raise the temperature of the insulation beyond its rated limits, and rapidly destroy it.

The greatest amount of current flow, and thus heat generation, in a motor occurs during locked rotor conditions or at the moment of start-up before the motor begins to generate "back electromotive force." Rotor current at start-up in a NEMA Design B motor is approximately six times as high as it is at its rated load. With some of the newer high-efficiency designs, it may reach eight to ten times full-load current (see Figure 6-2).

At six times the current, heat is generated 36 times as fast ($I^2 = 6^2 = 36$) as under full-load conditions. This is why repeated starts in a short period of time can be so damaging to motor life; the heat builds up faster than the ventilation system can dissipate it. Also, difficult-to-start loads that require a long time for the motor to reach synchronous speed create excessive heat buildup.

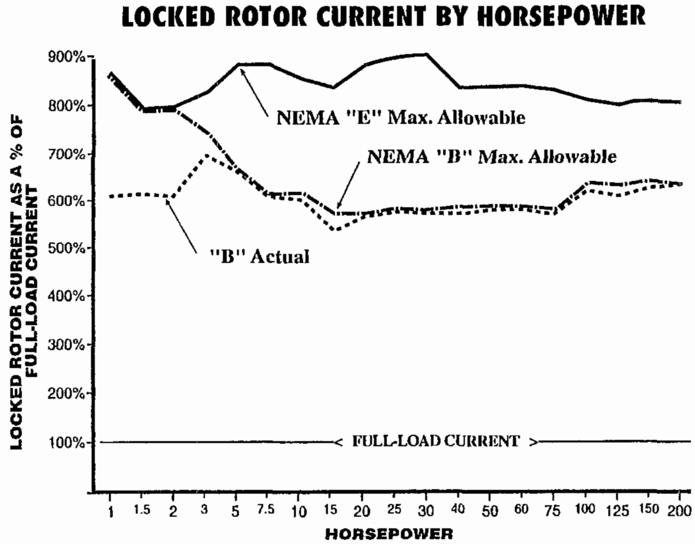


Figure 6-2. Locked rotor current by horsepower.

If a load cycle requires motor operation beyond its rated temperature rise for whatever reason, the operator must be willing to accept reduced motor life or use a larger motor.

A Bank of Motor Life

Through motor designs and selection of electrical and insulating materials, the manufacturer builds each motor with a "bank of motor life." The owner-operator determines how fast motor life will be expended by the way the motor is installed, operated and maintained.

The manufacturer cannot know how a standard motor will be applied. Thus the motor design engineer can only specify construction and materials that will meet the NEMA performance requirements for that motor design class and satisfy the manufacturer's target for maximum warranty claims under the terms of the warranty. Most standard motors are warranted for one year. However, some manufacturers warrant their standard products as long as five years.

Many motors are conservatively designed to carry their rated load with a low-temperature rise. In addition, these motors may use higher-grade insulation materials (G, H, H+), coil wire, and electrical steels as well as superior ventilation systems. This gives them a greater spread between their maximum operating temperature and the thermal capacity or limiting temperature of the insulation system materials. This adds significantly to the "bank of motor life" from which the motor owner can draw.

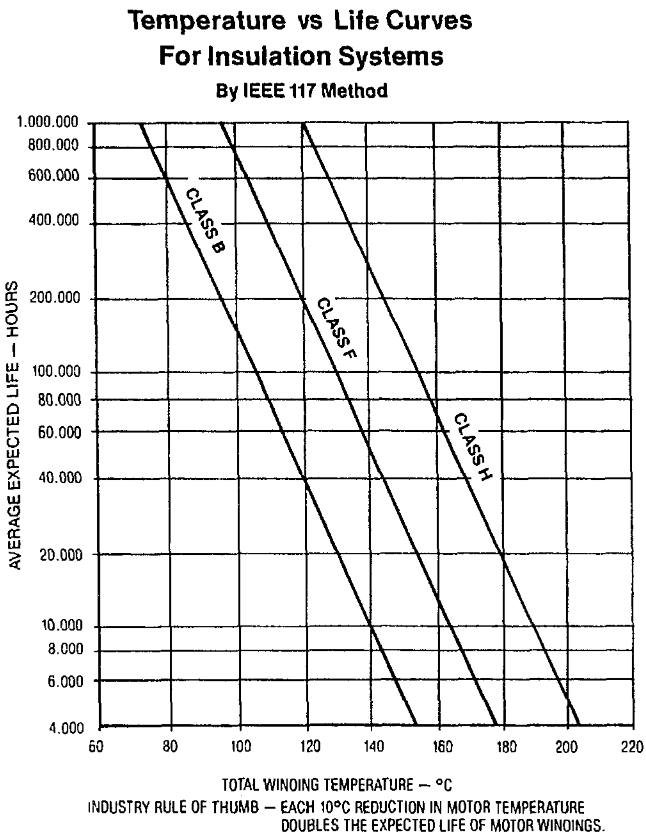


Figure 6-3. Temperature vs life curves for insulation systems. (*Source: Lincoln Bulletin E7, p. 7.*)

"Running Cooler"—A Relative Term

To determine whether one motor is "running cooler" than another requires knowledge of both the thermal capacity of the insulation and the final operating temperature of the windings (see Figure 6-3). As a general rule, *every 10°C increase in operating temperature will decrease insulation life by one-half.*

However, higher thermal capacity of the insulation system does not necessarily equate to longer insulation (and motor) life, nor does a lower operating temperature.

A motor with Class H insulation does not automatically last longer than a motor with Class B insulation. If the motor with the Class B insulation system (limiting temperature of 130°C) operates at a winding temperature of 110°C, it will have twice the insulation life (80,000 hours) of a motor with a Class H insulation system (limiting temperature of 180°C) operating at a winding temperature of 170°C (40,000 hours).

On the other hand, a motor operating at a higher temperature does not necessarily mean a shorter insulation (and motor) life either. A motor with a Class H insulation system operating at 160°C will have nearly twice the life of a motor with a Class B insulation system operating at 120°C.

Thermal Cushion

It is possible to determine if a motor is running "hot" or "cool" and how much "thermal cushion" it has to tolerate undervoltage or occasional overloads beyond its service factor. The reliability professional must investigate how close the motor is running to its designed thermal capacity. The more thermal cushion exists, the longer the motor will last.

Temperature rise figures should be added to the ambient temperature and compared to the limiting temperature, or design thermal capacity, of the insulation system. Both insulation system classification and allowable temperature rise are normally listed on the motor nameplate.

Enclosures

A motor with the proper enclosure must be chosen, depending upon the ambient conditions that the motor will experience. Where the ambient temperature is low and airborne contaminants are not a problem, open motor construction can be used. Where excessive moisture, other airborne corrosives or particulate matter are present, totally enclosed construction is required. When ambient temperatures are excessive, fans or other cooling provisions are needed. Motors used in explosive atmospheres require specially sealed enclosures, thermostats to limit surface temperatures and other features.

Other threats to insulation life include vermin (which the enclosure may be designed to exclude), mechanical damage, high dielectric stresses caused by voltage spikes, bearing system failure, and application-caused external vibration. Power conditions and mechanical concerns include voltage spikes, bearing systems and vibration.

Standard, but Different

While motor manufacturers subscribe to NEMA standards to simplify the classification and comparison of motors, there are wide latitudes within these standards that may complicate the direct replacement of one manufacturer's standard motor with that of another. Although two motors may be dimensionally interchangeable, designed to the same performance standards and full-load ratings, they can consume different amounts of energy to do the same work and will expend their bank of life at different rates.

Motors with increased efficiency, by definition, experience lower watt losses and thus generate less heat, permitting the use of a smaller cooling fan for further increases in efficiency. However, this reduces air flow and the cooling capability of the ventilation system, which will result in reduced insulation life if other steps are not taken to transfer heat away from the motor in order to limit temperature rise.

Void-free rotor construction, stagger stacking of laminations to expose more radiation area, fan and rotor paddle designs, frame material, and frame configuration can all affect the way heat is transferred away from the motor. In general, the manufacturer's quality parameters determine the differences in life of standard motors.

Learning from Failures

When a polyphase, squirrel-cage, AC motor fails, inspection of the windings will usually provide clues to the cause of the failure. A uniform overheating pattern results when there is poor ventilation or high ambient temperatures, continuous excess loading beyond the service factor (SF) for extended periods, or extended or frequent starts. Secondary failures such as phase-to-phase shorts may also occur after the insulation begins to degrade.

In extreme cases where the rotor shaft is locked, i.e., prevented from turning while power is applied for extended periods, the rapid heat buildup will not only destroy the winding insulation, but can even melt the rotor bars. This extreme heat buildup is often associated with failure of the motor control or another protective device required to prevent high input currents for extended periods. Such devices should trip out at 15 seconds or less under conditions of locked-rotor current.

Other causes of insulation deterioration include cyclical loading where the motor temperature regularly rises and falls, extended periods of slow acceleration, excessive "jogging" or "plugging" (reversing the motor to stop the machinery), or extended downtime, which may result in moisture accumulation within the motor. Such motors should be equipped with electrical heaters that are powered when the motor is not running to keep condensation from forming. Continuous motor operation within the load limitations of the motor will deliver the longest insulation and motor life.

More About Thermal Loading

Cases have been observed in kiln applications employing multiple fans where some of the fans were shut down while the kilns were operating to save energy; their insulation failed because the ventilation system was unable to get rid of the built-up heat. In such cases, all the fan motors could have been slowed down with adjustable speed drives to save the same amount of energy, while still allowing the ventilation systems to do their job. Blocked air passages from buildup of airborne contaminants, blower failures on TEBC enclosures and low air density at high altitudes can also lead to failure of the ventilation system to keep temperature rises within designed limits.

Fans, pumps, and dynamic compression equipment account for a large percentage of motors used in the process industries. Often it is desirable to run these centrifugal loads at reduced levels to control such process parameters as flow and temperature. This can be achieved by dampers or other restrictions to throttle the output, but the

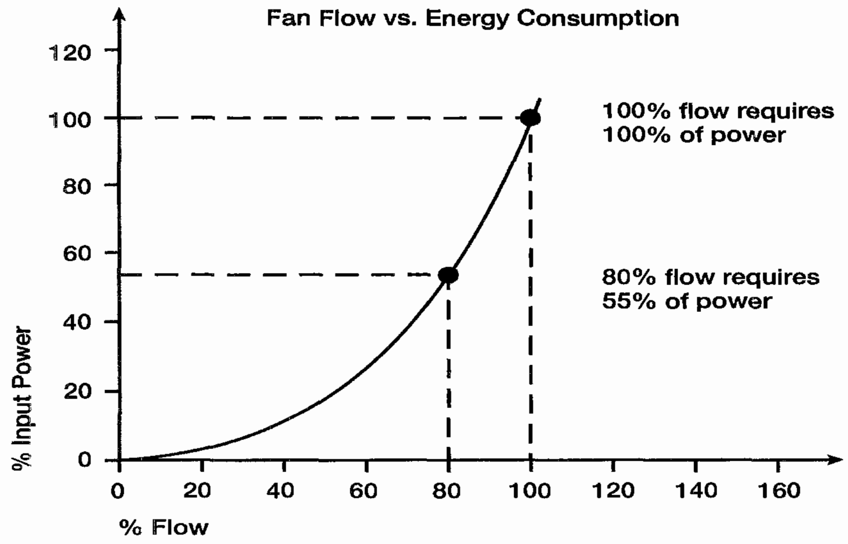


Figure 6-4. Fan flow vs fan energy consumption. (*Source: Lincoln Electric, St. Louis, Missouri.*)

motor is still running at full speed and consuming much of its maximum power requirement. In these centrifugal loads, the relationship between power consumption and flow is not linear. As can be seen from Figure 6-4, a fan will still deliver 80% of its maximum flow at only 55% of its speed. The use of an AC adjustable speed drive (ASD) gives precise output control, saves energy, and allows the motor to run cooler.

A word of caution: Many of today's adjustable speed AC drives rely upon pulse width modulation (PWM) and insulated gate bipolar transistors (IGBTs) to provide the needed frequency and voltage control. The wave forms generated by these drives can contain voltage spikes of 1,600 volts and higher. Motors used with ASDs should be designed to avoid damage to the winding insulation from these high-voltage spikes.

Economics of Oversizing

Oversizing or derating was once a common way of assuring long motor life and dependability, along with providing for unanticipated load fluctuations or future load increases. Driving a constant 25 HP load with a 50 HP motor will result in a much lower heat rise, thus extending insulation and motor life. However, today's energy consciousness has discouraged many motor purchasers from oversizing.

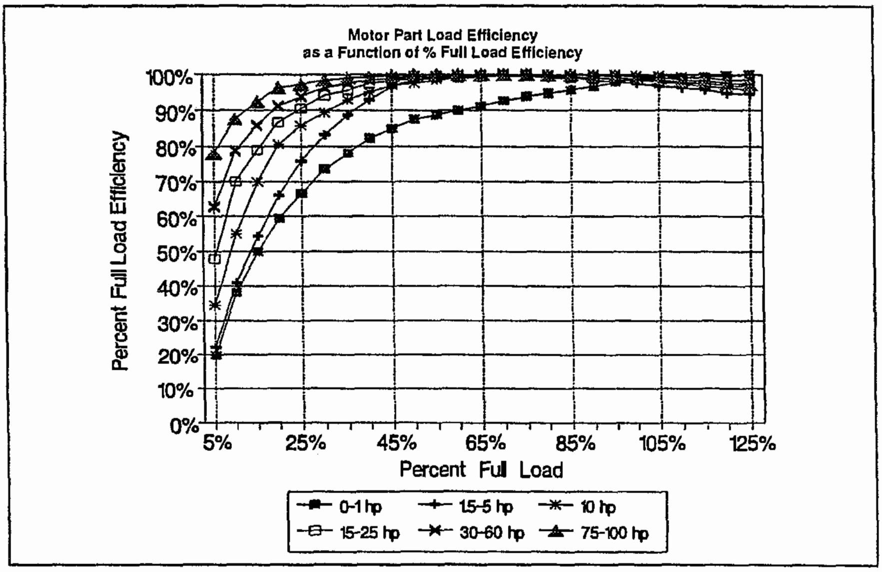


Figure 6-5. Motor part load efficiency as a function of percent of full load efficiency. (*Source: DOE EEE Handbook.*)

This notion may not necessarily be correct, since motor efficiency in the working range (50%–100%) is relatively flat for motors over 1 HP. This is illustrated in Figure 6-5. Also, because larger motors are inherently more efficient than smaller ones, a 50 HP motor constantly running at 50% load consumes even less energy than a 25 HP motor running at 100% load. Here is our proof:

Performance Characteristics of Typical 50 HP and 25 HP Motors—230 VAC (E)

	50 HP @ ½ Load	25 HP @ Full Load
Amps (I)	75.0	64.0
Power Factor (PF)	.70	.827
Efficiency	89%	88.5%

Energy Consumption (kW) = $\sqrt{3} \times E \times I \times PF \div 1000$
50 HP kW = $\sqrt{3} \times 230 \times 75 \times .70 \div 1000 = 20.91$ kW
25 HP kW = $\sqrt{3} \times 230 \times 64 \times .827 \div 1000 = 21.08$ kW

On the other hand, in applications in which the motor load cycle includes high idle time, energy costs will be greater with the 50 HP motor that will consume 70% more power.

	50 HP @ Idle	**25 HP @ Idle**
Amps (I)	47.8	25.0
Power Factor (PF)	.08	.09

Power Consumption (kW) = $\sqrt{3} \times E \times I \times PF \div 1{,}000$
50 HP kW = $\sqrt{3} \times 230 \times 47.8 \times .08 \div 1{,}000 = 1.523$ kW
25 HP kW = $\sqrt{3} \times 230 \times 25.0 \times .09 \div 1{,}000 = .896$ kW

Even so, looking at a typical application requiring the motors to operate 4,000 hours per year—half under load (2,000 hours) and half at idle (2,000 hours)—with a cost per kWh of $.06, the net energy savings with the 25 HP motor would be only $54.

Energy Cost ($) = kW × total hours × $/kWh

Loaded:

50 HP cost = 20.91 × 2,000 × .06 = $2,509

25 HP cost = 21.08 × 2,000 × .06 = $2,530

50 HP motor savings = $21

Idling:

50 HP cost = 1.523 × 2,000 × .06 = $183

25 HP cost = .896 × 2,000 × .06 = $108

25 HP motor savings = $75

($75 – $21 = $54 net annual savings with 25 HP motor)

This is a small price to pay for the knowledge that the 50 HP motor has a much larger thermal cushion than the smaller motor and will be far more able to withstand unforeseen overloads.

Keep Bearings in Mind

Rotor shaft bearings locate the rotor and maintain the desired air gap between the rotor and stator. In addition, they often support a superimposed radial load from a pulley, sprocket, or rotary tool, such as a saw blade or grinding wheel, attached directly to the motor shaft. Depending upon the size of the load and the capacity of the bearings, motor electrical performance, heat buildup, and bearing life can be seriously affected.

Bearing wear from overload or lack of lubrication leads to increased bearing friction and greater horsepower demand. The resulting heat buildup—more current means greater watt losses—can consume the motor's thermal cushion and lead to insulation failure.

Conversely, a load connected to the motor by a properly aligned coupling will impose almost no added load on the motor bearings. Vertical applications, and especially pump motors, place significant thrust loads on the motor shaft bearings as well. Medium-horsepower motors for such applications are generally equipped with deep-grooved ball bearings.

Larger vertically mounted motors (125 HP and above) driving pumps, fans, aerators, mixers, autoclaves, cooling towers, and similar applications may require grease-lubricated, angular-contact ball bearings, oil-lubricated spherical roller bearings, or sliding bearings that are oil-lubricated and water-cooled to handle extremely high thrust loads.

Bearing life (L_{10} Life) is defined by the Anti-Friction Bearing Manufacturers Association (AFBMA) as the life, in millions of revolutions or hours of operation that 90% of a group of bearings will complete at a given load and speed before fatigue failure occurs.

Motor bearings are normally selected with an L_{10} design life adequate to deliver approximately *five years* (40,000 to 50,000 hours) of continuous operation. Average motor bearing life is approximately *five times* design life and median bearing life is approximately *seven times* design life. A properly designed and maintained bearing system will normally outlast the insulation and will not be the determining factor in the life of the motor.

Whenever possible, motor bearings should be configured to allow re-lubrication. The housings should include a relief path for the spent lubricant. Preventive maintenance requires regular application of the same type of grease initially applied at the factory. Polyurea greases, for example, are not compatible with lithium greases and mixing may lead to loss of lubrication.

Care should also be taken not to introduce contamination when greasing. Contaminants create friction between the bearing elements and cause heat build-up. This heat can melt the grease, allowing it to run out, causing both bearings and motor to overheat. Motors operating in very hot or very cold environments may require special lubricants. Motor manufacturers occasionally select bearings that are lubricated for life at the time of manufacture. The mountings include contamination barriers such as shields and slingers to prevent the entry of liquid and solid contaminants, as well as sealing surfaces to prevent the loss of the factory installed lubricant. Needless to say, life-time lubricated bearings may limit both the anticipated and actually achieved life of the motor.

Some "lubricated-for-life" cartridge bearing mountings are, indeed, furnished with provisions for adding lubrication when the motor environment is particularly dirty, wet, or corrosive. The addition of a small amount of lubricant (¼ ounce every three months) is merely intended to refresh the lubricant seal between the shaft and the bearing cartridge or the motor end bracket. Keep in mind that the added grease, however, does not reach the bearing elements since the double seals or shields prevent the flow of lubricant into the bearing.

Oil mist and other types of oil lubrication systems are used on large-horsepower motors and the manufacturer's recommendations for lubricant selection and system maintenance may have to be followed closely.

Motor Mounting Basics

A motor is part of a system that also includes the driven machinery, the power source, the atmosphere surrounding the motor, and the surface or mounting base to which the motor is attached. Improper mounting or attachment of a motor to the base can lead to otherwise unexplained bearing failures, vibration, shaft breakage and, ultimately, premature motor failure.

The complete motor support structure—the understructure as well as the motor baseplate—must be adequate to resist forces resulting from the weight and operation of the system comprised of motor and driven machinery. The four motor attachment points must be fastened to a flat, rigid, machined surface. The motor should be dowelled as well as bolted; bolts alone are not always adequate to prevent lateral shifting. If a motor base is not flat, the frame will twist when the bolts are tightened down, causing misalignment with the driven machinery, vibration, and eventual bearing failure.

The motor base or mounting should be stiff enough to withstand deflection from belt pulls and other operating loads. Welded bracing can be added to improve mounting stiffness, but the heat of welding may also cause distortion and impose new stresses. Ideally, a base that has been modified should be stress-relieved and the mounting surfaces remachined.

Grouting should be properly installed between the steel motor base and the concrete block or cement pad on which it rests. A large base resting on only a few support points will sag and cause vibration problems. Inspect grouting periodically for cracking, powdering, or crumbling. The grout usually fails before the motor requires replacement, and the remaining motor life will be reduced without adequate support.

Often, when vibration is detected, loosening one attachment bolt at a time and noting the result will identify and locate a support problem. Adequate shimming should then be added under the affected mounting point and all bolts should be properly retightened.

When using shims to compensate for unavoidable irregularities in the motor base, or when making vertical adjustments to align components, as few shims as possible should be used. Shim packs of many thin shims are difficult to tighten down uniformly. Ideally, a single shim should be used at each point requiring shimming. Tapered shims are a last resort when the motor feet and base plate cannot be made parallel. This will avoid distortion of the feet when the bolts are tightened.

Consideration should always be given to alignment changes that occur while the motor system is in operation. Thermal expansion of tightly coupled shafts, for example, can transfer loads from one component to the next, affecting bearing life and even causing shaft breakage.

Changes in process operating conditions after original installation of the motor can cause new problems. Any time a motor is removed for servicing or shifted because of load changes, all dowels and shims should be inspected and original installation procedures followed.

Motor System Tuneup

Proper tuning of the motor system is required to avoid operation at natural (critical) frequencies or speeds that will create resonance and large damaging oscillations that can literally cause components to "shake themselves to death." The complete system must be connected and operating to check for these critical frequencies since they are determined by total mass as well as speed. "Bounce" from unresisted belt pulls, impact loads from hammer mills and vibrations from reciprocating compressors can all cause damaging resonance in the motor system.

With systems operating at a single speed, the critical frequency should be kept well to the high side of the operating speed. With multiple-speed systems, there are multiple critical frequencies (or harmonics) that are encountered as the speed changes. Where electronic adjustable speed drives are used to control the system, they may have to be programmed to avoid operation at these speeds, once they are determined.

Pumping and Piping

Vertical pump motor mountings commonly experience resonance when one end of the motor is unrestrained. The pump head itself is occasionally rather unsupported, leading to reed frequency vibration. Changing the mass of the vertical motor by adding weights, additional bracing, or spring washers under the hold-down bolts may eliminate vibrations, but proper system design is preferable.

External piping can also transmit vibrations to the pump drive system. Surges in the fluid system can cause unrestrained piping to "hammer" or shake uncontrollably and affect the motor and pump drive. Thermal expansion of long runs of pipe can also affect the alignment of the pump drive and put stresses on the motor bearings.

Power Points

All of the manufacturer's ratings for motor output, temperature rise, and expected life assume that the characteristics of the incoming power fall within certain variations of voltage and frequency (Hz). The total limit for these variations is ±10%. Operating the motor outside this limit affects both motor life and motor efficiency.

Operating a motor at reduced voltage causes the motor to draw increased current to satisfy the torque demand. The motor may be unable to develop adequate starting torque and higher current means higher I^2R losses.

Operating a motor at higher voltage increases core losses and insulation-damaging heat buildup. In general though, low voltage creates greater problems than high voltage and has the same damaging effect on winding insulation as overloading.

Voltage imbalance among the three phases is also damaging to motor life. Temperature rise in the windings increases at a rate equal to two times the square of the net voltage imbalance in the phase with the highest current ($2_{INET}{}^2$). *A 3.5% net voltage imbalance will cause a 25% increase in temperature.*

Assume the voltages to a motor are 456 volts to phase A, 440 volts to phase B, and 472 volts to phase C. The average voltage is 456 volts. The voltage imbalance is as follows:

$$\text{Voltage imbalance} = \frac{472 - 456}{456} \times 100 = 3.5\%$$

$$\text{Temperature increase} = 2 \times 3.5^2 = 25\%$$

Causes of voltage imbalance include unequal phase system loading, unequal tap settings, poor connections to the power supply, open delta transformer systems, capacitor bank (power factor correction) malfunctions, single-phasing (loss of a phase), etc. Voltage imbalances over 1% require derating the motor per NEMA standard MG1-14.35. The motor in the above example should be derated to operate within its specified temperature rise limits.

As mentioned earlier, another power problem affecting motor insulation and motor life is voltage surges caused by the high switching frequencies of modern PWM inverters (adjustable-speed drives or ASDs) utilizing IGBTs. While rapid IGBT switching eliminates audible noise and the "cogging" problems of earlier AC drives, the wave forms generated include transient spikes of 1600 volts or higher in a nominal 460 V system. This is more than enough to overcome the 600-volt "withstand capability" of the winding insulation.

Repeated exposure to these high voltage spikes punctures and breaks down the winding insulation, leading to early motor failure. Such problems can be minimized by special motor winding methods, special winding insulation or both.

Other damaging power surges can be caused by lightning strikes, capacitor discharges, and utility problems, as well as locked-rotor conditions resulting from motor system malfunction.

Over-Current Insurance

Over-current protection should be provided in the motor feeder circuit and good engineering practice emphasizes using an overload relay in each phase of the motor to give protection from voltage imbalance conditions. Controllers and overload relays must be sized in accordance with the National Electrical Code (NEC) and are nominally designed to accommodate 600% of the full-load current. The recently introduced NEMA Design E, high-efficiency motor may require a larger controller since the design parameters allow a locked rotor current up to 900% of full-load current.

Motor Life Insurance Terms

To maximize the life and performance of AC squirrel-cage induction motors in process industry applications, one must have:

- A motor with the proper design (A, B, C, D, or E) for the application
- An enclosure properly configured to protect the motor from environmental contamination
- An adequately sized motor to handle the load requirements
- A conservatively designed (or applied) motor that will allow it to work well within its designed thermal capacity
- A proper installation that assures good alignment, eliminates vibration, and does not transfer unwanted external loads to the motor
- An adequately designed and maintained (if required) bearing lubrication system
- A motor environment that provides adequate air circulation and maintains the effectiveness of motor heat removal or ventilation systems
- A regulated power supply that supplies the correct voltage, maintains voltage balance between phases and provides protection against over currents
- A load cycle designed to avoid frequent starts and other causes of motor overheating.

Notes

1. "Slip" refers to the difference between the synchronous speed (determined by the frequency of the alternating current and number of motor poles) and the actual full-load speed of an AC induction motor. Slip is required to induce current in the rotor. A high-slip motor (Design D) produces higher torque at lower speeds than other types of motors.
2. NEMA medium motors (open-motor Designs A, B and C; 1 HP to 200 HP) are required to have a "service factor" [SF] of 1.15. This means that such a motor is able to operate at 115% of rated load for a prolonged period while remaining within the allowable temperature limits of the insulation system.

Chapter 7

Equipment Reliability Improvement Through Reduced Pipe Stress*

It is intuitively evident that piping can, and will, cause forces and moments to act on equipment nozzles (see Figure 7-1). These forces and moments will ultimately act on the equipment centerline (see Figure 7-2).

What is perhaps not often appreciated is the fact that the load and stress imposed from a connecting piping system can greatly affect the reliability of equipment.

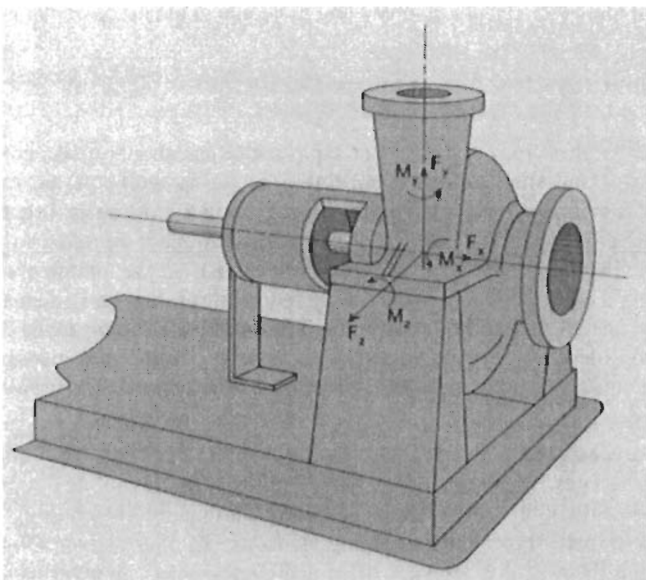


Figure 7-1. Forces and moments acting on a pump.

*Contributed by L. C. Peng, PE., Peng Engineering, Houston, Texas.

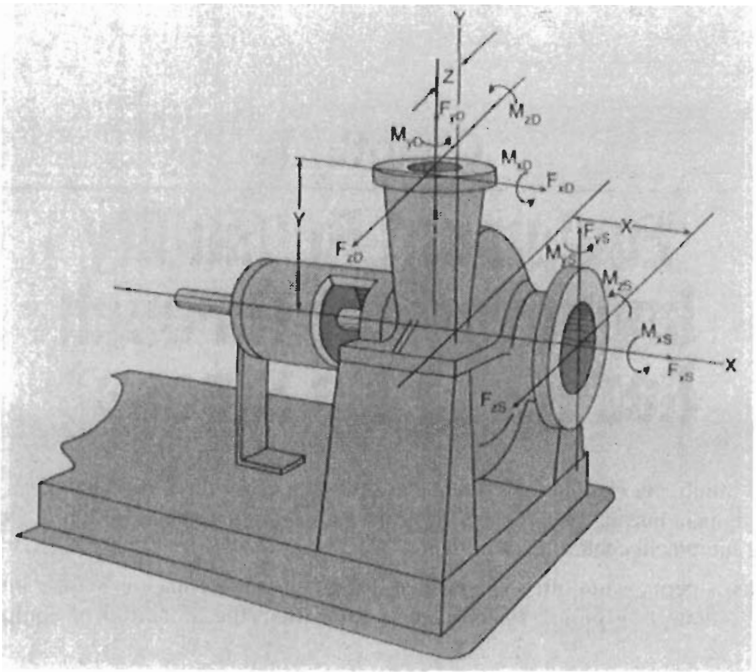


Figure 7-2. Resultant forces and moments at the pump centerline.

These loads, either from expansion of a pipe or from other sources, can cause shaft misalignment and shell deformation, thus interfering with internal moving parts. Therefore, it is important to design the piping system to impose as little stress as possible on the equipment and, ideally, it is preferred to have virtually no piping stress imposed on the equipment. The established practice is for an equipment manufacturer to specify a reasonable allowable piping load and for the piping designer to design the piping system to suit the allowables. The allowable piping loads are generally determined solely by the equipment manufacturers without any participation from the piping engineering community. The values so determined are usually too low to be practical.

Machines designed with unusually low allowable pipe load are almost certain to be weak machines. Weak machines also complicate the layout of the piping system in meeting allowable values. Unusual configurations and restraining systems are often used to make the calculated piping load stay within the given allowable range. However, all these efforts are very often just exercises in computer technology. The main reliability problem has not been solved. A better, sturdier equipment design with some common sense piping arrangement is the basis for improving reliability.

Allowable Load

Process equipment, and especially rotating equipment, generally tolerates only a very low allowable piping load. Piping engineers often think the manufacturers give low allowables to protect their own interests. This notion is not necessarily true, because many equipment items indeed cannot take too much load. The problem is that a weak link exists that is often overlooked in the design of an equipment auxiliary or subsystem. Figure 7-3 shows a typical pump installation that can be divided into three main parts: the pump body, the foundation, and the pedestal/baseplate. Without proper input from, or consultation by, both piping and equipment engineers, the routine design of a given pump assembly may overlook issues that affect different parts of the pump. The pump body is designed to be as strong as, if not stronger than, the piping so that the body can resist the same internal design pressure as the piping. The foundation, normally designed for the combined pump/motor assembly weight, is usually massive and stiff due to limitations in soil bearing capacity. However, the pedestal/baseplate is generally designed to take into account only the pump weight. This design basis creates a very weak pedestal/baseplate that can take very little load from the piping; hence, the usual claim that equipment cannot take any piping load. Although most vendors have at least some awareness of these facts, low allowable piping loads are still a very common occurrence. If they are exceeded, equipment misalignment and lack of concentricity of internal components will significantly curtail equipment life.

By understanding the situation, the problem can actually be rectified very easily. Improvement has already been seen in pump applications. Pump application engineers who long realized the low allowable piping load problem customarily specified double (2X) or triple strength (3X) base plates to increase the allowable piping load by two or three times, respectively. Surprisingly, to most engineers, the cost of a 2X or 3X pump set is only marginally higher than that of a regular pump set. Actually, it

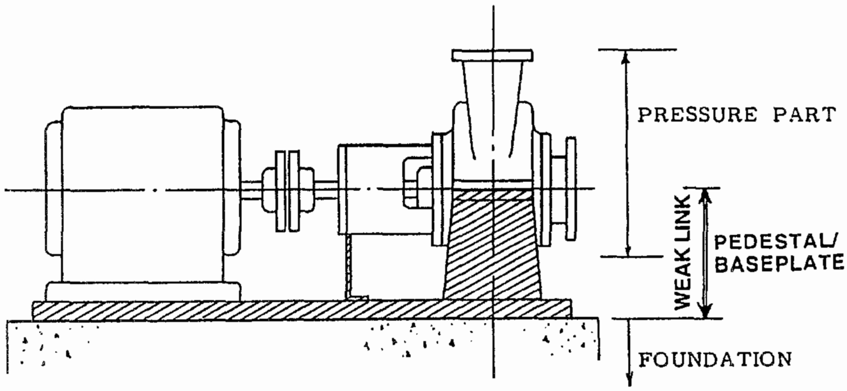


Figure 7-3. The weak link.

should not have been the least bit surprising, since all a vendor has to do to make it 2X or 3X is to provide a couple of braces or stiffeners. Recognizing the popular demand for the 2X or 3X baseplate, the API formally adopted it to its pump standards. Since the sixth edition of API Std-610, the allowable has been increased to a level that makes the 2X and 3X specification no longer necessary. In other words, the strength of the whole pump assembly has become fairly uniform and no additional allowable can be squeezed out without adding substantial cost. Unfortunately, at present this philosophy has not been shared by the manufacturers of most other fluid machines. For example, the 1956 NEMA turbine allowable load is probably the most unreasonable of its kind. The API 617 centrifugal compressor and ASME/ANSI B73.1 pump standards are not far behind. API Standard 617 uses 1.85 times the NEMA allowable, and ANSI B73.1 vendors often use 1.30 times the NEMA values for the allowables. Figure 7-4 shows the comparison of the pipe strength, the allowable API Std-610 piping load, and the NEMA allowable piping load. The pipe strength curve is based on a 7500 psi bending stress. It should be noted that the allowable pipe stress against thermal expansion can be as much as three times higher than 7500 psi.

Figure 7-4 clearly shows that the piping load that can be applied to equipment is much smaller than the strength of the pipe itself. Therefore, in designing the piping connected to an equipment item, the equipment allowable load is the controlling factor. For low-allowable items, such as a large-size steam turbine, an extensive expansion loop and a restraining system are generally required. This is a fact that should be understood by all parties concerned.

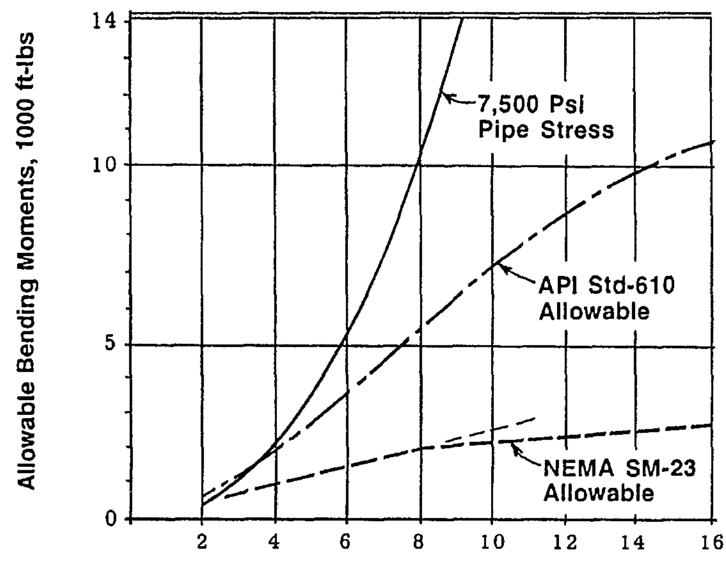


Figure 7-4. Allowable piping loads.

Because of the elaborate design of the piping system attached to sensitive equipment, engineers may sometimes get too trapped in the computer maze and overlook engineering fundamentals. Typical examples that can cause unreliable operation are discussed next.

Excessive Flexibility

Adequate piping flexibility is required to reduce the piping load at the equipment nozzle to the acceptable value. However, a good design should take into account the flexibility of the support structure and make proper use of protective restraints. Without properly located restraints, a piping system, no matter how flexible it is, has difficulty meeting the allowable load imposed by the equipment. Figure 7-5 shows a pump piping system that was designed without any restraints installed. This is a common mistake made by inexperienced engineers who think that a restraint can only increase the stiffness, thus increasing the load. It is true that a restraint will tend to decrease the flexibility of the system as a whole and will increase the maximum stress and force in the system. However, a properly designed restraint can shift the stress from the portion of piping near the equipment to a portion further away from the equipment.

Although extensive loops are used in the piping shown in Figure 7-5, the piping load still may not meet the equipment allowable due to the lack of a restraining system. Excessive flexibility makes the system prone to vibration, because it is easily

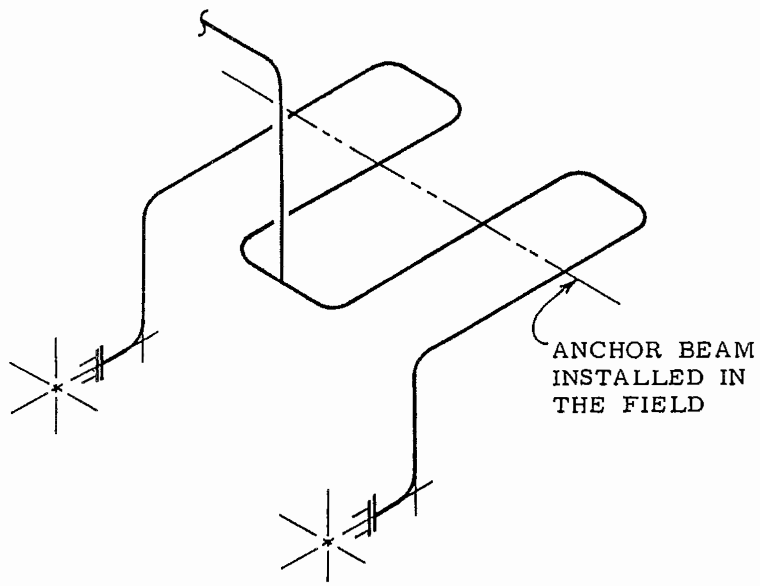


Figure 7-5. Too much flexibility.

excited by small, disturbing fluid forces. In addition, the piping loops enhance the internal fluid disturbance by creating cavities and other flow discontinuities associated with excessive pressure drops. A system similar to that shown in Figure 7-5 experienced very severe vibrations in one petrochemical plant. The responsible engineer had to install a large cross beam anchoring all the loops in efforts to reduce vibration to a manageable level. The function of the original loops was lost by the anchoring system. Moreover, the piping still experienced larger than normal vibrations due to flow disturbance caused by a loop which was now structurally fixed, but hydraulically still open to many changes in the direction of flow.

Theoretical Restraints

A properly designed piping system generally includes restraints to control the movements and to protect sensitive equipment. However, there are also restraints that are placed in desperation by piping engineers trying to meet the allowable load of the equipment. These so-called computer restraints give very good computer analysis results on paper, but are often very ineffective and sometimes even harmful. Figure 7-6 shows some typical situations that work on the computer, but do not work on a real piping system. These pitfalls are caused by the differences between the real system and the computer model. Here are some of the more important discrepancies:

- **Friction** is important in the design of the restraint system near the equipment. Figure 7-6 (a) shows a typical stop placed against a long Z-direction line to protect the

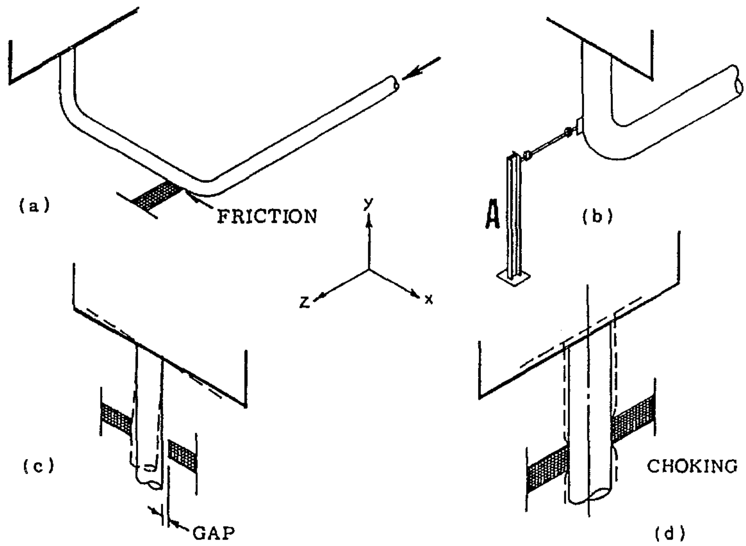


Figure 7-6. Problems with theoretical restraints.

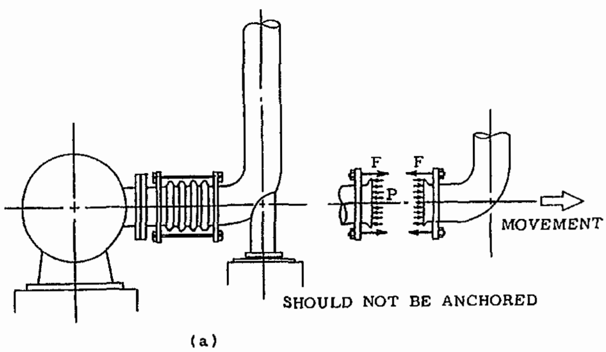
equipment. If friction is ignored in the design calculations, the calculated reaction at the equipment is often very small. However, in reality, friction acting at the stop surface will prevent the pipe from expanding in the positive X-direction. This friction effect can cause a high X-direction reaction at the equipment. Calculations including the friction will predict this problem beforehand. A proper type of restraint such as a low-friction plate or a strut would then be used.

- An ineffective support member is another problem often encountered in the protective restraints. Figure 7-6 (b) shows a popular arrangement to protect the equipment. The engineer's instinct is to always put the fix at the problem location. For instance, if the computer shows that the Z-direction reaction is too high, the natural fix is to place a Z-direction stop near the nozzle connection. This may be all right on the computer, but in reality it is very ineffective. For the support to be effective, the stiffness of support member A has to be at least one order of magnitude higher than the stiffness of the pipe. Here, the pipe stiffness is very high due to the relatively short distance from the nozzle to the support.
- A gap is generally required in the actual installation of a stop. Therefore, if a stop is placed too close to the nozzle connection, its effectiveness is questionable due to the inherent gap. As shown in Figure 7-6 (c), the pipe has to be bent or moved a distance equal to the gap before the stop becomes active. Due to the closeness of the stop to the equipment, nozzle stresses will often reach severe levels even before the pipe reaches the stop. This configuration is not acceptable because the equipment generally can only tolerate a much smaller deformation than the construction gap of the stop.
- Choking is another problem relating to the gap at the stop. Some engineers are aware of the consequences of the gap at the stop mentioned above and try to solve it by specifying that no gap be allowed at the stop. This gives the appearance of solving the problem, but another problem is actually waiting to occur. As shown in Figure 7-6 (d), when the gap is not provided, the pipe will be choked by the stop as soon as the pipe temperature starts to rise. We generally remember to pay attention to the longitudinal or axial expansion of a pipe, but we often forget that the pipe expands radially as well. When the temperature rises to a point when the radial expansion is completely choked by the support, the pipe can no longer slide along the stop surface. The axial expansion will then move upward, pushing the entire machine upward.

Expansion Joints

An alternative solution to keeping allowable nozzle loads in check involves the use of bellows expansion joints. Bellows expansion joints are popular in the exhaust systems of steam turbine drives which typically have extremely low allowable pipe loads for pipes 8 inches and above. Bellows joints are also often used for fitting units coming off a common header, as shown in Figure 7-7 (b). A properly installed and maintained bellows expansion joint should have the same reliability as other components, such as flanges and valves. However, in real applications, expansion joints are often considered undesirable due to anticipated maintenance problems. For instance, when covered with insulation, the expansion joint looks just like thickly insulated piping. Nobody knows exactly what is going on inside the mixed layers of covering. Due to "blindness anxiety," many installers have resorted to an uninsulated arrangement. This not only creates an occupational safety hazard, but can also lead to cracks due to thermal shock from the environment and/or weather changes. In refineries, fires around bellows-type expansion joints have often led to disaster.

One important factor often overlooked by engineers in the installation of a bellows expansion joint is the pressure thrust force inside the pipe. The bellows is flexible axially. Therefore, the bellows is not able to transmit or absorb the axial internal pressure end force. This pressure end force has to be resisted either by the anchor at the equipment or by the tie-rod straddling the bellows. With the exception of very low pressure applicators, such as the pipe connected to a storage tank, most equipment items are not strong enough to resist a pressure end force equal to the pressure times the bellows cross-sectional area. The pressure thrust force has to be taken by the tie-rod. These facts are not obvious to everyone and may result in some operational difficulties. Figure 7-7 illustrates two actual problems. Figure 7-7 (a) shows



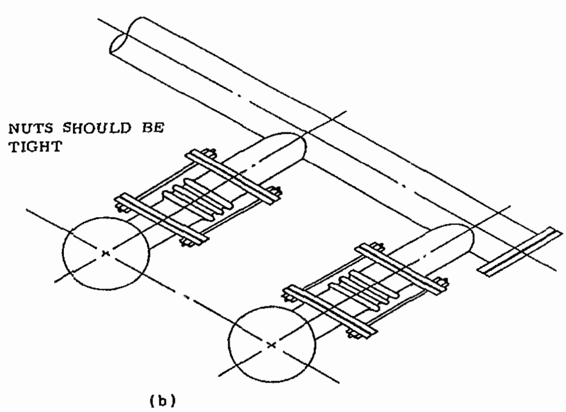


Figure 7-7. Tie-rods on expansion joints.

one of many steam turbine exhaust configurations installed in petrochemical plants. The expansion joint layout scheme appears to be sound, but the construction may not be done properly. When the base elbow is anchored, the tie-rod loses its function as soon as the pipe starts to expand. In this case, the pipe expands from the anchor toward the bellow joint, making the tie-rod loose and ineffective. The large pressure thrust force pushes the turbine, often causing shaft misalignment and severe vibrations. Figure 7-7 (b) depicts a similar situation. In one plant, the bellow expansion joints were used solely for fitting up the connections. The tie-rods were supposed to be locked; however, before start-up, an engineer had loosened the tie-rod nuts, apparently thinking the tie-rods defeated the purpose of the expansion joint. The turbine encountered serious vibration and it took quite a while before it was discovered that the problem was caused by the loose tie-rods. When the nuts are loose, the pressure end force simply pushes the machine out of alignment.

Other Practical Considerations

As can be seen, pipe stress reductions are not always easy to achieve. Especially when dealing with the low allowable nozzle loads specified for some equipment, the technique can become tricky and very often works only on paper. Other practical approaches may have to be explored to further improve overall reliability. One very important resource not to be overlooked is the experience found in operating plants. We have seen good, simple working layouts changed to complicated and questionable layouts only because a computer liked it that way. Undoubtedly, computers are important tools, but they are only as good as the information we give them. Since there are parameters such as friction, anchor flexibility, etc., that cannot be given accurately, computer results need to be interpreted carefully. It is time to realize that if something works well in a plant day in and day out, it should be considered good, regardless of whether or not the computer predicted it to be good. The process of examining and incorporating field experience is very important in designing a good, reliable plant.

Other solutions such as the use of sliding supports, spring supports, and more compact in-line arrangements as shown in Figure 7-8 also merit serious consideration. It is understood that engineers do not feel too confident about movable assemblies, but it is important to understand the difference between the movement of the whole assembly and the movement of only the pump or turbine. When the whole assembly moves, shaft alignment can still be maintained, provided the distortion of the equipment is not excessive. This pre-supposes that the piping load is still within the allowable range. It should be noted, however, that movable assemblies are just potential alternatives. One should not be oversold on the idea and blindly use sliding or spring-supported schemes in a plant. To make the sliding base or the spring-support scheme work, an extra strong baseplate is required. Then again, if we have that strong of a baseplate in the first place, it may well be possible to substantially increase the allowable piping load.

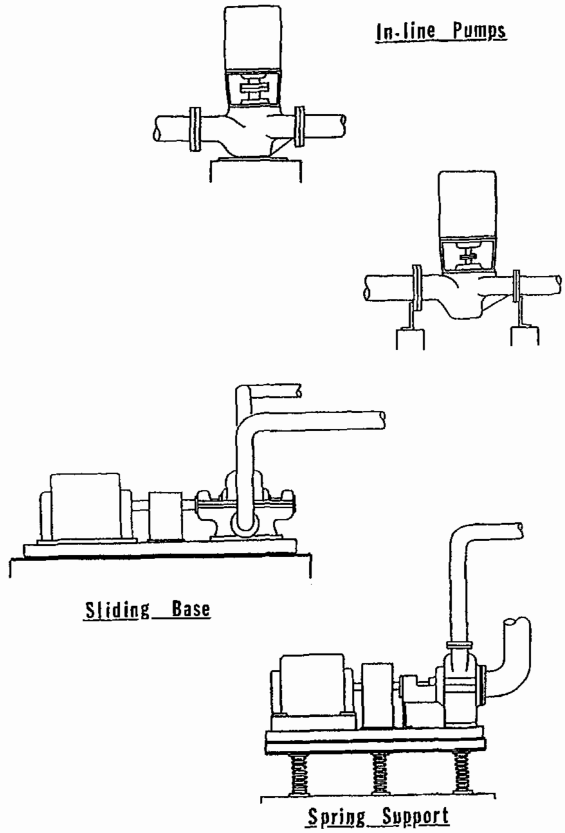


Figure 7-8. Alternative machine assemblies.

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NEMA SM-23, "Steam Turbines for Mechanical Drive Service," National Electrical Manufacturers Association, Washington, D.C.

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Chapter 8

Startup Responsibilities

Summary of Startup Preparations for Process Plant Machinery

A number of key elements are indispensable to ensuring successful commissioning of rotating machinery in process plants. These include the following mix of mandatory and contingency actions.

1. Develop equipment lists for general reference and progress tracking.
2. Assign mechanics during the erection period to observe or execute:
   a. Machine assembly:
      - Lifting and handling procedures.
      - Cleaning, inspection, and installation of bearings and seals.
      - Checking and recording critical clearances, and pre-operation settings and adjustments.
      - Full inspection of most centrifugal pumps.
      - Machinery-related instrument installation and adjustments.
   b. Correct shaft alignment.
   c. Parallelism and gaskets for major flange connections.
   d. External flush systems cleanup.
   e. Piping system cleaning for:
      - Compressors.
      - Turbines.
3. Set up training program for special machinery operation and repair:
   a. Consider vendor service representatives, as available.
   b. Use classroom instruction together with practical demonstrations.
   c. Use contract or on-loan startup advisors as time permits.
   d. Verify soundness of auxiliary systems, starting interlocks, and shut-down protection alarms, and provide routine checking and trouble-shooting during machinery operations.
   e. Review vendor's instruction books, cross-sectional drawings, and inspection procedures. Verify completeness of records.
   f. Expose mechanics to the same training as operators in such key subjects as:
      - Steam-turbine operation.
      - Centrifugal-compressor shaft seal system.
      - Recognizing machinery distress.

4. Assign the plant's machinery engineers to participate in startup activities full-time.
   a. Purposes and benefits:
      - Specialized training.
      - Improved communication and implementation of advisor's recommendations.
      - Permits round-the-clock specialist manning.
      - Assures continuity upon departure of temporary startup advisors.
   b. Duration of assignment:
      - Plant machinery engineers should continue in full-time startup assignment for one to two months after the plant goes on stream.
5. Train the plant's electricians and instrument mechanics on machinery accessories:
   a. Oil system.
   b. Governors.
   c. Starting interlocks, alarms, and safety shutdowns.
6. Train operators in machinery areas:
   a. Machinery auxiliary systems, including interlocks, alarms, and shutdown features.
   b. Testing of auxiliaries during operation.
   c. Machinery conditions requiring emergency shutdown.
   d. Avoiding the kinds of operating errors that can damage machinery.
7. Prepare specific operating instructions for major machinery units:
   a. Prepare before run-in, revise before plant startup, finalize after successful startup is completed.
   b. Integrate vendor's instructions for driver and compressor into process startup instructions.
   c. Prepare specific startup and shutdown procedures; checklists and data lists for monitoring: data on interlocks, alarms, and shutdown features; process factors; etc.
8. Ascertain availability of test equipment for run-in and operation:
   a. Stroboscope—high-speed coverage.
   b. Hand-held vibration-measuring equipment.
   c. Real-time vibration analyzer.
   d. Computerized data-acquisition systems.
9. Identify outside sources for special testing or balancing:
   a. Investigate nearby facilities for emergency service such as:
      - Vibration analysis.
      - Dynamic balancing of rotors.
      - Capacity to handle largest rotor.
      - Balance quality achievable with available machines per recent experience.
   b. Dynamic trim balancing in place:
      - Computerized techniques available.
      - Special equipment required.
      - Skilled technicians required.
   c. Metallurgical testing laboratory.

10. Investigate the availability of expert consultants:
    a. Vibration.
    b. Welding.
    c. Metallurgy.
11. Investigate plan and facilities for repair of remote vendors' equipment:
    a. Identify possible locations.
    b. Investigate qualifications.
    c. Make advance contacts.
12. Investigate local repair facilities:
    a. Larger machine tools than available in plant shop; special shop facilities or tools.
    b. Special casting repair techniques ("Metalock®" and "Metalstitch®")
    c. Welding and metallurgy.
13. Identify machinery vendors' service personnel:
    a. Establish procedures for obtaining services of vendor representatives for run-in and startup operations.
        - Determine official contact and responsible management.
        - Also make advance contacts for equipment where service representative is to be on an "on call" only status, such as for governors on steam turbines, materials-handling equipment, gearing, centrifuges, etc.
    b. Assess qualifications of assigned representatives quickly and obtain replacements if qualifications are unsatisfactory.
    c. If the erection advisor continues on as the startup advisor, verify that he is qualified in this area.
    d. Utilize vendor representatives for training plant personnel as time permits.
    e. Assign qualified plant mechanics to work supervised by vendor representatives.
14. Arrange preventive maintenance details:
    a. Prepare equipment:
        - Portable shelter.
        - Rotor lifting and supporting rig.
        - Verify access path and lifting position of mobile crane.
    b. Develop pre-planned inspections of critical equipment for execution during unplanned, brief plant operation interruptions.
    c. Develop overall plan for preventive maintenance services on all equipment items.
    d. Establish records of inspections, repairs, and part replacements:
        - Separate records for each major machinery item.
        - Use special forms, with sketches for recording vibration and other critical operating parameters.
        - Use vendors' instruction books and startup advisors for developing forms.
        - Plan overhaul technique of most critical machines in detail (e.g., instrument air compressor).
15. Understand spare parts situation:
    a. Status of deliveries.
    b. Warehouses properly organized for pre-startup period.

c. Spare rotor storage and protection.
d. Review stocking for completeness and adequacy.

16. Outline lubrication requirements:
    a. Determine requirements for each machine.
    b. Verify appropriate product equivalent of vendor-recommended lubricants.
    c. Procure ample quantity for startup. Stock quantity to replenish possible seal leakage. Allow for discard of initial charge.
    d. Periodically sample test during operations.

Machinery Startup Review Tasks

The preceding section outlined startup preparation in broad terms. These preparatory tasks can be further broken down into completeness reviews, quality assurance tasks, cleanliness checks, etc.

The following man-hour percentages could be considered representative for executing the field construction completeness review tasks associated with the startup of a world-size steam cracker (ethylene plant):

Review completeness of installation ("but-list")—21%

- Traps
- Vents
- Valves
- Plugs
- Coupling guards
- Lube supply lines

Review maintainability of equipment—7%

- Spool pieces
- Shims
- Auxiliary line interference
- Alignment devices

Ensure long-term, troublefree operation—4%

- Determine offset values needed to accommodate thermal growth
- Verify stress-free installation of machinery piping
- Gear-tooth contact checks

Driver solo runs and rotation checks—9%

- Steam-turbine overspeed trip settings
- Vibration measurements

Cleanliness checks—5%

- Lube and seal oil systems
- Steam-turbine inlet piping
- Pump mechanical seals

Autostart simulations (joint mechanical/instrument technician effort)—1.5%

- On lube and seal oil auxiliaries
- On other autostart drivers

Documentation, i.e., assembling the following data—3.5%

- Acceptance forms
- Equipment record folders
- Computerized record input data form
- API data sheets
- Cross-sectional drawings
- Spare parts availability
- Mechanical-seal dimensions
- Alignment form
- Startup failure history
- Startup checklists
- Lubrication survey

Machinery repairs—39%

- Precautionary disassemblies
- Component (material) modifications
- Cleaning, reassembling, and general repairs

Miscellaneous—11%

Executing the review tasks described on the preceding pages will be facilitated by a series of checklists. These must be developed to help personnel conduct the reviews and execute the various tasks in reasonably uniform fashion. First and most important is a checklist restating every construction- and installation-related item contained in the job specification documents. A typical checklist should look similar to the one represented in Figure 1-36. Machinery startup review personnel must use these checklists to verify that the installation complies with the job specification and that it is ready for imminent startup. Having observed compliance or deviations, the engineer, technician, or millwright should note his comments on the appropriate completeness summary.

Figures 8-1 and 8-2 represent completeness summary forms for centrifugal pumps and general-purpose steam turbines. Similar summaries should be used for all other categories of rotating machinery.

Assembling a Mechanical Procedures Manual

While performing the various startup review tasks, the review team can assemble a large number of documents which may be helpful later to the plant maintenance work forces.

Construction contractors executing large petrochemical projects are using a multitude of written guidelines dealing with the installation of rotating equipment. Many of these procedures describe future plant maintenance tasks and should, therefore, be assembled as soon as possible. Machinery startup engineers should be aware of this requirement and should assist the plant maintenance supervisor in identifying documentation to be collected and retained for future use. Table 10-4 represents a typical cross section of documents originating from various sources: design contractors, field contractors, startup engineers, manufacturer's field representatives, etc. At the termination of a major plant startup, these and other documents should be placed in a so-called "Mechanical Procedures Manual" and given to plant maintenance personnel for future use and reference.

Machinery Startup Reporting Structure

The startup responsibilities outlined earlier in this chapter can be effectively discharged by two organizations, one a mechanical startup section and the other a technical startup section. The technical startup section is manned by engineers and technicians with prior startup experience. Their task is to *define* all necessary steps leading to successful commissioning and satisfactory long-term operation of rotating machinery. The mechanical startup section is supervised by a machinery engineer

	Completeness Summary for Rotating Machinery	**Rev./Date** NOTE: This summary should be used with: **Detailed Checklist for Rotating Machinery**
Block #____ System # ____	**Centrifugal Pumps**	**Equipment #________**

Baseplate		Pipe Supports	
Case		Small-Bore Piping	
Packing or Seal Gland		Lubrication	
Packing or Seal Piping		Guards	
Suction Piping			
Strainer			
Discharge Piping			

Remarks, Incomplete Items, Deviations, etc:

Mechanical Inspector	(M)		Name ________
Process Inspector	(P)		Date ________
Mechanical Engineer	(T)		
Process Engineer	(T)		

Figure 8-1. Completeness summary form for centrifugal pumps. This summary should only be filled in by the reviewer after a careful on-site review has verified compliance with a "detailed checklist for rotating machinery." The reviewer will normally compile such a detailed checklist from the purchaser's original specification.

	Completeness Summary for Rotating Machinery	**Rev./Date** NOTE: This summary should be used with: **Detailed Checklist for Rotating Machinery**
Block # ___ System # ___	**G/P Steam Turbines**	**Equipment #______**

Baseplate		Small-Bore Piping	
Case		Lubrication	
Gland Condenser Piping		Guards	
Inlet Piping			
Strainer			
Exhaust Piping			
Pipe Supports			

Remarks, Incomplete Items, Deviations, etc:

Mechanical Inspector	(M)		Name ________
Process Inspector	(P)		Date ________
Mechanical Engineer	(T)		
Process Engineer	(T)		

Figure 8-2. Completeness summary form for general-purpose steam turbines. The reviewer should be guided by a careful scrutiny of the purchaser's original specification and procurement documents. For instance, if 12-gauge expanded sheet metal had been specified for "guard," it would be necessary to ascertain that 12-gauge material, expanded so as to permit on-line stroboscopic examination of coupling condition, has, in fact, been provided. Only then would the reviewer sign this form.

and is staffed by mechanical supervisors, foremen, and machinists whose principal task is to *execute* all necessary steps leading to the same results.

As can be seen from the organization chart in Figure 8-3, the two section supervisors are preferably reporting to the same startup leader or manager. The reporting route indicated in Figure 8-4 has been employed and can also be made to work. However, the latter's effectiveness can only be assured if the two section supervisors share mutual background and trust and respect each other. If their relationship is

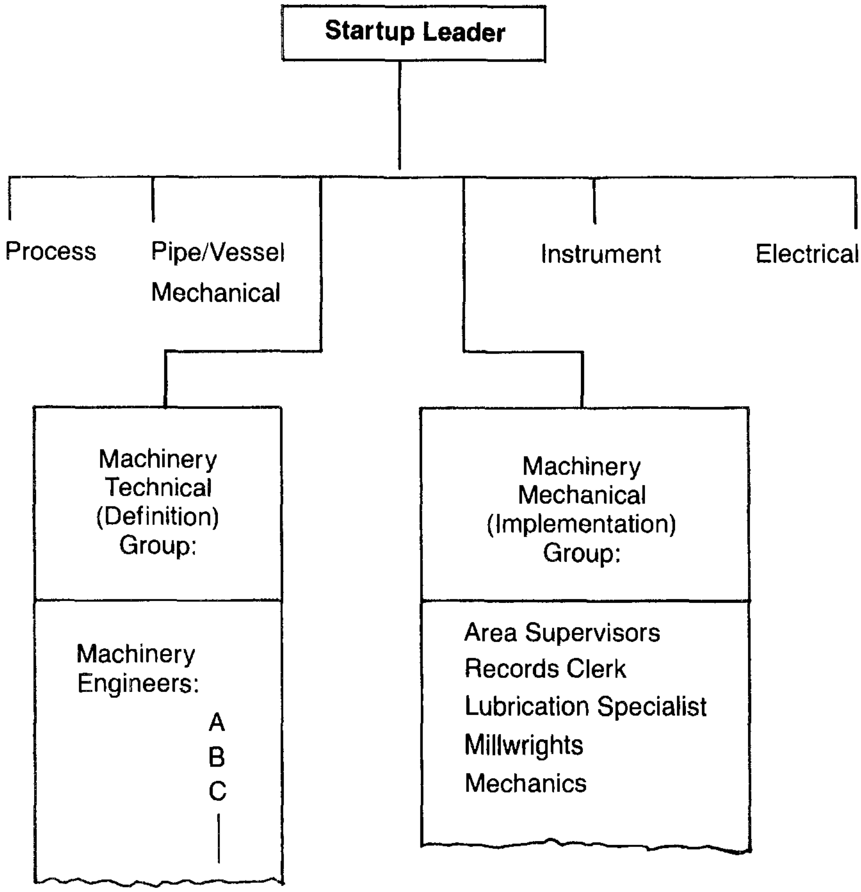


Figure 8-3. Machinery startup sections reporting to the same startup leader.

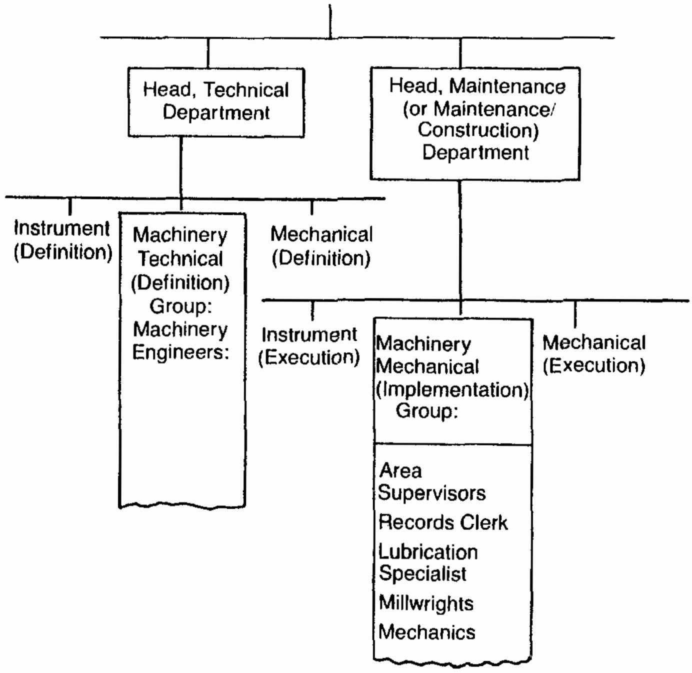


Figure 8-4. Machinery startup sections reporting to separate branches of organization.

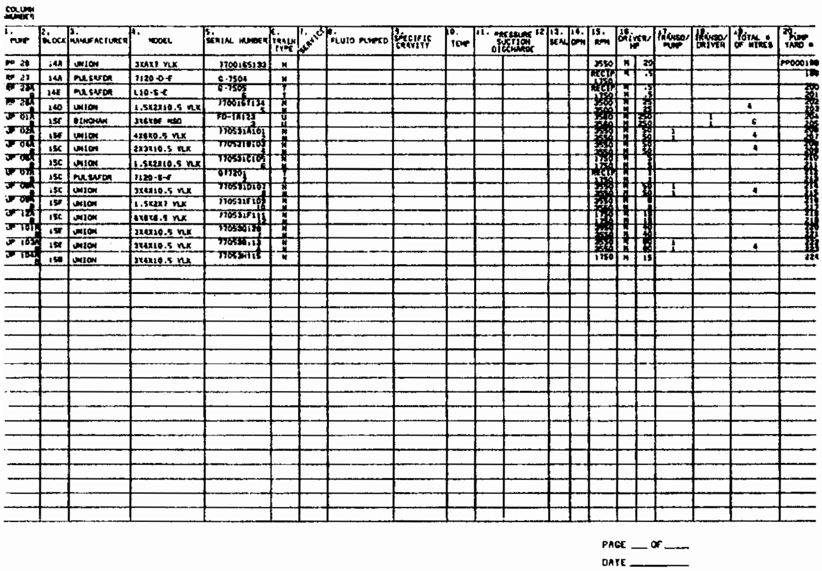
lacking in these elements, a game of "one-upmanship" and finger pointing may result. Startup progress and effective utilization of available resources could suffer from this kind of relationship.

Documentation for Effective Tracking of Progress

Machinery commissioning targets and accomplishments must be documented every step of the way if the startup is to progress in an orderly fashion.

Once a basic set of equipment listings has been prepared, it can be readily modified in the word processor/computer or by spreadsheets. This allows the document to be readily adapted to a variety of documentation needs. Figure 8-5 serves as an example.

To begin with, Figure 8-5, represents the basic tabulation of pertinent pump data in a given area of the plant. The basic, or master tabulation contains 20 columns of information:



COLUMN NUMBER

1. PUMP	2. BLOCK	3. MANUFACTURER	4. MODEL	5. SERIAL NUMBER	6. TRAIN TYPE	7. SERVICE	8. FLUID PUMPED	9. SPECIFIC GRAVITY	10. TEMP	11. PRESSURE SUCTION	12. DISCHARGE	13. SEAL	14. OPN	15. RPM	16. DRIVER/ HP		17. TRANSD/ PUMP	18. TRANSD/ DRIVER	19. TOTAL # OF WIRES	20. PUMP YARD #
PP 26	14A	UNION	3X4X7 VLK	7700165133	M									3550	M	20				PP0001[illegible]
PP 27	14A	PULSAFDR	7120-D-F	G-7504	M									RECIP	M	.5				[illegible]
PP 28A	14E	PULSAFDR	L10-S-E	G-7505	T									[illegible]	M	.5				[illegible]
PP 28A	140	UNION	1.5X2X10.5 VLK	7700161134	M									[illegible]	M	[illegible]			4	[illegible]
JP 01A	15F	BINGHAM	3X6X9F MSD	FD-1X123	U									[illegible]	M	[illegible]		1	6	[illegible]
JP 02A	15F	UNION	4X6X10.5 VLK	770531A101	M									[illegible]	M	[illegible]	1		4	[illegible]
JP 04A	15C	UNION	2X3X10.5 VLK	770531B103	M									[illegible]	M	[illegible]			4	[illegible]
JP 06A	15C	UNION	1.5X2X10.5 VLK	770531C105	M									[illegible]	M	[illegible]				[illegible]
JP 07A	15C	PULSAFDR	7120-S-F	G7720	T									[illegible]	M	[illegible]				[illegible]
JP 08A	15C	UNION	3X4X10.5 VLK	770531D107	M									[illegible]	M	[illegible]	1		4	[illegible]
JP 09A	15F	UNION	1.5X2X7 VLK	770531F109	M									[illegible]	M	[illegible]				[illegible]
JP 12A	15C	UNION	6X8X8.5 VLK	770531F111	M									[illegible]	M	[illegible]				[illegible]
JP 101A	15F	UNION	3X4X10.5 VLK	770530I[illegible]	M									[illegible]	M	[illegible]				[illegible]
JP 103A	15F	UNION	3X4X10.5 VLK	770530[illegible]13	M									[illegible]	M	[illegible]	1		4	[illegible]
JP 104A	15B	UNION	3X4X10.5 VLK	770530H115	M									1750	M	15				224

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Figure 8-5. Basic tabulation of pertinent pump data.

Column 1:	Plot plan or item number.
Column 2:	Block or plant area.
Column 3:	Manufacturer.
Column 4:	Model designation.
Column 5:	Manufacturer's serial number.
Column 6:	Train type. It is highly advantageous to produce isometric sketches identifying the various train types, as shown in Figure 8-6. Supervisory personnel can rapidly visualize the relative size and complexity of machinery trains by referring to these train isometrics. A second variant of these isometrics should be marked up to define where to install vibration transducers or to show operating technicians where to take vibration readings.
Column 7:	Pumping service designation.
Column 8:	Fluid pumped (pumpage).
Column 9:	Specific gravity. This information is important when mining whether the pump is suitable for run-in of water.
Column 10:	Pumping temperature.

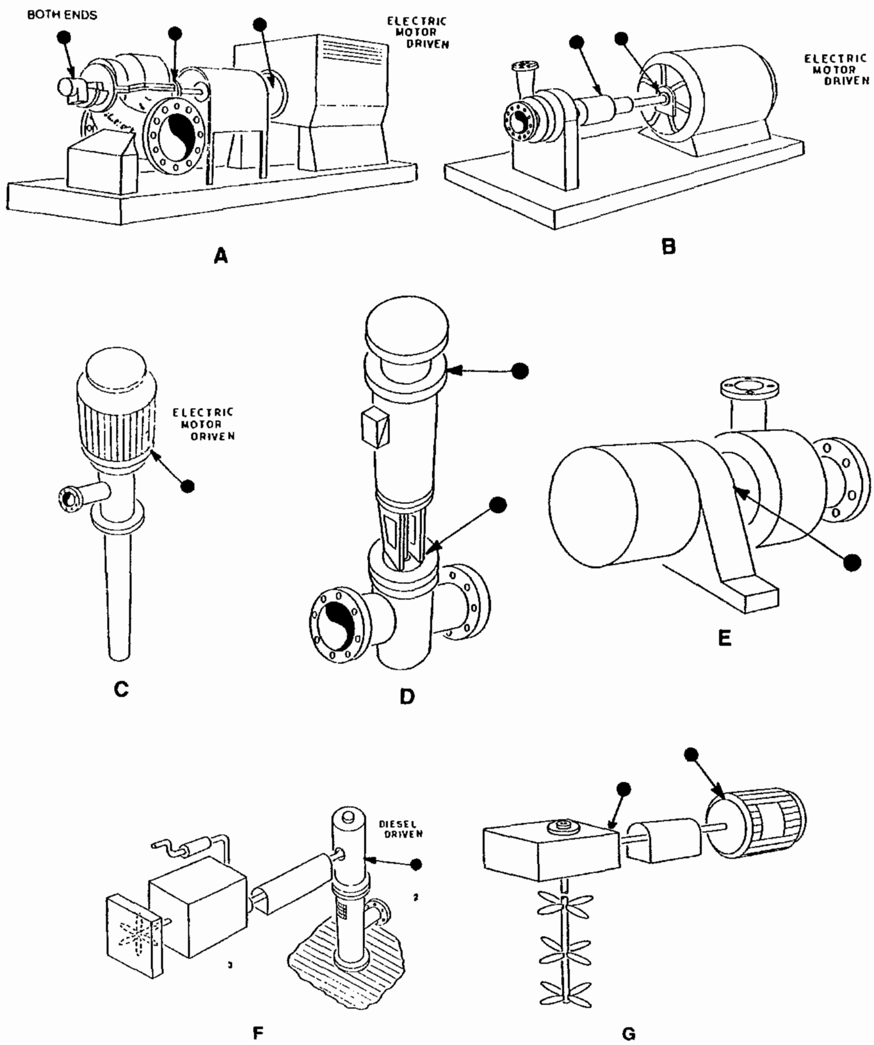

Figure 8-6. Typical isometrics of various train types.

Column 11:	Pump suction pressure.
Column 12:	Pump discharge pressure.
Column 13:	Type of mechanical seal. (T = tandem, P = packing, S = single, SG = single gas-type, D = double, DG = double gas-type).
Column 14:	Flow rate, gpm/or cubic meters per hour.
Column 15:	Shaft speed, rpm.
Column 16a:	Driver type (M = motor, ST = steam turbine).
Column 16b:	Driver power output, HP or kW.
Column 17:	Number of transducers for incipient failure detection on pump.
Column 18:	Same as column 17, on driver.
Column 19:	Total number of instrument wires available.
Column 20:	In-plant inventory identification (yard number).

Figure 8-7 shows the basic equipment tabulation modified to track major checkout segments as the startup progresses. The form is now essentially used for scheduling purposes. Columns 1, 2, 3, 4, and 6 have been retained. Subsequent columns identify anticipated and actual checkout dates, partial completion, etc.

COLUMN NUMBER

1. PUMP	2. BLOCK	3. MANUFACTURER	4. MODEL	TRAIN TYPE	GROUTING	AUX PIPING	STRAINER	ALIGNMENT	LUBRICATION	DRIVER SOLO	COUPLED	FINAL PIPING	RUN-IN	TARGET COMPLETION	ACTUAL COMPLETION	REMARKS
RP 26	14A	UNION	3XAX7 VLK	M												
RP 27	14A	PULSAFOR	7120 D-E	M												
RP 28A B	14E	PULSAFOR	L10-S-E	T T												
RP 29A B	14D	UNION	1.5X2X10.5 VLK	N N												
UP 01A B	15E	BINGHAM	3X6X9E MSD	U U												
UP 02A B	15E	UNION	4X6X0.5 VLK	N N												
UP 04A B	15C	UNION	2X3X10.5 VLK	N N												
UP 06A B	15C	UNION	1.5X2X10.5 VLK	N N												
UP 07A B	15C	PULSAFOR	7120-S-E	T T												
UP 08A B	15C	UNION	3X4X10.5 VLK	N N												
UP 09A B	15F	UNION	1.5X2X7 VLK	N N												
UP 12A B	15C	UNION	6X6X8.5 VLK	N N												
UP 101A B	15E	UNION	3X4X10.5 VLK	N N												
UP 103A B	15E	UNION	3X4X10.5 VLK	N N												
UP 104A	15B	UNION	6X6X10.5 VLK	N												

PAGE __ OF __
DATE: ______

Figure 8-7. Equipment tabulation for major checkout segments at startup.

PUMP	BLOCK	MANUFACTURER	MODEL	SERIAL NUMBER	TRAIN TYPE	SERVICE	FLUID PUMPED	SPECIFIC GRAVITY	SEAL	SEAL MODEL	SEAL DWG.	COUPLING TYPE	COUPLING DRAWING	RPM	DRIVER/HP	PUMP YARD #
RP 20	14A	UNION	3X4X7 VLK	7700185133	M				T					3550	M 20	PP0001[illegible]
RP 27	14A	PULSAFOR	7120-D-E	0-7504	M				P					RECIP	M .5	1[illegible]
RP 28A/B	14C	PULSAFOR	L10-S-E	0-7505/6	T/T				P					RECIP/1750	M .5/M .5	200/201
RP 29A/B	14D	UNION	1.5X2X10.5 VLK	7700181134/5	M/M				T					3500/3500	M 25/M 25	202/203
UP 01A/B	15E	BINGHAM	3X8XDE MSD	FD-1A123/4	U/U				T					3560/3560	M 250/M 250	204/205
UP 02A/B	15E	UNION	4X8X0.5 VLK	770531A101/2	M/M				T					3550/3550	M 50/M 50	206/207
UP 04A/B	15C	UNION	2X3X10.5 VLK	770531B103/4	M/M				T					3550/3550	M 50/M 50	208/209
UP 06A/B	15C	UNION	1.5X2X10.5 VLK	770531C105/6	M/M				S					1750/1750	M 5/M 5	210/211
UP 07A/B	15C	PULSAFOR	7120-S-E	037201/2	T/T				P					RECIP/1750	M 1/M 1	212/213
UP 08A/B	15C	UNION	3X4X10.5 VLK	770531D107/8	M/M				T					3550/3550	M 50/M 50	214/215
UP 09A/B	15F	UNION	1.5X3X7 VLK	770531E109/10	M/M				S					3550/3550	M 8/M 8	216/217
UP 12A/B	15C	UNION	8X8X8.5 VLK	770531F111/12	M/M				T					1750/1750	M 15/M 15	218/219
UP 101A/B	15E	UNION	3X4X10.5 VLK	770530126/7	M/M				T					3550/3550	M 40/M 40	220/221
UP 103A/B	15E	UNION	3X4X10.5 VLK	770530113/4	M/M				T					3550/3550	M 80/M 80	222/223
UP 104A/B	15B	UNION	3X4X10.5 VLK	770530115	M				T					1750	M 15	224

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Figure 8-8. Equipment tabulation emphasizing coupling and mechanical seal data.

Figure 8-8 represents another modification of the basic equipment tabulation. This time, the emphasis is on coupling and mechanical-seal data.

Pump outages experienced during pre-commissioning activities are logged in to keep track of both duration and primary cause. Figure 8-9 represents a typical outage log sheet with entries following a time sequence of events.

A different set of startup documentation is shown in Figures 8-10 and 8-11. These figures represent an event log for major compressors and a trip/shutdown log for a given machine, respectively. Note especially how Figure 8-11, representative of dozens of sheets generated during a major startup, affords an overview of several important entries: purpose of run, discipline responsible for shutdown, duration of shutdown, cumulative operating time, and downtime.

Figure 8-12 shows a checklist used for centrifugal-pump field installation and initial operation. Used in conjunction with field-posted startup instructions (refer back to Figure 1-26), a similar checklist containing work items reflecting a given project philosophy will go a long way toward reducing installation and commissioning oversights.

AREA AND ITEM	DATE DECOMMISSIONED	DATE RECOMMISSIONED	DAYS UNAVAILABLE	REASON FOR OUTAGE													
				VENDOR DESIGN	VENDOR ASSEMBLY	CONTRACTOR SPEC	PRODUCT VISCOSITY	IMPROPER FLUSH	DEBRIS IN LINE	LOSS OF SUCTION	SEAL LEAKAGE	BEARING FAILURE	COUPLING FAILURE	GASKET LEAKAGE	DESIGN MODIFIC.	OTHER	
MP-01A	4/20	4/21	1		√												
MP01B	4/20	4/22	1		√												
MP-02A	4/21	4/22	1		√												
MP-02B	4/21	4/24	2		√												
LP-10B	4/21	4/24	3								√						
MP-01B	4/22	4/23	1											√			
MP-01A	4/23	4/24	1											√			
LP-12B	4/26	4/27	1									√					
LP-12B	4/28	4/29	1							√							
LP-16	4/28	4/29	1							√							
LP-07A	4/28	4/30	1		√												
LP-08A	4/29	4/30	1						√								
VP-01B	4/30	5/2	3							√							
NP-08B	4/30	5/2	2						√								
NP-08A	5/1	5/2	1		√												
NP-05A	5/4	5/6	1						√								
LP-01A	5/5	5/9	2						√								
LP-02A	5/6	5/7	1						√								

PUMP OUTAGES EXPERIENCED PRIOR TO FEED-IN

Figure 8-9. Outage log sheet.

Similar checklists should be developed for other machinery categories and should be used by mechanical work forces and operating technicians involved in equipment commissioning activities. These forms will prove invaluable for any future CMMS implementation work.

(*text continued on page 360*)

Unit	Date	Solo Runs: Description of Run	Remarks
CT-01 Turbine Solo	7/30/98	Turbine solo run: checked mechanical bolt trip three times; running approximately six hours.	Vibrations good. Outboard bearing thermocouple reading Hi (215°F), trip set at 4275 rpm, had trouble with vacuum.
CT-01 Turbine Solo	7/31/98	Turbine solo: checked electronic trip three times; running time approximately four hours.	Vibrations good. Outboard bearing TC reading Hi 214°F, trip set at 4170 rpm, vacuum problems.
CT-02 Turbine Solo	8/02/98	Turbine solo: checked bolt trip and electronic trip each three times; running time approximately six hours.	Vibrations good, bearing temps good, trip set at 6600 mechanical bolt trip, 6540 electronic trip, had trouble with vacuum. Running time during solo runs for C-01: 10 hours, for C-02: 6.0 hours.
C-01 Train	8/04/98	Air run-in: First air run-in of compressor train: low pressure—first three stages of compression; running time approximately five hours.	Vibrations all good—high temperature turbine outboard bearing, some "stall" noticeable on double flow inboard side at 1500 rpm—none above critical speed, had troubles with condenser, ejectors and seal oil ΔP.

Figure 8-10. Event log for charge gas trains.

Date	Start (Hour)	Trip (Hour)	Purpose	Discipline	Cause of Shutdown	Tripped by	Shutdown Time	Accumulated Oper. Time	Accumulated Downtime
8/28	1715	(8/30) 1419	Air dry-out	Instruments	Problems with RCQ logic	RCQ	0.4	241.95	236.80
8/30	1445	1640	Air dry-out	Instruments	Problems with RCQ logic	RCQ	0.25	243.85	237.05
8/30	1650	(9/1) 1042	Air dry-out	Instruments Machinery	Problems with RCQ logic	RCQ	8.8	261.75	245.85
9/1	1930	1940	Air dry-out	Mechanical	Gasket in 1500# system blown	Manual	4.3	261.90	250.15
9/2	0000	(9/4) 0100	Air dry-out	Utilities	Loss of boilers	Manual	13.5	310.90	263.65
9/4	1430	(9/5) 1510	Air dry-out	Machinery	Woodward people worked on governor	Manual	0.15	335.55	263.80
9/5	1520	2040	Air dry-out	Machinery	Governor problem	Manual	0.65	340.9	264.45

Figure 8-11. Trip/shutdown log.

Work Request No. ______________ Date ______________
Pump Item No. ________ Yard No. ________ Location ________
Technician (Name) ______________ Supervisor ______________
Telephone Extension ________

How To Use This Checklist:

- During pre-startup operations

  Place your initials after items you have personally witnessed to be correct or attach an initialled copy of the Machinery Field Installation Checklist to cover items which Project Management personnel have reviewed for you.

- When the plant is fully operational

  Re-install pumps in accordance with this checklist only. Place your initials after items you have personally witnessed to be correct. Do not omit any steps.

Mechanical Preparation

	Initial
1. *Safety*	
a. Complied with lock-out and tag-out procedures and wore proper safety equipment.	
2. *Preparing pump and baseplate for installation*	
a. Cleaned and removed all burrs from bottom of pump feet and baseplate pads.	
b. Cleaned pump and piping suction and discharge flanges.	
3. *Pump installation on baseplate*	
a. Installed pump on base plate and checked the following:	
1. Pump feet and/or supporting flanges are square with baseplate pads and seated solidly.	
2. After tightening the base bolts, the pump was checked for level and leveled, if necessary.	
3. The pump was checked to make sure that it was slightly higher than the driver so that shims could be placed under the driver's feet during final alignment.	
b. Horizontal pumps and drivers rated 100 or more HP have been provided with tooling balls to facilitate Essinger alignment bar check. Verified balls to be in place.	
c. IFD isocoupler pads or IFD transducers have been bonded to equipment casings as listed below: ☐ driver inboard ☐ driver outboard ☐ pump inboard ☐ pump outboard ☐ pump central ☐ none	
4. *Pump and driver's coupling concentricity*	
a. Installed a coupling sweep and dial indicators and checked to make sure that the pump and driver's coupling halves' outside diameters and faces were true with their bores (maximum OD .002, face	

Figure 8-12. Centrifugal pump field installation and initial operation checklist.

(*checklist continued on next page*)

Initial

.001"). Used pry to hold axial movement while checking the coupling-face run-out on sleeve bearing equipment.

b. Recorded Essinger alignment bar readings on pumps and drivers rated 100 or more HP.

5. *Installation of suction and discharge piping*
   a. Checked piping and pump flanges for parallel, and when the pipe was pushed in place without strain the bolts fit free in the flange holes.
   b. Flange alignment is ☐ satisfactory ☐ unsatisfactory
   c. Placed and adjusted dial indicators on the top and side of the pump's coupling hub and/or flange. Observed the dial indicators while the pump's suction and discharge flange bolts were being tightened evenly to prevent piping strains on the pump.
   d. Pump suction strainer is ☐ installed ☐ not installed.

6. *Auxiliary piping*
   a. Cleaned and installed piping and plugs using thread compound. *Note:* All small piping (except water) is to be Schedule 160.

7. *Lubrication (sleeve and ball bearings)*
   a. Cleaned and installed bottom sight glass and snap drain on dry-sump lubricated pumps. Adjusted lubricator assembly to oil-level mark on bearing housing of wet-sump lubricated pumps. Used Enmist-100 on all pumps, Teresstic-32 on turbines. (Exception: Units with centralized pressure lubrication use Teresstic-32 throughout.)
   b. Verified that vertical in-line pumps in hydrocarbon service have thrust bearing installed at motor outboard end.
   c. On dry-sump oil-mist lubricated motors, verified that bearing identification code on nameplate shows thrust-loaded bearings *not* to be equipped with shields (no letter "P" in code sequence). Radial bearings are allowed one or no shield (one letter "P" or no letter "P" in code sequence).
   d. Cleaned oil-mist fittings at bearing housing, checked tubing for cracks, and connected it to distribution block. Adjusted tubing so that it sloped from the manifold to the oil reservoir without sagging, to prevent oil traps. Verified that drain tubing on horizontal motors is led to the drain rim of baseplates.
   e. Observed oil mist and escaping mist from bearings. *Note:* If mist is not observed, check "f" below.
   f. Checked oil-mist generator for proper operation. Check:
      ☐ Adequate oil in reservoir.
      ☐ Oil-mist pressure—20"-30" H_2O.
      ☐ Oil Temperature—95°F-125°F.

(*checklist continued on next page*)

Initial

☐ Air pressure—gauge outside cabinet approximately 30 psig higher than air inside cabinet.

☐ Air temperature 130°F-155°F

8. *Hot-pump piping strain check (300°F and above)*
   a. Placed and adjusted a dial indicator on the side of the pump's coupling hub and/or flange and observed the dial indicator while the pump was being heated as closely as possible to its operating temperature.
   Note: While making this check, do not tie down bearing bracket support on overhung-type pumps.
   ☐ Movement was more than .010" TIR. Made correction to piping supports.
   ☐ Movement was less than .010" TIR. Piping satisfactory.

9. *Alignment of driver to pump*
   a. Reviewed thermal growth calculation and verified pump and driver were offset accordingly.
   b. Connected coupling sweep (with less than .003" total sag) to pump. Installed and adjusted dial indicators to the OD and face of the driver's coupling and recorded the coupling's misalignment and calculated shim adjustments using two-indicator alignment method.
   Note: Do not stack shims; never use more than three shims.
   c. Final alignment after shim correction.

10. *Coupling and coupling guard*
   a. Checked motor rotation (before coupling).
   b. Adjusted flexible disc coupling so that sleeve-bearing motor driver will run in its magnetic center, and tightened set screws to lock it in place.
   c. Greased gear-type coupling and limited its end float to prevent bearing and shaft shoulder contact on sleeve-bearing-type motor
   d. Installed coupling guard.

STOP HERE	Do Not Proceed Unless All of the Above Items Have Been Fully Verified and Properly Initialled.

Operating Preparation

Initial

11. *Special procedures*
   a. Any additional process steps or checkout procedures pertaining to this pump or its driver have been reviewed and instructions issued to operating personnel.
      ☐ supplements issued
      ☐ no supplements required
   b. Verified pump is suitable for run-in on water.

(*checklist continued on next page*)

Initial

12. *Machinery/electrical specialist coverage*
    - ☐ stand-by coverage needed and requested.
    - ☐ stand-by coverage not requested. Operating personnel equipped with instrumentation to take and record necessary surveillance data.
13. *Prepare unit for start* (process operator to verify asterisked items)
    - *a. Verified tandem seal pots were filled with recommended fluid.
    - *b. Turned on seal oil and/or seal flush.
    - *c. Vented packing box of double seal to ensure the box was free of air and full of fluid.
    - d. Adjusted packing-gland bolt nuts to fingertight and observed slight leakage.
    - *e. Opened pump's suction valve. Bled air from pump.
    - *f. Opened pump's discharge valve (when line is under pressure). Cracked discharge valve when line was not under pressure.
    - *g. Started driver and checked for proper pressure and signs of cavitation to check for proper rotation.
14. *Initial startup data*
    - ☐ data below refers to run-in on water.
    - ☐ data below refers to process fluid other than water.

	Design (Values)	Actual (Values)
a. Pump suction pressure, psi.		
b. Pump suction temperature, °F.		
c. Pump discharge pressure, psi.		
d. Pump discharge temperature, °F.		
e. Motor reading, amps.		
f. Flow rate, gpm.		

Initial

15. *Test run*
    - a. Checked and adjusted packing for slight leak (packed pumps only).
    - b. Checked bearings for noise and acceptable temperature.
    - c. Checked for rubbing of seal throttle bushings and pump wear rings.
    - d. Verified that oil level in tandem seal pots remained unchanged.
    - e. Performed the following vibration check at conditions indicated.

      ☐ min flow ☐ rated flow ☐ max flow

      IB bearing—Horiz. Disp. Mils ______ Vel.—in sec ______
      IB bearing—Vert. Disp. Mils ______ Vel.—in sec ______
      OB bearing—Horiz. Disp. Mils ______ Vel.—in sec ______
      OB bearing—Vert. Disp. Mils ______ Vel—in sec ______
      Axial Disp. Mils ______
      Max Displacement: 1.5
      Max Vel. in/sec: .3

    - ☐ Initial vibration check showed excessive vibration and indicated misalignment, and a hot-check alignment adjustment was made.
    - ☐ Vibration check satisfactory.
    - ☐ Doweled pump and driver.

Note: After completion of work give completed checklist to your supervisor. Your supervisor will send it to the "B" Planner

Remarks:

(*checklist continued on next page*)

(text continued from page 353)

Vendor Assistance and Outside Facilities

Duties of a machinery startup team include the determination of how many vendor representatives should assist in sharing machinery preparation tasks. Not only their number but also their respective experience levels and even names should be identified for a given location.

Vendor assistance is required in cases where potential warranty disputes might arise, or where the owner's personnel simply don't have the expertise to supervise the construction contractor. Vendor personnel could be given such additional assignments as spare parts review and millwright training.

The examination and definition of potential outside facilities is a typical contingency step. These facilities may have to be relied on for rotor balancing, emergency repairs, or just plain routine work which cannot be reliably handled by the owner's or contractor's work forces. The dismantling, cleaning, and adjusting of mechanical or mechanical-hydraulic governors is a typical example. A company specializing in this work may be interested in cataloging all governors installed at the facility about to be started up. Similarly, companies with expertise in shaft, bearing, labyrinth, or impeller manufacturing may want to perform cataloging and spare parts dimensional sketching services at no cost to the owner. In any event, their particular expertise needs to be identified just like that of the nearest repair facility which could rapidly handle critically important restoration of major machinery involved in a disastrous failure incident.

Consultants and Contract Assistance

The best insurance against unexpected machinery problems lies in adequate predelivery audits and reliability reviews. This insurance is closely followed by diligent installation supervision and completeness reviews during the time of field installation. Finally, the commissioning instructions should be developed in cooperation with the owner's startup engineer or should, as a minimum, be submitted for his detailed review.

Ideally, then, the owner will draw on the expertise of his own senior professional personnel every step of the way. Unfortunately, these professionals are hardly ever available for the duration of a project, i.e. from its initial planning until the completion of the startup phase. Also, it would be rather presumptuous to assume that the owner's technical staff are thoroughly versed in all conceivable matters begging expeditious resolution during a major startup. Think, for instance, of difficulties with sophisticated shutdown logic, electronic governors, high-speed gearing and bearings, etc. Problems in any of these areas may be handled best by interfacing with capable consultants whose availability has been ascertained beforehand.

Contract assistance may also prove helpful in machinery startup situations. Oftentimes, a plant can draw on the experience of capable retirees whose background may allow them to serve as instructors, training coordinators, machinery repair supervisors, outside shop inspectors, or expeditors.

Chapter 9

Spare Parts and Their Effect on Service Factors

Spare Parts Philosophies

This many-faceted topic is sometimes neglected because it seems to defy solution. After all, the determination of required spare parts seems to be an educated guess at best. Yet unless a policy is established and spare parts procured, a startup can be in serious trouble before it even gets off the ground.

A solid review of failure statistics can provide help with and justification for spare parts recommendations. Historically, problems with major unspared machinery have caused commissioning or startup delays in one out of two train installations *without* reliability reviews, and in one out of six or seven train installations employing pre-installation reliability reviews. In either case, centrifugal compressors and steam turbines have experienced the bulk of the difficulties, followed by large gear-speed increaser units, electric motors in size categories above 1000 HP, reciprocating compressors, gas turbines, and gas expanders. Principal parts affected are tabulated in Table 9-1. Also, Table 9-1 makes an attempt to list spare parts recommended to be stored for initial startup operations and spare parts to be kept on hand for routine, post-startup operation. The relative frequency of replacement or repair is indicated by numbers ranging from 5 (high frequency) to 1 (low frequency).

Spare Parts Storage and Retrieval

The best source for information and sound recommendations on spare parts storage and retrieval is not necessarily an existing petrochemical plant. A reputable consulting company specializing in warehousing and inventory control is often better qualified to set up hardware and software systems to serve this function adequately and efficiently.

Inefficient storage and retrieval will result when major turbomachinery rotors are not hung vertically or when they have to be shipped to storage facilities away from the plant location. Inefficient storage and retrieval will also result when spare parts for one compressor are stored at one end of the building and parts for another compressor are stored near the opposite end, and when parts are not properly cross refer-

Table 9-1
Principal Component Failures Experienced by Major Machinery and Recommended Spare Parts

Machinery and Component	Frequency	S/U Spares	Post S/U Spares
Gear-speed increasers			
Bearings	3	full set	full set
Bull gear	2	full spare	full spare
Pinion	2	full spare	full spare
Coupling	2	1 spare	1 spare
Oil seals	1	1 set	1 set
Reciprocating compressors			
Crankshaft	1	—	—
Cylinders, liners	2	1 set	1 set
Piston rods	2	partial	partial
Valves	5	2 sets	1 set
Crosshead guides	3	1 set	1 set
Crossheads	2	—	—
Crankshaft bearings	2	1 set	1 set
Connecting rods	1	—	—
Connecting-rod bearings	1	1 set	1 set
Lubricators	2	1 unit	1 unit
Pistons	2	—	—
Piston rings	2	1 set	partial set
Centrifugal compressors			
Bearings (journal)	4	1 full set	1 full set
Rotors (shafts/impellers)	3	spare rotor	spare rotor
Labyrinth packing	3	1 full set	1 full set
Bearings (thrust)	2	disc and pads	pads
Couplings	2	full spare	full spare
Seals	5	1 full, 1 wear part set	1 wear part set
Steam turbines			
Gland packing	4	full set	full set
Interstage packing	4	full set	full set
Bearings	4	full set	full set
Rotors	3	full spare	full spare
Nozzles	2	partial set	partial set
Diaphragms	2	—	—
Admission valves	2	packing 2 sets	packing 1 set
Governor	3	full spare	full spare
Hydraulic actuator	2	rebuild kit	rebuild kit
Governor drive gear	4	full spare	full spare
Turning gear	1	—	—
Nozzle-ring bolting	2	full set	full set
Trip throttle valve	1	internals	internals
Electric motors			
Bearings	3	1 set	1 set
Oil seals	2	1 set	1 set
Stator	3	1 unit	1 unit
Rotor	1	—	—
Fan	2	1 unit	1 unit
Gas turbines			
Combustors	3	full set	full set
Compressor rotor	2	full rotor	full rotor

Table 9-1 (Continued)

Bearings	2	2 sets	1 set
Turbine rotor	2	full rotor	full rotor
Fuel controls	3	partial set	partial set
Speed controls	2	partial set	partial set
Gas expanders			
See applicable parts of steam turbines			

Key: No. Event/Train-Year
- 5 One every 1–2 years
- 4 One every 2–3 years
- 3 One every 3–4 years
- 2 One every 4–5 years
- 1 One every 6–8 years

enced, preserved, or labeled with the vendor's parts and drawing numbers in addition to the owner's spare parts identification.

The advent of sophisticated electronic controls for machinery-related instrumentation (e.g., electronic governors) makes it necessary to pay very special attention to appropriate preservation and storage. Dust, moisture, and crush-proof packaging are indispensable. While spare parts for major machinery should be stored in designated areas clustered in close proximity, sensitive electronic components for the same machine are best not located in the same bin with heavy mechanical components.

The storage of large turbomachinery rotors presents special problems and many opportunities for good or bad solutions. Modern user plants generally opt for the highly satisfactory storage method shown in Figure 9-1. Here, the rotors are preserved with an oil-derived coating and hung vertically from large cross beams. The building is humidity controlled, but does not require heating or cooling.

Storage and retrieval methods must represent a logical compromise satisfying several critical requirements. For instance, the night-shift repair crew must have access to spare parts without the help of daytime personnel. At the same time, procedures should be in place which give assurance that the removal of parts from stores is recorded so as to have an up-to-date reading of true inventory levels and reorder requirements.

Spare Parts Documentation

Money spent in documenting spare parts locations, inventory levels, and reordering quantities, and in cross referencing vendor designations, drawing numbers, bills of materials, owner's storage codes, etc. is money well spent. Cross-sectional drawings of major machinery should be combined with component number designations and all other relevant cross references to enable mechanical work forces to locate parts without wasted time or motion.

Complex spare parts documentation, time-consuming retrieval, and ineffective "automatic reordering" procedures have been shown to catalyze illicit substores. An illicit substore is the foreman's desk drawer. This is where he squirrels away special tools, a handful of Kalrez® O-rings, and an occasional stationary bellows mechanical

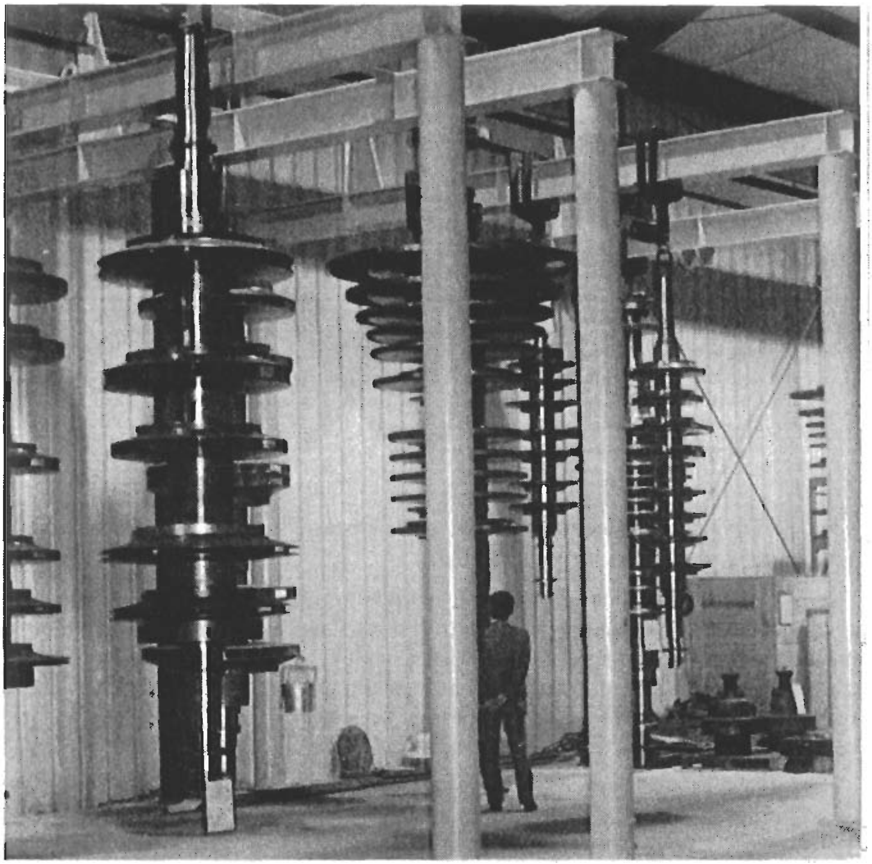

Figure 9-1. Vertically stored turbomachinery rotors. (*Courtesy of Mitsubishi America.*)

seal worth $2000. The foreman does this not because he is a kleptomaniac, but because management has turned a deaf ear to his pleas to remedy these frustrating impediments.

A good spare parts record contains elements similar to those shown in the spare parts identification sheet illustrated in Figure 1-32.

Spare parts identification sheets differ from conventional spare parts documentation or traditional storehouse information in a number of ways. They are primarily intended as an aid to mechanics, machinists, and turnaround planners. These persons require that spare parts information be contained on a single sheet, not in separate catalogs or on computer printouts. In many cases, illustrations are required for positive identification of parts by personnel unfamiliar with either the machinery or the storehouse routine.

Major machinery spare parts documentation sheets must contain all the information needed by mechanical work forces to locate the parts in the storehouse. These documentation sheets must allow mechanics, turnaround planners, and inspectors to verify stock levels, critical dimensions, and suitability of parts. Cross-reference, design-change, and inspection information complete the sheet and make it a standalone, highly useful document.

Chapter 10

Maintenance for Continued Reliability

Most industrial plants have well-defined maintenance procedures; however, they are under frequent and intensive scrutiny as to their cost-effectiveness. There must always be a positive return and value to every maintenance task performed. Without the value concept uppermost in mind, maintenance departments are often accused—sometimes justifiably—of "gold plating." This chapter conveys a few of the organizational and procedural means of avoiding either of the two obvious extremes, i.e., doing too much and not doing enough to optimally affect the bottom line.

Modern Maintenance Approaches and When to Apply Them

The best way to achieve high reliability is to design-out maintenance. Indeed, our text is supposed to make just this point; it has been our intent to approach the issue of designing-out maintenance from different angles. Nevertheless, it must be recognized that at least for the foreseeable future, plant and equipment designed for zero maintenance will be neither economically viable nor justifiable in the overwhelming majority of cases. Putting it another way: maintenance will be required, and we might as well try to identify and implement optimized maintenance methods.

Since this text does not purport to be a maintenance handbook, we will confine our scope of discussion to an overview of the two most prominent maintenance methods, TPM (total productive maintenance) and RCM (reliability-centered maintenance). While we do not take issue with either of these approaches, we would like to explain that neither of these highly touted methods represents a "magic bullet," or quick, inexpensive solution to a rather complex subject. It is simply our objective to deal openly with the many, often overlooked prerequisites to the successful implementation of modern management techniques.

What is TPM?

TPM (total productive maintenance) is a relatively recent addition to the plethora of hot management TLAs (three-letter acronyms) imported from Japan. It is closely tied to JIT (just-in-time) and TQM (total quality management), and is an extension of PM (preventive maintenance).

The basic idea of TPM is that equipment effectiveness must be maximized through elimination of all defects, downtime, slowdowns, and so on.[1]

Equipment Effectiveness. There are two objectives in improving equipment effectiveness:

- maximizing output
- minimizing input

Productivity defined as output/input is therefore increased.[2] Output is not simply product volume. Factors such as cost, delivery, and quality must be measured as well. Even safety, pollution, and attitude can be considered outputs to the production process, and must somehow be measured and controlled. The main inputs to production are labor, material, equipment, energy, facilities, and land. Cost of inputs must be monitored and reduced. It should be noted that although the cost of all inputs has risen in the past few decades, for many industries automation has decreased the labor, facilities, and land components at the expense of the remaining three.

Running equipment effectively under ideal operating conditions to achieve maximum productivity is the central concept of TPM. This means eliminating the losses and waste listed above. It is, therefore, necessary to establish a measurement system for these variables, and to track progress in eliminating them. The measures must be easy to record accurately; otherwise, people will not bother inputting data. There are other calculations that should be made. Equipment availability measures the percentage of time a machine is available for production. Operating efficiency incorporates the percent difference between ideal and actual machine cycle times, as well as the maintenance of a given machine output over the operating period. Overall equipment effectiveness is the product of the latter two percentages and the rate of good quality product (i.e., 100% minus defect rate). Other measures include MTBF (mean time between failures), the number of improvements suggested by employees, and per capita training expenditures. (See page 246 for more data.)

Autonomous Maintenance. In North American companies, labor unions and management have formed a clear separation between maintenance and production. This is the most obvious reason for the slow acceptance of TPM, or more specifically autonomous maintenance. However, some North American companies have overcome this stereotype and have benefited accordingly. Better communications, improved employee attitudes and placing responsibility for preventive maintenance and minor corrective maintenance at the front line, all lead to improved equipment effectiveness.

Autonomous maintenance requires not only a change in corporate culture, but a heavy investment in training. Operators who have always said, "That's not my job (problem), call maintenance," must now acquire a sense of ownership as well as the skills for implementing their new accountability. Operators are asked to keep the equipment clean, well-lubricated, and secure. Minor repairs and adjustments are also operator responsibility. Each operator is trained and empowered to inspect, measure, continuously diagnose, and fix problems.

The ability of operators to take fast corrective action is essential to improving equipment effectiveness. New standards must be set to incorporate the additional tasks, and targets established with supervisors for maximizing equipment uptime.

Team activities. The success of the TPM program depends on the formation of a formal organizational structure consisting of operations and maintenance teams that are superimposed on the existing structure. Groups must meet on a regular basis and on company time. Success or failure of the teams is linked to the motivation, skills, and training of team members. Likewise, management must create a working environment that fosters positive change. This can be accomplished through involvement in the program, providing physical surroundings conducive to work and meetings, and monetary support for the TPM program and the ideas it generates.

Training of team members must be both technical and behavioral. Technical training consist of, but is not limited to:

- Fastener and joint theory
- Bearing maintenance
- Basics of gears, belts, chains, and pulleys
- Sealing techniques
- Basic tribology
- Principles of electrical, pneumatic, and hydraulic systems

Safety is an essential component of all technical training.

Behavioral training includes motivation theory, group dynamics, brainstorming techniques, problem solving, and management of change. Behavorial training will ensure that technical training is exploited to its fullest.

To summarize, TPM concentrates on the elimination of waste. Waste exists as:

Downtime

1. Equipment breakdown
2. Set-ups and changeovers
3. Lack of material or manpower

Inefficiencies

1. Inadequate tools or parts to effect repair
2. Equipment design flaw
3. Substandard material input
4. Equipment deterioration
5. Poorly trained workers
6. Blockages
7. Pollution
8. Sporadic usage

Product Defects

1. Poor quality output
2. Decreased yield
3. Excessive scrap
4. Returns, rework, recycle, refeed

What Exactly is RCM?

RCM, or reliability-centered maintenance, is an outgrowth of the recognition that conventional, time-based maintenance may not be cost-effective. As the term suggests, it is an approach to the attainment of optimum equipment reliability by performing only known-to-be necessary maintenance. In an operations context, RCM encompasses the review of equipment functions, functional failures, and the consequences thereof. No predetermined maintenance actions are taken. Instead, each failure mode is subjected to a decision logic exercise to determine if a maintenance action is indeed required and what that action should specifically consist of. Generally speaking, these determinations would initially point to one of several categories of maintenance actions that fit under the collective term reliability-centered maintenance. These categories would include:

1. Scheduled discard/restoration maintenance
2. Fixed interval overhauls
3. Condition-based, or predictive maintenance
4. Servicing on an as-needed basis

As can be seen, this is a significant departure from the traditional approach of periodic overhaul maintenance. Failure modes, a detailed understanding of how equipment fails, and a thorough understanding of failure consequences become important in maintenance planning. Only those assets that show a clear age-related pattern are subjected to periodic, time-based maintenance. Equipment exhibiting evidence of random failure or likely to undergo progressive deterioration is subjected to *predictive* monitoring and allowed to stay in a service as long as it continues to meet the intended function and performance in a safe and economically viable fashion. These are important qualifiers to keep in mind, and we will have to come back to them later.

RCM: The Rest of the Story

An astute observer once commented that *RCM is about making the right choices.* This would include the right choice between replacing failed parts versus upgrading to improved parts, deciding whether these parts should come from OEM or non-OEM suppliers, making the right decision whether to engage in an often expensive failure mode and effect analysis (FMEA), picking the right hazard management technique, repair technique, installation method, oil replacement frequency, root-cause failure analysis technique, grout formulation, bearing-internal clearance—and literally hundreds of other issues that will affect equipment reliability and failure frequency. The success or failure of RCM is thus intimately linked to the experience background, training, motivation, and resourcefulness of an organization. Let's call this another qualifier or basic need; again, we will have to come back to this issue.

However, the success or failure of RCM is even more strongly influenced by management's perception of the daunting task at hand. All too often, managers have neither the patience nor the understanding to support and encourage the tedious learning, documentation, stewardship, and guidance-and-direction effort that must be expended to make RCM a success. And rare indeed is the advocate-purveyor of RCM technology training who will muster the courage to candidly discuss management misunderstandings, or what we earlier called qualifiers, while soliciting business. There have also been allegations that not every RCM trainer has sufficient practical experience to appreciate when, where, which, and why certain elements of aerospace-derived RCM are not applicable to, for example, a refinery. In those instances, the client could be saddled with cumbersome, procedure-driven exercises that add little, if any, value.

Essential Question to Be Asked

Let's suppose we condense the RCM approach into a few essential questions and then explore whether we have the talent, resources, and ability to answer each of them authoritatively and accurately. Being able to answer each question as we embark on RCM will be critically important to its success.

1. What are the functions and associated performance standards of the equipment? What is the life cycle cost of the machine or of its weakest, most failure-prone or critical component in a "best-of-class" plant?
   —If we don't know that a given pump in our plant fails four times as often as a comparable pump in identical service elsewhere, RCM may be premature. We should first concentrate our efforts on answering the simple questions, "What are *they* doing that *we aren't* doing" or "What are *we* doing that *they aren't* doing." What sense would it make for the owners of *our* pump to spend money ascertaining that the pump bearings typically last 14 months, and that they should therefore be scheduled for changeout every 13 months? Why not expend effort to ascertain that "others" have found it cost-justified to, let's assume, install superior magnetic bearing housing seals (see Figure 5-1), and that this upgrading prevents moisture contamination of the lube oil, the root cause of inadequate bearing performance at your site? You may find that the pump bearing life is now extended to 36 months or more. That, now, is a wise use of study time and monetary resources!
2. How might this asset fail to fulfill its intended function?
   —If we presently don't have the dedicated resources or experienced personnel, how and by whom will this question be answered? Are we prepared to invest in the time and training needed to answer this question? Should we engage a knowledgeable consultant who can provide the answer? How would it affect the morale of our organization if we hired this consultant? How would we ascertain that the consultant is, indeed, both qualified and knowledgeable?

3. What could cause each of a multitude of functional failures?
   —Since our systems are likely to include instrument, electrical, hydraulic and computer-electronic functions, do we really have the in-house talent and time to collect answers?
4. What exactly happens when each of these failures occurs? What are the short- and long-term consequences? What is the safety and environmental impact, downtime impact, community perception impact?
   —Can we afford to let our "lean" organization divert its attention away from the urgent day-to-day business? Will they find answers to some of the obviously subjective questions raised here? How much time can we afford to spend on getting these answers? Would a consultant know, or would he have to sit down in lengthy sessions with our own personnel to get at the answers?
5. What can be done to prevent each potential failure?
   —Are we prepared to arrange for the considerable interaction between project, operations, maintenance, technical, and managerial staff that will be necessary to deal with this question? The decisions, actions, and omissions of each of these different job functions will influence equipment reliability, failure risk, and plant profitability. Who chairs these meetings, and how much time are we prepared to allocate to the task?
6. What is the exact cost justification to implement a given failure prevention measure?
   —Do we have the talent to perform calculations with reasonable accuracy? Is someone already routinely engaged in this work, or will it be necessary to train a person?
7. What measurements can be made to predict and/or track failure development? How often should these measurements be made and who should make them? Who should interpret them?
   —Do we have personnel who have knowledge as to how often certain vibration measurements should optimally be performed on various machines? Do we know how, where, and when these measurements are typically being performed on equivalent equipment at "best-of-class" facilities around the world?

Prerequisites to RCM

By now the reader will perhaps appreciate why an estimated 60% of U.S. industry's *attempted* RCM implementations are being abandoned after a year or two, and why the affected plants revert back to their original or some other maintenance strategies. RCM is anything but a quick fix. If the basic problem is lack of training, then the implementation of proper training is an indispensable prerequisite to the pursuit of RCM. If, as in a typical refinery, most failures are related to operational or product-related upsets, it stands to reason that they will occur at essentially random intervals. Earlier in our discussion, we had made the statement that only equipment that exhibits wear-out failures will optimally respond to RCM, and that random failures are best detected by predictive, experience- and instrument-based data gathering and analysis techniques. It will thus make better economic sense to emphasize predictive and component upgrade approaches. These will have to be pursued in conjunction with cost-justified state-of-art work processes, sound and effective zero-defect driven operating and maintenance procedures, well thought-out checklists, and similarly appropriate material compiled and executed by a trained workforce.

A trained reliability engineering workforce is thus needed, and this workforce must obviously have a very thorough awareness of what constitutes *proven and readily available* upgrade components. This awareness comes from reading up-to-date technical books, from reviewing (and, occasionally, *reading*) a large number of trade and/or professional journals every month, from attending trade shows and exhibitions, from participation in vendor-sponsored courses and outside seminars, and so forth.

Emphasizing Reliability Instead of Maintenance: The Way to Increased Profits!

Some of the most knowledgeable world-scale petrochemical plants have realized in the early 1990s that moves toward "world-class maintenance"—read "RCM"—were costing them increasingly large amounts of money. They became convinced that rapidly bypassing the more time-consuming, low value-added maintenance routines in favor of tangible reliability enhancements made real economic sense. These plants had previously executed detailed shutdown plans, had trained their craftsmen in doing repairs safely and efficiently, were using the latest predictive and preventive maintenance techniques, and had implemented in-house facilities to test and evaluate instruments and electrical equipment. Leak detection teams were surveying all operating units on a rotating cycle. Thousands of oil samples were analyzed, and computers reminded operators and craftsmen when specific lubrication routines were due. Almost every piece of rotating machinery was monitored by someone, ranging from an operator with a portable monitor to a technician with the latest, most sophisticated data acquisition instrument. Vibration spectra were transmitted around the world electronically so as to have them analyzed by the most qualified expert in the company. Corrosion coupons and advanced corrosion monitoring instruments were used to evaluate materials of construction and protective coatings. At this world-renowned company, equipment maintenance records filled file cabinets and computer memory, allowing management to receive up-to-date maintenance measurements.[3]

But, while the company was considered profitable, maintenance productivity as indicated by the ratio-of-maintenance cost to original cost of assets was inferior to that of the best-of-class competition. More than one competing company had paid attention to Phillip Crosby's second dictum dealing with "The Absolutes of Quality Management." *The system for causing quality is prevention, not appraisal.*[4] Prevention is the infusion of reliability at every opportunity. Prevention is the "engineering-out" of maintenance requirements at the very inception of a project. The prevention of maintenance, or the elimination of the need to perform maintenance activities, starts with the cost estimating manual. This "price book" must reflect the purchase cost of equipment that requires little or no maintenance. Prevention of maintenance then moves through the compilation of a bidder's list from which nonconforming vendors are deleted, to the consistent adoption and implementation of a philosophy that views every future maintenance event as an opportunity to upgrade or to impart higher reliability to the equipment. When this world-class petrochemical company made it their goal to eliminate maintenance as it was then known, the benefits were drastic and immediate.

This company now recognized something to which we had alluded before: The reliability workforce members must be made up of well-motivated self-starters—men and women with inquisitive minds. They must be supported and valued by management, since no self-respecting reliability professional will be happy and productive in a stifling, business-as-usual work environment. In addition, this reliability workforce will only become optimally effective once the reliability statistics at the local plant site are being compiled and are routinely made available for comparison to representative or equivalent industry statistics. This is commonly known as benchmarking and implies that data collection is taken seriously by every job function in the plant.

Benchmarking: Comparing Yourself to Your Competition

If you wanted to know the extent to which it is reasonable to make improvements, you would compare your maintenance performance against that of the competition. Numerous different definitions and benchmarking routines exist and it is outside the scope of this segment to debate their merits and shortcomings. However, it should be intuitively evident that only a meaningful definition of availability will do; a randomly chosen availability statistic alone will not tell the full story. Say, for instance, an automobile that is being serviced twice a year will be unavailable for two days; hence, it has an availability of (365 minus 2)/(365), or 99.45%. Another car might suffer from an electronic glitch that randomly shuts it down once per day and for just one minute per event. It is thus unavailable for "only" 365 minutes each year. Since 360 minutes would be six hours, or one-fourth of a day, the availability claim of this car could be a seemingly attractive value of 364.75/365, which equals 99.93%. And yet, would not a reasonable person prefer to own the *less available car*?

We found the information given earlier in Chapter 4 (Reference 5) helpful; several tables in that chapter give strategic level measurements. We consider them rather self-explanatory, but wish to direct particular attention to the fact that in 1996, the certified training costs in best-of-class companies were $1,200, while the "average" plant spent a disappointing $400 per employee. Add to this that much of the so-called training does not necessarily impart real, useful, or implementable knowledge. And now draw your own conclusions as to the state of training and up-to-date technical competence at some plants that *talk* reliability but fall far short of pursuing the most cost-effective implementation steps.

Appendix A contains relevant statistics, some compiled over years of observation and data taking. Please note that some performance measurements are expressed as reliability, availability, duration of downtime, failures-per-million operating hours, mean time between failures (MTBF), or mean time to repair (MTTR). Knowing where you stand is without doubt an important prerequisite to either the "designing-out" of maintenance or to the adoption of RCM. Among other things, it will tell you whether your equipment is close to being life cycle cost optimized. Do you have these comparison data? If not, how will you get them?

Also in Appendix A, consider the maintenance cost breakdown by work order. Best-of-class companies use *predictive* maintenance in their efforts to determine when it's time to perform *preventive* maintenance. Remember that *the system for causing quality is prevention, not appraisal.* Where does your plant fit in on the cost breakdown table? Note, especially, how an increase in preventive maintenance effort will drive down costly emergency work and repairs.

Choosing Wisely in a Stressful Environment

Recall that by all accounts RCM does not respond to random failures. Aerospace and process industries represent two different worlds. Let us, nevertheless, suppose that we, just as Boeing, McDonnell-Douglas and the European Airbus Conglomerate, have installed equipment, subsystems, and components that are life cycle cost optimized. Assume further that, just as is the case in the aerospace field, voluminous statistical and experimental data attested to the fact that our assets will virtually always fail in one of the predictable wear-out modes, which is ideal for the application of RCM. Are we now able and willing to invest in the people and resources whose diligent efforts are *expensive initially,* but will pay out in the long run? Do we have the patience to defer seeing returns on this investment for some time, and meanwhile let the competitor brag about higher profits? How do we explain the whole thing to our shareholders? Are we committed to provide continuity to the RCM effort? Do we have an understanding of the limitations of contractor personnel, and can we separate fact from fiction when we hear marketing-driven, exuberant representations made by the technology consultant selling RCM training?

We have sometimes seen how, in their efforts to sell RCM technology to an ever wider spectrum of potential users, its marketers have plotted density functions, cumulative distributions and hazard functions that *may* apply to aircraft components for which pertinent statistics exist. One of our earlier books (Reference 6) describes the "monorail mistake," which involves going from one idea to another in an inevitable manner, ignoring all qualifying factors. Yet, there are many qualifying factors that have to be considered or overcome before valid parallels between the aerospace business and the process industries may be drawn. Indeed, two different worlds!

Specifically, for instance, there is often no valid failure cause or failure origin linkage between the aircraft fuel pump and the refinery's ethylene reflux pump. Where the former operates under highly consistent and predictable conditions, the latter may be subjected to operator error, excessive pipe strain, coupling friction, shaft misalignment, baseplate weakness, soft-foot conditions, lubricant contamination, low-flow recirculation due to over-sizing, and at least 40 additional factors that can be shown to influence life expectancy, reliability, and optimum service intervals of pumps in a typical process plant environment. The list could go on to steam turbines, mixers, extruders, compressors, and other equipment requiring some form of maintenance. Our premise is simply that, in spite of these influencing factors being discussed in the literature, not enough companies are engaged in remedying these *known* impediments to achievement of best-of-class maintenance performance. As this text shows, many of these impediments are relatively easy and inexpensive to rectify. It would thus be far more productive for process plants to start by systematically, patiently, and consistently implementing the many obvious and well-documented steps and procedures known to result in long-term cost savings.

Again, just to come back to our earlier example: Since it can be readily demonstrated that the overwhelming majority of centrifugal pump bearings fail prematurely due to the intrusion of atmospheric dust and moisture, it is economically attractive to "hermetically seal" the bearing housings with force-compensating (repulsion-type) magnetic face seals as shown in Chapter 5 (Reference 7). The long-term beneficial results and bottom-line cost-effectiveness of this upgrading method have been documented in several technical publications and trade journals. Neither failure mode and effect analysis (FMEA) nor any other time and resource-consuming investigative or analytical technique is needed to capitalize on this opportunity. And so we conclude that in this, and very many similar instances, looking to RCM as the "magic bullet" is both unnecessary and unproductive.

Indeed, then, RCM is about making the right choices, and these choices are made by trained, well-read, inquisitive, highly motivated and experienced self-starters who enjoy both full management support and management's respect. Every plant function must support efforts to eliminate the need to perform maintenance. The belief that these attributes and requirements can be taught in an aerospace-derived training course is plain wishful thinking. Yes, the *highly selective* application of RCM can be a money-making maintenance approach for the realistic, committed refinery or process plant manager. Likewise, RCM for RCM's sake will definitely be the wrong choice for someone who prefers "looking for the quick fix" instead of patiently and consistently implementing the many known and readily available reliability improvement techniques. And remember: That's what this book is really all about.

Maintenance Management Options*

In recent years we've witnesses a phenomenal growth of the worldwide petroleum, petrochemical, and chemical industries. Industry's appetite for more sophisticated chemicals and petroleum products continues to grow in spite of occasional short-term stagnation, and this increasing global demand for specialty products requires a corresponding increase in construction and plant expansion by petroleum and chemical companies. The investment required for new manufacturing facilities today has become staggering, and the daily financial penalty for having a plant "down" can, at times, be well in excess of $300,000.

Obviously, then, the plant maintenance managerial responsibility has grown dramatically. No longer can general management afford the luxury of a "knee-jerking" maintenance philosophy. Planned and tightly supervised maintenance is the only alternative to severe financial damage. In some cases, it can be the deciding factor between making or not making a profit. Further, there is an increasing awareness on the part of management that they can no longer determine the most economical type

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*Bolliger, U., and Wright, J. W. D., "Options for Planned Maintenance," *Hydrocarbon Processing*, January 1977. Adapted by permission.

of maintenance on the basis of maintenance costs alone; but now, must engage in an active dialogue with their colleagues on the production side to determine the "trade-off" between slightly increased maintenance cost and considerably reduced profit loss through plant outage.

The primary function of plant maintenance management is to keep a plant operating at maximum efficiency for a desired period of time. Here, we define "plant" as meaning boilers, furnaces, compressors, turbines, pumps, piping, and instrumentation, as well as buildings and grounds. While each of these components is important in its own way, items such as piping, valves, heat exchangers, and strainers are fairly easy to maintain. Nearly all journeymen craftsmen, whether staff or subcontract, can handle most normal maintenance. When it comes to more highly sophisticated machinery, however—machinery which by economic necessity has no standby, and which is in many instances the very heart of a plant—maintenance supervision must continually evaluate practical alternatives to ensure minimum downtime.

It seems that plant management, when considering the maintenance alternatives for key machinery in a single-stream plant, has three major options open to them. These three alternatives remain valid regardless of the size of the facility, in that units tend to be operated by sections, or "plants within plants."

The first and most obvious option is to try and handle the entire plant maintenance operation with captive, or "in-house," manpower capability. The second option is to employ full contract maintenance, and the third option is to employ the "peak shaving" maintenance concept.

Let's look a little more closely at each of the three, and evaluate the factors that might be considered when selecting the best alternative for your plant.

Captive, or In-House, Maintenance

Advantages

- Under most normal circumstances, you're dealing with craftsmen who are loyal employees, who respect the objectives of your maintenance program, who have a vested interest in the continued success of your company, and who can be trusted with any proprietary features of your process.
- They are aware of your company rules and procedures and recognize the threat of termination of their employment in cases of proven incompetence.
- Captive employees must live with the results of their work—they can't just walk away and expect someone else to "pick-up the pieces."
- Should an emergency arise, captive personnel are right there, ready to go to work.

Disadvantages

- What size crew is required to maintain your plant? While a full-time base-load staff is essential, manning in excess of the base-load workforce is demonstrably inefficient.
- This method of maintenance tends to encourage and justify overstaffing.

- It is more difficult to train and upgrade an existing maintenance force to keep up with current technology. Daily maintenance generally doesn't require in-depth inspection of, and familiarization with, sophisticated machinery. Planned shutdowns are scheduled for minimum duration and are usually not the time nor place to run a training course.
- Furthermore, even a trained captive staff will be short of in-depth experience if they seldom encounter problems on the same equipment at each shutdown. They will not have the confidence nor the ability that comes from years of successfully handling the unusual.

Full Contract Maintenance

Under the type of maintenance program, you are simply hiring an outside maintenance contractor to plan, supervise, and handle your entire maintenance operation.

Advantages

- The size of your maintenance force is constantly matched to a given workload: it is tailored to your needs, and you are only paying for what you need.
- It allows you to draw a very clear line between plant maintenance functions and those of your regular hourly employees. You no longer find yourself assigning "make-work" projects to keep your captive maintenance force busy: no longer might one group begin infringing on another group's territory, which results in duplications of work and cost.
- Outside contract personnel aren't on your payroll: they don't require you to make Social Security, hospitalization, tax, and pension fund payments. Furthermore, you can easily replace undesirable and unproductive personnel, retaining only the most productive people.
- Working with an outside contractor, you can generally expect and get higher productivity. The maintenance manager may also find that he can delegate tiresome detail and routine work to the outside contractor management and concentrate his efforts on the more important aspects of his job.

Disadvantages

- There is always the chance that after signing a long-term contract with an outside maintenance contractor, you find you are stuck with a vendor who isn't performing up to your standard.
- You may find also that you are being asked periodically to justify to your management inflated costs for services that had cost considerably less in the past.
- There is always the possibility that "outsiders" may irritate or cause friction with your regular employees.
- Most important, there is always the possibility that a crisis of one type or another will occur, and that your independent contractor will be unavailable—he won't be able to "jump on" your problem due to lack of people or his present logistical shortcomings.

Peak Shaving Maintenance

Peak shaving simply means that under normal periods of plant operation, the general maintenance load is handled by a small existing staff. However, during periods when you are performing turnarounds or rerates, correcting major breakdowns, or revamping your plant layout, the extraordinary work will be subcontracted to an independent service organization. Figure 10-1 illustrates this concept.

Advantages

- This method of operation allows you to hold your maintenance staff to the most cost-effective level at all times.
- You can maintain a planned and systematic work schedule, keeping overtime (resulting from turnarounds and emergencies) to a minimum. This will allow you to operate within the contractual guidelines agreed to by company management and unions.
- You are able to forecast and budget your annual maintenance costs with a high degree of accuracy.
- You can minimize the duration of your turnaround and the attendant loss of operating time, which will yield significant economic advantages.
- There is no need to keep equipment specialists employed full-time, and you can insist on, and get higher productivity from, suitably qualified outside maintenance personnel.

Disadvantages

- There is the tendency of your maintenance bargaining personnel to feel that a fully trained maintenance complement should be employed full-time.

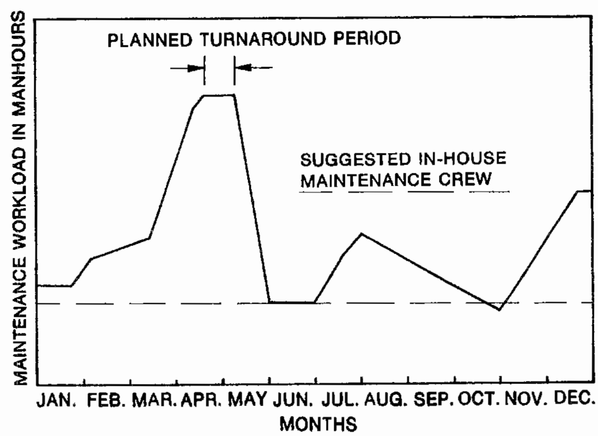

Figure 10-1. Peak shaving by outside contractor.

- Resentment may arise from having outsiders come in periodically and "take over," which may be seen as an implied criticism of your captive work force.
- Then there is the "love 'em and leave 'em" syndrome. In this case, after the subcontractor has completed his contract and left, the regular maintenance force may have to resolve possible problems that crop up later. This could cause resentment, and result in comments like: "Why didn't you let us do it in the first place?"

Many petrochemical companies in the industrialized world believe the peak shaving principle offers the most attractive long-term benefits to a maintenance manager:

1. It is a reasonable and practical compromise between the first and third alternatives—it combines the best points of the captive maintenance arrangement with the best points of the full maintenance contract.
2. It allows the maintenance manager to retain a skeleton crew to handle normal daily operations, yet leaves him free to employ high-priced specialists whose talents will be used on an infrequent basis.
3. It results in a reduced salary schedule, as well as fringe benefit economies (can amount to 30%–40% of employees' salary in some countries).
4. Contractor organizations are continually encountering and solving a multitude of complicated technical problems that the average maintenance man wouldn't normally see. They get constant practice in solving problems—under pressure.
5. Skill and experience developed by frequent problem solving enables a specialist contractor to complete an assignment and get a plant back on stream with the minimum amount of downtime. Obviously, this results in greatly increased overall savings.

If you are involved in planning and managing a maintenance program, there is one key additional factor to be taken into account when selecting an outside vendor organization.

Independent or OEM?

Over the past several years, the independent service contractor has come into his own. Today, the independent contractor vigorously competes with the service organizations of the original equipment manufacturer, or OEMs for short. Some positive points can be made for each.

The Independent Service Contractor

- The independent contractor often offers a maintenance manager a greater range of services and, at times, wage rates that are lower than the rates paid to the OEM. This can result in lower maintenance costs.
- Independent contractors tend to be more local-market oriented than today's OEM, who will generally operate on a global or regional geographic basis. Thus independents can, at times, provide faster service.

- Independent contractors tend to be more aggressive, and this may give the service supervisor the idea that the OEM isn't interested in "winning" his business.

Original Equipment Manufacturer Service Organization

- The OEM designed and built the original equipment; he respects his product and knows it inside out: in effect, the "pride of authorship" factor.
- The OEM has the comprehensive backup of his entire organization behind him to solve a problem—his engineering, testing, and manufacturing departments.
- Many of the OEM's service employees have spent time working for more than one manufacturer and have detailed knowledge of more than one manufacturers' equipment.
- Further, the OEM has a comprehensive grasp of the entire process, or system, within which his equipment was engineered to operate. This helps ensure an understanding and appreciation of all major system components, offering the added flexibility of being able to service or repair related equipment.
- As the process industries continue to expand around the world, OEMs have been reacting positively to the challenge of building and improving their global service organizations. It's no secret that "the sale of new apparatus tomorrow depends in great measure on the service you're giving today, for the equipment you sold yesterday."

While it is true that the peak shaving maintenance principle is often thought to be the most logical and practical system to use today in petroleum and petrochemical plants, there are some pitfalls to consider also. Very often, the OEM's "Peak Shaving Maintenance Contract Division" will man your turbomachinery turnaround with one or two qualified service technicians or supervisors; but the millwrights or machinists making up the bulk of the work force are frequently hired from the available local labor pool and may not have sufficient familiarity with the OEM's equipment to work to the equipment owner's fullest satisfaction.

True, the OEM may have capable personnel at his nearby satellite repair facility; however, these personnel can hardly be expected to become available to a petrochemical plant scheduled to perform turnaround maintenance on five or six major trains in the span of three or four weeks.

Modified Peak Shaving Maintenance Must be Considered. In view of the stated inadequacies of OEM-type peak shaving maintenance, a number of U.S. utilities and European process plants have found significant advantages in forming and maintaining teams comprised of a number of highly trained machinery technicians or mechanics at each of their separate affiliate locations. Thorough familiarity with large turbomachinery trains at their home location and continued exposure to machinery overhauls, advanced training, and occasional troubleshooting keep these teams proficient. They are high performers and take pride in their workmanship. Their work output is in the spotlight and a good job is rewarded accordingly.

The local team travels to an affiliated plant location across country, state, or national borders when the affiliate embarks on a major turnaround. The team leader—usually a working foreman—arrives at the affiliate plant location early enough to take an active part in turnaround planning. When the local plant schedules a turnaround, the roles are reversed and the affiliate teams travel to the local plant site.

This concept is graphically illustrated in Figure 10-2, which superimposes teams from affiliate plants B, C, D, and E on the resident turbomachinery turnaround team A. The efforts of these teams or crews are supported by solid documentation and detailed planning, and have resulted in highly satisfactory performance from time, cost, and reliability points of view.

Detailed Task Descriptions Improve Maintenance Effectiveness*

Renowned efficiency expert W. Edwards Deming noted that 85% of failures are the result of problems with the system, not the people. The percentages may be even higher for failures in a predictive and preventive maintenance program. Ineffective predictive and preventive maintenance can be attributed more to how the program is managed than to any lack of technology. With all the information tools and technologies available today there are still disconnects between the person performing a maintenance task and the supposed benefactor—the equipment. A multimillion dollar information system still requires a person to walk out to the motor and grease the inboard bearing. Who performs this task? How often? With what type of grease? How do we know it was done? How do we know that it has been effective? These are the simple questions that the people "responsible" for the work, surprisingly, cannot always answer. These answers require more attention to detail than to capital investment, but can have a significant impact on the effectiveness of a maintenance organization.

Job procedures are an extremely useful tool for outlining specific tasks and the order in which they should be performed. That level of detail is also useful for outlining entire maintenance programs, such as lubrication or vibration monitoring. It demands answers to questions such as: Who gets the report? Why? Where do we store this

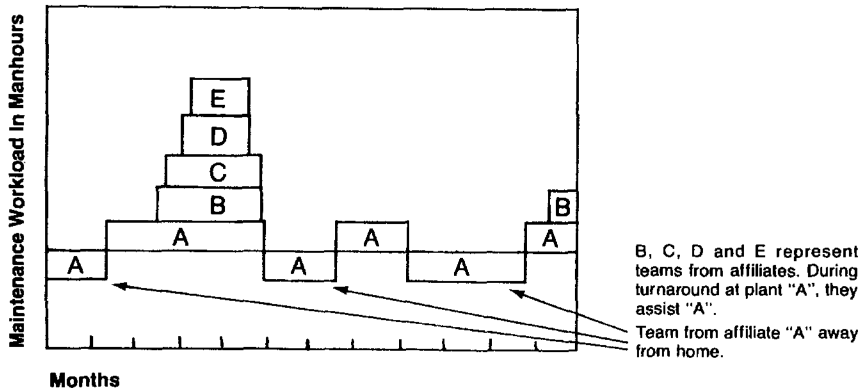

Figure 10-2. Peak shaving by affiliate turbomachinery turnaround teams.

information? When do we throw it away? and Who should be trained? Detailed task descriptions of maintenance processes do not always receive proper attention, but they are an important ingredient for an effective maintenance organization.

A proactive approach to maintenance should be common ground for anyone involved in plant reliability. For that reason, predictive, preventive, proactive, reliability-centered, total productive and all other maintenance philosophies that focus on prevention will be referred to throughout this section of our text as proactive maintenance or simply PM. This section describes a process for developing a PM program to a level of detail that is really dictated by the equipment itself. There are three aspects to developing the PM program: defining the plant; understanding, evaluating and selecting PM technologies; and documenting the PM program, which is where important details are captured. The next step is implementation, which is accompanied by measure and improve. The measure and improve step is an evolutionary process. It is never really complete.

Develop The PM Programs

Development of PM programs for a plant must begin by defining the plant. In other words, by defining the systems that exist within the plant and the components that make up the system, and then further defining the components or parts that make up the equipment. It is through selective application of PM technologies that preservation of component and thus system function can be ensured.

Define the Plant. In a hydrocarbon processing plant, one can typically find systems that provide plant air, electrical distribution, cooling water, boiler feedwater, steam, and nitrogen, just to name a few. This view of the plant might be called the "process sort" because the names of the systems describe the process or function that the system serves or supports in the production process. Every plant can be divided into its own unique series of systems.

Just as each plant can be broken up into systems, every system can be further divided into components. For example, a typical cooling water system might consist of the cooling tower, cooling tower basin, pumps, piping, and water treatment system. This view of the plant might be called the "component sort," because it describes the components that make up the system.

Advancing this concept one step further, each component in a system can be divided into still another set of components or parts. The cooling tower pump is an assembly of bearings, a rotor, the casing, the seals, a coupling, and the motor. If you are following the logic so far, you are probably asking yourself, "Okay, where does this end?" The answer to the question is, When you reach a point in the hierarchy where one of the PM technologies applies. For this cooling tower pump example, it is probably not necessary to go any further.

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*Contributed by Richard Ellis and Mark Galley, the Dow Chemical Company, Freeport, Texas. Presented at the 5th International Process Plant Reliability Conference, Houston, Texas, 1996.

If you are thinking in the larger context of all of the systems and components in your plant, you may ask how long it would take to break every component of every system in a plant down to this level. This question is valid to a certain extent. One thing to consider here is that you only have to do this one time, and the work you do can be leveraged across all the units in your facility. After all, a centrifugal pump, or cooling tower, or shell and tube heat exchanger in your plant has the same components in it as the plant next to yours. Moreover, all of the plants in your company would benefit from understanding equipment to the same level of detail without having to again go through the same development process.

Each level of the plant hierarchy, be it systems, components, or parts, has a specific and measurable impact on the production process. To ensure production, it is necessary to understand and document this impact. Defining a plant in terms of its systems, components and parts, and their function in the production process is a prerequisite to evaluating and selecting the PM technologies to be used in the preservation of function.

Understand, Evaluate, and Select PM Technologies. By understanding a plant's systems and components, their functions, and the required level of operating performance necessary to ensure production, it is possible to systematically evaluate and select PM technologies to maintain the required performance level. Part of PM program development is knowing and understanding what PM technologies are available. There are numerous books, training courses, magazines, and companies who specialize in PM technologies and it is outside the scope of our text to describe them all. Suffice it to say that one should familiarize oneself with the technologies and their application before using them.

Select PM Technologies. A non-spared process compressor warrants more attention in a PM program than does a storm water pump. In this and every other case, the criticality of the component in a plant is determined by the function it serves in the production process, and not by its complexity.

If the goal is production (and it is), then the selection of PM technologies must be driven by the goal to ensure production capability. One must resist the temptation to preserve the function of every component in every system simply because we can. Consider the following example.

A hydrocarbon processing plant knows that it is going to use oil sampling and testing as a PM technology, so it sets out to determine where to apply it. The plant can take either of two approaches. The first approach would be to apply oil sampling and testing to all oil-lubricated rotating equipment. This approach is certainly the easiest, but is also the least cost-effective. The second approach would be to apply the technology only to oil-lubricated rotating equipment that has a direct impact on plant production. The latter approach is obviously more effective.

The systematic approach is a combination of the earlier efforts to define a plant, its systems, components and their function in the production process. With the plant component list in one hand and the available PM technology list in the other, the selection process comes down to answering a few questions.

- Does failure of the component have a negative impact on the system (and thus production)?
- What does the component need, and how do we preserve it?
- What PM technology can we apply?

The answers to these questions, repeated again and again for every component in every system, provide the details for a PM program.

The list of questions here is not all-inclusive of the issues you may want to address in your plant. The list is provided for illustrative purposes. In the process of developing PM programs you may wish, or be required, to include questions such as: Does failure of the component have a negative impact on safety? Does failure of the component have a negative impact on the environment? The list you use is dependent on your goals and objectives.

Regardless of the amount of time and effort spent on the PM technology selection process, odds are that PM will be over-applied in some areas and under applied in others. And that's okay. The process of evaluating and selecting PM technologies is evolutionary, and there are other tools that will be addressed later in this section to help iron out the wrinkles.

Documenting the PM Program. Two types of PM program documentation are of interest. The first type, referred to as *equipment and PM task information,* is the detailed information that results from the work done earlier in defining the plant and selecting PM technologies. The second type, referred to as *work process documentation* addresses the need for documentation that defines the process by which PM tasks are accomplished. Both types of documentation are a necessary part of any PM program.

Equipment and PM Task Information. One-half of documenting a PM program involves writing down all information required by an individual to successfully accomplish a PM task. This information is specific to a piece of equipment and the PM task to be performed.

As an example, consider the simple PM task of lubricating a bearing. To successfully accomplish the task, the following information must be provided to the individual who will perform the work in the field: equipment number, what point is to be lubricated, what type of lubricant is to be used, what lubricant manufacturer is involved, how much lubricant is required, and how often the task is to be performed. Note that the information listed in this example is a combination of equipment and PM attributes. Equipment attributes include the tag number and the point that is to be lubricated. PM attributes make up the rest of the list.

Generally speaking, equipment and PM task information defines the *what, when, where,* and *frequency* of PM. It does not define who is going to do the task and why. It is a function of the PM technology one selects to preserve equipment function and usually resides in a computerized maintenance management system (CMMS). This information must be documented for each and every component in the plant.

Documenting the Work Process. The second-half of the documentation process involves writing down the process by which task-specific activities are to be accomplished. When it comes to a work process, there are two choices: the process by which work gets done can be ignored and one can hope that it will achieve the desired results, or, an effort can be made to understand and manage the work process to ensure achievement of the desired results. For PM to be effective, it must be managed.

Work process documentation should be written in a way that reflects the day-to-day work process. It should be considered a living document that is continuously revised and updated to reflect changes in the information it contains. In addition to defining *who* is going to complete the work and *why,* the work process documentation must contain everything required to ensure that the goals of a PM program are met. As a minimum, the work process documentation should clearly define:

- the scope of the work process
- the goal the work process is trying to achieve
- the expected benefits
- the work process itself, or how things get done
- the roles and responsibilities of the individuals involved in the work process
- the location of procedures, standards, compliance regulations, laws, etc.

The documentation is also valuable in that it

- establishes a benchmark for continuous improvement
- serves as a training tool for resident workforce members
- provides continuity within the maintenance function by capturing organizational knowledge and relaying this knowledge to contract personnel and newly hired workers.

A significant amount of understanding and eventual improvement will result from the documentation and examination of both the equipment and task-specific information, as well as the work process. An example of this can be seen by comparing two versions of the lubrication program from Unit A of the XYZ company. The first version, written in 1993, can be found in Table 10-1. The 1996 version can be found in Table 10-2.

Improvements to the lubrication program shown in Tables 10-2 and 10-3 resulted from continuous measurement of program effectiveness, and evaluation of the process by which work was accomplished. The differences between the documents reflect the ongoing learning of the organization.

It is important to keep in mind that "best practices" plants have documentation similar to the lubrication program illustrated here. Appropriate documentation is written for each PM program, including alignment, vibration monitoring, steam trap and utility leak surveys, cathodic protection, and crane, hoist, and elevator inspections, etc. The PM program template (Table 10-3) is generally used for guidance.

(*text continued on page 392*)

Table 10-1
Outdated Example of Work Process Documentation

Lubrication Program	
Purpose	The Lubrication Program has been established to ensure that all equipment receives and maintains the required levels of lubrication so that no equipment fails as a result of inadequate or improper lubrication.
Program Components	The Lubrication Program is comprised of the following: • regularly scheduled equipment lubrication (PM) • an oil sampling and testing program • lubrication training
Program Support	Lubrication Services will support the Unit A Lubrication Program as follows: • Serve as administrator of the Unit "A" Master Lubrication Schedule, incorporating revisions to the schedule as requested by the Reliability Specialist • Provide the Monthly Lubrication Schedule • Carry out the Oil Sampling and Testing Program • Assist in Failure Analysis Program as required • Recondition oil reservoirs as required • Recondition transformer oil as required • Perform biannual Lubrication Program health assessment
Regularly Scheduled Equipment Lubrication	Regularly scheduled equipment lubrication will be the responsibility of the Process Technicians. A Monthly Lubrication Schedule defining equipment requiring lubrication will be sent by Lubrication Services to the reliability Specialist who will coordinate with the Process Supervisor to schedule the work to be done. The technician(s) assigned to perform the lubrication shall initial and date the Monthly Lubrication Schedule as work is completed and return the completed schedule to the Reliability Specialist.
Standby Equipment	Experience indicates that false brinelling may occur in the bearings of standby equipment; a result of foundation vibration. To avoid this potential problem, standby equipment will be lubricated regardless of whether or not it has been operated. Prior to lubrication, equipment shall be "bump-started" using the jog-off-auto (JOA) switch.
Equipment History	Completed Monthly Lubrication Schedules will be filed in the records room located in building as part of the maintenance history for the plant. Retention period for completed Monthly Lubrication Schedules is current plus one-year.
Grease Lubrication	Where applicable, all grease lubricated equipment (specifically motors) shall be equipped with grease fitting(s) and grease relief(s). Prior to lubrication, it is the technician's responsibility to verify that equipment has been properly equipped. All deficiencies should be forwarded to the Reliability Department for correction prior to lubrication.

(table continued on next page)

Table 10-1 (Continued)

Training	Unit "A" lubrication training consists of the following six modules that cover the types and methods of lubrication used in the plant. • Introduction to lubrication • Constant level oilers • Bath-splash oil systems • Circulating oil systems • Grease guns and fittings • Storage and handling Training is accomplished through the use of self-paced training modules.
Lubrication Program Health Assessment	Biannual assessments will be conducted by Lubrication Services to evaluate the effectiveness of the Lubrication Program. The assessment will include a review of completed Monthly Lubrication Schedules and on-site inspections of lubricated equipment. Program deficiencies noted during the assessment will be prioritized and addressed through changes in procedures, additional training, etc., to continuously improve the program. The result of reviews, as it relates to the performance of the technicians, will be presented to the Process Supervisors as input for technicians in their respective areas.

Table 10-2
Up-To-Date Example of Work Process Documentation

Lubrication

Overview

Introduction	This document outlines the Plant XYZ lubrication discipline. Refer to the appropriate document listed under "Elements" for more detailed information about a particular element.
Goal	Ensure that all equipment receives and maintains the required levels of lubrication so that no equipment fails due to inadequate or improper lubrication.
Benefits	• Reduces friction (wear) • Removes heat • Protects from foreign material • Protects equipment against corrosion
Elements	The lubrication discipline, which will be coordinated with Lubrication Services, consists of the following elements: • Scheduled lubrication • Oil sampling and testing

Scheduled Lubrication

Introduction	Scheduled lubrication refers to equipment that receives routine lubrication
Frequency	Lubrication frequency is based on the speed (RPM), horsepower (HP), service, and type of bearing. The frequency of lubrication for Plant XYZ equipment is defined in the Computerized Maintenance Management System.

Table 10-2 (Continued)

Documentation	Lubrication history will be captured electronically and stored in the Computerized Maintenance Management System.
Document Retention	Life of equipment
Training	All Plant XYZ process technicians will be trained on the following: • Introduction to lubrication • Constant level oilers • Bath-splash oil systems • Circulating oil systems • Basic operator training and on the job • Grease guns and fittings • Storage and handling
Grease Lubrication	All great-lubricated equipment shall be equipped with grease fittings and grease reliefs. The process technician should verify that each lubricated piece of equipment has been properly equipped. Work orders should be entered for discrepancies.
Scheduled Lubrication Process	Below is the process for performing scheduled lubrication in the Plant XYZ facilities.

Step	Action
1	Lubrication scheduled service reports will autogenerate and print from the Computerized Maintenance Management System.
2	The Reliability Specialist picks up the schedule service report(s) and schedules the work for completion by the due date.
3	Each task on the lubrication schedule is initialed and dated by the person who completes the work. The completed schedule is returned to the Reliability Specialist who closes the primary PPM work order and forwards the scheduled service report to the PM Technician.

Work Orders	Work order numbers for scheduled lubrication are autogenerated by the Computerized Maintenance Management System.
Scheduled Service Reports	The following schedule service reports apply to the scheduled lubrication discipline.

	Report Name	Description
Unit "A"	UAFINFANGREA	Unit "A" - Fin-fan Lubrication
	UAMTRGREASE	Unit "A" Electric Motor Grease
	UAMTRGREASLR	Load Rack Electric Motor Grease
	UAOILCHGPMP1	Unit "A" Oil Change for Pumps
	UAOILCHGPMP2	Load Rack Oil Change for Pumps
	UAAGITOILCHG	Unit "A" Agitator Oil Change
	UABLOWOILCHG	Unit "A" Blower Oil Change
	UAGROILCHG	Unit "A" Gear Reducer Oil Change
	UAMTROILCHG	Unit "A" Motor Oil Change
	UASEALPOT	Unit "A" Seal Pot Oil Change
	UASEALPOTLR	Load Rack Seal Pot Oil Change

(*table continued on next page*)

Table 10-2 (Continued)

Responsibilities	This section defines the responsibilities of the individuals associated with the lubrication discipline.

Person	Responsibilities
Process Technicians	1. Perform scheduled lubrication for equipment in their area as assigned by Reliability Specialist. 2. Return initialed and dated schedule service reports to the Reliability Specialist.
PM Technician	1. Maintain up-to-date status of schedule services, schedule service reports and equipment specifications in the Computerized Maintenance Management System. 2. Owner of the Plant XYZ Lubrication Discipline. 3. Assist in training process technicians in proper lubrication methods.
Lubrication Services	1. Participate in lubrication discipline reviews as requested.

Lubrication Services	Questions concerning the lubrication discipline that cannot be answered by Plant XYZ personnel can be addressed to one of the Lubrication Services contacts shown below.

Calling Order	Person	Phone	Pager
Primary Contact			
1st Backup			

Plant XYZ Contacts	The Plant XYZ contacts for the lubrication discipline are listed below.

Calling Order	Person	Phone
Primary Contact		
1st Backup		
2nd Backup		

Oil Sampling and Testing

Introduction	Oil sampling and testing refers to the scheduled collection and laboratory analysis of oil samples taken from specific equipment items.
Goal	Proactively monitor the integrity of lubricating oils in order to improve equipment reliability and prevent unscheduled outages.
Benefits	• Improves equipment reliability • Allows for early detection of equipment problems • Helps schedule outage repairs by providing insight of where to look for problems and what to look for • Detects lubricant breakdown or contamination
Method(s)	Oil samples are taken manually by Lubrication Service technicians.
Frequency	The sample frequency for Plant XYZ equipment is identified in the Computerized Maintenance Management System.
Documentation	Oil sampling history will be captured electronically and stored in the Computerized Maintenance Management System.
Document Retention	Life of equipment.

Table 10-2 (Continued)

Oil Sampling and Testing Process	Below is the process for performing oil sampling and testing in the Plant XYZ facilities.

Step	Action
1	Oil sampling and testing scheduled service reports will autogenerate to Lubrication Services work order backlog.
2	Lubrication Services will collect and analyze oil samples according to the Computerized Maintenance Management System scheduled service report(s).
3	Oil sample test results will be handled as follows:

IF . . .	THEN . . .
The test results are acceptable	Go to step 4
The test results are not acceptable	Lubrication Services will submit a work request the plant.

Step	Action
4	Lubrication Services closes PPM work order.

Work Orders Work order numbers for oil sampling and testing are autogenerated by the Computerized Maintenance Management System.

Scheduled Service Reports The following scheduled service reports apply to the oil sampling and testing discipline.

Report Name	Description
UAOILANALYS	Unit "A" Oil Sampling and Testing Schedule

Responsibilities This section defines the responsibilities of the individuals associated with the oil sampling and testing discipline.

Person	Responsibilities
Lubrication Services	1. Collect and analyze oil samples according to scheduled service reports. 2. Close the Computerized Maintenance Management System PPM work order, recording history and comments as appropriate.
PM Technician	1. Serve as the primary Plant XYZ contact. 2. Maintain up-to-date status of schedule services, schedule service reports and equipment specifications in the Computerized Maintenance Management System. 3. Coordinate action(s) required as a result of oil sample analysis.

Lubrication Service Contacts Questions concerning the oil sampling and testing discipline that cannot be answered by Plant XYZ can be addressed to one of the following Lubrication Service contacts.

Calling Order	Person	Phone	Pager
Primary Contact			
1st Backup			
2nd Backup			

Plant XYZ Contacts The Plant XYZ contacts for the oil sampling and testing discipline are listed below.

Calling Order	Person	Phone	Pager
Primary Contact			
1st Backup			
2nd Backup			

Table 10-3
PM Program Template

PM Program Name

Overview

Introduction	Define what this document is about.
Goal(s)	State the expected goals and objectives. Use this opportunity to analyze how the goals you are setting align with production or company goals. Is what you are doing adding value?
Benefit(s)	State the desired benefits you expect the program to provide.
Elements	If the PM program has more than one element, use this section to list them. For example: • the lubrication program may be made up of scheduled lubrication and an oil sampling and testing program • the vibration program may be made up of permanently mounted vibration monitoring instruments as well as routine readings that are taken by hand If the program you are defining does not contain more than one element, an Overview section is not necessary.
Introduction	Define what this document is about.
Goal(s)	If this is one element of a larger PM program, list any goals that may be unique to the element. Although it may be redundant, consider restating goals from the overview if they apply. The reason for this is that sometimes documentation for individual elements gets used independent of the overview, and meaning may be lost if taken out of context.
Benefit(s)	Refer to the comments in the goals section above.
Definitions	Provide definitions of terms used within the program that may not be familiar to everyone.
Methods	State inspection, testing, or sampling methods to be used in the program.
Reviews	State the purpose and frequency of the program reviews if there will be any, and who will conduct the review. The review process should really be an ongoing process. However, it is sometimes valuable to have someone unfamiliar with the process review and ask questions.
Training	Define any training that will be provided as a part of the PM program, or what training is necessary to ensure program effectiveness.
Engineering Standards	List any engineering standards that might apply to the program. This facilitates quick reference.
Safety References & Standards	List applicable safety standards and/or references. This too facilitates quick reference and can be used to identify what safety standards or references were used or considered in the development of the program.
Environmental Regulations	List applicable environmental regulations that may pertain to the PM program. This facilitates quick reference and can be used to identify what regulations were used or considered in the development of the program.
Procedures	List applicable plant procedures associated with the program.
Documentation	Define the media (paper, electronic, Post-It® Notes, etc.) that will be used to document completed PM tasks, and where the documentation will be stored.

Table 10-3 (Continued)

Document Retention	Define how long documentation related to PM tasks are to be kept.
Scheduled *Program Name* Process	As a minimum, outline the steps that occur in the work process of the program.
Work Orders	Define how work orders will be generated, or how the costs for the PM program are to be tracked.
Scheduled Service Reports	This section is optional and it depends on the process used to define task specific PM activities. This type of list is useful in developing budgets for your PM programs. If you know the number of items per report, the number of times the report generates per year, and include a known or estimated cost for each activity on the report, it is easy to develop a budget. This list is also useful because it provides a visual representation of scope of PM work carried on in the plant.
Responsibilities	List the name(s) or job title(s) and specific, measurable responsibilities for each individual associated with the PM program in the table below.
Name of additional contact(s) if available	If you have access to experts, either internal or external to your company, that can answer questions about issues related to the PM program when no one from the plant is available, list their names below. The intent here is to empower plant personnel to initiate trouble calls to people when plant personnel cannot be contacted.
***Plant Name* Contacts**	List the names of people within your plant that should be contacted to answer questions or help resolve problems.

Scheduled *Program Name* Process:

Step	Action
1	

Scheduled Service Reports:

Report Name	Description

Responsibilities:

Person	Responsibilities
Persons name or job title	List the responsibilities for each person or job title associated with the PM program. The compilation of responsibilities for each person or job title from each PM program into one document results in the creation of a job description.

Name of additional contact(s) if available:

Calling Order	Person	Phone	Pager
Primary Contact			
1st Backup			

Plant Name Contacts:

Calling Order	Person	Phone	Pager
Primary Contact			
1st Backup			
2nd Backup			

(*text continued from page 384*)

Implement the PM Programs

Implementing the PM programs means turning the task-specific information and work process documentation into action. Success here is a function of the degree to which you communicate and how others understand the desired goals and their individual roles in the process. The implementation process is an ongoing process, because PM programs are constantly evolving. Evolution is the result of measuring and improving the tasks that are done and the processes by which they are accomplished.

Measure and Improve. Measure and improve should be considered another element of effective PM. It is not a specific technology, but it is just as important as any other element in ensuring production capability. Implementing a PM program is not the final step in the PM process. As soon as a program is developed, it must become the starting point for regular improvements. An effective PM program continues to evolve both from a technology standpoint, with better tools becoming available, and from the standpoint of the data that is collected to evaluate program effectiveness. The opportunity here is that measurement is not necessarily a difficult task. The most important issue is to develop a plan to collect data and to continue to improve the data-collection process. Measuring the effectiveness of PM through collection and analysis of data is typically overlooked or ignored. It is another aspect of managing the program.

Those most closely involved with PM know to what degree the work is performed, but its effectiveness or the need of performing a particular PM task often go unmeasured. Many times it's difficult to measure the improvements because they have not been defined within the proper context. It is important to know the significance of a particular PM program. It answers the question: Why do we even do this—why is it important? The purpose of a production facility is *production*—safe and environmentally sound. Each PM task should contribute to the production capability of a plant. It should add value to the raw material passing through the process by ensuring that a particular component remains functional. This in turn keeps the system functional, which keeps the plant producing. Another aspect of measurement is the cost of PM in dollars. This can be weighed directly with the risk, the probability and consequences of failure, associated with a particular system or component.

The maintenance spending of an organization can be improved by separating it into at least two specific areas and carefully measuring each: proactive and reactive (breakdown). The spending for each area should be trended. The cost of PM, which most likely will grow as a result of this detailed operating discipline, should be compared to the reactive cost which should diminish. The effectiveness of an organization's implementation and management of the work process can be determined by measuring the reduction in reactive spending.

All failure modes causing reactive maintenance should be investigated to determine if they were economically preventable. Failure analysis is a powerful tool in a plant's efforts to improve its maintenance effectiveness. The results of failure analysis: what failed, why did it fail, what could be done to prevent it from occurring, the maintenance cost and the production impact should be reviewed in weekly or twice-monthly meetings. Reviews should be attended by those directly involved in the work process. The discussion should address ideas on how PM programs or the work processes could be improved.

Often maintenance as a percent of replacement asset base (RAB) is used as a benchmark maintenance measurement. It is sometimes very insightful to look at maintenance as a percent of RAB for just rotating equipment or instruments. Even calculating maintenance as a percent of RAB by manufacturer or process service can shed light on the subject of high-cost maintenance items. There are many interesting information sorts that can be "thought up" or realized by those responsible for the program once information scope and accountability have been assigned to them.

Of all the different maintenance philosophies in industry, the bottom line is "whatever is effective is effective." Just as there is no single financial measurement that an individual or a major industrial company can rely on to monitor and ensure its financial health, there is no single maintenance task or work process that a maintenance organization can rely on to monitor and ensure its effectiveness. This means that there is no "right answer" to PM. There is no manual that clearly outlines what is needed. An organization's PM program and its work processes are a function of its goals and objectives.

This overview represents what has proved effective in helping two maintenance organizations fulfill their role in plant production, that of ensuring production capability. The whole is the sum of the detailed parts. A detailed understanding of the plant, its systems, its components and parts, its equipment and task-specific PM information, its work processes, and its measure-and-improve process, all are necessary to make maintenance effective.

Some lessons that have been learned and the key points that have proved to be effective are summarized below.

- The role of maintenance, that of ensuring production capability, should be a key driver in maintenance related decisions.
- In many cases, ineffective maintenance is more the result of a failure to manage the process by which work gets done than lack of available technology. For PM, or any maintenance work process to be effective, it must be managed.
- Unfortunate as it may be, the devil is in the details. Failures, or loss of function, that do occur within a manufacturing or production unit always seem to be a detail that was overlooked or never known.
- Companies have the resources, the people, within their gates to solve most of their problems. The key is to become more effective at utilizing them. Who else is more familiar with the problems at hand than those left with the task of fixing them?
- Documenting the work process using the template found in Table 10-3 has proved invaluable both as a training and communication resource and a benchmarking tool.

And so, it would be appropriate and beneficial to ponder:

- Does our plant have defined work processes, or does work just somehow get done?
- Are we supplying the people who do the work in the field the information they need to do their job?
- Is what we are doing today going to change the way we do business tomorrow?

Machinery Turnaround Planning

Figure 10-3 shows the familiar bar chart used for typical turnaround planning. But bar charts of the type shown can be no more accurate than the experience level of their developer allows them to be. Unless the developer is thoroughly familiar with turbomachinery overhauls and remembers task sequences as well as durations, turnaround plans will probably not be too accurate. The job may take longer than anticipated and could delay re-starting of the plant. Conversely, the job could be overmanned to the point where it is done faster than necessary. Here, the plant could lose out by thus drawing skilled personnel away from other jobs. Work execution may suffer from lack of continuity or the job may risk being executed in less than diligent fashion.

To avoid the pitfalls of inaccurate planning, process plants would be well advised to use a "Mechanical Procedures Manual," which could comprise such detailed out-

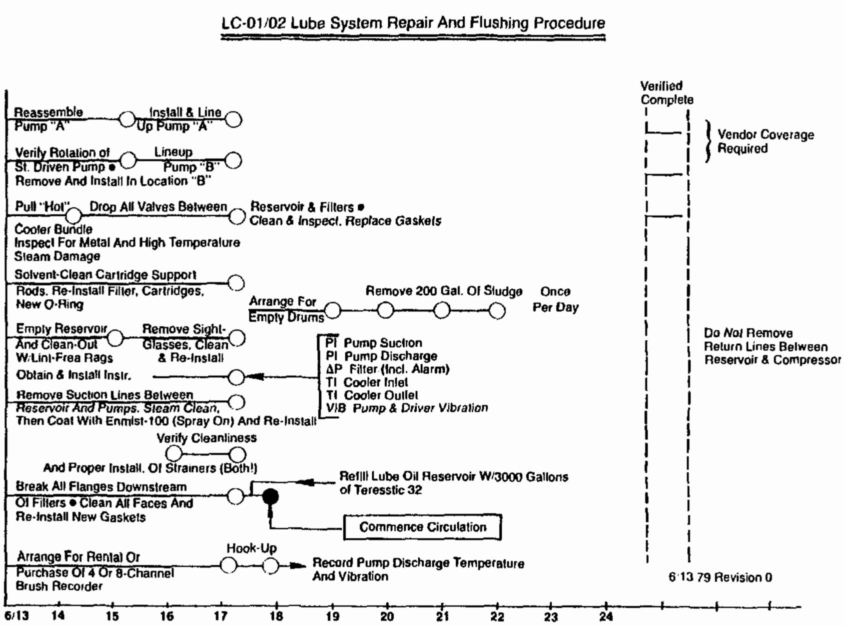

Figure 10-3. Detailed bar chart for turnaround task.

38 M Elliott Compressor
Re-sealing of Casing Joint

Contractors:

A: Beta Maintenance And Machining Co.
Manpower (1) 1 Supervisor, 5 Millwrights, 2 Days.
(2) 1 Supervisor, 3 Millwrights, 6 Days.
(3) Total Manhours = 282

B: McRae Field Machining Services, Inc.
Manpower (1) 2 Field Service Technicians
(2) Total Manhours = 16

Remarks:

1. No unusual problems encountered. All times representative of similar machinery.
2. On future jobs, allow contingency of 24 hours for possible problems with overhead cranes.

Date:__________ Responsible Supervisor:__________

Figure 10-4. Cover sheet for illustrated machinery turnaround book.

lines as indicated in the typical table of contents shown in Table 10-4. Sample sheets from one such procedure, "38M Elliott Compressor—Re-sealing of Casing Joint" (See Figure 10-4) are represented in Figures 10-5 through 10-10. A similar procedure, "Mitsubishi Steam Turbine Reassembly" was included in Chapter 1, Figure 1-31. Note that on the steam turbine reassembly sheet we had indicated a margin column with the heading "Hours." It contains two numbers, one representing the time it takes to execute the task shown in the picture and narrative, the other representing the total man-hours into the job. Turbomachinery turnaround planning accuracy is obviously more precise when pictorial records and elapsed-time tabulations are available to the planning staff.

Further, accurate turnaround planning is enhanced by separating the plant, area, or process unit into "work zones." This planning procedure requires that the planning staff avail themselves of process flow schematics (piping and instrument diagrams), equipment listings, and instrument listings. The work-zone turnaround planning concept is illustrated in Figure 10-11. In this example, a large condensing turbine driving a centrifugal compressor requires overhaul. Recognizing that the top half of the turbine and the rotor will have to be removed and transported to some other location for cleaning, the planner would make the upper half of the turbine a work zone separate from the rest of the machine. Similarly, he may recognize that surface condenser and condensate pump repairs could be executed by another crew; accordingly, these items would comprise another work zone.

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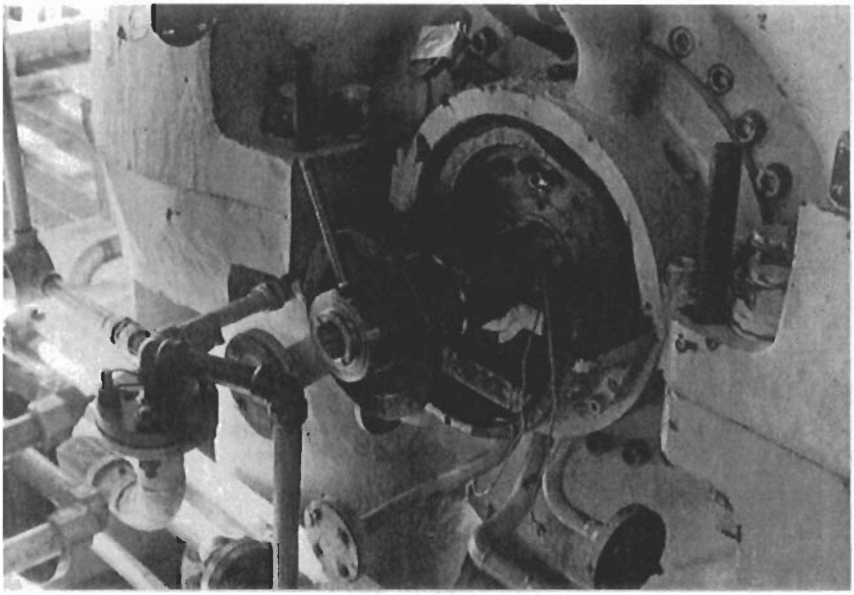

Figure 10-5. Installing hydraulic equipment to remove the compressor coupling half. (1 hour into the job.)

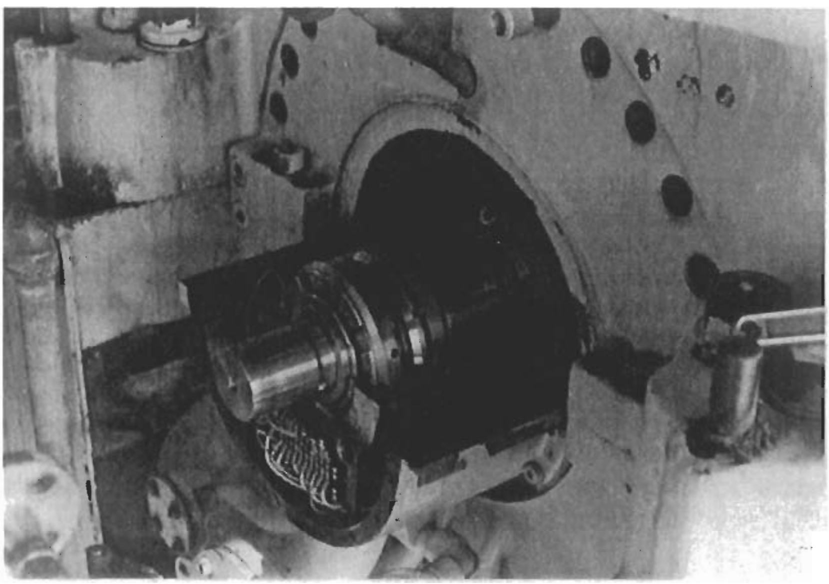

Figure 10-6. Kingsbury-type thrust bearing, OB end. (2 hours into the job.)

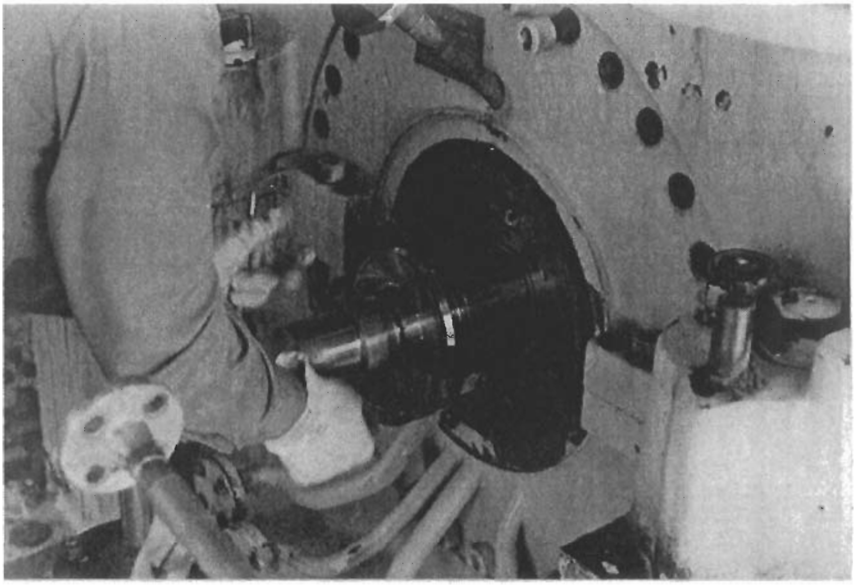

Figure 10-7. Removing the thrust disc of the Kingsbury assembly, OB end (2½ hours into the job).

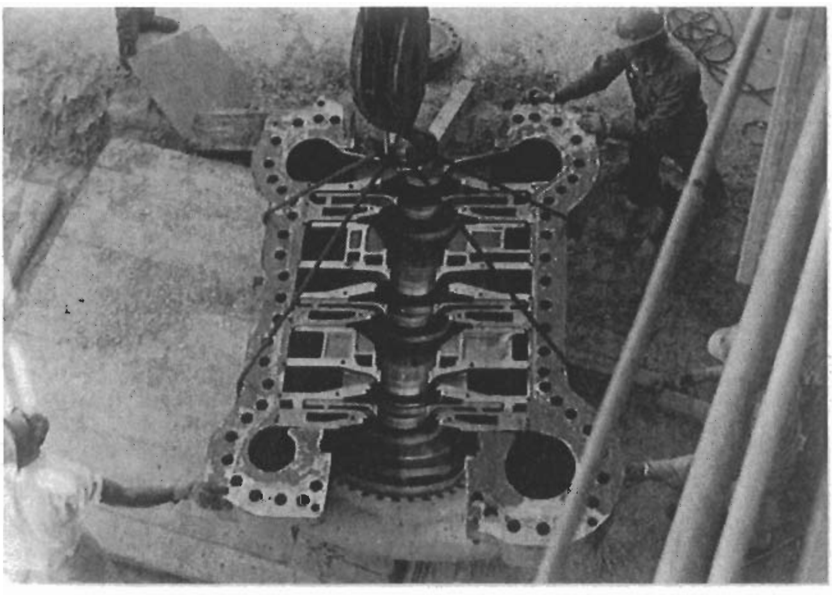

Figure 10-8. Compressor top half rolled over so it could be cleaned (7 hours into the job).

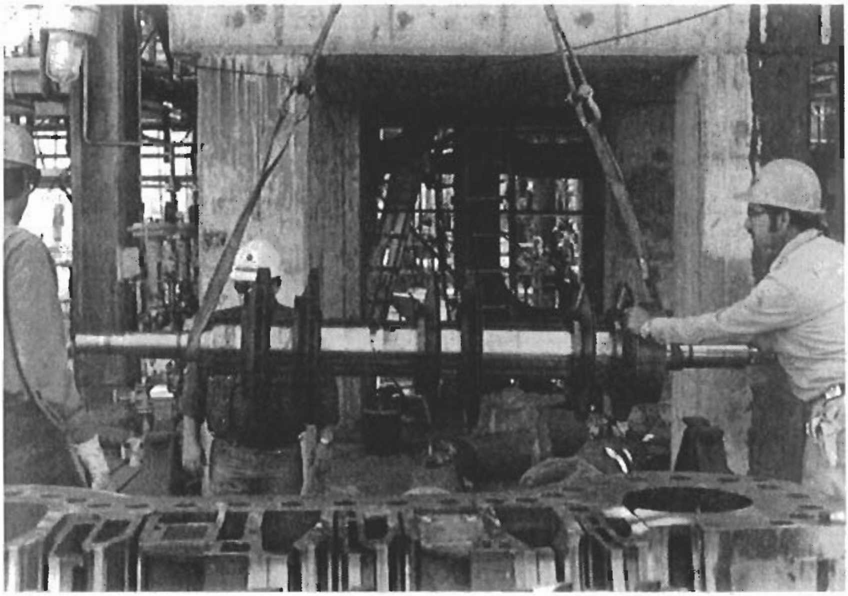

Figure 10-9. Compressor rotor being brought to grade (9 hours into the job).

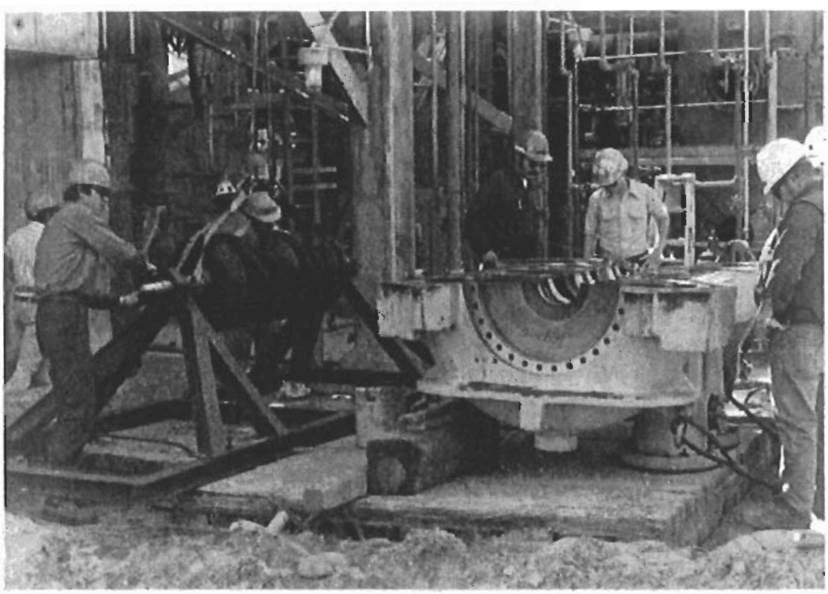

Figure 10-10. Inspection and cleaning of rotor and top half of case (10–16 hours into the job).

Table 10-4
Partial Table of Contents, Mechanical Procedures Manual

Centrifugal pump pipe-up criteria
Calibration of dial indicator brackets
Cleaning and flushing of lube systems
Electric motor run-in
Eliminating cooling water from process pumps
Critical pump and driver lubrication follow-up
Procedure for shutting down problem pumps
Procedure for reinstallation of problem pumps
Housing fits for antifriction bearings
Shaft fits for antifriction bearings
Routine modifications for centrifugal pumps
Allowable rotor unbalance for pumps and turbines
Conversion from double-row to duplex angular contact bearings
Centrifugal pump field removal
Centrifugal pump field installation
Rotating equipment repair form
Determining cleanliness of lube systems
Steam turbine latching procedures
Replacement of pump stuffing box packings
Sentinel valves on small steam turbines
Lubrication procedures and listings, plant-wide
Oil-mist lubrication for electric motors
Lubrication procedures for diaphragm pumps
Oil-mist lube conversions for ANSI pumps
Frequency of periodic in-plant machinery testing
Criteria for pump run-in on water
Rotating equipment correction documentation
Completeness review for rotating machinery
Grouting of pump bases
Insulation procedure for cold-service pumps
Installation of mechanical seals

(text continued from page 395)

Manpower assignments, job preplanning, and general scheduling are thus greatly facilitated. Again, we wish to make the point that, for best results, the preparation of bar charts should be preceded by a thorough review of illustrated turbomachinery turnaround procedures and by separation of the entire task into "work zones."

Coming full circle, remember that preparation for your *first* turnaround should begin when you specify the machinery and invite potential vendors to submit bids. Potential vendors should be under contractual obligation to furnish illustrated turnaround procedures and critical dimension diagrams as shown earlier in Figures 1-30 and 1-31. During the actual turnaround, machinist supervisors should measure the record critical dimension in the appropriate columns of Figure 1-30.

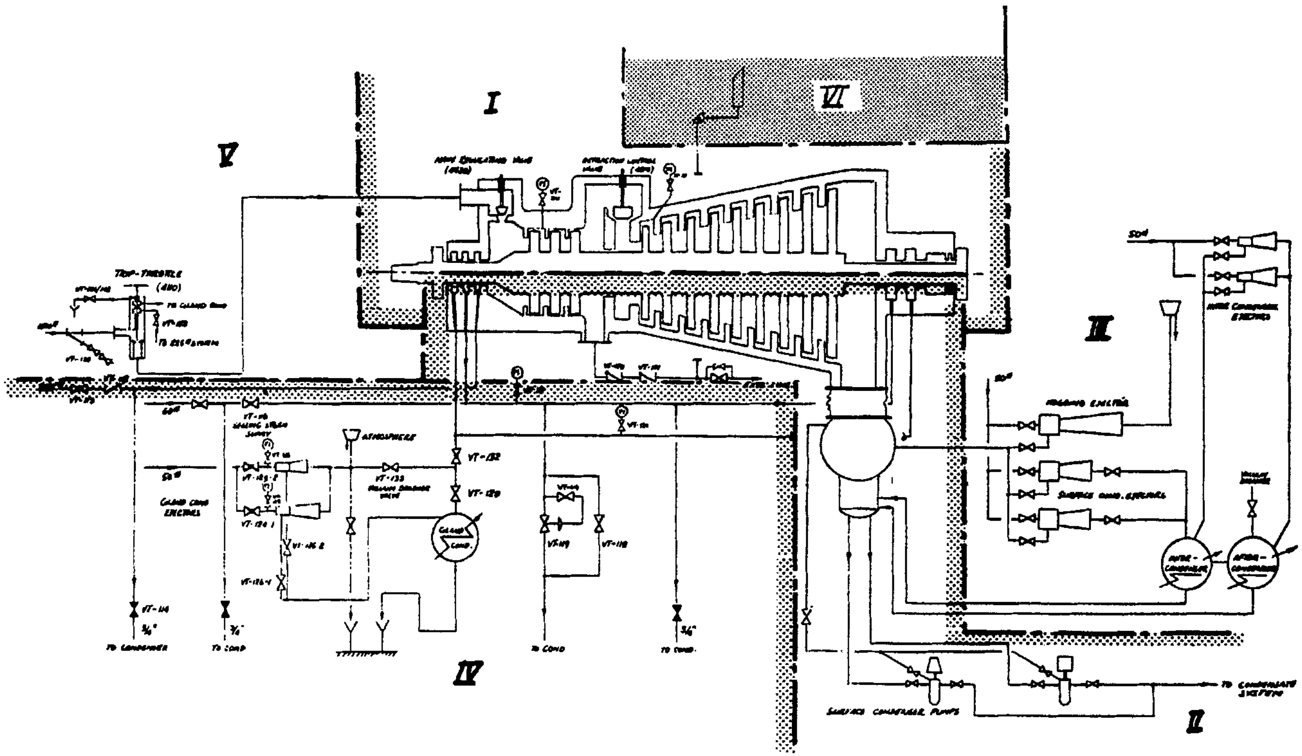

Figure 10-11. Illustration of work-zone turnaround planning.

Turnaround Scope Development through Reliability, Availability, and Maintainability Analysis*

The challenge for refineries and chemical plants is to optimize the turnaround scope and its execution so that turnaround costs, duration, and frequency are reduced while plant production availability is improved. Cost, duration, execution strategy, etc., all are dependent upon and usually driven by the scope. Therefore, it is very critical to develop a comprehensive turnaround scope that is supported by economics, by reliable operational and maintenance data, and by input from all personnel concerned.

An effective method for developing a turnaround scope uses reliability, availability, and maintainability (RAM)[8] principles. It is a team approach that uses the expertise and experience of key personnel to analyze plant operational and maintenance data and to economically justify every scope item.

Previous Practice. In the past, the turnaround scopes at a typical refinery were developed by collecting the concerns and requirements from various groups, such as operations, inspection, machinery, etc. These concerns were then combined and became the basis for a turnaround. The scope was finalized and funds were made available to execute the turnaround. There was no detailed analysis of the defined items to justify them. These turnarounds were extensive and expensive.

Recently, however, because of financial constraints on a refinery's cash flow, many facilities can no longer support these large and expensive turnarounds. Therefore, it will become even more critical to define the minimum scope required for a turnaround that will provide the desired RAM of the units until the next scheduled turnaround. This goal can only be achieved by carrying out a thorough assessment of the units and by economically justifying every scope item. To achieve this objective, consider developing a turnaround scope definition process and assembling a small team comprised of process, technical, and maintenance functions to thoroughly assess the plant equipment based upon RAM principles. The team should carry out a cost-benefit analysis to justify and prioritize each item for the turnaround scope. The scope form, Figure 10-12, facilitates the process.

Developing Preturnaround Requirements. About a year prior to the scheduled turnaround date, initial discussions for preturnaround requirements should begin. A flow chart of the typical preturnaround activities must be developed (Figure 10-13). Estimated time for each major activity is also given in the flow chart. As shown, it takes about a year from defining the preliminary turnaround objectives to final review of the detailed plans. Time estimates are based on the idea that the turnaround activities are part of the day-to-day responsibilities of the people involved. This duration, if required, can be reduced by allocating proper resources to complete

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*Based on a paper by S. K. Gupta and J. E. Paisie, originally presented at the *Hydrocarbon Processing*/Gulf Publishing Co. 5th International Conference on Process Plant Reliability, Houston, Texas, Oct. 2–4, 1996.

(text continued on page 404)

SUN COMPANY, Inc. (R&M) Rev. 7 Aug./95

T/A SCOPE FORM PHASE - 1 (SCOPEFRM.WK4)

Guiding Principle for T/A Scope:

Scope shall not include any work which can be accomplished during normal operation. exceptions shall be approved by the Refinery Manager or his designee.

Date:	Prepared By:	T/A Date:	Refinery:
Plant #:	Sub System:		Equipment #:
Equipment Name:		Service:	
Design Press.:	Design Temp.:	Last Inspected:	

RECOMMENDED SCOPE FOR T/A:

DESIGN CHANGE REQUIRED: ☐ YES ☐ NO

JUSTIFICATION:

COMMON: (Check One or More)		Explanation
☐	Legal Inspection Req'd	
☐	Other Inspection (Rerip/wet Mag) Req'd	
☐	Leaking/Repair Req'd (Explain)	
☐	Poor Process Performance (Explain)	
☐	Future Production Requirements (Explain)	
☐	Technical Recommendation	
☐	VOC Repair	
☐	Reliability/Preventative Maintenance	
☐	Capital Project/Equip. Modification	
☐	Other (Explain)	

APPROVAL:

Operations:	Technical:	Maintenance:
Date:	Date:	Date:

OPERATIONS/TECHNICAL/MAINTENANCE HAS VERIFIED T/A REQ'D: ☐ YES ☐ NO

Estimated Cost:	Estimated Man-hours:

APPROVED BY MANAGEMENT FOR T/A:	Date:
Planned By:	Date:

EQUIPMENT HISTORY:

Component	Year Installed	# of Maint./Proc. Incidents Since Last T/A	Mean Time to Repair	Explanation
		Total	Avg.	

PRESENT/EXPECTED CONDITION AT T/A:

Component	Mechanical Condition	Process Performance	Explanation

N:\123R5W\SCOPEFRM.WK4 Page 1 of 2 PTO

Figure 10-12. Scope form.

EQUIPMENT RELATED OTHER QUESTIONS:

1. Does this equipment meet the present operating requirements?
2. Does this equipment meet the future production requirements?
3. Are there any process or feed changes from the design?
4. What is a consequence of a failure (fire, explosion, etc.)?
5. Can it be isolated on line?
6. What is the production impact (BPD or $$), if this equipment is out of service?
7. Do we have a spare?
8. What work was done during the 1992 T/A?
9. Are there any HAZOP or loss prevention concerns with this equipment?
10. Do we need any additional information (inspection or performance test, etc.)
11. Are there any relief system concerns?

ECONOMIC ANALYSIS:

CORRECTIVE ACTION OPTIONS:

	Corrective Action Options	Predicted Failure/yr.	Mean Time to repair
1			
2			
3			
4			

ANNUALIZED COST:

Option	Total Cost	A/P i,n	Annualized Cost
1			
2			
3			
4			

RELIABILITY ANALYSIS (SAVINGS):

Historical Availability	Predicted Availability	Maint / Prod. Cost/Day	Prob. of Loss Prod.	Reliability Savings

TOTAL SAVINGS PER YEAR:

Option	Reliability Savings	Process Impv't	Energy Savings	Total Savings
1				
2				
3				
4				

NET BENEFIT ANALYSIS:

Annualized Cost	Annualized Savings	Net Savings	Impact on Safety

IMPACT ON SAFETY

COMMENTS:

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Figure 10-12. Scope form (continued).

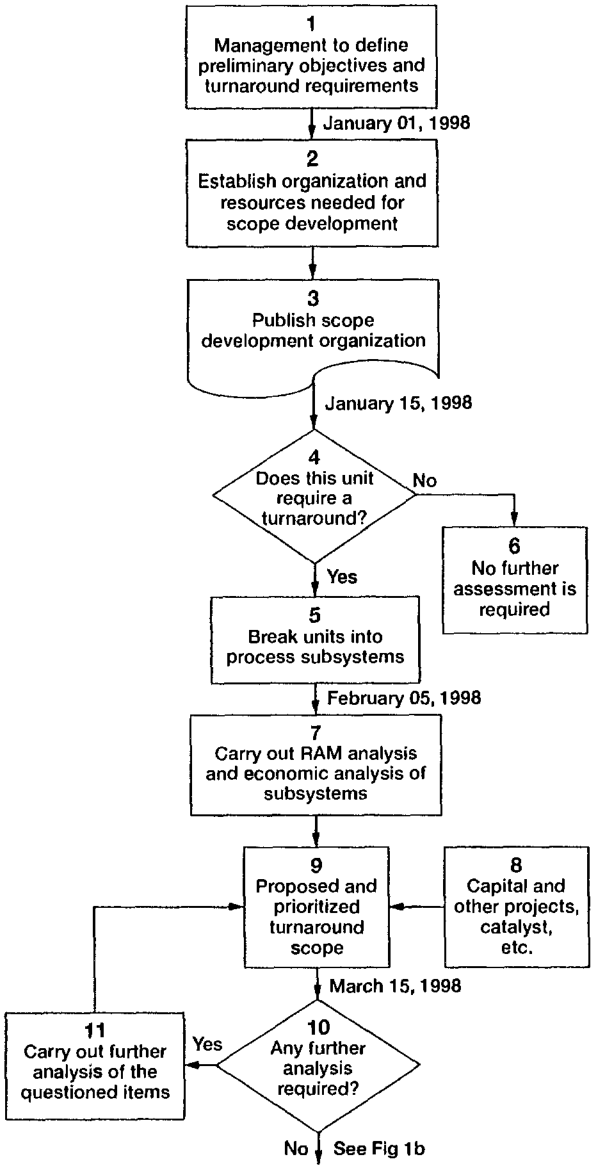

Figure 10-13. Flow chart of typical preturnaround activities.

(*text continued from page 401*)

various activities within the required time. Also, if a major capital project is to be implemented during the turnaround, more than one year may be required for project engineering, material delivery, and fabrication, etc.

Once the turnaround objectives are clearly defined by refinery management (budget, required run length, capital projects, etc.), a turnaround management team with a representative from each discipline—technical, operations and maintenance—should be

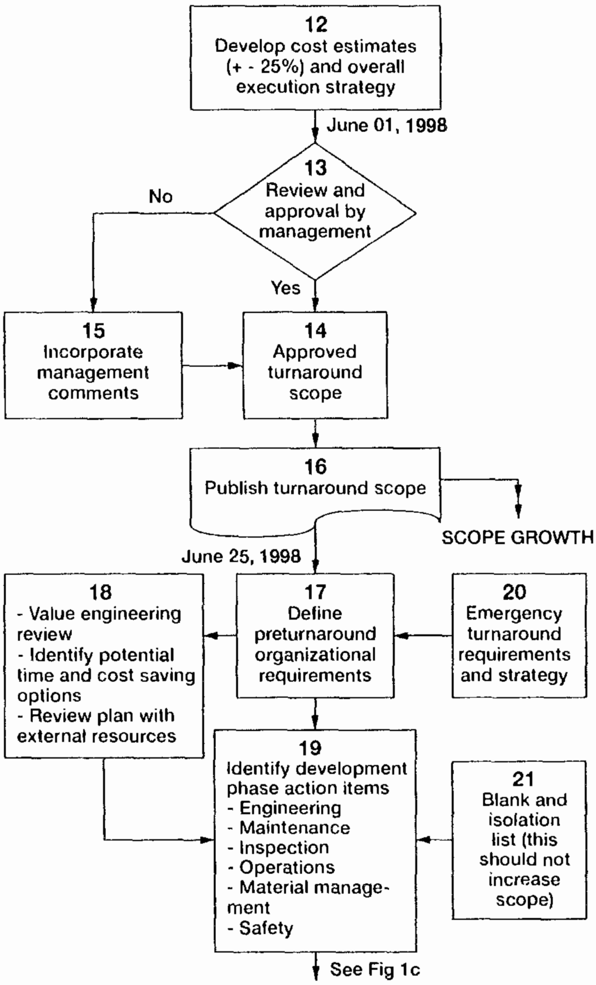

Figure 10-13. Flow chart of typical preturnaround activities (continued).

established. One member of this team is identified as a leader and the team becomes responsible for turnaround preparation and execution as shown in the flow chart.

The turnaround management team first reviews overall refinery operation to identify units requiring turnaround based upon the refinery configuration. The impact and relationship of the units scheduled for turnaround with the rest of the refinery is reviewed next to optimize the turnaround and production requirements during the turnaround period and to meet management objectives. For example, should the hydrogen plant or a crude unit be part of a hydrocracker plant turnaround?

Scope Development Team. A scope development team typically consisting of the following personnel should assess and analyze the unit condition, and develop turnaround scope recommendations.

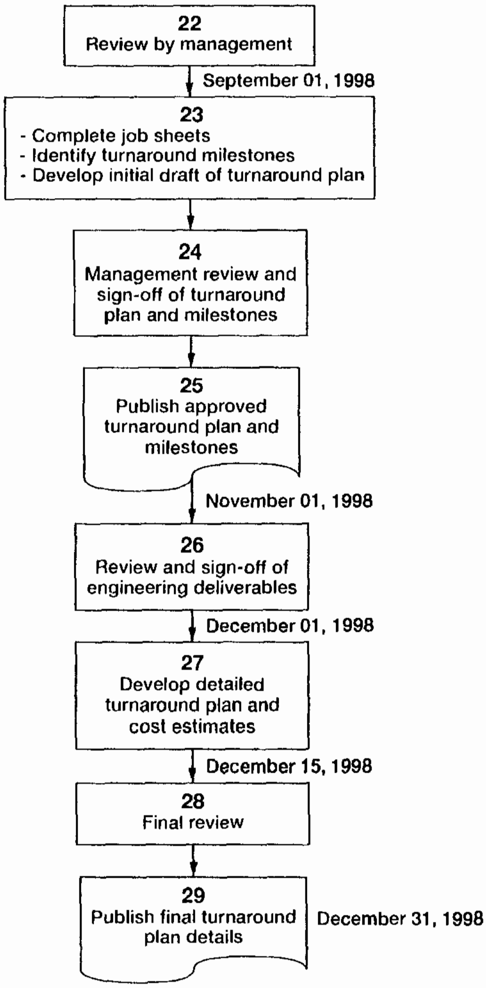

Figure 10-13. Flow chart of typical preturnaround activities (continued).

- Unit reliability/maintenance engineer
- Unit tech service/contact engineer
- Unit operations representative
- Inspector responsible for the unit
- Turnaround planner.

Other resources, such as a maintenance representative and personnel representing electrical, instrumentation, machinery, etc., should be invited to the meetings as needed.

More than one scope team may be required depending on the units involved in the turnaround. A midsize facility may have three teams evaluating seven units within the refinery. It is recommended that the turnaround management team members lead these scope development teams.

Unit Assessment. Each unit is usually broken down into one or more subsystems based on the similar process streams to address the common corrosion, erosion, and process concerns throughout the subsystem. This will also make the analysis much more manageable. For example, a hydrocracker unit can be broken down into two subsystems: reactor and fractionator.

Equipment Assessment. An equipment assessment form should be developed to review and analyze all the equipment consistently throughout the subsystem and unit. Historical failure rate and maintainability data, along with good understanding of current or expected equipment condition are required to predict the future failure rate for various options. During the assessment process, you may find that your historical data are poor, with sketchy information. However, understanding all the issues and the group discussions are the key elements in equipment assessment and developing potential solutions. Team members should share their experience to add to the equipment history, thus contributing positively to the group discussions. Plan to identify the requirements for collecting equipment history and maintaining it for the future.

Each potential solution will now be evaluated from an economic and safety standpoint using the following categories:

- Reliability savings
- Process improvements
- Energy savings
- Impact on safety.

"Do nothing" will be your first option and can be used as the basis for comparison against the other options. The evaluation is typically done annually.

It is important to understand that the numbers generated during the analysis are often based on assumptions and, in many cases, the numbers may have high uncertainty. However, accuracy is not critical since the numbers are intended for comparison. Certainly, data with higher certainty will produce better results. The true objective of the analysis is to make sure the right questions and issues are discussed during the decision-making process. Once the results of the initial analysis are completed, just the high priority items can be further reviewed, if required.

Economic calculations

Availability, $A_j = MTTF/(MTTF + MTTR)$ based on an average or constant failure rate[8]

Where j = option number (1, 2, 3 . . .); option 1 is "do nothing"

$MTTF$ = mean time to failure based upon historical data—reliability

$MTTR$ = mean time to repair—maintainability

Reliability savings, $R_j = (A_j - A_o) * (L_j) * (365\ \text{days/year}) * (P_j)$

Where A_o = historical availability
A_j = predicted availability associated with each option
L_j = loss of production cost and maintenance/day due to an incident
P_j = loss of production probability due to an incident as given below:

Failure effect	Probability, P
Actual production loss	1
Probable production loss	0.1–0.9
Possible production loss	0.1
No production loss	0

Annualized cost, C_j = (*cost of option*) * $(A/P_{i\%,n})^2$

Where $A/P_{i\%,n}$ = *capital recovery rate at i% rate of return in n years*

Process improvement, PI_j = *estimated extra revenues per year due to more production and/or better product as a result of the proposed option j*
Energy savings, E_j = *estimated savings per year due to an increase in energy efficiency as a result of the proposed option j*

Total savings, $TS_j = R_j + PI_j + E_j$

Net savings, $NS_j = TS_j - C_j$

Note that for a particular option, one or more of these benefits (R_j, etc.) could be negative and may appear to be a bad option. However, net savings for each option compared with the base option "do nothing" will provide real economics for that option.

Impact on safety due to an option can be ranked high, medium, and low, where high is an unacceptable safety risk and low is a minimal and acceptable safety risk.

Process Piping Assessment. Major process piping should typically be reviewed for each subsystem similar to that equipment. Here, many lines are often grouped together to speed up the process without any major impact on the quality of analysis. Economic analysis should also be carried out using the process described previously.

Review of Major Infrastructure Systems. Major infrastructure items include the following:[9]

- Instrumentation
- Cooling water
- Electrical power
- Sewer
- Fire system

- Nitrogen/instrument air/plant air/fuel gas
- Flare relief system.

Instrumentation should be analyzed separately for each process unit subsystem. The other infrastructure systems can be analyzed on a unit-wide basis.

Figures 10-14 through 10-17 and the following questions are typically used as a framework for these analyses. The economic analysis for these systems is then carried out in the same manner as the process equipment and piping.

Instrumentation

1. Check if any HAZOP or other outstanding issues are associated with the instrumentation.

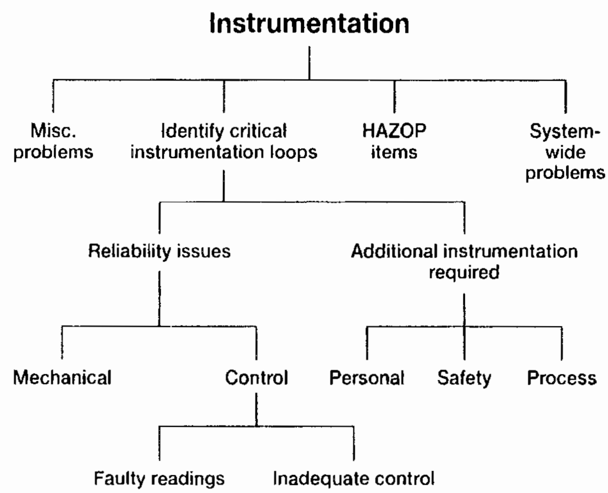

Figure 10-14. Instrumentation analysis.

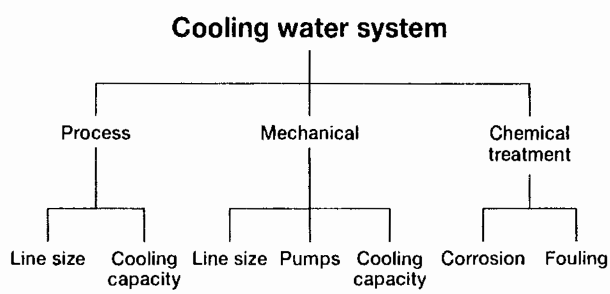

Figure 10-15. Cooling water system analysis.

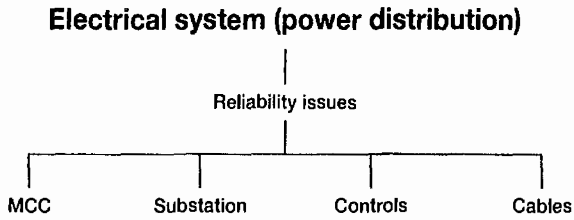

Figure 10-16. Electrical system analysis.

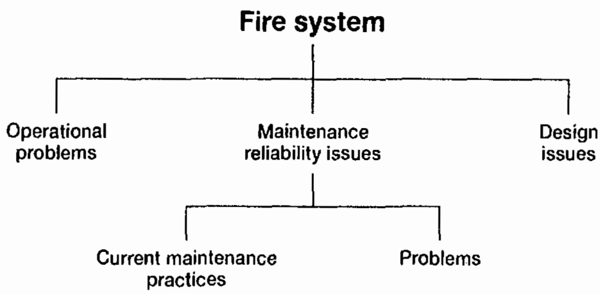

Figure 10-17. Fire system analysis.

2. Identify any problems that have a wide-range effect on the process control system.
3. Identify the critical control loops from a process and safety standpoint. Critical control loops are loops required to run the plant safety at maximum efficiency.
4. Review the critical loops as shown in Figure 10-13.
5. Review known instrumentation problems not associated with critical control loops to determine if action is warranted.

Cooling water

1. Identify any process issues, such as cooling capacity.
2. Identify mechanical issues, such as thin lines and leaking valves.
3. Review the system corrosion and fouling history to ensure there are no reliability/performance issues.

Electrical power

1. Identify reliability issues associated with the motor control centers, substation, control schemes and feeder cables.
2. Use the single-line diagrams for this discussion.
3. Consider a simple event tree logic diagram. This diagram would show the failure consequences. Note: Reliability/maintenance issues associated with electric drivers can and should be discussed together with the process equipment and not as part of this analysis.

Fire system

1. Identify any operational problems.
2. Review the design to ensure proper coverage.
3. Review current fire system maintenance practices.
4. Identify any maintenance/reliability issues.
5. Use the fire system P&IDs for this exercise.

Nitrogen/instrument air/plant air/steam

1. Identify any process problems related to supply.
2. Identify any maintenance/reliability problems.
3. Use the plant air and instrument air P&IDs as information sources for part of this analysis.

Sewers

1. Determine if any safety issues exist, such as, flooding, vapors, etc.

Case study 1. Depropanizer feed bottoms exchanger. Design conditions for this exchanger are given in Table 10-5A.

This exchanger was last inspected in 1992 and both shell and bundle were in good condition. It was originally installed in 1967 and had no failures since. Process performance has been good. If this exchanger were to fail, the hydrocracker unit would have to be shut down. The production loss would be 24,000 bpd at $3.00/bbl. Table 10-5B shows the results of group discussions.

Table 10-5A
Design Conditions for E-92-73A

Parameters	Shell Side	Tube Side
Design pressure	345 psig	285 psig
Design temperature	520°F	430°F
Service	Depropanizer feed	Butane

Table 10-5B
RAM Analysis Data for E-92-73A

Option	MTTF (years)	MTTR (days)	Option Cost
1 Do nothing	10	4	0
2 Open, clean, inspect and repair	10	4	28,500

Availability calculations:

$A_0 = \text{MTTF}/(\text{MTTF} + \text{MTTR}) = 28/(28 + (0/365)) = 1.000$
$A_1 = 10/(10 + (4/365)) = 0.999$
$A_2 = 10/(10 + (4/365)) = 0.999$

Reliability savings calculations:

$R_1 = (A_1 - A_0)*(L_1)*(365 \text{ days/year})*(P_1) = (0.999 - 1.000)*(77{,}000)*(365)$
$*(1.0) = -\$30{,}612$
$R_2 = (0.999 - 1.000)*(77{,}000)*(365)*(1.0) = -\$30{,}612$

Annualized cost calculations:

$C_1 = (\textit{cost of option})*(A/P_{i\%,n}) = (0)*(0.334) = 0$ ($A/P_{i\%,n}$ is @ 20% return over 5 years)
$C_2 = (28{,}500)*(0.334) = \$9{,}519$

Total saving calculations:

$TS_1 = R_1 + PI_1 + E_1 = -30{,}612 + 0 + 0 = -\$30{,}612$
$TS_2 = R_2 - PI_2 + E_2 = -30{,}612 + 0 + 0 = -\$30{,}612$

Net saving calculations

$NS_1 = TS_1 - C_1 = -30{,}612 - 0 = -\$30{,}612$
$NS_2 = TS_2 - C_2 = -30{,}612 - 9{,}519 = -\$40{,}131$

Calculations results are summarized in Table 10-6.

The obvious conclusion is to choose option 1, "do nothing" and not include this exchanger in the turnaround scope.

Table 10-6
Economic Analysis Summary for E-92-73A

Option	Annualized cost, C_j	Total annualized savings, TS_j	Net savings, NS_j	Incremental savings compared to "option 1"	Impact on safety
1	0	–30,612	–30,612	0	Nil
2	9,519	–30,612	–40,131	–9,519	Nil

Note: *In this case, total annualized savings are reliability savings only, since there are no process improvements or energy savings.*

Case Study 2. Lean Oil Contact Cooler. Design conditions for this exchanger are given in Table 10-7.

This exchanger was last inspected in 1992 and both shell and bundle were found in good condition. It was originally installed in 1967. Since 1992, however, there have been three incidents of tube failure. Process performance has been poor, primarily due to cooling water fouling, especially during the summer months. A consequence of this exchanger failing was a 12,000 bpd (at $3.00/bbl) production loss and $6,500/day due to flaring light ends (propane, butane, etc.) Tables 10-8, 10-9, and 10-10 show group discussion results and RAM and economic analysis summaries.

Table 10-7
Design Conditions for E-92-58A

Parameter	Shell side	Tube side
Design pressure	200 psig	150 psig
Design temperature	300°F	500°F
Service	Absorber overhead vapors	Cooling water

Table 10-8
RAM analysis data for E-92-58A

Option	MTTF (years)	MTTR (days)	Option cost
1 Do nothing	1	4	0
2 Clean in situ	0.5	4	8,000
3 Open, clean, inspect and repair	5	4	24,800
4 Replace bundle	10	4	40,000
5 Install a bypass around the exchanger	10	0	50,000

Table 10-9
RAM analysis summary for E-92-58A

Option	Hist. avail. A_0	Pred. avail., A_j	Prod./ main't. cost/ day, L_j	Prob. of prod'n loss, P_j	Reliability savings, R_j	Process impv't. PI_j
1	0.993	0.989	47,000	1	−73,899	0
2	0.993	0.979	47,000	1	−255,873	185,000
3	0.993	0.998	47,000	1	74,545	185,000
4	0.993	0.999	47,000	1	93,284	185,000
5	0.993	1	47,000	1	112,063	185,000

Note: *Process improvements are based on having the exchanger clean for summer months, which would allow for 144 bpd propane recovery for six months per year.*

Table 10-10
Economic analysis summary for E-92-58A

Option	Annualized Cost, C_j	Total Annualized Savings, TS_j	Net Savings, NS_j	Incremental Savings Compared to "Option 1"	Impact on Safety
1	0	–73,899	–73,899	0	Nil
2	2,672	–70,873	–73,545	354	Nil
3	8,283	259,545	251,262	325,161	Nil
4	13,360	278,284	264,924	338,823	Nil
5	16,700	297,063	280,363	354,261	Nil

Obviously, as shown, option 5, "install a bypass around the exchanger," is the best option and would be recommended for the turnaround scope. In this case, "do nothing" is not a preferred option.

Case Study 3. Main Fractionator Tower. Design conditions for this tower are:

Design pressure	101 psig
Design temperature	650°F
Material	Carbon steel
Service	Hydrocracker main fractionator
Size	10 ft-0 in. ID × 170 ft – 3 in. tall

The main fractionator tower was inspected through external manways only in 1989 and 1992. Full internal inspection (trays, internal manways, etc.) was done in 1985. This tower has been identified for wet-mag (WFMP) inspection, which was carried out at the accessible locations only during the 1989 and 1992 external manway inspections. Some wet H_2S indications were found in 1989, which were removed by light grinding. The same locations were inspected again in 1992 and showed no wet H_2S indications.

The consequence of a failure due to this tower would be a total hydrocracker unit shutdown at 24,000 bpd production loss. Tables 11, 12 and 13 summarize the group discussion results and RAM and economic analysis summaries.

As shown, option 2, "open, clean, and inspect (including internal trays, manways, etc.)" would be an obvious recommendation. With option 2, we should also realize some process improvements because this option would make sure that the tower internals are in good condition, which would improve tower efficiency.

Case Study 4. Make-Up Compressor Discharge Drum. Design conditions for this tower are:

Design pressure	1,789 psig
Design temperature	543°F
Material	Carbon steel
Service	Hydrogen

Table 10-11
RAM analysis data for T-92-52

Option	MTTF (Years)	MTTR (Days)	Option Cost
1 Do nothing	5	14	0
2 Open, clean and inspect (including internal trays, manways, etc.)	10	14	78,500

Table 10-12
RAM analysis summary for T-92-52

Option	Hist. Avail. A_0	Pred. Avail., A_j	Prod./ Main't. Cost/ Day, L_j	Prob. of Prod'n Loss, P_j	Reliability Savings, R_j	Process Impv't. PI_j
1	1	0.992	85,000	1	–236,186	0
2	1	0.996	85,000	1	–118,544	Unknown

Table 10-13
Economic analysis summary for T-92-52

Option	Annualized Cost, C_j	Total Annualized Savings, TS_j	Net Savings, NS_j	Incremental Savings Compared to "Option 1"	Impact on Safety
1	0	–236,186	–236,186	0	?
2	26,219	–118,544	–144,763	91,423	Nil

This drum was last inspected in 1992 and was found to be in good condition. It was originally installed in 1967 and has had no failures to date. Process performance has been good. If this drum failed, the hydrocracker unit would have to be shut down. Production loss would be 24,000 bpd at $3.00/bbl.

Group discussion results are shown in Tables 10-14, 10-15 and 10-16.

The obvious conclusion is to choose option 1, "do nothing" and not include this vessel in the turnaround scope.

Table 10-14
RAM analysis data for V-92-13

Option	MTTF (Years)	MTTR (Days)	Option Cost
1 Do nothing	10	4	0
2 Open, clean, mine and inspect	10	4	23,800

Table 10-15
RAM analysis summary for V-92-13

Option	Hist. Avail. A_0	Pred. Avail., A_j	Prod./ Main't. Cost/ Day, L_j	Prob. of Prod'n Loss, P_j	Reliability Savings, R_j	Process Impv't. PI_j
1	1	0.999	77,000	1	–30,612	0
2	1	0.999	77,000	1	–30,612	0

Table 10-16
Economic analysis summary for V-92-13

Option	Annualized Cost, C_j	Total Annualized Savings, TS_j	Net Savings, NS_j	Incremental Savings Compared to "Option 1"	Impact on Safety
1	0	–30,612	–30,612	0	Nil
2	7,949	–30,612	–38,561	–7,949	Nil

RAM Approach Advantages. Some key advantages are:

- Systemic approach with very little or no chance of missing items of concern
- Analysis, results, recommendations and decisions were documented for future reference
- Involvement throughout the organization (operations, technical, and maintenance) in the process—a team approach
- Thorough assessment of the unit based on RAM principles by the key stake holders
- Economic justification
- Total buy-in from the organization
- Identifies and defines future data collection requirements
- Establishes a baseline assessment and a process to continually improve RAM of the units.

Effective Maintenance and Maintenance Effectiveness Surveys*

Increasingly, modern process plants are intent on deferring equipment maintenance until the need for this work has been clearly established. These plants typically apply predictive, instrument-based condition-appraisal techniques, including vibration and corrosion monitoring, lube oil analysis, and continuous determination of

*Based on Bloch/Carroll, NPRA paper MC-90-74, presented at the National Petroleum Refiners Association Maintenance Conference, San Antonio, Texas, 1990.

machine efficiency deterioration. However, it can be shown that not all equipment responds to purely predictive techniques and that carefully chosen preventive, periodically implemented maintenance methods make economic sense.

Introduction

Predictive maintenance methods have become an indispensable part of the maintenance planning and shutdown strategies of modern process plants.

As is well known, the predictive approach has been used for many decades on rotating machinery such as pumps, compressors, steam turbines, and electric motors. Information is gathered over a long period on vibration severity; lubricating oil samples are collected, analyzed and trended; and, in some cases, considerably more sophisticated measurements such as noise, temperature, or vibration signature analysis are included in predictive maintenance routines.

Very often, though, predictive or condition-based maintenance is practiced to the exclusion of preventive maintenance. This can be a serious and costly mistake. An example would be waiting for lubrication deficiencies in regreasable rolling element bearings to manifest themselves through excessive vibration. In this case, it would be far more cost-effective to simply implement a regular program of periodic relubrication, the benefits of which will be shown later.

Another very important area where preventive maintenance neglect has caused extended equipment outages and even loss of life relates to steam turbine trip systems. Here, periodic exercising of trip valve stems and proper lubrication of sensitive linkages will prove inexpensive and far more effective than most "predictive" approaches.

A modern process plant is thus confronted with the task of determining which of the two maintenance approaches, preventive or predictive, should be chosen. A few examples may provide guidance in this regard.

Understanding How Components Fail: A Key Ingredient

A clear understanding of how components fail usually will assist in defining whether preventive or predictive programs should be implemented for a given equipment category or discipline. A representative illustration is metals inspection.

A typical metals equipment inspection program used in today's processing plants is based on the collection of extensive data involving the rate of corrosion or erosion of specific equipment, usually piping or vessels. These data are then extrapolated to determine the remaining life of the equipment. The resulting predictions can, and often do, set the run length of the equipment and determine a good portion of future maintenance to be undertaken.

A good metals inspection program can result in modifications to operating conditions or selection of alternative materials. Furthermore, such a program provides reasonable assurances that the equipment, given similar operating environments, will operate reliably and safely for the predicted period. Obviously then, metals equipment inspection programs are largely predictive in nature.

Similar approaches are valid for most pump maintenance programs. Data are taken, trended, and sometimes analyzed to evaluate the hydraulic and mechanical performance of these machines. The trended data provide a reasonable certainty that the equipment will operate satisfactorily. More importantly, predictive techniques can indicate an incipient failure if maintenance is not performed quickly. This analysis does not avoid repair work, but in the majority of cases prevents more costly repairs or even catastrophic failures. This essentially is one of the major goals of an effective predictive maintenance program for pumps and, to some extent, motors. Other major objectives are to extend the operating life of the equipment and to allow for proper work execution planning. Achieving all these objectives would certainly reduce overall maintenance costs.

Obviously, the above programs have in common the advantage that measurements and data can be collected while the equipment is in service. These data are then used to predict the future with a high degree of credibility. Unfortunately, all equipment does not lend itself to this approach. Many types of equipment require visual inspection or off-line measurements to collect the data necessary to determine physical condition.

Coping with Valve Failures That Cannot Be Predicted

Because of the potential seriousness of steam turbine rotor overspeed events, proper stop valve or trip and throttle (T/T) valve maintenance deserves high priority. Most process plants will disassemble line valves for inspection and repair when problems such as excessive seat leakage, or external leakage, or sluggish operation are apparent. But unless turbine stop valves and T/T valves are periodically tested and inspected, the problem may be discovered too late. Accordingly, T/T valve maintenance must emphasize preventive techniques.

Figure 10-18 depicts a fairly common and well-proven T/T valve assembly. This valve, and many similar models in use worldwide, will best respond to two modes of preventive maintenance: lube oil purification and exercising of the screw spindle, Item 36. Water-contaminated lube oil is prevalent in steam turbine hydraulic systems. With wet, contaminated lube oil on the hydraulically activated side of the oil cylinder, Item 73, corrosion products have sometimes lodged in the close-fitting clearance between piston rings and cylinder wall.

In one catastrophic failure event at a medium-sized U.S. refinery, water-contaminated lube oil was indirectly responsible for causing major damage to a large steam turbine. The initial problem was related to coupling distress and severe unbalance vibration. When the coupling bolts sheared, the steam turbine was instantly unloaded and the resulting overspeed condition activated a solenoid dump valve. Although the oil-pressurized side of the trip piston was thus rapidly depressured, the piston stem, Item 64, refused to move and the turbine rotor sped up and disintegrated. The emergence of reliable means of on-stream oil purification might have prevented the disaster,[10] and the use of an inexpensive lube oil purifier similar to the one illustrated in Figures 12-11 and 12-12 would have provided the required degree of oil dryness.

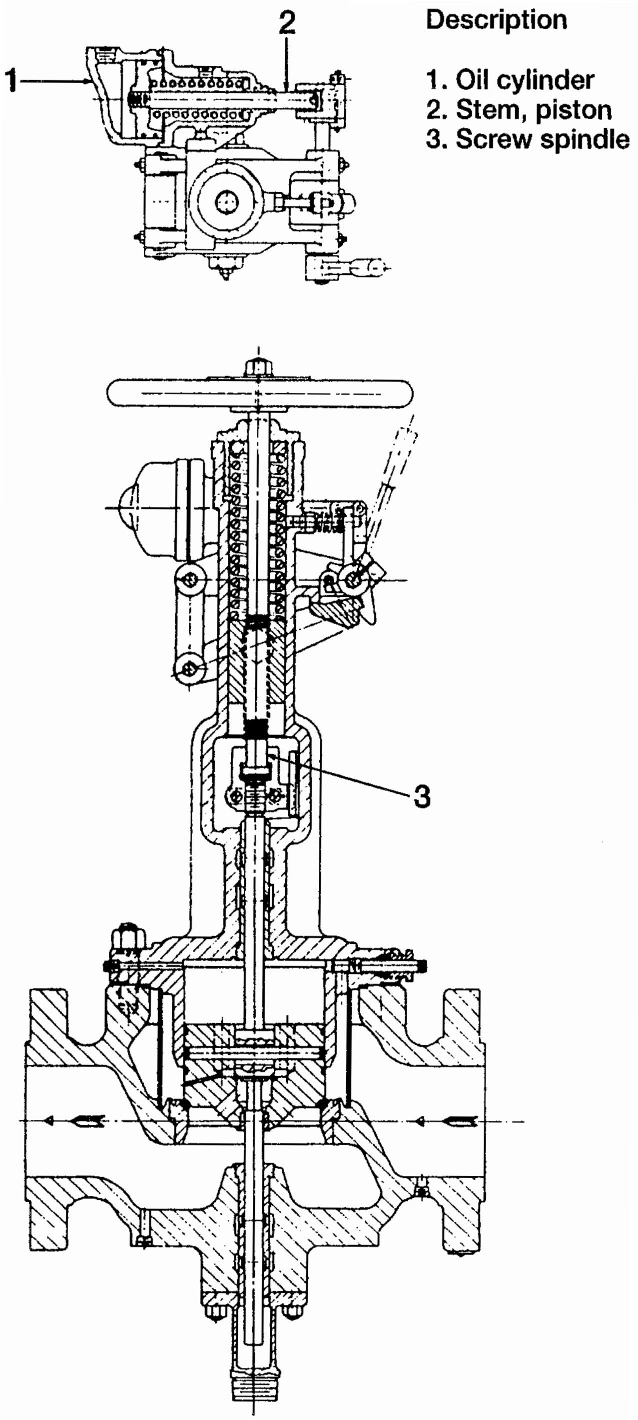

Figure 10-18. Common T/T valve assembly.

This device continuously removes water from lube oil by means of air-stripping, and is designed for permanent installation on oil reservoirs.

A conscientiously implemented *predictive* program of lube oil analysis[11] would assist in determining lube oil quality. However, a thorough cost comparison for such a predictive program with that of a routine lube oil purification program may show advantages for the latter.

In other words, *preventive* reliability assurance via continuous on-line purification of turbine lube oil can often be ranked ahead of the *predictive* method of lube oil condition analysis. This is shown in the comparison study, Table 10-17.

Many T/T valves also fail to perform when called upon because impurities in the steam collect on sliding parts in the steam passageways of the valve. The extent to which these deposits have collected, or the extent to which they impede proper sliding motion is very difficult, if not impossible, to predict. There is, however, one almost perfect way to ensure that sliding surfaces will remain in serviceable condition, i.e. exercising the valve periodically.

As will be noted from Figure 10-18, the T/T valve has a hand wheel (or hydraulic actuator) to allow "stroking" or partial closing of the valve stem with the steam turbine in normal operation. Since T/T valves are typically oversized, partial closing will not cause the steam turbine to lose speed. Exercising the valve verifies the integrity of sliding surfaces, and any deposit layers are wiped off before their presence would cause serious problems.

Preventive Maintenance of Electrical Equipment May Impact Machinery Reliability

Quite obviously, there are other equipment categories that also will not respond well to the predictive approach. For the sake of illustration, we choose to highlight

Table 10-17
Comparison Study

Yearly cost of a program of *preventive* reliability assurance via on-stream purification of lube oil:	
Hardware and installation amortized cost	$3,200
Utilities (electric power and heating steam at $0.05/kwh – equivalent)	660
Lube oil replacement every 5 years, prorated on 1-year basis, 1,000 gallons at $2.70/gallon	540
Labor at $30/hour, including such related services as lube oil disposal	500
TOTAL COST PER YEAR	$4,900
Yearly cost of a program of *predictive* reliability assurance via monthly lube oil analysis:	
Labor, at $30/hour (sample taking)	$ 180
Sample analysis, trend plotting and reporting	660
Lube oil replacement, 1,000 gallons at $2.70/gal	2,700
Labor, at $30/hour, including such related services as lube oil disposal	2,480
TOTAL COST	$6,020

Note: *The above comparison does not take into account the inherently greater risk of incurring machinery damage with lubricating oil supplies that do not apply on-line means of oil purification.*

two prominent ones: the electrical and instrument portions of typical refineries and other process plants.

In many instances, electrical and electronic equipment gives little or no warning of an incipient failure. It may operate as designed one moment, and fail the next. Circumstances that may ultimately lead to failure usually are present beforehand, but may defy detection by predictive means. Examples of deterioration can be corrosion of connections and contacts, deterioration of insulation, the presence of water or moisture, vibration-caused loose connections or wear, overheating, poor initial design or construction, exceeding design conditions, momentary overload, or combinations of the above. In instances of equipment that is not normally in operation, such as shutdown systems or auto-start equipment, a component may already be inoperative, but the condition may remain undetected until the apparatus is required to operate.

It is unrealistic to expect that all failures can be avoided. However, with a well-developed preventive maintenance program faithfully implemented and managed, much higher reliability will be achieved. Unfortunately, many organizations are still skeptical of preventive electrical maintenance, while others make only token efforts. In some cases upper management is convinced that they have a good program in place, but upon closer analysis, this may not be the case. In several recent maintenance effectiveness audits conducted by the authors, this situation existed. Upper management believed they had an effective preventive maintenance program for critical equipment, but little or no preventive effort was in fact carried through to completion. The reasons for such complacency are sometimes subtle, while others are very obvious. However, complacency is not unique and exists far more often than one would like to admit. It is often only after a serious outage or safety event that the weaknesses in the program are exposed.

Preventive vs. Predictive Maintenance for Typical Centrifugal Pumps

There can be no question on the merits of predictive vibration monitoring techniques in determining the existence of bearing defects, rotor unbalance, shaft misalignment, and other defects in centrifugal pumps. However, relying on vibration monitoring alone is tantamount to waiting for defects to announce their presence instead of preventing defects from developing in the first place. Think of the disparity between the practice of changing oil in automobiles and changing oil in process pumps. Most people would not even consider going against the manufacturer's edict to drive no more than 7,500 road miles (roughly 200 operating hours at 37 mph). Nevertheless, not many process plants practice oil replacement intervals as recommended in Figure 10-19.[12] This attitude prevails inspite of clear evidence that periodic oil changes will significantly reduce bearing failure events.

Failure to periodically change the lube oil in typical centrifugal pumps allows water, corrosion products, and other contaminants to contact the various bearing components, thus causing accelerated wear. The lube oil sometimes is replaced only when pumps are taken to the repair shop. And often, many pumps require repair sim-

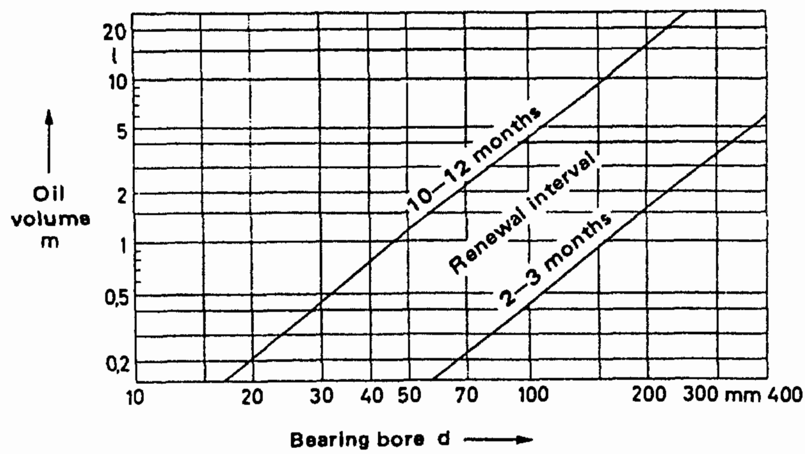

Figure 10-19. Oil quantity and renewal interval for oil sump lubrication.[12]

ply because the lube oil either was not, or could not be, drained in accordance with proper and cost-effective preventive maintenance practices.

In a recent study, the authors reviewed bottom-line maintenance and repair costs for pumps using properly applied conventional lubrication techniques and as-recommended oil drain intervals. These were found to compare quite favorably against oil-mist lubrication on economic grounds. Conversely, oil-mist lubrication looked attractive compared to conventional lubrication that is either incorrectly applied or applied without the benefit of periodic oil changes as the preventive maintenance method.

Again, and simply to re-emphasize, the record shows that *preventive* lube oil maintenance in the form of biannual oil changes makes economic sense on centrifugal pumps. Over-dependence on vibration monitoring can prove costly because it does not *prevent* problems, but only detects problems after the fact.

Electric Motor Bearing Maintenance Misconceptions

The two most prevalent reasons for electric motor bearing failures are over-greasing and the inadvertent mixing of lithium-soap grease (used by an estimated 85% of U.S. process plants) with polyurea synthetic grease (supplied until recently in an estimated 85% of electric motor bearings in the U.S.). A large number of electric motor bearings are prone to fail due to overgreasing because the shielded side of the bearing is installed *away* from the grease cavity (Figure 10-20) instead of *adjacent* to the grease cavity (Figure 10-21). Others fail because regreasing is done without removing the drain plug (item 2), which has been specifically provided for purposes of expelling spent grease and prevention of over-greasing.[13]

Compared to a conscientiously applied program of periodic regreasing, the occasional preference for "lifetime-lubricated" electric motor bearings is a poor choice

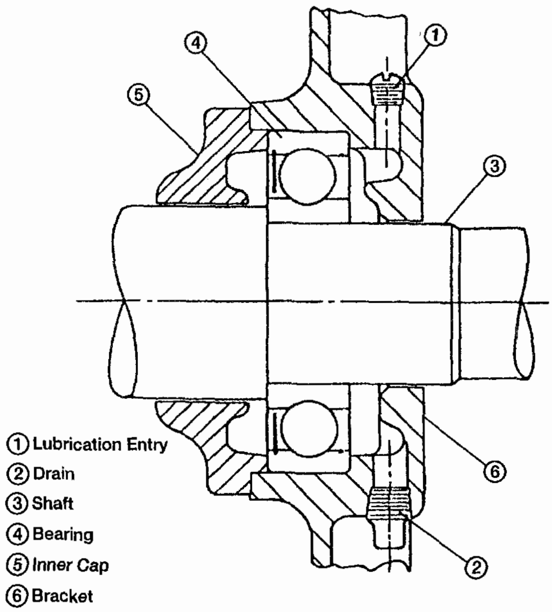

Figure 10-20. Bearing installation with shielded side away from grease cavity increases risk of over-greasing. Drain plug (2) should always be removed while regreasing this bearing.

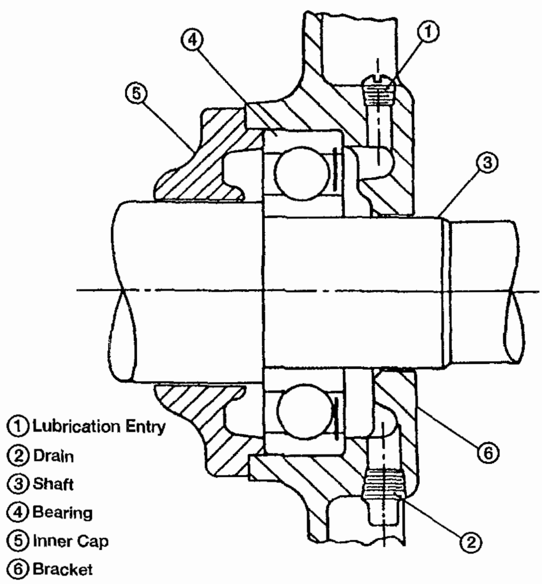

Figure 10-21. Placing shielded side of bearing towards grease cavity will reduce likelihood of over-greasing. So will removal of drain plug (2) while regreasing.

indeed. As indicated in Table 10-18, sealed, non-regreasable bearings rank at the very bottom of the bearing manufacturers' life expectancy tables. In fairness we should add, however, that over-greasing or mixing incompatible grease types are even less desirable.

Table 10-19 highlights the cost of incurring 156 electric motor bearing replacements per 1,000 motors per year[14] in a refinery practicing "occasional," and probably incorrect grease lubrication. This is contrasted with the cost of only 18 electric

Table 10-18
Influence of Lubrication on Service Life

	Oil	**Oil**	**Grease**	**Dry lubricant**
	Rolling bearing alone	Rolling bearing with gearwheels and other wearing parts	Rolling bearing alone	Rolling bearing alone
Decreasing service life**	Circulation with filter, automatic oiler	Circulation with filter	Automatic feed	
	Oil-air Oil-mist			
		Oil-air Oil-mist		
	Circulation without filter*			
		Circulation without filter*		
	Sump, regular renewal		Regular regreasing of cleaned bearing	
		Sump, regular renewal* Rolling bearing (a) in oil vapour		
		(b) in sump (c) oil circulation	Regular grease replenishment	
	Sump, occasional renewal			
		Sump, occasional renewal Rolling bearing		Regular renewal
		(a) in oil vapour	Occasional renewal	
		(b) in sump	Occasional replenishment	
		(c) oil circulation		
			Lubrication for -life	
				Lubrication for-life

**By feed cones, bevel wheels, asymmetric rolling bearings.*
***Condition: Lubricant service life < Fatigue life.*

Table 10-19
Cost of Electric Motor Bearings Failure

Number and cost of electric motor bearings failing without preventive maintenance ("occasional" regreasing):	
156/1,000 motors/years, at $1,800 per failure	$280,800
Number and cost of electric motor bearings failing with preventive maintenance (periodic regreasing):	
18/1,000 motors/year, at $1,800 per failure	32,400
Labor component of periodic regreasing, twice/year, $24/hour, 8 motors/hour	6,000
Materials component of periodic regreasing	4,600
	$ 43,000
Advantage realized by 1,000 motor plant practicing preventive maintenance:	$280,800
	− 43,000
TOTAL =	$237,800

motor bearing replacements per 1,000 motors per year, which we observed both at a petrochemical plant in the U.S. and a midsized refinery in the Middle East.

Why Some Preventive Maintenance Programs Prove to Be Ineffective

Some well-intended programs are often doomed to failure from the start due to the manner in which they are originated, developed, structured, implemented, or supported. By this we mean that the relevance of these programs perhaps has not been communicated to all parties affected, or input may not have been solicited from them. A further potential impediment to the successful implementation of a sound preventive maintenance or critical on-stream component verification program is the reluctance of equipment owners to risk what they perceive, quite often erroneously, as a procedure that could cause an inadvertent plant outage event. Since this is a valid concern, the issue merits significant attention. It should be addressed during the development stages of any preventive maintenance program, and may require training, simulations, detailed procedures, and similar actions.

Considering the above, how then does one go about implementing an effective preventive maintenance program? The key lies in the approach used in its development, the participation of all appropriate personnel functions, and the accountability and reporting of results. The following example primarily considers the approach that has proven successful for both instrument and electrical PM programs. Not only is it applicable to machinery maintenance as well, but since critical instruments are involved in machinery protection, a sound instrument and electrical PM approach is part of machinery reliability assurance.

Structuring an Instrument and Electrical PM Program

There is no single approach for a critical instrument checking program. All plants differ, organizations and manpower are unique, equipment is different and operating environments can vary widely.

But, personnel will more often support a program to which they or their peers have had input and participation. Conversely, if key personnel are not involved in the planning stage, support can be marginal. If an isolated section or group, such as the instrument engineers, develops the program, others will not be fully receptive to using it. Not encouraging full participation of all affected parties results in a missed opportunity for valuable input to ensure that the program is as well though out and workable as possible. The "package approach" in which one group or person develops the entire program should be discouraged.

Instead, the team developing such a program should be composed of technical/operations/and maintenance personnel. Each has critical input that can resolve a variety of problems—identified or potential—facing such a program. Ideally, the team should be composed as follows:

Site Instrument Engineer–an individual to lead the effort who is familiar with both the hardware configuration and proper design.

Site Instrument Technician–one or two people who have worked in the plant or unit with applicable hands-on experience to guide them.

Site Operating Specialist–an experienced operations person familiar with the equipment and one who knows the implications of its operation.

Nonresident Specialist–an experienced specialist from outside the plant. This person would serve as a source of new ideas, experience, and suggestions. Using a nonresident specialist can avoid reinventing the wheel. This person is an advisor only, serving as a resource person.

Management Sponsor–although not a part of the working team, visible management sponsorship is a critical success factor. Resources, both financial as well as personnel, are often necessary to correct existing deficiencies. A sincere commitment to implement the program must be more than mere words or memos. Support has to be more visible and can be implemented in numerous ways, e.g., through semiformal briefing sessions or status presentations.

Establishment of Objective and Schedule

Not all equipment requires routine maintenance, since its loss or failure may have little or no impact. The PM program objective should reflect this. It could be stated as "Improve the instrument reliability in those loops/systems that would cause a plant shutdown, significant economic loss, or severe safety hazard." Unless such guidelines are proposed, the end result can be far different than originally intended. It is also at this stage that key participants endorse the intent, effort, and schedule. They also must be willing to provide the necessary resources. A great deal of wasted effort can be avoided if agreement is reached at this early stage.

Approach and Content of a PM Program

First, one must determine which equipment should receive attention. This would normally be that which can cause a plant shutdown, major upset, or excessive economic loss. Each piece of equipment that falls into this category should then be reviewed for:

- Proper hardware
- Proper design
- Proper installation
- Ability to safely inspect/test

In addition, the organization should be reviewed for:

- Adequate experience
- Adequate skills
- Sufficient manpower
- Documentation and records
- Appropriate financial support or budgeting

There are other items that can have an impact on reliability, and these also should be addressed during the development period. Some are:

- Quality and reliability of utilities, particularly instrument air and electricity.
- Freezing, overheating, dirt, corrosion, general environment, and the presence of toxic or restrictive conditions.

Lists should be prepared, not only of the equipment to be examined, but the frequency and nature of specific work to be undertaken on each item, any special precautions required, specific approvals necessary, a means of recording results, detailed test procedures, and equipment used, etc.

One extremely important feature of the program is to have one individual or position clearly accountable for its development and implementation. Another is to have an effective means to present the results and progress. Only those directly involved in the implementation of the program require access to extensive details. However, the management sponsor and supporting organizations should be routinely presented with key statistics providing feedback on how the program is working. The status and progress of a given program and the general health of the equipment are thus better understood. In some of the more successful programs, brief status presentations are made to a top management group on a monthly schedule. This gives visibility to the entire effort and builds team spirit.

Proving the Program on a Small Portion of the Plant

Most programs must be debugged when first implemented. A trial area provides the learning experience before more significant effort has been expended and precludes extensive modification at a later date. It is prudent to select a more modern portion of the plant, especially one with good documentation. This initial trial area should be reviewed to evaluate the effort. Full implementation of the program in this selected area should be a precondition to the task. In addition to testing the effort, the trial area will provide data on the magnitude of the total task, its ongoing cost in manpower and other resources, and potential required modifications.

Maintenance Effectiveness Surveys Uncover Vulnerabilities

We generally assume that preventive maintenance programs will ensure that the equipment remains in serviceable condition. Similarly, many predictive maintenance programs are carried out for the distinct purpose of verifying that the equipment is presently in serviceable condition without necessarily taking steps that it will remain in that condition. This could lead to oversights and potential problems.

To be cost-effective, preventive maintenance must be applied with a good deal of forethought, experience, and judgment. Likewise, predictive maintenance must be confined to areas that lend themselves to prediction of impending distress. Preventive maintenance must lead to cost-effective failure avoidance; predictive maintenance must result in limiting the damage or must lead to the determination of remaining life.

The two approaches are often complementary, but at times they are mutually exclusive. Hence, the merits of each method should be reassessed periodically by a survey team of two or more engineers with broad-based experience.

Periodic *maintenance effectiveness surveys* are considered a highly suitable means of uncovering areas of vulnerability and areas where bottom-line maintenance cost savings can be realized. These surveys resemble machinery reliability audits that are aimed at identifying factors that can minimize forced machinery outages. However, maintenance effectiveness surveys are far more comprehensive in both scope and detail than pure machinery reliability audits. And, unlike *maintenance management* studies that concentrate heavily on manpower and organizational matters, a maintenance effectiveness survey goes into the when, how, why, and what to do with instrument, electrical, machinery, and related hardware. They should be scheduled at least every two years, and should be conducted by personnel whose experience and continuing work exposure gives them access to state-of-the-art techniques that transcend both industry and national boundaries.

Maintenance effectiveness surveys emphasize practical, implementable steps toward achieving plant-wide state-of-the-art reliability and availability to the limit. They are an extremely effective way to identify inappropriate design, inadequate equipment, poor installation, marginal applications, inadequate documentation, as well as repetitive problem areas. Maintenance effectiveness surveys also identify equipment upgrade opportunities. They have been shown to shift the maintenance emphasis from unplanned to planned work.

Conclusion

An effective maintenance program is one that places the emphasis on failure prevention, rather than failure correction. The net result of such an approach is safer operations, stable production, higher service factor, and overall lower costs. This, however, requires a proactive mentality rather than a reactive one. It also requires a "business" approach to maintenance rather than one that is just "service" oriented.

In order to achieve such an approach to maintenance, the proper use of either predictive or preventive maintenance is a key factor. In simple terms, predictive maintenance means using projected data or trends to determine the trouble-free service life of equipment. Preventive maintenance, on the other hand, means doing the minimal routine work necessary to ensure the equipment remains in proper operating condition. Although complementary to each other, the two are not necessarily interchangeable. And, while each has its own application within an operating plant, experience shows that the wrong approach is often pursued.

Maintenance effectiveness surveys can serve to sort out which of the two approaches is more appropriate in a given situation. Conducted by two or more engineers experienced in both maintenance management and equipment reliability assessment, these surveys provide rapid and valuable information on how to best utilize all available maintenance resources. The result will be the achievement of greater reliability of plant and equipment while, at the same time, minimizing bottom-line maintenance and repair expenditures.

How to Be a Better Maintenance Engineer

Today's maintenance or reliability professional is faced with many demands, and volumes of advice have been written on the need to organize, prioritize, and manage tasks, efforts, and schedules. Why, then, do many capable individuals still fall short of achieving these intuitively evident requirements? Could it be that they lack the basic foundation—certain prerequisites that would enable them to organize work and effectively manage time?

I believe that prerequisites exist and that fulfilling them is mandatory if the maintenance engineer wants to be productive and efficient. These prerequisites include, but are not limited to, establishing peer group and mentor contacts (networking), maximizing vendor engineering and sales force contributions, and searching and retrieving literature—all of these being activities of a resourceful person. A maintenance and reliability professional cannot afford to laboriously rediscover through trial and error what others have experienced and very often *documented* years earlier.

How to Practice Resourcefulness

Contact with a peer group can be established in a number of ways. Technical society and continuing education meetings promote information sharing and are certain to facilitate, as well as accelerate, the learning process, especially for relatively young or recently designated technicians and engineers. The speakers at such gatherings are often seasoned professionals, consultants, or recent retirees. It is implicit in their education and experience that they might fit the mentor role. No worthy mentor would ever refuse answering someone's phone call or verbal request for guidance and direction.

Asked a question about turbomachinery, he or she would direct the conversation to the activities of the Turbomachinery Laboratories of the Texas A&M University in College Station, Texas. Since 1972, the proceedings of the annual Turbomachinery Symposia have represented an easy-to-read collection of up-to-date, user-oriented technology, usually encompassing machinery design, operation, maintenance, reliability upgrading, and failure analysis/troubleshooting. The cross-referenced index to these symposia is without a doubt worth a small fortune. The same can be said about Texas A&M's International Pump Users Symposia and proceedings. These have been available since 1984 and will be of immense value to those earnestly seeking to put their industrial education on the fast track. And that's perhaps one of the keys to achieving true proficiency as a maintenance engineer/technician. Prior formal education will, at best, *prepare* us for a business or professional career; it will not, however, take the place of mandatory self-education. This self-education is, by definition, an ongoing and continuous effort in a competitive work environment.

What about trade journals? Reviewing at least their tables of contents is part of ongoing familiarization and technological updating that the maintenance professional must pursue. Imagine its value by considering the following scenario:

Your boss asks you to find a dependable long-term solution to repeated mechanical seal failures on your high-pressure ammonia pumps. You remember tucking away an article on high-pressure ethylene seals, without necessarily reading it at that time. But you find it and call up its author, Marlin Stone, who works for the Elephant Seal Company. You've never met him or even heard of him, but you know a lot about him! He's a communicator or he wouldn't have written this article. He's aware and perhaps even ahead of high-pressure seal developments, because the journal isn't known for rehashing old data. You call and tell him you've read his two-year-old article and found it of real interest. . . . I happen to believe that before you're close to telling Marlin Stone that *your* problem concerns not ethylene, but ammonia, Mr. Stone has already made up his mind to hear you out and either assist you outright or find the name of an ammonia expert who will do so.

Now let's look at the alternative. Since you don't have access to trade journals (honest now, is that the truth?), you call the local representative of Pickme Packing Ltd. who will instantly assure you that George Pickme, Jr. is the expert on that service and they would be delighted to be your partner supplier. Two years and seven modifications later, you realize that Pickme Packing Ltd. used your plant as a test facility to hone their skills in sealing a nasty product at your expense.

To be resourceful also implies that the maintenance engineering practitioner maintains contact with several competing vendors in an open and ethical manner. Suppose you spot excessive wear on your pulverizer gears. You know it's excessive because you spoke to the maintenance managers at three other user sites ("networking," in its implemented form), and you recall reading about the benefits of synthesized hydrocarbon lubricants. You recall picking up literature at a recent trade show and proceed to call three apparently prominent manufacturer-formulators of these advanced lubricants.

After explaining the situation, you follow up with a confirming fax to each vendor. You disclose relevant material specifications, configuration, speed and load details and request written replies by a certain deadline. Two replies arrive on time, the third vendor will need a more urgently worded reminder. When the three replies are available for review and closer scrutiny, you discover that one of the various defining lubricant parameters listed by vendor "A" differs from the ones quoted by

"B" and "C." This prompts you to ask "A" for an explanation of the significance of the deviation: continuing education at work.

Once the maintenance/reliability professional learns to tackle similar component and equipment upgrade issues by simultaneously using this approach, repeat problems will burden the organization less frequently. At this point, our professional will clearly be more productive and management may take notice.

If access to management personnel needs a boost, prepare monthly highlights; a one-page (maximum) summary of activities, work progress, accomplishments and value added. If a draft copy of these monthly highlights is discussed with operations and maintenance workers and credit is given where it is due, the maintenance professional will gain the respect and rapport of a surprisingly large number of appreciative and cooperative fellow employees.

And *now, only now,* will it make sense to address organizing, prioritizing and time management strategies. The maintenance/reliability professional should document daily how time was spent. An ordinary desk calendar or PC will do, and both today's activities as well as planned activities days and weeks ahead should be retrievable. *Weeks* ahead? Yes, goals, deadlines, vendor followup target dates, meetings, etc., should be listed. The desk calendar or PC screen represents your informal training plan. Telephone numbers are punched into an electronic organizer; remember, proper vendor contacts are part of the engineer's/technician's training and productivity enhancement approach. Work requests without stated or implied deadlines go into a "suspend file;" requests that are difficult to tackle will be discussed with the mentor.

Try this approach; you'll be surprised how well it works.

The Role of the Maintenance Engineer in the Knowledge Age*

While our earlier segment was meant to convey how resourcefulness can be acquired, it is fair to say that modern maintenance, i.e., plant availability management, requires rigorous methodology, adherence to processes, and a profound knowledge of cause and effect. Plant availability management cannot be accomplished by relying entirely on skill and experience, as maintenance departments have done in the past. The typical maintenance organization of the 1990s is technically backward, even by Industrial Age standards, and is currently unprepared for the information age. To find the optimal availability solution between appropriate reliability and maintainability options and match the plant's output to current market conditions is a capability that the maintenance organization cannot attain without the involvement of highly skilled maintenance engineers.

In the past, maintenance engineers played a minor role in setting manufacturing strategies and policies. The maintenance engineer was used primarily to solve problems that could not be solved by the skill and experience of the maintenance supervisor. It was not uncommon for the maintenance engineer's position to be filled by

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*Based on a presentation by Paul Smith, Electronic Data Systems, Houston, Texas. Adapted, by permission, from the Proceedings of the 5th International Process Plant Reliability Conference, Houston, October 1995.

employees trained in other disciplines. The position was used as a training position to produce generalists who later became managers. In the information age this position will be filled by highly trained specialists. The maintenance engineer must become an interpreter who can translate the output of *applying knowledge to work* into daily activities that can be performed by the maintenance staff.

The maintenance engineer must now become active in setting manufacturing strategies and policies and in determining solutions to daily problems. What the maintenance organization does, when they do it, and how they do it will be determined by rigorous methodology and analysis of information. The maintenance engineer will move from being an occasional problem solver to becoming active in the daily decision making and goal setting of the maintenance organization. Tasks performed by the maintenance engineer of the first decade of the 21st century will almost certainly include:

- Failure mode and effects analysis
- Fault tree analysis
- Weibull analysis
- Interpretation of plant availability modeling
- Establishing and managing effective preventive maintenance programs
- Cost analysis
- Maintenance strategy development
- Failure analysis
- Risk analysis
- Maintenance task analysis

The output of these knowledge-based tasks will become the basis of all work done by the maintenance organization. The maintenance organization of the next decade can no longer rely on skill, experience, and past practices, but must now be able to predict with great accuracy the financial consequences of all of its actions. These will not be abstract theoretical exercises, but ongoing actions that translate the plant's knowledge base into daily maintenance activities. The maintenance engineer interpreting the information in the plant's maintenance computer systems will give the maintenance organization the capability to control the plant's availability in a real-time mode.

The definition of these tasks cannot be performed without the formal education that the maintenance engineer either possesses or will have to acquire. As the maintenance engineer becomes a highly trained maintenance specialist, his contribution will become critical to the success of the process plant of the future.

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Chapter 11

Maintenance Cost Reduction

Maintenance cost reductions are possible through implementation of appropriate organizational procedures, optimum supervision, and judicious utilization of contract labor in certain circumstances. However, we are primarily concerned with engineered reliability improvement items. Specifically, we want to provide some insights into the rationale that prompted:

1. Elimination of cooling water from general-purpose pumps and drivers
2. Use of dry-sump oil-mist lubrication for pumps and electric motors
3. Adoption of non-lubricated couplings for all classes of rotating equipment
4. Widespread usage of laser-optic alignment verification
5. Machinery condition monitoring with operator-friendly vibration meters.

Eliminating Cooling Water from General-Purpose Pumps and Drivers

Extensive experimentation with removal of cooling water from pumps and general-purpose turbine drivers in large petrochemical plants indicates that the elimination of cooling water may, in fact, increase machinery reliability. The obvious savings in capital expenditures for piping and water-treatment facilities, and savings in operating cost alone, provide good incentives to take a closer look at this topic. But, in attempting to explain the merits of eliminating cooling water from this equipment category *entirely,* machinery engineers must not only look at the effect on bearings. They must also be prepared to deal with questions relating to pedestal cooling and stuffing-box cooling. Fortunately, experience exists and can be readily summarized.[1]

It has been shown conclusively and over a period of many years that pedestal cooling is not required for any centrifugal pump generally found in petrochemical plants. Pumping services with fluid temperatures as high as 740°F (393°C) require nothing more than hot alignment verification between driver and pump.

Pump stuffing-box jacket cooling, while reducing heat migration from the pump casing toward the bearing housing, will not serve as an effective means of lowering the temperature in the seal environment. A changeover to high-temperature mechanical seals may be possible and is preferred by U.S. plants. If mechanical seals need cooling because the flush liquid has a low boiling point, the least troublesome way to control seal temperatures may be to circulate a coolant such as water, steam, or cool flushing oil through a jacket which is part of the mechanical seal package. Figure 11-1 shows a well-proven design of this type. Note the bellows configuration (1) at the high temperature and pusher configuration (2) at the lower temperature region of the seal package.

An alternative solution would be to route some of the pumpage from the pump discharge line through a small cyclone separator, a flush cooler, a filter, and an orifice, and then into the stuffing box. The cooled flush would provide the proper temperature environment for the seal components and prevent solid contaminants from entering the stuffing-box area through the throat bushing of the pump. However, clean hot services may well be ideally suited for a maintenance-free dead-ended flush arrangement after converting to the cooling jacket configuration shown in Figure 11-1 or after installing high temperature gas seals, pages 550–558.

Bearing Cooling Is Not Usually Needed

Cooling water can be deleted from many sleeve bearings on centrifugal pumps and on small turbine drivers after experimentally verifying that oil sump temperatures do not exceed an operating limit of 180°F (82°C). This limit was found to be extremely conservative from a bearing-life point of view. If it is exceeded by a few degrees, more frequent oil sampling or oil replacement may be appropriate. A good synthetic lubricant may be ideally suited in this event and is easily cost-justified.

Since most general-purpose machinery is equipped with anti-friction bearings, attention is primarily directed to the significant maintenance credits which can result from eliminating cooling water from anti-friction bearings on pumps and small steam turbines. Experience shows that equipment life can actually be extended by removing cooling water from bearings. Cooling of bearing oil sumps invites moisture condensation, and bearings will fail much more readily if the oil is thus contaminated by water. Laboratory tests show that even trace amounts of water in the lube oil are highly detrimental. Hydrogen embrittlement on the steel granular structure can reduce the expected bearing life to less than one fifth of normal or rated values.

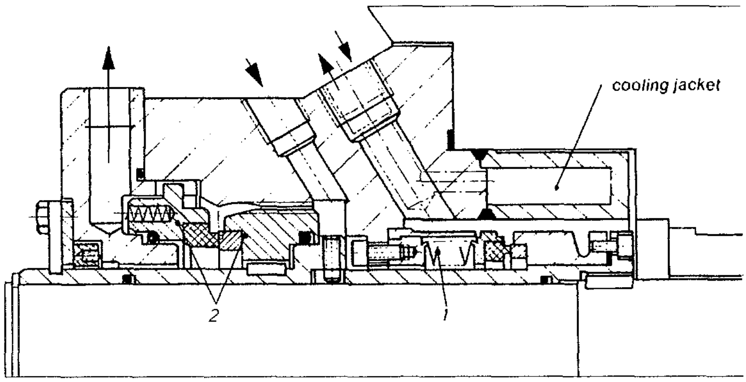

Figure 11-1. Seal cooling jacket separate from pump. (*Courtesy of Burgmann Seals America, Houston, Texas.*)

Another reason for not cooling the bearing housings of pumps and drivers is to maintain proper bearing internal clearances. Hot-service pump bearings have often failed immediately after startup when the bearing housings were cooled by water. When it was recognized that high temperature gradients were responsible for reducing bearing clearances to unacceptably low values, a heating medium was introduced into the bearing bracket to heat the housing: The problem was solved.

Parameters Which Influence the Need for Bearing Cooling

The minimum permissible viscosity of ball-bearing lube oils at the operating temperature of the bearing is a function of bearing size and speed as defined in Figure 11-2. As a rule of thumb, and valid for most bearings operating in typical centrifugal pumps, rated bearing life will be obtained if metal temperatures of operating bearings remain low enough to ensure minimum viscosities of 150 SUS (32.1 cSt) for spherical roller bearings in thrust-loaded services, 100 SUS (20.6 cSt) for radially loaded

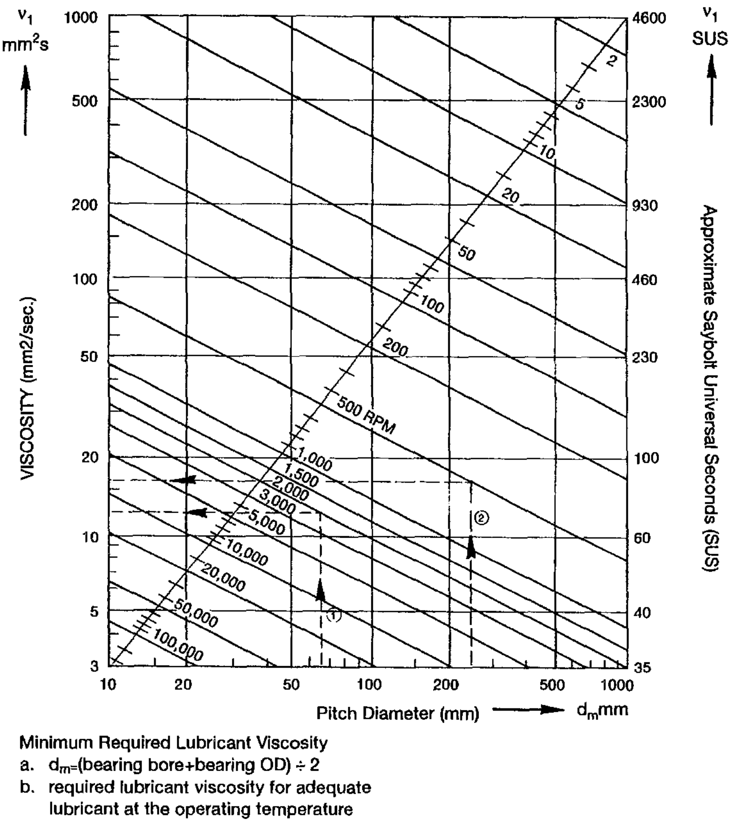

Figure 11-2. Minimum required lubricant viscosity as a function of bearing size and speed. *(Courtesy of SKF Bearing Co.)*

spherical roller bearings, and 70 SUS (13.1 cSt) for ball and for cylindrical roller bearings. If the viscosities drop below the given values, the oil film may have insufficient adhesion or strength, and metal-to-metal contact could result. While this indicates that lube oils should be selected primarily on the basis of maximum bearing temperature, consideration should also be given to oil viscosity at startup of idle standby equipment in cold climates. Pump warmup bypasses or oil viscosity selection based on minimum ambient condition may be required in some isolated instances. However, the majority of pumps operating in low ambients will start up and perform without difficulty as long as these higher viscosity oils have low pour points.

Many pump bearings will experience only a surprisingly small temperature rise after cooling has been discontinued, and an average temperature rise of 8°F (5°C) on a sample of 36 centrifugal pump bearings is typical of our observations. The additional heat input can either be *removed* by dissipating some of the heat traveling along the shaft, or accommodated by selecting a lubricant which will exhibit satisfactory viscosity even at the increased bearing operating temperature.

It is safe to assume that standard anti-friction bearings will show no loss of life as long as metal temperatures do not exceed 250°F (121°C). Maintaining oil temperatures within given limits is thus aimed at satisfying only two requirements:

1. Oil viscosities must remain sufficiently high to adequately coat the rolling elements under the most adverse operating temperature.
2. Oil additives, such as oxidation inhibitors, *must not be boiled off,* i.e., adequate service life of the lubricant must be maintained.

Experience shows that the additional heat input could be accommodated by selecting a lubricant with higher viscosity. Figure 11-3 can be used to determine the safe allowable operating temperature for several types of anti-friction bearings using two

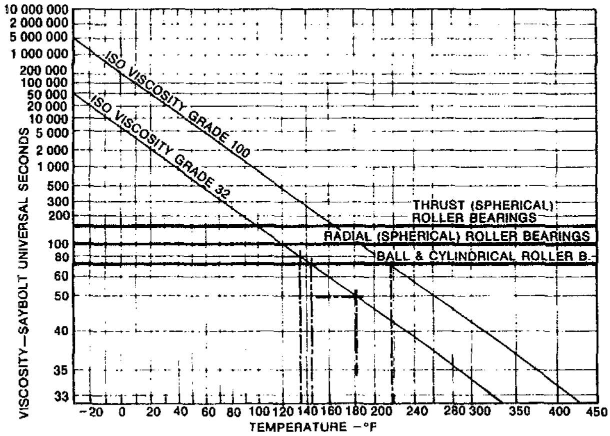

Figure 11-3. ASTM standard viscosity-temperature chart for liquid petroleum products (D341-43.)

grades of lube oil. ISO viscosity grade 32 (147 SUS at 100°F or 28.8–35.2 cSt at 40°C) and grade 100 (557 SUS at 100°F or 90–110 cSt at 40°C) are shown on this chart. Other viscosity grades can be sketched in as required. The chart shows, for instance, a safe allowable temperature of 145°F (63°C) for ball bearings with grade 32 lubrication. Switching to grade 100 lubricant, the safe allowable temperature would be extended to 218°F (103°C).

If a change from grade 32 to grade 100 lube oil should cause the bearing operating temperature to reach some intermediate level, a higher oil viscosity would result and bearing life would actually be extended. This can best be illustrated by an example.

A ball bearing with a pitch diameter of two inches (50 mm) operates at 3600 rpm. The lubricant is ISO viscosity grade 32 and, with water cooling, the bearing operating temperature is observed to be 135°F (57°C). Figure 11-3 shows this operating temperature corresponding to a viscosity of 80 SUS (15.7 cSt), which exceeds the rule-of-thumb minimum requirement of 70 SUS and makes this an acceptable installation. Reference to Figure 11-2 places the intersection of the 50 mm line with the bearing speed line below 80 SUS, thus reconfirming that the required lube-oil viscosity is exceeded by the available lube-oil viscosity.

Let us say we remove cooling water without going to a higher viscosity oil and find the bearing operating temperature has climbed to 185°F (85°C). This would result in a viscosity of only 50 SUS (7.4 cSt), which is below the safe acceptable value of 70 SUS (13.1 cSt) given in Figure 11-3, and places the pitch diameter/bearing speed line intersection, i.e. required lube-oil viscosity, above the available lube-oil viscosity in Figure 11-2. Safe long-term operation of typical centrifugal pumps requires compliance with the acceptability criteria of Figure 11-2 and 11-3. Let us assume now that changing to a lube oil with ISO grade 100 finds the bearings operating at 195°F (91°C). In Figure 11-3, this corresponds to a comfortably increased viscosity of 90 SUS (18.2 cSt) and, as expected, a shift towards adequate lubrication in Figure 11-2. The only penalty to be paid for switching to higher viscosity lubricants is a slight increase in friction horsepower which must be overcome by the pump driver.

Very few of our experiences with cooling-water removal from anti-friction bearings showed temperature increases as drastic as those given in the example. In fact, on quite a number of occasions, deletion of cooling water has resulted in decreased bearing operating temperatures. What at first appeared to be a puzzling observation was soon explained. As indicated earlier, fully jacketed water-cooled bearing brackets may thermally load a bearing because the bearing outer race is not allowed to expand freely. This may cause the bearing clearances to be uniformly reduced and operating temperatures to rise. Partially jacketed water-cooled bearings may cause thermal distortion of the bearing housing and tend to invite bearing distress in this fashion. Very significant increases in bearing life were obtained after thus recognizing that certain cooling methods may achieve exactly the opposite of their intended purpose.

We know of a 150-HP bottoms product pump with a stream temperature of 690°F (366°C). This pump, like dozens of others with product temperatures ranging as high as 740°F (393°C), does not require bearing cooling water and continues to operate with dry-sump oil-mist lubrication in once-through application of the lubricant. Only fresh oil containing the required amounts of rust and oxidation inhibitors originally compounded by the lube processing plant reaches the rotating elements. Dry-sump oil mist is an ideal lubrication method for anti-friction bearings operating in cost-conscious petrochemical facilities. Additional details on oil-mist lubrication are given later in this chapter under "Economics of Dry-Sump Oil Anti-Friction Mist Lubrication for Anti-Friction Bearings."

Implementing a Program of Removing Cooling Water

Petrochemical plants can easily implement a program of removing cooling water from pumps and drivers in well-planned, step-by-step fashion. Highest priority should be assigned to removing cooling water from pedestals and making the necessary hot alignment checks and adjustments. Eliminating cooling water from anti-friction bearings should be next on the priority list. As confidence is gained and maintenance cost reductions are realized, the program can be extended to cover sleeve-bearing and mechanical-seal applications.

After removing cooling water from existing pumps, or after commissioning new pumps without cooling water, measurements may be made to ascertain that the viscosity limits given earlier are not exceeded. Resistance-type thermometers are well suited for measuring either sump or bearing metal temperatures. Plain immersion of the wire tip into the oil sump will give an almost instantaneous reading. However, these temperatures may not reflect the bearing metal temperature. Viscosity determinations should, therefore, be based on the assumption that actual temperatures at the rolling elements are approximately 10°F higher. The preferred measuring method would be to detect bearing metal temperatures via a small hole drilled through the bearing cover and extending to the thrust-bearing outer race periphery. The thrust bearing is chosen because it is generally more highly loaded than the radial bearing.

A typical program for eliminating cooling water from pump and driver bearings is outlined as follows. These guidelines apply to single and multi-stage centrifugal pumps; other types of pumps should be considered separately for removal of cooling water. It should be noted that temperature measurements should preferably be made under the most adverse ambient conditions.

- Cooling water should be removed from all pump bearing brackets with dry-sump oil-mist lubrication, regardless of the pump product temperature. As a general precaution, we may wish to take temperature measurements on the bearing caps of pumps with pumping temperatures in excess of 500°F (260°C). These measurements can be discontinued after about two hours.
- Pumps with rolling element bearings and product temperatures below 350°F (177°C) should have all cooling water removed from bearing bracket, gland, and stuffing box. Cooling-water piping should be dismantled on a planned basis if temperature monitoring for a period of two hours shows lube oil temperature-viscosity relationships in the acceptable range, as defined earlier.
- For pumps with rolling-element bearings and pumping temperatures of 350°F (177°C) and higher, shut off cooling-water supply to bearing bracket and monitor oil temperature for four hours. Final oil temperatures in excess of 200°F (93°C)

would require diester or polyalpha-olefin synthetic lubes, or conversion to dry-sump oil-mist lubrication.

- Pumps with sleeve bearings, pumping temperatures below 250°F (121°C), and bearing diameter less than three inches at shaft speed of 3600 rpm or less than six inches at a shaft speed of 1800 rpm should be subjected to the four-hour temperature monitoring test. All cooling water should then be removed.
- For pumps with sleeve bearings and pumping temperatures of 250°F (121°C) and higher, reduce cooling-water flow while standing by. Subject pumps to the four-hour temperature test. Final oil temperatures in excess of 180°F (82°C) would require suitable synthetic hydrocarbon lubricants or some means of blowing forced air (from TEFC motors) over the bearing housing.

Summary

Bearing cooling water can be deleted from virtually all centrifugal pumps normally encountered in petrochemical plants. Experience shows that uncooled bearings will often operate more reliably than cooled bearings.

Pedestal cooling is not required, but hot alignment verification is needed.

Mechanical-seal cooling water can often be eliminated if a high-temperature seal is substituted for the conventional mechanical seal, or if a cool, external flush stream is routed to the seal faces or through extended seal seats available from experienced seal manufacturers. Gas seals eliminate the problem altogether.

Economics of Dry-Sump Oil-Mist Lubrication For Antifriction Bearings*

Optimized bearing lubrication is not necessarily achieved by choosing a lubricant resulting in moderate sump temperatures.[2] Higher viscosity lubricants, although causing slightly higher operating temperatures, may extend the life of bearings and rotating equipment by forming thicker, or better adhering, oil films. These beneficial effects can be quantified and were graphically represented in the preceding section.

Further optimization can be achieved by selecting appropriate oil-mist lubrication methods. Dry-sump lubrication is explained here in detail and relevant examples will be presented.[3]

This section also discusses the demonstrated merits of proprietary pump bearing housing seal designs that promise to prevent environmental contaminants from reaching critical parts of operating or "on standby" pump bearings.

Selecting the Correct Oil Viscosity

Contrary to long-held belief, optimized bearing lubrication is *not* usually achieved by simply choosing a lubricant resulting in moderate sump temperatures. Bearing

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*The reader may also wish to consult H. P. Bloch and A. Shamim's comprehensive text on this subject, *Oil Mist Lubrication: Practical Applications*, Fairmont Press, Lilburn, Georgia, 1998, ISBN 0-88173-256-7.

speed, loading, and lubricant viscosity are important parameters which have been shown to influence bearing life. These factors merit close consideration if optimum bearing lubrication is to be defined.

Proper lubrication requires that an elastohydrodynamic oil film be established and maintained between the bearing rotating members. Thus the proper lubricant is one which will form a thick oil film between the rotating parts. This oil film must ensure that no metal-to-metal contact takes place under foreseen speed and load conditions.[4]

Maintaining a minimum base oil viscosity of 70 Saybolt Universal Seconds (SUS) or 13.1 centistokes (cSt) has long been the standard recommendation of many bearing manufacturers. It was applied to most types of ball and some roller bearings in centrifugal pump services, with the understanding that bearings would operate near their published maximum rated speed, that naphthenic oils would be used, and that the viscosity would be no lower than this value even at the maximum anticipated operating temperature of the bearings. Figure 11-3 showed how higher viscosity grade lubricants will permit higher bearing operating temperatures. Most ball and roller bearings can be operated satisfactorily at temperatures as high as 250°F (121°C), from the metallurgy point of view. The only concern would be the decreased oxidation resistance of common lubricants, which might require more frequent oil changes. However, the once-through application of oil mist solves this problem.

These findings prompted many major petrochemical plants to standardize on ISO grade 100 lubricants, although a number of centrifugal pump manufacturers persist in recommending lower viscosity grade oils for anti-friction bearings in their products. Still, the results of the conversion proved highly affirmative. Application of ISO grade 100 lubricant allowed users to reduce their maintenance expenses further when it was recognized that cooling water could be eliminated from anti-friction bearings in a large number of centrifugal pumps. Services with pumping temperatures as high as 740°F (393°C) were involved, and cooling water was safely removed from even these!

Oil-Mist Lubrication for Pumps

Several large petrochemical pants in North and South America have extensive and long-term experience with automated oil-mist lubrication systems. This application method has proven to be particularly suitable for lubricating centrifugal pumps and their electric motor drivers.

Oil-mist lubrication is a centralized system which utilizes the energy of compressed air to supply a continuous feed of atomized lubricating oil to multiple points through a low-pressure distribution system, approximately 20 inches H_2O.[5] Oil mist then passes through a reclassifier nozzle before entering the point to be lubricated. This reclassifier nozzle establishes the oil-mist stream as either a mist, spray, or condensate, depending on bearing configuration and operating parameters. Figure 11-4 shows a typical oil-mist lube system in schematic form.

Rolling-element bearings in centrifugal pumps are lubricated by one of two different mist application methods: purge mist or dry sump. Purge mist, or wet sump as it is sometimes called, involves the use of a conventional oil sump, with oil mist being

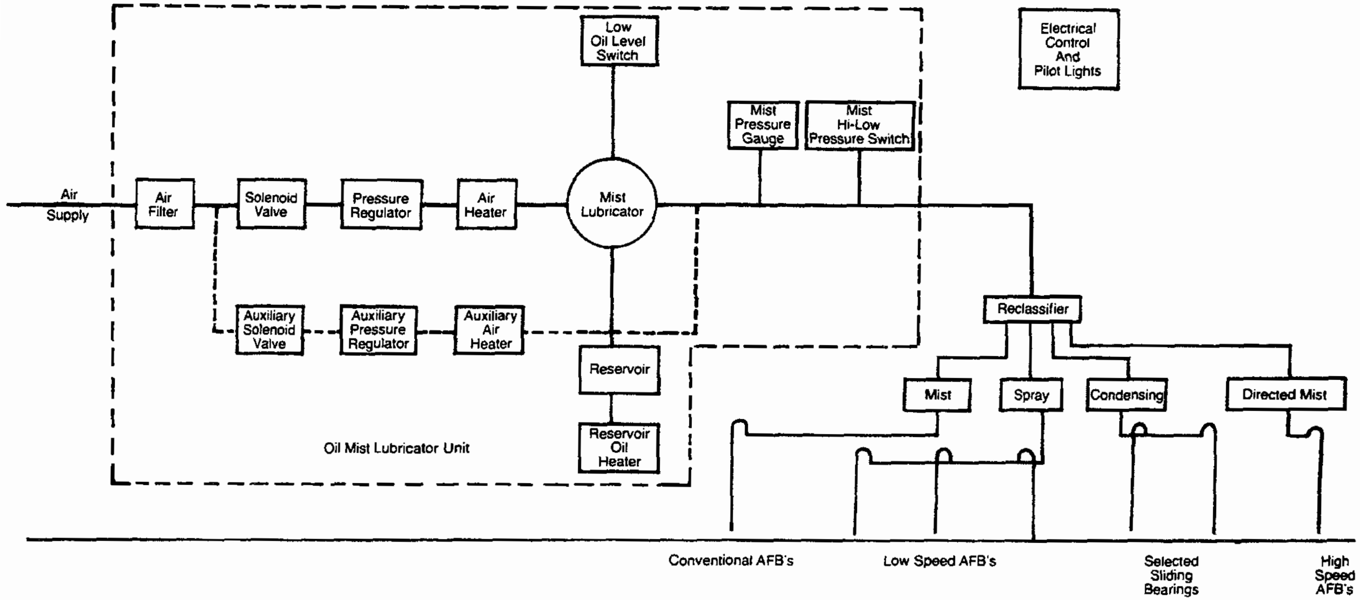

Figure 11-4. Schematic layout of oil-mist lube system.

used to purge the bearing housing and replenish nominal oil losses. When correctly applied, purge mist provides adequate lubrication if for any reason the oil level in the sump drops below the reach of the oil-ring, flinger, or lowermost ball of the bearing. By providing slightly higher than atmospheric pressure inside the bearing housing, purge mist effectively prevents the intrusion of ambient air and moisture. It does not, however, prevent oil-sump contamination resulting from oil-ring deterioration or loss of lube oil additives safeguarding against oxidation.

With dry-sump oil mist, the need for a lubricating-oil sump is eliminated. If the equipment shaft is arranged horizontally, the lower portion of the bearing outer race serves as a mini oil sump. The bearing is lubricated directly by a continuous supply of fresh oil condensation. Turbulence generated by bearing rotation causes oil particles suspended in the oil-mist stream to coalesce on the rolling elements as the mist passes through the bearings and exits to the atmosphere. This technique offers four principal advantages:

- Bearing wear particles are not recycled back through the bearing but washed off.
- The need for periodic oil changes is eliminated.
- Higher bearing operating temperatures are permitted if dry-sump oil-mist lubrication is used.
- By collecting mist condensate in a transparent pot located at the bottom of the now empty oil sump, oil discoloration can be seen at a glance. A snap fitting at the base of the transparent pot makes sampling for spectrometric analysis simple, and early trouble detection is thus facilitated. Due to low oil volumes, metals content will show up as higher ppm than in wet-sump systems.[6]

Contrary to a maintenance person's intuition, loss of mist to a pump or motor is not likely to cause an immediate and catastrophic bearing failure. Tests by various oil-mist users have proven that bearings operating within their load and temperature limits can continue to operate without problems for periods in excess of eight hours. Furthermore, experience with properly maintained oil-mist systems has demonstrated incredibly high service factors. Backup mist generator modules and supervisory instrumentation are available and can be made part of a well-engineered installation. In this context, "well-engineered" refers to a system which pays attention to such installation criteria as flow velocity in piping and optimum reclassifier nozzle configuration and location. Moderately and heavily loaded bearings may require directed classifiers. Unlike mist classifiers, directed classifiers generate a coarser spray which condenses easily. This requires the discharge end of the reclassifier to be within 1 inch (25 mm) of the bearing rotating element. If the bearing surface speed exceeds 2000 linear feet (610 m) per minute, the oil mist must offset windage from the rotating element. In this case, the reclassifier discharge end should be located within ⅛–¼ inch (3–6 mm) of the bearing surface. The flow of mist in lines must be laminar. This lessens the probability of oil droplets contacting one another to form large drops that fall out of suspension. It requires that the mist velocity be maintained below 20 feet per second (6.1 m per sec)—a factor that is easily overlooked in installations that make it a practice to place several feet of small-diameter tubing between reclassifier nozzle and bearing housing.

This installation practice may serve exceedingly well for services incorporating small bearings and thus demanding a low volume of mist. A large bearing will, of course, demand a larger volume of mist. Sizing the reclassifier to accommodate this demand is only one requirement. Forgetting to use larger diameter tubing may result in excessive mist velocity, causing large drops of oil to fall out of suspension and only relatively oil-free air to reach the rolling elements.

The relationship between droplet size, impingement velocity, and wetting ability has been quantified in Figure 11-5.[7] The larger the droplets, the more likely they are to wet out and form an oil film at low impingement velocities. A stable mist can be

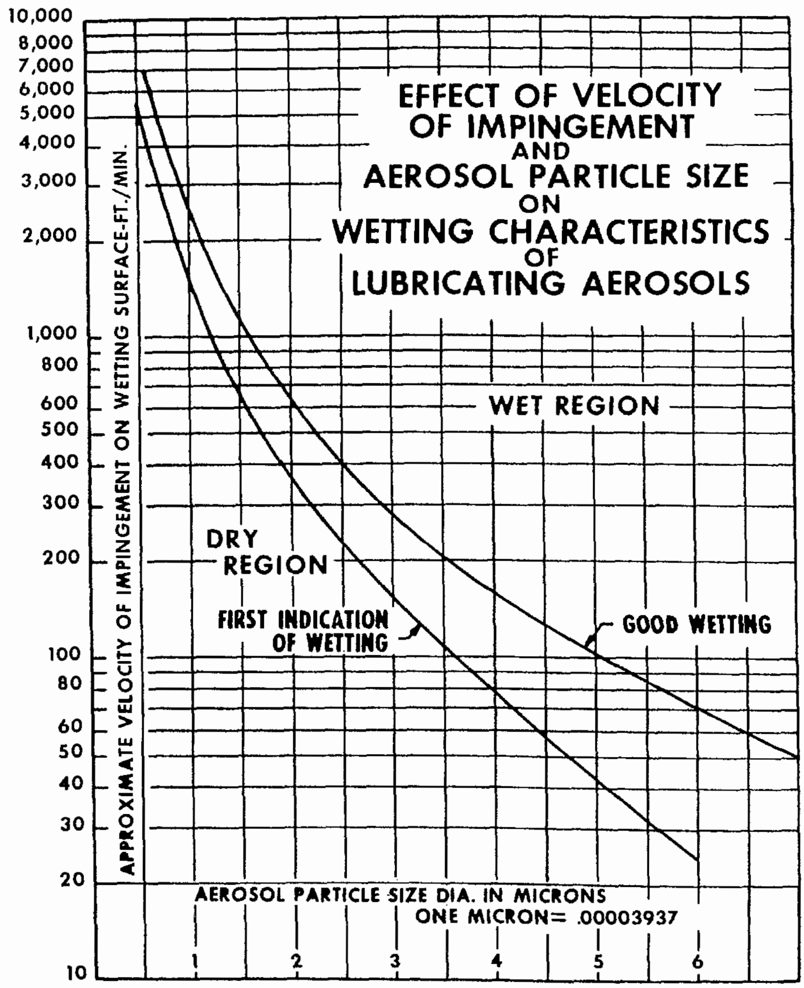

Figure 11-5. Effect of velocity of impingement and aerosol particle size on wetting characteristics of lubricating aerosols. (*Courtesy of C.A. Norgnen Co.*)

maintained and effectively transported in supply headers if the droplet size does not exceed 3 microns. Unless the rotating elements of antifriction bearings create relatively high impingement velocities, reclassifier nozzles must be used to coalesce this mist into larger droplets.

The size of the venturi throat or vortex generator, oil feed line, and pressure differentials imposes limits on oil viscosities that can be misted. However, oil heaters, or supply air heaters, can be utilized to lower the viscosity of heavier lube oils to the point where dependable misting is possible. Systems without oil heaters are generally limited to 1000 SUS at 100°F (216 cSt at 38°C) when operating in 70°F (21°C) environments. If ambient temperatures drop below 70°F (21°C) or if the viscosity of the oil exceeds 1000 SUS 100°F (216 cSt at 38°C), heating should be used to reduce the effective viscosity of the oil and to make the formation of a stable mist possible. However, oils from 1000 to 5000 SUS at 100°F are now successfully misted in many applications. The use of air heaters is encouraged regardless of oil viscosity.

Properly engineered *dry-sump* oil-mist systems have proven so reliable and successful that grass-roots ethylene plants in the 500,000+ metric ton/year category rely entirely on this lubrication method for their often critical and sophisticated centrifugal pumps. Dry-sump oil mist, properly applied, will discharge virtually no spray mist into the atmosphere. Closed systems are responsible for this achievement. Total oil consumption is generally only 40% or 50% of oil used in conventional lubrication.

Finally, feedback from petrochemical units using dry-sump oil-mist lubrication showed them to experience far fewer bearing problems than similar units adhering to conventional lubrication methods. Failure reductions of 75% seem to be the rule and have been documented. Larger reductions have sometimes been achieved.

Oil Mist Proven for Motor Lubrication

Since the mid-1970s, oil mist has also demonstrated its superiority for lubricating and preserving electric motor bearings. By then, petrochemical plants in the U.S. Gulf Coast area, the Caribbean and South America had converted in excess of 1,000 electric motors to dry-sump oil-mist lubrication. In 1986, there were more than 4,000 electric motors on oil-mist lube in the U.S. Gulf Coast area alone.

However, universal acceptance did not come overnight. And to this day, we hear occasional questions relating to such issues as oil intrusion and explosion hazard.

Today's epoxy motor winding materials will not deteriorate in an oil-mist atmosphere. This has been conclusively proven in tests by several manufacturers. Windings coated with epoxy varnish were placed in beakers filled with various types of mineral oils and synthetic lubricants. Next, they were oven-aged at 170°C (338°F) for several weeks, and then cooled and inspected.

Final proof was obtained during inadvertent periods of severe lube oil intrusion. In one such case, a conventional oil-lubricated, 3,000 hp, (~2,200 kW), 13.8 kV motor ran well even after oil was literally drained from its interior. The incident caused some increase in dirt collection, but did not adversely affect winding quality.

Regarding explosion hazards with oil mist in plant-wide systems, the potential for such occurrences was investigated and confirmation obtained that the oil/air mixture was substantially below the sustainable burning point. Experiments had shown the concentration of oil mist in the main manifold ranging from 0.005 to as little as 0.001 of the concentration generally considered flammable. The fire or explosion hazard of oil-mist lubricated motors is thus not different from that of NEMA-II motors. No signs of overheating were found, and winding resistance readings conformed fully to the initial as-installed values.

Converting from Grease Lube to Oil-Mist Lube. Conversion to dry-sump oil-mist lubrication does not necessarily require that the motor be removed and sent to the shop. Motors with regreasable bearings are easiest to convert because they generally don't incorporate oil-rings or bearing shields. Most oil-lubricated bearings can be modified for dry-sump lubrication by adding only the piped oil-mist inlet, vent and overflow drain passages. Oil rings must be removed because there is, of course, no longer an oil sump from which oil is to be fed to the bearing. Figure 11-6 shows the bearing shields removed to establish unimpeded passage from the oil-mist inlet pipe through the bearing rotating elements and finally the vent pipe to atmosphere or collection header. However, ample experience shows that the inboard bearing shield need not be removed to ensure a successful installation.

Our reference[8] describes a petrochemical plant area with a series of vertical motors. One such motor, rated 125 hp/3,560 rpm, experienced frequent thrust bearing failures with conventional oil lubrication. Installing dry-sump oil mist solved the chronic lubrication problem. Bearing housing temperatures were lowered from 160°F (71°C) to 110°F (43°C) after conversion to dry-sump lubrication.

A properly installed and maintained oil-mist lubrication system will result in a high percentage reduction in bearing failures. It must be noted, however, that such bearing failure reductions will not be achieved if the basic bearing failure problem is

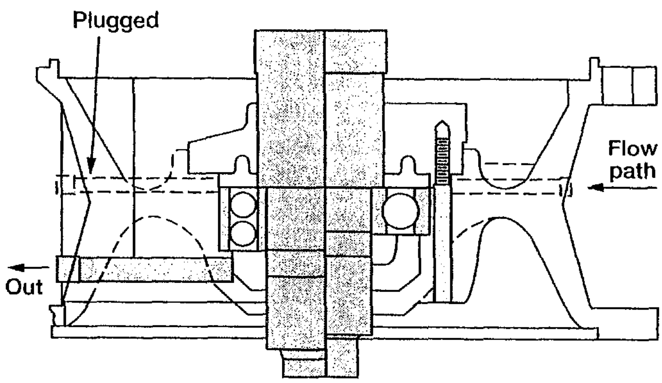

Figure 11-6. Vertical electric motor bearings with both shields removed to promote unimpeded passage from the oil-mist inlet pipe through the rotating elements to vent pipe and atmosphere or collection header.

not lubrication related. Oil-mist cannot eliminate problems caused by defective bearings, incorrect bearing installation, excessive misalignment or incorrect mounting clearances.

Nevertheless, oil-mist excels as a preservative and protective "blanket," preventing ingress of atmospheric contaminants into standby equipment. Bearing friction losses are kept low, and with through-flow oil-mist lubrication,[8] electric motor bearings tend to run considerably cooler than with grease or oil ring lubrication.

Sealing Against External Contamination

It is a well-documented fact that the overwhelming majority of rolling element ("anti-friction") bearings used in industrial machinery fail due to lubricant contamination originating from dirt, dust, water or airborne vapor condensate. Major bearing manufacturers have estimated that only 9% of the millions of bearings replaced every year in the U.S. alone are "normal wear-out" failures.

The situation is even worse in centrifugal pumps that, next to electric motors, are assumed to be the most widely used machines on earth. Approximately one-third of all pump failures involve premature distress of rolling element bearings. Contaminants such as airborne dust and water vapor or sand and water from cleaning operations are the main culprits.

To ward off contaminant intrusion into bearing housings, many machines incorporate lip seals, labyrinth seals, noncontacting rotor-stator seals, or contacting face seals of one type or another. But, while any one of these different seal configurations is certainly better than no seal at all, each has certain features that merit closer investigation before a cost-effective selection can be made.

A large number of machines are equipped with elastomeric lip seals; centrifugal pumps built to ANSI B73 standard dimensions are strongly represented in any tabulation of lip seal users.

However, lip seals tend to generate a fair amount of heat and shaft wear while they are tight-fitting. Typical mean lives have been reported in the 2,000 to 3,000 operating hour category when operating conditions were close to ideal.

Rotating labyrinth seals (Figure 11-7) are a distinct improvement. They do not, however, qualify as hermetic seals and should be applied on grease, but not oil-lubricated equipment.

Where conventional seals use either springs or elastomeric lips to apply sealing force, magnetic seals (Figure 11-8) use magnetism. When no fluid pressure exists, magnetic force holds the two sealing surfaces tightly together. This force minimizes friction between sealing faces while ensuring proper alignment of surfaces through equal distribution of pressure. Magnetic face seals are hermetic seals in the true sense of the word. Many thousands of them have been used in military and commercial aircraft since 1948.

It should be noted that only the three-piece *repulsion type* magnetic seal shown in Figure 11-8 incorporates the beneficial attribute of reduced face loading if wear of seal faces should ever take place. In two-piece magnetic seals, magnetic *attraction*

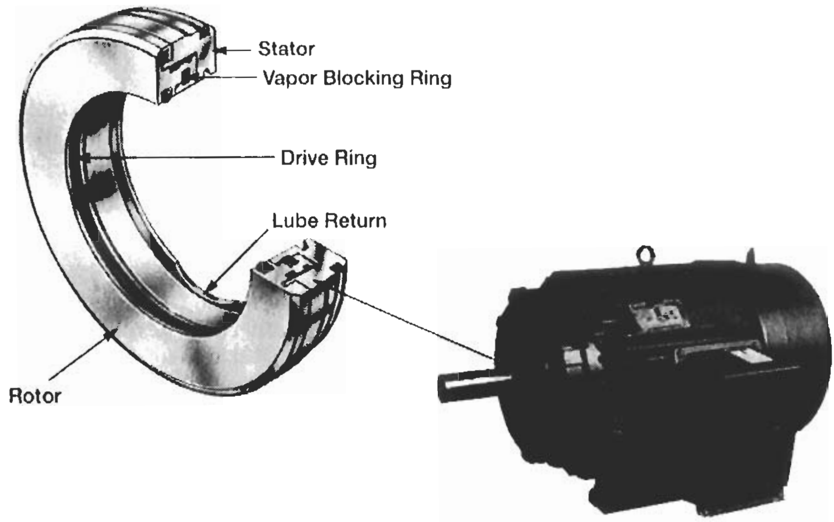

Figure 11-7. Rotating labyrinth seal suitable for grease-lubricated gearings. (*Courtesy of INPRO Companies, Rock Island, Illinois.*)

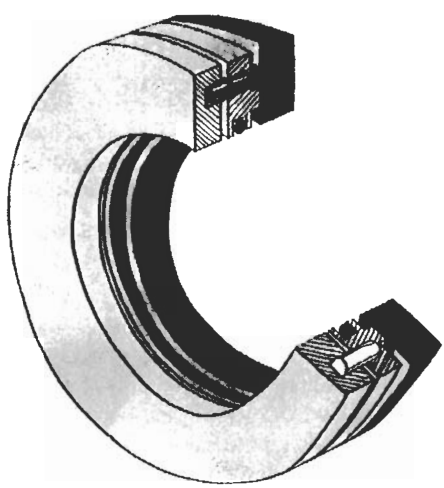

Figure 11-8. RMS-700 Repulsion Magnetic Seal. (*Courtesy of INPRO Companies, Rock Island, Illinois.*)

will increase as the seal faces wear. The resulting load increase causes wear rates to increase exponentially.

Magnetic seals have often been in continuous service for several years. Since they are customarily used in oil-lubricated services, the ever-present oil fog environment provides ample lubrication and, thus, exceedingly low rates of wear.

Using worn lip or labyrinth-type bearing housing seals without paying attention to oil contamination has been shown to give HPI plants typical bearing life expectancies of only about 2.5 years. With bearing replacements assumed to cost $3,000 per pump, plants applying hermetically sealed bearing housings can expect bearings to last an average of six years under identical operating conditions. For a petrochemical plant with 2,000 installed pumps, the value of avoided pump bearing failures exceeds $1,000,000 each year. Needless to say, the use of hermetic bearing housing sealing devices is eminently justified and should be advocated on an attrition basis. In other words, when oil-lubricated pumps go to the shop with bearing failures, they should be retrofitted with hermetic bearing housing seals of the type illustrated in Figure 11-8, since the configuration shown in Figure 11-7 cannot really perform as a "hermetic" seal.

Figure 11-9 shows a typical oil-mist console, with Figure 11-10 giving details of downstream piping. A closeup of a pump bearing housing is shown in Figure 11-11.

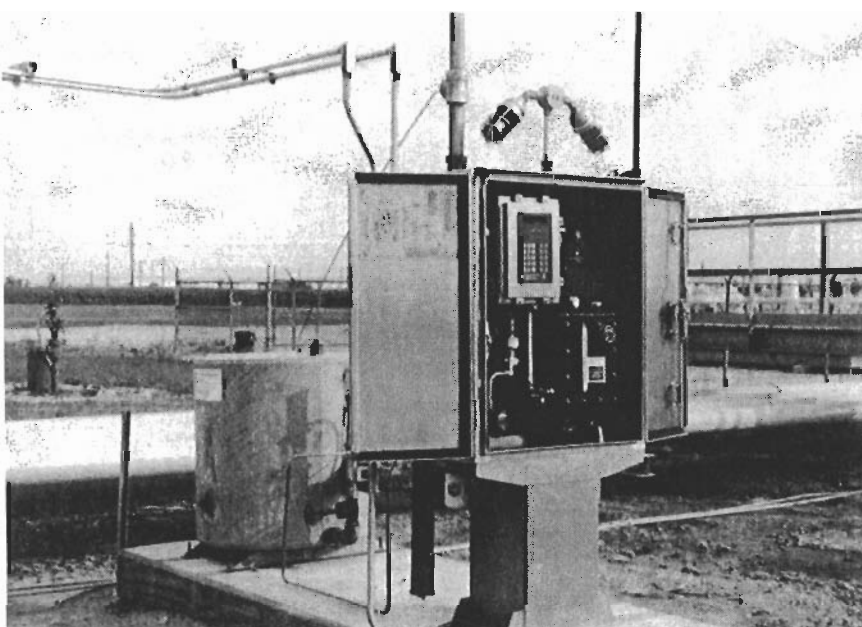

Figure 11-9. Typical oil-mist console. (*Courtesy of Lubrication Systems Company, Houston, Texas.*)

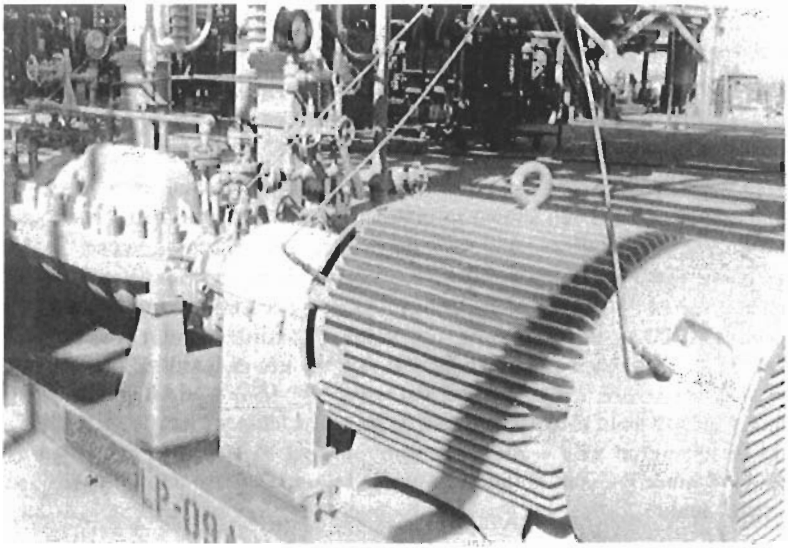

Figure 11-10. Downstream piping details.

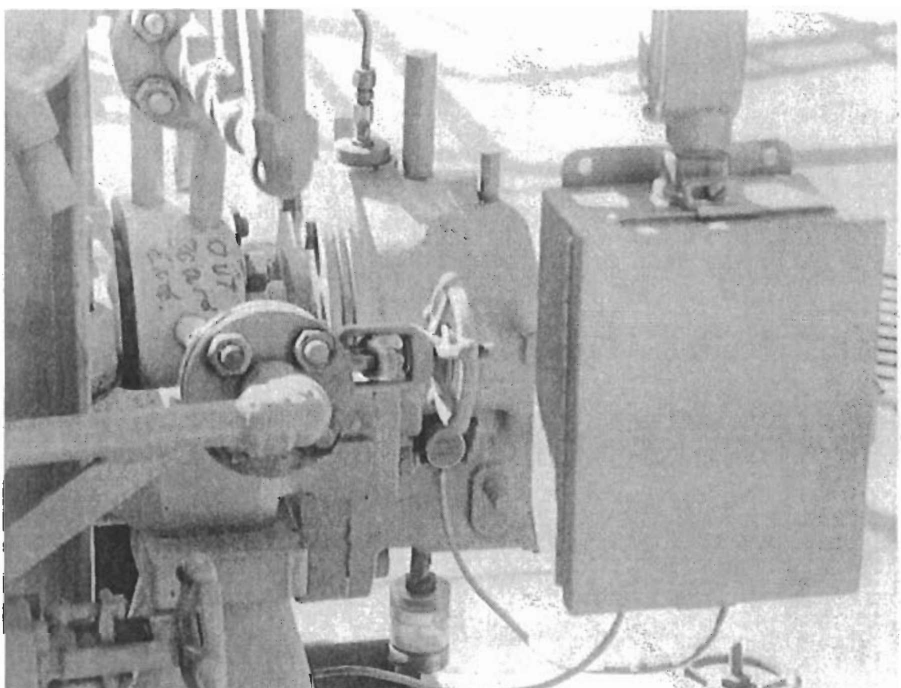

Figure 11-11. Pump bearing housing.

Gear Couplings Versus Non-lubricated Couplings

For years, gear-type couplings (Figure 11-12) have been the petrochemical industry standard for centrifugal pump-driver connections. These couplings excel through their simplicity; they consist of three fundamental components only—two hubs, mounted on the coupled shafts, and one floating member. The hubs may have external or internal teeth; the mating member embodies the reverse design. Fully crowned teeth, (Figure 11-13) are preferred. Gear-tooth lubrication is generally provided by way of grease or heavy lube oil which is retained in the gear mesh region by O-rings or similar sealing means.

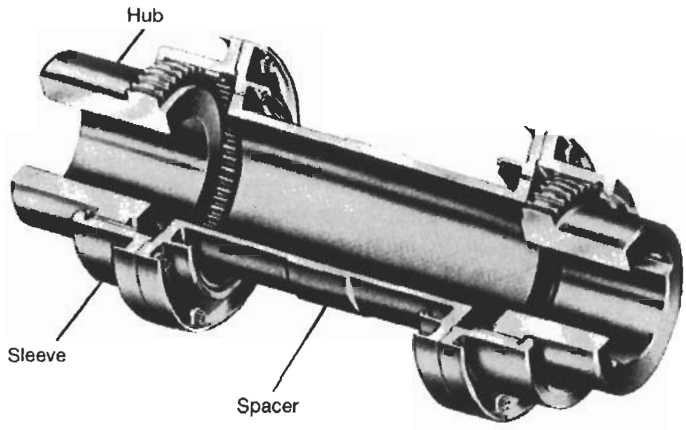

Figure 11-12. Gear-type spacer coupling. (*Courtesy of Ameridrives International, Erie, Pennsylvania.*)

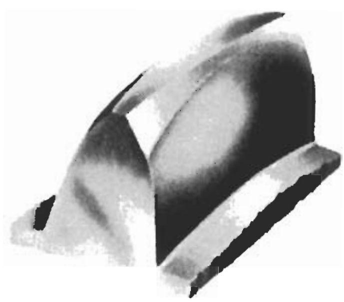

Figure 11-13. Fully crowned coupling tooth. (*Courtesy of Ameridrives International, Erie, Pennsylvania.*)

However, the applicability of gear couplings to the customary wide spectrum of centrifugal pump services deserves closer scrutiny.[9] Advances in mechanical-seal designs, more reliable bearing lubrication, and the infusion of qualified design follow-up and pump troubleshooting talent have all contributed to pushing the pump mean time between failures toward and frequently beyond the three-year range. Gear-coupling servicing requirements were often found to preclude these extended run lengths. Also, component failure analysis applied to general-purpose equipment resulted in the "rediscovery" of a key phenomenon which can work against gear couplings: the potential transmission of very substantial axial forces and moment loads to the connected shafts. This happens whenever the coefficient of friction between mating teeth undergoes a significant increase. The problem is amplified by excessive misalignment, coupling unbalance, and inadequate lubrication.

No Coupling Is Perfect, But Non-lubricated Construction Is Preferred Choice

One type of non-lubricated coupling takes its name from a diaphragm-shaped metallic flexing element. Diaphragm couplings exhibit a number of highly advantageous characteristics which account for their rapid acceptance in high-speed, high-power turbomachinery applications. Unlike most other coupling types, they are generally designed to operate indefinitely at the manufacturer's stated maximum allowable misalignment angle. A multi-convoluted diaphragm coupling is depicted in Figure 11-14, while Figure 11-15 shows a standard industrial diaphragm-type cou-

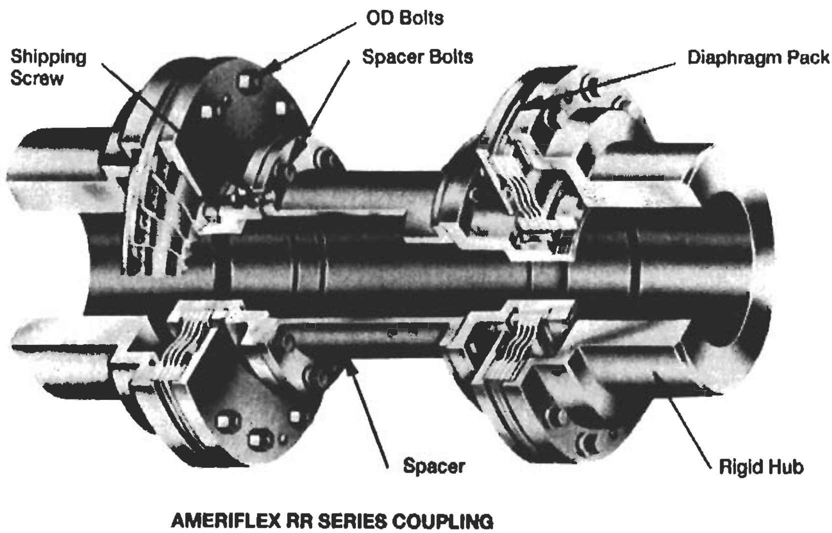

Figure 11-14. Multi-convoluted diaphragm coupling. (*Courtesy of Ameridrives International, Erie, Pennsylvania.*)

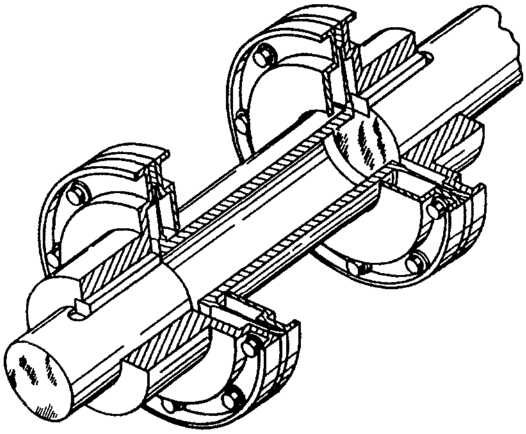

Figure 11-15. Typical diaphragm coupling. *(Courtesy of Bendix Division of Lucas Aerospace, Utica, New York.)*

pling cut-away view. Figure 11-16 illustrates the cross-section of a similar coupling with emergency gear coupling backup.

A generally less-expensive non-lubricated flexible coupling employs sets of stainless steel membranes, or disc packs, to transmit driving torque. Although not quite on a par with diaphragm couplings, disc-pack couplings will probably serve exceptionally well in more than 90% of centrifugal pump services. While not totally immune to mistreatment and misapplication, they have been shown to overcome many of the problems which can beset gear couplings. The principal advantages of metallic disc-packs over conventional gear couplings are that:

- No lubrication is required.
- They transmit low, known thrust forces to connected equipment shafts.

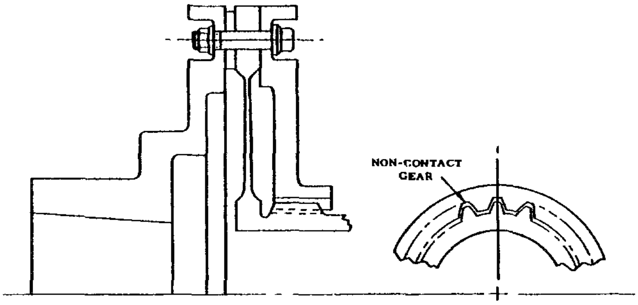

Figure 11-16. Diaphragm coupling with backup gear (non-contacting). *(Courtesy Bendix Div. of Lucas Aerospace, Utica, N.Y.)*

- Reaction forces and moments at bearings are kept low.
- They can be designed for infinite life.
- Better balance is usually maintained.

The metal diaphragm coupling transmits power from driving to driven machine by use of a solid rigid hub on each machine, a flexible metal diaphragm bolted to the outer rim of each hub, and a spacer connecting the two diaphragms. All misalignment, both angular and axial, is accommodated by elastic flexure of the thin web of the diaphragm, as shown in Figure 11-17. There is no relative (or "rubbing") movement of any parts of the coupling.

The dynamic behavior and component properties of diaphragm couplings can be predicted with the highest degree of accuracy. Consequently, critical pumping services will often provide justification to select the more costly diaphragm coupling over other designs.

Rating Basis Should be Reviewed

There is an abundance of literature on disc-pack-type general-purpose couplings, and many manufacturers have issued detailed and well-presented documentation for their products. Unfortunately, they sometimes contain "fine print" or neglect completely to point out the rating basis for their products.[9]

Just as most gear couplings, if operating at the quoted maximum allowable misalignment angle, will risk greatly accelerated tooth wear or the transmission of excessive loads to coupled equipment, disc-pack coupling life will be influenced by operation at certain misalignment angles. More often than not, the manufacturer will list maximum misalignment angles because they are dimensionally possible to achieve without causing mechanical interference. But, the catalog rating may have been on the assumption that the user will operate the equipment at parallel or angular offsets *well below* those listed as maximum allowable. Operation at maximum allowable offsets is likely to cause fatigue failure of the flexing members. It is therefore advisable to obtain a full understanding of the rating basis of any coupling.

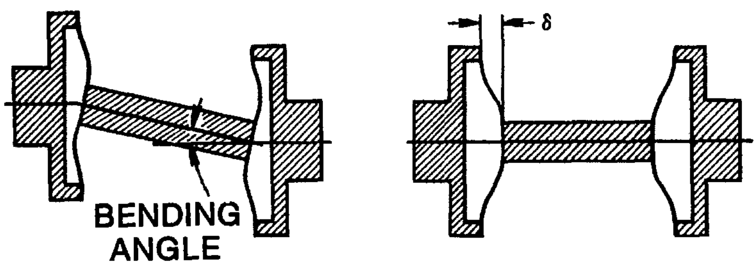

Figure 11-17. Diaphragm couplings in flexing modes. (*Courtesy of Kopflex Coupling Corporation, Baltimore, Maryland*)

Safety Features May Differ Among Disc-Pack Designs

Metallic disk-pack couplings are manufactured by several knowledgeable companies in the U.S. and overseas. These manufacturers have experience in selecting suitable materials for such critical elements as flexible membranes or coupling hubs in high-speed applications. They understand the relevance of engineering parameters relating to torsional critical speeds, bearing loads, and intermittent shock loading. Nevertheless, desirable safety or ease of maintenance features may well make the difference between reliable, low-maintenance operation at one plant, and less reliable, maintenance-intensive operation at another. Two of these features, spacer removal and spacer retention, merit closer examination.

- It should be possible to remove coupling spacers without disturbing the disc pack. Dismantling the disc packs is not only time consuming but also very risky. When flexible disc packs are assembled at the factory, careful attention is given to the placement of adjacent discs. They must be assembled flat and without buckling. Moreover, they may require grain orientation in random direction so as to ensure near-uniform spring rate throughout a full turn of the shaft.

  To preclude incorrect disassembly and reassembly of disc packs, coupling construction should be as shown in Figures 11-18 and 11-19. Unlike some slightly less expensive three-part models, the couplings shown in Figures 11-18 and 11-19 consist of a five-part spacer element incorporating a torque tube (1), two disc packs (2), and two nesting adaptors (3) which are bolted into coupling hubs (4). The entire five-part spacer element can be removed or inserted between coupling hubs without disturbing the disc packs.

- Couplings should incorporate provisions preventing the spacer from becoming a flying missile in the event of disc-pack failure. Often one side will break before the other, causing the resulting unbalanced overhung load to take off. After bending an equipment shaft and wrecking seals and bearings, hubs or spacers have sometimes torn coupling guards off their base mounts and become airborne. Floating piloted spacer arrangements will confine the torque tube at location "P" (Figure 11-18) without allowing it to contact the nesting adaptor (3). The coupling components shown in Figures 11-18 and 11-19 are dimensioned to prevent this contact as long as the disc packs remain serviceable. Should disc packs fail, an elastomeric ring (5) as shown inserted into the nesting adaptor of Figure 11-18 can be used to limit torque-tube misalignment and thus avoid vibration caused by extreme unbalance.

The coupling shown in Figure 11-19 maintains reasonably close concentricity after failure of a disc pack by providing "overload bushings" (6). Machined to normally avoid contact with their respective mating holes, these bushings will act as emergency drive lugs after such failures.

In at least one documented event, failure of a disc pack still allowed a centrifugal pump to run for two more days without danger of flinging out the torque tube. Equipment rotation continued by engagement of bolt heads or "overload bushings" with counterbored holes in the nesting adaptor. Although failure was evident from the increased noise, there was no damage to shafts, bearings, or seals. Conversely,

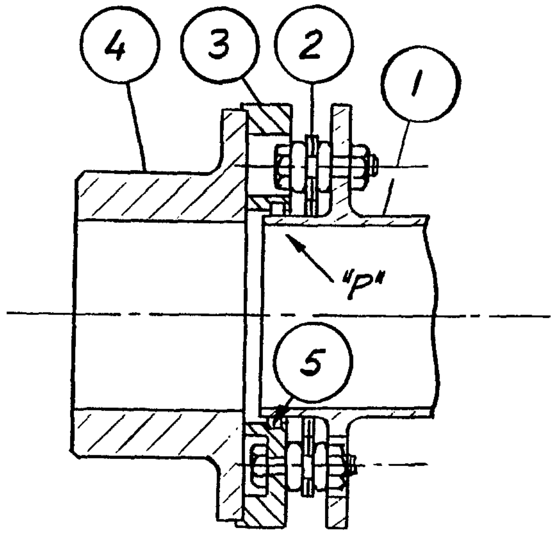

Figure 11-18. Disc pack coupling with "captured" center member nested at "P." (*Courtesy of Thomas Coupling Co., Warren, Pennsylvania.*)

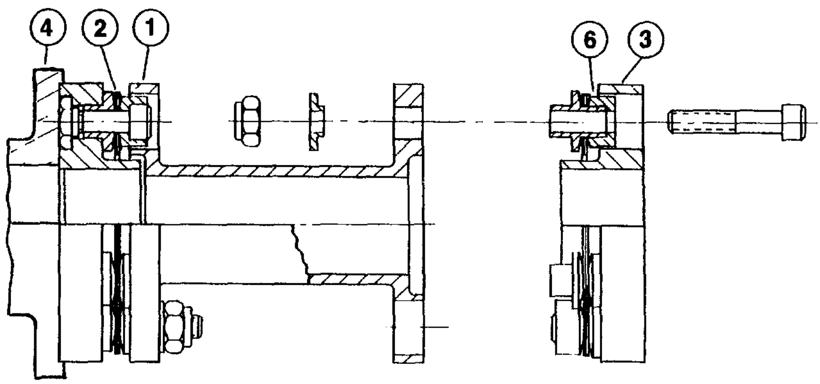

Figure 11-19. Disc pack coupling with "captured" center member and overload bushings. (*Courtesy of Flexibox Inc., Houston, Texas.*)

the disc pack couplings *without* confined torque tubes have been involved in serious incidents causing damage to equipment and potentially serious injury to personnel.

Failure Records Are Revealing

The importance of proper alignment between pump and driver is best demonstrated by the failure statistics of several refinery units. The replacement rates for metallic disc pack couplings of the type shown in Figures 11-18 and 11-19 are generally fewer than 3 per 100 pumps per year. When a seven-fold replacement rate at another unit prompted an investigation, unacceptably high piping loads were found to be at fault. Coupling failures occurred most often on the "A" pump after a parallel "B" pump had been removed for shop repairs and was subsequently reinstalled at its site location. Removal of the "B" pump had resulted in piping disturbances which placed undue loads on the operating pump and adversely influenced its alignment.

These failure incidents illustrate the importance of wide separation between pumps and drivers. Spacer-type couplings will accommodate far more misalignment than will close couplings without spacers. They are not an API-pump requirement, but are well worth the small additional investment on lower cost ANSI-pump installations as well.

Another study compared the replacement rate for metallic disc packs not requiring dismantling with that of metallic disc packs requiring disassembly. It was found that for every replacement of disc packs incorporating the features of Figures 11-18 and 11-19, *four* disc pack replacements were needed with couplings requiring disc pack disassembly.

Elastomeric Couplings

Elastomeric couplings are often considered by process plants personnel because they are low in cost and are thought to have high misalignment capability. Some elastomeric couplings are configured as illustrated in Figure 11-20; the design intent is for operation in compression. Others allow the elastomer to be severely twisted while operating in tension (Figure 11-21). Figure 11-20 shows that some of these couplings do not incorporate means to positively secure a metal ring, item 4. In fact, at least one manufacturer explained that operation with excessive misalignment will cause this ring to slide off, allowing the flexible element, item 3, to be flung out. The manufacturer, quite correctly, admonishes the user/purchaser to be certain to have the coupling guard in place before starting the machine. When this advice was overlooked, a worker was struck in the eye and became permanently disabled.

And are you aware of the relative performance of some of the elastomers commonly used in some coupling models? Table 11-1 may offer some help, but some forethought is recommended, nevertheless. It would seem that coupling selection based on price alone may be inappropriate in a safety and reliability-conscious process plant environment. We have the failure and injury statistics to prove it.

Finally, Table 11-2 gives a general performance comparison of industrial flexible couplings. Remember that operation with *any* misalignment will impose an additional load on the bearing system of virtually any process machine. Quoted misalignment capacities may reflect coupling capabilities that may affect bearing and rotor system capabilities or life expectancies.

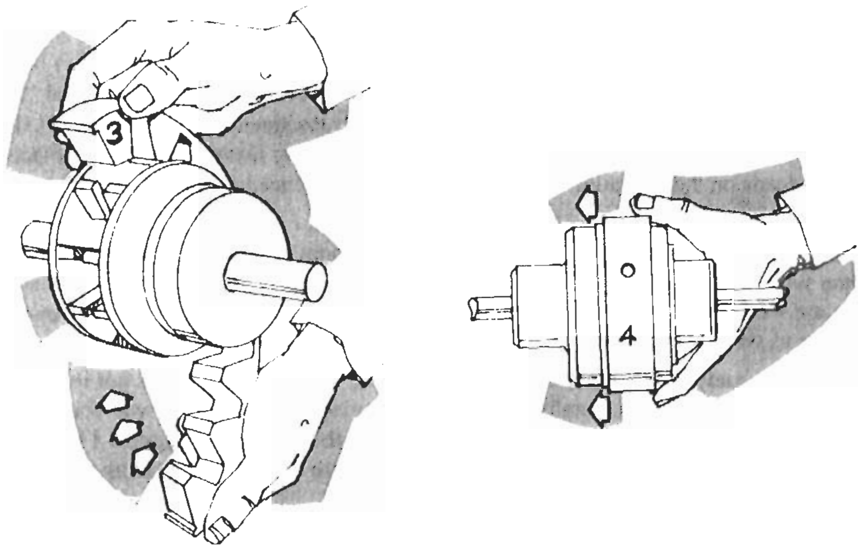

Figure 11-20. Elastomeric coupling designed for operation in compression. ***(Courtesy of Atra-Flex® Inc., Santa Ana, California.)***

Figure 11-21. Elastomeric coupling designed for operation in tension. ***(Courtesy of Rexnord Corporation, New Berlin, Wisconsin.)***

Table 11-1
Relative Performance of Two Commonly Used Coupling Elastomers

Environment	Relative performance of: Polyurethane	Rubber (Polyisoprene)
Abrasion	Excellent	Excellent
Acids—dilute	Fair	Good
Acids—concentrated	Poor	Good
Alcohols	Fair	Good
Aliphatic hydrocarbons	Excellent	Poor
Gasoline, fuel	Excellent	Poor
Alkalies—dilute	Fair	Good
Alkalies—concentrated	Poor	Good
Animal and vegetable oils	Excellent	Fair
Aromatic hydrocarbons	Excellent	Poor
Benzol, toluene	Poor	Poor
Degreaser fluids	Good	Poor
Heat aging	Good	Good
Hydraulic fluids	Poor	Poor
Low temperature embrittlement	Excellent	Good
Oil	Excellent	Poor
Oxidation	Excellent	Good
Ozone	Excellent	Poor
Radiation	Good	Good
Silicate and phosphate	Poor	Poor
Steam—hot water	Poor	Unknown
Sunlight aging	Excellent	Poor
Synthetic lubricants	Poor	Poor
Water swell	Excellent	Good

Source: Rexnord Corporation, New Berlin, Wisconsin

Summary

1. Non-lubricated coupling construction is to be preferred over lubricated gear couplings in most centrifugal pump-driver applications. Unless the critical nature of a given pumping service justifies diaphragm couplings, flexible disc pack couplings should be used.
2. Some flexible disc pack couplings are capable of sustained operation at stated maximum permissible misalignment angles only at the risk of decreased life.
3. Wider separation between pump and driver is an effective way of accommodating misalignment and reducing risk of disc pack fatigue failure.
4. Highly significant safety and reliability credits may accrue from selecting disc pack couplings incorporating two principal features:
   - A piloted spacer arrangement that confines the spacer to the coupling hub area in case of disc pack failure.
   - An arrangement allowing the machinery to be uncoupled without loosening the bolts that keep the disc pack together.
5. Elastomeric couplings should be used with extreme caution on high-speed equipment. They are appropriate for low-speed, high-torque, and highly damped applications, but require review of the elastomer selection.

Table 11-2
Performance Comparison of Industrial Flexible Couplings

	Mechanically Flexible						Elastomeric		Metallic Membrane			
Function	**Gear Straight**	**Gear Crowned**	**High Speed**	**Chain**	**Grid**	**Universal Joint②**	**Shear**	**Compression**	**Laminated Disc**	**Laminated Diaphragm**	**Convoluted Diaphragm**	**Tapered Diaphragm**
Maximum continuous Torque (kN-m)	4500	5000	1500	200	450	3000	60	1700	1000	150	700	700
Maximum speed (rpm)	12000	14500	40000	6500	4000	8000	5000	8000	30000	24000	30000	30000
Maximum bore (mm)	900	900	300	250	500	500	250	600	475	300	300	300
Angular misalignment (°)	0.5	1.5	0.25	2	0.3	20	3	3	0.5	0.5	0.5	0.5
Parallel offset ① mm/mm mm total	.008	.026	.004	- 1.0	- 0.2	0.34	- 1.5	- 1.0	.008	.008	.008	.008
Axial travel	very high	very high	high	moderate	moderate	zero	moderate	low	moderate	moderate	high	moderate
Lubrication required	yes	yes	yes	yes	yes	yes	no	no	no	no	no	no
Backlash	high	medium	low	high	medium	none	none	low	none	none	none	none
In-service balance	fair	fair	good	poor	poor	poor	poor	poor	very good	very good	very good	very good
Bending moment	high	high	high	high	medium	high	low	medium	low	low	low	medium
Axial restoring force	high	high	high	high	medium	very high	low	medium	low	low	low	medium
Torsional stiffness	high	high	high	high	medium	high	low	medium	high	high	high	high
Damping	low	low	low	medium	high	low	very high	high	medium	medium	low	low

Notes: 1) Parallel offset is quoted as mm per mm of flexible element separation for designs which require a double engagement (spacer) design to accommodate lateral misalignment. Other designs which can accommodate lateral misalignment within a single flexing element have a typical value quoted for parallel offset.

2) The data quoted for the universal joint is for an unmodified design. The inclusion of a splined feature will change criteria such as axial travel.

Source: World Pumps, December 1996

Quantifying the Reliability Impact of Laser Alignment Techniques*

The need for proper alignment between different machine cases of a machine train has been well established. Misalignment has been determined to be one of the more frequently occurring malfunction conditions, especially in installation of process compressor trains and gas-turbine-driven-pump machine trains. The normal alignment procedures generally allow for the determination of an alignment drawing through calculations, using the growth of the various machines involved. This has often been shown to be a "guesstimate" at best, and some means of determining the final, hot running condition of the machine has been proven to be desirable and necessary.

Considerable progress has been made in the development of devices that will facilitate and speed up making alignment corrections, once their magnitude is known. Figure 11-22 shows one such element (Vibracon SM), which comes with a spherical top section capable of accommodating up to 4° between the bed plate and the machine foot. Figure 11-23 illustrates typical field installations.

While adjustable chocks will obviously cost more than a set of shims, they should be considered typical of the numerous small improvements which, when summed up, make the difference between a long-term reliable, maintainable, and "surveill-

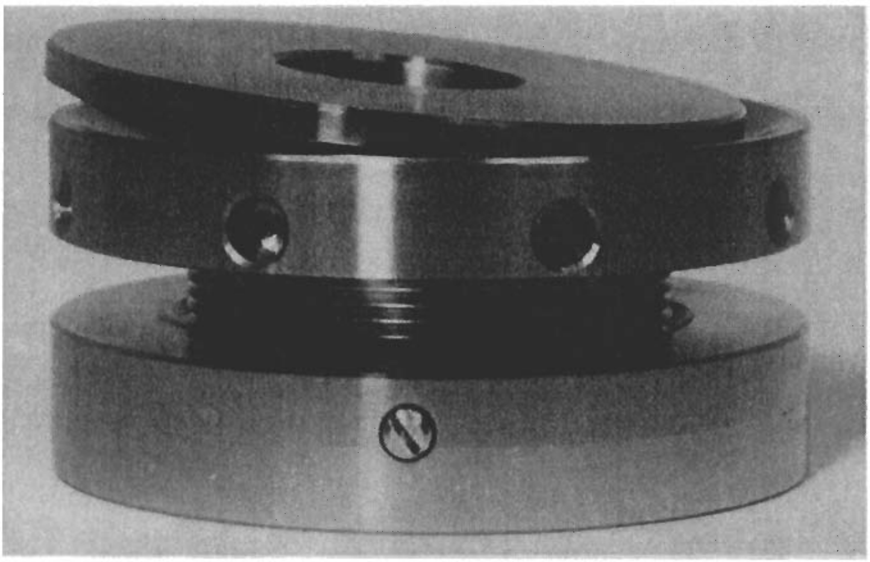

Figure 11-22. Adjustable chock. (*Courtesy of Machine Support, B. V., Zoeterwoude, The Netherlands.*)

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*Source: Galen Evans, Ludeca, Inc., Miami, Florida. Based on a presentation at the 5th International Process Plant Reliability Conference, Houston, Texas, 1996.

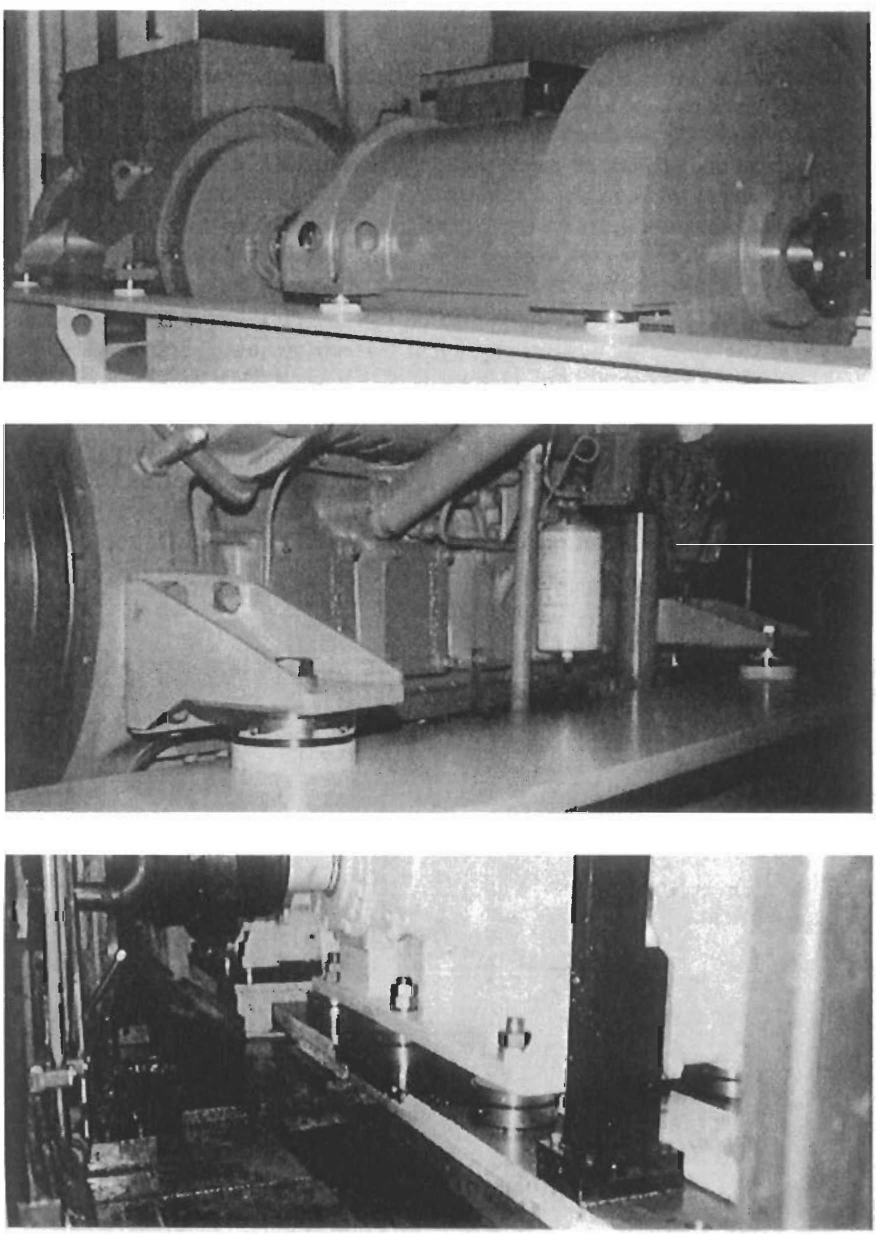

Figure 11-23. Machinery installations using adjustable chocks. (*Courtesy of Machine Support, B. V., Zoeterwoude, The Netherlands.*)

able" installation on the one hand and a more vulnerable "business-as-usual" installation on the other hand.

The tracking of operating alignment accuracy between major machinery drivers and their respective connected (driven) machines is also an indication of how reliability concepts are implemented at a given facility. Considerable work has been done in the area of hot alignment measurements. The more popular techniques include optical measuring techniques, using optical instruments similar to those used for surveying land, and the proximity reference system. The proximity probe has been used to determine the relative position changes between shafts of different machine cases. Until the advent of modern laser optic techniques described earlier in this text, the most popular techniques used proximity probes and were called the "Jackson," "Dodd Bar," and "Indikon" techniques.

It should be noted, however, that proper *initial* shaft alignment is necessary in any case and that laser optic techniques are clearly among the most advantageous techniques because they are both fast and precise. As shown in Figure 11-24, measurements are taken using a proven type of emitter system and prism, which are fastened to the shafts or coupling halves of the machines to be aligned facing one another using universal quick-fit brackets.

A semiconductor laser (1) in the emitter system produces a laser beam that is collimated by the lens system (2). As the beam is aimed toward the prism (3), mounted

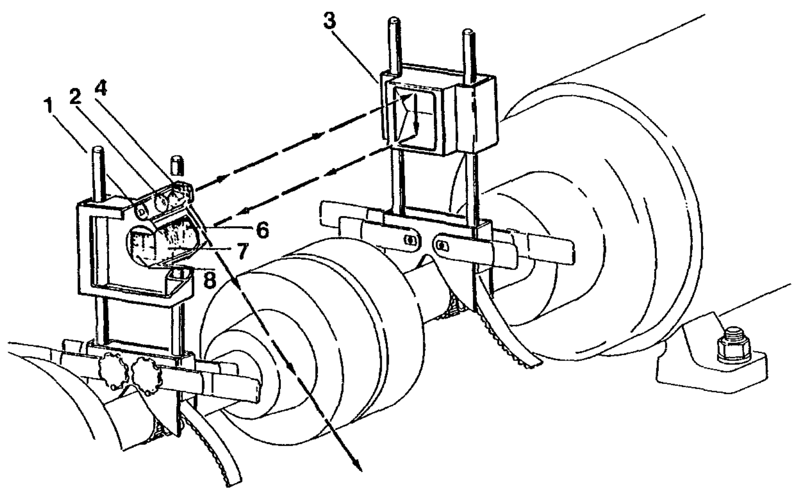

Figure 11-24. Principle of operation of laser optic alignment devices. (*Courtesy of Prueftechnik Dieter Busch, D-85730 Ismaning, Germany.*)

across from the emitter), it passes through a beam splitter (4). The prism reflects the beam into the receiver opening.

A filter (6) and an additional lens system (7) project the beam onto a photoelectronic position detector (8), which determines the location of incidence of the laser beam in terms of X and Y components.

As the shafts are then rotated together (or separately), the parallel dislocation of the laser beam causes a deviation in the Y direction, and angularity results in a shift along the X axis from the original starting point.

A portion of the beam is directed onto the ground by the beam splitter (4) to serve as a geometric ground marking for exact determination of distance measurements.

The beam deviation detected by the emitter system is automatically transferred to the alignment computer via cable. The computer thereby guides the operator along in dialog fashion. LCD graphics show the display schematically, and the built-in digital display assumes the function of dial indicators.

Figure 11-25 shows the laser optic devices in use on a close-coupled pump.

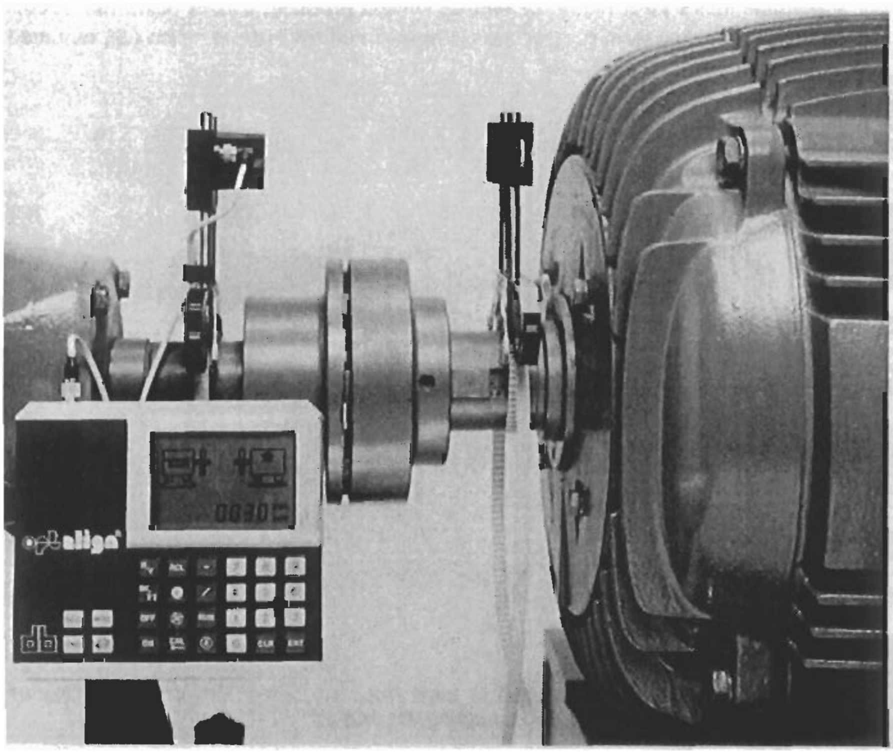

Figure 11-25. Laser optic alignment in progress. (*Courtesy of Prueftechnik Dieter Busch, D-85730 Ismaning, Germany.*)

How Precise?

For the purposes of quantifying reliability, the real issue is not what kind of system was used to perform alignment. The issue is, "How accurately are the alignments performed?" To wit, with what precision and with what certainty is the alignment system used? Or, to what tolerances are the alignments performed and how certain are the measurements? Table 11-3 shows one common tolerance scheme based on speed range.

This tolerance chart is typical of those espoused for precision alignment. Notice that at 3600 RPM it requires final alignment readings within 1.0 mils offset. According to the 4:1 rule for measurement, the alignment system must be able to measure four times better than the desired tolerance to give reasonable certainty of result. So, an alignment system must measure within 0.25 mils of offset to *indicate* the correct offset alignment within reasonable certainty. However, to *correct* alignment, the 10:1 rule applies. The system should be able to measure ten times better than the tolerance in order to produce accurate corrections. Therefore, an alignment system must be able to measure to a resolution of 0.1 mils (0.0001 inch) in order to be able to reliably produce the alignment results in the table above for 3600 RPM. For higher speeds, the minimum requirement is 0.05 mils (0.00005 inch) offset.

Table 11-3
Tolerances for Shaft Alignment

Tolerances for Shaft Alignment

Short Couplings				Spacer Shafts
	Offset	Angularity		(mils per inch of
	(mils)	(mils)/	(mils)/	spacer length)
RPM		(inch)	(10 inch)	mils/inch
600	5.0	1.0	10.0	1.8
900	3.0	0.7	7.0	1.2
1200	2.5	0.5	5.0	0.9
1800	2.0	0.3	3.0	0.6
3600	1.0	0.2	2.0	0.3
7200	0.5	0.1	1.0	0.15

These suggested tolerances are the maximum allowable deviations from desired values (targets), whether such values are zero or nonzero. These recommended tolerances should be used in the absence of in-house specifications or tighter tolerances from the machinery manufacturer.

Courtesy of Ludeca, Inc., Miami, Florida.

Applying the 4:1 and 10:1 rules to angular resolution requires the system to measure minimum angular resolutions of 0.05 and 0.02 mils per inch (mRad) respectively for 3600 RPM. At higher speeds the requirements are 0.025 and 0.01 mils per inch.

So, the definition of a precision alignment system is simple. The system must be able to align machines within precision alignment tolerances. Accordingly, it must measure at least 0.1 mils offset and 0.02 mils per inch angle. Any system that cannot perform at least this minimum threshold of excellence is simply not a precision alignment system.

Reliability Impact

The normal context of reliability is in terms of equipment availability or process operation. This section will interpret reliability to have the broadest possible meaning as it applies to assessing the effects of correcting misalignment. Specifically, we will expand it to include the variety of topics discussed below. This is not done in any attempt to change the definition of the word, but simply to maximize the opportunity to justify precision alignment. For example, strictly speaking, energy savings would not likely be considered a reliability issue. Yet, alignment-related energy savings can pay for an alignment program that has many other hard to quantify, yet beneficial effects on reliability.

Mean Time Before Failure

Achieving precision alignment is credited, by nearly every author on the subject, with extending machine life compared to misaligned machines. Certainly the vast body of anecdotal evidence and equipment operating experience indicates that misalignment contributes to equipment failure rates. Therefore, it is not much of a leap to say that improving alignment increases equipment life or mean time before failure (MTBF). Yet, there are few if any studies indicating how close an alignment will produce how much improvement in MTBF. To date, no one has published any method of scientifically quantifying the benefits of improved alignment.

The largest problem is that alignment does not occur in a vacuum. In a given facility, maintenance practices are in a constant state of change. In an ongoing search for improvement, new materials are constantly tried in everything from couplings to seals to bearings to lubricants. Improved parts designs are incorporated at frequent intervals. Training (or the lack thereof!) changes workmanship norms. New maintenance tools, measurement methods, and evaluation techniques constantly exert their influence and bring about change. In short, everything changes and isolating the impact of one particular change, such as an alignment program, is difficult.

Not only do maintenance practices change, but operating parameters change as well. Processes are run faster or slower to produce more or less product according to market demands. Product specifications change, which in turn change the process average running conditions. Changes in temperature, pressure, speed, flow, and viscosity can all affect machinery life in unknown or unquantifiable ways.

Some or all of these changes are almost certain to take place within a facility during the long time span required to do a meaningful mean time before failure (MTBF) analysis. The certainty of change makes any study of MTBF in an operating environment highly suspect to interpretation and criticism. Any facility sufficiently interested in machine life to study precision alignment is certainly likely to use or investigate other predictive or preventive maintenance technologies. Each technology will have its proponents and claims to MTBF improvements.

Alas, one might conclude there is not yet a reasonable way to separate all the various contributing factors to an improved MTBF and assign to each factor a percentage contribution with certainty. In the absence of a sure-fire method, the author proposes an analysis of equipment history. One can compare the MTBF of machines precision aligned to the MTBF of machines not precision aligned and also to the MTBF of machines of indeterminate (presumably bad) alignment. Although the comparison will not be certain, it can be suggestive. Such a comparison would be especially effective if used in conjunction with the results of other analytical methods.

This calculation method compares the lives of machines precision aligned against some similar machines not precision aligned. It is only valid to the degree that whatever other changes take place in a facility, besides improved alignment practices, occur to both groups of machines equally. The more similar the service and environment of the two machine groups, the more valid the comparison. The two groups can actually be the same machines, but compared at two different periods of time.

For example, the author once had eight identical pumps added to his maintenance responsibilities. All were ANSI pumps directly driven through elastomeric couplings by 75hp electric motors in paper stock preparation service. There were excellent equipment records for those machines dating back for many years. For the purposes of comparison at the time, machine "life" was defined to be the time from last repair until next failure of a seal, bearing, shaft, or coupling. This simplistic definition had the advantage of eliminating most, if not all, failures not possibly related to alignment, such as electrical problems or worn out impellers.

A review of the maintenance records showed that during the previous five years those eight pumps were collectively experiencing one failure every 15.1 days. The average "life" was less than four months! This astonishingly poor service life was often explained by that "known fact" that stock preparation service was especially hard on pumps. Indeed, the constant repairs proved it. Over the years, extra pumps, piping, and a valve system had been installed specifically to prevent the frequent failures from impacting operations. In this way, the breakdowns and failures were not a problem; they were just a normal operating expense!

Implementing an alignment program with an early model Optalign® laser shaft alignment system led to an average equipment life of 42 months. The usual failure mode became low discharge pressure due to a worn impeller. Since there were no other changes in the maintenance practices affecting those pumps during that time, the comparison is valid and indicative *for that facility.* However, the results may not apply to other facilities, because prior to the alignment program these pumps had for all practical purposes never been aligned at all. Since the machines had elastomeric couplings, the maintenance practice was to skip the alignment altogether so long as the coupling could be installed. After all, they reasoned, the rubber coupling made alignment unnecessary. That assumption, of course, was wrong.

Process Uptime

All of the comments about MTBF can be applied to studies attempting to link precision alignment with improved uptime. It certainly seems that a relationship must exist, but quantifying it seems problematical. About the best available method is to compare the uptime before an alignment program to uptime achieved afterward.

Unscheduled Outages

Unscheduled equipment maintenance or outages are certainly valid indicators of maintenance effectiveness. Yet, it is extremely difficult to decide how much of any change in unscheduled outages is due to a precision alignment program. One method that seems reasonable is to count unscheduled outages due to failures normally associated with misalignment. A comparison of outages before and after a precision alignment program would then be insightful.

For example, one could count seal failures, coupling failures, broken shafts, and bearing failures not attributed to inadequate lubrication. Certainly such failures are not proof of misalignment in and of themselves. Likewise, not all such failures are due to misalignment. Yet, misalignment will cause, or contribute to, all of these failure modes. So, an improvement in facility-wide alignment practices should reduce the number of failures causing unscheduled outages.

Maintenance Costs

A different approach is to look at the overall expenditures for such items as bearing, seals, shafts, and couplings over a given period before and after precision alignment. In one steel mill, after several years of intense alignment activity, the reduction in parts costs amounted to over one million dollars per year. Since there were no known changes to the lubrication, purchasing, or maintenance programs, the entire reduction was attributed to a mill-wide precision alignment program.

In other instances, the benefits may not be as easy to isolate, but persistent digging for the numbers will yield results. If precision alignment leads to fewer repairs and a reduction in parts usage, there may also be some cost savings as a result of reduced spare parts inventory.

Overtime

In some work environments calls-outs and overtime hours may be an excellent indicator of reduced repair activity due to a precision alignment program. This is especially applicable if alignment-related failure repair is performed by a relatively small part of the workforce, and their labor data are available in breakout form.

For example, in a given facility perhaps only millwrights repair rotating machinery, and their overtime hours go down after implementing a precision alignment program. Since there are likely to be relatively few millwrights compared to the overall maintenance workforce, the factors having an impact on their overtime can be relatively easy to isolate.

Energy Usage

There have been several papers documenting the energy savings of a precision alignment program. The savings claims vary from 3% to 12%, but there is considerable debate about the measurement methods and validity of the data. As of 1996, there were at least two studies under way that attempted to more definitively study the issue. One is sponsored by the U.S. Department of Energy and the results of that study will eventually be made public.

All of the work and proposed calculations about energy savings due to alignment have been applicable only to machinery driven by electric motors. Although one might expect any savings to apply to turbine-driven machinery, we are unaware of data to substantiate this. Our worksheets (pages 470–476) should apply equally well to any type of driver, but the illustrations will all use electric motors.

Methods for evaluating energy savings realized from improved alignment fundamentally fall into two categories: real power and apparent power. To simplify a rather technical definition, apparent power savings are calculated from changes in volts and amps. Real power, by contrast, is a sophisticated measurement involving not only volts and amps, but also power factor or phase.

In most industrial facilities, real power is the basis upon which the electrical bill is computed, so it would be the better value to use. The caveat is that real power is problematical to measure with common volt and amp meters. One must use an energy analyzer, such as the Dranetz Model 8000.

Even with the use of an energy analyzer, measuring power consumption before and after alignment may not be enough. In most cases, the load varies considerably with respect to the expected savings. For example, an expected alignment energy savings may be 3%, but the load variation may be 10% or more. Load, or horsepower transmitted across the coupling, may vary with pressure, flow, speed, temperature, viscosity, or almost any other process variable. It is thus necessary to ascertain that the load parameters are the same when energy consumption is measured. When the process has changed so that the load is different, it may still be possible to do an energy savings calculation based on over-all drive train efficiency. This will require access to the efficiency curves for the driver and driven machines. The curves are usually available from the original equipment manufacturer for pumps, motors, and turbines.

Quantifying Impact

MTBF Savings Worksheet

Any evaluation of equipment life must account for the time value of money. Although the inclusion of interest rates makes the calculations more complex, it greatly improves the validity of the results. Although many methods are available, this calculation is based on present worth.

Formula

$$Savings = \left[R_B \times \left(\frac{1}{(1+i)^{n_B}} \right) - R_A \times \left(\frac{1}{(1+i)^{n_A}} \right) \right] \times M$$

Where:

Savings	=	Dollar savings per year due to MTBF improvements
R_B, R_A	=	Average repair cost before and after alignment program, respectively. Can include both parts and labor, or only labor, or only parts, but be consistent.
n_B, n_A	=	MTBF before and after alignment, respectively. (in years)
i	=	Interest rate for comparison
M	=	number of similar machines in facility

Calculations

Line	Parameter	Example	Actual
L1	Interest rate per year	.09	
L2	Before precision alignment MTBF	3	
L3	After precision alignment MTBF	3.5	
L4	Before alignment average direct repair cost	$2,300	
L5	After alignment average direct repair costs	$2,100	
L6	Number of similar machines in facility	200	
L7	(1 + L1) ^ L2	3.31178	
L8	(1 + L1) ^ L3	3.91773	
L9	(1 ÷ L7) × L4	694.49	
L10	(1 ÷ L8) × L5	536.03	
L11	**Savings** = (L9 - L10) × L6	**$31,692.00**	

Process Uptime Savings Worksheet

If the plant can run more hours during a year it can make more product. That product can be sold, but has an offsetting incremental cost of production. To account for the various incremental sales factors, this calculation is based on incremental profit. Typically the incremental profit per unit of production is a figure available from the accounting or industrial engineering departments. In this instance, "savings" is actually increased profit.

Formula

$$Savings = U \times Hrs \times IP$$

Where:

Savings	=	Dollar savings per year due to uptime improvements
U	=	Typical unit production rate, units per hour.
Hrs	=	Hours of increased production per year
IP	=	Incremental profit per unit

Calculations

Line	Parameter	Example	Actual
L1	Production rate	1600	________
L2	Hours of increased production	48	________
L3	Incremental profit	$8.00	________
L11	**Savings** = L1 × L2 × L3	**$614,400**	________

Unscheduled Outages Cost Savings Worksheet

In some facilities there is an unscheduled downtime cost that is readily available. Note that it generally overlaps with the increased uptime profit, so most evaluations should only claim one savings or the other, but not both.

Formula

$$Savings = D \times Hrs$$

Where:

Savings	=	Dollar savings per year due to downtime improvements
D	=	Unscheduled downtime costs per hour.
Hrs	=	Hours of reduced unscheduled downtime per year

Calculations

Line	Parameter	Example	Actual
L1	Unscheduled downtime costs per hour	$10,000	______
L2	Hours improvement	16	______
L11	**Savings** = L1 × L2	**$160,000**	______

Simple Energy Savings Worksheet

This is a simple, yet accurate, energy savings worksheet based on overall improvement in machine efficiency. It is useful when estimating savings potential given a typical improvement. For example, perhaps another unit or facility has proven a percentage savings that is applicable to the equipment under study.

Formula

$$Savings = 0.746 \times HP \times Hrs \times Cost \times \left(\frac{100}{eff_B} - \frac{100}{eff_A} \right)$$

Where:

Savings	=	Dollar savings per year due to alignment
0.746	=	kilowatt per horsepower
HP	=	Horsepower ratings of motors being aligned
Hrs	=	Hours the motor is run per year, 330 days × 24 $^{hr}/_{day}$ = 7,920 hr.
Cost	=	Cost of electricity in dollars per kilowatt-hour, typical = \$0.078 per kWh
eff_A	=	Efficiency after alignment, 2% illustrative improvement
eff_B	=	Efficiency before alignment, range for new motors at optimum (full) load is 78% to 96.5%. Plant typical 82%.

Calculations

	Parameter	Single Motor	Entire Plant	Actual
Line 1:	Horsepower	200	30,000	
Line 2	Line 1 times 0.746	149.2	22,380	
Line 3	Hours per year or per month	350 × 24=8,160	340 × 24=8,160	
Line 4	Cost of electricity per kWh	\$0.078	\$0.078	
Line 5	Line2 × Line3 × Line4	\$94,962.82	\$14,244,422.40	
Line 6	Efficiency before alignment	82%	82%	
Line 7	Efficiency after alignment	84%	84%	
Line 8	100 ÷ Line 6	1.2195	1.2195	
Line 9	100 ÷ Line 7	1.1905	1.1905	
Line 10	Line8 - Line9	0.0290	0.0290	
Savings	Line10 × Line5	**\$2757.34**	**\$413,601.11**	

Measured Energy Savings Worksheet

For most applications, there is no applicable and proven energy savings information available. Thus, if one wants the numbers, one is forced to measure it. The following procedure has the advantage that it looks at the overall performance of a machine before and after alignment. The drawback is that it may require permanent installation or servicing of instrumentation during a process shutdown, prior to beginning the study and well before alignment is performed.

Procedure

1) Install and calibrate instrumentation to measure power output from the driveN. For most pumps, this includes pressure and flow. A data logger or strip recorder would be best, but in stable operating conditions visual monitoring is sufficient. Also prepare to measure shaft RPM in order to account for efficiency, which can vary significantly with speed.

2) Connect Dranetz or similar instrument to motor, usually at the motor control center. Program the analyzer to collect *real power* (kWh) for 2 hours. Average the data for energy accumulation over 1 hour. If driver is not an electric motor, install and calibrate the necessary instrumentation to measure power input to the driver.

3) Operate machinery train and record data as found. Two hours of steady state operation would be a practical minimum time, with a printout from the Dranetz every hour. Carefully note any power factor fluctuations. Only use the data if the power factor variations remain less than 0.1; ideally it would remain constant. Also collect process parameter data such as temperature (of the bearings, fluids, coupling, and ambient air), vibration (overall and spectrum), noise level, and pressure.

4) Perform the alignment.

5) Repeat step 3.

Formula and Theory

$$Savings = P \times hrs \times \cos t \times \left(\frac{100}{eff_B} - \frac{100}{eff_A} \right)$$

$$eff = \frac{hp_{driveN}}{hp_{driveR}}$$

Where:

Savings	=	Dollar savings per year due to alignment
P	=	Power measured before alignment, kWh
hp_{driveN}	=	Horsepower calculated from output parameters of driveN
hp_{driveR}	=	Horsepower measured or calculated for driveR (typically from power analyzer)
Hrs	=	Hours the motor is run per year
Cost	=	Cost of electricity, dollars per kilowatt-hour
eff_A	=	Measured efficiency after alignment
eff_B	=	Measured efficiency before alignment

For a pump, the output horsepower can be computed as follows:

$$hp_{pump} = \frac{Q \times \gamma \times H}{550} = \frac{GPM_{water} \times PSI}{9126}$$

In the case where the speed, flow, or pressure changes significantly, then the pump hydraulic efficiency curve should be checked and the *shaft* horsepower used. Changes in load parameters could make the pump operate at a more inefficient place on its curve and thereby offset or even augment gains due to alignment. Since the goal is to quantify alignment benefits, the method used must account for changes in efficiency due to changed operating conditions.

$$hp_{shaft} = \frac{Q \times \gamma \times H}{550 \times e_h} = \frac{GPM_{water} \times PSI}{9126 \times e_h}$$

Where:

hp_{pump}	=	Calculated hydraulic, or output, horsepower
hp_{shaft}	=	Calculated shaft, or input, horsepower
Q	=	Flow of fluid, cubic feet per second
γ	=	Density of fluid, pounds per cubic foot
H	=	Pressure, feet of head
GPM_{water}	=	Flow of water, gallons per minute
PSI	=	Pressure, pounds per square inch
e_h	=	Hydraulic efficiency

Calculations

Line	Parameter	Single Motor	Actual
L1	Initial motor power, kWh	150	
L2	Hours per year or per month	350 × 24=8,160	
L3	Cost of electricity per kWh	\$0.078	
L4	L1 × L2 × L3	\$95,472	
L5	motor hp = L1 × 1.340	201	
L6	Initial flow of Water, ft^3/sec	10.5	
L7	Initial density of fluid, lb/ft^3	62.4	
L8	Initial pressure, ft of head	115	
L9	Initial efficiency from pump curve	.72	
L10	pump hp = (L6 × L7 × L8) ÷ (550 × L9)	190.3	
L11	eff_B = (L10 ÷ L5) × 100	94.66	
L12	Final motor power, kWh	147	
L13	Final motor hp = L12 × 1.340	196.98	
L14	Final flow of Water, ft^3/sec	10.5	
L15	Final density of fluid, lb/ft^3	62.4	
L16	Final pressure, ft of head	115	
L17	Final efficiency from pump curve	.72	
L18	hp = (L14 × L15 × L16) ÷ (550 × L17)	190.3	
L19	eff_A = (L18 ÷ L13) × 100	96.61	
L20	100 ÷ L11	1.0564	
L21	100 ÷ L19	1.035	
L22	L8 - L9	0.0213	
L23	**Savings** = L22 × L3	**\$2035.74**	

Why and How to Monitor Centrifugal Pump Condition

It is not unusual to find 1,200 centrifugal pumps installed in a good-sized petrochemical plant, with approximately 600 of these running at a given time. If only 400 of them require shop repairs during the course of a year, the plant has a better than average repair record.[10] Not counting product losses or fire damage caused by approximately one event per 1,000 pump failures, the average pump repair costs $9,800. These are valid 1998 accounting figures that include field removal, installation, failure analysis, and burden. It is easy to visualize how pump failure reductions may save the average petrochemical plant a very sizable amount of money every year. The desired reduction in failure incidents or failure severities can be achieved by monitoring the condition of centrifugal pumps and initiating corrective action at the right time.

Let us look at it from another vantage point. A survey of "condition monitoring" in British industry[11] estimated that industries already using condition monitoring could readily increase their savings at least six-fold if available instrumentation were applied more widely. The same survey estimates that approximately 180 industries could benefit from condition surveillance; however, only ten industries are presently utilizing these cost-saving techniques. Finally, if improved surveillance techniques and their proper management could be implemented in these 180 industries, British industry is thought to be able to realize annual net savings in excess of $1 billion, or about 40 times the amount presently saved by the limited application of machinery surveillance in only ten industries.

Centrifugal pumps and their drivers probably represent a large portion of this total. From the point of view of condition monitoring, they are simple machines, and much data are available which allow reliable determination of component integrity.

Condition Monitoring Defined

The concept of condition monitoring encompasses the detection of mechanical defects as well as fluid-flow disturbances. Typical mechanical defects include bearing flaws, mechanical-seal defects, coupling malfunctions, rotor unbalance, erosion, corrosion, and wear. Fluid-flow disturbances include inadequate NPSH, insufficient flow, and gas entrainment or cavitation. Abnormal flow conditions can lead to mechanical defects and vice versa. Also, abnormal flow conditions may or may not manifest themselves in pump vibration. Similarly, mechanical defects may or may not manifest themselves in pump vibration. True condition monitoring should not be confined to mere vibration data logging, because measureable vibration increases are sometimes occurring only after irreversible mechanical damage has taken place. Condition monitoring should, therefore, be defined as the detection of any abnormal parameters which must be corrected if pump damage is to be avoided.

How Abnormal Parameters Can Be Detected

A number of diagnostic means are usually available to determine the condition of centrifugal pumps. Fluid-flow disturbances are best measured by pressure deviations, or changes in flow patterns and, sometimes, temperatures. Mechanical distress, both existing or potential, can manifest itself as changes in lube-oil temperature, lube-oil particulate contamination, bearing noise, and, of course, pump vibration. All of these parameters can be measured with available techniques. However, the determination of mechanical distress by continuous lube-oil analysis, bearing surface contact roughness measurements, etc., is not presently considered practical for centrifugal pumps in the petrochemical industry.[12]

On the other hand, the determination of machinery distress by measuring increases in vibration levels has achieved widespread acceptance. Monitoring vibration levels of operating machinery through the point of failure has shown that 90% of the time this indicator moves up sharply prior to actual failure.[13] Consequently, machinery conditioning monitoring is often confined to vibration monitoring and/or vibration analysis. However, state-of-the-art condition monitoring methods go beyond vibration monitoring. As will be seen later, these up-to-date methods attempt to capture incipient failure events by monitoring stress waves and/or shock pulses which can *precede* machinery vibration by days or sometimes weeks.

Conventional (low-frequency) vibration monitoring. Excessive vibration of centrifugal pumps can lead to internal rubbing, overstressing of pipe flanges and hold-down bolts, grout failures, mechanical-seal leakage, bearing damage, coupling wear, and a host of other difficulties. Vibration detection and remedial action are necessary if equipment life and safety of personnel are to be ensured. Vibration *detection and monitoring* are commonly used to determine existence and severity of the problem, while *vibration analysis* is needed to define the cause of deviations from normal equipment behavior. With the exception of spring-operated, hand-held vibrographs, conventional vibration instrumentation makes use of transducers which change mechanical energy into electrical energy. Transducers can embody one or both of the following principles:

1. Proximity measuring techniques employing non-contacting, eddy-current probes to determine distance or change in distance to a conductive material. Proximity measurements are indispensable for the surveillance of large turbomachinery. Centrifugal pump applications include nuclear reactor coolant circulators and large boiler feedwater pumps.[14] Figure 11-26 shows a typical pump installation incorporating proximity probes.
2. Velocity transducer techniques operating on the inertial mass, moving-case principle. The inertial mass consists of a copper wire coil suspended inside the pickup case. The pickup case incorporates a permanent magnet. Machine vibration induces a current in the coil. Within frequencies ranging from approximately 10 Hz to 2kHz, the induced current is proportional to the velocity of vibration.[15] A centrifugal pump equipped with a velocity transducer is shown in Figure 11-27.

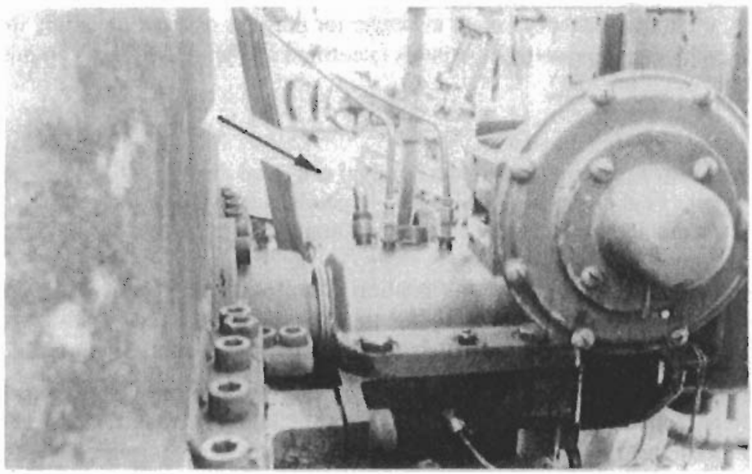

Figure 11-26. Pump drive turbine incorporating proximity probes. ***(Courtesy of Bently-Nevada Company, Minden, Nevada.)***

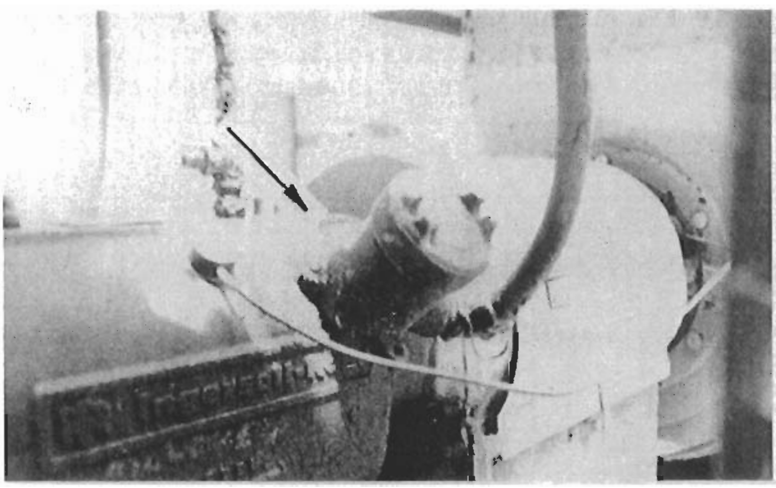

Figure 11-27. Centrifugal pump with case-mounted velocity (seismic) transducer. ***(Source: Metrix Company, Houston, Texas.)***

3. Accelerometer-based measuring techniques using a piezoelectric crystal sandwiched between the accelerometer case and an inertial mass. Machine vibration causes the crystal to be strained and a displaced electric charge to migrate to the opposite side of the crystal. The resulting voltage is proportional to the acceleration along the axis perpendicular to which the accelerometer casing is normal. Typical frequency response would be 3 Hz to 10 kHz.
4. Dual probes incorporating proximity and velocity transducers in a single housing.

All of these transducer types are available for portable or fixed mounting installation. In each case, the electrical output is quantified and processed to accomplish one or more of the following.

1. Display on a suitable meter.
2. Signal storage, followed by data retrieval and trend display.
3. Annunciation or machine shutdown if a pre-selected value is exceeded.
4. Determination of frequency components contained in the signal.

State-of-Art (High Frequency) Vibration Monitoring. In practice, display on a suitable high frequency meter is by far the most common mode of usage. These readings are taken with portable meters similar to the one shown in Figure 11-28.

Modern data collectors give the answer quickly by measuring overall vibration severity (effective RMS velocity) at the touch of a key. ISO Standard 2372 (Figure 11-29) then rates this vibration level in terms of "good" to "unacceptable" for the particular machine type at hand. Even inexperienced personnel can tell instantly whether further analysis is needed.

A single broadband measurement (Figures 11-30 and 11-31) can give an objective benchmark for following vibration trends over time—the perfect solution for machines that do not warrant the effort and expense of complicated FFT-based

Figure 11-28. Portable instrument used to monitor pump condition. (*Courtesy of Prueftechnik AG, D-85730 Ismaning, Germany.*)

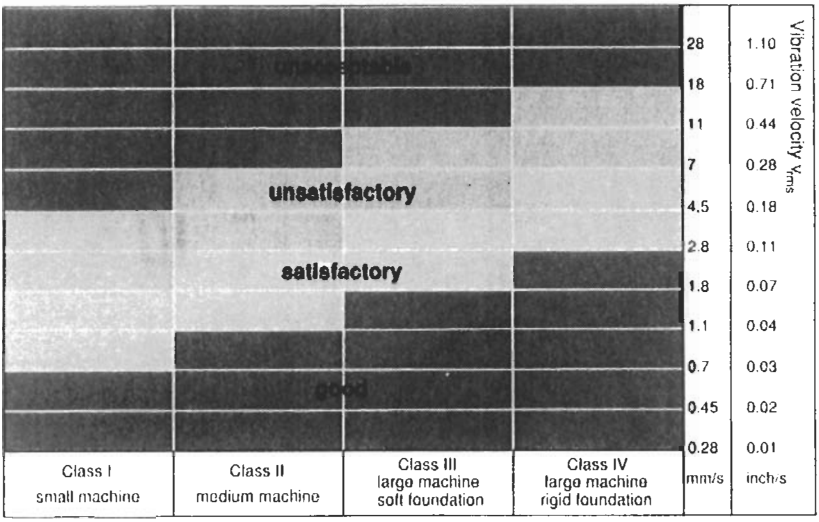

Figure 11-29. ISO Standard 2372 rates vibration levels.

vibration monitoring, yet deserve periodic observation with trending. Zero-to-peak and peak-to-peak values can also be measured if desired.

The instrument depicted in Figure 11-28 allows measurements to be taken even in locations where access or visibility is poor, requiring only that a hand-held probe be plugged into the accessory jack on the top of the instrument. Used in conjunction with a coded stud (see Figure 11-32) locations are registered automatically during measurement.

A conscientiously followed vibration monitoring program can be expected to reduce pump and driver maintenance costs by approximately 5% per year for perhaps five or six years. At the end of this time period, maintenance costs for pumps and drivers should level at approximately 70% of expenditures incurred before initiation of the program. Data from an average U.S. petrochemical company indicate that pump and driver maintenance without conventional vibration monitoring would be approximately $25 (prorated to 1997 dollars) per horsepower per year of installed pump-and-driver combination. Vibration-related mechanical problems experienced an average decline of 16% per year over a period of three years. The number of problem incidents then stabilized at 60% of the previous total, bringing the maintenance cost *with* vibration monitoring to approximately $17 per horsepower per year.

The estimated direct cost for conducting a conventional vibration monitoring program on a plant-wide basis is about $2.25 per horsepower per year of installed pump-and-driver combined horsepower. Even if we assume the indirect costs to exceed direct costs by a substantial margin, the net savings are far too significant to be ignored. This is especially true because earnings resulting from reduced product losses, or possibly, extended unit runs have not been included in the estimate, and

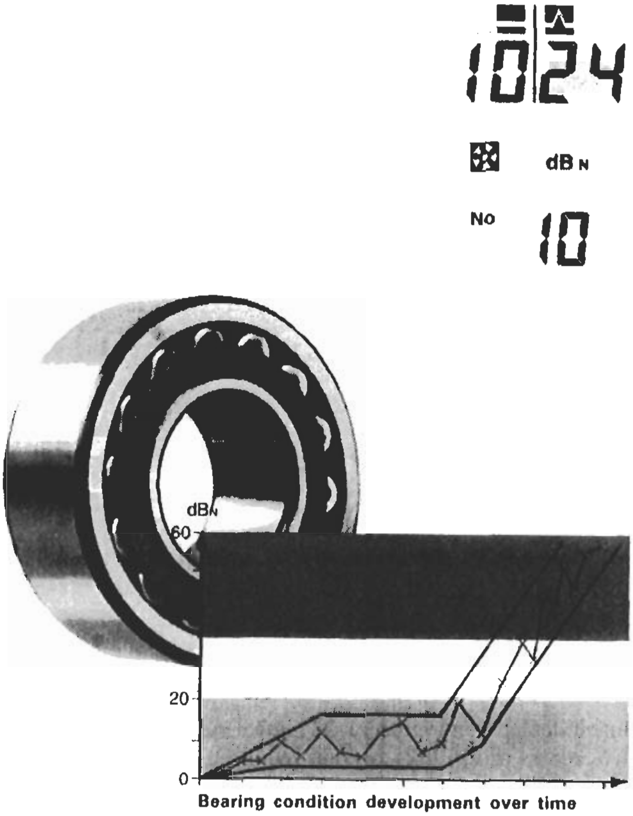

Figure 11-30. Broadband measurements monitor bearing condition. (*Courtesy of Prueftechnik AG, D-85730 Ismaning, Germany.*)

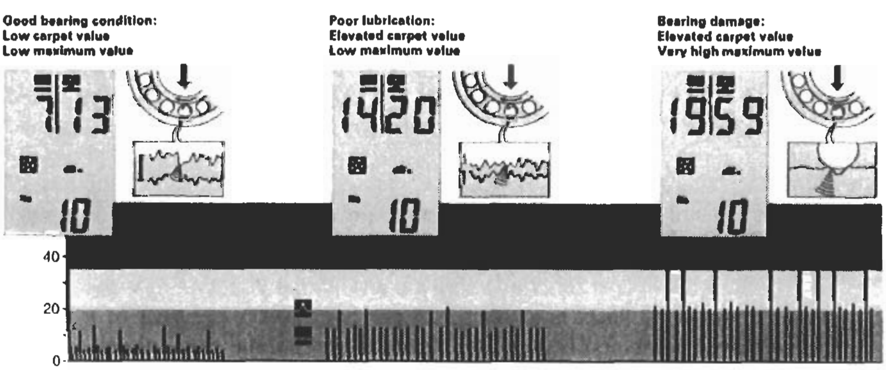

Figure 11-31. Shock pulse measurements contain valuable information. (*Courtesy of Prueftechnik AG, D-85730 Ismaning, Germany.*)

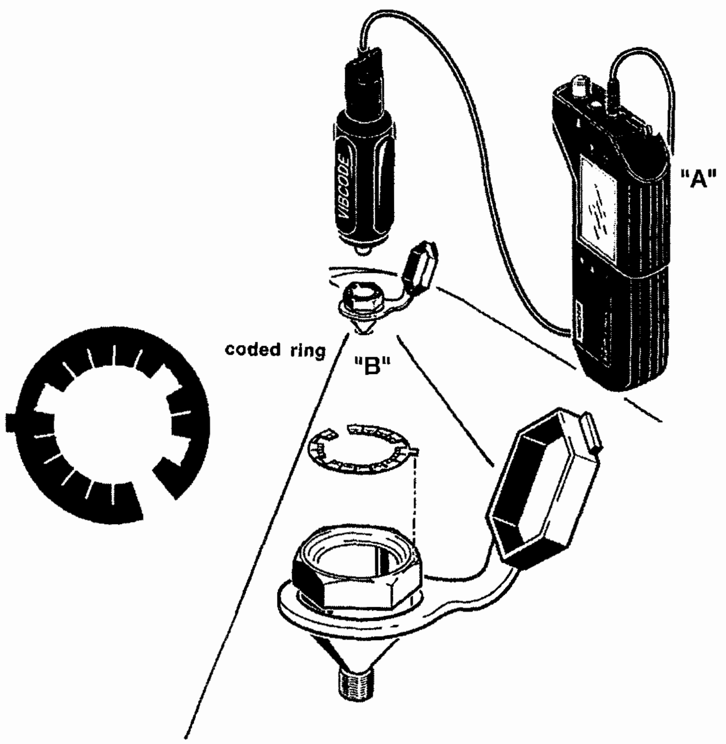

Figure 11-32. Modern condition monitoring instrument "A" connected to coded stud "B" will automatically register measurement locations. (*Courtesy of Prueftechnik AG, D-85730 Ismaning, Germany.*)

also because forward-looking plants have found it advantageous to let the *operator* do the data collecting. Using the data collector shown in Figure 11-32 as a screening tool allows the operator to spot deviations and abnormalities such as cavitation, bearing distress, excessive temperatures and speed deviations. Since all captured data are transferred to a computer, it is quite evident that trending and also the tracking of equipment operating hours are made easy with just this one 300 gram tool!

Finally, readers in need of a highly practical, up-to-date text dealing with vibration analysis of machinery are encouraged to look at R. C. Eisenmann's *Machinery Malfunction Diagnosis and Correction—Vibration Analysis for the Process Industries,* Prentice PTR, Upper Saddle River, New Jersey; 1997 ISBN 0-13-240946-1. Written by an expert father-and-son engineering team, the Eisenmann book deals with many different types of process machinery vibration case histories.

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Chapter 12

Lubrication and Reliability

The importance of properly lubricating process machinery is so obvious that little time needs to be spent in discussing the subject. Yet, there are issues that keep coming up: lube-oil purification, synthetic lubricants, cost justifications, claims and counterclaims. Within the scope of this text, we have to limit coverage of the subject to the essentials, to the often misunderstood, and to approaches that are perhaps only practiced by the leading process plants. Bear in mind, though, that these are the areas where being proactive may yield significant reliability improvements and in some cases, rapid payback.

Methods and Criteria for Lube-Oil Purification

The average petrochemical plant has about 40 rotating machinery trains equipped with pressure lubrication systems. Each of these systems incorporates a self-contained lube-oil reservoir ranging in capacity from perhaps 50 gallons ($0.2m^3$) for a process pump, to approximately 6000 gallons ($24m^3$) for large turbocompressor lube systems.[1]

Experience shows that self-contained lube-oil reservoirs are subject to contamination and deterioration from particulate matter and water. Some gas-compressor lube-oil systems are further exposed to potential dilution from lighter hydrocarbons. Excess water and hydrocarbon constituents which depress the viscosity or flash point of steam-turbine lube oils must be removed periodically if machinery distress is to be avoided.

While the need for lube-oil purification has generally been recognized, operators have been confronted with a profusion of guidelines attempting to define maximum allowable water contents of turbine lube oils. Table 12-1 illustrates how a large U.S. compressor manufacturer allows 40 wppm maximum, several turbine manufacturers allow 100 wppm, the U.S. Navy permits a maximum water content of 500 wppm, and so on. This range of possible contamination levels prompted a detailed investigation of prior research into the effects of contaminated lube oil on turbine reliability.[1]

The study showed that acceptable contamination levels can be defined as a function of lube-oil temperature and relative humidity of the systems environment. It identified several lube-oil dewatering methods as potentially suitable for long-term

Table 12-1
Steam-Turbine Lube-Oil Purification Practices

Guideline Issued by User or Manufacturer	Maximum Allowable Water Content Quoted	Sampling Frequency	Purifier Operation
Chemical plant, Texas	4000 wppm	monthly	intermittent
Consulting firm	2000 wppm (steam turbines only)	—	—
Refinery, Europe	1000 wppm (BFW and cooling water pumps)	monthly	intermittent
Refinery, Kentucky	1000 wppm	—	as required
U.S. Navy (MIL-P-20632A)	500 wppm	—	continuous
Consulting firm	500 wppm (gas turbines only)	—	—
Chemical plant, Canada	200	visual	as required
Utility, Michigan	100	monthly	continuous
Chemical plant, Texas	100	6 months	intermittent
Chemical plant, Louisiana	100	monthly	intermittent
Refinery, Texas	100	—	intermittent
Consulting firm	100	weekly	as required
Utility, Europe	100	—	continuous
Refinery, Texas	100	—	continuous
Utility, Michigan	100	—	—
Turbine manufacturer, Japan	100	—	—
Compressor manufacturer, Pennsylvania	40	—	—
Refinery, Louisiana	—	—	intermittent
Refinery, California	—	—	intermittent

trouble-free operation in petrochemical plants or on offshore platforms. However, only soundly engineered systems can be expected to give satisfactory results. Accordingly, a number of highly relevant, although frequently overlooked, design considerations are explained in detail.

Sources of Water Contamination

Analytical studies and field experience show that even under the best of circumstances, lube-oil drain headers and reservoirs are saturated with moist air.[2] The systems are usually vented to atmosphere. Temperature differences and cyclic variations in ΔT between vent areas and ambients promote condensation. The possibility of ingesting wet or contaminated air exists also at the shaft seals. Large amounts of oil draining from the bearing area back to the reservoir are known to create suction effects or slightly lower pressure regions in the bearing housing. This promotes the inflow of ambient air through labyrinth seals and, together with condensation in reservoirs and vents, explains the fact that even motor-driven turbomachinery experiences lube-oil contamination.

The potential problems are compounded on steam-turbine-driven machinery. Figure 12-1 illustrates a typical mechanical drive steam turbine. Steam leakage past the

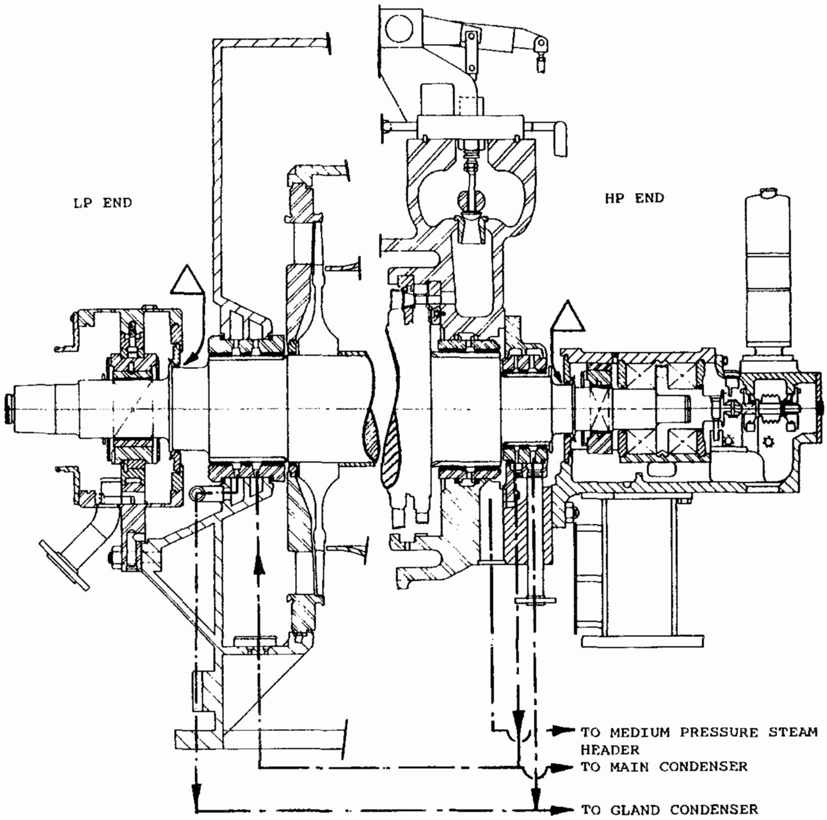

Figure 12-1. Labyrinth seals of mechanical drive steam turbine with typical water ingestion points indicated "Δ." ***(Courtesy of Elliott Company, Jeannette, Pennsylvania.)***

last set of labyrinths at either the high-pressure or low-pressure end of the turbine will find its way easily past the bearing seal labyrinth and into the lube-oil drain piping.

Water Contamination Can be Minimized

While it is quite appropriate to be concerned with selecting effective lube-oil dewatering methods, it would be a mistake to overlook how moisture intrusion can be minimized in the first place.

At least one multinational chemical company located in the U.S. Gulf Coast area has obtained excellent results by using an air purge in the labyrinth seal separating the turbine bearings from the adjacent casing. This is accomplished by bleeding small volumes ($1.4 m^3/hr$ or 50 scfh) of shop or instrument air into the annulus formed between sets of labyrinth teeth, as shown in Figure 12-2. Most of the air escapes

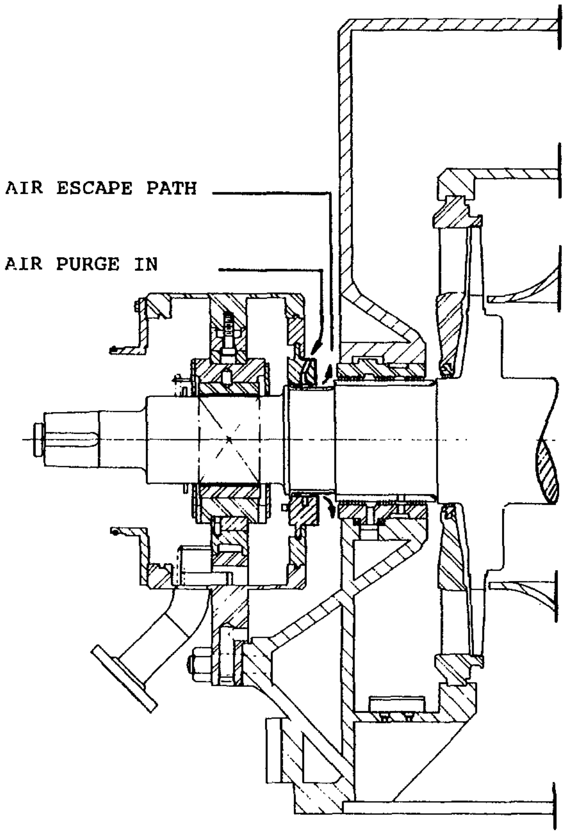

Figure 12-2. Air-purged bearing housing labyrinth for large steam turbines.

toward the turbine exterior, and in doing so swirls away any steam or airborne contaminants migrating toward the bearing interior from the external environment.

A second, at least equally effective and more energy efficient, way of minimizing moisture intrusion involves the use of rotor-stator seals. Figure 12-3 depicts one such seal that may prove valuable on small steam turbines.[3] This special seal is a metallic, non-contact unit that has no wear parts. It consists of a rotor section with a locking ring that clamps onto the shaft and a stationary section mounted into the housing. The split execution allows for field installation without the need to remove the shaft. Because of this feature, most installations can be made in two hours or less. Since it does not incorporate any elastomers, temperature is not usually a limiting factor.

This modified labyrinth design represents a "controlled leakage path." Steam trying to enter into the bearing housing is either expelled outright or begins to follow the labyrinth configuration of the seal. It first makes a 90° turn around the labyrinth nose and then a 90° turn into the inboard cavity.

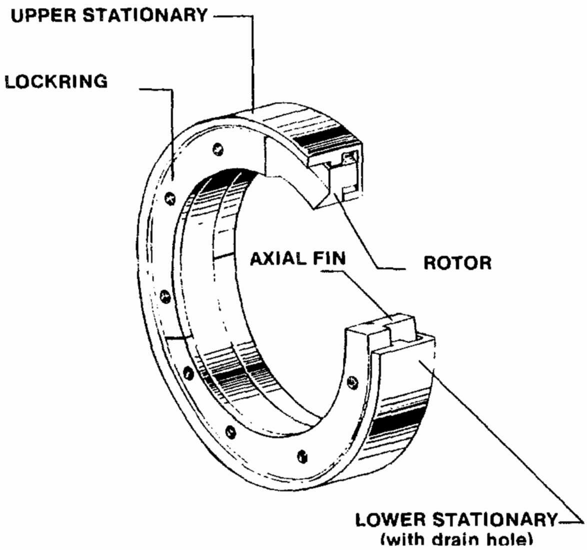

Figure 12-3. Advanced bearing housing seal for steam turbines. (*Courtesy of ROC Carbon Company, Houston, Texas.*)

The seal nose has machined recesses ("pumpers") revolving at the same rpm as the shaft, gathering steam, water, and contaminants and rapidly moving them around the labyrinth periphery, which allows time for cooling and condensation. The collection of contaminants uses both the principles of dynamic pumping as well as radial centrifugal force action. Most steam and contaminants will then be expelled from this inboard cavity through the drain hole. The other side of the rotor acts as a slinger, retaining oil in the bearing housing. While application of this seal to large special-purpose machinery may require redesign efforts, its merits have been demonstrated on smaller machinery.

Airborne water vapor has also been observed to condense on the interior of vent lines connected to bearing sumps, gear cases, and lube-oil reservoirs. Using a simple dip leg, condensing vapors can be collected and drained before they reenter the system as free water. Finally, a nitrogen sweep could be employed to reduce the risk of moist air contacting large quantities of lube oil in reservoirs and drain piping.

Free Water Judged Detrimental

Research conducted at the U.S. Navy Marine Engineering Laboratory has confirmed the detrimental effects of free water in turbine lubricating oil. Free water is known to remove some of the rust inhibitors from these oils. Summarizing his investigations, MacDonald[2] describes the probable sequence of events as follows:

Small amounts of free water condensed from the air, and from gland leaks, begin to collect in the oil. It settles in the quiescent areas of the sump, reservoirs, bearing pedestals, and governors. Although the quantity is small, it continues to collect until it is sufficient to carry through the system. Even oil containing a good rust inhibitor will not give indefinite protection against settled, free water. Thus rusting begins in the quiescent oil-wetted regions of the system. Meanwhile, the water concentration in the vapor spaces of bearing pedestals, gear cases, bearing oil-return lines, and sumps has also increased, and in the presence of the air being released from the oil, rusting begins wherever condensation occurs. The rust that forms in the bearing oil-return lines consists of extremely small particles. It enters the system and continues to circulate. Because of its small particle size it is often overlooked and it usually collects only in such places as governor reservoirs or other quiescent areas. When it is mixed with water and oil, a solid-stabilized emulsion may form. It is an excellent polishing agent, dry or wet, and is most likely to cause governor and overspeed trip troubles. Rust flakes may be formed by subsequent agglomeration of the small particles and in areas where larger water droplets form on condensation. Such flakes are commonly found in gear cases and sumps. While they are usually too large and dense to circulate to any great extent, if drawn into the oil pumps they become fragmented and pass through the system. Such coarse materials are the most likely sources for solids which cause grooving and scoring of bearings and journals.

On many occasions, the catastrophic failure of steam turbines has been attributed to the presence of free water in the oil. As long as the oil is contaminated with free water there exists the risk of relay valve and piston sticking, and overspeed trip bolts may seize. This is due to features of equipment design which allow water separation in feed lines, valves, pistons, and trip bolts. Even the occasional overspeed trip testing and exercising of moving parts cannot eliminate these risks as long as wet oil is introduced into the turbine lube and governing system.[4]

Free water in turbine lubricating oils has also been found responsible for journal and bearing corrosion. While the corrosion of lead in lead-base whitemetals is quite well known, engineers should be aware that tin-base whitemetals are not necessarily immune from similar attack. Reference 19 contains photographic evidence of smooth, hard, black deposits of tin dioxide on the surface of marine-turbine whitemetal bearings caused by water in the oil. Wilson[5] describes equipment that soon after startup suffered severe pitting and "water marking" of journals and gears along with discoloration and hardening of the bearings. Investigation showed evidence of galvanic corrosion caused by water and corrosion debris resting in the bottom of the bearings and oil wiper slots.

The detrimental effects of free water in turbine lubricating oil have recently been observed by at least two petrochemical companies. In one location there appeared significant babbitt damage on bearings; the other location attributed repeated failure of eddy-current probe tips to the presence of free water in the oil.

Quantifying the Effects of Water

The observations of MacDonald[2] are supported by more definitive investigations by Appeldoorn, Goldman, and Tao[6] which sought to quantify the influence of water and oxygen in lubricants. This latter reference concludes that water and oxygen cause a significant increase in wear under non-scuffing conditions, and that this process appears to consist of the formation and rubbing away of metal oxides.

Research by Schatzberg[7] dealt with the time-dependent wear of lubricants under non-scuffing conditions. Schatzberg was able to quantify wear scar versus time for a premium turbine lubricating oil (ISO Grade 32) with oxidation and rust inhibitors. In the same reference, he depicted his findings of wear scar versus load for the same lubricant. It is of extreme interest to note that *only in this particular lubricant,* Schatzberg found the effect of dry oxygen as severe as that of humidified oxygen plus 1% water. This leads us to conclude that excessive aeration of turbine lube oils must be avoided, and that dry nitrogen blanketing is highly desirable on large, expensive machinery trains.

It is well known that either dissolved or free (suspended) water in lubricating oils can cause a significant reduction in the fatigue life of ball bearings and other highly stressed alloy steel components. For instance, 0.01% *dissolved* water in a lubricant decreased the specimen fatigue life by 32–48%. It was concluded that microcracks in the ball surface of anti-friction bearings act as capillaries into which water condenses. This forms a water-rich phase and the resulting aqueous corrosion leads to hydrogen embrittlement within the cracks.[8]

These findings are highly relevant for installations with anti-friction bearings. There is clearly a powerful incentive to find lubrication methods which rule out water contamination of the bearing lubricant. Incidentally, this explains why once-through dry-sump oil-mist application is a superior lubrication method for anti-friction bearings.

Cost Justification and Latest Technology for the On-Stream Purification of Turbomachinery Lube Oil*

Contaminants that interfere with reliable lubrication of compressors, gas turbines, steam turbines, and their respective auxiliaries will inevitably collect in the lube oil. Their timely and dependable removal can have great significance for repair cost, disposal cost, and plant emissions. Until recently, only somewhat complex vacuum oil purifiers were available for this decontamination duty. This section will describe state-of-art, maintenance-friendly vacuum oil purifiers and their latest adjuncts: packaged, positive pressure ambient air, or inert gas stripping units.

How Process Plants Benefit From Dedicated Oil Purifiers

In the 1975–1995 time frame, the benefits of oil purification were viewed primarily as reducing oil consumption and improving machinery reliability, thereby contributing to plant operating cost reductions.

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*Adapted from Bloch, Allen, Russo presentation at 5th International Turbomachinery Maintenance Congress, Singapore, 1991.

While these benefits are self-evident, it is becoming increasingly important to focus on methods of reducing plant emissions. Oil purification, on-site and on-stream, has now moved to the forefront because of environmental pressures and oil disposal concerns. In addition to social cost justification, economic benefits accrue from the avoidance of potential steep fines from the environmental regulatory agencies.

The viability of on-stream oil purification has been documented by a world-scale chemical plant that has had in service an inventory of 22,000 U.S. gallons of lube oil for more than twenty years with no plans to replace it (Reference 9).

Lube oils of any grade or specification used in machinery operation and maintenance generally suffer from three common sources of contamination: dirt; hydrocarbon, gas, or other process dilutants; and water intrusion.

Of these, the first one, dirt, is usually filterable; hence, it can be readily controlled. However, dirt is often catalyzed into sludge if water is present. Experience shows that if water is kept out of lube oil, sludge can be virtually eliminated.

The second contaminant, process-dependent dilution, is seen in internal combustion engines and gas compressors where hydrocarbons and other contaminants blow past piston rings or seals and are captured within the lube and/or seal oils. Dilution results in reduced viscosity, lower flash points, and noticeable reduction of lubrication efficiency.

The last one, water, is perhaps both the most elusive and vicious of rotating machinery enemies. In lube oil, water acts not only as a viscosity modifier but also actively erodes and corrodes bearings through its own corrosive properties and the fact that it dissolves acid gases such as the ones present in internal combustion engines. Moreover, water causes corrosion of pumps, and rusts cold steel surfaces where it condenses.

In some systems, water promotes biological growth which, in itself, fouls oil passages and produces corrosive chemicals. Water-contaminated lube oils can truly be termed "the enemy within." Because of the elusive nature and the many misunderstandings associated with water contamination, the reliability professional must be concerned with water, its effects, and means of removing it from a lube-oil system.

Forms of Water Contamination Vary

In oil systems associated with process machinery, water can, and will, often exist in three distinct forms: free, emulsified, and dissolved. But, before examining the effects of water contamination, it may be useful to more accurately define these terms.

Free water is any water that exists in excess of its equilibrium concentration in solution. This is the most damaging form for water to be in. Free water is generally separable from the oil by gravity settling.

Emulsified water is a form of free water that exists as a colloidal suspension in the oil. Due to electro-chemical reactions and properties of the oil/water mixture in a particular system, some or all of the water that is in excess of the solubility limit forms a stable emulsion and will not separate by gravity even at elevated temperatures. In this respect, emulsified water behaves as dissolved water, but it has the damaging properties of free water and modifies the apparent viscosity of the lubricant.

Dissolved water is simply water in solution. Its concentration in oil is dependent upon temperature, humidity, and the properties of the oil. Water in excess of limits imposed by these conditions is free water. Figure 12-4 gives the equilibrium concentration of water in typical lube oils. Dissolved water is not detrimental either to the oil or the machinery in which it is used.

For corrosion to occur, water must be present. Free water, in particular, will settle on machinery surfaces and will displace any protective surface oil film, finally cor-

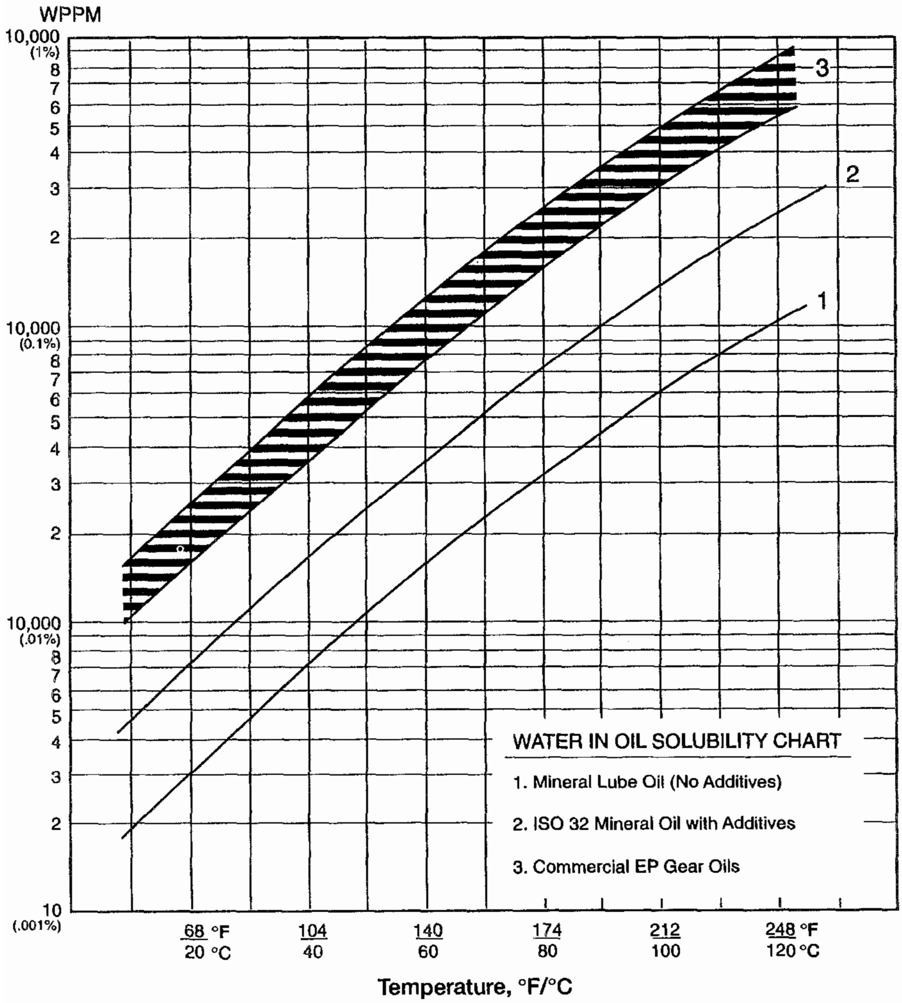

Figure 12-4. Water in oil solubility chart.

roding the surface. Emulsified water and dissolved water may vaporize due to frictional heat generated as the lube oil passes through bearings. Very often, though, the water vapors recondense in colder pockets of the lube-oil system. Once recondensed, the free water continues to work away at rusting or corroding the system.

Larger particles generated by corrosion slough off the base metal surface and tend to grind down in the various components making up the lube system, i.e., pumps, bearings, control valves, and piping. The mixing of corrosion products with free and emulsified water in the system results in sludge formation which, in turn, can cause catastrophic machinery failures.

Next, it should be pointed out that water is an essential ingredient for biological growth to occur in oil systems. Biological growth can result in the production of acidic ionic species and these enhance the corrosion effects of water. By producing ionic species to enhance electro-chemical attack of metal surfaces, biological activity extends the range of corrodible materials beyond that of the usual corrodible material of construction, i.e., carbon steel.

While corrosion is bad enough in a lube oil system, erosion is worse because it usually occurs at bearing surfaces. This occurs through the action of minute free water droplets explosively flashing within bearings due to the heat of friction inevitably generated in highly loaded bearings.

Additive loss from the lube oil system is another issue to contend with. Water leaches additives such as anti-rust and anti-oxidant inhibitors from the oil. This occurs through the action of partitioning. The additives partition themselves between the oil and water phase in proportions dependent upon their relative solubilities. When free water is removed from the oil by gravity, coalescing or centrifuging, the additives are lost from the oil system, depleting the oil of the protection they are designed to impart.

Figure 12-5 illustrates the severity of the effects of water on bearing life due to a combination of the above. We see that bearing life may be extended by 240% if water content is reduced from 100 wppm to 25 wppm. However, if water content is permitted to exist at 400 wppm, most of which will be free water, then bearing life will be reduced to only 44% of what it could be at 100 wppm.

Ideal Water Levels Difficult to Quantify

As illustrated earlier in Table 12-1, manufacturers, operating plants, and the U.S. Navy all have their own water contamination limits.

Extensive experience gained by a multinational petrochemical company with a good basis for comparison of competing water removal methods points the way here. This particular company established that ideal water levels are simply the lowest practically obtainable and should always be kept below the saturation limit. In other words, the oil should always have the ability to take up water rather than a propensity to release it.

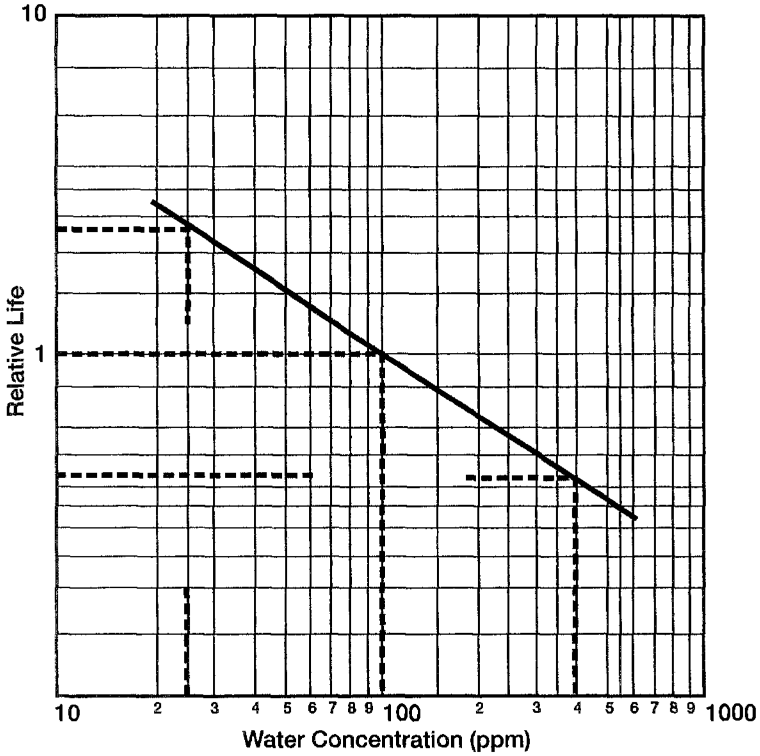

Figure 12-5. Effect of water concentration on bearing life.

Methods Employed to Remove Water

Lube-oil dewatering *centrifuges* (Figures 12-6 and 12-7) have been installed by many foreign and domestic petrochemical plants. They are also in extensive use on shipboard and offshore platform locations as well as by public utilities and private power plants.

Centrifuges have been used for decades. They operate on the principle that substances of different specific gravities (or densities) such as oil and water can be separated by centrifugal force. Centrifuges achieve a form of accelerated gravity settling, or physical separation. At a given setting, centrifuges are suitable for a narrow range of specific gravities and viscosities. If they are not used within a defined range, they may require frequent, difficult readjustment. They will not remove entrained gases such as hydrogen sulfide, ethane, propane, ethylene, or air. Although centrifuges

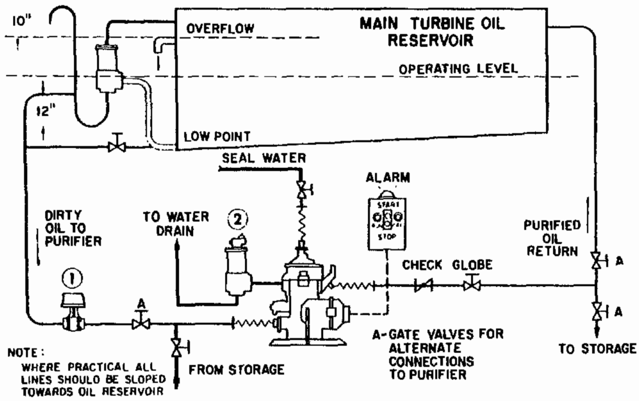

Figure 12-6. Typical centrifugal separator installation. (*Courtesy of De Laval Separator Company, Poughkeepsie, New York.*)

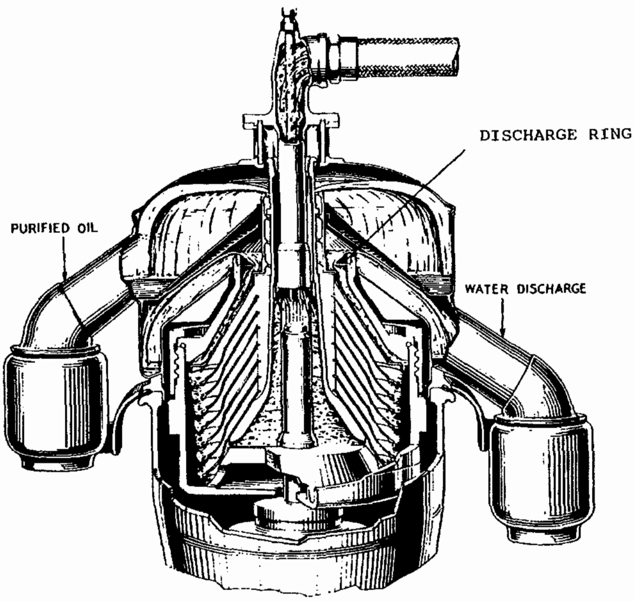

Figure 12-7. Cutaway view of oil purifier bowl. (*Courtesy of De Laval Separator Company, Poughkeepsie, New York.*)

provide a quick means to separate high percentages of free water, they are maintenance intensive because they are high-speed machines operating at up to 30,000 RPM. More importantly, they can only remove free water to 20 wppm above the saturation point in the very best case, and none of the dissolved or emulsified water. In fact, centrifuges often have a tendency to emulsify some of the water they are intended to remove.

Coalescers (Figure 12-8) are available for lube oil service and have found extensive use for the dewatering of aircraft fuels.[10] Unfortunately, coalescers remove only free water and tend to be maintenance-intensive. More specifically, a coalescer is a type of cartridge filter that operates on the principle of physical separation. As the oil/water mixture passes through the coalescer cartridge fibers, small dispersed water droplets are attracted to each other and combine to form larger droplets. The larger water droplets fall by gravity to the bottom of the filter housing for drain-off by manual or automatic means. Since coalescers, like centrifuges, remove only free water, they must be operated continuously to avoid long-term machinery distress.

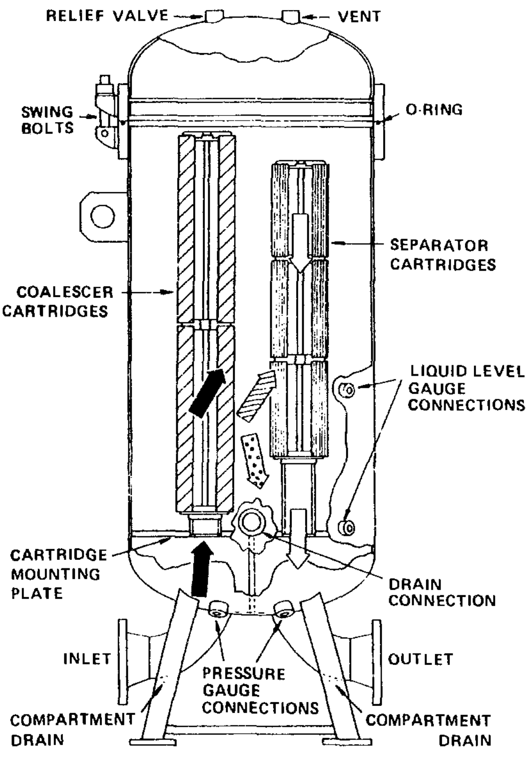

Figure 12-8. Cross section of typical vertical two-stage coalescer. (*Courtesy of Special Fluid Products, Inc.*)

The moment they are disconnected, free water will form and begin to cause component damage. Again, because they are based on a physical separation principle, they are only efficient for a narrow range of specific gravities and viscosities.

Coalescers are used in thousands of airports throughout the world to remove water from jet fuel. Water, of course, freezes at high altitudes. Refining of jet fuels is closely controlled to rigid specifications that allow successful water removal by this method. There are several disadvantages to coalescers. They are only efficient over a narrow range of specific gravities and viscosities. They do not remove dissolved water, which means they must be operated continuously, and it is expensive to change elements.

Filter/Dryers are also cartridge-type units that incorporate super-absorbent materials to soak up the water as the wet oil passes through the cartridges. They remove free and emulsified water, require only a small capital expenditures, and are based on a very simple technology. However, they do not remove dissolved water, and their operation might be quite costly because the anticipated usage rate of cartridges is highly variable due to the changing water concentrations. The amount of water contamination at any given time would be difficult, if not impossible, to predict. Additionally, high cartridge use creates a solid waste disposal problem.[11]

Next, we examine *vacuum oil purifiers* that have been used since the late 1940's. Low-cost versions generally tend to suffer reliability problems. Therefore, the user should go through a well-planned selection process and should use a good specification. For a more rugged, mobile vacuum oil purifier see Figures 12-9 and 12-10. Good products will give long, trouble-free service.

A vacuum oil purifier operates on the principle of simultaneous exposure of the oil to heat and vacuum while the surface of the oil is extended over a large area. This differs from the other methods we have discussed in that it is a chemical separation rather than a physical one. Under vacuum, the boiling point of water and other contaminants is lowered so the lower boiling point constituents can be flashed off. Typical operating conditions are 170°F (77°C) and 29.6 in. Hg (10 Torr). Because water is removed as a vapor in a vacuum oil purifier, there is no loss of additives from the oil system. The distilled vapors are recondensed into water to facilitate rejection from the system. Non-condensibles such as air and gases are ejected through the vacuum pump.

Vacuum Oil Purifiers More Closely Examined

As illustrated in Figure 12-9, the typical components of a vacuum oil purifier are an inlet pump; a filter typically rated at 5 microns; some method of oil heating such as electric heaters, steam, hot water, or heat transfer fluid; a vacuum vessel; and a vacuum source such as a mechanical piston vacuum pump or water eductor. Vacuum oil purifiers may, or may not, incorporate a condenser depending upon the application. A discharge pump is employed to return the oil to the tank or reservoir, and an oil-to-oil heat exchanger may be employed for energy conservation.

Vacuum oil purification is applied across a broad spectrum of industries: power generation and transmission, automotive, aluminum, refining and petrochemicals,

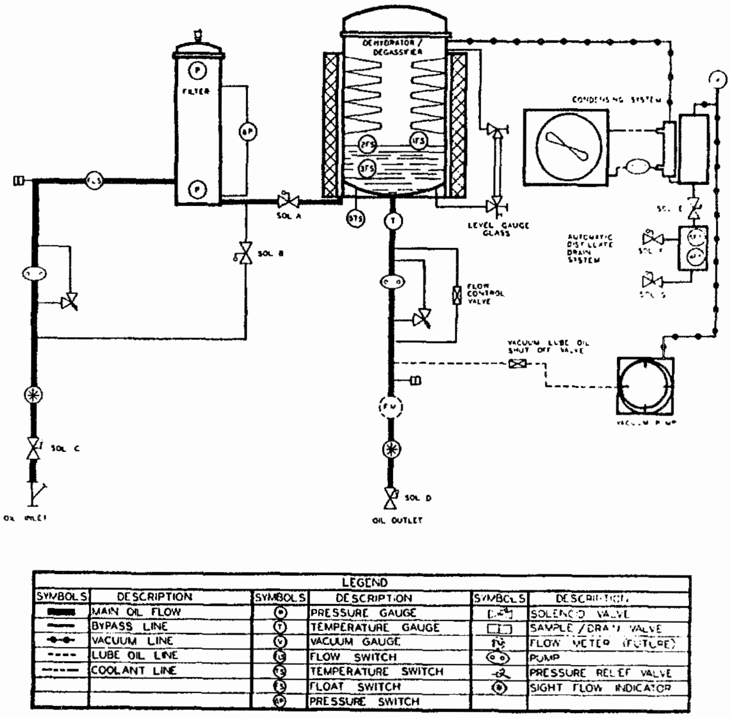

Figure 12-9. Typical flow diagram of a vacuum dehydrator system. *(Courtesy of Allen Filters, Inc. Springfield, MO.)*

steel, mining, construction, plastic injection molding, metalworking and food processing.

Vacuum oil purification is the only extended range method capable of removing free, emulsified, and dissolved water. Since vacuum oil purifiers can remove dissolved water, they can be operated intermittently without the danger of free water forming in the oil. Furthermore, they are the only method of oil purification that will simultaneously remove solvents, air, gases, and free acids.

In virtually all instances, a cost justification study by medium and large users of industrial oils will favor well-engineered vacuum oil purifiers over centrifuges, coalescers, and filter/dryers. Cost justification is further influenced in favor of vacuum oil purifiers by bottom-line analyses that look at the cost of maintenance labor and parts consumption.

Sound Alternatives Available For On-Stream Purification

Not every application needs or can justify a vacuum oil purifier. Because of monetary constraints, some users are willing to give up flexibility, but not effectiveness.

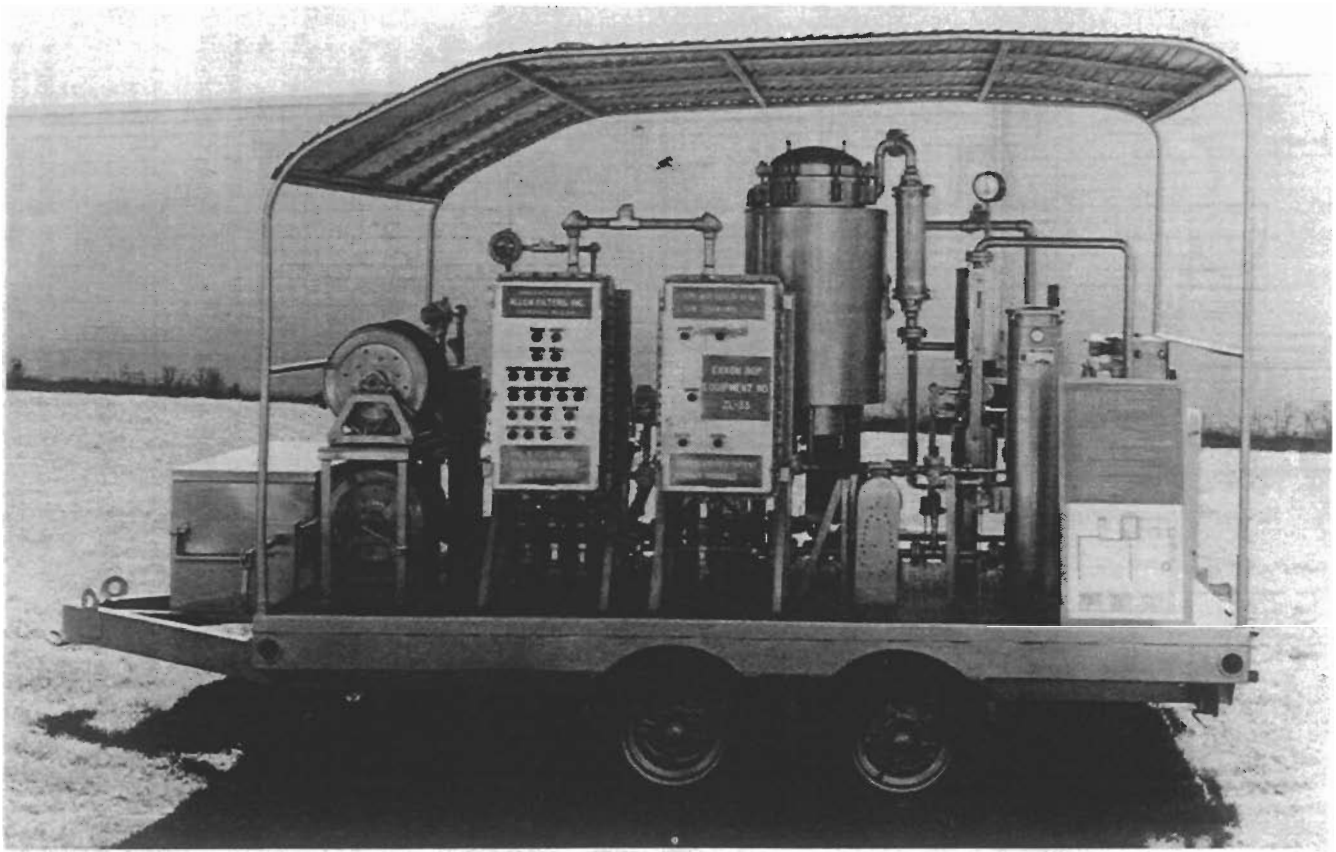

Figure 12-10. Mobile vacuum dehydrator. *(Courtesy of Allen Filters, Inc., Springfield, MO.)*

Other users may have troublesome machines that are continuously subject to free water contamination and require an inexpensive, dedicated dehydration device operated continuously to purify the lube oil system. These users may be well served by the very latest method of oil purification that operates on the chemical separation principle of *air stripping*. It is important to recognize that this chemical separation principle, too, removes free, dissolved, and emulsified water. It, therefore, ranks as a viable alternative and close second to vacuum oil purification as the preferred dehydration method, particularly for smaller systems.

Ambient air stripping units are intended for light-duty application on small or medium-sized reservoirs. They are specifically designed for dedicated use on individual machines for water removal only. No operator attention is required, and such units are simple to install. Since they are compact and lightweight, they may be set on top of a lube oil reservoir. They are available at low initial cost in the $12,000 to $22,000 range, and are extremely easy to maintain. These modern self-contained stripping units remove water at or above atmospheric pressure in the vapor phase and, therefore, conserve oil additives. Units similar to the one shown in Figures 12-11 and 12-12 are capable of removing free, emulsified and dissolved water to well below the saturation levels, and yield a product that is, in most cases, as good as fresh lubricant. Such units can readily reduce water concentration to below 15 wppm, which is considered safe for the long-term protection of fluid machinery.

Operating Parameters For State-of-Art Stripping Units

As depicted in Figure 12-11, air stripping units draw oil from the bottom of an oil reservoir by a motor-driven gear pump, which comprises the only moving parts of the unit. The oil is then forced through a filter that removes particulates and corro-

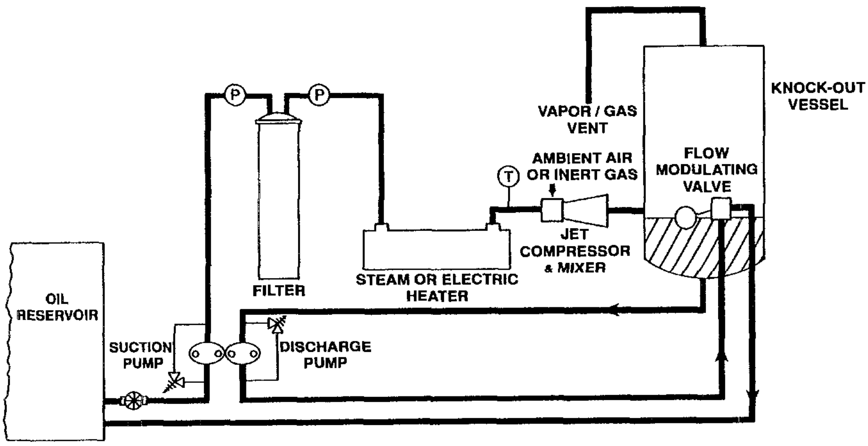

Figure 12-11. Schematic representation of a gas stripping device used for lube oil purification. (*Courtesy of Ausdel, PTY, Cheltenham, Victoria, Australia.*)

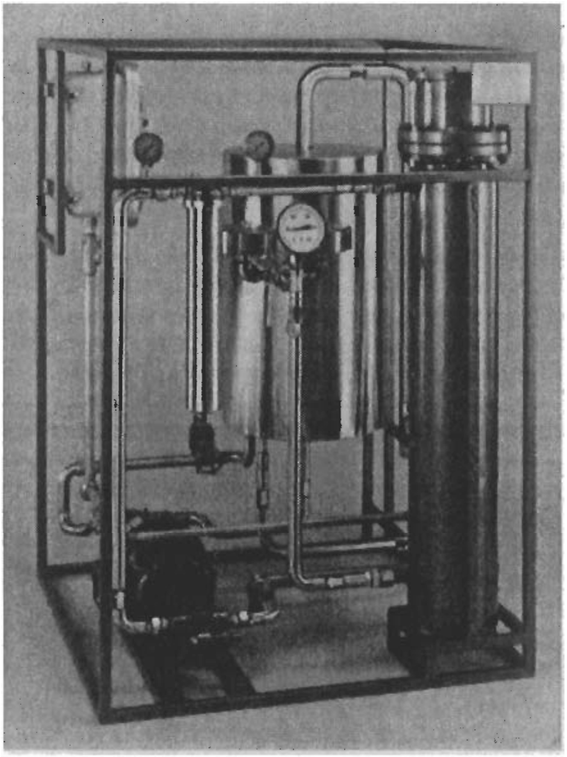

Figure 12-12. Typical, self-contained gas stripping device used for lube oil purification. ***(Courtesy of Ausdel, PTY, Cheltenham, Victoria, Australia.)***

sion products, and then to a steam or electric heater for temperature elevation. From there the oil goes to a very efficient mixer/contactor (jet pump) where ambient air, or low pressure nitrogen, is aspirated into the wet oil mixture. The air is humidified by the water in the oil during its period of intimate contact, and this is the method by which the oil is dehydrated. Since even relatively humid air can absorb even more moisture when heated, ambient air is usually a suitable carrier gas for this water stripping process. The wet air is then vented to atmosphere, while the oil collects in the bottom of a knock-out vessel. A gravity or pump-equipped return loop allows the now-dehydrated oil to flow back to the reservoir.

The choice of air versus nitrogen depends on the oxidation stability of the oil at typical operating temperatures between 140°F (60°C) and 200°F (93°C). The choice is also influenced by flammability and cost considerations.

Normally, low pressure or waste steam is used to heat the oil as it passes through the unit. Models ranging from 5 to 500 liters of oil per minute are manufactured in the U.S. and Australia. A number of different design options are available. For example, if electricity or steam costs are expensive; or, if hot oil returning to the reservoir is a problem, then an oil feed/effluent exchanger can be added to the basic model to significantly reduce heating costs.

Narrowing the Choice to State-of-Art Units

Vacuum oil purifiers and modern self-contained stripping units produce comparable oil quality, and are generally preferred over the less effective and maintenance-intensive physical separation methods. In general, the cost of a self-contained stripping unit (see Figure 12-12) is only a fraction of the cost of a vacuum oil purifier, and operating and maintenance costs are markedly lower due to its much simpler concept and construction.

Mobile vacuum oil purifiers offer substantial performance flexibility, especially when rapid de-watering is of importance. Vacuum units would be required in critical situations when an unexpected rate of contaminant ingression into an oil system occurs that is beyond the capabilities of a dedicated air/gas stripper. This is because the fixed geometry of the air stripper mixer/contactor does not provide the same flexible contaminant-handling capabilities inherent in a vacuum oil purifier. The vacuum units excel also in cases in which combustible gases are present, such as compressor seal oil applications. In these cases, the availability and cost of nitrogen must be considered for the air strippers; but with a vacuum oil purifiér, nitrogen use is a non-issue. Plants that intend to be at the forefront of technology should optimize machinery reliability by dedicating an air stripper to each reservoir and should use one or more mobile vacuum units as required to facilitate uptime on critical equipment until the next turnaround/planned maintenance period. The user should consider the foregoing discussion when deciding what type to utilize.

Cost Justifications Will Prove Merits of On-Stream Purification

The following cost justification will enable potential users to determine for themselves the value of dedicated on-stream oil purification units. On many occasions, plant personnel were astonished at the credits they could experience when they filled in their numbers on the tabular justification matrix reproduced on pages 504–506.

Synthetic Lubricants and Reliability Improvement*

Most plant engineers will agree that there's plenty of debate in today's lubrication world about whether it is better to use natural mineral oils or synthetic ones. In fact, the situation is quite polarized: Some users are highly "prejudiced" against synthetic oils—but can't tell you why; others believe that synthetic oils are better than mineral oils—but they, too, often don't have a clear understanding of the reasons.

Based on H. P. Bloch and A. R. Pate, Jr., "Consider Synthetic Lubricants For Process Machinery," *Hydrocarbon Processing*, January 1995, and H. P. Bloch and J. B. Williams, "Synthetic Lubricants Measure Up To Claims," *Chemical Engineering*, January 1995.

(*text continued on page 506*)

Air Stripper Return on Investment
Annual Cost Savings

		Example Plant	Your Plant
Oil			
Oil inventory in reservoir (liters)	1.	2,000	________
Annual number of oil changes WITHOUT purifier (once per 2 years)	2.	0.5	________
Annual number of oil changes WITH purifier (once per 10 years)	3.	0.1	________
Annual liters of oil saved: Line 1 × (Line 2 – Line 3)	4.	800	________
Cost/liter of oil	5.	$1.50	________
Annual value of oil saved	6.	$1,200	________
Labor charge to change oil (30 manhours @ $20/hr including overhead)	7.	$600	________
Annual value of labor saved: Line 7 × (Line 2 – Line 3)	8.	$240	________
Subtotal (Line 6 + Line 8)	9.	$1,440	________
Filters			
Annual number of filter changes WITHOUT purifier (3 per year)	10.	3	________
Annual number of filter changes WITH purifier (once per 2 years)	11.	0.5	________
Replacement filter cost (parts and labor)	12.	$600	________
Annual savings in replacement filter cost: Line 12 × (Line 10 – Line 11)	13.	$1,500	________
Planned Maintenance			
Scheduled major machine shutdown WITHOUT purifier (once per 2 years)	14.	0.5	________
Scheduled major machine shutdown WITH purifier (once per 10 years)	15.	0.1	________

Air Stripper Return on Investment (Continued)
Annual Cost Savings

		Example Plant	Your Plant
Maintenance labor to inspect and overhaul worn parts (100 man-hours @ $20/hr) per shutdown	16.	$2,000	______
Cost of worn parts	17.	$10,000	______
Value of lost production per shutdown	18.	$10,000	______
Annual planned maintenance credit (Line 16 + Line 17 + Line 18) × (Line 14 - Line 15)	19.	$8,800	______
Unplanned Maintenance			
Number of unscheduled major machine shutdowns for oil-related wear WITHOUT purifier (once per 5 years)	20.	0.2	______
Number of unscheduled major machine shutdowns for oil-related wear WITH purifier	21.	0	______
Value of lost production, parts and labor per event	22.	$100,000	______
Annual unplanned maintenance credit: Line 22 × (Line 20 - Line 21)	23.	$20,000	______
Total Annual Savings			
Line 9 + Line 13 + Line 19 + Line 23	24.	$31,740	______
Operating Costs			
Annual on-stream hours	25.	8,000	______
Power consumption (KW)	26.	1.1	______
Cost of power ($/KWH)	27.	0.04	______
Annual power cost: Line 25 × Line 26 × Line 27	28.	$352	______
Steam consumption (kg/hr)	29.	75	______
Cost of steam ($/ton)	30.	4.0	______
Annual steam cost: Line 25 × Line 29 × Line 30/1000	31.	$2,400	______
Nitrogen consumption (m^3/hr)	32.	2.5	______

(text continued on next page)

Air Stripper Return on Investment (Continued)
Annual Cost Savings

		Example Plant	Your Plant
Cost of nitrogen ($\$/m^3$/zero if air is used)	33.	0.08	________
Annual nitrogen cost: Line 25 × Line 32 × Line 33	34.	$1,600	________
Subtotal operating costs: Line 28 + Line 31 + Line 34	35.	$4,352	________
Net Annual Benefit			
Total benefits (Line 24)	36.	$31,740	________
Total operating costs (Line 36)	37.	$4,352	________
Net annual benefit (Line 36 × Line 37)	38.	$27,388	________
Profitability Calculation			
Cost of air stripper purifier	39.	$17,250	________
Payback period (months) (Line 38/Line 39) × 12	40.	7.6	________
Annual return on air stripper purifier	41.	>100%	________

(*text continued from page 503*)

Indeed, although synthetic lubricants have gained considerable acceptance in many plants, there are still widespread misconceptions that impede their greater acceptance. For example, one erroneous notion is that to justify the cost of a synthetic lubricant—about $15/gal in 1997—the drainage or replacement interval should be five times that of a mineral oil costing $3/gal.

But, switching from mineral oils to synthetics often smoothens the operation of a machine and reduces its vibration, resulting in lower energy consumption and less maintenance. Further, the time period for oil changes is extended and, overall, less spent oil has to be disposed of. Since machines run more efficiently and smoothly, their lives are also extended.

There's also the wrong notion that synthetic lubricants are only cost-effective in very limited applications. In fact, judicious application of properly formulated synthetic lubricants can benefit a wide spectrum of process machinery. This informed usage is very likely to drive down overall maintenance and downtime expenditures and can markedly improve plant profitability.

Origin of Synthetic Lubes

Synthetic-based fluids, used in the production of synthetic lubricants, are manufactured from specific chemical compounds that are usually petroleum derived. The base fluids are made by chemically combining (synthesizing) various low molecular weight compounds to obtain a product with the desired properties. Thus, unlike petroleum oils which are complex mixtures of naturally occurring hydrocarbons, synthetic base fluids are man-made and have a controlled molecular structure with predictable properties.[12]

There is no typical synthetic lubricant. The major classes are as different from each other as they are from petroleum lubricants. Synthesized base fluid are classified as follows:

1. Synthesized hydrocarbons (polyalpha-olefins)
2. Organic esters (diesters and polyol esters)
3. Polyglycols
4. Phosphate esters
5. Silicons
6. Blends.

The first four base fluids account for more than 90% of the synthetic fluids used worldwide. The first three contain only atoms of carbon, hydrogen and oxygen. The first two are of greatest interest to machinery engineers in modern process plants.

Examining Synthetic Lubes

Understanding the principal features and attributes of the six base fluids will place the potential user in a position to prescreen applicable synthetics and to question suppliers whose offer or proposal seems at odds with these performance stipulations.

Synthetic hydrocarbon fluids (SHF), such as those with a polyalphaolefin (PAO) base, provide many of the best lubricating properties of petroleum oils but do not have their drawbacks. (Even the best petroleum oils contain waxes that gel at low temperatures and constituents that vaporize or easily oxidize at high temperatures.) The SHF base fluids are made by chemically combining various low molecular weight linear alpha olefins to obtain a product with the desired physical properties. They are similar to cross-branched paraffinic petroleum oils because they consist of fully saturated carbon and hydrogen.

These man-made fluids have a controlled molecular structure with predictable properties. They are available in several viscosity grades and range from products for low temperature applications to those recommended for high temperature uses. They are favored for their hydrolytic stability, chemical stability and low toxicity.

Organic esters are either dibasic acid or polyol types. Dibasic acids have shear-stable viscosity over a wide temperature range (–90°F to 400°F), high film strength, good metal wetting properties and low vapor pressure at elevated temperatures. They easily accept additives, enhancing their use in many commercial applications and especially as compressor lubricants.

Polyol esters have many of the performance advantages of dibasic acid esters and can be used at even higher temperatures. They are used principally in high-temperature chain lubricants, for industrial turbines and in some aviation applications.

Polyglycols were one of the first synthetic lubricants developed. The polyglycols can be manufactured from either ethylene oxide, propylene oxide or a mixture of both. The propylene oxide polymers tend to be hydrocarbon soluble and water insoluble, while the ethylene oxide tends to be water soluble and hydrocarbon insoluble. In many applications, the physical properties of the finished product can be engineered by adjusting the ratio of ethylene oxide and propylene oxide in the final molecular structure.

Polyglycols have excellent viscosity and temperature properties; they are used in applications from –40°F to 400°F and have low sludge-forming tendencies. A major application for polyglycol lubricants is in compressors that handle hydrocarbon gases. This is due to the non-hydrocarbon-diluting properties inherent in polyglycols. The polyglycols' affinity for water results in poor water separability.

Phosphate esters are organic esters that, when used with carefully selected additives, provide a group of synthetic fluids that can be used where fire resistance is required. Even when ignited, the phosphate esters will continue to burn only if severe conditions required for ignition are maintained. Some phosphate esters are less stable in the presence of moisture and heat. The products of the resulting degradation are corrosive and will attack paints and rubbers. The poor viscosity index (VI) limits the operating temperature range for any given phosphate ester product.

Silicones have been in existence for many years and offer a number of advantages as lubricants. Silicones have good viscosity versus temperature performance, excellent heat resistance, oxidative stability and low volatility. Silicones are chemically inert and have good elastomer compatibility. Poor metal-to-metal lubricating properties and high cost limit their use to specialized applications where their unique properties and high performance can be justified.

Blends of the synthetic lubricants with each other or with petroleum lubricants have significant synergistic results. In fact, many of the synthetic lubricants being sold consist of a blend of two or more base materials to enhance the properties of the finished product. For instance, the combination of PAO and diester gives superior seal protection, since any contraction of the seal due to the PAO is balanced by swelling induced by the diester. Also, additives are more easily soluble in diesters than in PAO, permitting the formulation of stable PAO-diester blends that can be operated over a wide temperature range.

Synthetic lubricants have been steadily gaining industrial acceptance since the late 1950s. In many applications today, they are the specified lubricant of the compressor manufacturer. This is especially true in rotary screw and rotary vane air compressors.

While the greatest industrial acceptance has been with air compression, many other industrial applications can be economically justifiable. Synthetic lubricants are currently being used in compressors processing such diverse materials as ammonia, hydrogen, hydrocarbon gases, natural gas, hydrogen chloride, nitrogen and numerous other gases.

Synthetic lubricants are not limited to compressors but are used in gear boxes, vacuum pumps, valves, diaphragm pumps and hydraulic systems. Synthetic lubricants are being used in applications that need more efficient, safe lubrication or where the environmental conditions preclude the use of traditional petroleum products.

Properties and Advantages

Synthetic lubricant fluids provide many of the best lubricating properties of mineral oils but do not have their drawbacks. In fact, synthetics have these advantages over comparable petroleum-based lubricants:

- Improved thermal and oxidative stability
- More desirable viscosity-temperature characteristics
- Superior volatility characteristics
- Preferred frictional properties
- Better heat transfer properties
- Higher flash point and autoignition temperatures.

Experience clearly shows that these advantages result in the following economic benefits:

- Increased service life of the lubricant (typically four to eight times longer than petroleum lubricants)
- Less lubricant consumption due to its low volatility
- Reduced deposit formation as a result of good high-temperature oxidation stability
- Increased wear protection resulting in less frequent maintenance
- Reduced energy consumption because of increased lubricating efficiency
- Improved cold weather flow properties
- Reduced fire hazard resulting in lower insurance premiums
- Higher productivity, lower manufacturing costs and less downtime because machines run at higher speeds and load with lower temperatures
- Longer machinery life because less wear results in more production during life of machine and tools.

Synthetic lubricant base stocks, while possessing many of the attributes needed for good lubrication, require fortification with additives relative to their intended use.

Additives are the Key

The most important ingredients of synthetic oils are the additives. Some additives impart new and useful properties to the oil, others enhance properties that are already present, while others minimize undesirable effects during the product's life. Additives may be present from a few hundredths of a percent to 30% or more.

Typically, mineral and synthetic oils contain at least a rust, corrosion and oxidation inhibitor. High-performance oils are blended with one or more additives to impart desirable anti-wear, defoaming, wetting (oiliness), anti-welding and anti-seizure, metal bonding and extreme-pressure properties. The additives are selected based on performance and price considerations.

However, formulating oils is not an exact science; it is an art that involves a great deal of trial, error and experience. There are many trade-offs that have to be overcome when formulating high-performance oils. Too often, an additive that enhances the performance of one property is detrimental to another. Thus, it is important to obtain synergistic combinations where one additive improves the performance of the other without introducing negative effects.

Most synthetic lubricants, such as the well-proven PAO-diester blends, contain a variety of proprietary additives to provide the needed performance. Additionally, the lubricant must ionically bond to bearing metals to reduce the coefficient of friction, and to greatly increase the oil film strength.

This ionic bonding results in a tough, tenacious, yet slippery film, which makes equipment run cooler, quieter, smoother, more efficiently and longer. In other words, these additives "micropolish" surfaces, reducing friction, vibration and energy consumption.

The most effective synthetic lubricants for chemical process machinery are ones that possess high film strength and oxidation stability, are noncorrosive, and have superior water resistance. Engineers should evaluate these lubricants, since a growing and compelling body of evidence suggests that PAO or diester lubricants, or a combination of these base stocks, will significantly reduce bearing and gear operating temperatures. An experienced formulator thus takes into consideration a range of requirements:

Dispersion of Contaminants. It is important to keep internally and externally generated oil-insoluble deposit-forming particles suspended in the oil. This mechanism reduces the tendency of deposits, which lower operating efficiency, to form in critical areas of machinery. Additives that impart dispersing characteristics are called "dispersants" and "detergents." A dispersant is distinguished from a detergent in that it is nonmetallic, does not leave an ash when the oil is burned and can keep larger quantities of contaminants in suspension.

Protecting the Metal Surface From Rust and Corrosion. Humidity (water) type rust and acid type corrosion must be inhibited for long surface life. An oil film itself is helpful but this film is easily replaced at the metal surface by water droplets and acidic constituents. Additives that have an affinity for a metal surface, more so than water or acids, are used in oils to prevent rust and corrosion and are generally referred to as simply "rust inhibitors."

Oxidative Stability. Oils tend to thicken in use, especially under conditions where they are exposed to the atmosphere or where oxygen is present. This phenomenon is chemically termed "oxidation."

Oxygen reacts with the oil molecule initiating a chain reaction that makes the molecule larger, thereby decreasing fluidity. Conditions that assist the oxidation process are heat, oxidation catalyzing chemicals, aeration, and perhaps other mechanisms that allow the oxygen to easily attach itself. Additives that retard the oxidation process are termed "oxidation inhibitors."

Wear Prevention. Inevitably the metal surfaces being lubricated come in contact. Whenever the speed of relative motion is low enough, the oil film does not stay in place. This can also happen if the loading on either or both surfaces is such that the oil film tends to be squeezed out. When moving metal surfaces come in contact, certain wear particles are dislodged and wear begins. Additives that form a protective film on the surfaces are called "anti-wear agents."

Viscosity index improvers function to improve viscosity/temperature relationships, that is, to reduce the effect of temperature on viscosity change.

Foam suppressants allow entrained air bubbles to collapse more readily when they reach the surface of the oil. They function by reducing surface tension of the oil film.

Oiliness additives are materials that reduce the oil friction coefficient.

Surfactants improve the ability of the oil to "wet" the metal surface.

Alkalinity agents imparts alkalinity or basicity to oils where this is a desirable feature.

Tackiness agents impart stringiness or tackiness to an oil. This is sometimes desirable to improve adhesive qualities.

Obviously then, the lubricant supplier or formulator has to choose from a number of options. There are technical considerations to weigh and compromises to make. Close cooperation between supplier and user is helpful; formulator experience and integrity is essential.

Case Histories

The following are highlights from the many successful case histories in the 1975 to 1995 time frame.

Circulating Oil System for Furnace Air Preheaters. Several major refineries in the U.S. and Europe had experienced frequent bearing failures on these slow-rotating heat exchangers while operating on the manufacturer-recommended mineral oil. With bearing housings typically reaching temperatures around 270°F, the cooled and filtered mineral oil would still overheat to the point of coking. Bearing failures after six months of operation were the norm. After changing to a properly formulated synthetic, a lubricant with superior high-temperature capabilities and low volatility, bearing lives were extended to several years. One refinery alone has documented savings of approximately $120,000 per year since changing lubricants.

Right Angle Gear Drives for Fin Fan Coolers. A European facility achieved a disappointing mean-time-between-failures (MTBF) of only 36 months on 36 hypoid gear sets in a difficult to reach elevated area. In fact, using mineral oil (ISO VG 150), a drain interval of six months was necessary to obtain this MTBF. Each oil change required 12 man-hours and temporary scaffolding at a cost of $1,000. Changeover to an appropriate synthetic, i.e., a synthetic with optimized temperature stabilizers, wear reducers and oxidation inhibitors, has allowed drain intervals to be increased to two years while obtaining a simultaneous increase to three years MTBF. Detailed calculations showed a net benefit of $1,950 per year per gear set. Combined yearly savings: $70,200 with no credit taken for power reduction or avoided production curtailments.

Plantwide Oil Mist Systems. An oil mist lubrication system at a Southeast Texas chemical plant experienced an unscheduled shutdown as a result of cold weather. Twenty-seven mist reclassifiers in this system were affected. These reclassifiers provided lubrication to several fin fans, two electric motors and the rolling element bearings in 14 centrifugal pumps. Wax plugging of the mist reclassifiers brought on by the cold weather caused the unexpected shutdown. As a result, several bearings failed because of lubricant starvation. An ISO VG 68 grade conventional mineral oil was the source of the wax.

The oil mist system had to be isolated and blown out to avoid further bearing failures. In addition to the downtime costs, significant labor and hardware costs were required to restore the unit to normal operation.

For this reason, a synthetic wax-free lubricant replaced the mineral oil. Neither the oil feed rate nor the air-to-oil ratio required adjusting after switching to the synthetic.

Since converting in March 1979 to a diester-based oil mist system, the following has resulted:

- No cold weather plugging of the mist reclassifiers has been experienced.
- No lubricant incompatibility has been detected with other components of the oil mist system.
- The synthetic lubricant is providing proper bearing wear protection as evidenced by no increase in required maintenance for pumps, fans, or motors serviced by the oil mist system.
- Downtime, labor and hardware replacement costs attributed to cold weather operational problems have been eliminated.
- Savings in contractor and plant manpower used to clean the reclassifier equaled $25,100 per year.
- Two failures of pumps and motors were assumed to be prevented via use of wax-free lubricant. The savings equaled $7,000 per year.

Total net credit has been $49,375 per year. This does not include any process losses associated with equipment outages.

Pulverizing Mills in Coal-Fired Generating Plant. A large coal-fired power generating station in the southwestern U.S. was having lubrication problems with their coal pulverizing mill. The equipment, a bowl mill pulverizer, was experiencing the following problems lubricating the gears that drive the mill:

- The lubricant was losing viscosity and had to be changed every four to six months.
- Air entrainment in the lubricant was causing cavitation in the pumps that circulated the lubricant.
- The gears were experiencing an unacceptable level of wear as measured by a metals analysis on the lubricant.
- On very cold mornings, the lubricant was so viscous it had to be heated before the unit could be put in service.
- The initial viscosity of the petroleum-based lubricant varied significantly.

After evaluating the options, it was decided that a synthetic-based lubricant offered the best solution. In cooperation with a major synthetic lubricant manufacturer, the facility decided a synthetic hydrocarbon-base stock with the proper additive package would be the best choice. Additive package concentrations were evaluated in a number of bowl mills simultaneously to establish the optimum level and composition. Figure 12-13 shows the dramatic effect on metal gear wear accomplished over a 1,000 hour trial period.

The synthetic hydrocarbon-base stock has proven to be extremely shear resistant. One particular bowl mill has been closely monitored during 54 months of operation (Figure 12-14) to establish viscosity stability. The data represent only operating hours, not total time elapsed, since the unit is not operated continuously. The performance has been excellent and lubricant life has exceeded 60 months.

The synthetic hydrocarbon lubricant was compared to two petroleum-based lubricants supplied by major oil companies. The tests were run on three bowl mills that

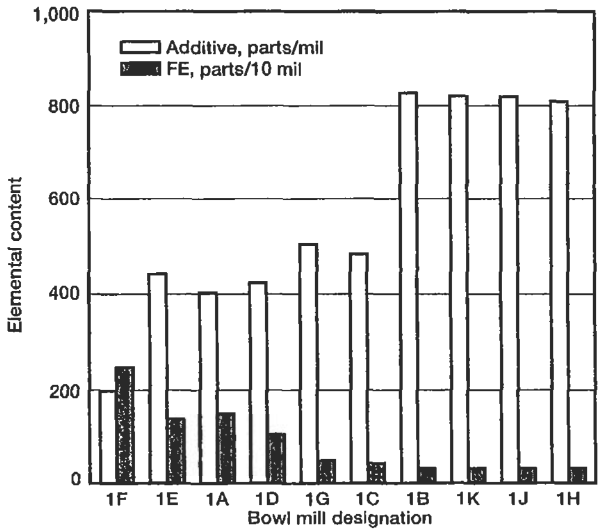

Figure 12-13. Bowl mill wear, 1,000 operating hours.

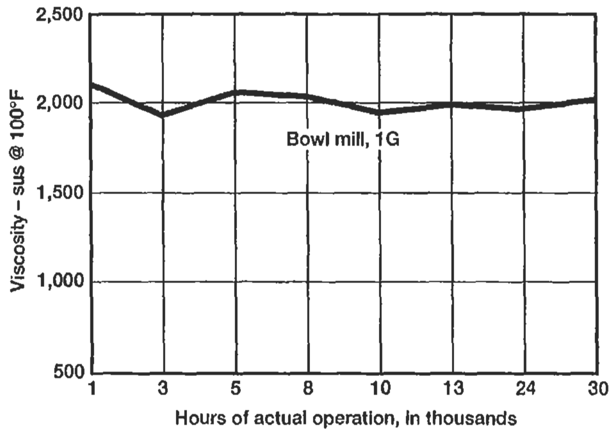

Figure 12-14. Viscosity stability, bowl mill.

had recently been reworked and tested. All three bowl mills were fed the same amount of coal during the test period. All three gear oils were the same ISO 320 viscosity grade.

The average current draws were:

Product	Amps
Petroleum #1	70
Petroleum #2	75
Synthetic hydrocarbon gear oil	68

The lower amp difference shown by the synthetic hydrocarbon is the result of the lower coefficient of friction shown in Table 12-2.

In summary, the synthetic hydrocarbon gear oil has solved the original problems and provided additional benefits not anticipated. The switch to synthetic lubricants has clearly improved performance and achieved significant savings in operating costs, as shown in the following tabulation.

Table 12-2
Physical Properties of ISO VG 320 Gear Oil

	Petroleum	Synthetic hydrocarbon
Viscosity index	95	140
Thermal conductivity, Btu/hr/ft^2	0.071	0.085
Coefficient of friction	0.101	0.086
Pour point, °F	+5	−45

The extended drain interval provides savings in three areas:

1. Lubricant consumption cost savings:

Petroleum oil cost per gal	$4.00
Petroleum oil changes per yr	2
Volume of gear box, gal	300
Petroleum oil cost per yr	
($4.00/gal)(2 changes/yr)(300 gal/unit) =	$2,400
Synthetic oil cost per gal	$16.00
Synthetic oil changes per yr	0.2
Volume of gear box, gal	300
Synthetic oil cost per yr	
($16.00/gal)(0.2 changes/yr)(300 gal/yr) =	$960

Annual savings on lubricant cost—$1,440 per unit

2. Reduced maintenance cost savings

Petroleum oil changes per yr	2
Maintenance cost per change	$500
Petroleum oil maintenance cost per yr	
(2 changes/yr)($500/change) =	$1,000
Synthetic oil changes per yr	0.2
Maintenance cost per change	$500
Synthetic oil maintenance cost per yr	
(0.2 changes/yr)($500/change)	$100

Annual savings in scheduled maintenance costs—$900

3. Lubricant disposal costs

Petroleum oil used per yr, gal	600
Disposal cost per gal	$0.50
Cost of disposal	$300
Synthetic oil used per yr, gal	60
Disposal cost per gal	$0.50
Cost of disposal	$30

Annual savings in disposal cost per year—$270

The reduction in energy consumption also provides significant savings:

Average annual power cost using petroleum oil lubricant	$33,278
Average annual power cost using synthetic hydrocarbon lubricant	$31,211

Annual savings in power consumption—$2,067

The total annual savings for all of the above categories amount to $4,677. In addition, savings in reduced wear and thus fewer repairs are certain to be realized.

Vibration Performance Improved With Synthetics

Rolling element bearings can experience significant reductions in vibration amplitude and, thus, increased life expectancy when lubricated with synthetic oils. The higher film strengths of the synthetic oils reduce the severity of impact when the rolling elements of a bearing move across spall marks and other discontinuities. In fact, many defective bearings have been "nursed along" by the high-strength bonding characteristics of these films.

A major gas-transmission company documented the vibration-shock pulse activity of a compressor turbocharger and oil supply pump before and after switching to a synthetic oil. After the changeover, a strong, tenacious and slippery oil film reduced vibration severity by "peening over" asperities on the various metal surfaces of bearings.

A variety of equipment has been rejuvenated by switching to synthetics:

- An external washer-filter in a Willamette, Ind., pulp mill had been operating well above 1.5 G for shock pulse activity and 0.2 in./s for vibration. Upon switching to a synthetic oil, the values dropped to 0.75 G for shock pulse activity and 0.15 in./s for vibration. The change occurred immediately and continued for a month
- A multistage air blower at a fiber spinning plant had its vibration decreased from 0.155 in./s to 0.083 in./s by switching oils. Further, the temperature of the bearing dropped by 20°F
- A 10-hp centrifugal pump had an acceptable vibration of 0.068 in./s, but the bearing housing temperature of 175°F was borderline. After changing to a synthetic oil, the vibration dropped to 0.053 in./s, and bearing housing temperature went down to 155°F. Further, the motor amperage was cut from 5.7 A/phase to 4.4 A/phase

To obtain such improvements, however, it is important to choose a well-formulated synthetic oil. As typified in Figure 12-15, there are noticeable differences in the operating temperatures of spur gear units, reactor pump bearings and bevel gear enclosures using products from different vendors.

Further, while there are many excellent products on the market today, many may not be appropriate for use in process machine applications. For example, high-film-strength oils based on extreme pressure (EP) technology and intended for gear lubrication typically incorporate additives containing sulfur, phosphorus and chlorine. These EP industrial oils cannot be used as bearing lubricants for pumps, air compressors, steam turbines, high-speed gears and similar machinery, since the sulfur, phosphorus and chlorine will cause corrosion at high temperatures and in moist environments.

Testing Provides Proof

The best indication of which oil—synthetic or mineral—will excel in an application can be obtained by comparing their specific performance. There are numerous laboratory tests that are good indicators of how well an oil will perform in service. These includes tests for viscosity, pour point, residue, strip corrosion, rust, demulsibility and so on.

On the other hand, it should be pointed out that these tests are only predictors. Realistic tests under simulated field conditions are better, while the true measure of a lubricant's performance can only be determined in actual service. For example, in 1992, Kingsbury, Inc., completed the testing of a well-compounded ISO Grade 32 synthetic lubricant in a thrust-bearing test machine.

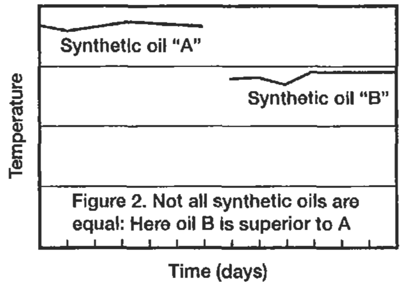

Figure 12-15. Not all synthetic oils are equal. Here oil B is superior to A.

At low speeds and loads, there appeared to be little difference between this lubricant and identical premium-grade mineral oils of the same viscosity. However, at high loads and speeds, above 550 psi and 10,000 rpm, Kingsbury found that the synthetic oil cut bearing temperature by 15°F, and decreased frictional losses by 10%.

Similarly, engineers from SKF and Exxon conducted a series of tests on rolling contact bearings.[13] The objective was to compare the properties of a specially formulated diester lubricant with those of a premium-grade mineral oil that was in service in an Exxon petrochemical plant.

Two synthetic lubricants and two mineral oils of varying viscosities were experimentally compared. The test results indicated that the synthetic lubricant, having a viscosity of 32 centistokes (cSt) at a temperature of 100°F, offered long-term surface protection equivalent to that of the base line mineral oil with a viscosity of 68 cSt, without reducing bearing surface life below the theoretically estimated levels.

The same good wear protection could not be achieved with a reduced viscosity mineral oil. The use of the lower viscosity synthetic lubricating fluid could provide projected energy savings of $140,000 per year—prorated to 1998 dollars, when the test was conducted.

Other companies, too, have been operating successfully with synthetic oils for many years, and are reluctant to publicize their experience so as not to give away a competitive advantage. Suffice it to say that a forward-looking process plant needs to explore the many opportunities for often substantial cost savings that can be achieved by judiciously applying properly formulated synthetic lubricants.

Automatic Grease Lubrication as a Reliability Improvement Strategy

Experienced maintenance and reliability professionals have seen rapid progress from reactive maintenance through preventive, predictive and "hybrid" approaches toward today's proactive maintenance methods. Recognizing that there are still some cost reduction measures that could—and should—be implemented, a number of "Best-of-Class" companies are now focusing on maintenance as part of strategy and profit potential. This realization has led these companies to fundamentally reassess their lubrication options for both new as well as existing machinery.

Lubrication Should Not Be An Afterthought

The most advanced, truly bottomline-oriented owners/purchasers of high-speed paper producing machinery are among a growing number of buyers that base their specification and procurement decisions on life cycle cost calculations. These calculations have shown, in the majority of cases, the long-range maintenance and downtime cost avoidance advantage of incorporating automatic oiling or grease feed provisions in modern process machines. Best-of-class design contractors and owner companies insist on automatic lubrication to be part of the design specification and permanent plant operating strategy.

This new thinking supersedes the old notion that lubrication automation is difficult to cost-justify, or that automatic lubrication provisions can be retrofitted as needed! In a highly cost-competitive environment or at a time when profit margins can vanish because of a single, unplanned outage event, buying equipment with proven lubrication provisions makes eminent sense. Experience shows that procurement of suitable automated lubricant delivery units at the inception of a project may well be the only low-risk opportunity that presents itself to buyers who take reliability seriously. Buying critically important machinery with the thought that upgrading to full automation at a later date will always be an option could be fallacious and may prove to be a costly mistake.

Bearing Manufacturers Prefer Automatic Lube Option

The disadvantages of manual lubrication have long been recognized by the leading bearing manufacturers. As can be seen from Table 12-3, the service life of rolling element bearings with automatic grease feed provisions ranks well ahead of most other means of lubricant delivery. This is why many European process plants much prefer engineered automatic lube application systems over traditional static oil sumps or manual regreasing. These plants are often especially mindful of the shortcomings of single-point automatic lubricators. Although occasionally found in the United States, these spring or gas-pressurized plastic grease containers offer little control over grease quantity and grease homogeneity. Greases under constant pressure tend to separate into their respective oil and soap constituents. This is highly undesirable since the virtually all-soap matrix is now likely to enter the bearing without oil. A properly engineered grease injection system will provide near-instantaneous pressure pulses with adjustable rest periods interspersed to suit the requirements of a specific application.

Although static oil sumps are still used in the majority of centrifugal pumps installed in U.S. paper mills, petrochemical facilities, and general process plants, the experience of Scandinavian pump users should prompt a fundamental reassessment and serious questioning of the role of contamination-prone static oil sumps in an era of reduced maintenance manpower availability. Moves away from "sump, occasion-

Table 12-3
Lubrication Methods Ranked by Order of Decreasing Service Life. (Ref. 14)

Decreasing service life**	Oil: Rolling Bearing Alone	Grease: Rolling Bearing Alone
	Circulation with filter, automatic oiler	Automatic feed
	Oil-air	
	Oil-mist	
	Circulation without filter*	
	Sump, regular renewal	Regular regreasing of cleaned bearing
		Regular grease replenishment
	Sump, occasional renewal	
		Occasional renewal
		Occasional replenishment
		Lubrication for-life

**By feed cones, bevel wheels, asymmetric rolling bearings.*
***Condition: Lubricant service life < fatigue life.*

al renewal" to "automatic grease feed" as ranked in Table 12-3 have resulted in reliability increases that can no longer be overlooked. This readjusted thinking has paid off handsomely in a number of Finnish and Swedish paper mills by allowing manpower reductions and resulting in plant availability extensions.

Comparing Manual and Automatic Grease Lubrication Provisions

Three principal disadvantages of manual lubrication are generally cited:

- Long relubrication intervals allow dirt and moisture to penetrate the bearing seals. Well over 50% of all bearings experience significantly reduced service life as a result of contamination.
- Overlubrication occurring during grease replenishment causes excessive friction and short-term excessive temperatures. These temperature excursions cause oxidation of the oil portion of the grease.
- Underlubrication occurring as the previously applied lube charge is being depleted at this time and prior to the next regreasing event.

In contrast, automated lubrication has significant technical advantages. Time and again, statistics compiled by SKF (Figure 12-16) and other major bearing manufacturers have shown lubrication-related distress responsible for at least 50%, and perhaps as much as 70% of all bearing failure events worldwide. Thoroughly well-engineered automatic lubrication systems, applying either oil or grease, are now

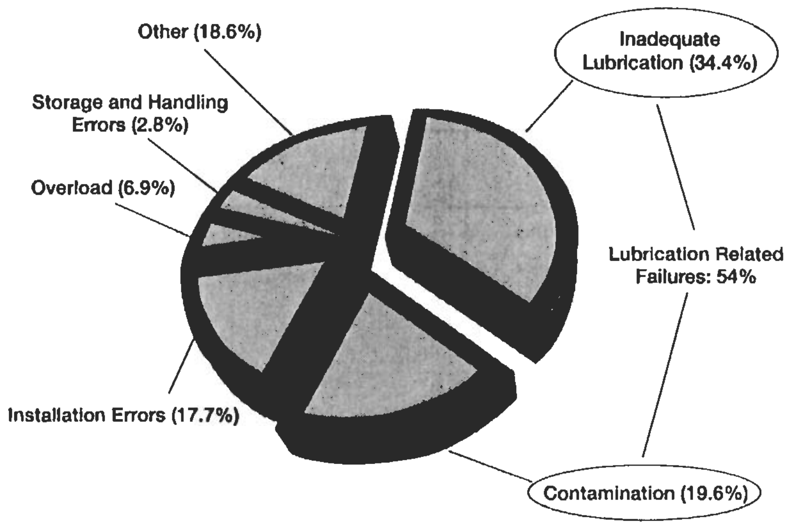

Figure 12-16. Bearing failures and their causes. ***(Courtesy of SKF USA, Inc. also TAPPI 1995 Engineering Conference Proceedings.)***

available to forward-looking, bottomline-oriented user companies. These systems (Figure 12-17) ensure that:

- The time elapse between relubrication events is optimized.
- Accurately predetermined, metered amounts of lubricant enter the bearing "on time" and displace contaminants.
- The integrity of bearing seals is safeguarded.
- Supervisory instrumentation and associated means of monitoring are available at the point of lubrication for critical bearings.

How Automated Lubrication Works

Depending on the type of installation, engineered lubricant injection systems are configured for either grease or liquid oil delivery. Modular in design and easily expandable, they are suitable for machinery with just a few lubrication points, as well as installations covering complete manufacturing or process plants involving thousands of points. Automated grease lubrication systems are designed for the periodic lubrication of rolling element bearings, as in the centrifugal pump depicted in Figure 12-18, or for different types of sleeve bearing. Also, automated grease lubrication systems are used on guides (shown in the soot blower in Figure 12-19) and on open gears, chains, and coupling devices.

Depending on plant and equipment configuration, engineered automatic lubrication systems consist of a single or multi-channel control center (Figure 12-17, Item 1), one or more pumping stations (Item 2), appropriate supply lines (Item 3), tubing

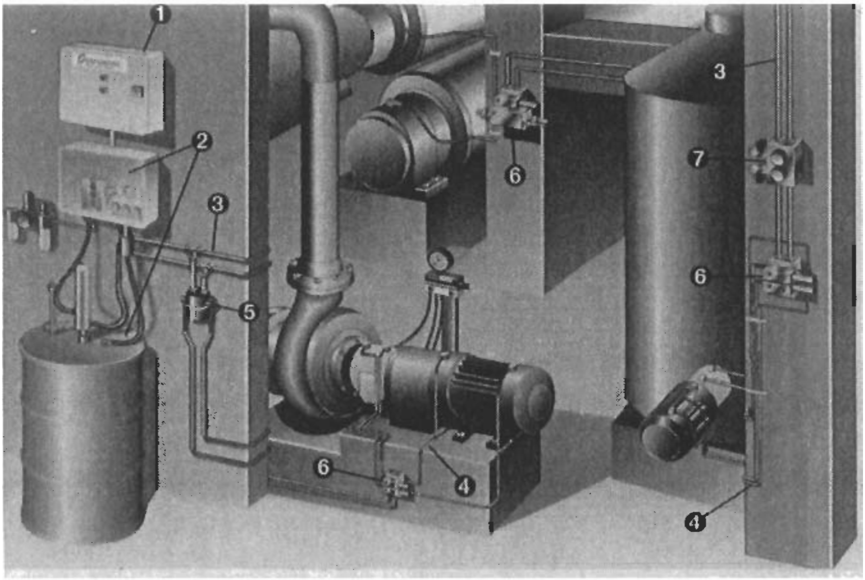

Figure 12-17. Single-header, state-of-the-art automatic lubrication system. *(Courtesy of Safematic, Muurame, Finland and Alpharetta, GA.)*

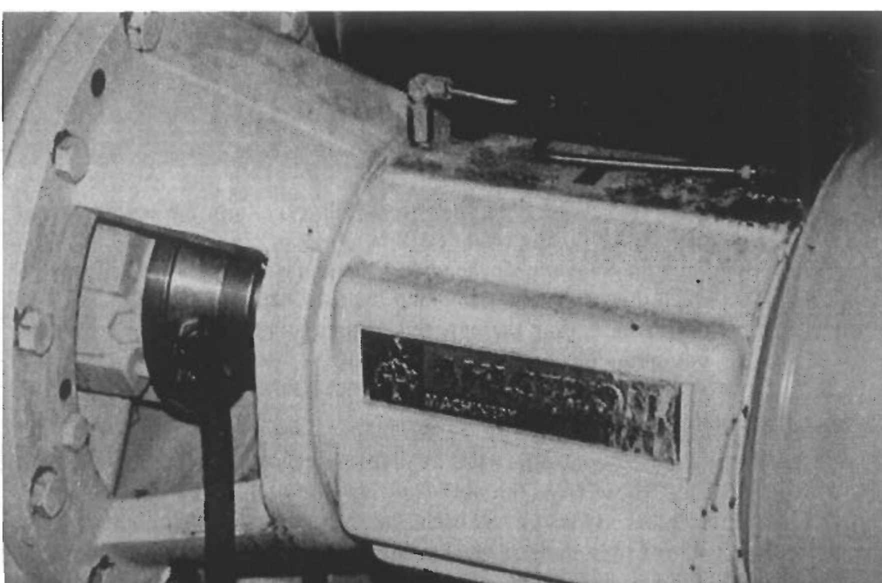

Figure 12-18. Centrifugal pump with automated grease lubrication. *(Courtesy of Safematic, Muurame, Finland and Alpharetta, GA.)*

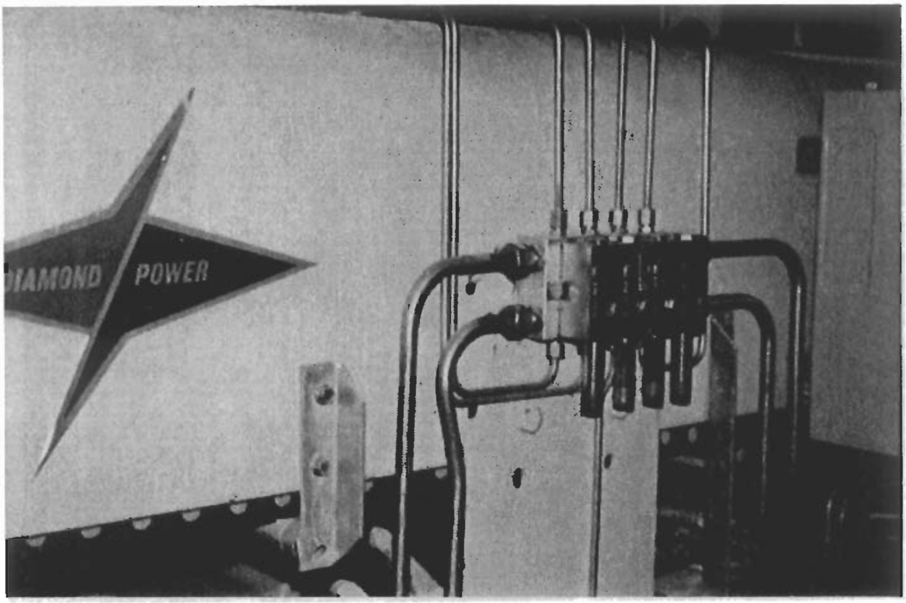

Figure 12-19. Soot blower assembly fitted with automatic grease lubrication provisions. *(Courtesy of Safematic, Muurame, Finland and Alpharetta, GA.)*

(4), which links a remote shutoff valve (5) and lubrication dosing modules (6), and also interconnects dosing modules and points to be lubricated. Different sized dosing modules are used to optimally serve bearings of varying configurations and dimensions. The dosing modules themselves are individually adjustable to provide an exact amount of lubricant and to thus avoid overlubrication. A pressure sensing switch (Item 7), completes the system.

The control center starts up a pump that feeds lubricant from the barrel through the main supply line to the dosing modules. When pressure in the system rises to a preset level, the pressure switch near the end of the line transmits an impulse to the control center, which then stops the pump and depressurizes the pipeline. The control center now begins measuring the new pumping interval. If for some reason the pressure during pumping does not rise to the preset level at the pressure switch, an alarm is activated and the lubrication center will not operate until the problem has been rectified and the alarm subsequently reset.

Special multi-channel controllers are available with state-of-the-art automatic lubrication systems. These have the ability to provide lubrication to installations requiring a variety of lube types or consistencies. Even different timing intervals can be controlled from a single multi-channel controller location. These systems have proven their functional and mechanical dependability in operating environments ranging from *minus* 35°C to plus 150°C. One Finnish manufacturer tests every type of grease supplied by user/client companies under these temperature extremes and leaves no reliability-related issues open for questioning.

Cost Studies Prove Favorable Economics of Automated Lubrication Systems

A Finnish paper mill, Enso Oy, has documented the production increases, labor savings, and downtime reductions shown in Figure 12-20. Downtime hours for a total of 31 process units encompassing over 7,500 lubrication points are illustrated in Figure 12-21. Here, the Kaukopaa mill documented the 11-year trend from 9,700 hours of downtime in 1985 to approximately 280 hours in 1995. In the same time period, production went from 620,000 tons (1985) to 950,000 tons (1995). Figure 12-22 shows how, from 1990 until 1995, total maintenance expenditures decreased 26%, and maintenance costs per unit of production were reduced by 46%. Needless to say, Enso Oy has realized millions of dollars in extra profits from the timely introduction of engineered automatic lube systems. They have included these systems in the mandatory scope of every new project and Enso Oy's mill standard ("EGO") defining automatic lubrication systems has been adopted as a National Industrial Standard in Finland, a "high-tech" country in every sense of the word.

Washers, agitators, pumps, electric motors, soot blowers, barking drums, chippers, screens, presses, conveyors, and other equipment are automatically lubricated at a modern facility. A machine, which without lube automation often experiences five lubrication-related bearing failures per year, is likely to experience none with an engineered grease injection system. This often translates into 30–60 hours of additional machine time and profit gains of $90,000–$180,000 annually.

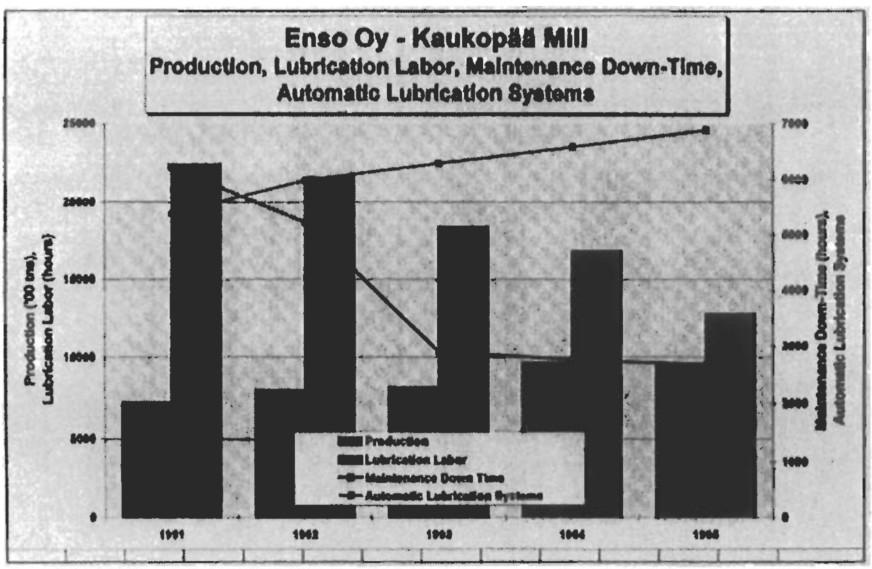

Figure 12-20. Production, lubrication labor and maintenance downtime statistics. *(Courtesy of Safematic, Muurame, Finland and Alpharetta, GA.)*

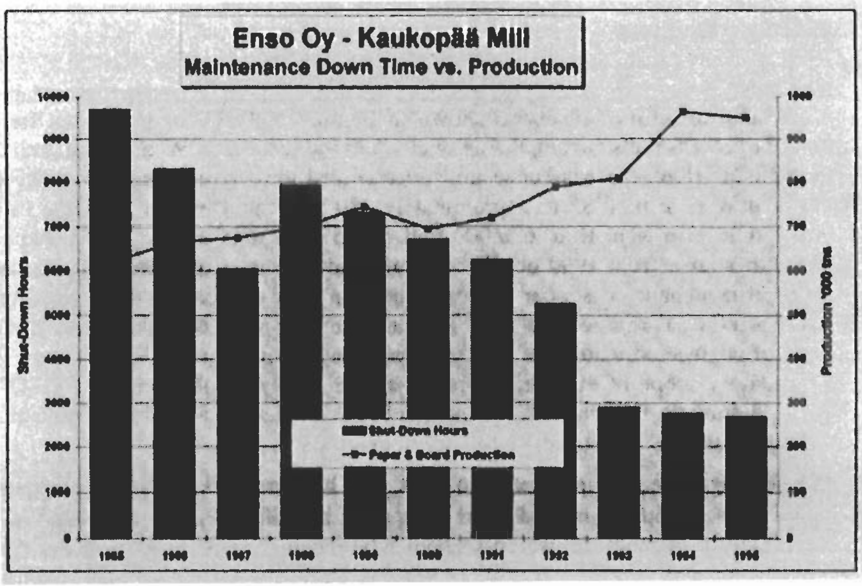

Figure 12-21. Maintenance downtime vs. production. *(Courtesy of Safematic, Muurame, Finland and Alpharetta, GA.)*

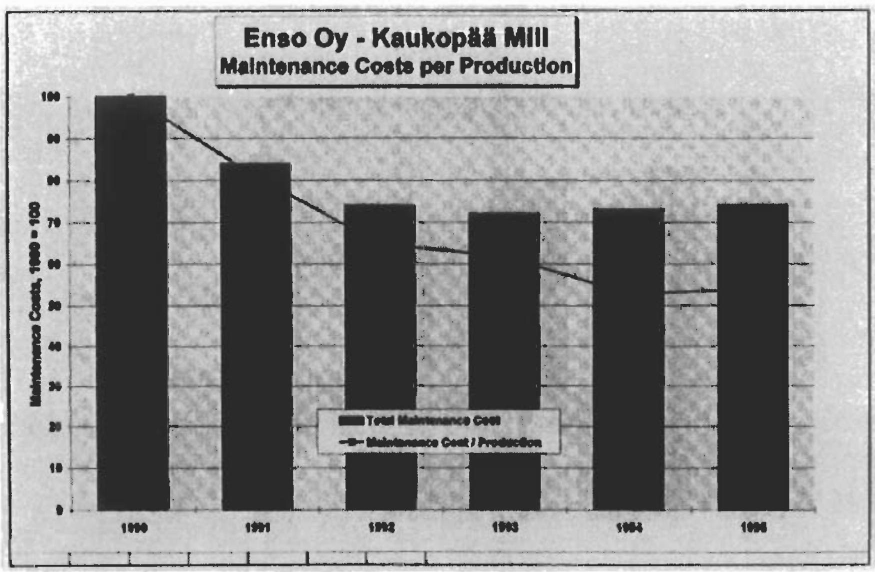

Figure 12-22. Maintenance costs referred to production. *(Courtesy of Safematic, Muurame, Finland and Alphretta, GA.)*

Payback for these systems, both originally supplied as well as retrofitted, typically ranges in the six-months to three-year timeframe. This might be one of the explanations why Scandinavian paper producers, whose workers have higher incomes than most of their American counterparts, are profitable and able to compete in the world markets. Automated lubrication has consistently yielded increased plant uptimes ranging from 0.1% to 0.5%.

What Lessons Can Be Drawn from All This

Engineered automated lubrication systems are no longer a luxury. Instead, they are one of the key ingredients of any well-thought-out maintenance cost reduction and failure avoidance strategy. Procurement specifications for a wide range of machinery used in process plants, chemical manufacturing plants, petroleum refineries, and paper mills deserve to be updated wherever life cycle cost studies demonstrate the reliability and general maintenance advantages of this mature lube application method. Retrofitting of engineered automatic lubrication systems, although somewhat more costly to implement than initial installations, is nevertheless justified in the numerous critically important machines that operate in adverse temperature and moisture environments. Since the bearing failure history of these machines is generally painfully evident, cost justifications are easy to develop with often remarkable accuracy. More importantly, these studies may point the way toward greater profitability.

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Chapter 13

Providing Safety and Reliability Through Modern Sealing Technology

Government agencies in the United States are imposing increasingly stricter limits on emissions of Volatile Organic Compounds (VOCs) from rotating equipment. These fluids change from the liquid to the vapor phase upon leaking to atmospheric pressure and thus become airborne pollutants. Leakage of VOCs from valves, fittings, and pump seals is known as "fugitive emissions."

Emission readings in the U.S. are done per EPA Method 21.[1] An organic vapor analyzer draws in a continuous sample of air from the space within 1 cm of the seal end plate and shaft interface. VOC emissions are measured in parts per million (ppmv) concentration in air. Emission readings are affected by factors such as wind speed, shaft rotation rate, and proximity of the probe to the emission source. Since the measurement is not in mass units (e.g., grams/hour), the EPA considers Method 21 to provide only an emissions *screening* value.

The U.S. Environmental Protection Agency (EPA) has established a national emissions limit of 1000 ppm for most pumps in organic hazardous air pollutant service.[2] In response, mechanical seal manufacturers have developed extremely low-emission, reliable seals.

Nevertheless, there are additional benefits that can be attributed to these government-initiated moves. Higher reliability means reduced product losses, fewer downtime events, and reduced maintenance expenditures. These are issues that are equally important to modern process plants, and reliability professionals must be thoroughly acquainted with the topic. Suffice it to say that the average refinery has 1,000 centrifugal pumps that undergo, on average, 300 seal replacements per year. "Best-practices" refineries replace as few as 150 seals, and the resulting savings easily exceed $1,000,000 each year.

This chapter* starts with a brief review of API Standard 682 specifications for seals in flashing hydrocarbon services. Next, it gives an in-depth presentation of design guidelines for low-emission single seals. This latter work is based on more

*Contributed by Bill Key, BW/IP International, Inc., Temecula, California.

than ten years of development using mechanical seal testers and employing propane as the operating fluid. Low-emission single mechanical seals are shown to be an effective means to comply with government emission limits.

It also reviews dual seal options to control emissions to very low levels. Included are liquid/liquid dual seals and their auxiliary equipment, dry gas secondary seals, and dual gas seals using pressurized barrier gas.

API Standard 682

API Standard 682[3] covers the minimum requirements for sealing systems for rotary and centrifugal pumps in refinery services. The objective of the API 682 Committee is reflected in their Mission Statement, "*To produce a reliable sealing system that has a high probability of operating 3 years of uninterrupted service, meeting or exceeding environmental emission regulations.*" The API 682 document 3 covers seal standards for most refinery applications. This discussion will only present highlights of the standards for seals in flashing hydrocarbon services. Seals designed to these standards should perform well in most other volatile fluid services.

Standard Seal for Flashing Hydrocarbon Services

Flashing hydrocarbon service includes all hydrocarbon services in which the fluid has a vapor pressure greater than 14.7 psia (0.101 MPa) at pumping temperature. The API 682 standard seal for flashing hydrocarbon services is a pusher seal with special features to maintain adequate vapor suppression. Other specifications include:

- Inside mounted cartridge seal
- Balanced seal
- Flexible element rotates

This seal is specified as a type "A" seal (Figure 13-1). When requested, the standard alternate to the type A seal is a totally engineered sealing system with an engineered metal bellows (type "B" seal). A standard single seal (one rotating face per seal chamber) is classified as Arrangement 1.

The operating window for type A seals in flashing hydrocarbon service is:

- Temperatures from –40°F to +500°F (–40°C to +260°C)
- Pressures to 500 psi (34.5 bar)
- Seal face surface speed < 5,000 ft/min (25 meters/sec)

A totally engineered sealing system is required for operation outside the above parameters.

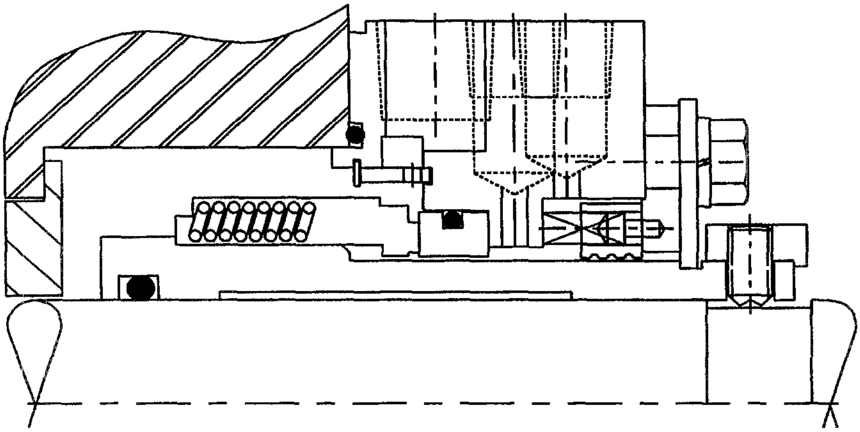

Figure 13-1. Type "A" seal. Arrangement 1—single seal.

Dual Seal Arrangements

For arrangements of two mechanical seals, the terminology "tandem" and "double" are no longer in use. A seal configuration composed of two mechanical seals is now known as either an *unpressurized* dual seal or a *pressurized* dual seal.[3] Furthermore, the lubricating fluid between unpressurized dual seals is a *buffer* fluid and it has a pressure less than that of the pump process fluid. Pressurized dual seals employ a *barrier* fluid between the seals that is at a higher pressure than the pump process fluid.

The standard *unpressurized* dual (Arrangement 2) mechanical seal is an inside balanced type cartridge mounted mechanical seal with two rotating flexible elements and two mating rings in series, Figure 13-2. The inner seals of Arrangement 2 are designed with a positive means of retaining the sealing components and sufficient closing force to prevent the faces opening to pressurization of the buffer fluid to 40 psig (2.75 barg). Also, the outer seal must be capable of handling the same operating pressure as the inner seal.

The standard *pressurized* dual mechanical seal is an inside balanced type cartridge mounted mechanical seal with two rotating flexible elements and two mating rings in series (Figure 13-3). It is classified as Arrangement 3. The inner seal must be designed to stay in place in the event that pump process fluid is lost. This criterion ensures that the faces do not open up in the event that the highest pressure is at the ID of the inner seal.

Vapor Pressure Margin

API 682 requires the seal chamber pressure to be a minimum of 50 psi (3.4 bar) or 10% above the maximum fluid vapor pressure at seal chamber fluid temperature.[3] This margin can be achieved either by raising the seal chamber pressure or lowering the seal

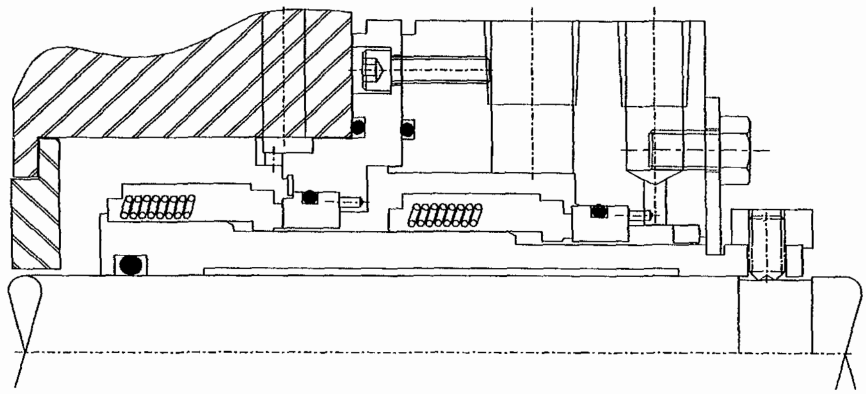

Figure 13-2. Arrangement 2—standard dual seal with buffer fluid pressure lower than process fluid pressure.

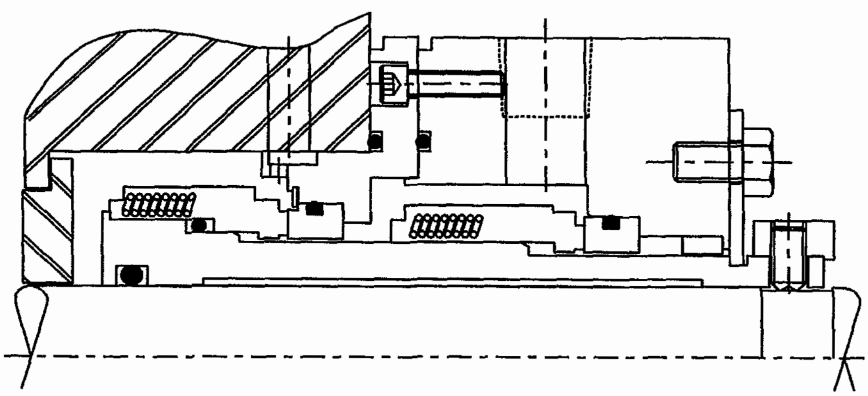

Figure 13-3. Arrangement 3—dual seal with barrier fluid pressure higher than process fluid. Inner seal is double balanced to withstand reverse pressure without opening.

chamber fluid temperature. If the temperature is above 140°F (60°C), an internal circulating device and an API Plan 23 closed-loop cooling system should be used.

Distributed Seal Flush

A distributed seal flush system such as a circumferential or multiport (see Figures 13-1 and 13-12) arrangement should be provided for all single seals with rotating flexible elements.[3] Ports with a minimum diameter of ⅛″ (3 mm) must be used in multiport systems. Distributed flush systems are not mandated for stationary single seals and dual seals.

Throat Bushings

API 682 requires that throat bushings be provided unless otherwise specified.[3] A throat bushing provides a restrictive clearance around the shaft (or sleeve) between the seal and the impeller. It is used to increase seal chamber pressure and control seal flush flow rate. A floating bushing allows a tighter clearance and, hence, higher seal chamber pressure. For Plan 23 flush systems, a throat bushing acts to isolate the seal chamber fluid from the pump process fluid.

Throttle Bushings

A throttle bushing forms a close clearance around the sleeve (or shaft) at the outboard end of a mechanical seal. For single seals, and when specified for dual seals, a non-sparking *floating* throttle bushing should be installed in the seal gland or chamber and positively retained against pressure blowout to minimize leakage if the mechanical seal fails.[3]

Seal Face Materials

One of the seal rings must be premium grade, blister resistant carbon graphite; the mating ring should be reaction-bonded silicon carbide.[3] Self-sintered silicon carbide should be furnished, when specified.

Seal Manufacturer Qualification Testing

To assure the end user that a seal will perform reliably, each seal/system must undergo qualification testing on an appropriate test rig.[3] Testing is done on two seal sizes, 2″ (50 mm) and 4″ (100 mm). The test sequence consists of dynamic, static, and cyclic phases run consecutively without disassembly of the seal. Seals for flashing hydrocarbon services are tested on propane.

Dynamic testing is for a minimum of 100 hours, and the static (0 rpm) phase is for at least four hours. The cyclic phase includes startup, dropping pressure to vaporize seal chamber fluid, turning off flush for one minute, and shutdown. A total of five cycles are run.

Low-Emission Single Seal Design

Single mechanical seals provide reliable sealing for most VOC services when the following conditions are satisfied:

- Fluid specific gravity > 0.45
- Vapor pressure margin in the seal chamber > 25 psi
- Flush fluid provides good lubrication of the faces

A dual seal with a barrier fluid is recommended if any of these conditions are not met.

Low-emission single seals are designed to run with contacting faces. To minimize heat generation and wear, contact loading must be light. Face contact promotes longer seal life by minimizing the possibility of abrasive particles getting between the faces. Contact occurs on face high spots (asperities). A small amount of fluid migrates across the sealing dam in the spaces between asperities. Typically this leakage is on the order of 1 gm/hr for low emission single seals.

Seals on flashing hydrocarbon services often operate with seal chamber pressure close to vapor pressure. Seal face surfaces may be hotter than the fluid boiling point. In these cases, the entire fluid film between the faces is a vapor. Well-designed seals can run successfully with vapor between the faces provided that careful attention is paid to the following:

- Face deflections
- Seal balance ratio
- Face width
- Materials
- Flush
- Vapor pressure margin
- Recoverability from an upset

Face Deflections

Seal faces are lapped to about 1 light band flatness. In operation, however, the faces distort (cone) due to pressure and thermal loading. Generally faces are designed so that pressure deflects the faces toward OD contact and thermal loading deflects the faces toward ID contact. Optimized seals run with total coning (sum of pressure and thermal deflections for both faces) close to zero light bands. Figure 13-4 shows faces that are in ID contact and slightly open to the high pressure fluid.

Wear of the carbon face under steady running conditions usually produces full face contact, from OD to ID across the sealing dam (Figure 13-5). Contact occurs on the asperities. Asperity size is greatly exaggerated in Figure 13-5. Roughness average (R_a) is on the order of 0.10 μm (4 μin) for a well-running seal. A fluid film, with a pressure drop from OD to ID, exists between the asperities. The fluid film provides hydraulic load support and allows some leakage in spite of face contact.

Vapor temperature margin is often less than 20°F on flashing hydrocarbon services. Laboratory tests on propane show that seal interface temperature is typically 40°F (or more) hotter than the seal flush.[4,5] Thus, vaporization of the fluid film usually occurs near the face OD (Figure 13-5) due to face temperature rise. Some researchers have stated that seals will not be adequately lubricated unless at least 50% of the fluid film radial width is in the liquid state. However, flashing hydrocarbons have a *liquid* viscosity on the order of 0.1 cP, which is less than 1% the viscosity of lubricating oils. Due to the very low viscosity of light hydrocarbon liquids, there cannot be any significant *hydrodynamic* lubricating effect. Lubrication may be enhanced by a tribochemical reaction of the fluid film hydrocarbon vapor and the seal faces.[6] Laboratory testing and field experience show that five-year life, with low emissions, is attainable for optimized seals running with a vapor film from OD to ID.

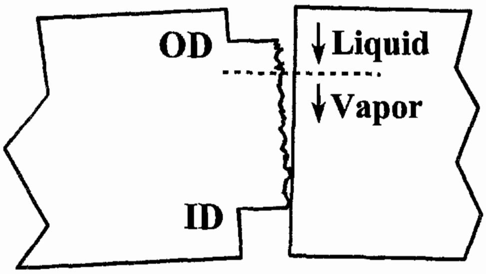

Figure 13-4. Faces usually contact at ID or OD at startup. Vaporization often occurs near the OD on flashing services due to face temperature rise.

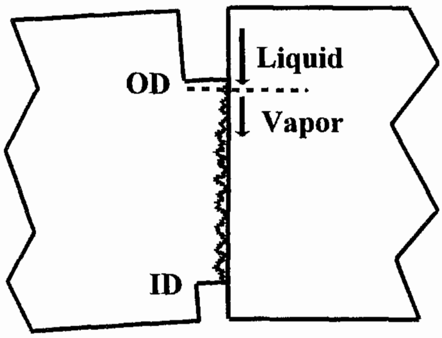

Figure 13-5. Wear results in full face contact from OD to ID.

Competent seal designers use finite element analysis and fluid mechanics models[7] to predict seal face deflections. Computer aided seal analysis has proven to be a valuable tool in designing seals for optimal performance. Due to model assumptions, however, the predictions need to be adjusted based on lab tests of seals running under simulated field conditions. Testing on propane is used to guide the design process for seals running on flashing fluids. For an optimized seal, a post-test surface trace of the carbon shows a smooth profile, with worn-in coning less than four light bands.

Seal Balance Ratio

The force acting to push the faces together is proportional to the seal balance ratio (ignoring spring forces). Reduced leakage can be achieved by using a higher balance ratio to increase the closing forces. A very high balance ratio must be avoided to prevent heavy contact loading and rapid wear. Too low a balance ratio results in excessive leakage.

An opening force, tending to separate the faces, is generated by the pressure distribution in the fluid film. When the sealed fluid changes from liquid to vapor as it moves across the faces, the opening force can be significantly greater than if the fluid remains in the liquid phase.[8] Figure 13-6 illustrates typical pressure distributions in the fluid film for three different fluid conditions: all liquid leakage, liquid flashing to vapor half way across the face, and all vapor leakage. The pressure profiles are for faces that are running with full-face contact. It is clear that two-phase flow results in higher film pressure and thus higher opening force. Higher balance ratios are thus required for seals operating on volatile liquids. If the balance ratio is not sufficiently high, the faces can "blow open."

Ideal balance ratio for two-phase seals can be estimated using theoretical models.[8] Actual seal performance is usually somewhat different than that predicted. Laboratory testing on propane shows that optimum balance ratio is in the range of 77% to 85% for flashing hydrocarbon services.

Face Width*

Selection of face width involves contrasting criteria. A narrow face generates less heat, but a wider face is more resistant to pressure deflection and provides a longer leak pathway. A narrow face is more sensitive to O-ring drag effects.

For narrow faces, O-ring drag is a larger fraction of net closing force. Net closing force consists of hydraulic closing force, spring loading, and O-ring drag. O-ring drag can either add to or subtract from the axial closing force.[9] Since hydraulic closing force is proportional to face area, O-ring drag has a larger *relative* effect on closing force for narrow faces.

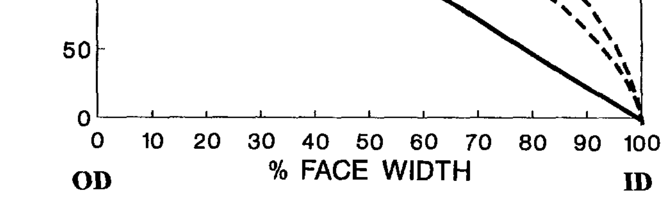

Figure 13-6. Possible pressure profiles. Vaporization results in greater opening force and requires higher balance ratio to prevent face separation.

*Face width narrative modified to reflect H.P. Bloch and T.P. Will experience, 1965–1997.

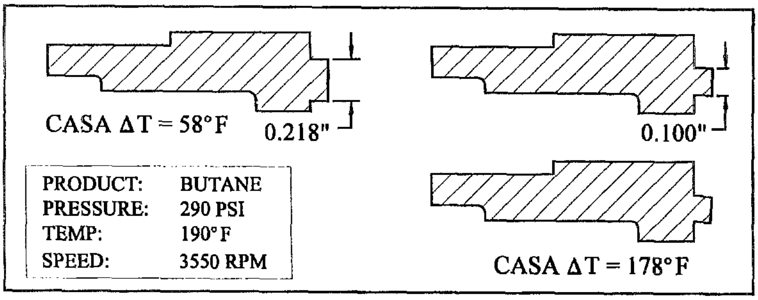

Figure 13-7. An application in which a narrow face failed in three days due to OD contact caused by pressure deflection. Wider face ran more than two years.

Seals with very narrow face widths of 0.100" (2.5 mm), as recommended by Will,[10] require different geometries from those using wide faces (0.200", or 5 mm). An example using identical geometries is reported in [4] and depicted in Figure 13-7. The service is butane at 290 psi. Narrow faces of 0.100" (2.5 mm) were found to last less than three days, with the carbon nose worn off. Replacement seals with the nominal face width of 0.218" ran for over two years. Computer-aided seal analysis (CASA) predicts a large amount of pressure deflection for the narrow face, causing OD contact. The divergent gap results in low opening forces developed within the fluid film and, thus, high contact loading. For this identical geometry case, the computer model predicts a face ΔT of 58°F for the standard width face, and a ΔT of 178°F for the narrow face. However, highly satisfactory experience is reported with narrow face seals and optimally configured seal components in light hydrocarbon services.[10]

Optimum face width to achieve both low emissions and long service life is thus a function of both seal design and service conditions. Properly designed and operated seals with face widths ranging from 0.140" to 0.280" can comply with strict emission limits and attain long life. Narrow faces typically perform satisfactorily at pressures below 10 atmospheres (150 psi), but may require extensive redesign in higher pressure applications.

Face Materials

Low-emission seals run with asperity contact and the lubricating film is often mostly vapor. Best performance under these severe lubricating conditions is obtained by running carbon-graphite against silicon carbide. This material combination has the lowest friction and heat generation of currently available seal face materials.[11] The high thermal conductivity of silicon carbide facilitates the removal of heat from the sealing interface. Carbon-graphite provides lubricity and it readily conforms to the hard face to minimize the leakage gap.

Reaction-bonded silicon carbide is preferred over sintered SiC because it is perceived to have higher chip resistance. The use of large chamfers improves the chip resistance of both types of silicon carbide.[12] Sintered SiC has superior chemical resistance and it performs well in corrosive applications such as HF acid.

More important than the grade (reaction-bonded or sintered) is microstructure quality. High grade reaction-bonded SiC contains minimal anomalies such as silicon streaks, unreacted carbon, and contamination with foreign material.[13] Figures 13-8 shows the microstructure of a "bad" silicon carbide seal ring. Many of the silicon carbide particles (gray or black) are weakly bonded to each other; they are surrounded by low strength free silicon (white). SiC grain pull-out easily occurs if the mating faces experience moderately loaded contact. Figure 13-9 shows "good" SiC with relatively small pools of silicon filling the gaps between the SiC particles. Each SiC particle is strongly bonded to its neighbors. It is imperative that seal vendors have a microstructure specification to assure obtaining high quality silicon carbide.

Carbon-graphite rings must also be of high quality. Carbon vendors list from 30 to 50 grades of material suitable for mechanical seal faces. Laboratory tests show that only a few grades can reliably handle vapor phase operation in VOC services.[4] Carbon rings are molded in powder form and baked at high temperature. This drives off volatiles and creates a network of interconnected pores. Impregnation, with either a resin or metal, makes the ring leak tight and strengthens the structure. Testing conducted with propane as the sealed fluid shows that antimony filled carbons run cooler and are more blister resistant than resin filled carbons.[4] Carbons impregnated with antimony provide an optimized seal with the ability to recover from some short-term pump upsets.

Recoverability. A mechanical seal should have the ability to absorb short-term upsets and recover to provide long life and low emissions. For example, during start-up, the seal chamber may be filled only with vapor. This is referred to as "running the pump dry."

A laboratory test was devised to evaluate face materials under dry running conditions. Each test started with a 3–4 hour run-in on liquid propane at 225 psig (15.5 barg). Then pressure was decreased to about 160 psig (11 barg) to vaporize the seal

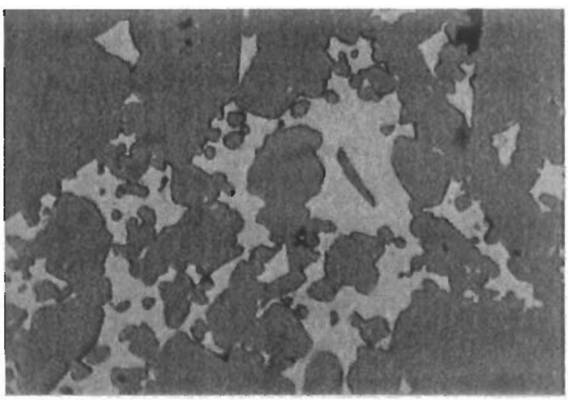

Figure 13-8. "Bad" reaction-bonded silicon carbide (500X). SiC particles are dark; white areas are free silicon. Note weak bonding of SiC particles.

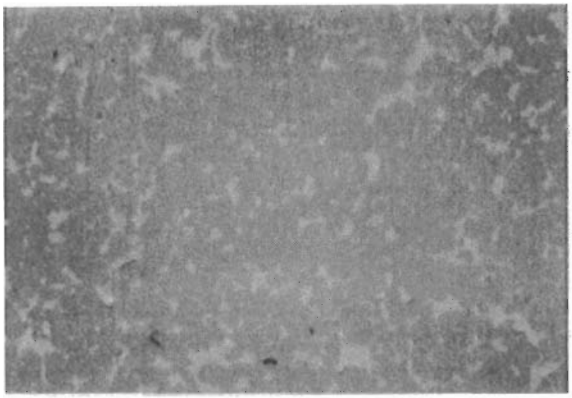

Figure 13-9. "Good" reaction-bonded silicon carbide (500X). Strong bonding between SiC particles (gray). Small pools of silicon (white).

chamber fluid. After two minutes of "dry running" at 3600 rpm, pressure was increased to 225 psig for a final hour on liquid propane.

More than two dozen grades of carbon-graphite recommended by vendors were run against a high quality silicon carbide mating face. The silicon carbide faces survived this two-minute dry running test with no visible wear. Of the carbon grades evaluated, only high quality antimony filled faces survived without blistering or other signs of face damage. Two high quality resin impregnated faces experienced minor blistering. Other grades showed heavy damage.

The effect of vapor pressure margin on face temperature rise (above bulk fluid temperature) is shown in Figure 13-10.[4] This result is for an optimized seal with multiport flush and an antimony impregnated carbon face. At about 3.9 hours into the test, chamber pressure was decreased in steps until vaporization occurred at about 160 psig (11 barg). Note that the right side vertical axis is in units of bar, indicating vapor pressure *margin*. Face ΔT was stable until vapor margin decreased below 15 psi (1 bar). Further reduction to 0 psi margin resulted in a large increase in face ΔT to 140°F (78°C). Seal recoverability is demonstrated on the right hand side of Figure 13-10. As ΔT dropped to 40°F (22°C), the vapor margin was increased to 58 psi (4 bar). Both the carbon and silicon-carbide faces were in excellent condition after this test.

Flush Arrangement

Seal flush is used to cool the faces and to maintain bulk fluid temperature below the vapor point. In flashing hydrocarbon services, it is common for pump suction pressure to be near the fluid vaporization pressure. Experience shows that satisfactory seal performance requires that seal chamber pressure be 25–50 psi (1.7–3.4 bar) above the vapor pressure at bulk fluid temperature.[4] Special flush arrangements are used to provide an adequate vapor pressure margin in the seal chamber.

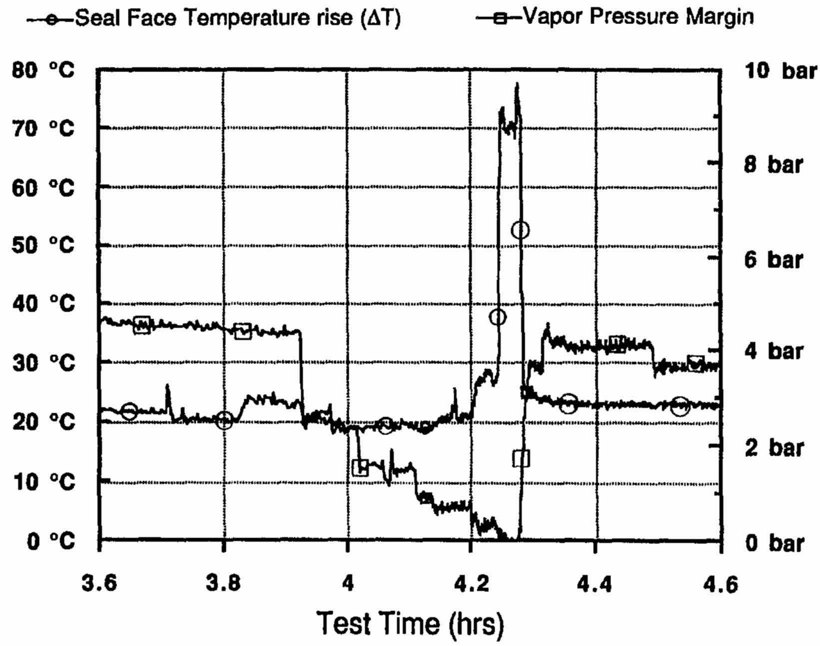

Figure 13-10. Effect of vapor pressure margin on face temperature rise. Chamber fluid is vapor at 0 psi margin.

Boiling Heat Transfer. Seal face temperature may be hot enough to cause boiling on wetted surfaces as shown in Figure 13-11. If limited in extent, boiling is an effective mechanism to cool the seal. First, the phase change from liquid to gas absorbs heat from the faces. Second, boiling creates intense turbulent agitation that brings cooler bulk fluid into contact with the faces. Seal flush must be sufficient to remove the vapor bubbles and prevent formation of a vapor pocket in this region. A vapor pocket results in negligible cooling and rapid face wear.

An API 11 flush[3] is frequently used in flashing hydrocarbon services. Flow is circulated from pump discharge through a flow control orifice to the seal chamber and then returns to pump suction. Chamber pressure can be increased by forcing the return flow to pass through a close clearance throat bushing located between the seal chamber and pump suction. Vapor margin can also be increased by cooling the flush with a heat exchanger in the flow line from pump discharge, known as API Plan 21. For applications above 140°F (60°C), API Standard 682 recommends use of a Plan 23 flush. In a Plan 23 flush arrangement, seal chamber fluid is continuously circulated by means of a pumping ring through a heat exchanger. This plan minimizes heat load on the cooler by cooling only the small amount of liquid that is recirculated. Plan 23 piping recommendations can be found in. References 14 and 15.

Laboratory testing[4] of mechanical seals on propane shows that performance is enhanced by:

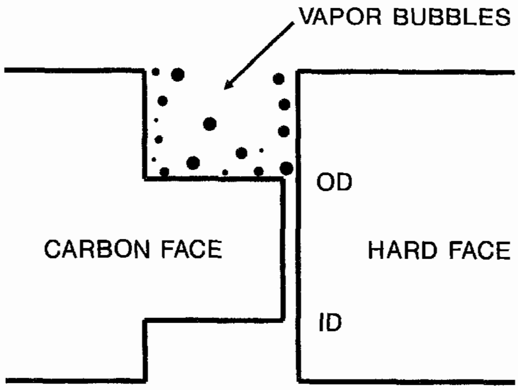

Figure 13-11. Hot surfaces may cause boiling. Flush must be sufficient to prevent formation of a vapor pocket.

- Use a multiport injection flange with 4 to 8 ports equally spaced circumferentially around the seal. Figures 13-1 and 13-12 show an annular plenum with multiple ports surrounding the seal. This arrangement provides more uniform cooling.
- Injection directly at the seal interface to provide cooling near the source of heat generation. Sweep away vapor bubbles to prevent formation of a vapor pocket around the carbon nose.
- API 682 specifies a minimum diameter of 0.125″ (3 mm). Injection velocity should be on the order of 15 ft/sec (5 m/sec).[3]
- Ports must have a small radial clearance over the seal faces to minimize velocity dissipation. Radial gap should be on the order of 0.200″ (5 mm).
- A flow rate of about 1.5 gpm per inch (2 lpm/cm) of shaft diameter is required when sealing flashing hydrocarbons. This is about twice the recommended rate for nonflashing fluids.
- Chamber pressure should be 25–50 psi (1.7–3.4 bar) above the vapor pressure at bulk fluid temperature. Too low a vapor margin results in unstable seal performance. Conversely, increasing the seal chamber pressure to very high levels can result in excessive seal-generated heat and rapid wear.

Multiport versus single port flush systems were evaluated on a laboratory seal tester using propane as the process fluid.[4] The multiport system had eight outlet ports positioned over the seal interface. Both seals were an optimized design and ran carbon filled with antimony against SiC. Operating conditions, including total flush flow rate, were identical for both single and multiport arrangements. The seals were run on the *recoverability* test previously described. After a three- to four-hour run-in, pressure was dropped from 225 psig to about 160 psig to vaporize the chamber fluid. After two minutes of "dry running," pressure was increased and a final hour run performed on liquid propane.

Face temperature rise above bulk fluid temperature is shown in Figure 13-13. The multiport arrangement resulted in cooler and more stable face temperatures. After

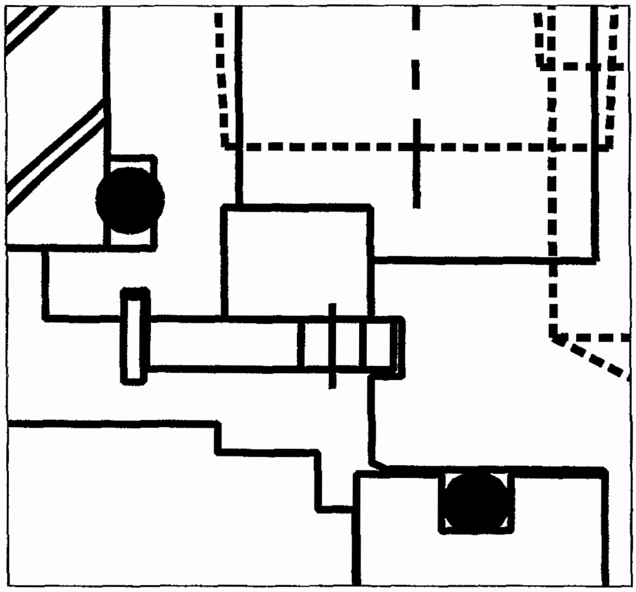

Figure 13-12. Multiport flush.

the two minutes of dry running, the single port seal required about 20 minutes to return to a stable face temperature. Apparently the single port flush is not very effective in eliminating the vapor pocket that formed during the running period.

Figure 13-4 shows emission rates for the two seal arrangements. Average emissions were 74 ppm for the multiport injection and 108 ppm for the single port flush. The multiport arrangement ran considerably more stable, as evidenced by reduced spiking in emissions levels. Note some spiking to ~1,000 ppm during the first 20 minutes after startup. Emissions then drop to low levels after a brief run-in. The multiport seal shows a spike of about 500 ppm shortly after the two minutes of dry running.

Vapor Pressure Margin

Seal chamber vapor pressure margin can be understood by examination of the vapor pressure curve for propane, Figure 13-15. Propane is in the liquid phase for conditions above the curve, and in the gas phase for conditions below the curve. Consider a situation in which the bulk fluid in the seal chamber is at 90°F. Figure 15 shows that propane has a vapor pressure of about 150 psig at that temperature. If 50 psi vapor pressure margin is achieved, then chamber pressure is about 200 psig and the temperature required to vaporize the fluid is about 110°F. Thus, for a 50 psi vapor pressure margin, there is only about a 20°F temperature margin. If pressure margin is 25 psi, then temperature margin is only 10°F. (Note that fluids lighter than

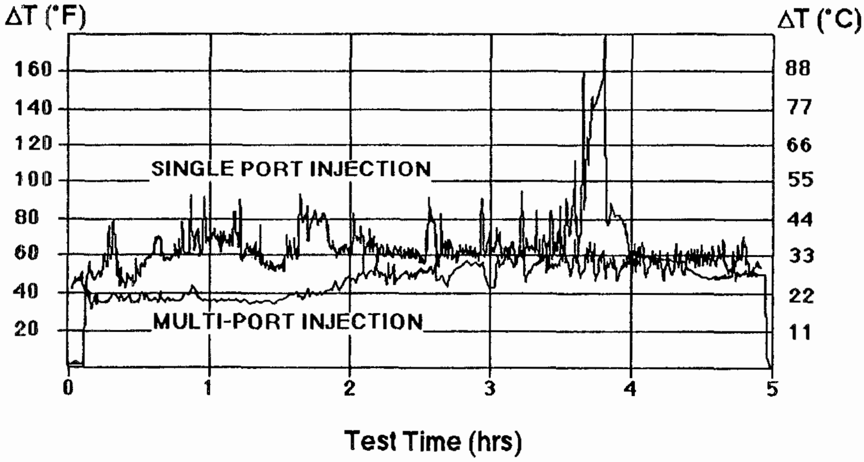

Figure 13-13. Seal face temperature rise: multiport vs. single port flush. Two minutes of dry running occurred at about 3.5 hours.

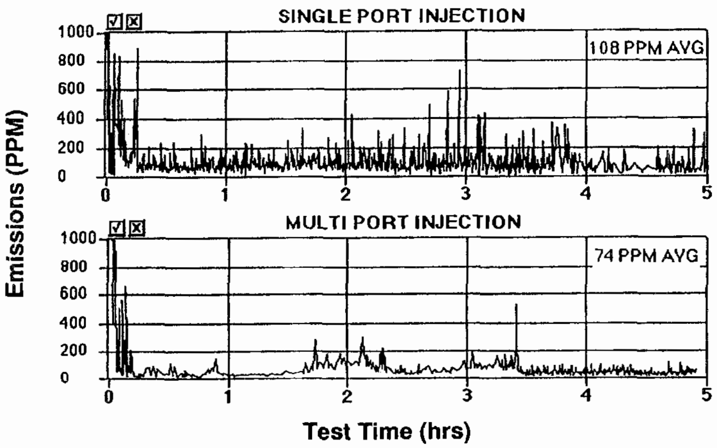

Figure 13-14. Emission levels: multiport vs. single port flush. Multiport flush results in lower emissions and more stable performance.

propane generally show a smaller temperature margin for a given pressure margin, while heavier fluids generally have a larger temperature margin).

Now consider what happens to a volatile liquid as it migrates across a seal face, from high to low pressure. The fluid experiences both a pressure drop and an increase in temperature. The fluid temperature is approximately equal to face temperature, since low-emission seals operate with extremely thin fluid films. Laboratory test measurements of seals running in propane show that face temperature is typically 40°F (20°C) above the bulk fluid temperature. Note, from above, a 50 psi vapor

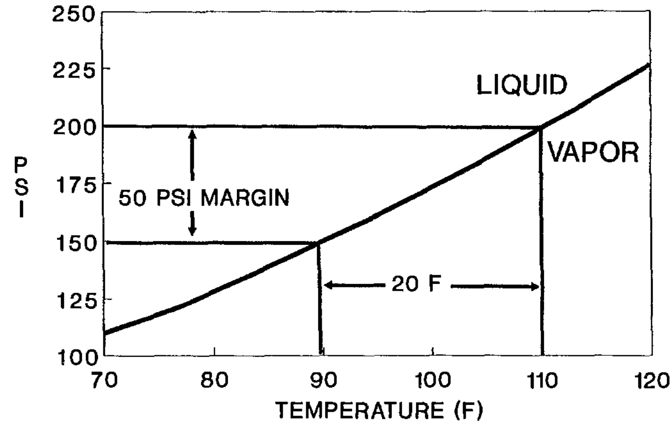

Figure 13-15. Propane vapor pressure curve. Even with a 50 psi vapor pressure margin, there is only a 20°F temperature margin.

pressure margin for propane results in only about a 20°F temperature margin. Thus, on flashing hydrocarbon services, almost the entire fluid film, from OD to ID of the face, is often in the vapor phase. Seals designed and operated along the guidelines presented in this segment can handle this condition with low emissions and long life.

Hydropad Faces

Hydropads are recesses or slots on the sealing surface of one of the faces. An example is shown in Figure 13-16. The slots act to promote hydrodynamic lift and enhance removal of heat from the faces.[16] Hydropads are often used under marginal lubrication conditions. On volatile liquid services, hydropads should be considered when:

- Pressure > 400 psig (28 barg)
- Vapor pressure margin < 10 psi (0.7 bar)
- Both faces are SiC (e.g., when sealing highly corrosive fluids, such as HF acid)

Work at BW/IP's lab showed that properly designed seals with hydropads can achieve emissions less than 1,000 ppm. Hydropad faces should not be used in dirty services that could clog the slots or allow abrasive particles to enter the fluid film between the faces.

Emissions from Pipe Fittings and Gaskets

Threaded pipe fittings in the seal flush line can be significant leak sources, with readings above 1,000 ppm.[4,17] Similar emission levels may be measured near the gasket region on the seal chamber face. Any leakage from these areas may drift into the emission measurement area for the mechanical seal. The mechanical seal may then be erroneously implicated as a leaker. It should be standard practice to sniff

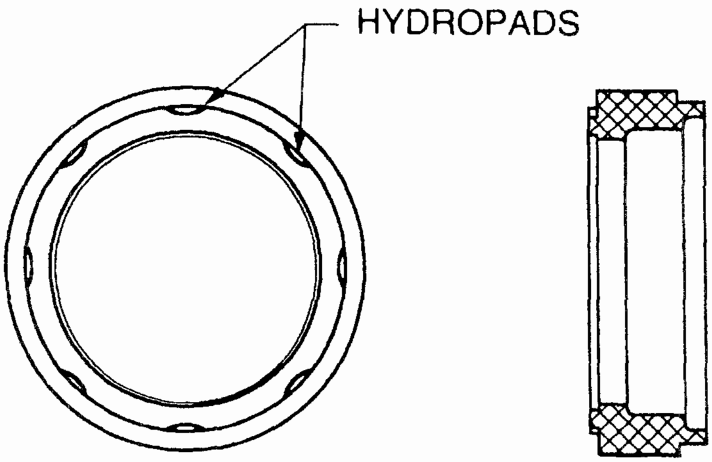

Figure 13-16. Hydropad face.

nearby pipe fittings and the flange gasket area if excessive VOC concentrations are detected adjacent to the mechanical seal.

Leak-tight threaded pipe fittings can be more easily attained using anaerobic paste-type sealants rather than PTFE tape. The seal chamber face must be smooth to be emission tight. Gaskets and O-rings must be free of nicks and scratches.

Dual Seal Arrangements

Dual seals consist of two single mechanical seals per seal chamber. A barrier (or buffer) fluid usually fills the space between the primary and secondary seals. This fluid may be either pressurized or unpressurized. Typically the barrier fluid is circulated through a cooler to remove seal generated heat. Some secondary seals are designed to run on gas and do not require a barrier fluid. Dual seals with a buffer fluid nearly eliminate product leakage to the environment, achieving emission levels less than 10 ppm.[17]

Unpressurized dual seals are simpler to operate and are more reliable than pressurized systems. In unpressurized systems, process fluid migrates past the primary seal faces into the buffer fluid. For volatile liquid process streams, this fluid is typically a vapor since the buffer fluid is at low pressure. In refineries, the vapor is usually vented to a flare. Other means of vapor disposal include carbon adsorption, condensation, or injection back into the pump.

In a pressurized dual seal system, the primary mode of migration is the barrier fluid into the pump process fluid. Face wobble and run-out can cause some reverse migration of the process fluid into the barrier fluid, causing barrier fluid contamination. A pressurized barrier fluid will trap more of the volatile gases in solution because solubility is approximately proportional to pressure. Pressurized barrier fluid systems have a higher probability of losing barrier fluid through the secondary seal faces and piping connections as compared to unpressurized systems.

Dual Seals with Unpressurized Buffer Fluid

In this arrangement, the inner (primary) seal is lubricated by the pump process fluid and the outer seal by the buffer fluid. The buffer fluid is at a pressure lower than the process fluid. Per API Standard 682, the normal arrangement is for both seals to be in series (Figure 13-2). Process fluid leakage is into the buffer fluid. In volatile liquid service this leakage is typically a vapor since the buffer fluid is at low pressure. Buffer fluid circulation transports the vapor into the external reservoir (Figure 13-17), where it is vented to a vapor disposal system (e.g., flare). Secondary seal leakage to the environment is low due to the small pressure difference across this

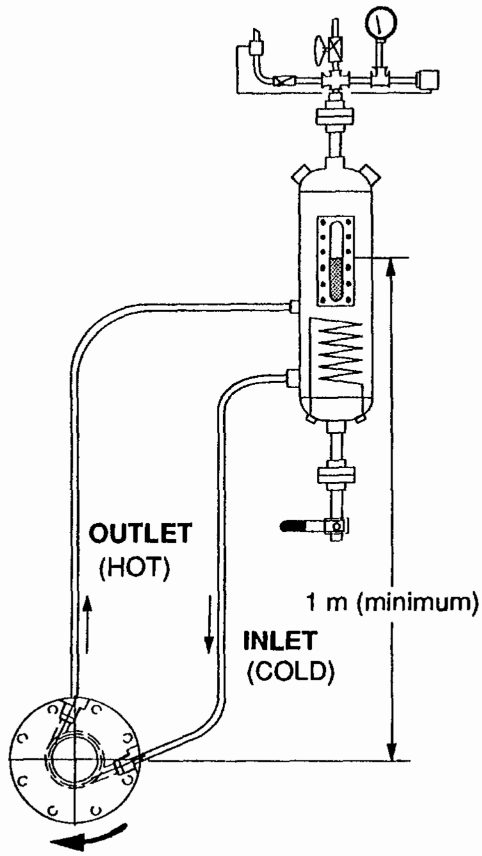

Figure 13-17. Dual seal with reservoir. Note tangential ports for improved flow and self-venting chamber with top discharge.

seal. The secondary seal, however, should be able to accommodate full system pressure to function as a safety backup seal.

Flow Circulation

Figure 13-17 shows the standard flush, known as API plan 52, for the secondary seal with unpressurized buffer fluid.[3] Buffer fluid flow circulation is usually produced by a pumping action generated by the rotating seal assembly. This circulation acts to cool both the secondary and primary seal faces. Tangential inlet and outlet ports are strongly recommended for flow enhancement. Tangential ports result in about four times the flow of radial ports.[17] Flow discharge **must be** located on top of the seal chamber for self-venting of vapors to the reservoir. On vertical pumps, the discharge port must be above the seal interface.

The return line to the seal chamber should be below the shaft centerline to promote thermosyphon flow. Thermosyphon flow results from fluid density difference between the "hot" outlet line and "cold" return line. A mathematical model and experimental verification of thermosyphon flow for seal circulation systems is contained in Reference 18.

Inadequate flow circulation can result in overheating of the fluid and damage to the seal faces. To assure sufficient flow, the pumping ring must have sufficient capability and the piping system must not hinder the flow. Per API Standard 682, the seal vendor must supply the head versus flow performance of the internal circulating device based on actual test results.[3] Performance of the internal circulating device should exceed the flow resistance of the piping system. A typical pumping ring head vs. flow curve and a piping system resistance curve are shown in Figure 13-18. The operating point is the intersection of the two curves. Poor piping practices can shift the system resistance curve far to the left, resulting in low flow.[19]

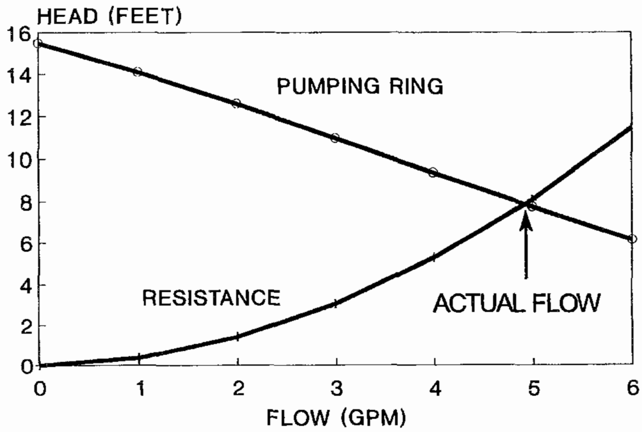

Figure 13-18. Pumping ring head-flow curve and system resistance curve. Actual flow is at curve intersection.

Good piping practices, shown in Figure 13-17, include:

- Large bore tubing (generally ¾″ ID)
- Minimal run lengths (locate reservoir close to seal)
- Large radius bends
- Continuous up-slope from seal to reservoir (no gas pockets)
- Minimal restrictions (use full port valves and connectors)
- Liquid level at least 3 feet (1 m) above the shaft

Examples of "bad" piping practices are illustrated in Figure 13-19.[19] Elbows and tees are used instead of bends. An up and down section of pipe is a vapor trap. Return to the reservoir is above the bottom of the sight gauge. Flow circulation through such systems is often inadequate for proper seal cooling.

Barrier/Buffer Fluids

The choice of barrier/buffer fluid can have a significant effect on secondary seal performance.[17, 20, 21, 22, 23] Figure 13-20 shows face temperature rise above seal

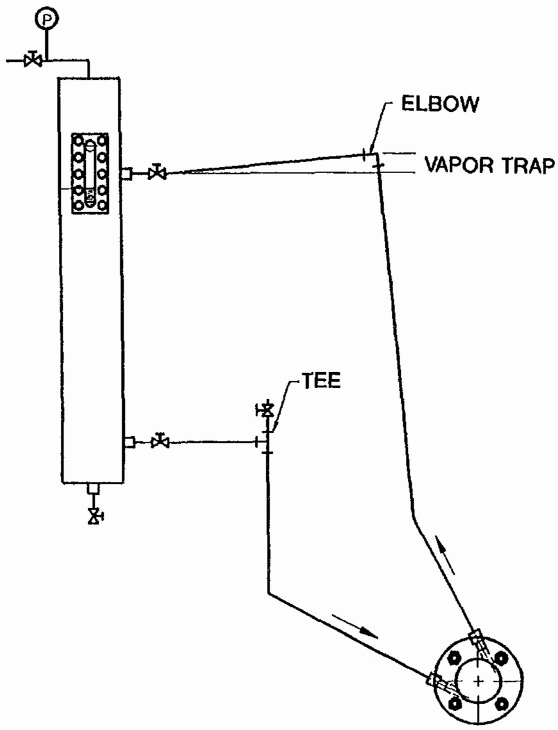

Figure 13-19. "Bad" piping practices.

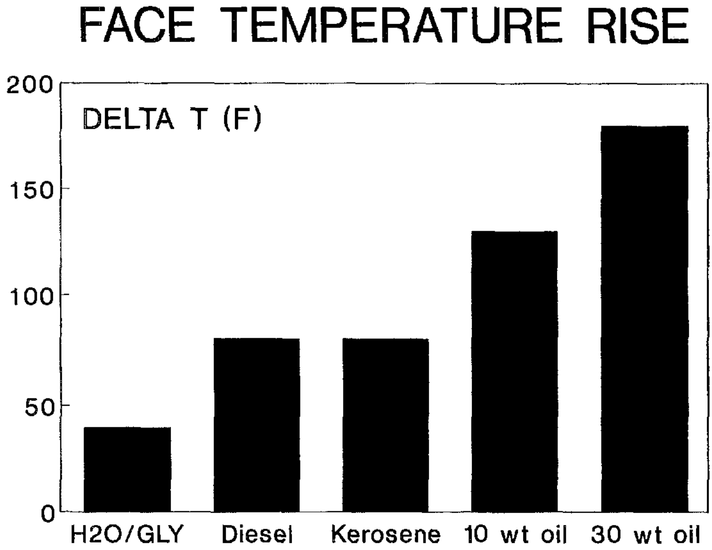

Figure 13-20. Face temperature rise (ΔT) for different barrier fluids. Diesel is recommended for most refinery services.

chamber fluid for different barrier fluids. Data are for a 2.625″ seal at 3,600 rpm, ~3 ft of fluid head, and carbon vs SiC faces. The secondary seal was designed for no visible leakage. A 50/50 mixture of water/ethylene glycol ran with the coolest faces. Diesel and kerosene resulted in relatively low face ΔT and low wear rate. Diesel #2 has proven to be an excellent barrier fluid in refinery services. Automatic Transmission Fluid (ATF), not shown, resulted in very high temperatures and rapid wear. Using only half the spring load, ATF resulted in a face ΔT 38% hotter than that for 30 weight oil. Cooler running faces are expected to have longer life and are less likely to have coking problems.

Optimum fluid viscosity is in the range of 1–5 cSt (at bulk fluid temperature). Higher viscosities generate more heat and may cause blistering of carbon faces. Viscosity should be at least 1 cSt to provide adequate lubrication. In addition, the fluid should:

- Have an initial boiling point at least 50°F above the temperature to which it is exposed
- Never be operated above autoignition temperature
- Not freeze at minimum site temperature

The barrier/buffer fluid must be compatible with the pumped process fluid and it must not be an organic hazardous air pollutant (organic HAP) as defined by the EPA.[2] Ethylene glycol is listed as an organic HAP. Fortunately, a 50/50 mix of propylene glycol and water has been found to result in slightly higher ΔT, but lower face wear than EG/H_2O.[20] Propylene glycol is not listed as an organic HAP. If used, glycol should be bought premixed with water. Automotive antifreeze, which is primarily ethylene glycol, may not be suitable because some additives tend to plate out and cause damage to seal faces.

Face ΔT is only one measure of barrier/buffer fluid performance. Another important parameter is worn-in surface roughness, or R_a.[20] The R_a of a surface is a measurement of the deviation in height relative to a reference as one moves across a surface. The rougher a surface, the larger the value of R_a. A typical carbon face will have an R_a of about 2 μ-inches (0.05 μm) in the as-lapped condition. Figures 13-21 and 13-22 show profilometer traces of carbon faces that ran for six days in two different buffer fluids. The seal that ran on ATF is seen to have a considerably rougher face, indicating more rapid wear than the one that ran on a synthetic oil. The vertical scale is 10 μ-inches (~1 light band) per division. The seal ran cooler on the synthetic fluid with a face ΔT of 67°F versus 124°F for ATF.

Diesel fuel worked well in field applications. One Southern California refinery installed 75 unpressurized dual seals to meet the area's stringent emission require-

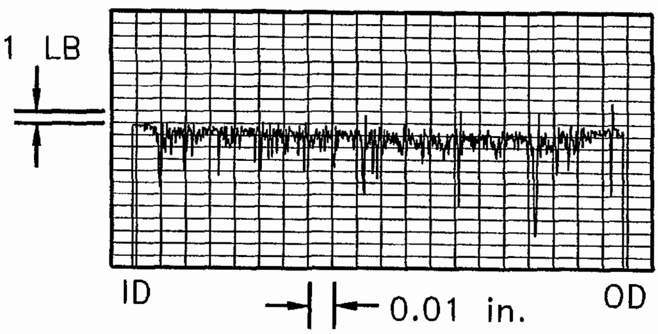

Figure 13-21. Surface trace of carbon face that ran on ATF > R_a = 11 μ-inches.

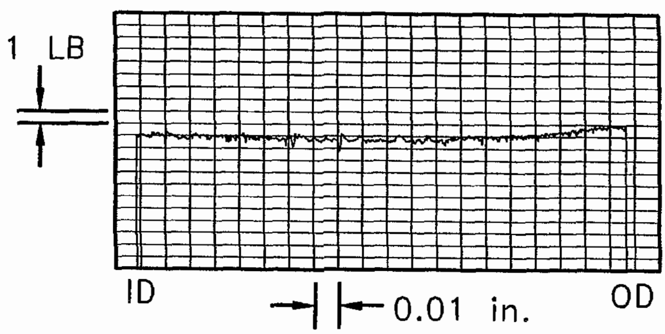

Figure 13-22. Surface trace of carbon face that ran on a synthetic fluid. R_a = 2μ -inches.

ments. The refinery is mostly using five gallon water-cooled reservoirs with diesel as the buffer fluid. Both primary and secondary seals are low-emission designs. Almost all of the installations have run more than five years and continue to be very low VOC emitters. Degraded diesel is disposed of by either feeding it to the refinery flare or into the pump process fluid.

Additional testing shows that synthetic fluids result in less face wear and lower face ΔT compared to petroleum barrier fluids.[22, 23] On other applications, life expectancy of synthetic fluids is up to ten times that of petroleum.[24]

NGL, Ethane Seals

Unpressurized dual seals with a liquid buffer fluid work well for NGL and ethane applications. High vapor pressure fluids are difficult to seal reliably with single seals. The liquid buffer fluid helps cool and stabilize the primary seal. Typical buffer fluids for these applications are either diesel or water/glycol.

Dual Seals with a Pressurized Barrier Fluid

Pressurized systems are used to minimize migration of process fluid into the environment. They are also used in applications in which the process fluid provides poor lubricity for the seal faces or in which the process fluid may change frequently, as in pipeline services. Both primary and secondary seals are lubricated by the pressurized barrier fluid. The typical arrangement is for both seals to be in series (Figure 13-3), as per API Standard 682.

Circulation of the barrier fluid is by means of an internal pumping ring (API plan 53) or by an external pump or pressure system (API plan 54). Barrier fluid pressure is usually set to be at least 10% higher than the pump process pressure at the seal chamber throat bushing. (It should be at least 25 psi greater). Too low a barrier fluid pressure may result in process fluid migration into the barrier fluid. Too high a pressure results in heavier face loading and more rapid wear.

API Standard 682 notes that pressurization with a gas blanket may result in foaming and loss of circulation at pressures above 150 psig (10 barg).[3] Gas solubility in liquids is approximately proportional to absolute pressure. Hence, at high pressure operation, more gas will come out of solution upon a drop in pressure in the flow loop. In most applications, helium has a lower solubility than nitrogen and thus the use of helium may alleviate foaming.[25] Nevertheless, for operation above 150 psig it is recommended that pressurization be achieved by using a piston or bladder-type accumulator.

Pressurized Dual Seal for HF Acid

A series dual seal arrangement with a pressurized barrier fluid was designed for HF acid application.[26] Both seals are low emission spring pusher designs. The system was tested at BW/IP's laboratory using propane as the primary seal flush and a

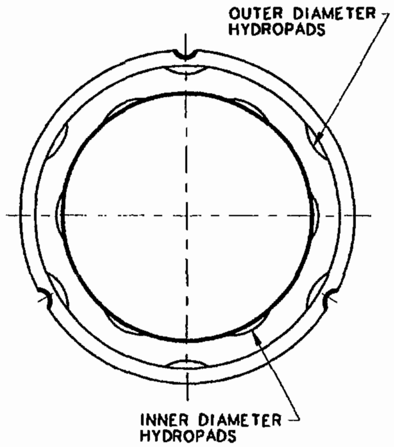

Figure 13-23. SiC seal face with OD and ID hydropads for HF acid application.

mixture of 95% isobutane and 5% propane for the secondary seal flush. Primary seal faces are sintered SiC, with one face having a proprietary hydropad design (Figure 13-23). The secondary seal uses a special HF resistant carbon face mating against a sintered SiC face.

The seal was tested at 3,600 rpm with primary seal flush at 300 psi and barrier fluid at 325 psi. Emission readings from the secondary seal were consistently less than 100 ppm and all seal faces had an excellent post-test appearance.

An additional short-term test was run to simulate loss of pump suction. The primary seal cavity was run dry (0 psig). Isobutane/propane barrier fluid pressure remained at 325 psig. The seal was run in this reverse pressure condition at 3,600 rpm for 15 minutes. The SiC versus SiC primary seal faces were heavily scored but remained intact, thus proving the seal design can handle this type of short-term upset without resulting in catastrophic leakage. This test was rerun using 325 psig unleaded gasoline as the barrier fluid and 0 psig air in the primary seal chamber. Primary seal leakage was 60–120 ppm and secondary emissions were 65–80 ppm during the three hour run. All faces were in good post-test condition. This seal design has been operating successfully for about four years in two HF acid pumps.

Compact Gas Seal Technology For Pumps*

The gas seal technology used in gas compressors has been successfully applied for emission-free sealing of liquid pumps in the past few years. The seals with pressur-

*Source: J. Nosowicz and W. Schöpplein, Feodor Burgmann, Wolfratshausen, Germany. Based on a paper originally presented at the *Hydrocarbon Processing*/Gulf Publishing Co. International Conference and Exhibition on Process Plant Reliability—Europe, Nov. 11–13, 1996, Amsterdam, The Netherlands.

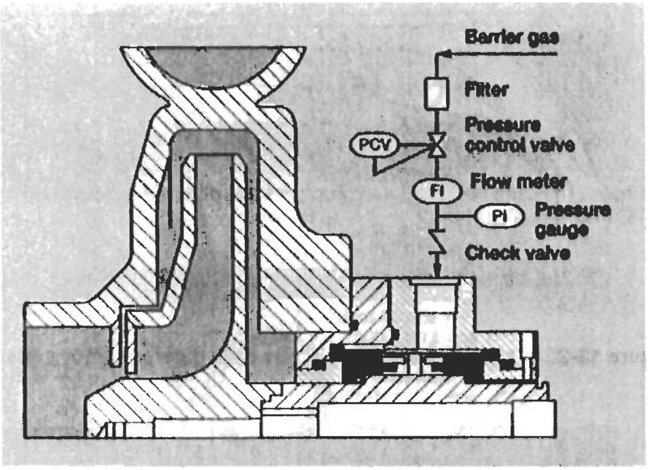

Figure 13-24. Double (dual) gas seal uses pressurized barrier gas.

ized gas supply systems are used as single or dual (tandem) seals. Gas seals, mainly as single seals, are frequently used as safety seals as well. Applying this noncontacting sealing system will result in reduced investment and operating cost.

Sealing Concept. Gas-lubricated mechanical seals for liquid pumps are operated with pressurized gas (inert gas) supply systems (barrier systems), Figure 13-24. The difference between the product and the barrier pressure is approx. 0.2 MPa, or 29 psi.

In contrast to a conventional mechanical seal, a gas-lubricated seal has a considerably wider sealing surface. In addition, one sliding face is contoured and the specific spring force is significantly smaller. The rotating seal faces compress the gas in the sealing gap via pumping grooves (Figure 13-25). During normal operation there will be a gap width of several micrometers. Seal face and seat design result in a convergence to parallel sealing gap geometry. Figure 13-26 shows an example of the static and dynamic pressure distribution in the sealing gap.

Similar to compressor seals, liquid pump gas seals can be designed with uni- or bidirectional grooves (Figures 13-25 and 13-28). The pressure distribution in Figure 13-27 shows the difference in pressure buildup in the circumferential direction of the ring.

Gas stiffness in the uni-directional seal sealing gap with a V-groove is larger than that of the bi-directional version with a U-groove.

The seal aerodynamic opening force, depending on the gap width resulting from the pressure build-up, shows the difference in function of uni- and bidirectional sliding face designs (Figure 13-28).

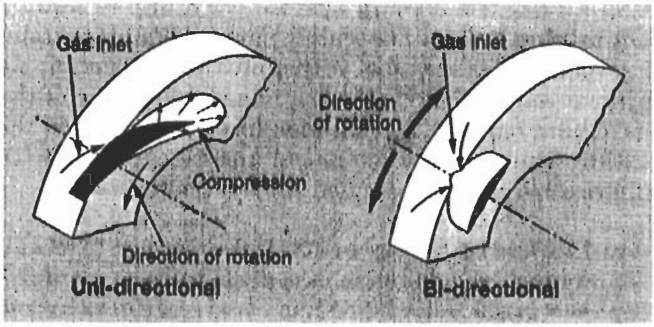

Figure 13-25. Uni- and bidirectional seals have different groove designs.

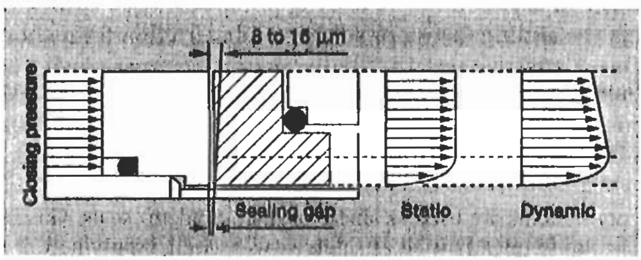

Figure 13-26. Pressure distribution in a dry gas seal sealing gap.

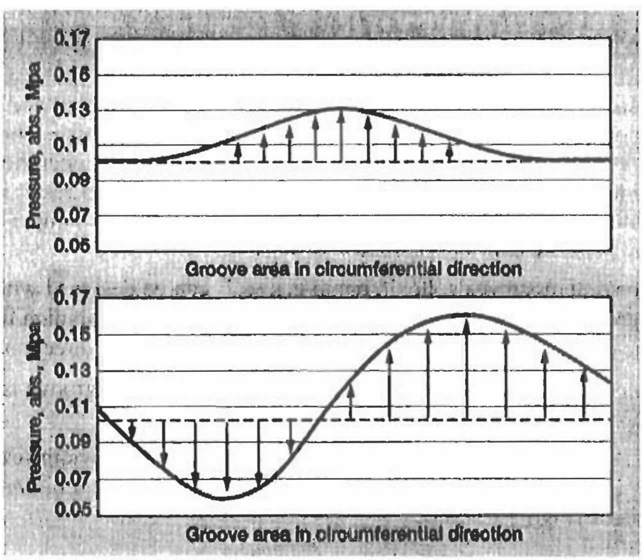

Figure 13-27. Uni- and bidirectional grooves have different pressure buildup profiles.

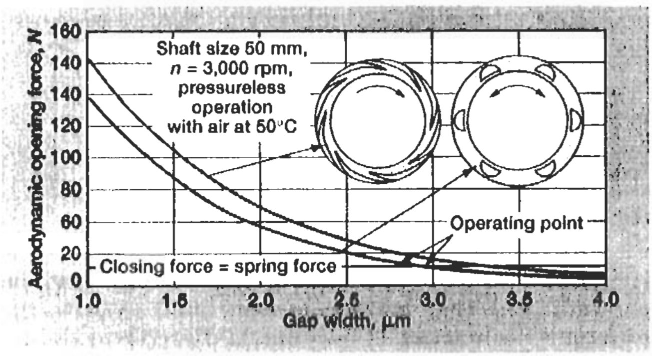

Figure 13-28. V and U grooves have different aerodynamic opening forces.

The unidirectional seal version is particularly suited for pressureless operation at low speeds. Bidirectional seals with a pressurized gas supply system are recommended in process pumps.

Operating Performance

Barrier Gas Consumption/Leakage Rate. At a pressure difference of 0.2 MPa (29 psi) normally available on the product side, barrier gas consumption is very low. The total barrier gas consumption for a dual gas seal as a function of pressure and sliding speed can be determined from Figure 13-29.

If a gas-lubricated mechanical seal is used as a safety seal behind a conventional mechanical seal, it also has to seal a liquid medium in case of conventional seal failure. Figure 13-30 shows the leakage rate of a unidirectional gas seal pressurized with water.

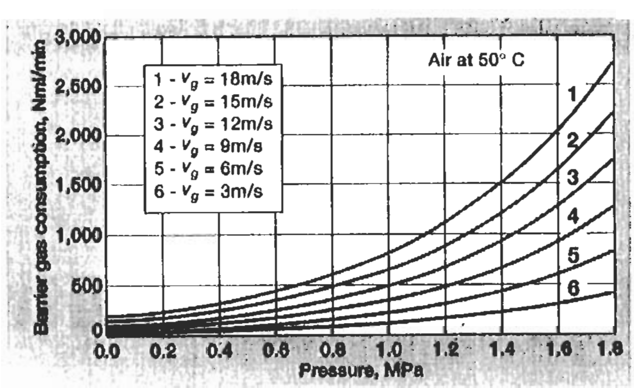

Figure 13-29. Barrier gas consumption of dry gas seals is very low.

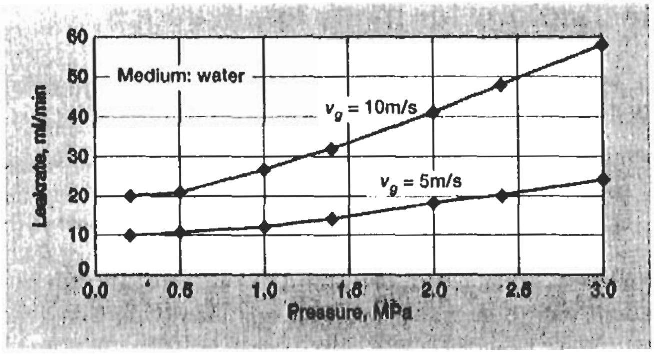

Figure 13-30. Leakage rates of a gas seal sealing liquid.

Power Consumption. Power consumption is very low; approximately 1% to 2% that of a conventional mechanical seal (Figure 13-31). Table 13-1 shows the power requirements of the different sealing systems.

Operating Limits. Operating limits depend on the allowable load of the individual seal parts. The temperature and pressure limits are defined by the elastomers, materials of the secondary sealing elements, and the sliding materials.

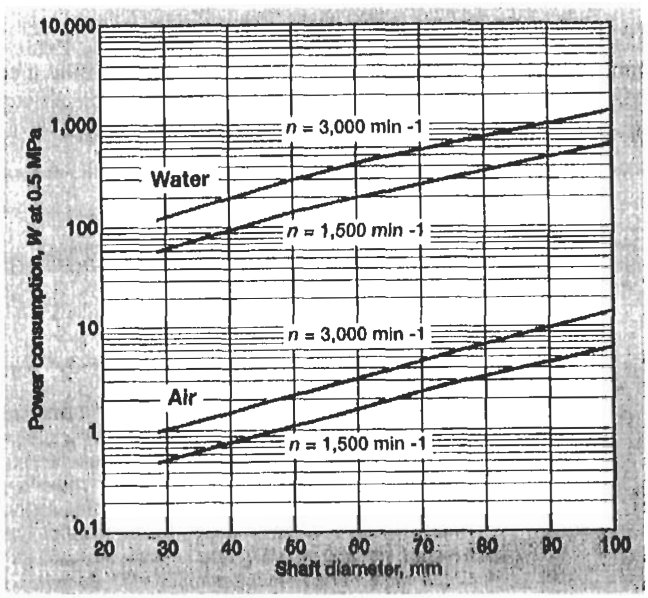

Figure 13-31. Power consumption of gas seals is considerably less than that of conventional mechanical seals.

Table 13-1
Power Requirement Comparison: Gas Seal/Liquid Seal/Magnetic Drive, Shaft Size 50 mm, n = 3,000 rpm, p = 0.5 MPa

Sealing System	Power Consumption, Watts
CGS-dual gas seal in a chemical standard pump	3
Liquid dual seal in a standard chemical pump	300
Magnetic drive with Hastelloy can/canned motor pump	1,300

Table 13-2
Gas Seal Materials

Sliding materials	hard/soft	Antimony-impregnated carbon/silicon carbide
	hard/hard	Silicon carbide/silicon carbide with special coating
Elastomers		All standard elastomers
Springs, metal parts		stainless steel, Hastelloy, etc.

For secondary sealing elements, temperature, extrusion performance and pressure-dependent gas absorption are relevant. For example, pressure variations in the sealing chamber can lead to gas absorption in the O-ring. In turn, O-ring destruction caused by explosive decompression of the absorbed gas may result. Such damage can be prevented by using suitable elastomer qualities, e.g., suitable for gases containing CO_2.

Sliding materials have a major influence on how gas-lubricated seals function. During pump start and stop periods, the sliding faces are subject to dry friction for a short time. Consequently, the sliding materials must have good emergency-running properties. Another important functional criterion is the sealing gap geometry, which is also influenced by the stiffness of the sliding materials. Table 13-2 lists gas-lubricated seal materials for liquid pumps.

Reversed Pressure. If the barrier pressure should decay, noncontacting face and seat separation on the product side is no longer ensured. Due to the increased pressure difference on the product side, the dynamically loaded O-ring would now move axially to the face housing wall (Figure 13-32).

A hydraulic closing force, F_H, acts on the seal face. The sliding faces on the atmospheric side still run without contacting. If the gas supply to the seal chamber is interrupted so that it can no longer be vented, the seal chamber is evacuated. This can result in seal face and seat contact on the atmosphere side. The resulting pressure profile is shown in Figure 13-33.

Gas-Lubricated Sealing System for Pumped Liquids Containing Solids. For sealing pumped liquids containing solids, a dual gas seal with a pressurized gas supply system is available (Figure 13-34). The sliding face of the hard/hard material

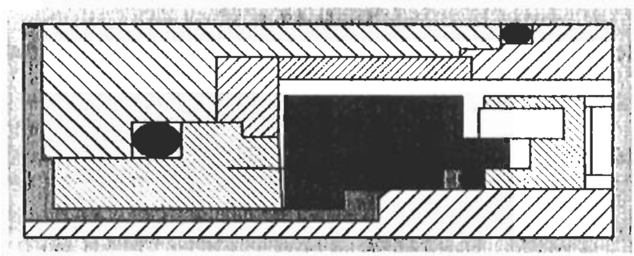

Figure 13-32. The seal face for OD and ID pressurization stays closed in case of reversed pressure.

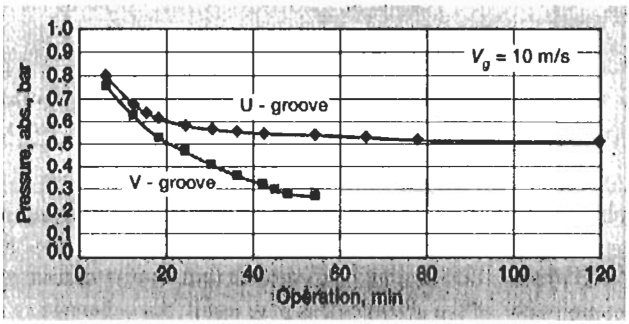

Figure 13-33. Pressure developed in an unvented seal chamber of a dual, pressurized gas seal with reverse pressure capability.

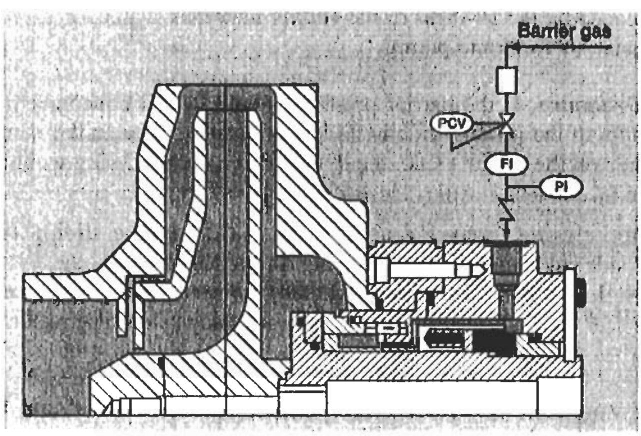

Figure 13-34. This dual gas-lubricated seal handles solids-containing liquids.

combination has gas grooves on the inside. The sealing system can also be operated with unpressurized or pressurized supply fluids.

Gas-Lubricated Safety Seals. If the German "Störfallverordnung" (regulation in case of emergency or failure) applies, a product-side liquid-lubricated seal can be backed up with a compact gas seal (CGS) as a safety seal behind the other seal. During normal operation, this safety seal runs contact-free. In case of emergency, the CGS is pressurized with the product and operates as a liquid seal until the pump has been shut down.

Hermetically sealed pumps with magnetic drives can be equipped with conventional seals or noncontacting seals as safety seals. The compact gas seal will be mounted behind the magnetic coupling can. In case of failure the gas seal will take over the sealing function until the pump has been shut down.

Field Experience. Since the mid-1990s, gas-lubricated dual seals have been increasingly applied for sealing pumps and blowers. Operating conditions of some bi-directional gas seals are listed in Table 13-3.

Compact gas seals ensure trouble-free field operation. Seal supply gas consumption corresponds to the calculated test values. Only in very few cases were there some irregularities. One of the CGS-D/33 seals, for example, (Table 13-3, item 1) had to be inspected after 650 hours of operation when supply gas consumption rose above the permitted value. Upon disassembly, the seal was found with sliding faces still in good operating condition. A detailed inspection showed that the product-side seal face could not move axially under the specified operating temperature. This was due to the temperature-dependent volume increase of the dynamically loaded O-ring.

Table 13-3
Operating Conditions of Compact Gas Seals (CGS) in Field Operation

Item	Seal Type	Medium	Characteristic Pressure, p, (bar) Product Gas	Supply Gas	Speed Min-1	Temp., °C	Notes
1	CGS-D/33	Phenol	5	7	2,895	170	Secondary sealing elements made of perfluororubber. Supply gas: N_2
2	CGS-D/53	Isocyanate	3	5	1,450	180	Consumption of supply gas, N_2, approx. 0.18 Nl/min.
3	CGS-D/60	Mixed acid 15% DNA 20% HNO_3 30% H_2SO_4 remainder water	3	5	1,450	80	Product-side secondary sealing elements made of perfluororubber, supply gas N_2
4	CGS85	Air	—	0.2	2,900	190	Blower seal

The resulting increased O-ring compression blocked the seal face movability. After reducing the O-ring compression, the seal could function again.

The single gas seal (Table 13-3, item 5) installed in a blower failed shortly after startup. The stationary seat was destroyed. An inspection showed that the radial clearance between seat and shaft was equivalent to the maximum shaft radial displacement. When starting the blower, the shaft contacted the ID of the seat and destroyed it. These problems could be eliminated by modifying the seal design.

Advantages. The main advantages when applying a compact gas seal are:

- Uncomplicated gas supply system
- No product emission to atmosphere
- No wear
- No additional cooling
- Pump efficiency increase.

The self-dynamic and seal-regulating stable gas film of the compact gas seal ensures high operating safety in pump applications.

The Reliability Impact of Special Seals for Non-Pump Applications*

Fundamentals of Seal Reliability

To achieve optimum equipment performance and reliability, it is often best to step back and look at the fundamental technology and approaches for designing and sealing equipment before one selects a given sealing system.

What was a good sealing solution ten years ago may not be a good solution for today. In our dynamic industry, the reasons for technical change in seals and sealing systems are being affected by a new set of variables. Currently, some of the variables that affect the sizing and selection of seals and sealing systems include the following:

- Tighter government controls limiting allowable leakage of fluids to the atmosphere. Rates deemed acceptable in the mid-1980s may not be acceptable leakage rates today. This is particularly true in the hydrocarbon processing industry where safety, environmental control, and health hazard considerations are becoming a bigger concern.
- The user community is demanding sealing systems that provide increased reliability, have longer life, and are more forgiving to the off-design operation of the equipment. Major refineries and chemical plants around the world establish programs to make seal installations simpler and more foolproof. Expected seal life, which in 1986 averaged nine months in many of the fluid processing industries, is

*Contributed by W. V. Adams, W. S. Binning, and R. L. Phillips, Flowserve Corporation, Kalamazoo, Michigan.

being pushed toward a five-year goal in refineries and chemical plants in the United States and around the world.

- Increased emphasis on economics and energy conservation is forcing seal manufacturers and end users to select new seal designs that will run at higher temperatures, pressures, and speeds than ever before. In some of these instances, the requirements are met by simple changes in materials, but more frequently, radical changes in design and fundamental modes of operation are required in order to achieve the performance advantages sought by the end user.
- Another factor adding to the changes in seal design and sealing systems is the basic understanding of seal technology. The basic way we do things has been changing. Fundamental modes of lubrication, seal materials, and analytical tools have advanced significantly, greatly expanding the range of application and enhancing the performance of mechanical seals.

A broad overview of fundamental seal technology will help end users understand how fundamental changes may result in more economical and reliable sealing solutions in the future.

Seal Classification

All sealing devices can be classified into two generic classifications: static and dynamic (Figure 13-35).

Static seals include such products as gaskets, sealants, and direct-contact sealing devices. Dynamic seals, which will be the broader point of discussion in this segment, are classified into two broad categories: rotating seals and reciprocating seals. Our focus will primarily be on rotating sealing devices.

Types of Seals

Under the broad classification of axial end-face mechanical seals, there are two types of seals: pusher and non-pusher seals, which can be found in four different arrangements: single, double, tandem, and staged.

Pusher-type seals refer to axial end-face mechanical seals with a semi-dynamic secondary sealing device (Figure 13-36).

The term *semi-dynamic secondary sealing device* is used to describe the O-ring or other secondary sealing device that must move backward and forward to accommodate wear at the seal faces and to accommodate vibration and axial run-out of the seal faces.

Pusher-type seals with elastomers or PTFE secondary seals are fundamentally susceptible to hang-up and fretting damage, which are characteristics of this seal design.

Hang-up is a seal term used to define the failure of components to move axially along the shaft under applied spring loads and hydraulic forces (Figure 13-37).

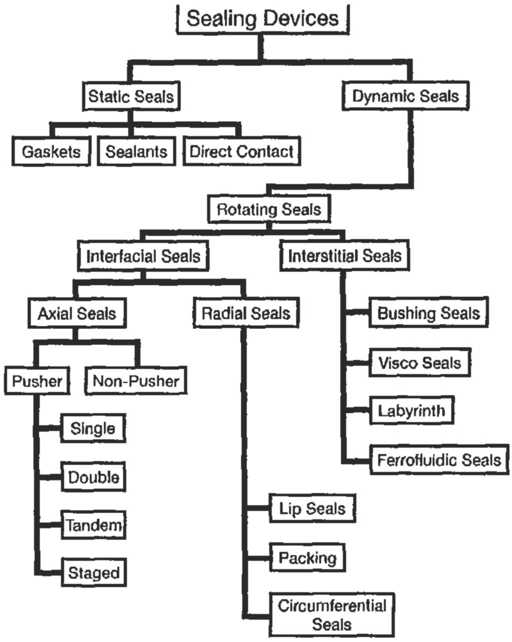

Figure 13-35. General classifications of seals.

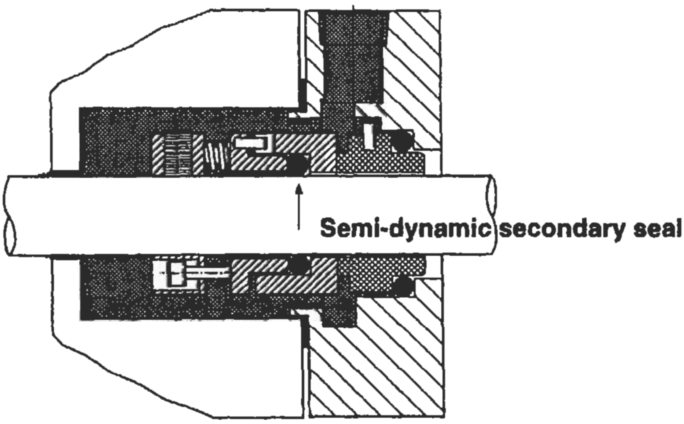

Figure 13-36. Typical inside pusher seal.

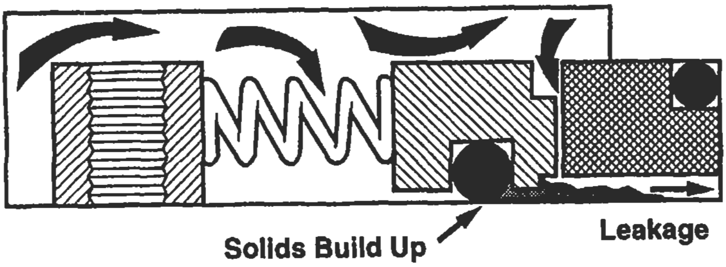

Figure 13-37. Pusher seal hung up.

Secondary seal hang-up may result from deposits that form on the atmospheric side of the seal as shown in Figure 13-37. Other sources of secondary seal hang-up are internal friction between the secondary sealing device and the shaft sleeve. This may be the result of a rough surface finish, lack of lubrication, or swelling of the secondary sealing device due to temperature or chemical attack.

Non-pusher-type seals include seals such as metal bellows, PTFE bellows, and elastomer bellows seals that do not require the semi-dynamic secondary seal to accommodate axial movement due to wear, vibration, and run out (Figure 13-38).

Axial movement is accommodated internally in the bellows portion of the seal. Bellows seals, however, are generally less commercially available for wide ranges of pressures and materials of construction.

Modes of Lubrication

Recent technology advancements have provided seals that operate in one of four basic modes of lubrication for various pieces of rotating equipment. One of these four basic modes of lubrication will provide the lubrication that is necessary between the seal faces. Knowledgeable seal manufacturers and end users quite frequently choose between two or more of these modes of lubrication in selecting seals and sealing systems for a given application. In doing so, reduced leakage, longer seal

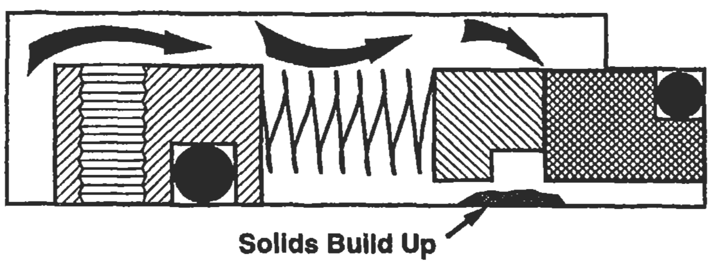

Figure 13-38. Non-pusher seal.

life, reduced power consumption, and reduced emissions to the atmosphere can be achieved. The four basic modes of lubrication include full-fluid film lubrication with either liquids or gases and boundary lubricated seals with either liquids or gases (Figure 13-39).

Full-Fluid Film Gas and Liquid Lubrication. This terminology is applied to seals that for all practical purposes are constantly separated by a full-fluid film. Because there is no contact between the mating faces, there is virtually no wear, and seal designers are able to predict long life and stable seal performance. The pressure profile generated across the seal faces is a function of the seal face geometry and seal face deflections that occur under pressure and temperature transient conditions in the seal cavity. Evaluation of these seals defies older methods that assumed a linear pressure drop across the seal faces. This technology is applied to a new family of high-performance seals for both liquids and gases. Until a few years ago, most seal manufacturers were not seriously involved in applying full-fluid film lubrication to axial end-face mechanical seals. Today, most seal manufacturers work with the four fundamental modes of lubrication to achieve higher performance levels in their sealing devices. Figure 13-40 shows the relative performance that is anticipated for full-fluid film liquid seals in comparison with conventional sealing devices. In general, an order of magnitude higher in pressure and speed capability is expected over conventional boundary lubricated seals. Also, full-fluid film seals typically consume 95% less power than do double contacted liquid lubricated seals (Figure 13-41).

Boundary Lubricated Gas and Liquid Seals. This terminology refers to the lubrication that occurs between two seal faces that rub under light or moderate loads. In general, closing forces vary from a few pounds per square inch to several hundred pounds per square inch. Lubrication that occurs between the faces is the result of surface waviness, porosity, and surface roughness. The pressure profile of the seal faces closely approximates the linear pressure drop that has been proposed in many commercial publications over the past years. In the hydrocarbon processing industry, this has been the primary mode of lubrication for the past 30 years. Boundary lubricated seals provide minimal single-seal leakage rates and allow for substantial spring loads to overcome pusher-type seal secondary seal hang-up.

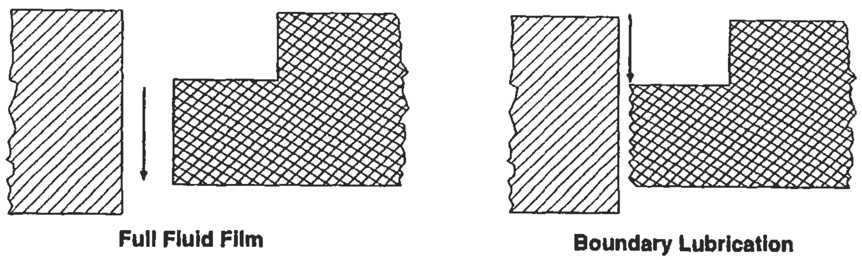

Figure 13-39. Lubrication modes.

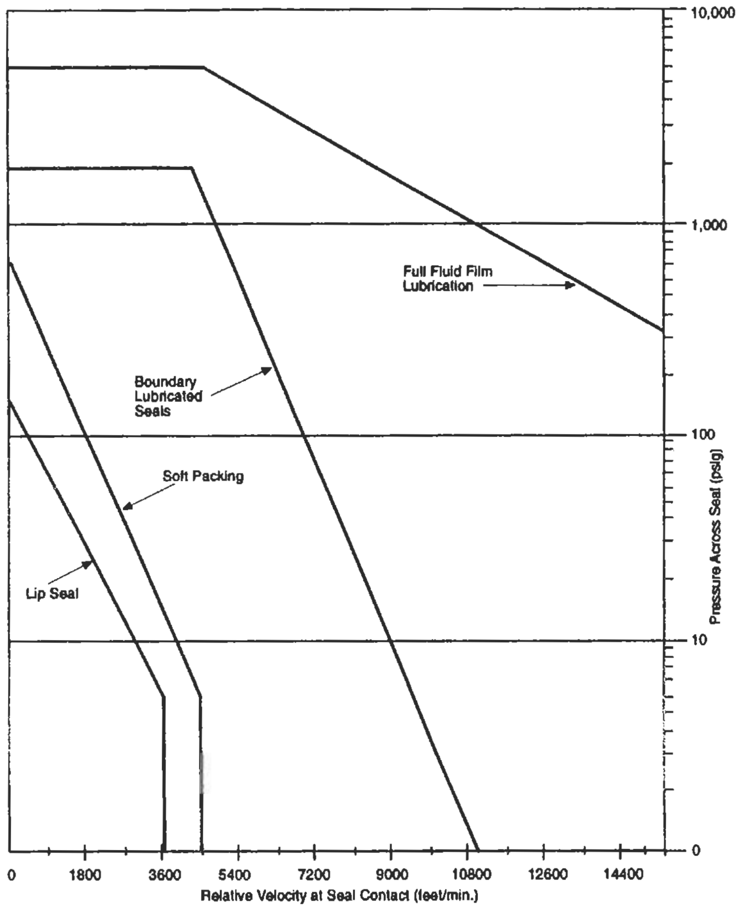

Figure 13-40. Operating limits of boundary and full-fluid film seals.

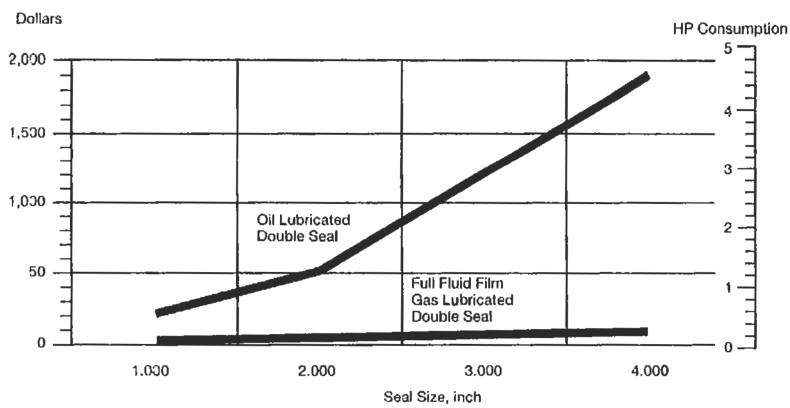

Figure 13-41. Annual energy cost and power consumption for oil vs. gas lubricated seals.

Basic Seal Geometry

Another factor having significant impact on seal selection for various applications is our understanding of basic seal geometry and the advantages and disadvantages that various seal geometries offer. In general, most commercial seal designs can be classified by one of four basic seal geometries (Figures 13-42 through 13-45).

None of these arrangements is considered a superior design for all applications. But, under given operating conditions, each design has recognized advantages and disadvantages that should be understood. Table 13-4 summarizes the four basic seal geometries and what are recognized as some of their basic advantages and disadvantages.

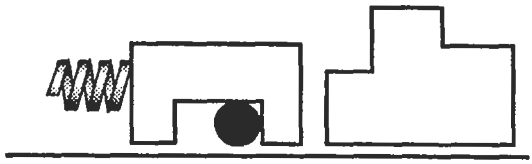

Figure 13-42. A flexible rotor with the wear face on the stator.

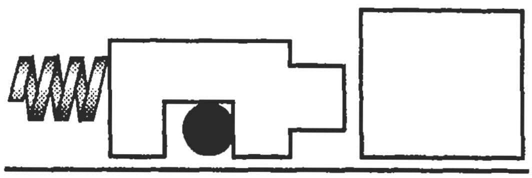

Figure 13-43. A flexible rotor with the wear face on the rotor.

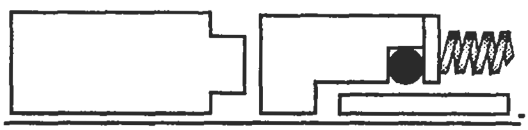

Figure 13-44. A flexible stator with the wear face on the rotor.

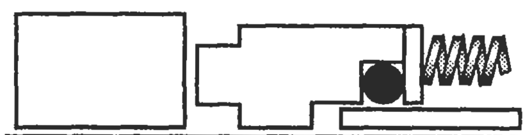

Figure 13-45. A flexible stator with the wear face on the stator.

Table 13-4
Basic Seal Geometry

Design Varieties	 **Shaft**	 **Shaft**	 **Shaft**	 **Shaft**
Questions relating to seal performance	1. Flexible Rotor Wear Face on Stator	2. Flexible Rotor Wear Face on Rotor	3. Flexible Stator Wear Face on Rotor	4. Flexible Stator Wear Face on Stator
Secondary seal fretting due to stator misalignment	Yes	Yes	No	No
Secondary seal fretting by non-wearing face being out-of-square	No	Yes	No	Yes
Secondary seal fretting by wearing face being out-of-square	Yes	No	Yes	No
Parallel misalignment of seal faces results in hydraulic load imbalance, tilt stability	Yes	No	Yes	No
Seal design limited speed wise due to radial load support of the secondary seal	Yes	Yes	No	No

As a result of industry's understanding of basic seal geometry, some trends are expected over the next five to ten years away from the more conventional flexible rotor design (with the wear face on either the stator or the rotor) to a flexible stator design with the wear face on the stator. The potential advantage of this seal geometry has been recognized on critical high-pressure and high-speed applications. This seal geometry offers these two advantages.

1. Parallel misalignment of the seal faces with respect to the shaft or seal housing will not cause hydraulic load imbalance in the flexible stator design with the wear face on the stator. This is not the case with the flexible rotor design when the wear face is on the stator (Figures 13-46 and 13-47).
2. The flexible stator design eliminates fretting damage due to out-of-perpendicularity between the gland or seal flange and the shaft axis. This is not the case with the flexible rotor design with the seal face mounted on either the stator or rotor portion of the seal (Figures 13-48 and 13-49).

Specialty Seals for Non-Pump Applications

Nowhere is it more important to apply the fundamentals of basic seal geometry, seal types, and modes of lubrication than in the application of mechanical seals to

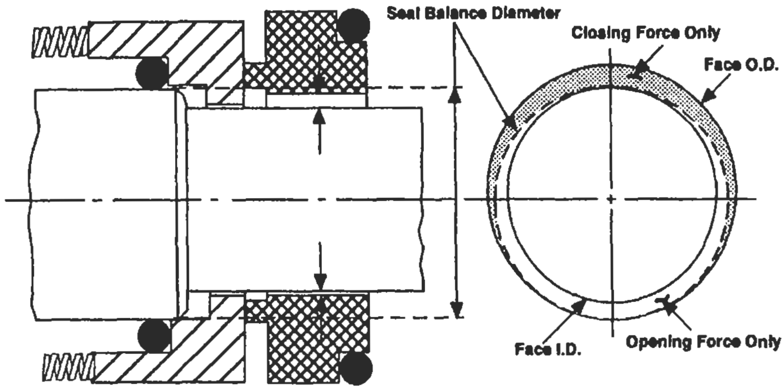

Figure 13-46. Flexible rotor with wear face on the stator.

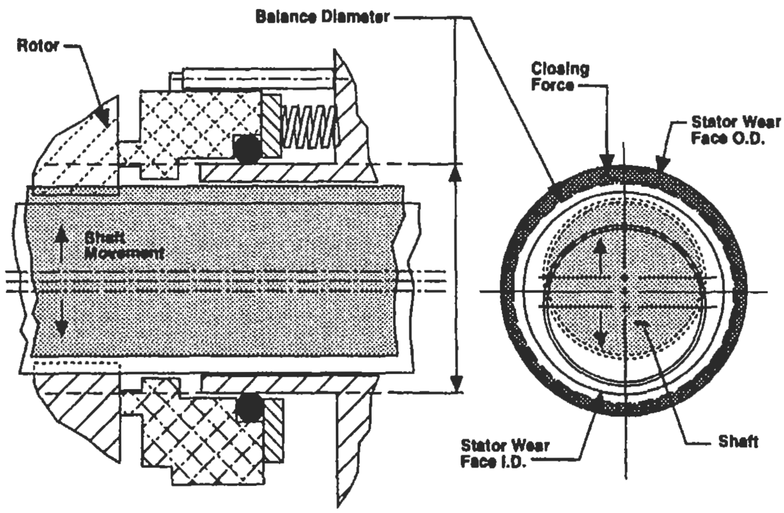

Figure 13-47. Flexible stator with wear face on stator.

specialty non-pump equipment such as compressors, mixers, centrifuges, and steam turbines.

Traditional mechanical seals that have been designed for pumps simply won't work on many of these specialty pieces of equipment. Table 13-5 shows some of the predominant operating conditions that must be considered when designing seals for these pieces of equipment.

The remainder of this section of this chapter will address the application of specialty mechanical seals to these non-pump applications and the impact that can be achieved on the reliability of the rotating equipment.

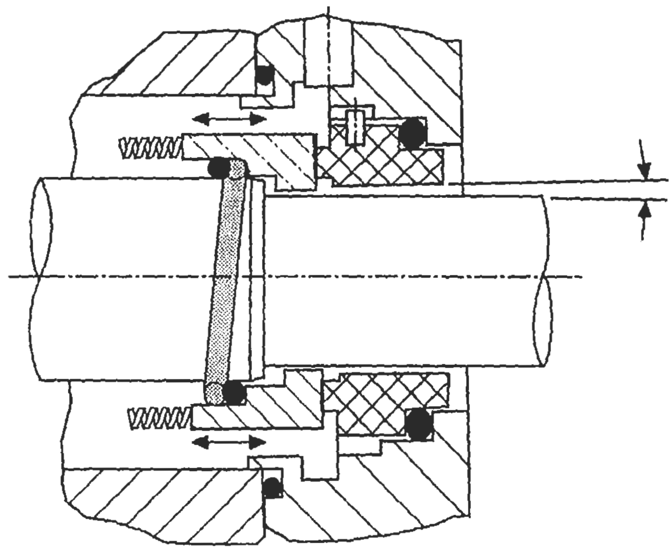

Figure 13-48. Flexible rotor with wear face on stator.

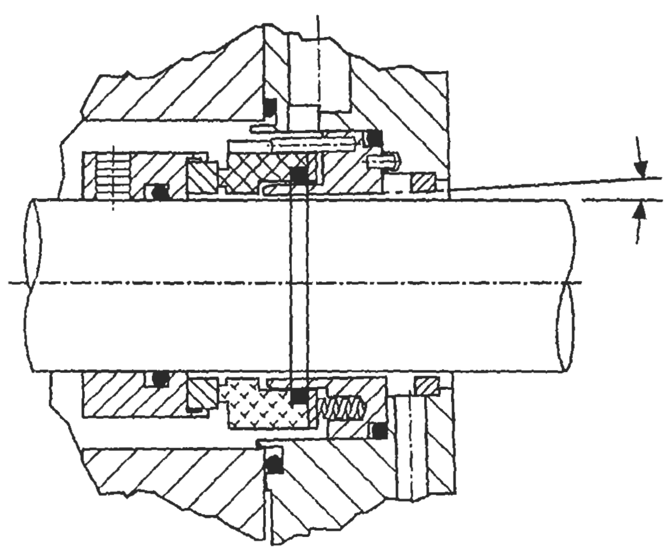

Figure 13-49. Flexible stator with wear face on stator.

Table 13-5
Specialty Equipment Performance Characteristics

Equipment	Axial Movement	Vibration	High Speed	High Pressure	Perpendicularity	Concentricity	Run-Out	High Temperature
Mixer				•	•	•	•	
Compressor			•	•				
Centrifuge	•	•						
Steam Turbine			•		•			•

Mixers

Approximately 30% of the 1.5 million mixers installed in the United States are sealed with some type of mechanical seal. More will undoubtedly be sealed in the future as reliability, safety, and environmental issues become greater factors. Examples of common mixer configurations are shown in Figure 13-50.

Mixer Seal Reliability

Depending on the style and type of mixer and whether the mixer was originally designed for packing or mechanical seals, the operating conditions for the seal may vary widely. Since the mid-1970s, major reliability problems have been found in six major areas that include:

- Excessive shaft orbiting
- Stationary seal face warpage
- Pressure reversals on double seals
- Wear debris contaminating the product
- Barrier fluid leakage into the vessel
- Non-coupled equipment designs resulting in costly maintenance cycles

The following paragraphs discuss each one of these reliability issues and how they can be overcome using current seal technology.

Excessive Shaft Orbiting. Shaft run-out or orbiting is measured by using a dial indicator and measuring the F.I.M. (full indicator movement) runout at the O.D. of the shaft at the face of the seal chamber (Figure 13-51).

Many mechanical seals are installed on mixers with a bearing support in the seal canister that limits shaft deflection at the seal faces. In other instances, especially when retrofitting packed mixers to mechanical seals, the bearing may not be present and shaft deflection or orbiting can occur in the seal chamber area to levels that will cause contact between the shaft and stationary components of the mechanical seal. Conventional seals designed for mixer canisters with integral bearing support can only tolerate small runouts, less than 0.062 inch. When the packing is removed, orbiting of the shaft in the stuffing box area may be as much as 0.150 inch F.I.M. One should be aware of these runout conditions before selecting a seal for a mixer.

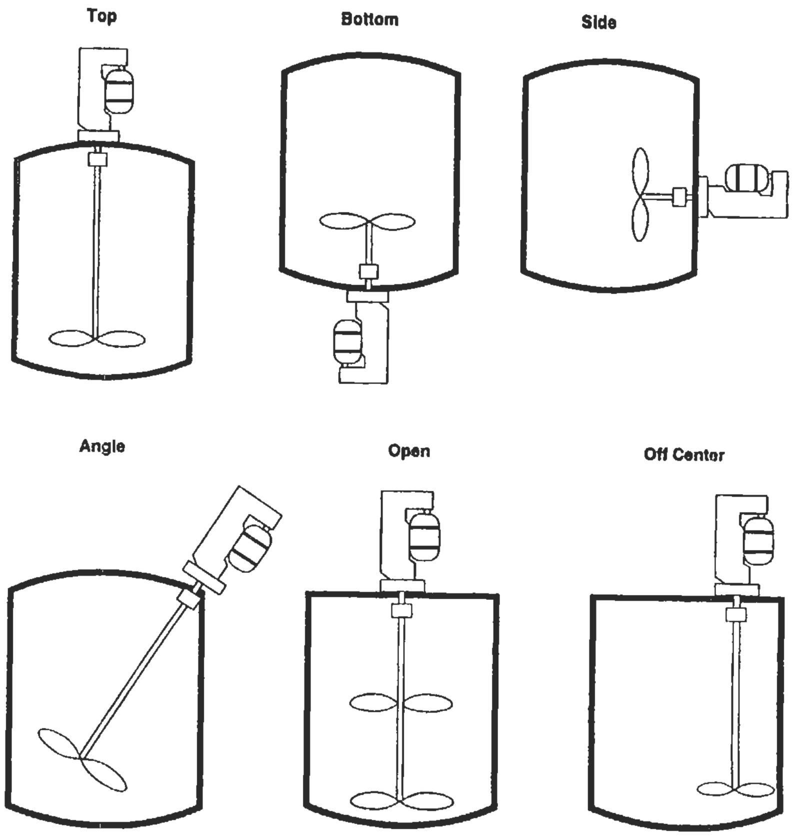

Figure 13-50. Various types of mixers.

New mixer seal technology is available making tolerances up to 0.250 inch F.I.M. at the seal chamber possible. These seals have larger radial clearances between the shaft and wider seal faces to prevent over-wipe of the seal faces during standard operation. Figures 13-52 shows an example of conventional mixer seal technology. Figure 13-53 shows the same seal design with greater runout capabilities.

Stationary Seal Face Warpage. One major problem found to cause excessive leakage on mechanical seals adapted to mixers is warpage of the stationary seal face on either the inboard or the outboard seals. Clamping loads on the stationary seal face between the seal housing and upper mixer seal flanges as shown in Figure 13-54 should be avoided. Some manufacturers choose to clamp the stationary seat with the seal housing against the flange using a gasket to cushion these loads. While clamping loads are somewhat reduced, deflections at the seal faces of 50 to 300 lightbands still occur. This can cause unstable startup problems with the mechanical seals in

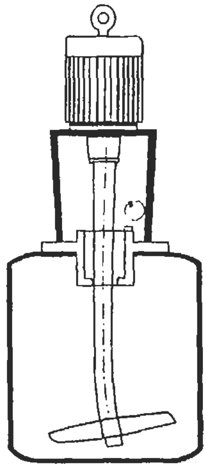

Figure 13-51. Mixer shaft "orbiting," i.e., operating with excessive runout.

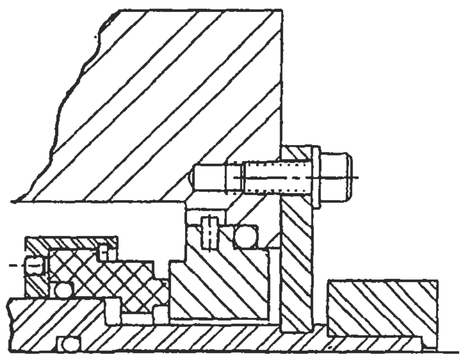

Figure 13-52. Conventional mixer seal clearances.

terms of leakage to atmosphere and into the vessel. A better solution, shown in Figure 13-55, is to isolate their inboard stator face from these large clamping loads and thermal and mechanical distortion of the vessel flange.

Pressure Reversals Unseat Seal Faces. Pressure reversals or loss of barrier fluid pressure can result in a reversal of hydraulic loads on the inboard seal hardware. If the seal hardware is not properly retained using a lock ring design for the inboard insert and reverse pressure capabilities for the rotating seal ring, the seal faces can blow open and unseat the stationary seal face. Pressure reversals on mixer seals is a common failure mode in industry. This can be prevented by designing for and ordering mixer seals with full pressure reversal capabilities as is typical of the seal shown in Figure 13-56.

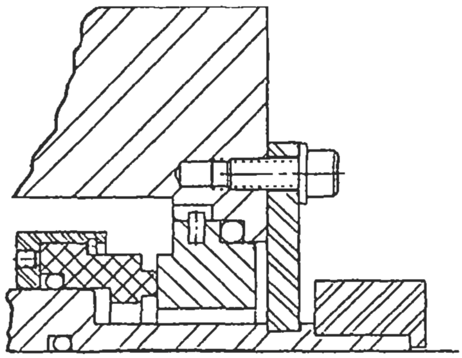

Figure 13-53. Mixer seal with increased runout capabilities.

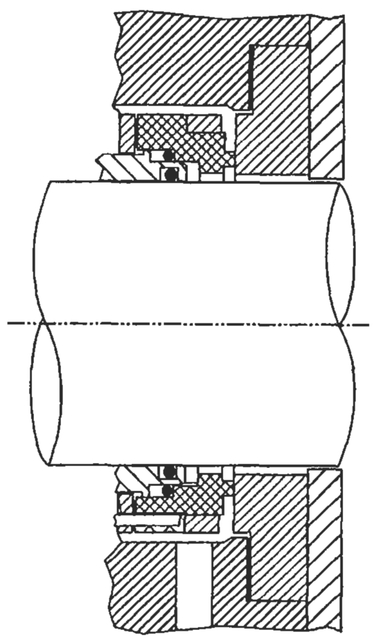

Figure 13-54. Clamped-style stationary seal face.

Wear Debris Contaminates Product. Typical wear that occurs at the seal face may over time accumulate and drop into the mixer vessel. This is normally not objectionable because only minute amounts of carbon-wear debris are actually occurring. If the product, however, is an injectable drug or has extremely tight purity specifications, this wear debris may become objectionable. Objectionable wear debris can be eliminated from falling into the mixer vessel by use of the sanitary

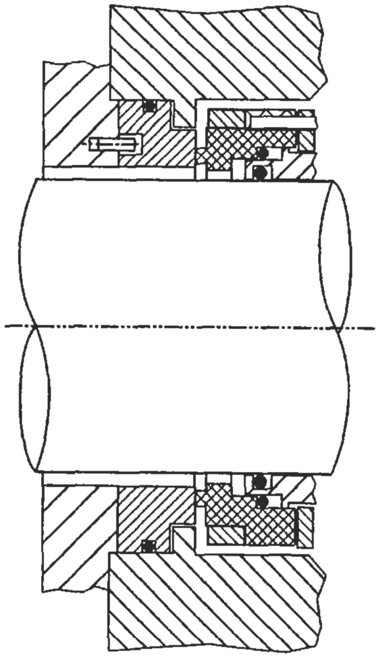

Figure 13-55. Non-clamped-style stationary seal face.

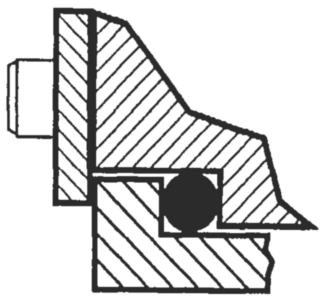

Figure 13-56. Retained stationary seal face.

design feature shown in Figure 13-57. Another solution to objectionable wear debris is full-fluid film gas or liquid seal technology.

Barrier Fluid Leakage Into Vessel. Similar to wear-debris contamination of the product is product contamination due to seal barrier fluid leakage into the process stream. Quite often this is assumed to be a norm and processes are designed to clean up the purity of the product later in the cycle. This is not necessarily the only solution. Both the sanitary design feature shown in Figure 13-57 and gas barrier sealing technology can be considered as well. Double gas seal technology for mixers operating at almost any pressure is available today.

Non-Coupled Equipment. A problem that often occurs when installing mechanical seals on older equipment is that the large non-coupled mixer shaft does not have a

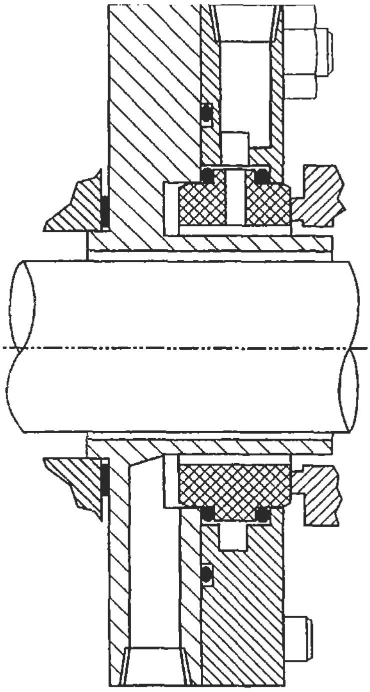

Figure 13-57. Mechanical seal with debris catcher.

coupling either inside or outside the tank to facilitate the installation of a mechanical seal. This kind of installation usually requires pulling the entire mixer drive and reworking the equipment, which can result in hiring large cranes and maintenance crews and spending literally days of process time to retrofit the technology. In these instances, split seal technology may be a wise choice in order to avoid costly maintenance problems while providing near zero leakage characteristics from the mechanical seal (Figure 13-58). Both liquid and dry running split seal technologies are available for mixers.

Table 13-6 summarizes the most common mixer rotating equipment reliability issues and their solutions.

Centrifuges

Solid bowl and screen bowl centrifuges are used to separate solids from liquids. Usually the internal parts of the centrifuge, the bowl and the screw conveyor, rotate at a speed between 1,000 and 4,000 rpm. The bowl, located over the conveyor, rotates in the same direction as the conveyor but at a small differential speed that is designed into the differential gear box on the equipment. The differential speed is usually between 10 and 100 rpm, either faster or slower than the conveyor speed. There are three distinctly different sealing locations on a typical centrifuge that must be properly application-engineered in order to achieve adequate centrifuge performance. These sealing points include:

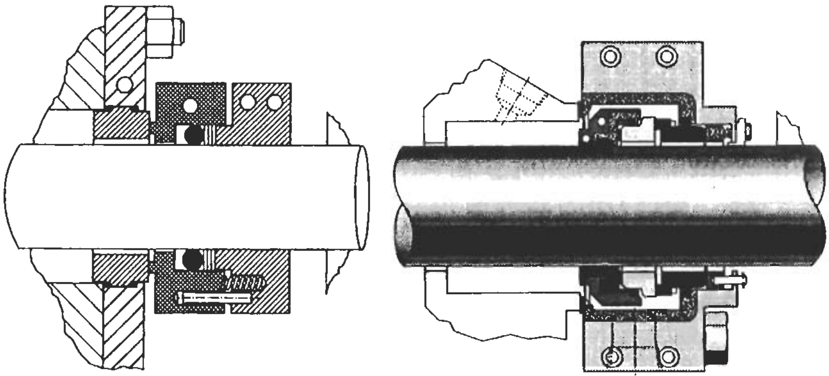

Figure 13-58. Split seal technology.

Table 13-6
Rotating Equipment Reliability Problems

Mixers	Reliability Issue	Recommended Solution
	Shaft orbiting	Make sure basic seal design and geometry accommodate orbiting and runout.
	Stationary seal face warpage	Avoid clamp-style stationary seal face.
	Pressure reversal unseats seal faces	Make sure seal design offers pressure reversal.
	Wear debris contaminate product	Investigate stationary design feature.
	Barrier fluid leakage into vessel	Consider gas seal versus liquid seal technology.
	Non-coupled equipment	Split seal technology.

- The case seal that seals between the centrifuge casing and the rotating shaft conveyors
- The feed pipe seal that seals between the static feed pipe and the rotating bowl head shaft
- Internal seals that seal the bearing internal to the centrifuge

Pressurized centrifuges typically use one of three different types of sealing devices depending on the temperature, pressure, speed, and level of emission control that is desired for the application. These sealing devices include:

- Dynamic seals—usually a repeller or expeller type liquid ring seal
- Segmented bushing seals
- Axial end face mechanical seals

Centrifugal Seal Reliability

Major reliability factors found over the years relative to these seal selections have included:

- Concentricity and perpendicularity and axial shaft growth problems with casing seals
- Dynamic seal leakage under static conditions
- Product contamination by the buffer fluid
- O-ring retention problems

Concentricity, Perpendicularity, and Axial Growth. Case seal designs assure good concentricity, perpendicularity, and axial growth capabilities up to ±0.500 inches by mounting the casing seals perpendicular and concentric with the bearing housing. Case seals are typically piloted and registered off the bearing housing to assure concentricity and perpendicularity with the axis of the shaft. Thermal growth differences between the shaft and the centrifuge casing are accommodated through the use of an expansion joint between the casing seal housing and the centrifuge housing. This joint may be sealed with packing, O-rings, or a flexible metal bellows (Figure 13-59).

Dynamic Seal Leakage. Upgrades from dynamic seals to either mechanical end face seals or segmented bushing seal housings with a constant barrier fluid purge are often required in order to achieve true leak-free performance while centrifuges are static. Proper consideration of environmental legislation concerning the process fluid being handled should address these conditions. Retrofit packages ranging from dynamic seals to segmented bushing seals or mechanical seals to achieve lower emissions under static standby conditions are available.

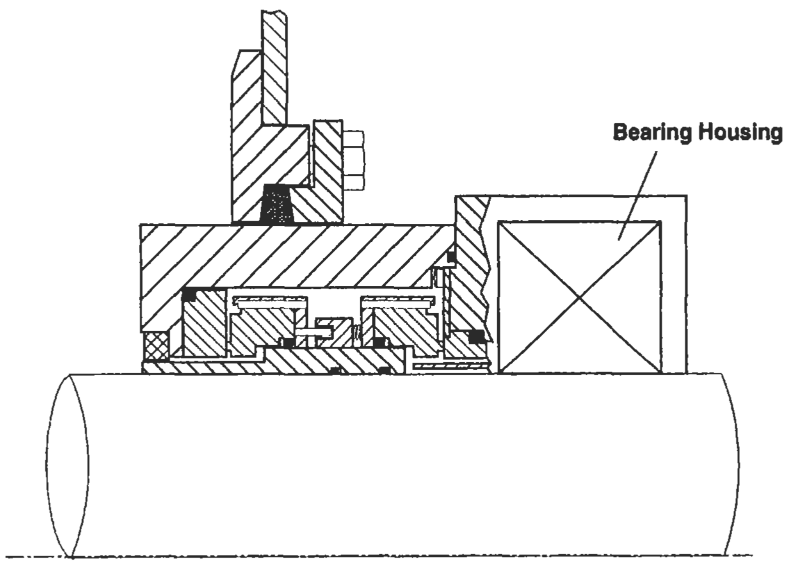

Figure 13-59. Seal canister secured to bearing housing.

Product Contamination by Buffer Fluid. In order to avoid liquid contamination of the product by a buffer fluid, retrofits to segmented bushing seal packages or mechanical seals using gas barrier technology are options. Usually an inert gas barrier fluid can be used as a substitute for liquid barriers in order to avoid product contamination.

O-Ring Retention Problem. One of the most annoying maintenance and reliability problems is assembling a seal cartridge with large diameter O-rings at the I.D. of the shaft sleeve or at the faces of glands and have it fall down and get crimped, pinched, or out of place when the equipment is assembled into position. These loose gaskets at the I.D. of sleeves and at faces of glands can be avoided through the use of dovetail-type O-ring grooves (Figure 13-60), rather than conventional rectangular O-ring grooves.

Table 13-7 summarizes centrifuge seal reliability problems and solutions.

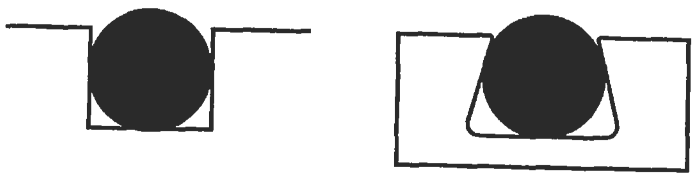

Figure 13-60. Rectangular versus dovetail O-ring groove.

Table 13-7
Rotating Equipment Reliability Problems

CENTRIFUGE	Reliability Issue	Recommended Solution
	Axial shaft growth	Seal cartridge mounted to bearing frame with axial expansion accommodated in seal housing design.
	Dynamic expeller seal leaks static	Upgrade to mechanical seals or segmented bushing with constant barrier fluid pressure.
	Product contamination by buffer fluid	Segmented bushing seal or mechanical seal with gas barrier technology.
	O-ring retention problem	Use dovetail O-ring groove to aid installation.

Steam Turbines

Steam turbines are used in many industries as mechanical drives where electric power is either unreliable or unavailable. Figure 13-61 shows the components of a single-stage steam turbine. Sealing devices are used to prevent steam from leaking past the shaft where it passes through the turbine casing. The majority of general-purpose, single-stage steam turbines are sealed with a series of segmented carbon bushings, (Figure 13-62). Labyrinth seals are also used as sealing devices. Mechanical seals using gas sealing technology were used at an Exxon plant in 1982 and are now gaining acceptance as a superior sealing alternative for many steam turbine applications.[27]

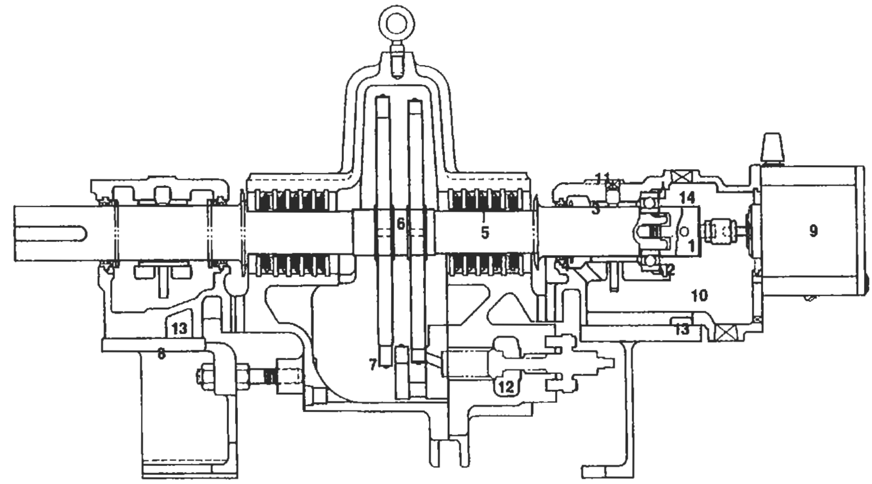

Figure 13-61. Single-stage steam turbine showing carbon ring seals, Item 5. *(Courtesy of Dresser-Rand, Wellsville, New York.)*

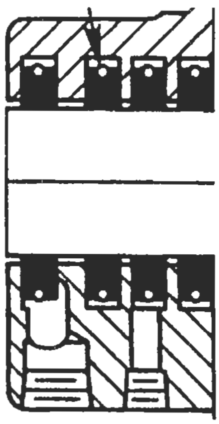

Figure 13-62. Segmented carbon bushing seal.

1. Bolt-type overspeed trip
2. Thrust bearing
3. Journal bearing
4. Labyrinth-type bearing case oil seals
5. Carbon-ring packing
6. Turbine wheels
7. Blading
8. Support bracket
9. Governor
10. Internal oil reservoir
11. Instrument port

12. Hand-operated nozzle control
13. Water cooling jackets
14. Overspeed trip adjustment port

Table 13-8 summarizes the typical characteristics of the predominant seal types that are used in general-purpose, single-stage, steam turbines with a 3-inch shaft size. The initial capital cost represents the cost of the two sets of seals that are required to seal a steam turbine. The operating cost per year includes parts, labor, and the cost of steam loss based on a steam cost of $0.005 per pound. The repair costs include labor as well as the replacement of worn seal components and any damaged steam turbine components. Due to the high leakage rates associated with segmented carbon bushings and labyrinth seals and the detrimental effect that this leakage has on bearing life, the cost of replacing the bearings is included in the repair costs for these two seal types.

The cost of the segmented carbon bushings is based on between 8 and 12 bushings per steam turbine at $50 per bushing. The mechanical seals are estimated at $5,000 per seal. The expected life of the labyrinth seal is not listed because, assuming proper material selection and sizing, the labyrinth should not wear or corrode and should last indefinitely. Although the mechanical seals that are applied in steam turbines are typically non-contacting, the secondary seals are subject to degradation in the high temperature steam, thus generating the expected life of five years. A cost analysis of the three sealing options reveals that, in spite of the high initial cost of a mechanical seal, its payback period as compared to either segmented carbon bushings or labyrinth seals is less than six months due to the reduction in operating costs (primarily due to significantly lower steam leakage) and lower repair costs.

Steam turbines offer a harsh environment in which seals must operate. This environment creates seal reliability issues that are dependent on the seal design. These issues are discussed in the following paragraphs for the three seal designs most commonly applied to steam turbines.

Segmented Carbon Bushings

The most significant problem associated with the segmented carbon bushings shown in Figure 13-62 is the high rate of steam leakage. In addition to the value associated with lost steam, the escaping steam enters the bearing lubricating oil and caus-

Table 13-8
Steam Seal Characteristics

Seal Type	Cost: Capital	Cost: Operating	Cost: Repair	Leakage	Expected Life
Segmented Carbon Bushing	$400–500	$1,500	$2,000–10,000	200 lbm/hr	6–24 months
Labyrinth Seal	$200–500	$500–1,000	$2,000–10,000	> 200 lbm/hr	n/a
Mechanical Seal	$10,000	$0	$2,000–7,500	0.4 lbm/hr	5 years

es premature bearing failure. The corrosiveness of the steam also shortens the life of other components in the vicinity of the steam turbine. The life of the carbon bushings themselves is relatively short, ranging from six months to two years. The manner in which the carbon bushings are broken in can significantly affect bushing life. Although the bushings are designed to maintain a clearance over the turbine shaft, thermal expansion rates and shaft misalignment can cause contact between the bushing and the shaft. For this reason, special treatments are often applied to the shafts under the bushings. Depending on the degree of contact between shaft and bushings, turbine maintenance procedures may require the shaft to be repaired and recoated.

Labyrinth Seals

The labyrinth seals that are used in steam turbines may include features such as steps or active loading mechanisms. As is the case with segmented carbon bushings, labyrinth seals also have high steam leakage rates that can damage bearings and other components. Clearances between the shaft and the labyrinth seal vary as a function of operating temperature due to the different rates of thermal expansion of the materials for these two components. The labyrinth seal leakage is proportional to the cube of the clearance so any variations in clearance can have a significant impact on the steam leakage. The potential for contact between the labyrinth seal and the shaft may require the application of special treatments to the shaft.

Mechanical Seals

When properly designed and installed, mechanical seals provide a greatly reduced leakage rate as compared to the segmented carbon bushings and labyrinth seals. The leakage rates of the mechanical seal are on the order of 500 times less than the alternative sealing technologies. Due to the poor lubricating properties of steam, the mechanical seals applied to steam turbines are effectively dry running seals and consequently employ gas seal technology. Figure 13-63 shows cross-sectional views of

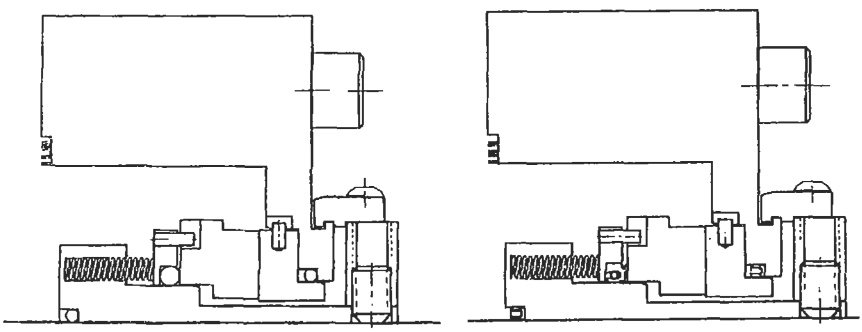

Figure 13-63. Typical flexible rotor steam turbine seal designs.

a typical steam turbine mechanical seal using both elastomer and spring energized secondary seals.

Because of the high operating temperature of steam turbines, the metallurgy of the sleeve, gland, and drive collar must be selected to match the thermal expansion rates of the turbine shaft and casing. Typically 400-series stainless steel is used for these components. The high steam temperatures also mandate that special consideration be given to the choice of secondary seals. Elastomers are not acceptable at the operating temperatures of most steam turbines. Steam turbines were not designed to accommodate mechanical seals and consequently slight rework to the turbine casing may be required to provide good surfaces from which the seal can be piloted and aligned. Additional conversion costs to install a mechanical seal are typically minor, especially for steam turbines designed with a separate, bolt-on housing for the sealing devices. The mechanical seals must also be capable of operating under slow roll conditions. When proper preheating procedures are not followed, mechanical seals may start up in water that requires them to be able to operate with liquid lubrication. Contaminants in the steam can act to clog the face patterns on non-contacting gas seals, diminishing the available load support and possibly resulting in contact between the seal faces. If mechanical end face seals are adapted to a steam turbine, quality and cleanliness of the steam must be carefully reviewed.

Table 13-9 summarizes major steam turbine seal reliability issues and potential solutions.

Table 13-9
Rotating Equipment Reliability Problems

STEAM TURBINE	Reliability Issue	Recommended Solution
	Short life and high leakage of segmented carbon rings	Axial end face mechanical seal
	Bearing housing oil contamination with water	Bearing closure device upgrade (See Figures 11-8 and 12-3)
	Dissimilar material of construction	Seal sleeve and housing must be of material with same thermal expansion rate as shaft and casing

Conclusion

Specialty mechanical seals will have a major impact on the reliability of rotating equipment in the future. The increased application of mechanical seals to non-pump rotating equipment is expected to require larger diameter seals that operate at higher pressures, speeds, and temperatures than ever before. Achieving these goals will require making practical use of our basic understanding of seal geometries and the various modes of lubrication available to us today. Technological advances to conventional seal designs will carry us beyond the simple mechanical seal with two lapped faces that operate under boundary lubrication with liquid or gas. Future seals will incorporate multifaceted seal face geometries that will operate in one of the four known modes of lubrication. To gain control over these technologies will require greater application of new technical tools, innovations in seal design, new materials of construction, and new manufacturing technology. Computer-aided design and engineering techniques such as CAD, Finite Element Analysis (FEA), and Computational Fluid Dynamics (CFD) will become common tools in the development of mechanical seals.

Seals and sealing systems will operate at a fraction of the power and energy consumption of older sealing systems. Requirements for longer life and more forgiving seals for the off-design operation of equipment will become commonplace. The impact of this changing seal technology is expected to touch every segment of our industry as we move from simple compression packing and end face mechanical seals to more cost-effective seals. In the next ten years, the sealing industry will move forward quickly as we apply new technology to a broader range of applications.

Dry Gas Compressor Seals*

Dry gas seals can eliminate some of the problems associated with conventional seal oil systems. Increased safety, lowered maintenance and improved reliability are key considerations for retrofitting with gas seals. Understanding the dry gas sealing technology and optimizing the seal selection for a given set of operating parameters is crucial for the successful application on rotating equipment. Control systems incorporating filtration, leakage, and pressure monitoring can provide real time diagnosis of seal performance.

The gas seal design discussed here is the spiral groove type. Various types of face configurations have been developed by several manufacturers since the early 1970s. As of 1997, the spiral groove face configuration design has been successfully applied on over 2,000 centrifugal compressors.

An attempt has been made to provide some seal design criteria and the different types of seal arrangements and to emphasize the need for optimum seal and control system selection. The process conditions, namely pressure, temperature, gas composition, and contaminants in the gas stream are an integral part of the seal design process. Of equal importance are the key characteristics of the compression equipment involved, namely, the rise in temperature from suction to discharge, the surface speed involved, and whether it is an overhung or beam unit, rotordynamics, etc.

Review of the location of the compression equipment in the overall process loop can provide important information on the contaminants in the gas stream. P&ID reviews can provide possible alternative choices of buffer gas supply for the seal. Above all, a change from the existing wet seal system to dry seals can have a beneficial effect on plant efficiencies by eliminating oil contamination of downstream equipment, catalysts, etc.

*Based on technical literature provided by John Crane Inc., Morton Grove, Illinois. Adapted, by permission, from papers authored by Piyush Shah;[28] J. R. Dugas, B. X. Tran, J. F. Southcott;[29] and G. G. Pecht & D. Carter.[30]

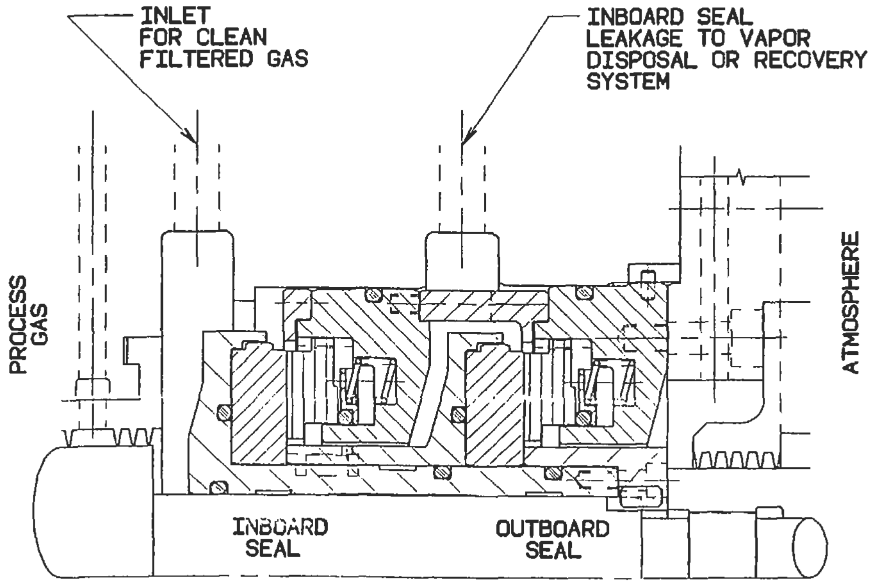

Figure 13-64. Cross-sectional view of a spiral-grooved gas seal. (*Courtesy of John Crane Inc., Morton Grove, Illinois.*)

Spiral Groove Gas Seal Design

Figure 13-64 shows a cross-sectional view of a spiral grooved gas seal. The rotating assembly consists of the mating ring (with spiral grooves) mounted on a shaft sleeve held in place axially with a clamp sleeve and a locknut. It is typically pin or key driven. The stationary assembly consists of the primary ring mounted in a retainer assembly held stationary within the compressor housing. Under static conditions, the primary and mating rings are held in contact due to the spring load on the primary ring.

Operating Principle

The operating principle of the spiral grooved gas seal is that of a hydrostatic and hydrodynamic force balance. Under pressurization, the forces exerted on the seal are hydrostatic and are present whether the mating ring is stationary or rotating. Hydrodynamic forces are generated only upon rotation. The mating ring, incorporating the logarithmic spiral grooves, is the key to generating these hydrodynamic forces.

Figure 13-65 shows the spiral groove pattern on the mating ring, rotating in a clockwise direction. As gas enters the grooves, it is sheared towards the center. The sealing dam acts as a restriction to the gas outflow, thereby raising the pressure upstream of the dam. The increased pressure causes the flexibly mounted primary ring to separate from the mating ring.[31,32] The mating ring with spiral grooves and the primary ring held within the retainer assembly are shown in Figure 13-66. Note that in this illustration, the mating ring is configured for counter-clockwise rotation.

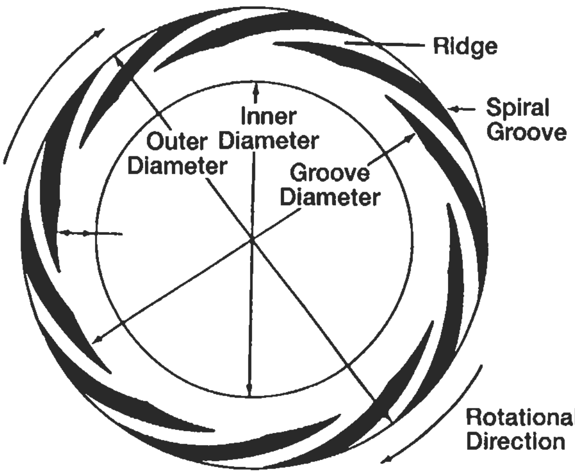

Figure 13-65. Dry gas seal mating ring for clockwise rotation. (*Courtesy of John Crane Inc., Morton Grove, Illinois.*)

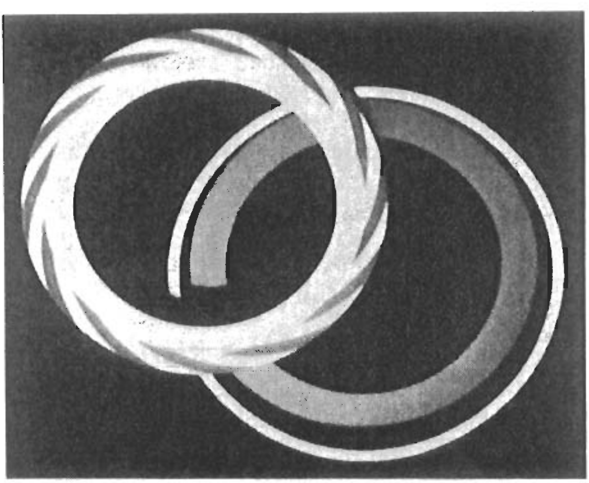

Figure 13-66. Dry gas seal mating ring for counter-clockwise rotation, shown on primary ring. (*Courtesy of John Crane Inc., Morton Grove, Illinois.*)

Performance Characteristics

Seal Leakage. Since the typical operating gaps between the two sealing surfaces range from 0.0001 in. to 0.0003 in., the resultant leakage is very small in magnitude. The size and speed effect on leakage are shown in Figure 13-67. Under conditions of static pressurization beyond 50–75 psi, the seal leaks a very small amount. This leak-

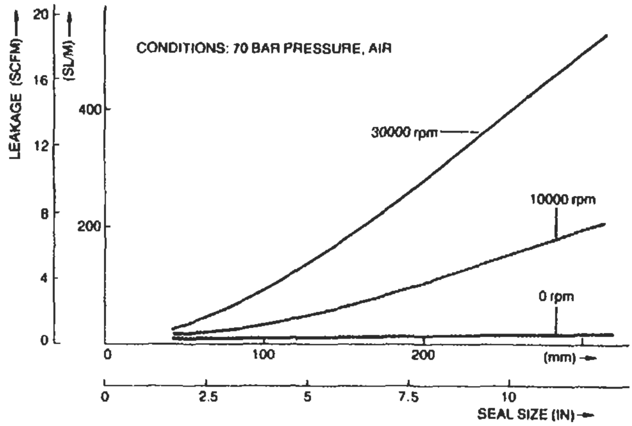

Figure 13-67. Size and speed effect on leakage. (Source: Ref. 28.)

age increases with increasing pressure. For example, a four-inch shaft seal on a natural gas compressor statically pressurized to 1,000 psi will leak about 1 scfm. Under dynamic condition, due to the pumping effect of the spiral grooves, the leakage increases as well.

Figure 13-68 shows the pressure, temperature, and gas effect on leakage. Increased viscosity of gases at higher temperatures reduces the amount of seal leakage.

Power Loss. Since the sealing surfaces are non-contacting under dynamic conditions, the power loss associated with gas seals is very low. The pressure and thermal distortions of the sealing surfaces are computed and compensated for in the seal

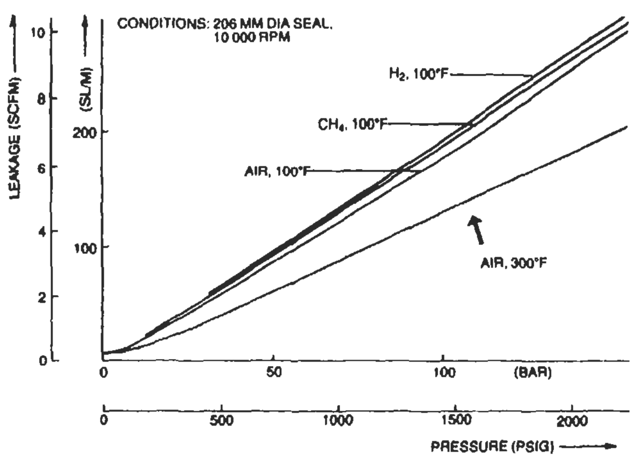

Figure 13-68. Pressure, temperature, and gas effect on leakage. (Source: Reference 28.)

design. For spiral grooved seals, the power loss associated with increasing shaft speeds is shown in Figure 13-69. The comparative economic evaluation of the wet seal and dry seal systems is listed in Table 13-10.

Seal Arrangements

There are five basic arrangements that can be prescribed for a diverse group of applications: single-straight, single educted, tandem, double-opposed, and triple.

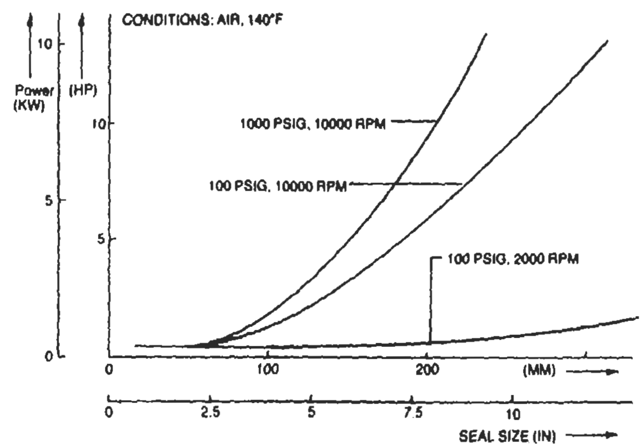

Figure 13-69. Gas seal power loss. (Source: Reference 28.)

Table 13-10
Comparative Economic Evaluation of Wet Seal vs Dry Gas Seal

	Wet Oil Seals	Dry Gas Seal
Seal oil support systems costs	Pumps, reservoirs, filters, traps, coolers, consoles.	None
Seal oil consumption	1–100 gallons/day	No seal oil
Maintenance cost	A major expenditure over equipment life.	Negligible
Energy costs	Seal Power loss: 10–30 HP Unit driven pumps: 20–100 HP	1–2 HP
Process gas leakage	Gas Leakage: 25SCFM & higher	Less than 2 SCFM
Oil contamination	Of Pipelines: High clean up costs Of Process: Catalyst Poisoning	None
Toxic and corrosive Applications	Buffer Gas Consumption (egN_2) 40–70SCFM	2–4SCFM
Unscheduled shutdowns	High downtime costs	Very reliable
Aborted startups	Frequent	Rare

Gases that are inert or nontoxic are typically sealed by a single-straight seal arrangement. The leakage from the seal is either vented or flared. On wet gas compressors in refinery services the leakage can be educted by a steam eductor and flared.

A majority of the hydrocarbon mixtures, chemical and petrochemical process gases and gases having toxic and corrosive contaminants such as hydrogen sulphide have to be sealed from the environment and the lubrication systems. In a tandem seal arrangement (Figure 13-64), two seal modules are oriented in the same direction behind each other. The first seal (inboard) handles full pressure, while the second seal (outboard) would run as standby or backup seal with zero pressure differential. The backup seal then functions as an additional barrier between the process gas and atmosphere. The primary leakage from the first seal can be safely vented or flared. The control system for a tandem seal arrangement is shown in Figure 13-70.

Triple seals incorporate three seals oriented in the same direction. This creates space for an additional buffer gas inlet, or leakage gas outlet.

In processes where zero leakage to the atmosphere is the plant safety requirement, a double opposed seal arrangement with plant nitrogen buffer should be utilized. The nitrogen buffer leaks into the process at a selective rate (<1 scfm) and also leaks (<1

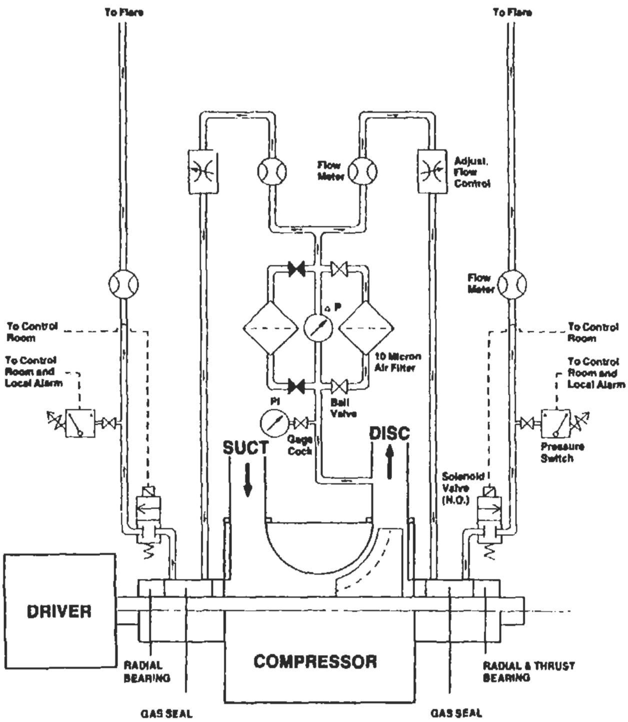

Figure 13-70. Control systems: tandem seal arrangement. (Source: Ref. 28.)

scfm) to the atmosphere. This prevents atmospheric and bearing oil contamination. A thorough investigation of the plant safety procedures must be undertaken and should incorporate the review of control system requirements for the double opposed seals.

Safety*

Increased safety of the dry running seals on compressors is attributed to the elimination of high pressure oil systems common with liquid lubricated seals. This increased safety is associated with the fact that high pressure oil lines are susceptible to leaks and at high pressures, the resulting oil spray may be fuel for a fire that could destroy an entire compressor gas turbine train.

Economic Justification

Tables 13-11 and 13-12 show the cost of ownership and payback periods for oil seals and gas seals on a 4.500-in. shaft size compressor operating at 10,000 rpm. Because the current up-front costs for gas seals are nearly identical to oil seal costs, over 75% of the new compressors purchased use gas seal technology. Cost justifications for retrofits indicate that 6 to 18 month payback period is typically required to cover the cost of retrofitting from oil seals to gas seals. The variation in the payback periods is a function of the actual compressor operating conditions and maintenance and reliability issues related to the particular compressor in question.

Compressor Seal Reliability

Compressor gas seal reliability studies have shown three primary areas of failure that include:

- Secondary seal hang-up
- Rotor disintegration
- Tolerance ring installation and assembly problems

Secondary Seal Hang-Up

Market studies indicate that semi-dynamic secondary seal hang-up is one of the primary causes for compressor gas seal failures and instabilities in seal performance. This problem is a result of not adequately compensating for O-ring volumetric swell at elevated temperatures, chemical attack, and standard deviations in O-ring manufacturing tolerance (Figure 13-71).

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*Source: W. V. Adams, W. S. Binning, and R. L. Phillips, Flowserve Corporation, Kalamazoo, Michigan.

(text continued on page 590)

Table 13-11
Oil Seals vs. Gas Seals Cost Comparison

Retrofit from Oil Seals to Gas Seals

Oil Seals — Years of Operation

Item	Start Up	1st	2nd	3rd	4th	5th
Replacement Seal Cost	na	na	$12,000	na	$12,000	na
Oil Supply System	na	na	na	na	na	na
Oil Consumption & Contam.	na	$18,980	$20,878	$22,966	$25,262	$27,789
Buffer Gas Losses	na	$31,536	$33,113	$34,768	$36,507	$38,332
Horsepower Losses	na	$130,086	$130,086	$130,086	$130,086	$130,086
Operation and Maintenance	na	$25,000	$31,250	$39,063	$48,828	$61,035
Fouling and Cleanup	na	$15,000	$15,000	$15,000	$15,000	$15,000
Unscheduled Shutdown	na	$0	$0	$0	$0	$0
Annual Total Cost	**na**	**$220,602**	**$242,327**	**$241,883**	**$267,683**	**$272,242**
Cumulative Total Cost	**na**	**$220,602**	**$462,929**	**$704,812**	**$972,495**	**$1,244,737**

Gas Seals — Years of Operation

Item	Start Up	1st	2nd	3rd	4th	5th
Seal and Control Panel Cost	$115,000	na	na	na	na	na
Installation and Commissioning	$80,000	na	na	na	na	na
Compressor Modifications	$0	na	na	na	na	na
Buffer and Process Leakage	na	$4,205	$4,205	$4,205	$4,205	$4,205
Horsepower Losses	na	$6,938	$6,938	$6,938	$6,938	$6,938
Operation and Maintenance	na	$5,000	$5,000	$5,000	$5,000	$5,000
Annual Total Cost/seal	**$195,000**	**$16,143**	**$16,143**	**$16,143**	**$16,143**	**$16,143**
Cumulative Total Cost	**$195,000**	**$211,143**	**$227,285**	**$243,428**	**$259,571**	**$275,714**

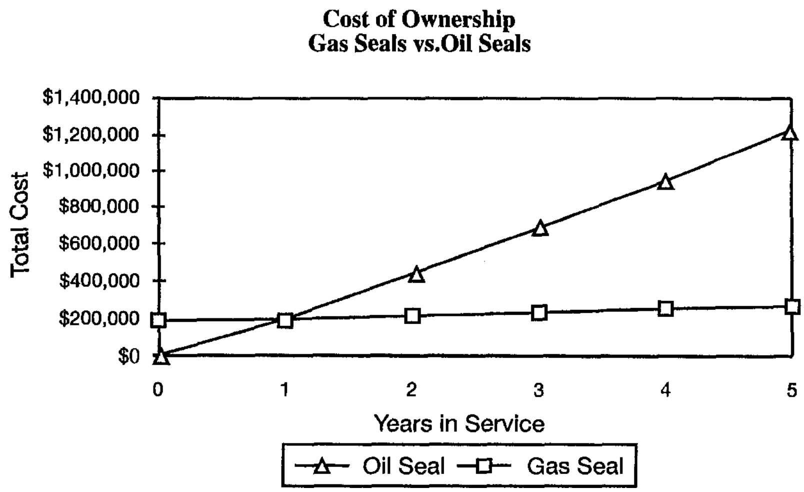


Table 13-12
Oil Seals vs. Gas Seals Cost Comparison

New Compressor with Oil Seals or Gas Seals

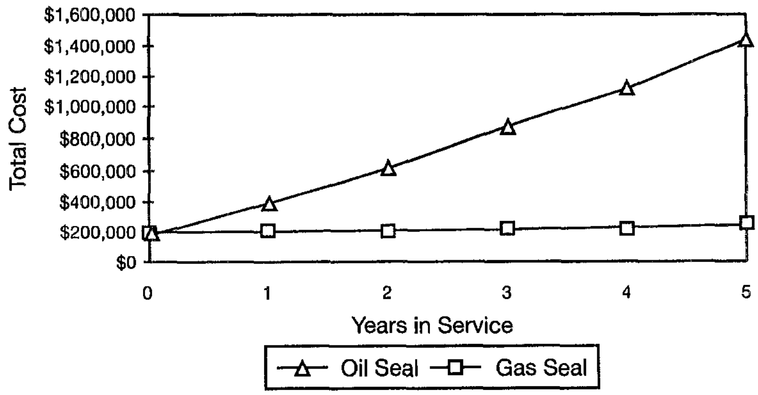
Oil Seals

		Years of Operation				
Item	**Start Up**	**1st**	**2nd**	**3rd**	**4th**	**5th**
Oil Seal Cost	$24,000	na	$12,000	na	$12,000	na
Oil Supply System	$150,000	na	na	na	na	na
Oil consumption & contam.	na	$18,980	$20,878	$22,966	$25,262	$27,789
Buffer Gas Losses	na	$31,536	$33,113	$34,768	$36,507	$38,332
Horsepower Losses	na	$130,086	$130,086	$130,086	$130,086	$130,086
Operation and Maintenance	na	$25,000	$31,250	$39,063	$48,828	$61,035
Fouling and Cleanup	na	$15,000	$15,000	$15,000	$15,000	$15,000
Unscheduled Shutdown	na	$0	$0	$0	$0	$0
Annual Total Cost	**$174,000**	**$220,602**	**$242,327**	**$241,883**	**$267,683**	**$272,242**
Cumulative Total Cost	**$174,000**	**$394,602**	**$636,929**	**$878,812**	**$1,146,495**	**$1,418,737**

Gas Seals

		Years of Operation				
Item	**Start Up**	**1st**	**2nd**	**3rd**	**4th**	**5th**
Seal and Control Panel Cost	$115,000	na	na	na	na	na
Spare Seals	$70,000	na	na	na	na	na
Installation and Commissioning	na	na	na	na	na	na
Compressor Modifications	na	na	na	na	na	na
Buffer and Process Leakage	na	$4,205	$4,205	$4,205	$4,205	$4,205
Horsepower Losses	na	$6,938	$6,938	$6,938	$6,938	$6,938
Operation and Maintenance	na	$5,000	$5,000	$5,000	$5,000	$5,000
Annual Total Cost/seal	**$185,000**	**$11,143**	**$11,143**	**$11,143**	**$11,143**	**$11,143**
Cumulative Total Cost	**$185,000**	**$196,143**	**$207,285**	**$218,428**	**$229,571**	**$240,714**

Cost of Ownership
Gas Seals vs.Oil Seals



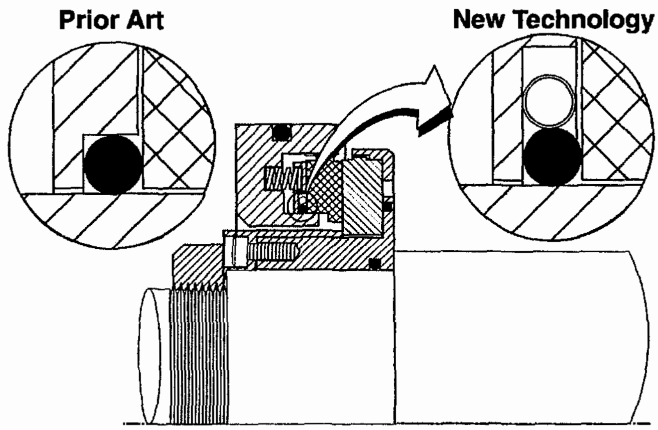


Figure 13-71. Semi-dynamic secondary seal design to reduce drag.

(text continued from page 587)

Solutions to date have included the addition of lubricants and fillers to the O-ring material in order to reduce secondary seal drag and reducing the radial squeeze on the O-ring from 10% to 5% and even 0% to minimize hang up.

The standard or conventional O-ring cavity, however, still results in variations in secondary seal drag because of its confined dimensions and the higher thermal expansion rates of various elastomers than the metal parts surrounding them. To overcome this problem, some manufacturers have chosen to use a design that confines the O-ring only in an axial direction and allows for thermal and chemical expansion without increasing the load of the secondary seal on the shaft sleeve. One means of achieving this goal is by putting a garter spring around the outside diameter of the secondary seal to hold it in constant contact with the secondary sealing surface.

Rotor Disintegration. Silicon carbide and tungsten carbide rotors are potential sources of stress risers and fatigue-type failures if they are not properly mounted and driven with respect to the shaft sleeves. Past methods of installing a horizontal drive pin in silicon carbide and tungsten carbide rotors as shown in Figure 13-72 are being abandoned in favor of O.D. drive mechanisms and centering methods that greatly reduce the stress risers in the rotating face. Stresses are also reduced by using an O.D. centering strip versus the conventional I.D. centering strip. Figure 13-73 shows this centering and drive mechanism. The result of the centering and drive mechanism is a reduction in stress in the rotating ring and increased reliability.

Elimination of Tolerance Ring Problems. Figure 13-74 shows the typical tolerance ring configuration found underneath shaft sleeves and rotating seal faces of compressor gas seals. These tolerances rings have typically been installed in shallow rectangular grooves. One common mode of failure with this configuration of tolerance

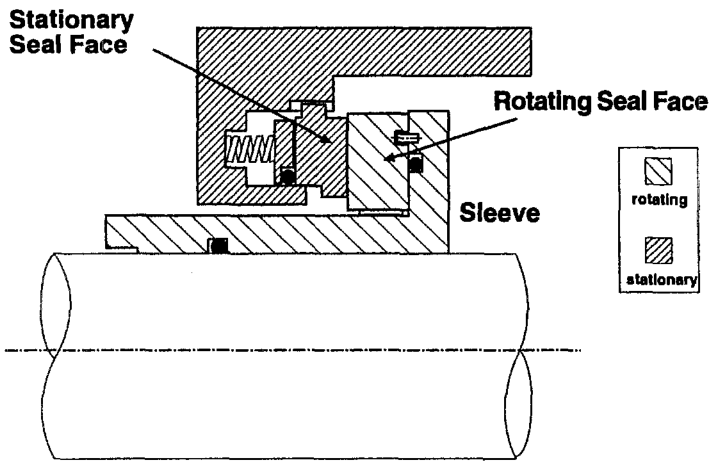


Figure 13-72. Rotor design with high internal stresses.

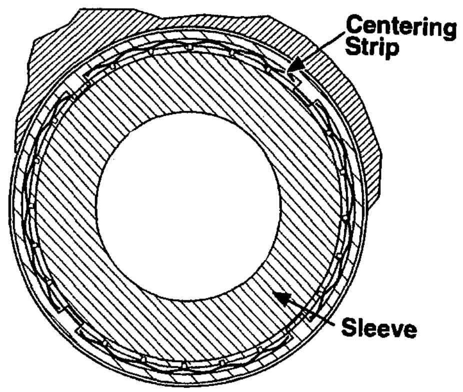


Figure 13-73. Seal rotor design minimizing internal stresses.

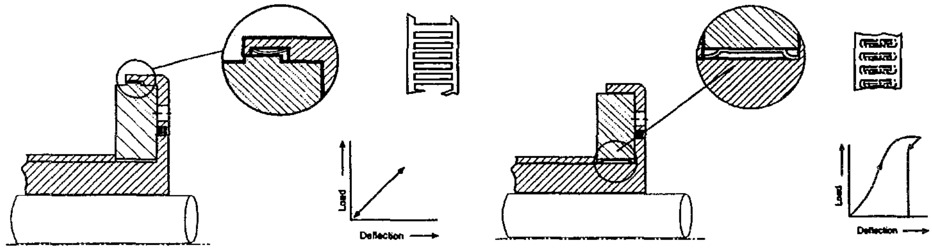


Figure 13-74. Tolerance ring centering at OD versus ID.

rings occurs when the ring slips up underneath the rotating seal face and gets trapped between tungsten carbide or silicon carbide element and the shaft sleeve. This can result in catastrophic failure during startup and operation of the seal due to the higher rate of thermal expansion of the shaft sleeve over the rotating seal face. New designs incorporating the unique centering strip groove and centering strip design have been developed and are in active use today. These centering strips, which contain a slight curvature, are snapped into a dovetail-type groove in the sleeve that centers the silicon carbide rotor from either the I.D. or O.D. This design eliminates the possibility that the centering strip can slide out of the groove during assembly or disassembly of components. When mounted at the O.D. of the rotating seal, the centering strip also provides compressive loads distributed around the circumference of the rotating seal for superior centering of the rotor while subtracting from the centrifugal forces that are common on rotating hard faces.

Silicon Carbide Chipping. Another critical feature of rotating seal faces is the means used to drive the rotating seal face, especially when silicon carbide is used. Silicon carbide in its best configuration is still somewhat fragile and sensitive to chipping. Several drive mechanisms have been developed that provide a soft drive with the silicon carbide rotor rather than jamming it against a hard metal drive pin. Figure 13-75 shows one such drive mechanism, which incorporates multiple coil springs that are positioned evenly around the O.D. of the rotating silicon carbide face. The multiple coil springs provide a distributed load along the periphery of the silicon carbide drive slot rather than one single point contact from a drive pin. This method has been shown to be superior in eliminating chipping and fracture of silicon carbide components. Other methods, including elastomers and more flexible keys, have also been found suitable.

Table 13-13 summarizes compressor gas seal reliability issues and recommended solutions as a quick reference.

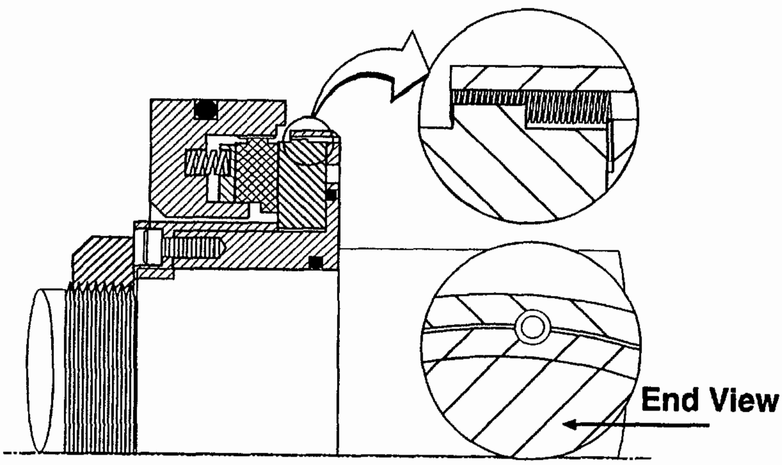


Figure 13-75. Soft drive of rotor.

Table 13-13
Rotating Equipment Reliability Problems

COMPRESSOR	Reliability Issue	Recommended Solution
	Safety-high pressure oil leaks are a potential source for fire	Use gas barrier seal technology.
	Rotor disintegration	Eliminate stress riser from drive pins and tolerance rings.
	Secondary seal hang-up	Use low drag secondary seals that acommodate O-ring tolerances and swell.
	Tolerance ring assembly problem	Use confined cavity.
	Silicon carbide chipping	Use soft drive distributed loading.

Conclusions

It has been demonstrated in operating applications that a thorough review of the service conditions, plant safety, and reliability requirements is essential prior to consideration of gas seals.

Proper selection of seal arrangements and control systems incorporating plant operating philosophies is crucial to a successful retrofit and high availability of that piece of equipment. Whereas, with oil seals, little or no consideration was given to the process gas sealed, with dry seals the process gas itself is used in the seal as a buffer. Hence, it is essential to understand the seal dynamics, gas contaminants, and process fluctuations.

Since the late 1970s, an ever increasing number of centrifugal compressors are being sealed with modern non-contacting gas lubricated seals. This seal technology has become the industry standard for refinery, petrochemical, gas processing, pipelines, and offshore applications. Properly engineered, these seals have proven to be highly reliable, saving industry literally millions of dollars in downtime avoidance, increased energy efficiency, maintenance cost savings and associated expenditures. Even including the required support system, compressor gas seals are often less expensive than the traditional liquid seals with their sometimes elaborate oil systems. In many retrofit applications, dry gas seals for process gas compression services have yielded payback periods of less than one year.

Warding Off Equipment Reliability Setbacks: A Postscript

In late 1997, the Maintenance Manager of a world-scale petrochemical plant located away from our shores sent us a distress message. Its content is rather typical of a disturbing trend towards decreased equipment reliability in some locations. His detailed telefax showed the long-term effects of "re-engineering" by blindly downsizing, or even deleting, the reliability assurance and improvement function in a number of modern plants.

This manager explained that his unit designs and installations ranged in vintage from 1960 to about 1980 and that the facility has "operated somewhat successfully" over the past 35 years. He went on to write:

> Nevertheless, we are continuously plagued with problems associated with plant rotating equipment and turbomachinery. Maintenance on rotating equipment is high in terms of downtime and costs. The quality of maintenance on our rotating equipment has deteriorated to the point where we suffered severe losses, including fatalities.
>
> To this end, I am seeking assistance, support, and guidance in how I should proceed with this. I am hoping through this fax we can at least start the communication process and build on it as we go along. Specifically, we may need to have an audit done to find out where we are with respect to maintenance of rotating equipment, some recommendations on what needs to be done, and the process to get there. We may also need some followup support from you or your organization.

This Maintenance Manager deserved considerable credit for recognizing the seriousness of the situation and for asking for help. Encouraged by an immediate response, he sent us additional details on plant history and changes in ownership that had taken place since 1980. He gave a recap of his company's Vision Statement (chances are *yours* reads about the same):

> The company places great emphasis on the training of its human resource personnel, plant reliability, safety, technology, and on maintaining a healthy environment.

His next statements were transmitted in bold, underlined letters. He candidly outlined what went wrong and we considered his insight and honesty refreshing and commendable:

> During the past eight years or so, a number of experienced people have left the organization, some through natural attrition, some through promotions of various sorts, some have migrated to other jobs. We have therefore been left with a competency gap which we were unable to fill due to various reasons and for which we must take full responsibility. In addition, our plants are fairly old; reliability problems, maintenance costs and the duration of maintenance downtime events are on the rise.
>
> A key area in which expertise and competence were lost is that of rotating equipment. In that field, we are left with very few experienced people, a few helpers and some trainees. Our task is to quickly accelerate the level of competence and expertise in rotating equipment maintenance at the "hands-on" level, supervisory level, and at the management level.

Next, this perceptive Maintenance Manager described to us a suggested approach which we will share with those of our readers who, before long, are almost certain to be confronted with similar issues of serious concern.

Suggested Approach Described by the Client

The Maintenance Manager suggested the following task sequence:

1. Carry out an audit of our maintenance systems, procedures, and personnel specific to rotating equipment maintenance.
2. Set up a team of professionals and experts to work with us at plant site for an initial period of six months.
3. Provide consulting support and followup service for some period thereafter.

He continued by first highlighting the overall purpose, and then giving a more detailed description of the team composition he envisioned:

Overall Purpose

The manager cited the overall purpose—To complement the existing maintenance team at plant site by injecting professionalism, quality, experience, and expertise in the following disciplines:

- Rotating equipment maintenance
- Electrical and instrument maintenance
- Maintenance supervision
- Maintenance planning

In the manager's words:

> The maintenance organization, especially the people, need to become proactive in their way of work. We need a paradigm shift in our culture, attitude, and our approach in the way we do maintenance today. To this end, we are seeking professionals to assist us in bridging that gap in the areas that we need most. These are the craft levels, in planning and in maintenance supervision, and these will be assigned in the first instance, to the plant maintenance organizations.

He continued by specifying the desired team composition as follows:

Rotating Equipment Expertise

- 10 to 15 years of hands-on experience in maintaining and overhauling rotating equipment and turbomachinery, including general-purpose steam turbines, high speed/high horsepower turbomachines, pumps, compressors, blowers, gearboxes, lubrication systems, bearing maintenance, mechanical seals, alignment, leak repairs.
- Mechanical engineering technician's diploma is a minimum requirement.
- Systematic approach to troubleshooting and diagnostics of problems associated with plant rotating equipment through quality procedures and checklists.

Electrical and Instrument Maintenance Expertise

- 10 to 15 years hands-on experience in the repair and maintenance of instruments and electrical equipment, including DCS and PLC troubleshooting and maintenance, high voltage electrical systems, electric motor and switchgear repair and maintenance, pneumatic and hydraulic systems, instrument calibrations, uninterrupted power supplies, analyzer maintenance, emergency shutdown systems, transmitter maintenance, etc.
- Minimum of an electrical and electronic engineering technician's diploma
- Systematic approach to diagnostic and troubleshooting of E&I systems using quality procedures and checklists.

Planning Expertise

- 10 to 15 years of planning experience in the petrochemical industry with specific expertise in front-line maintenance planning.
- B.S. degree in mechanical or industrial engineering or equivalent indepth experience and knowledge in contracts and contractor administration.
- Planners must be able to review work requests for completeness and priority, estimate the resource requirements, determine the optimum method, procedures, etc., for executing the work, prepare material requisitions and reservations, tag and secure availability of parts, prepare time schedules, perform critical path planning, and manage work order backlog, etc.

Supervisory Experience

- 10–15 years experience in maintenance field supervision in each of the disciplines described above
- B.S. degree in mechanical or electrical engineering or equivalent
- Must be results-oriented, adhering to objectives and targets
- Must be able to facilitate employee performance improvements
- Must possess delegating and supportive skills
- Must be able to communicate effectively and must be a team player
- Must be flexible and be able to adjust to changing priorities even in crisis conditions.

General Specifications

- All personnel must be between 35 and 45 years of age
- All personnel must be computer literate and be familiar with Lotus Suite and Microsoft Project
- Must be able to communicate professionally to all levels of the organization
- Must be proactive in their approach to work
- Must be able to influence, train, transfer their knowledge to their peers, especially in areas of quality, safety, and sound working practices
- Must be dynamic, energetic and be able to remain professional even in crisis conditions.

- Must be able to adapt to the local environment, local organizational and societal culture in a very short time period
- Must be meticulous and look at details to ensure that the job gets done right the first time
- Must have good housekeeping practices
- Must be quality conscious and be able to influence peers along the lines of quality.

We had no disagreement with the Maintenance Manager's request and perception of needs. In working with this client, we realized we would have much to say about the proper interface relationship that his operating, mechanical-technical, and management personnel had to develop and nurture. We knew there would have to be consistent backing from the top and accountability where permissiveness was perhaps the rule. And at least one other important area, age, needed to be dealt with from the start.

We simply couldn't help but notice the age qualifications sought in this instance. The expert retiree was ruled out, and we got to wonder about which *qualified* 35 to 45 year old resident of a North American or Western European country would be willing to take on the cultural, technical, and procedural challenges that were either laid out or implied in this communication. That brings us to another important point.

Where Maintenance and Reliability Professionals Come From

Many managers are unaware of the fact that "best-in-class" companies routinely "design-out" maintenance at the inception of a project. That, clearly, is the first key to highest equipment reliability and plant profitability. It requires the up-front investment in time and resources that has been proven time and again to pay tremendous dividends.

Whenever maintenance events occur as time goes on, the real industry leaders see every one of these events as an opportunity to upgrade. Indeed, upgrading is the second key, and upgrading is the job of highly trained, well-organized, knowledgeable reliability professionals. World-class performance is impossible to achieve without qualified professionals, and the notion that these professionals could always be hired on a moment's notice is rather unrealistic. Similarly, the idea that contractors can fill the gap surely lacks merit and forethought. Where would the contractor's young engineers have received their training? Look again at the list of qualifications given above!

And, while we surely decided to do our best to cooperate with this client, here's our advice to the petrochemical plant manager who understands the value of a thoroughly well-trained maintenance-reliability work force: Develop one and hold on to them. Start by compiling a role statement, then progress to mapping out a training plan. Interview a number of interested candidates; select the right ones. Give them periodic performance feedback; defend their goals and contributions as necessary and appropriate. Don't ever allow the trained reliability professional to become just a pair of hands or a person whose entire time is spent fighting repair deadlines rather than being immersed in proactive failure prevention. Groom this reliability professional's abilities, judgment, and motivation; do it by encouraging access to his or her peers. Ask this person to use analytical skills to the utmost, to read, write, to communicate. Rest assured that all parties will benefit from the experience if you carefully and consistently implement this "grow-your-own" formula.

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Appendix A

Useful and Interesting Statistics

Maintenance Cost Breakdown by Work Order

Table A-4
Petrochemical Plant
Maintenance Cost Breakdown by Work Order (1996)

	Percentage of Direct Cost	
	Average	World Class
Predictive	1.5	10
Preventive	2.5	20
Improvement	8	10
Emergency (Overtime)	13	7
Repair	75	53

This table shows that more preventive maintenance will drive down emergency work and repairs.

Table A-2
Machinery Availability/Reliability

Basis: 8760 hrs/yr.

Industry Covered: HPI
Year: 1996

Equipment	Availability		Reliability		Downtime						Forced Outage					
					IMR&O hrs./yr.		Forced DT hrs./yr.		Proactive Mtc. hrs. per 5 yrs.		MTBF yrs.		lambda* #		MTTR hrs.	
	Best	Average	Best	Average	Best	Average	Best	Average	Best	Average	Best	Average	Best	Average	Best	Average
Compressors																
Axial	0.997	0.991	1.000	0.998	26.2	76.4	2.2	16.4	120	300	22.8	9.1	5.0	12.5	50	150
Centrifugal																
- Clean Service	0.999	0.997	1.000	0.998	5.7	24.8	3.3	18.8	12	30	15.0	8.0	7.6	14.3	50	150
- Fouling Service	0.996	0.990	0.999	0.995	36.0	90.6	11.0	40.6	125	250	4.6	3.7	25.0	30.9	50	150
- Gas Transmission**	0.960	0.938	0.999	0.983	347.0	540.2	7.0	140.2	1700	2000	0.1	0.1	800.0	1600.0	1	10
Reciprocating																
- Lubricated	0.995	0.973	0.998	0.978	44.0	237.2	20.0	189.2	120	240	1.5	0.5	76.0	240.0	30	90
- Conv. Non-Lube	0.986	0.913	0.991	0.923	122.9	766.1	74.9	670.1	240	480	1.0	0.3	114.0	340.0	75	225
- Integral Gas Engine	0.963															
- Labyrinth Piston	0.995	0.976	0.998	0.983	44.1	207.2	20.1	147.2	120	300	5.0	2.0	23.0	56.0	100	300
Screw																
- Dry	0.996	0.990	0.999	0.997	35.0	90.0	5.0	30.0	150	300	10.0	5.0	11.4	22.8	50	150
- Liquid Injected	0.992	0.977	0.998	0.988	66.6	199.9	16.6	99.9	250	500	3.0	1.5	38.0	76.0	50	150
Electric Motors																
- Induction < 500 kW	0.998	0.997	0.999	0.998	17.0	28.7	10.0	18.7	35	50	12.0	8.0	9.5	14.2	120	150
- Induction > 500 kW	0.995	0.992	0.998	0.996	42.0	67.4	18.0	37.4	120	150	20.0	12.0	5.7	9.5	360	450
- Synchronous	0.996	0.993	0.998	0.997	38.2	60.0	14.2	30.0	120	150	25.4	15.0	4.5	7.6	360	450
Gas Turbines																
- Industrial	0.994	0.990	0.996	0.994	53.3	89.9	33.3	49.9	100	200	6.0	5.0	19.0	22.8	200	250
- Aircraft Derivatives	0.995	0.990	0.998	0.997	46.6	85.0	16.6	25.0	150	300	6.0	5.0	19.0	22.8	100	125
Gears																
- General Purpose	0.999	0.995	0.999	0.997	13.0	40.0	5.0	30.0	40	50	10.0	5.0	11.4	22.8	50	150
- Special Purpose	0.998	0.997	0.999	0.998	15.9	26.6	7.9	16.6	40	50	25.4	15.0	4.5	7.6	200	250
Mixers	0.999	0.995	0.999	0.997	13.0	40.0	5.0	30.0	40	50	10.0	5.0	11.4	22.8	50	150
Pumps - Centrifugal																
- Process, API	0.998	0.996	0.999	0.998	17.2	30.8	9.2	20.8	40	50	6.5	3.5	17.5	33.0	60	72
- Process, ANSI	0.998	0.996	0.999	0.998	17.4	31.1	9.4	21.1	40	50	6.4	3.4	17.8	33.5	60	72

(table continued on next page)

Table A-2 (Continued)

	Availability		Reliability		Downtime						Forced Outage					
					IMR&O hrs./yr.		Forced DT hrs./yr.		Proactive Mtc. hrs. per 5 yrs.		MTBF yrs.		lambda* #		MTTR hrs.	
	Best	Average	Best	Average	Best	Average	Best	Average	Best	Average	Best	Average	Best	Average	Best	Average
Pumps - Continued																
- Utility Service	0.998	0.996	0.998	0.997	21.3	39.3	13.3	29.3	40	50	6.4	3.4	17.8	33.5	85	100
- Canned Motor	0.995	0.990	0.998	0.996	40.0	87.6	15.0	37.6	125	250	10.0	8.0	11.4	14.3	150	300
- Magnetic Drive	0.996	0.993	0.999	0.995	32.5	64.9	12.5	39.9	100	125	8.0	5.0	14.3	22.8	100	200
Pumps, Pos. Displ.																
- Metering	0.998	0.997	0.999	0.998	13.8	26.0	5.8	16.0	40	50	3.5	1.6	33.0	73.0	20	25
- Large Recip.	0.996	0.994	0.999	0.998	33.5	50.0	8.5	20.0	125	150	10.0	5.0	11.4	22.8	85	100
- Screw	0.998	0.997	0.999	0.998	14.0	25.0	6.0	15.0	40	50	10.0	5.0	11.4	22.8	60	75
Steam Turbines (MD)																
- Condensing	0.992	0.989	0.997	0.994	71.8	99.9	29.8	49.9	210	250	7.0	5.0	16.2	22.8	210	250
- Extraction/Adm.	0.993	0.991	0.998	0.996	59.5	81.3	17.5	31.3	210	250	12.0	8.0	9.5	14.3	210	250
- Back Pressure	0.994	0.991	0.998	0.997	56.0	75.0	14.0	25.0	210	250	15.0	10.0	7.6	11.4	210	250

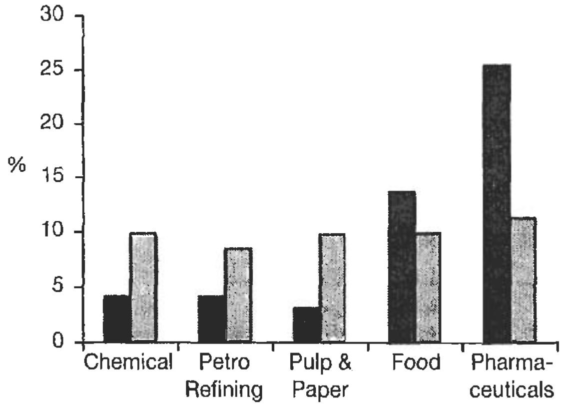
* per MMhrs.

** Includes Driver, i.e. Total Train

Notes: Inputs to this Table:

1. MTBF
2. lambda (MTBF)
3. Proactive Mtc.Hours (Off-line Inspections, Planned Maintenance, Repairs and Overhauls/Turnarounds)

Source: H.P. Bloch and F.K. Geitner's experience, 1998.



Data Source: Based on top ten companies in each industry per Fortune 1994 listings. Capital performance data per Stems Stewart Performance 1000 Report-1994 data.

Figure A-1. Not all process based industries are returning their cost of capital.

Maintenance Cost Is the Largest Cost Lever Available to Improve Profitability

Maintenance costs often range from 11% to 30% of the total operating cost of a process control plant. A multinational chemical company's annual report shows that maintenance cost exceeded 18% in 1996. According to the U.S. Department of Commerce, 40% of the manufacturing revenues are spent on maintenance. These costs can be controlled through changes in current manufacturing practices. Out of necessity, maintenance has undergone several progressive changes over the years. Maintenance began with a mentality of "if it is broken, fix it." Now reliability based maintenance (RBM) allows you to determine what will fail before it actually does.

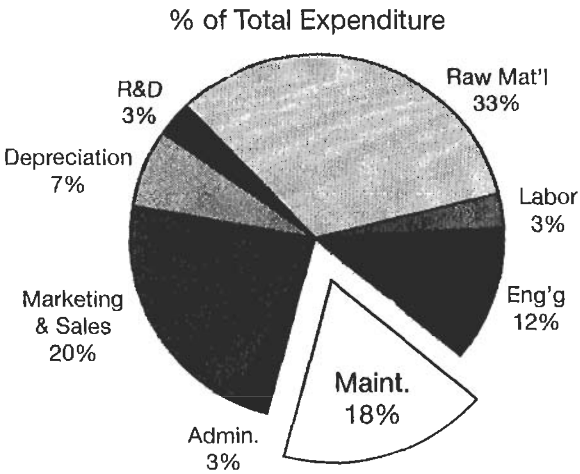


Figure A-2. Maintenance cost of a multi-national chemical company exceeds 18%.[1]

In spite of relatively high maintenance costs, the Society of Maintenance and Reliability (SMRP) Professionals reported in 1996 that more than 56% of U.S. industries do not have a comprehensive maintenance and reliability program. *Modern Power Systems*, July 1994, reported results from an EPRI study, which was independently collaborated by Chevron, that showed the execution of a predictive maintenance program on electric motors could reduce the cost of maintenance by more than 50%.

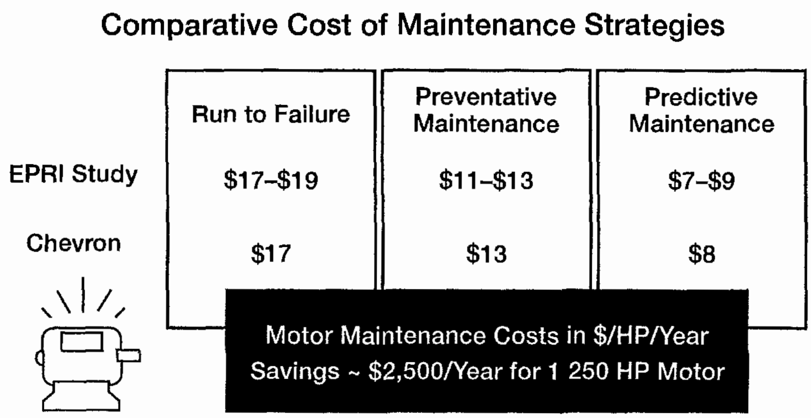


Figure A-3. Predictive maintenance practices for electric motors can reduce maintenance costs by 50%.[1]

Current Maintenance Methods Differ Significantly from Best Practice

SMRP surveyed their membership in the Spring 1996 and reported that only 44% of their membership have integrated reliability and maintenance programs. Surprisingly, 17% of the companies surveyed reported the absence of any reliability or predictive maintenance program.

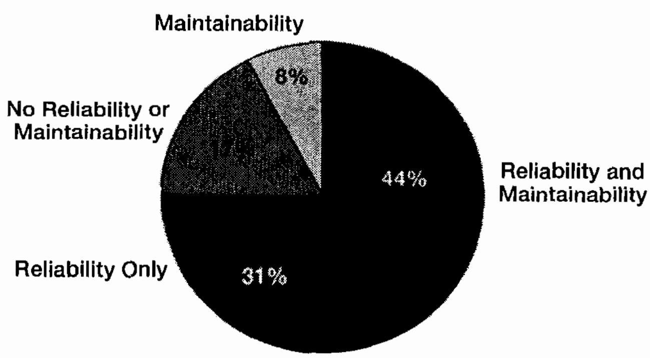


Figure A-4. Do you have a reliability and maintainability process?[1]

Deloitte and Touche in their study, "Maintenance Practices and Technologies," report that management recognizes the importance of maintenance. Diagnostics and predictive maintenance practice rank as high as traditional cost reduction programs.

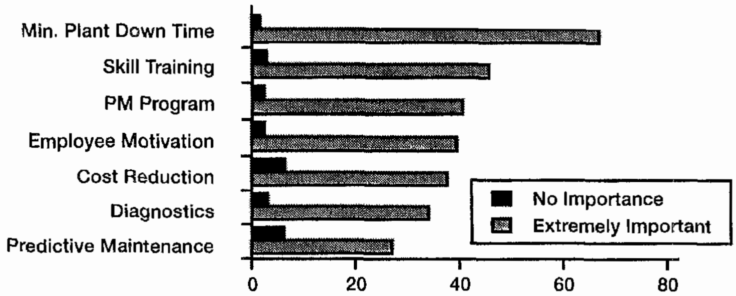


Figure A-5. Maintenance programs recognized to be of importance.[1]

The same Deloitte and Touche study shows that the majority of companies surveyed employed reactive maintenance strategies versus the preferred predictive and proactive capability with a high degree of automation.

C. L. Hays, in his paper entitled, "Plant Maintenance and Diagnostics, Current Practice and Future Trends" at the P/PM conference in late 1996 summarized the disadvantage of a strict preventive maintenance policy based on a time-based overall of a mechanical component.

Figure A-6 shows the traditional "bathtub curve" that represents the probability of a failure versus time for a random system. Traditional preventative maintenance practice is based on "guessing" the appropriate time to replace a critical component to avoid the expected downtime from a failure. The premature replacement of a critical component under the preventive maintenance strategy does not always result in improved uptime because the system may experience an untimely failure of the new component.

Condition-based maintenance strategies (CBM) are key to improving the uptime of a mechanical system. CBM is based on the ability to proactively identify the root cause of failures, eliminate the source of failures, and to provide for predictive maintenance methods. These methods provide early detection of impending failures and thus provide for maximum system life without prematurely introducing an overall cycle as is shown in Figure A-6.

Plant-wide Asset Management Provides Platform for Integrating Assets

The second Offshore Reliability Data Survey (OREDA) of ten large offshore petroleum producers in the North Sea provides an interesting view of the failure rates of major plant assets used for offshore oil production.

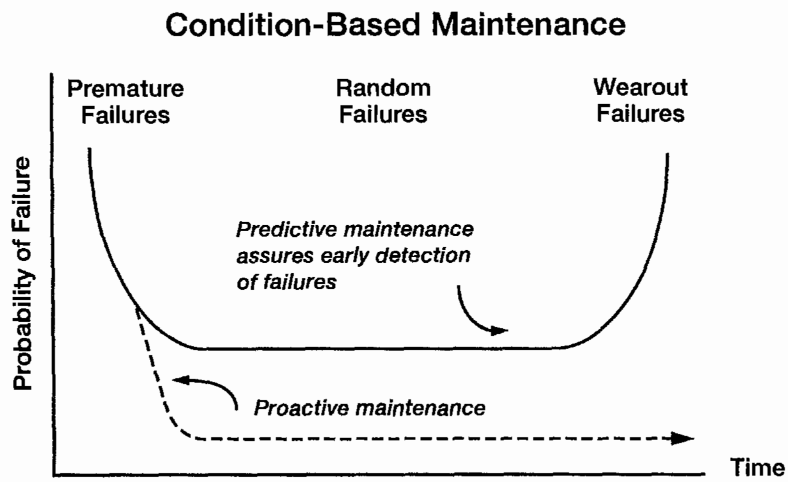


Figure A-6. Proactive maintenance reduces the probability of failure and extends the MTBR.

Figure A-7 is based on the failure rate data provided in the OREDA report and confirms the premise that rotating equipment affords the most significant opportunity for increased reliability and reduction of maintenance costs.

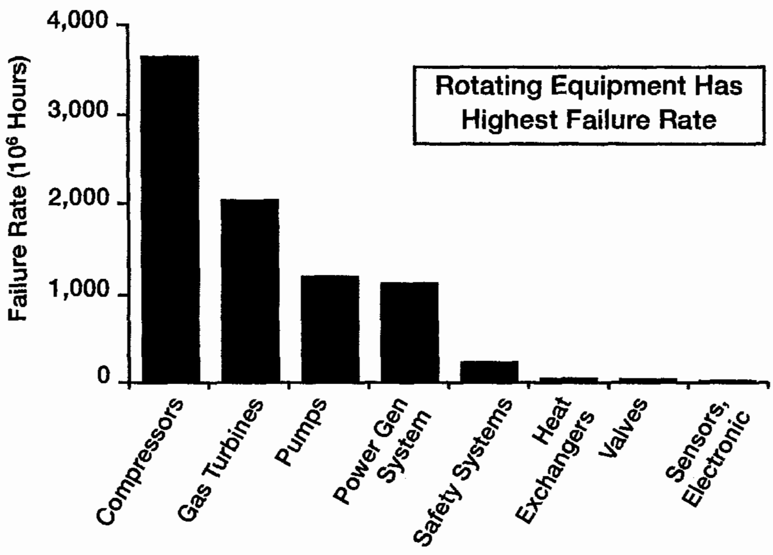


Figure A-7. OREDA failure data.

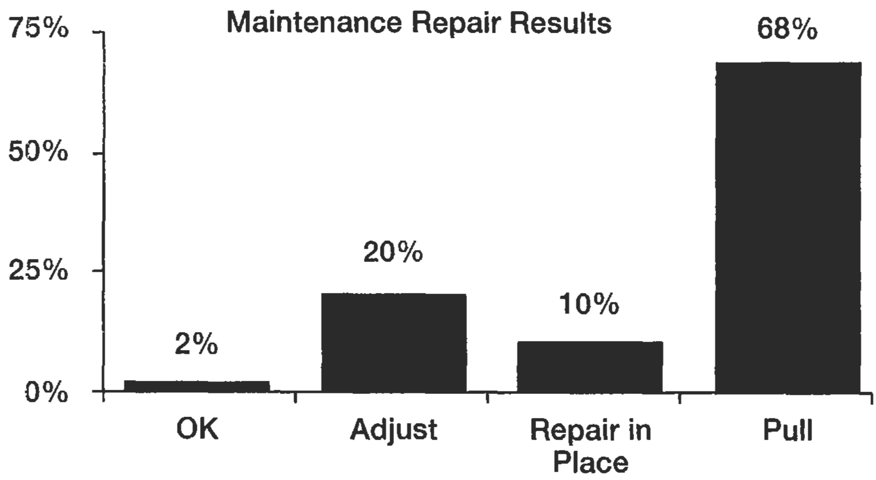


Figure A-8. Rotating equipment maintenance costs can be reduced by condition monitoring.[1]

The Innovation Management Group shows that the implementation of condition monitoring strategies for rotating equipment can reduce maintenance costs by 32% as a result of the opportunity to detect impending failures provided for in situ maintenance versus the traditional pull and replace method used.

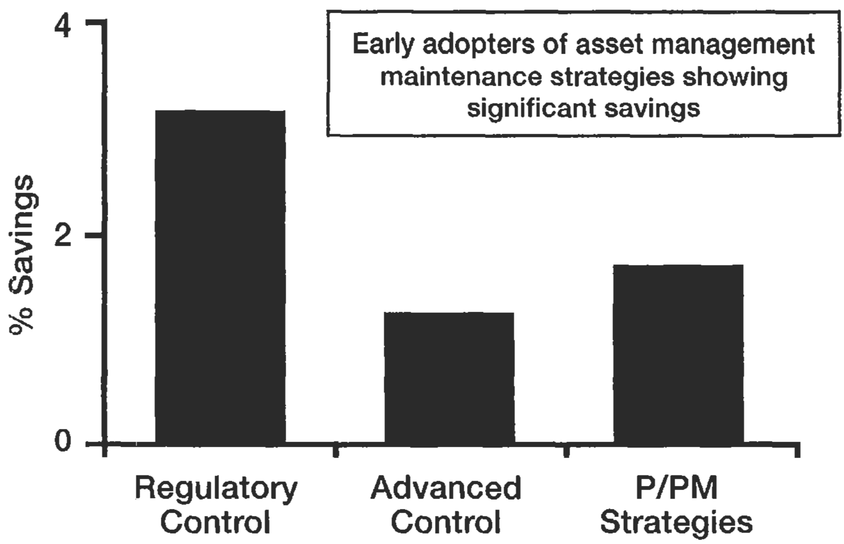


Figure A-9. Early adopters measuring savings from asset management.[1]

The December 1995 *P/PM Journal* chronicles a study of varied pump maintenance practices and the value of using predictive maintenance strategies versus run to fail. An overall savings of 40 times the initial purchase price of the pump is reported.

Plantwide asset management provides the platform for integrating emerging maintenance technologies built on the foundation of field device diagnostics. Adoption of open communication networks and planning with equipment suppliers will permit realization of the vision of a plantwide asset management system that will reduce maintenance costs by up to 70%.

Early adopters of asset management strategies are already reporting benefits.

One early adopter of the "islands of automation" approach has measured savings of 1.75% versus 3.2% for regulatory control with less than 20% of the assets being monitored.

References

1. Boynton, B. and Lenz, G., "Plant Asset Management: An Integrated Maintenance Vision," presented at 6th International Process Plant Reliability Conference, Houston, Texas, October 1997.

Appendix B

Common Sense Reliability Models
A Heuristic Approach

Contributed by:

Paul Barringer, P.E.

Barringer & Associates, Inc.
Manufacturing, Engineering, and Reliability Consultants
P.O. Box 3985
Humble, TX 77347-3985
Phone: 281-852-6810 FAX: 281-852-3749
hpaul@barringer1.com http://www.barringer1.com

Reliability *measures the capacity of equipment or processes to operate* without failure for a specified interval when put into service and used correctly.

Reliability Definitions

- **Reliability:**
  - **The probability that an item can perform its intended function for a specified interval under stated conditions.**

Reliability Is	Reliability Is Not
Probability of failure-free interval	Certainty of no failures
Performing the intended function	Performing any possible functions
Working for specified time intervals	Functioning forever
Working under stated conditions	Working under all possible conditions

- **Reliability is not the same as availability**
  - **Availability is uptime/(uptime + downtime) that tells about use of available time**

Primary Measures Of Reliability

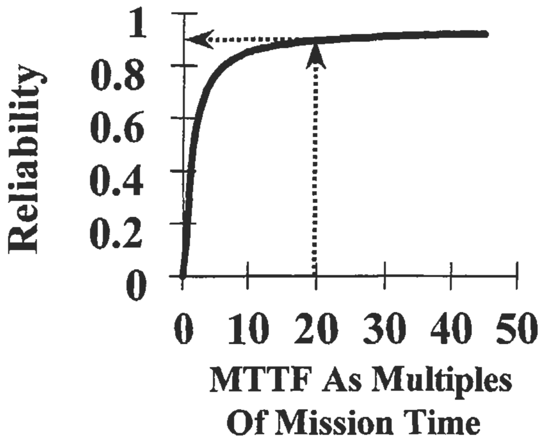
- **Mean time between failure is a primary yardstick for measuring reliability**
  - If MTTF is large compared to the mission-->reliable
  - If MTTF is short compared to the mission-->unreliable!!
  - Do you know your MTTF for rotating equipment?
  - Do you know your MTTF for non-rotating equipment?
  - Which cost you the most money for failures?
- **The business measure for reliability is the cost of UNreliability (lost gross margin + maintenance $'s)**
  - How much UNreliability can you afford?
  - How much UNreliability can you correct?
  - Where do you start the corrections and how much can you afford?
  - What's your Pareto list of the top 10 ten cost items?

Reliability Problems = Business Problems

Reliability Sensitivity to MTTF

High Reliability Requires Long MTTF

$$R(t) = e^{-(t/MTTF)} = e^{-(\lambda t)}$$



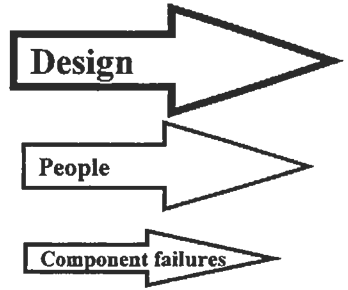
- **The graph shows MTTF as multiples of the mission time**
- **Long MTTFs, compared to the mission time, give high reliability**
- **The results:**

t*MTTF	Reliability
0.5	0.135
1.0	0.368
5.0	0.819
10.0	0.905
15.0	0.936
20.0	0.951
30.0	0.967
40.0	0.975

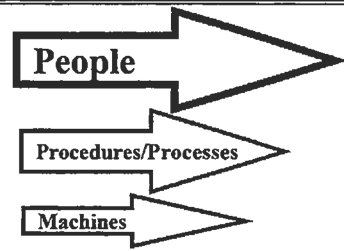
!

Failures: Roots Of Reliability Problems

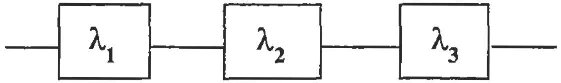
Early Plant Life	Frequency %
• Design Error	35
• Fabrication Error	1
• Random Component Failure	18
• Operator Error	12
• Procedure Error & Unknowns	10
• Maintenance Error	12
• Unknown	12
	100



Mature Plants	
• People	38
• Procedures + Processes	34
• Equipment	28
	100



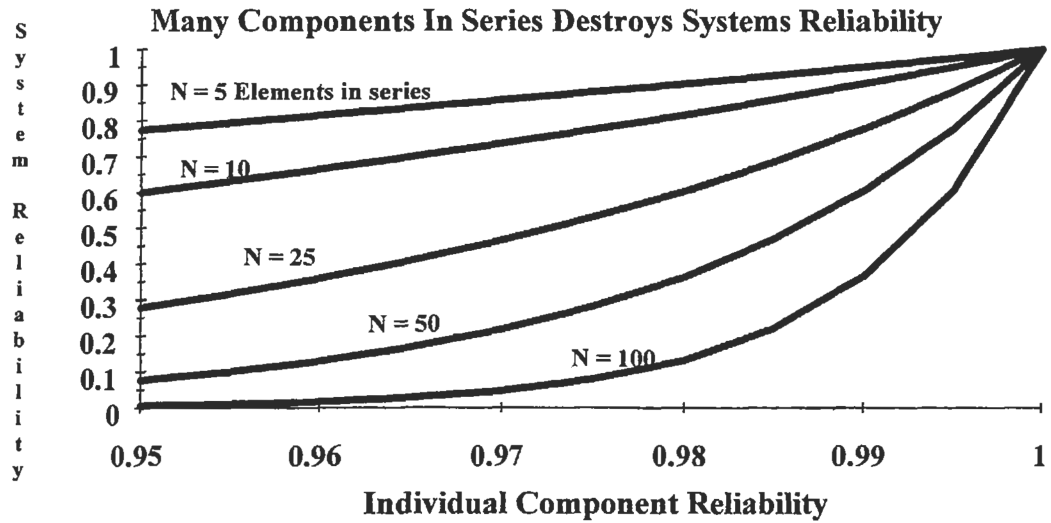
Basic Serial Reliability Models



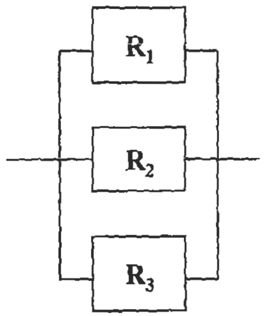
$R_1 = e^{(-\lambda_1 t)}$ $\quad R_2 = e^{(-\lambda_2 t)}$ $\quad R_3 = e^{(-\lambda_3 t)}$

$$\lambda_T = \Sigma\lambda_i = \lambda_1 + \lambda_2 + \lambda_3$$

$$\boxed{R = \prod_{i=1}^{i=n} R_i = R_1 * R_2 * R_3 = e^{(-\lambda_T t)}}$$



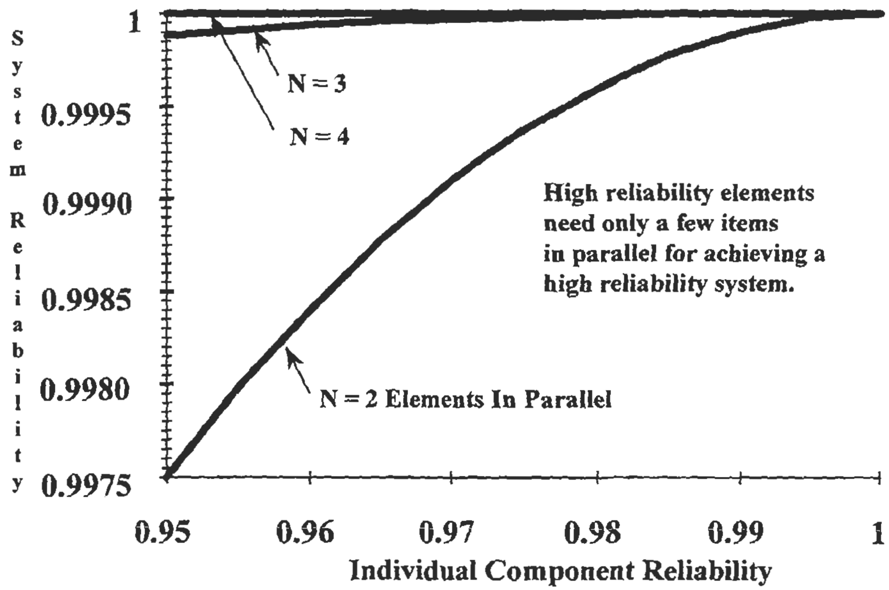
Basic Parallel Reliability Models

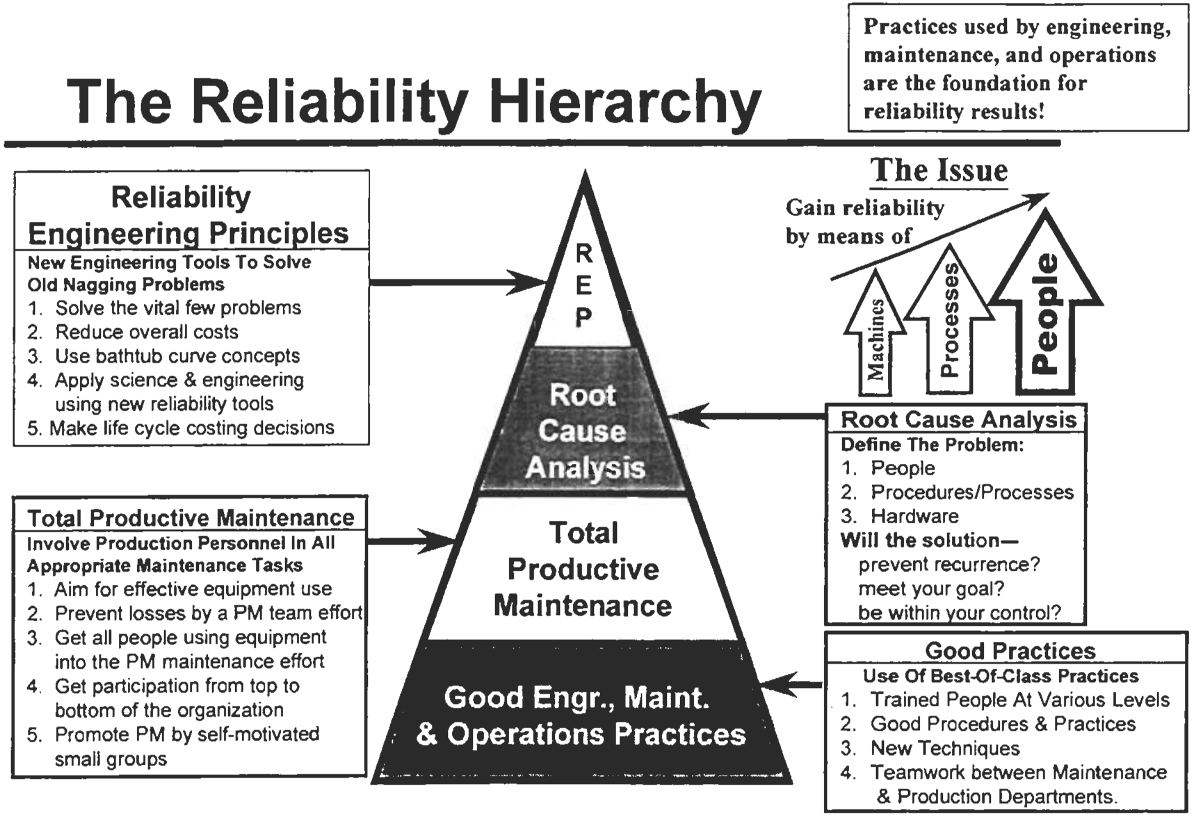


Each element must be able to carry the full load

$$R_{overall} = 1 - (1 - R_1)*(1 - R_2)*(1 - R_3)*(....)$$

Components In Parallel Improves Systems Reliability



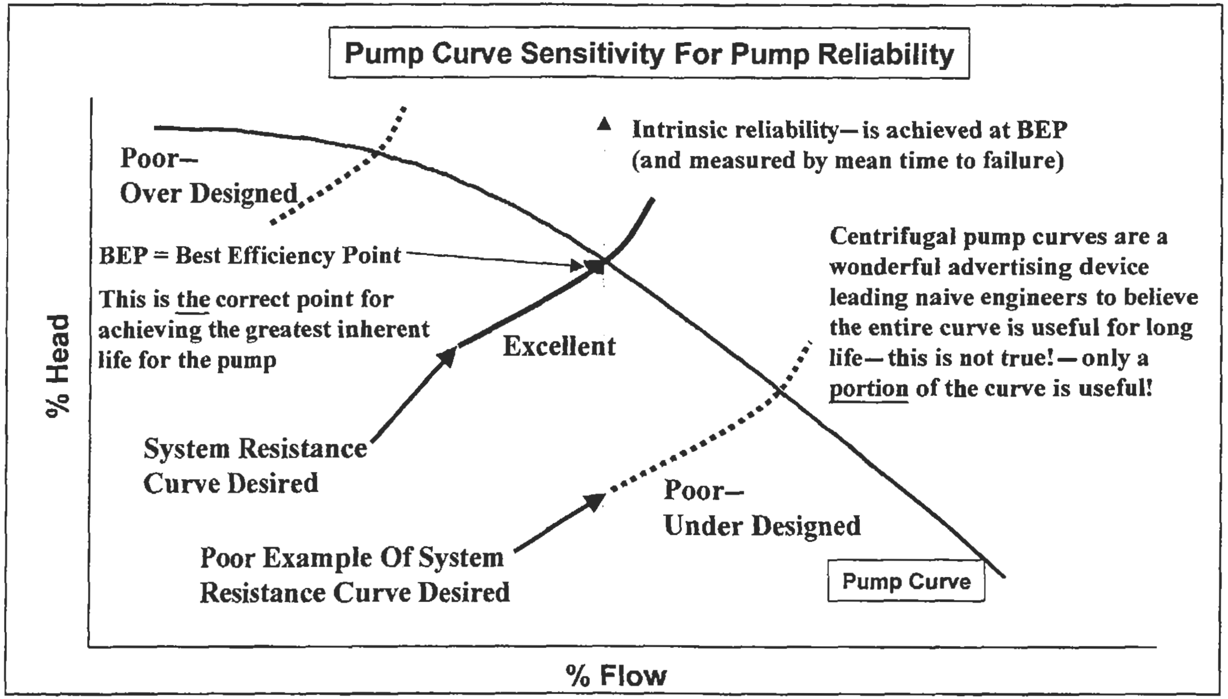


The Reliability Hierarchy
Practices used by engineering, maintenance, and operations are the foundation for reliability results!
Reliability Engineering Principles
New Engineering Tools To Solve Old Nagging Problems
1. Solve the vital few problems
2. Reduce overall costs
3. Use bathtub curve concepts
4. Apply science & engineering using new reliability tools
5. Make life cycle costing decisions
REP
Root Cause Analysis
Total Productive Maintenance
Good Engr., Maint. & Operations Practices
The Issue
Gain reliability by means of
Machines
Processes
People
Root Cause Analysis
Define The Problem:
1. People
2. Procedures/Processes
3. Hardware
Will the solution—
prevent recurrence?
meet your goal?
be within your control?
Total Productive Maintenance
Involve Production Personnel In All Appropriate Maintenance Tasks
1. Aim for effective equipment use
2. Prevent losses by a PM team effort
3. Get all people using equipment into the PM maintenance effort
4. Get participation from top to bottom of the organization
5. Promote PM by self-motivated small groups
Good Practices
Use Of Best-Of-Class Practices
1. Trained People At Various Levels
2. Good Procedures & Practices
3. New Techniques
4. Teamwork between Maintenance & Production Departments.

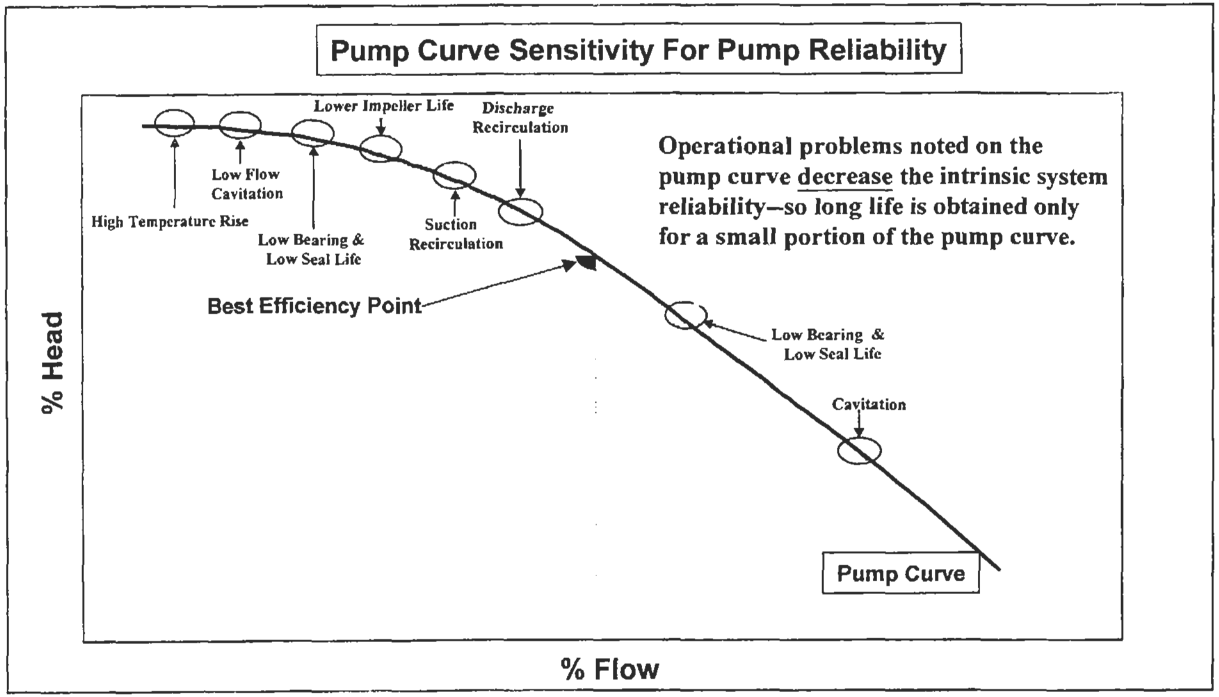
Survey Data On MTTF vs Practices

- Survey sheets were sent to many engineers asking for their opinions on how life of pump components was effected by installation/operation/maintenance practices– opinions are used since hard facts are not available
- Data returned by experts was consolidated, and (generally) median results are reported below
- The characteristics of best, good, and inferior practices were identified and losses in component life are listed
- In general, results from the survey seem realistic although inferior trend data may be too pessimistic
- The data follows trends reported in Bloch & Geitner's *Machinery Reliability Assessment* book
- Simple serial models are used to assimilate the data

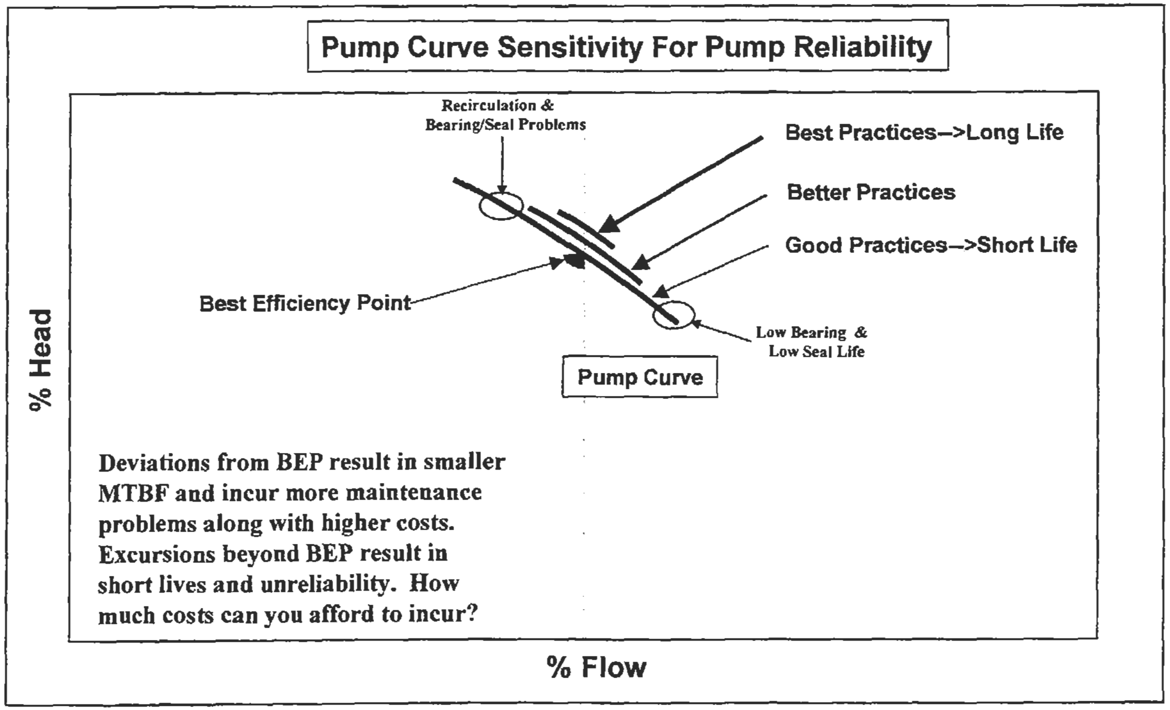
Pump Curve Characteristics



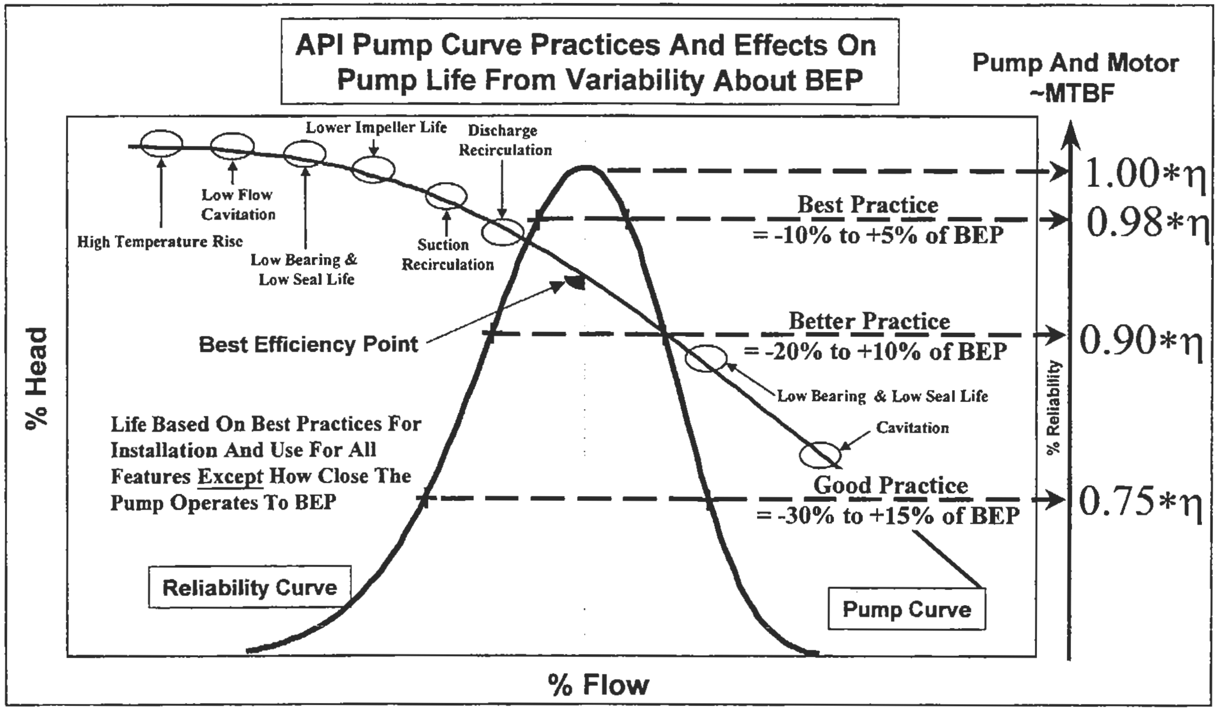
Problems Causing Short Pump Life



Usable Portion Of Pump Curves



Pump Curve Practices–A Model



Loss Of Life From BEP Practice

Inherent Component Reliability	100 hp pump: 8" discharge * 10" suction * 14" case			Best Practices For Installation & Use	Effects Of Practices on Component Life	Dual Pump	Good Practice Replacements		
	beta	eta		Multiplier	beta	eta			
API	shape factor (no dimensions)	location factor (hrs)	MTTF (yrs) = $\eta*\Gamma(1+ 1/\beta)$	Life Multiplier * eta From Practices	shape factor (no dimensions)	location factor (hrs)	MTTF (yrs) = $\eta*\Gamma(1+ 1/\beta)$		
Impeller	2.5	400,000	40.51	0.9726	2.50	389,045	39.40		
Housing	1.3	400,000	42.17	0.8547	1.30	341,882	36.05	MTBF	
Pump Bearings	1.3	400,000	42.17	0.9712	1.30	388,476	40.96	For All	
Seals	1.4	400,000	41.62	0.9677	1.40	387,090	40.27	**Mech.–>**	**6.25**
Shafts	1.2	400,000	42.95	0.9712	1.20	388,476	41.71	Items	
Coupling	2	300,000	30.35	0.9801	2.00	294,030	29.75		
Motor Bearings	1.3	150,000	15.81	1.0000	1.30	150,000	15.81	MTBF	
Motor Windings	1	150,000	17.12	1.0000	1.00	150,000	17.12	For All	
Motor Rotor	1	300,000	34.25	1.0000	1.00	300,000	34.25	**Elect.–>**	**5.50**
Motor Starter	1.2	300,000	32.21	1.0000	1.20	300,000	32.21	Items	
			2.99	=~Mean time between system failures=			**2.93**		
			26,209 hours		Δ = **Loss**	or=	**25,627**	hours	
					2%	loss=	**582**	hours	

Best Practices For Installation And Use Including Best BEP Practices Achieves 98% of Inherent System Life

Best BEP Practices = -10% to +5% of BEP

Loss Of Life From BEP Practice

Inherent Component Reliability	100 hp pump: 8" discharge * 10" suction * 14" case			Best Practices Except For Better BEP Practices	Effects Of Practices on Component Life	Dual Pump	Good Practice Replacements		
	beta	eta		Multiplier	beta	eta			
API	shape factor (no dimensions)	location factor (hrs)	MTTF (yrs) = $\eta*\Gamma(1+ 1/\beta)$	Life Multiplier * eta From Practices	shape factor (no dimensions)	location factor (hrs)	MTTF (yrs) = $\eta*\Gamma(1+ 1/\beta)$		
Impeller	2.5	400,000	40.51	0.8706	2.50	348,250	35.27		
Housing	1.3	400,000	42.17	0.7236	1.30	289,455	30.52	MTBF	
Pump Bearings	1.3	400,000	42.17	0.7785	1.30	311,400	32.83	For All	
Seals	1.4	400,000	41.62	0.8663	1.40	346,500	36.05	**Mech.–>**	**5.39**
Shafts	1.2	400,000	42.95	0.7785	1.20	311,400	33.44	Items	
Coupling	2	300,000	30.35	0.9059	2.00	271,755	27.49		
Motor Bearings	1.3	150,000	15.81	0.9400	1.30	141,000	14.87	MTBF	
Motor Windings	1	150,000	17.12	1.0000	1.00	150,000	17.12	For All	
Motor Rotor	1	300,000	34.25	0.9400	1.00	282,000	32.19	**Elect.–>**	**5.33**
Motor Starter	1.2	300,000	32.21	1.0000	1.20	300,000	32.21	Items	
			2.99	=~Mean time between system failures=			**2.68**		
			26,209	hours	Δ = **Loss**	or=	**23,462**	hours	
					10%	loss=	**2,747**	hours	

Best Practices For Installation And Use Except For Better BEP Practices Achieves 90% of Inherent System Life

Better BEP Practices = -20% to +10% of BEP

Loss Of Life From BEP Practice

Inherent Component Reliability	100 hp pump: 8" discharge * 10" suction * 14" case			Best Practices Except For Good BEP Practices	Effects Of Practices on Component Life	Dual Pump	Good Practice Replacements		
	beta	eta		Multiplier	beta	eta			
API	shape factor (no dimensions)	location factor (hrs)	MTTF (yrs) = $\eta * \Gamma(1+1/\beta)$	Life Multiplier * eta From Practices	shape factor (no dimensions)	location factor (hrs)	MTTF (yrs) = $\eta * \Gamma(1+1/\beta)$		
Impeller	2.5	400,000	40.51	0.6716	2.50	268,650	27.21		
Housing	1.3	400,000	42.17	0.6965	1.30	278,600	29.37	MTBF	
Pump Bearings	1.3	400,000	42.17	0.6435	1.30	257,400	27.14	For All	
Seals	1.4	400,000	41.62	0.5074	1.40	202,950	21.12	**Mech.–>**	4.26
Shafts	1.2	400,000	42.95	0.6435	1.20	257,400	27.64	Items	
Coupling	2	300,000	30.35	0.7549	2.00	226,463	22.91		
Motor Bearings	1.3	150,000	15.81	0.7750	1.30	116,250	12.26	MTBF	
Motor Windings	1	150,000	17.12	0.9650	1.00	144,750	16.52	For All	
Motor Rotor	1	300,000	34.25	0.7750	1.00	232,500	26.54	**Elect.–>**	4.74
Motor Starter	1.2	300,000	32.21	1.0000	1.20	300,000	32.21	Items	
			2.99	≈~Mean time between system failures≈			**2.24**		
			26,209	hours	Δ = **Loss**	or=	**19,656**	hours	
					25%	loss=	**6,552**	hours	

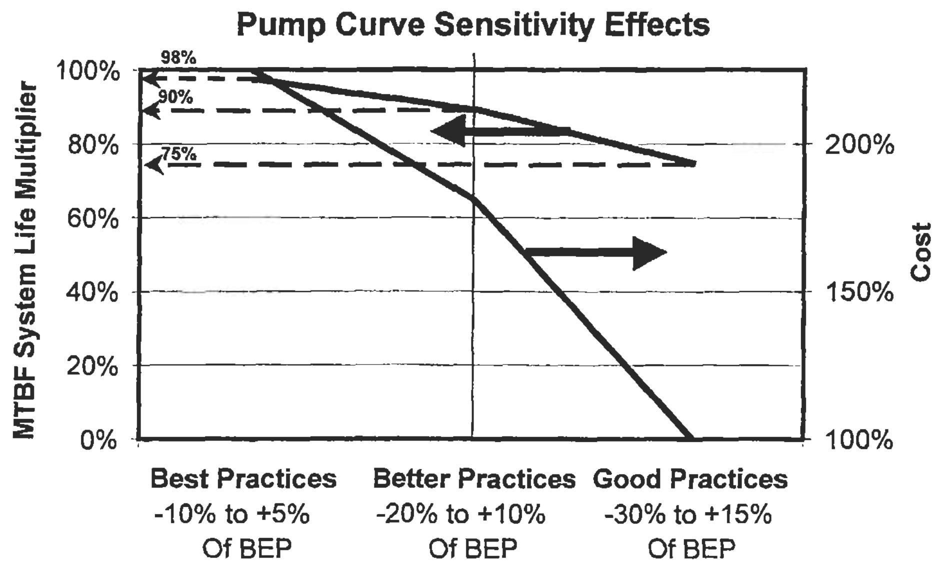
Best Practices For Installation And Use Except For Good BEP Practices Achieves 75% of Inherent System Life

Good BEP Practices = -30% to +15% of BEP

Responsibilities For Pump Curves

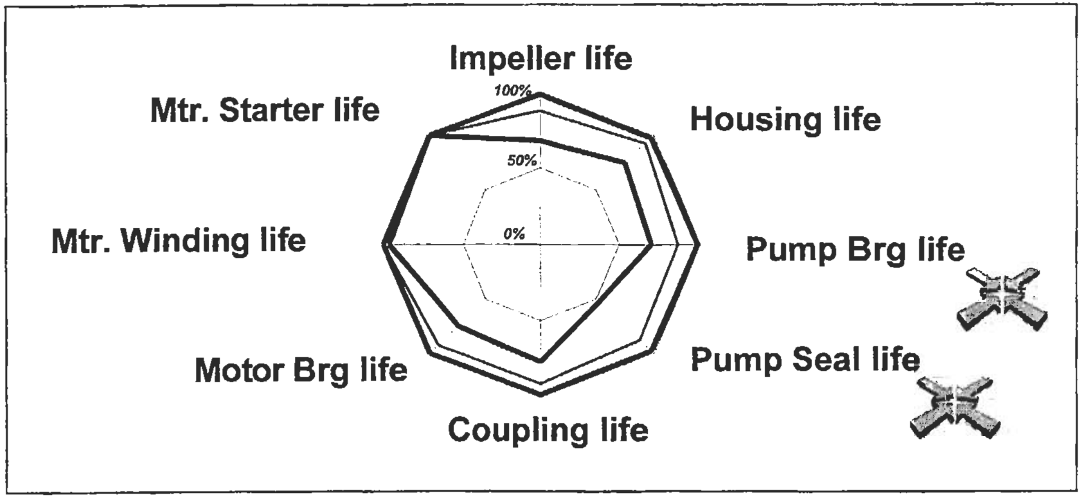
- Engineering is responsible for designing the system to the BEP
- Operations is responsible for keeping the system at the BEP and issuing maintenance work orders for correcting impellers/pumps to re-center the BEP when operating conditions change and BEP is no longer in the target area
- Maintenance is responsible for correcting the problems identified, i.e., removing/replacing the impeller to meet a size specified by engineering
- Pumps operating off the curve are a major reason for short life of bearings and seals

Reliability & Costs Effects



System Life Includes Pump And Motor—All Other Features Use Best Practices Except As Noted

Pump Curve Effects On Component Characteristic Life ~MTTF



—— Best Practices = -10% to + 5% Of BEP
—— Good Practices = -20% to +10% Of BEP
—— Inferior Practices = -30% to +15% Of BEP

Straight Runs Of Suction Piping

- Long runs of pipe are required to smooth flow and create equal mechanical loads on pumps
- Short runs of pipe produce unequal impeller loads and cause undesirable loads on the system
- Unbalanced loads from short pipe runs produce undesirable vibration loads
- The inlet pipe should be 2 sizes larger with special reducers for smooth entry of fluids into the pump to achieve long life
- Short runs of straight pipe result in short MTTF
- Long runs of straight pipe result in long MTTF

Loss Of Life From Intake L/D Practice

Inherent Component Reliability	100 hp pump: 8" discharge * 10" suction * 14" case			Best Practices Except For Better L/D Intake Pipe Practices	Effects Of Practices on Component Life	Dual Pump	Good Practice Replacements		
	beta	eta		Multiplier	beta	eta			
API	shape factor (no dimensions)	location factor (hrs)	MTTF (yrs) = $\eta*\Gamma(1+ 1/\beta)$	Life Multiplier * eta From Practices	shape factor (no dimensions)	location factor (hrs)	MTTF (yrs) = $\eta*\Gamma(1+ 1/\beta)$		
Impeller	2.5	400,000	40.51	0.9240	2.50	369,593	37.43		
Housing	1.3	400,000	42.17	0.8120	1.30	324,788	34.24	MTBF	
Pump Bearings	1.3	400,000	42.17	0.8741	1.30	349,628	36.86	For All	
Seals	1.4	400,000	41.62	0.8798	1.40	351,900	36.61	**Mech.–>**	**5.80**
Shafts	1.2	400,000	42.95	0.8741	1.20	349,628	37.54	Items	
Coupling	2	300,000	30.35	0.9311	2.00	279,329	28.26		
Motor Bearings	1.3	150,000	15.81	0.9250	1.30	138,750	14.63	MTBF	
Motor Windings	1	150,000	17.12	1.0000	1.00	150,000	17.12	For All	
Motor Rotor	1	300,000	34.25	0.9250	1.00	277,500	31.68	**Elect.–>**	**5.28**
Motor Starter	1.2	300,000	32.21	1.0000	1.20	300,000	32.21	Items	
			2.99	=~Mean time between system failures=			**2.76**		
			26,209 hours		Δ = Loss	or =	**24,216**	hours	
					8%	loss=	**1,993**	hours	

Best Practices For Installation And Use Except For Better L/D Practices Achieves 92% of Inherent System Life

Better Intake L/D Practices = 6 to 8

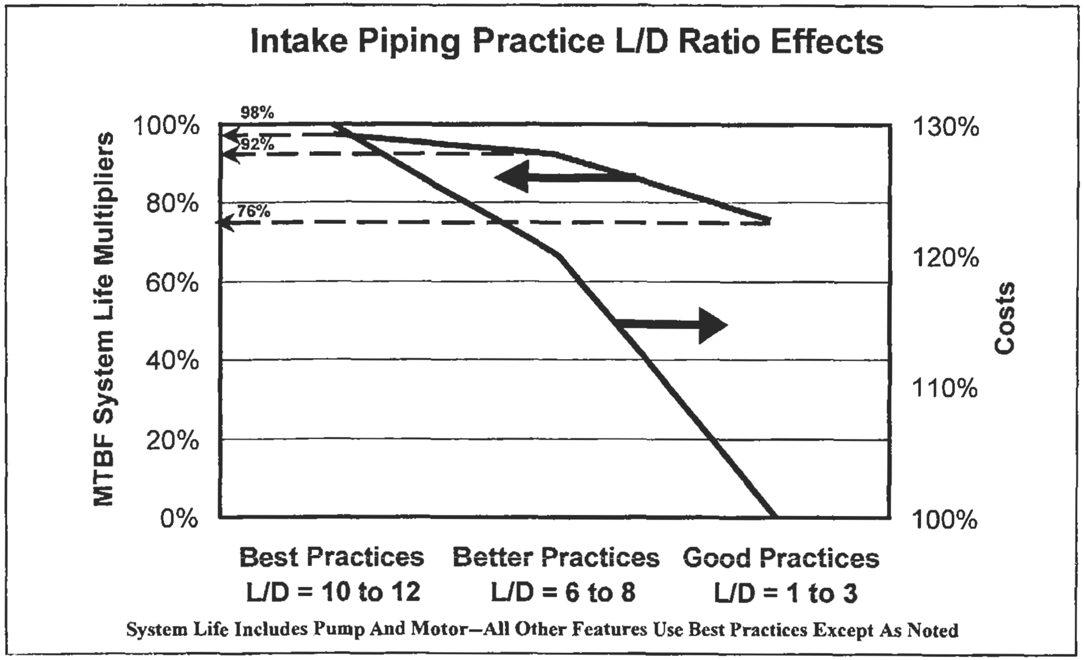
Loss Of Life From Intake L/D Practice

Inherent Component Reliability	100 hp pump: 8" discharge * 10" suction * 14" case			Best Practices Except For Good L/D Intake Pipe Practices	Effects Of Practices on Component Life	Dual Pump	Good Practice Replacements		
	beta	eta		Multiplier	beta	eta			
API	shape factor (no dimensions)	location factor (hrs)	MTTF (yrs) = $\eta*\Gamma(1+1/\beta)$	Life Multiplier * eta From Practices	shape factor (no dimensions)	location factor (hrs)	MTTF (yrs) = $\eta*\Gamma(1+1/\beta)$		
Impeller	2.5	400,000	40.51	0.7295	2.50	291,784	29.55		
Housing	1.3	400,000	42.17	0.6838	1.30	273,506	28.84	MTBF	
Pump Bearings	1.3	400,000	42.17	0.5827	1.30	233,086	24.57	For All	
Seals	1.4	400,000	41.62	0.5865	1.40	234,600	24.41	**Mech.–>**	4.31
Shafts	1.2	400,000	42.95	0.5827	1.20	233,086	25.03	Items	
Coupling	2	300,000	30.35	0.7841	2.00	235,224	23.80		
Motor Bearings	1.3	150,000	15.81	0.8000	1.30	120,000	12.65	MTBF	
Motor Windings	1	150,000	17.12	1.0000	1.00	150,000	17.12	For All	
Motor Rotor	1	300,000	34.25	0.8000	1.00	240,000	27.40	**Elect.–>**	4.88
Motor Starter	1.2	300,000	32.21	1.0000	1.20	300,000	32.21	Items	
			2.99	=~Mean time between system failures=			**2.29**		
			26,209 hours		Δ = **Loss**	or=	**20,040**	hours	
					24%	loss=	**6,169**	hours	

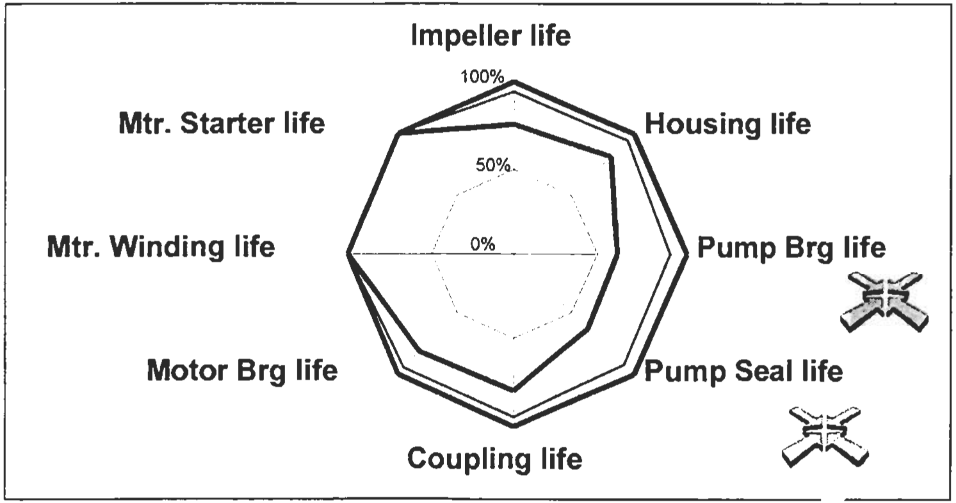
Best Practices For Installation And Use Except For Good L/D Practices Achieves 76% of Inherent System Life

Good Intake L/D Practices = 1 to 3

L/D Suction Straight Run Effects On Reliability & Cost



L/D Suction Straight Run Effects On Component Characteristic Life ~MTTF



Best Practices =	10 to 12 L/D
Good Practices =	6 to 8 L/D
Inferior Practices =	1 to 3 L/D

Rotational Shaft Alignment

- Lack of rotational shaft alignment imposes higher mechanical loads on rotating elements which shortens life
- Poor alignment results in short MTTF
- Good alignment results in long MTTF
- Alignment is a three dimensional requirement
- Jacking screws and shims are requirements for precision alignment–with limits for the number of shims allowed for corrections before a solid riser block is constructed and used
- Standards must be set and maintained at operating conditions

Loss Of Life From Rotational Alignment Practice

Inherent Component Reliability	100 hp pump: 8" discharge * 10" suction * 14" case			Best Practices Except For Better Rotational Alignment Practices	Effects Of Practices on Component Life	Dual Pump	Good Practice Replacements		
	beta	eta		Multiplier	beta	eta			
API	shape factor (no dimensions)	location factor (hrs)	MTTF (yrs) = $\eta * \Gamma(1+ 1/\beta)$	Life Multiplier * eta From Practices	shape factor (no dimensions)	location factor (hrs)	MTTF (yrs) = $\eta * \Gamma(1+ 1/\beta)$		
Impeller	2.5	400,000	40.51	0.9240	2.50	369,593	37.43	MTBF	
Housing	1.3	400,000	42.17	0.8120	1.30	324,788	34.24	For All	
Pump Bearings	1.3	400,000	42.17	0.8498	1.30	339,917	35.84		
Seals	1.4	400,000	41.62	0.8710	1.40	348,381	36.25	**Mech.–>**	5.69
Shafts	1.2	400,000	42.95	0.8498	1.20	339,917	36.50	Items	
Coupling	2	300,000	30.35	0.8910	2.00	267,300	27.04		
Motor Bearings	1.3	150,000	15.81	0.9350	1.30	140,250	14.79	MTBF	
Motor Windings	1	150,000	17.12	1.0000	1.00	150,000	17.12	For All	
Motor Rotor	1	300,000	34.25	0.9350	1.00	280,500	32.02	**Elect.–>**	5.31
Motor Starter	1.2	300,000	32.21	1.0000	1.20	300,000	32.21	Items	
			2.99	=~Mean time between system failures=			**2.75**		
			26,209 hours		Δ = Loss	or=	**24,060**	hours	
					8%	loss=	**2,148**	hours	

Best Practices For Installation And Use Except For Better Rotational Alignment Achieves 92% of Inherent System Life

Better Rotational Alignment Practices = ±0.003 inches or ±0.025 mm

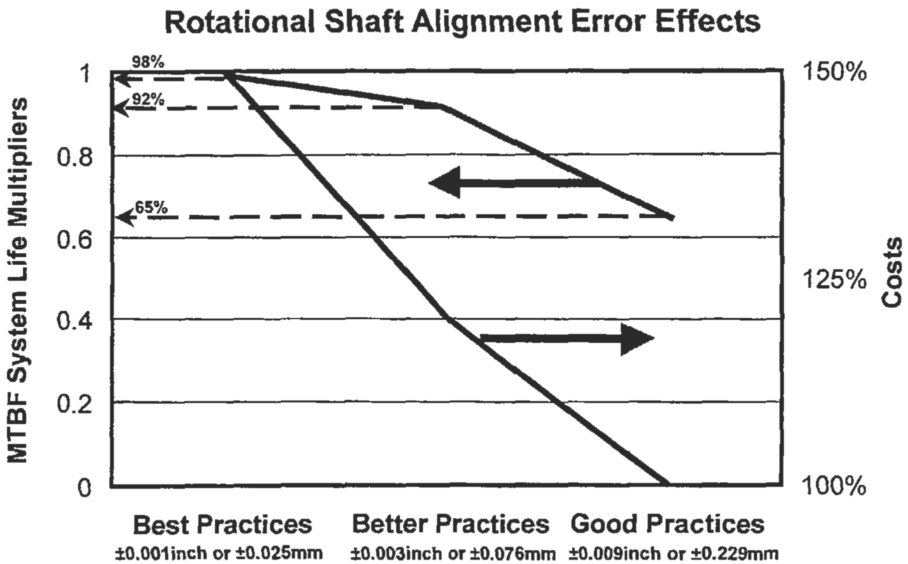
Loss Of Life From Rotational Alignment Practice

Inherent Component Reliability	100 hp pump: 8" discharge * 10" suction * 14" case			Best Practices Except For Good Rotational Alignment Practices	Effects Of Practices on Component Life	Dual Pump	Good Practice Replacements		
	beta	eta		Multiplier	beta	eta			
API	shape factor (no dimensions)	location factor (hrs)	MTTF (yrs) = $\eta\bullet\Gamma(1+1/\beta)$	Life Multiplier * eta From Practices	shape factor (no dimensions)	location factor (hrs)	MTTF (yrs) = $\eta\bullet\Gamma(1+1/\beta)$		
Impeller	2.5	400,000	40.51	0.8754	2.50	350,141	35.46		
Housing	1.3	400,000	42.17	0.7051	1.30	282,053	29.74	MTBF	
Pump Bearings	1.3	400,000	42.17	0.5584	1.30	223,374	23.55	For All	
Seals	1.4	400,000	41.62	0.3871	1.40	154,836	16.11	**Mech.->**	3.86
Shafts	1.2	400,000	42.95	0.5584	1.20	223,374	23.99	Items	
Coupling	2	300,000	30.35	0.6435	2.00	193,050	19.53		
Motor Bearings	1.3	150,000	15.81	0.5500	1.30	82,500	8.70	MTBF	
Motor Windings	1	150,000	17.12	1.0000	1.00	150,000	17.12	For All	
Motor Rotor	1	300,000	34.25	0.5500	1.00	165,000	18.84	**Elect.->**	3.88
Motor Starter	1.2	300,000	32.21	1.0000	1.20	300,000	32.21	Items	
			2.99	=~Mean time between system failures=			**1.94**		
			26,209 hours		Δ = Loss	or=	**16,952**	hours	
					35%	loss=	**9,257**	hours	

Best Practices For Installation And Use Except For Good Rotational Alignment Achieves 65% of Inherent System Life

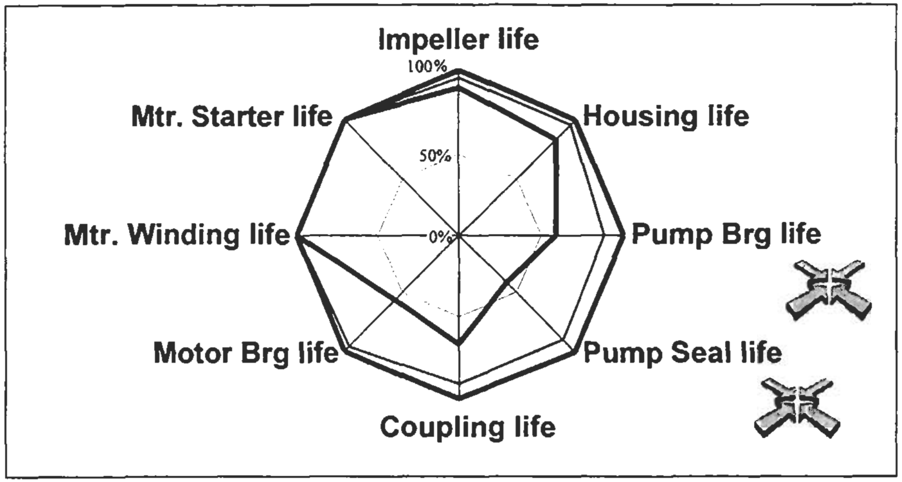
Good Rotational Alignment Practices = ±0.009 inches or ±0.229 mm

Rotational Alignment Effects On Costs & Reliability



System Life Includes Pump And Motor—All Other Features Use Best Practices Except As Noted

Rotational Shaft Alignment Effects On Characteristic Life ~MTTF



—— Best Practices ±0.001" or ±0.025mm
—— Good Practices ±0.003" or ±0.076mm
—— Inferior Practices ±0.009" or ±0.229mm

Mechanical Pipe Alignment

- **Lack of pipe alignment imposes high mechanical loads**
- **Poor pipe alignment results in short MTBF**
- **Good pipe alignment results in long MTBF**
- **Pipe alignment is a three dimensional requirement**
- **No-load pipe alignment requires consideration of temperature changes and the resulting physical changes in pipe dimensions to reduce mechanical loads**
- **Good pipe alignment practices require hand alignment without use of mechanical assist to achieve long life in service**

Loss Of Life From Piping Alignment Practice

Inherent Component Reliability	100 hp pump: 8" discharge * 10" suction * 14" case			Best Practices Except For Better Rotational Alignment Practices	Effects Of Practices on Component Life	Dual Pump	Good Practice Replacements		
	beta	eta		Multiplier	beta	eta			
API	shape factor (no dimensions)	location factor (hrs)	MTTF (yrs) = η*Γ(1+ 1/β)	Life Multiplier * eta From Practices	shape factor (no dimensions)	location factor (hrs)	MTTF (yrs) = η*Γ(1+ 1/β)		
Impeller	2.5	400,000	40.51	0.8250	2.50	330,007	33.43		
Housing	1.3	400,000	42.17	0.6470	1.30	258,783	27.28	MTBF	
Pump Bearings	1.3	400,000	42.17	0.4886	1.30	195,452	20.61	For All	
Seals	1.4	400,000	41.62	0.3232	1.40	129,288	13.45	**Mech.–>**	3.43
Shafts	1.2	400,000	42.95	0.4886	1.20	195,452	20.99	Items	
Coupling	2	300,000	30.35	0.6065	2.00	181,950	18.41		
Motor Bearings	1.3	150,000	15.81	0.5335	1.30	80,025	8.44	MTBF	
Motor Windings	1	150,000	17.12	1.0000	1.00	150,000	17.12	For All	
Motor Rotor	1	300,000	34.25	0.5335	1.00	160,050	18.27	**Elect.–>**	3.81
Motor Starter	1.2	300,000	32.21	1.0000	1.20	300,000	32.21	Items	
			2.99	=~Mean time between system failures=			**1.80**		
			26,209 hours		Δ = **Loss**	or=	**15,809**	hours	
					40%	loss=	**10,400**	hours	

Best Practices For Installation And Use Except For Better Piping Alignment Achieves 60% of Inherent System Life

Better Rotational Piping Practices = ±0.010 inches or ±0.254 mm

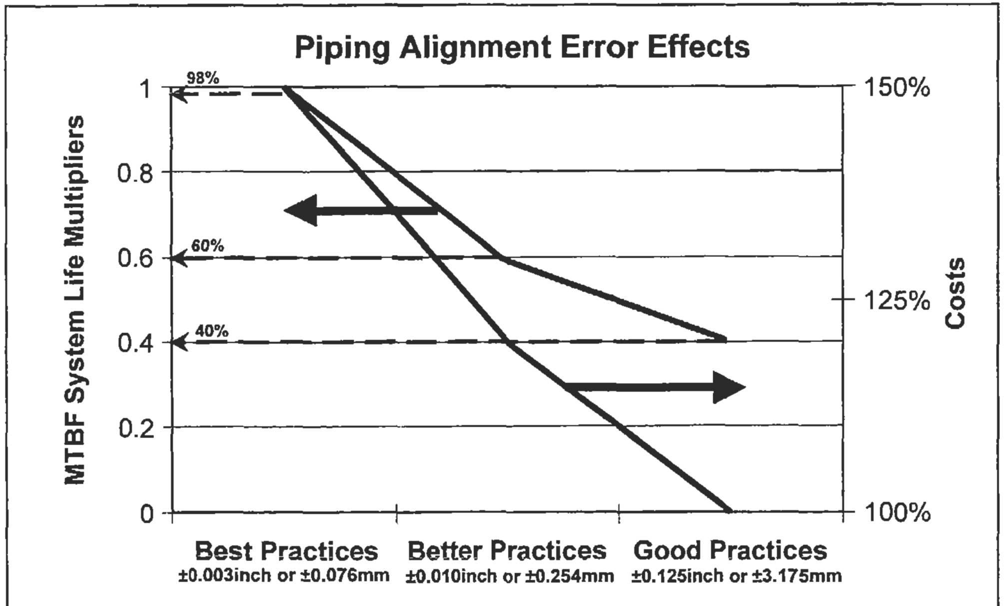
Loss Of Life From Piping Alignment Practice

Inherent Component Reliability	100 hp pump: 8" discharge * 10" suction * 14" case			Best Practices Except For Good Rotational Alignment Practices	Effects Of Practices on Component Life	Dual Pump	Good Practice Replacements		
	beta	eta		Multiplier	beta	eta			
API	shape factor (no dimensions)	location factor (hrs)	MTTF (yrs) = $\eta * \Gamma(1+ 1/\beta)$	Life Multiplier * eta From Practices	shape factor (no dimensions)	location factor (hrs)	MTTF (yrs) = $\eta * \Gamma(1+ 1/\beta)$		
Impeller	2.5	400,000	40.51	0.6018	2.50	240,722	24.38		
Housing	1.3	400,000	42.17	0.4495	1.30	179,809	18.96	MTBF	
Pump Bearings	1.3	400,000	42.17	0.2234	1.30	89,349	9.42	For All	
Seals	1.4	400,000	41.62	0.1548	1.40	61,934	6.44	**Mech.-->**	**1.88**
Shafts	1.2	400,000	42.95	0.2234	1.20	89,349	9.59	Items	
Coupling	2	300,000	30.35	0.4585	2.00	137,548	13.92		
Motor Bearings	1.3	150,000	15.81	0.4400	1.30	66,000	6.96	MTBF	
Motor Windings	1	150,000	17.12	1.0000	1.00	150,000	17.12	For All	
Motor Rotor	1	300,000	34.25	0.4400	1.00	132,000	15.07	**Elect.-->**	**3.34**
Motor Starter	1.2	300,000	32.21	1.0000	1.20	300,000	32.21	Items	
			2.99	=~Mean time between system failures=			**1.20**		
			26,209 hours		Δ = **Loss**	or=	**10,545**	hours	
					60%	loss=	**15,664**	hours	

Best Practices For Installation And Use Except For Good Piping Alignment Achieves 40% of Inherent System Life

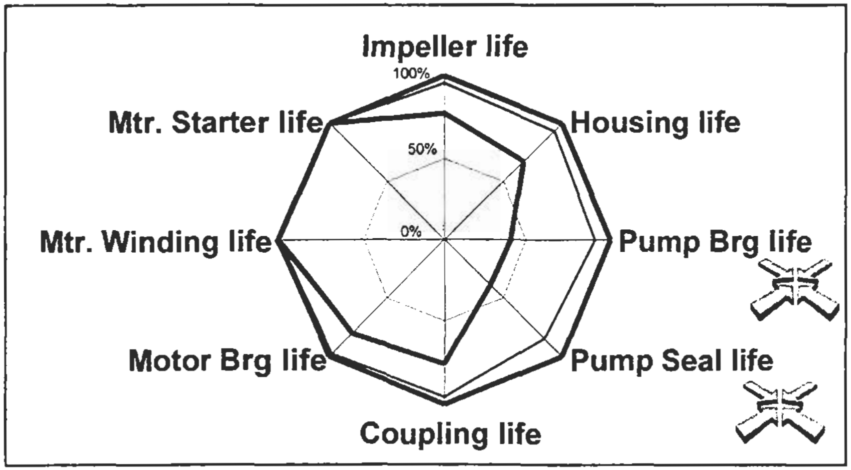
Good Rotational Piping Practices = ±0.125 inches or ±3.175 mm

Piping Alignment Effects On Costs & Reliability



System Life Includes Pump And Motor–All Other Features Use Best Practices Except As Noted

Piping Alignment Effects On Component Characteristic Life ~MTTF



▬	Best Practices	±0.003" or ±0.076mm
—	Good Practices	±0.010" or ±0.254mm
▬	Inferior Practices	±0.125" or ±3.175mm

Dynamic Rotational Balance

- Lack of rotational balance imposes high loads on shafts, bearings, housings, and couplings
- Poor balance results in short MTBF
- Good balance results in long MTBF
- Rotational balance requires multiple plane balancing of <u>all</u> rotating elements
- Rotational balance to achieve no-load conditions is desired and different standards apply to high rotational frequencies than for low rotational frequencies

Rotational Balance Effects On Costs & Reliability

Inherent Component Reliability	100 hp pump: 8" discharge * 10" suction * 14" case			Best Practices Except For Better Rotational Balance Practices	Effects Of Practices on Component Life	Dual Pump	Good Practice Replacements	
	beta	eta		Multiplier	beta	eta		
API	shape factor (no dimensions)	location factor (hrs)	MTTF (yrs) = $\eta * \Gamma(1 + 1/\beta)$	Life Multiplier * eta From Practices	shape factor (no dimensions)	location factor (hrs)	MTTF (yrs) = $\eta * \Gamma(1 + 1/\beta)$	
Impeller	2.5	400,000	40.51	0.8358	2.50	334,305	33.86	
Housing	1.3	400,000	42.17	0.6699	1.30	267,950	28.25	MTBF
Pump Bearings	1.3	400,000	42.17	0.5077	1.30	203,067	21.41	For All
Seals	1.4	400,000	41.62	0.3484	1.40	139,352	14.50	Mech.--> 3.57
Shafts	1.2	400,000	42.95	0.5077	1.20	203,067	21.81	Items
Coupling	2	300,000	30.35	0.6113	2.00	183,398	18.55	
Motor Bearings	1.3	150,000	15.81	0.5198	1.30	77,963	8.22	MTBF
Motor Windings	1	150,000	17.12	1.0000	1.00	150,000	17.12	For All
Motor Rotor	1	300,000	34.25	0.5198	1.00	155,925	17.80	Elect.--> 3.74
Motor Starter	1.2	300,000	32.21	1.0000	1.20	300,000	32.21	Items
			2.99	=~Mean time between system failures=			1.83	
			26,209 hours		Δ = Loss	or=	15,996	hours
					39%	loss=	10,213	hours

Best Practices For Installation And Use Except For Better Rotational Balance Achieves 61% of Inherent System Life

Better Rotational Balance Practices = ±0.0396 inches/sec or ±1.006 mm/sec

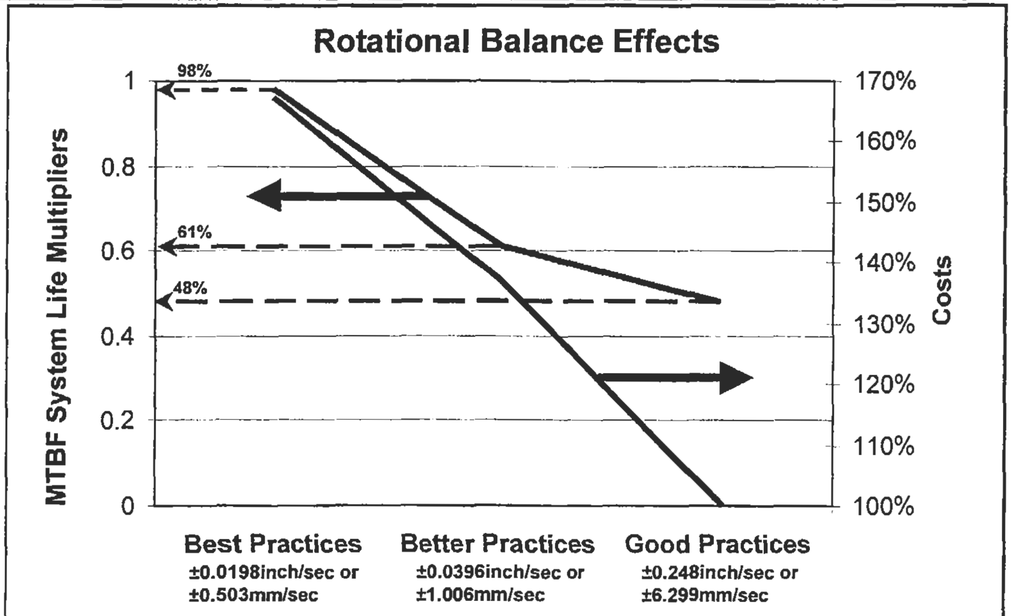
Rotational Balance Effects On Costs & Reliability

Inherent Component Reliability	100 hp pump: 8" discharge * 10" suction * 14" case			Best Practices Except For Good Rotational Balance Practices	Effects Of Practices on Component Life	Dual Pump	Good Practice Replacements		
	beta	eta		Multiplier	beta	eta			
API	shape factor (no dimensions)	location factor (hrs)	MTTF (yrs) = $\eta * \Gamma(1+ 1/\beta)$	Life Multiplier * eta From Practices	shape factor (no dimensions)	location factor (hrs)	MTTF (yrs) = $\eta * \Gamma(1+ 1/\beta)$		
Impeller	2.5	400,000	40.51	0.7148	2.50	285,919	28.96		
Housing	1.3	400,000	42.17	0.5553	1.30	222,116	23.42	MTBF	
Pump Bearings	1.3	400,000	42.17	0.3455	1.30	138,198	14.57	For All	
Seals	1.4	400,000	41.62	0.2468	1.40	98,708	10.27	**Mech.->**	2.65
Shafts	1.2	400,000	42.95	0.3455	1.20	138,198	14.84	Items	
Coupling	2	300,000	30.35	0.4987	2.00	149,614	15.14		
Motor Bearings	1.3	150,000	15.81	0.4125	1.30	61,875	6.52	MTBF	
Motor Windings	1	150,000	17.12	0.9500	1.00	142,500	16.27	For All	
Motor Rotor	1	300,000	34.25	0.4125	1.00	123,750	14.13	**Elect.->**	3.16
Motor Starter	1.2	300,000	32.21	1.0000	1.20	300,000	32.21	Items	
			2.99	=~Mean time between system failures=			**1.44**		
			26,209 hours		Δ = Loss	or=	**12,635**	hours	
					52%	loss=	**13,573**	hours	

Best Practices For Installation And Use Except For Good Rotational Balance Achieves 48% of Inherent System Life

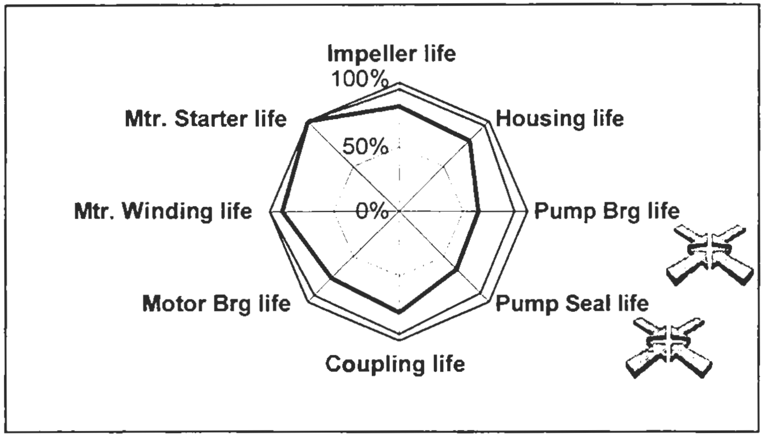
Good Rotational Balance Practices = ±0.248 inches/sec or ±6.299 mm/sec

Rotating Balance Effects On Costs & Reliability



System Life Includes Pump And Motor–All Other Features Use Best Practices Except As Noted

Rotating Balance Effects On Component Characteristic Life ~MTTF



—— Best Practices = 0.0198 ips or 0.503 mm/sec

—— Good Practices = 0.0396 ips or 1.006 mm/sec

—— Inferior Practices = 0.2480 ips or 6.299 mm/sec

Pump Foundations

- Lack of robust foundations result in high mechanical loads imposed onto the rotating system from stilt mountings, low mass foundations, or simply bolting pumps to the floor
- Poor foundations result in short MTTF
- Good foundations results in long MTTF
- Pump foundations require high mass, high rigidity, long design life, robust designs, and good installation procedures to isolate the foundation from other vibrating structures

Pump Foundation Effects On Costs & Reliability

Inherent Component Reliability	100 hp pump: 8" discharge * 10" suction * 14" case			Best Practices Except For Better Pump Foundation Practices	Effects Of Practices on Component Life	Dual Pump	Good Practice Replacements		
	beta	eta		Multiplier	beta	eta			
API	shape factor (no dimensions)	location factor (hrs)	MTTF (yrs) = $\eta*\Gamma(1+\ 1/\beta)$	Life Multiplier * eta From Practices	shape factor (no dimensions)	location factor (hrs)	MTTF (yrs) = $\eta*\Gamma(1+\ 1/\beta)$		
Impeller	2.5	400,000	40.51	0.9240	2.50	369,593	37.43		
Housing	1.3	400,000	42.17	0.8161	1.30	326,420	34.41	MTBF	
Pump Bearings	1.3	400,000	42.17	0.8741	1.30	349,628	36.86	For All	
Seals	1.4	400,000	41.62	0.8710	1.40	348,381	36.25	**Mech.–>**	5.73
Shafts	1.2	400,000	42.95	0.8741	1.20	349,628	37.54	Items	
Coupling	2	300,000	30.35	0.8821	2.00	264,627	26.77		
Motor Bearings	1.3	150,000	15.81	0.9000	1.30	135,000	14.23	MTBF	
Motor Windings	1	150,000	17.12	0.9875	1.00	148,125	16.91	For All	
Motor Rotor	1	300,000	34.25	0.9000	1.00	270,000	30.82	**Elect.–>**	5.18
Motor Starter	1.2	300,000	32.21	1.0000	1.20	300,000	32.21	Items	
			2.99	=~Mean time between system failures=			**2.72**		
			26,209 hours	Δ = **Loss**		or=	**23,846**	hours	
					9%	loss=	**2,363**	hours	

Best Practices For Installation And Use Except For Better Pump Foundation Achieves 91% of Inherent System Life

Better Pump Foundation Practices = 3.5 Pump System Mass

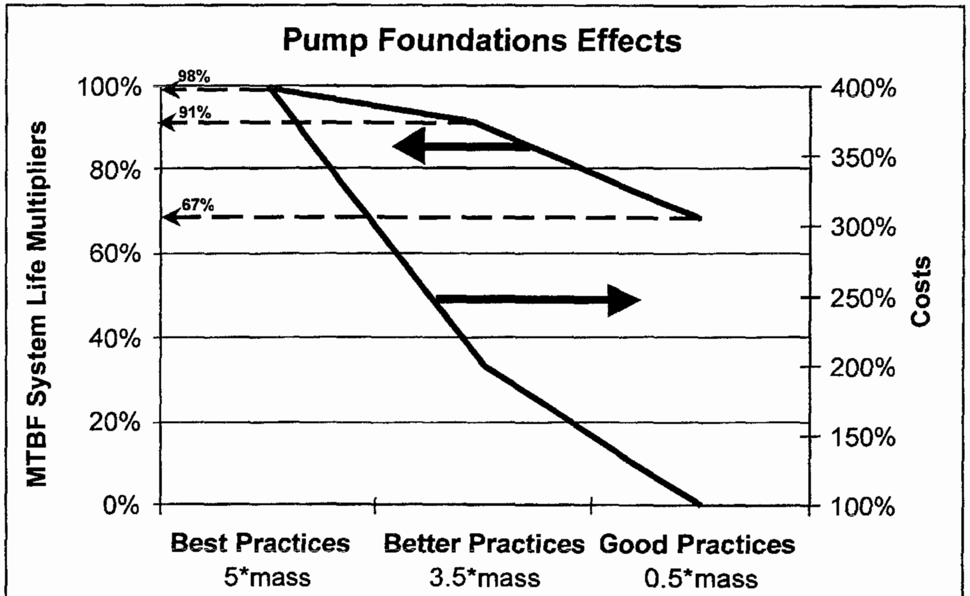
Pump Foundation Effects On Costs & Reliability

Inherent Component Reliability	100 hp pump: 8" discharge * 10" suction * 14" case			Best Practices Except For Good Pump Foundation Practices	Effects Of Practices on Component Life	Dual Pump	Good Practice Replacements		
	beta	eta		Multiplier	beta	eta			
API	shape factor (no dimensions)	location factor (hrs)	MTTF (yrs) = $\eta*\Gamma(1+\ 1/\beta)$	Life Multiplier * eta From Practices	shape factor (no dimensions)	location factor (hrs)	MTTF (yrs) = $\eta*\Gamma(1+\ 1/\beta)$		
Impeller	2.5	400,000	40.51	0.8510	2.50	340,414	34.48	MTBF For All Mech.--> Items	3.93
Housing	1.3	400,000	42.17	0.6657	1.30	266,290	28.08		
Pump Bearings	1.3	400,000	42.17	0.4856	1.30	194,238	20.48		
Seals	1.4	400,000	41.62	0.5322	1.40	212,900	22.15		
Shafts	1.2	400,000	42.95	0.4856	1.20	194,238	20.86		
Coupling	2	300,000	30.35	0.6861	2.00	205,821	20.82		
Motor Bearings	1.3	150,000	15.81	0.6000	1.30	90,000	9.49	MTBF For All Elect.--> Items	4.07
Motor Windings	1	150,000	17.12	0.9600	1.00	144,000	16.44		
Motor Rotor	1	300,000	34.25	0.6000	1.00	180,000	20.55		
Motor Starter	1.2	300,000	32.21	1.0000	1.20	300,000	32.21		
			2.99	=~Mean time between system failures=			2.00		
			26,209 hours		Δ = Loss	or=	17,503	hours	
					33%	loss=	8,706	hours	

Best Practices For Installation And Use Except For Good Pump Foundation Achieves 67% of Inherent System Life

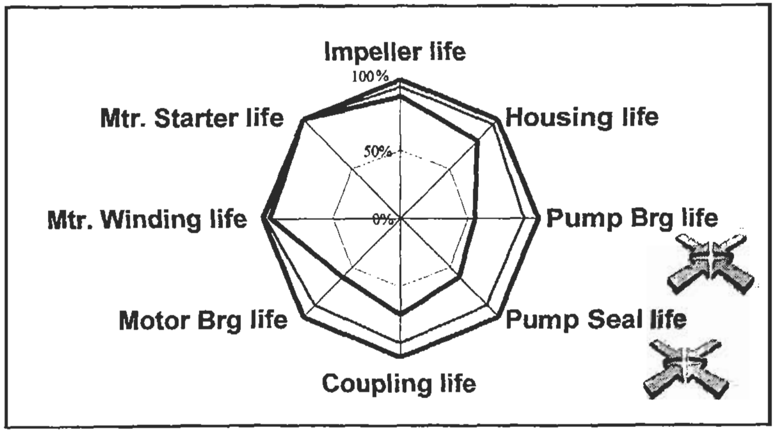
Good Pump Foundation Practices = 0.5 Pump System Mass--Stilt Mounted

Foundation Effects On Costs & Reliability



System Life Includes Pump And Motor–All Other Features Use Best Practices Except As Noted

Foundation Effects On Component Characteristic Life ~MTTF



—— Best Practices = 5*mass

—— Good Practices = 3.5*mass

—— Inferior Practices = 0.5*mass

Grouting Of Pump Bases/Foundations

- **The purpose of grout is to make the pump base and foundation monolithic to reduce vibrations**
- **Poor grouting allows pump bases to have high amplitude vibrations which destroy inherent reliability and yield short MTTF**
- **Good grouting attenuates vibrations and results in long MTTF**
- **Grouting of pumps to foundations require a void-free, adhesive attachment between pump base and concrete foundation with moisture-free materials that will not crack or allow entrance of moisture for long periods of time**

Grouting Effects On Costs & Reliability

Inherent Component Reliability	100 hp pump: 8" discharge * 10" suction * 14" case			Best Practices Except For Best Grout Practices	Effects Of Practices on Component Life	Dual Pump	Good Practice Replacements		
	beta	eta		Multiplier	beta	eta			
API	shape factor (no dimensions)	location factor (hrs)	MTTF (yrs) = $\eta*\Gamma(1+ 1/\beta)$	Life Multiplier * eta From Practices	shape factor (no dimensions)	location factor (hrs)	MTTF (yrs) = $\eta*\Gamma(1+ 1/\beta)$		
Impeller	2.5	400,000	40.51	0.9532	2.50	381,264	38.62		
Housing	1.3	400,000	42.17	0.8120	1.30	324,788	34.24	MTBF	
Pump Bearings	1.3	400,000	42.17	0.8741	1.30	349,628	36.86	For All	
Seals	1.4	400,000	41.62	0.8710	1.40	348,381	36.25	**Mech.–>**	5.76
Shafts	1.2	400,000	42.95	0.8741	1.20	349,628	37.54	Items	
Coupling	2	300,000	30.35	0.8870	2.00	266,097	26.92		
Motor Bearings	1.3	150,000	15.81	0.9000	1.30	135,000	14.23	MTBF	
Motor Windings	1	150,000	17.12	0.9900	1.00	148,500	16.95	For All	
Motor Rotor	1	300,000	34.25	0.9000	1.00	270,000	30.82	**Elect.–>**	5.19
Motor Starter	1.2	300,000	32.21	1.0000	1.20	300,000	32.21	Items	
			2.99	=~Mean time between system failures=			2.73		
			26,209 hours	Δ = Loss		or=	23,913	hours	
					9%	loss=	2,296	hours	

Best Practices For Installation And Use Except For Better Grout Achieves 91% of Inherent System Life

Better Grout Practices = Slightly Porous But Adhesive

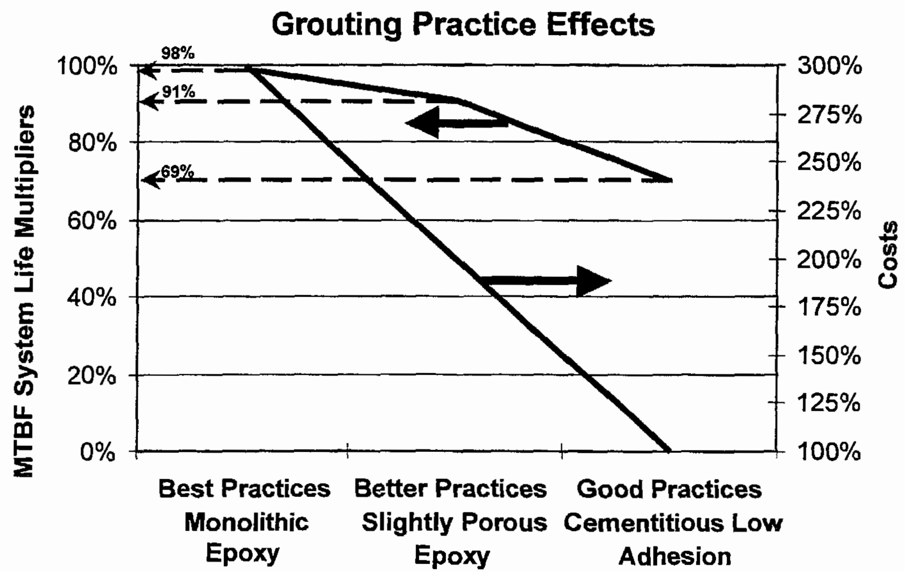
Grouting Effects On Costs & Reliability

Inherent Component Reliability	100 hp pump: 8" discharge * 10" suction * 14" case			Best Practices Except For Good Grout Practices	Effects Of Practices on Component Life	Dual Pump	Good Practice Replacements		
	beta	eta		Multiplier	beta	eta			
API	shape factor (no dimensions)	location factor (hrs)	MTTF (yrs) = $\eta * \Gamma(1+1/\beta)$	Life Multiplier * eta From Practices	shape factor (no dimensions)	location factor (hrs)	MTTF (yrs) = $\eta * \Gamma(1+1/\beta)$		
Impeller	2.5	400,000	40.51	0.8510	2.50	340,414	34.48		
Housing	1.3	400,000	42.17	0.6838	1.30	273,506	28.84	MTBF	
Pump Bearings	1.3	400,000	42.17	0.5342	1.30	213,662	22.53	For All	
Seals	1.4	400,000	41.62	0.5322	1.40	212,900	22.15	**Mech.-->**	**4.14**
Shafts	1.2	400,000	42.95	0.5342	1.20	213,662	22.94	Items	
Coupling	2	300,000	30.35	0.7351	2.00	220,523	22.31		
Motor Bearings	1.3	150,000	15.81	0.6000	1.30	90,000	9.49	MTBF	
Motor Windings	1	150,000	17.12	0.9800	1.00	147,000	16.78	For All	
Motor Rotor	1	300,000	34.25	0.6000	1.00	180,000	20.55	**Elect.-->**	**4.09**
Motor Starter	1.2	300,000	32.21	1.0000	1.20	300,000	32.21	Items	
			2.99	=~Mean time between system failures=			**2.06**		
			26,209	hours	Δ = **Loss**	or=	**18,013**	hours	
					31%	loss=	**8,196**	hours	

Best Practices For Installation And Use Except For Good Grout Achieves 69% of Inherent System Life

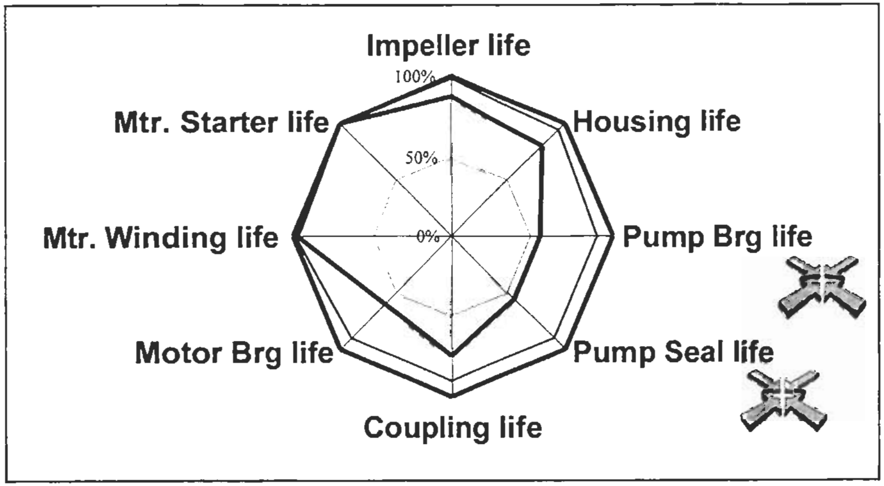
Good Grout Practices = Cementitious & Low Adhesion

Grouting Effects On Costs & Reliability



System Life Includes Pump And Motor—All Other Features Use Best Practices Except As Noted

Grouting Effects On Component Reliability



—— Best Practices = Monolithic Epoxy
—— Good Practices = Slightly Porous Epoxy
—— Inferior Practices = Cementitious Low Adhesion

Summary Of Best Practices

This is the highest grade for installation and use--little life is lost from these practices

		Pump Curve % Off BEP	L/D Suction Straight Runs	Rotational Shaft Alignment	Piping Alignment	Rotational Balance	Foundation Design	Grouting
Best Practices	**Resulting eta Multiplier**	+ 5% to -10% of BEP	L/D = 10 to 12	±0.001 inches/inch error	±0.003 inch error	Smooth at 0.0198 ips	5 Times Equipment Mass	Monolithic And Adhesive Epoxy
Impeller	0.9726	98%	100%	100%	100%	100%	100%	100%
Housing	0.8547	86%	100%	100%	100%	100%	100%	100%
Pump Bearings	0.8719	98%	100%	100%	100%	99%	100%	100%
Seals	0.9533	98%	99%	100%	100%	100%	100%	100%
Shafts	0.8719	98%	100%	100%	100%	99%	100%	100%
Coupling	0.9801	99%	100%	99%	100%	100%	100%	100%
Motor Bearings	1.0000	100%	100%	100%	100%	100%	100%	100%
Motor Windings	1.0000	100%	100%	100%	100%	100%	100%	100%
Motor Rotor	1.0000	100%	100%	100%	100%	100%	100%	100%
Motor Starter	1.0000	100%	100%	100%	100%	100%	100%	100%

These life results are from a pump survey conducted from expert sources from around the world

Loss Of Inherent Reliability- Best Practices

Use life cycle costs. Justify each practice. Find the lowest long-term cost of ownership

Best practices is the highest grade for installation and use—little system life is lost

Inherent Component Reliability	100 hp pump: 8" discharge * 10" suction * 14" case			Best Practices For Installation & Use	Effects Of Practices on Component Life	Dual Pump	Good Practice Replacements		
	beta	eta		Multiplier	beta	eta			
API	shape factor (no dimensions)	location factor (hrs)	MTTF (yrs) = $\eta * \Gamma(1+ \ 1/\beta)$	Life Multiplier * eta From Practices	shape factor (no dimensions)	location factor (hrs)	MTTF (yrs) = $\eta * \Gamma(1+ \ 1/\beta)$		
Impeller	2.5	400,000	40.51	0.9726	2.50	389,045	39.40		
Housing	1.3	400,000	42.17	0.8547	1.30	341,882	36.05	MTBF	
Pump Bearings	1.3	400,000	42.17	0.9712	1.30	388,476	40.96	For All	
Seals	1.4	400,000	41.62	0.9677	1.40	387,090	40.27	**Mech.->**	6.25
Shafts	1.2	400,000	42.95	0.9712	1.20	388,476	41.71	Items	
Coupling	2	300,000	30.35	0.9801	2.00	294,030	29.75		
Motor Bearings	1.3	150,000	15.81	1.0000	1.30	150,000	15.81	MTBF	
Motor Windings	1	150,000	17.12	1.0000	1.00	150,000	17.12	For All	
Motor Rotor	1	300,000	34.25	1.0000	1.00	300,000	34.25	**Elect.->**	5.50
Motor Starter	1.2	300,000	32.21	1.0000	1.20	300,000	32.21	Items	
			2.99	=~Mean time between system failures=			**2.93**		
			26,209 hours		Δ = **Loss**	or =	**25,627**	hours	
					2%	loss=	**582**	hours	

All best practices are used simultaneously to achieve 98% of the inherent system life

Summary Of Better Practices

Note the deteriorating effects of the lower grade for installation and use

		Pump Curve % Off BEP	L/D Suction Straight Runs	Rotational Shaft Alignment	Piping Alignment	Rotational Balance	Foundation Design	Grouting
Better Practices	**Resulting eta Multiplier**	+ 10% to -20% of BEP	L/D = 6 to 8	±0.003 inches/inch error	±0.010 inch error	Good at 0.0448 ips	3.5 Times Equipment Mass	Slightly Pourous But Adhesive
Impeller	0.6583	88%	95%	95%	94%	95%	95%	98%
Housing	0.5163	73%	95%	95%	92%	95%	95%	95%
Pump Bearings	0.3950	79%	90%	88%	88%	90%	90%	90%
Seals	0.4314	88%	90%	90%	84%	90%	90%	90%
Shafts	0.3950	79%	90%	88%	88%	90%	90%	90%
Coupling	0.5705	92%	95%	90%	94%	95%	90%	91%
Motor Bearings	0.6036	94%	93%	94%	97%	95%	90%	90%
Motor Windings	0.9776	100%	100%	100%	100%	100%	99%	99%
Motor Rotor	0.6036	94%	93%	94%	97%	95%	90%	90%
Motor Starter	1.0000	100%	100%	100%	100%	100%	100%	100%

These life results are from a pump survey conducted from expert sources from around the world

Loss Of Inherent Reliability- Better Practices

Frequently these costly, life consuming , low-grade practices are used simultaneously

Note the *substantial loss of* system life from the lower grade for installation and use

Inherent Component Reliability	100 hp pump: 8" discharge * 10" suction * 14" case			Better Practices For Installation And Use	Effects Of Practices on Component Life	Dual Pump	Good Practice Replacements		
	beta	eta		Multiplier	beta	eta			
API	shape factor (no dimensions)	location factor (hrs)	MTTF (yrs) = $\eta*\Gamma(1+ 1/\beta)$	Life Multiplier * eta From Practices	shape factor (no dimensions)	location factor (hrs)	MTTF (yrs) = $\eta*\Gamma(1+ 1/\beta)$		
Impeller	2.5	400,000	40.51	0.6583	2.50	263,312	26.67	MTBF For All Mech.--> Items	
Housing	1.3	400,000	42.17	0.5163	1.30	206,529	21.77		
Pump Bearings	1.3	400,000	42.17	0.3950	1.30	158,005	16.66		
Seals	1.4	400,000	41.62	0.4314	1.40	172,571	17.95		3.17
Shafts	1.2	400,000	42.95	0.3950	1.20	158,005	16.97		
Coupling	2	300,000	30.35	0.5705	2.00	171,161	17.32		
Motor Bearings	1.3	150,000	15.81	0.6036	1.30	90,544	9.55	MTBF For All Elect.--> Items	
Motor Windings	1	150,000	17.12	0.9776	1.00	146,644	16.74		
Motor Rotor	1	300,000	34.25	0.6036	1.00	181,089	20.67		4.10
Motor Starter	1.2	300,000	32.21	1.0000	1.20	300,000	32.21		
			2.99	=~Mean time between system failures=			1.79		
			26,209 hours	Δ = Loss		or=	15,650	hours	
					40%	loss=	10,559	hours	

All better practices are used simultaneously to achieve 60% of the inherent system life

Summary Of Good Practices

This lowest grade for installation and use results in the loss of substantial life!

		Pump Curve % Off BEP	L/D Suction Straight Runs	Rotational Shaft Alignment	Piping Alignment	Rotational Balance	Foundation Design	Grouting
Good Practices	**Resulting eta Multiplier**	+15% to -30% of BEP	L/D = 1 to 3	±0.009 inches/inch error	±0.125 inches error	Rough at 0.248 ips	0.5 Times Equipment Mass or Stilt-Mounted	Cementitious & Low Adhesion
Impeller	0.1949	68%	75%	90%	69%	81%	88%	88%
Housing	0.1438	70%	80%	83%	64%	79%	78%	80%
Pump Bearings	0.0151	65%	60%	58%	40%	61%	50%	55%
Seals	0.0095	51%	60%	40%	40%	64%	55%	55%
Shafts	0.0151	65%	60%	58%	40%	61%	50%	55%
Coupling	0.1149	76%	80%	65%	71%	78%	70%	75%
Motor Bearings	0.0737	78%	80%	55%	80%	75%	60%	60%
Motor Windings	0.8625	97%	100%	100%	100%	95%	96%	98%
Motor Rotor	0.0737	78%	80%	55%	80%	75%	60%	60%
Motor Starter	1.0000	100%	100%	100%	100%	100%	100%	100%

These life results are from a pump survey conducted from expert sources from around the world

Loss Of Inherent Reliability- Good Practices

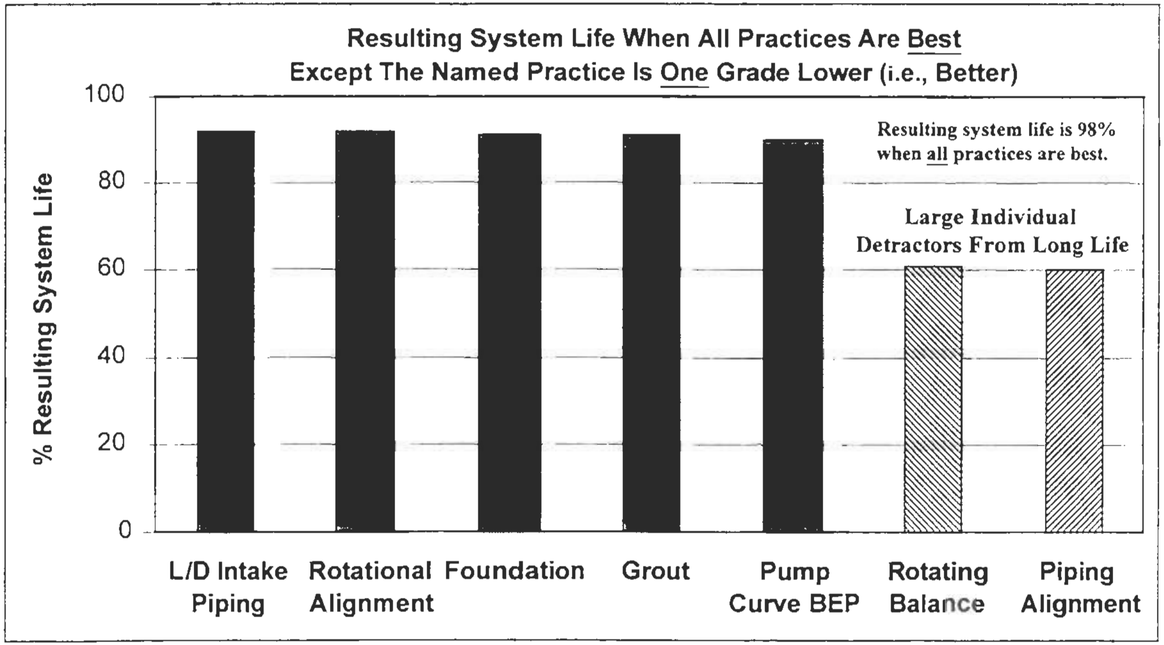
Seldom are all these inferior practices used simultaneously

Note that nearly all system life is lost from this lowest grade for installation and use

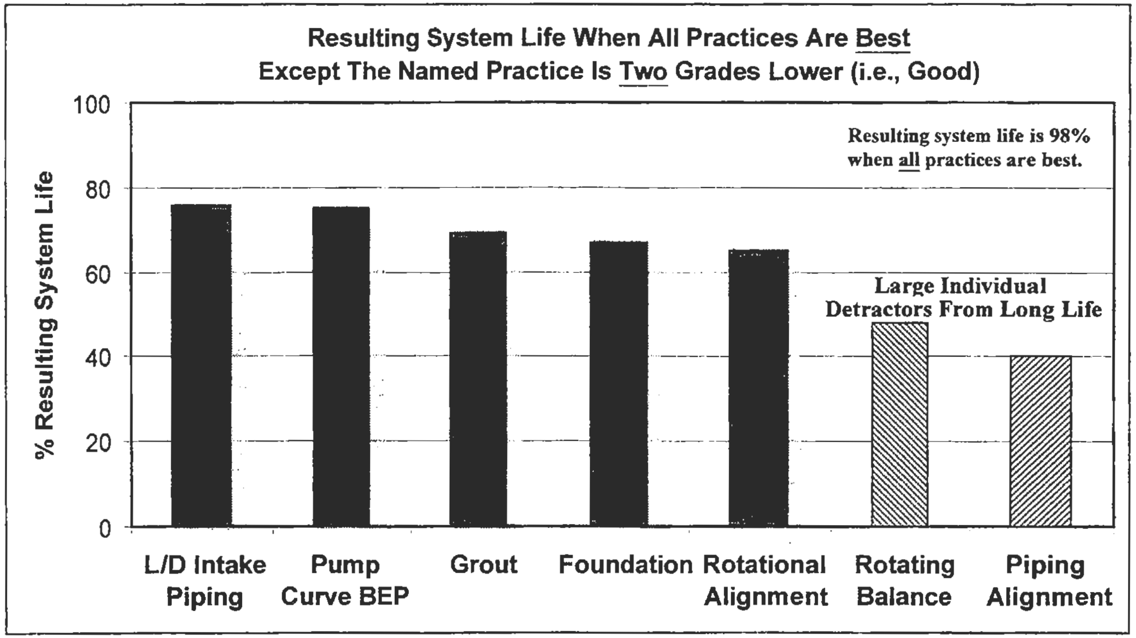
Inherent Component Reliability	100 hp pump: 8" discharge * 10" suction * 14" case			Good Practices For Installation And Use	Effects Of Practices on Component Life	Dual Pump	Good Practice Replacements		
	beta	eta		Multiplier	beta	eta			
API	shape factor (no dimensions)	location factor (hrs)	MTTF (yrs) = $\eta*\Gamma(1+ 1/\beta)$	Life Multiplier * eta From Practices	shape factor (no dimensions)	location factor (hrs)	MTTF (yrs) = $\eta*\Gamma(1+ 1/\beta)$		
Impeller	2.5	400,000	40.51	0.1949	2.50	77,943	7.89		
Housing	1.3	400,000	42.17	0.1438	1.30	57,521	6.06	MTBF	
Pump Bearings	1.3	400,000	42.17	0.0151	1.30	6,044	0.64	For All	
Seals	1.4	400,000	41.62	0.0095	1.40	3,795	0.39	**Mech.-->**	0.16
Shafts	1.2	400,000	42.95	0.0151	1.20	6,044	0.65	Items	
Coupling	2	300,000	30.35	0.1149	2.00	34,483	3.49		
Motor Bearings	1.3	150,000	15.81	0.0737	1.30	11,048	1.16	MTBF	
Motor Windings	1	150,000	17.12	0.8625	1.00	129,372	14.77	For All	
Motor Rotor	1	300,000	34.25	0.0737	1.00	22,097	2.52	**Elect.-->**	0.74
Motor Starter	1.2	300,000	32.21	1.0000	1.20	300,000	32.21	Items	
			2.99	=~Mean time between system failures=			**0.13**		
			26,209 hours		Δ = **Loss**	or=	**1,156**	hours	
					96%	loss=	**25,052**	hours	

All good practices are used simultaneously to achieve only 4% of the inherent system life

Pareto Distributions Of Severity–1 Grade Lower



Pareto Distributions Of Severity—2 Grades Lower



How To Correct Short MTTF

- Increase equipment strength–larger inherent MTTF
- Decrease equipment loads–use better practices to decrease loads or operate the equipment derated
- Mix and match loads/strengths for cost effectiveness
  - Use life cycle costs to decide load/strength mix and match strategy
  - Use higher grade equipment/practices as found by life cycle costs
  - Train engineering personnel to select and specify both equipment and installation practices, and operating practices which are cost effective
  - Involve production personnel in maintenance problems to stop equipment abuses which increase loads and reduce life
  - Train maintenance personnel to repair equipment correctly with a view toward correcting cost ineffective practices causing decreased reliability

Why Work On Reliability Issues?

- Solving reliability problems solves cost problems
- Set reliability goals as business items for cost reductions
- Solving reliability problems requires new tools and training to both predict and solve root causes of failures
- Reliability problems are often people/procedure problems and teamwork helps solve the root cause of the problem
- Use reliability engineers as strategic resources to prevent failures while using maintenance engineers as tactical failure restorers until problems are permanently resolved
- Make reliability improvements pay their way by working toward the lowest life cycle cost for the business

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MECHANICAL ENGINEERING

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About the Author

Heinz P. Bloch, P.E., served as Exxon Chemical's regional machinery specialist for the U.S. and in machinery-oriented staff and line positions with Exxon affiliates throughout the world. He retired from Exxon Chemical Company in 1986 after more than 20 years of service, and is presently a licensed consulting engineer with offices in Montgomery, Texas. He co-authored *Reciprocating Compressors* also from Gulf Publishing Company.

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