BEST PRACTICE MANUAL

FLUID PIPING SYSTEMS

Prepared for

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1 INTRODUCTION

1.1 Background

Selection of piping system is an important aspect of system design in any energy consuming system. The selection issues such as material of pipe, configuration, diameter, insulation etc have their own impact on the overall energy consumption of the system. Piping is one of those few systems when you oversize, you will generally save energy; unlike for a motor or a pump.

Piping system design in large industrial complexes like Refineries, Petrochemicals, Fertilizer Plants etc are done now a day with the help of design software, which permits us to try out numerous possibilities. It is the relatively small and medium users who generally do not have access to design tools use various rules of thumbs for selecting size of pipes in industrial plants. These methods of piping design are based on either “worked before” or “educated estimates”. Since everything we do is based on sound economic principles to reduce cost, some of the piping design thumb rules are also subject to modification to suit the present day cost of piping hardware cost and energy cost. It is important to remember that there are no universal rules applicable in every situation. They are to be developed for different scenarios.

For example, a water piping system having 1 km length pumping water from a river bed pumping station to a plant will have different set of rules compared to a water piping system having 5 meter length for supplying water from a main header to a reactor. Hence the issue of pipe size i.e. diameter, selection should be based on reducing the overall cost of owning and operating the system.

This guidebook covers the best practices in piping systems with a primary view of reducing energy cost, keeping in mind the safety and reliability issues. The basic elements of best practice in piping systems are:

1. Analysis & optimum pipe size selection for water, compressed air and steam distribution systems
2. Good piping practices
3. Thermal insulation of piping system

2 FUNDAMENTALS

2.1 Physical Properties of Fluids

The properties relevant to fluid flow are summarized below.

Density: This is the mass per unit volume of the fluid and is generally measured in kg/m3. Another commonly used term is specific gravity. This is in fact a relative density, comparing the density of a fluid at a given temperature to that of water at the same temperature.

Viscosity: This describes the ease with which a fluid flows. A substance like treacle has a high viscosity, while water has a much lower value. Gases, such as air, have a still lower viscosity. The viscosity of a fluid can be described in two ways.

- Absolute (or dynamic) viscosity: This is a measure of a fluid's resistance to internal deformation. It is expressed in Pascal seconds (Pa s) or Newton seconds per square metre (Ns/m^2). [1Pas = 1 Ns/m^2]

- Kinematic viscosity: This is the ratio of the absolute viscosity to the density and is measured in metres squared per second (m^2/s).

Reynolds Number: A useful factor in determining which type of flow is involved is the Reynolds number. This is the ratio of the dynamic forces of mass flow to the shear resistance due to fluid viscosity and is given by the following formula. In general for a fluid like water when the Reynolds number is less than 2000 the flow is laminar. The flow is turbulent for Reynolds numbers above 4000. In between these two values (2000<Re<4000) the flow is a mixture of the two types and it is difficult to predict the behavior of the fluid.

$$\mathrm{Re} = \frac{\rho \times \mathrm{u} \times \mathrm{d}}{1000 \times \mu}$$

Where:
ρ = Density (kg/m^3)
u = Mean velocity in the pipe (m/s)
d = Internal pipe diameter (mm)
μ = Dynamic viscosity (Pa s)

2.2 Types of Fluid Flow:

When a fluid moves through a pipe two distinct types of flow are possible, laminar and turbulent.

Laminar flow occurs in fluids moving with small average velocities and turbulent flow becomes apparent as the velocity is increased above a critical velocity. In laminar flow the fluid particles move along the length of the pipe in a very orderly fashion, with little or no sideways motion across the width of the pipe.

Turbulent flow is characterised by random, disorganised motion of the particles, from side to side across the pipe as well as along its length. There will, however, always be a layer of laminar flow at the pipe wall - the so-called 'boundary layer'. The two types of fluid flow are described by different sets of equations. In general, for most practical situations, the flow will be turbulent.

2.3 Pressure Loss in Pipes

Whenever fluid flows in a pipe there will be some loss of pressure due to several factors:

a) **F**riction: This is affected by the roughness of the inside surface of the pipe, the pipe diameter, and the physical properties of the fluid.

b) **C**hanges in size and shape or direction of flow

a) Obstructions: For normal, cylindrical straight pipes the major cause of pressure loss will be friction. Pressure loss in a fitting or valve is greater than in a straight pipe. When fluid flows in a straight pipe the flow pattern will be the same through out the pipe. In a valve or fitting changes in the flow pattern due to factors (b) and (c) will cause extra pressure drops. Pressure drops can be measured in a number of ways. The SI unit of pressure is the Pascal. However pressure is often measured in bar.

This is illustrated by the D'Arcy equation:

$$h_f = \frac{fLu^2}{2gd}$$

Where:
L = Length (m)
u = Flow velocity (m/s)
g = Gravitational constant (9.81 m/s²)
d = Pipe inside diameter (m)
h_f = Head loss to friction (m)
f = Friction factor (dimensionless)

Before the pipe losses can be established, the friction factor must be calculated. The friction factor will be dependant on the pipe size, inner roughness of the pipe, flow velocity and fluid viscosity. The flow condition, whether 'Turbulent' or not, will determine the method used to calculate the friction factor.

Fig 2.1 can be used to estimate friction factor. Roughness of pipe is required for friction factor estimation. The chart shows the relationship between Reynolds number and pipe friction. Calculation of friction factors is dependant on the type of flow that will be encountered. For Re numbers <2320 the fluid flow is laminar, when Re number is >= 2320 the fluid flow is turbulent.

The following table gives typical values of absolute roughness of pipes, k. The relative roughness k/d can be calculated from k and inside diameter of pipe.

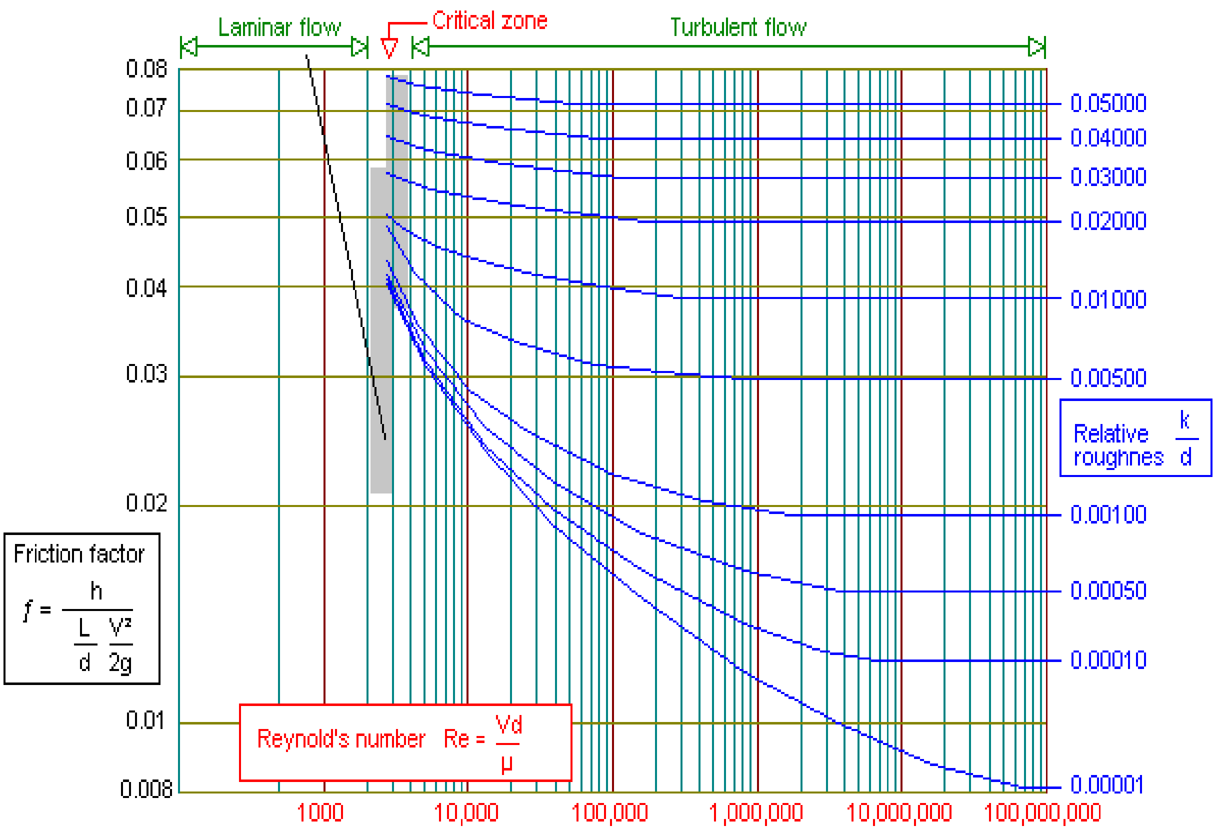


Figure 2-1: Estimation of friction factor

The absolute roughness of pipes is given below.

Type of pipe	k, mm
Plastic tubing	0.0015
Stainless steel	0.015
Rusted steel	0.1 to 1.0
Galvanised iron	0.15
Cast iron	0.26

A sample calculation of pressure drop is given below.

A pipe of 4" Dia carrying water flow of 50 m^3/h through a distance of 100 metres. The pipe material is Cast Iron with absolute roughness of 0.26.

$$\text{Velocity}, m/s = \frac{\text{Flow}, m^3/h}{3600 \times \text{Pipe Cross Section Area}, m^2} =$$

$$= \frac{\text{Flow}, m^3/h}{3600 \times \left[3.14 \times (d/1000)^2 / 4\right]}$$

$$= \frac{50}{3600 \times \left[3.14 \times (100/1000)^2 \big/ 4 \right]}$$

= 1.77m/s

$$Re = \frac{\rho \times u \times \left(d \big/ 1000 \right)}{\mu}$$

Where:
ρ = Density (kg/m^3) = 1000
u = Mean velocity in the pipe (m/s) = 1.77
d = Internal pipe diameter (mm) =100
μ = Dynamic viscosity (Pa s). For water at 25º C, the value is 0.001 Pa-s

$$Re = \frac{\rho \times u \times \left(d \big/ 1000 \right)}{\mu} = \frac{1000 \times 1.77 \times \left(100 \big/ 1000 \right)}{0.001} = 177000$$

Relative roughness, k/d = 0.26/100= 0.0026

From fig 2.3, corresponding to Re = 177000 and k/d of 0.0026, friction factor in the turbulent region is 0.025.

Head loss = $h_f = \frac{fLu^2}{2gd} = \frac{0.025 \times 100 \times 1.77^2}{2 \times 9.81 \times (100/1000)}$ = 4.0 m per 100 m length.

2.4 Standard Pipe dimensions

There are a number of piping standards in existence around the world, but arguably the most global are those derived by the American Petroleum Institute (API), where pipes are categorised in schedule numbers. These schedule numbers bear a relation to the pressure rating of the piping. There are eleven Schedules ranging from the lowest at 5 through 10, 20, 30, 40, 60, 80, 100, 120, 140 to schedule No. 160. For nominal size piping 150 mm and smaller, Schedule 40 (sometimes called ‘standard weight’) is the lightest that would be specified for water, compressed air and steam applications. High-pressure compressed air will have schedule 80 piping.

Regardless of schedule number, pipes of a particular size all have the same outside diameter (not withstanding manufacturing tolerances). As the schedule number increases, the wall thickness increases, and the actual bore is reduced. For example:

- A 100 mm Schedule 40 pipe has an outside diameter of 114.30 mm, a wall thickness of 6.02 mm, giving a bore of 102.26 mm.

- A 100 mm Schedule 80 pipe has an outside diameter of 114.30 mm, a wall thickness of 8.56 mm, giving a bore of 97.18 mm.

2.5 Pressure drop in components in pipe systems

Minor head loss in pipe systems can be expressed as:

$$h_{minor\_loss} = \frac{ku^2}{2g}$$

where h_{minor_loss} = minor head loss (m)
k = minor loss coefficient
u = flow velocity (m/s)
g = acceleration of gravity (m/s^2)

Minor loss coefficients for some of the most common used components in pipe and tube systems are given in table 2.1.

Table 2-1: Minor loss coefficients

Type of Component or Fitting	Minor Loss Coefficient, k
Flanged Tees, Line Flow	0.2
Threaded Tees, Line Flow	0.9
Flanged Tees, Branched Flow	1.0
Threaded Tees, Branch Flow	2.0
Threaded Union	0.08
Flanged Regular 90^o Elbows	0.3
Threaded Regular 90^o Elbows	1.5
Threaded Regular 45^o Elbows	0.4
Flanged Long Radius 90^o Elbows	0.2
Threaded Long Radius 90^o Elbows	0.7
Flanged Long Radius 45^o Elbows	0.2
Flanged 180^o Return Bends	0.2
Threaded 180^o Return Bends	1.5
Fully Open Globe Valve	10
Fully Open Angle Valve	2
Fully Open Gate Valve	0.15
1/4 Closed Gate Valve	0.26
1/2 Closed Gate Valve	2.1
3/4 Closed Gate Valve	17
Forward Flow Swing Check Valve	2
Fully Open Ball Valve	0.05
1/3 Closed Ball Valve	5.5
2/3 Closed Ball Valve	200

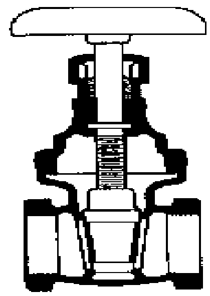
The above equations and table can be used for calculating pressure drops and energy loss associated in pipes and fittings.

2.6 Valves

Valves isolate, switch and control fluid flow in a piping system. Valves can be operated manually with levers and gear operators or remotely with electric, pneumatic, electro-pneumatic, and electro-hydraulic powered actuators. Manually operated valves are typically used where operation is infrequent and/or a power source is not available. Powered actuators allow valves to be operated automatically by a control system and remotely with push button stations. Valve automation brings significant advantages to a plant in the areas of process quality, efficiency, safety, and productivity.

Types of valves and their features are summarised below.

- **Gate Valves** have a sliding disc (gate) that reciprocates into and out of the valve port. Gate valves are an ideal isolation valve for high pressure drop and high temperature applications where operation is infrequent. Manual operation is accomplished through a multi turn hand wheel gear shaft assembly. Multiturn electric actuators are typically required to automate gate valves, however long stroke pneumatic and electro-hydraulic actuators are also available.



Gate Valve

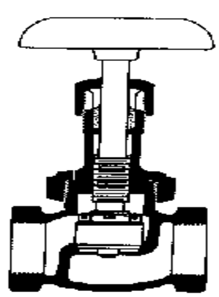
Recommended Uses:
1. Fully open/closed, non-throttling
2. Infrequent operation
3. Minimal fluid trapping in line

Applications: Oil, gas, air, slurries, heavy liquids, steam, non-condensing gases, and corrosive liquids

Advantages:
1. High capacity
2. Tight shutoff
3. Low cost
4. Little resistance to flow

Disadvantages:
1. Poor control
2. Cavitate at low pressure drops
3. Cannot be used for throttling

- **Globe Valves** have a conical plug, which reciprocates into and out of the valve port. Globe valves are ideal for shutoff as well as throttling service in high pressure drop and high temperature applications. Available in globe, angle, and y-pattern designs. Manual operation is accomplished through a multi-turn hand wheel assembly. Multiturn electric actuators are typically required to automate globe valves, however linear stroke pneumatic and electro-hydraulic actuators are also available.



Globe Valve

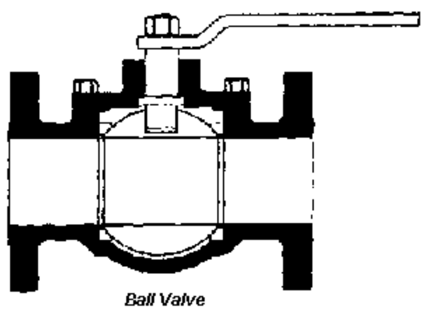
Recommended Uses:
1. Throttling service/flow regulation
2. Frequent operation

Applications: Liquids, vapors, gases, corrosive substances, slurries

Advantages:
1. Efficient throttling
2. Accurate flow control
3. Available in multiple ports

Disadvantages:
1. High pressure drop
2. More expensive than other valves

- **Ball Valves** were a welcomed relief to the process industry. They provide tight shutoff and high capacity with just a quarter-turn to operate. Ball valves are now more common in 1/4"-6" sizes. Ball valves can be easily actuated with pneumatic and electric actuators.



Ball Valve

Recommended Uses:
1. Fully open/closed, limited-throttling
2. Higher temperature fluids

Applications: Most liquids, high temperatures, slurries

Advantages:
1. Low cost
2. High capacity
3. Low leakage and maint.
4. Tight sealing with low torque

Disadvantages:
1. Poor throttling characteristics
2. Prone to cavitation

Butterfly valves are commonly used as control valves in applications where the pressure drops required of the valves are relatively low. Butterfly valves can be used in applications as either shutoff valves (on/off service) or as throttling valves (for flow or pressure control). As shutoff valves, butterfly valves offer excellent performance within the range of their pressure rating.

Typical uses would include isolation of equipment, fill/drain systems, and bypass systems and other like applications where the only criterion for control of the flow/pressure is that it be on or off. Although butterfly valves have only a limited ability to control pressure or flow, they have been widely used as control valves because of the economics involved. The control capabilities of a butterfly **valve** can also be significantly improved by coupling it with an operator and electronic control package.

Butterfly Valve

Recommended Uses:
1. Fully open/closed or throttling services
2. Frequent operation
3. Minimal fluid trapping in line

Applications: Liquids, gases, slurries, liquids with suspended solids

Advantages:
1. Low cost and maint.
2. High capacity
3. Good flow control
4. Low pressure drop

Disadvantages:
1. High torque required for control
2. Prone to cavitation at lower flows

3 COMPRESSED AIR PIPING

3.1 Introduction

The purpose of the compressed air piping system is to deliver compressed air to the points of usage. The compressed air needs to be delivered with enough volume, appropriate quality, and pressure to properly power the components that use the compressed air. Compressed air is costly to manufacture. A poorly designed compressed air system can increase energy costs, promote equipment failure, reduce production efficiencies, and increase maintenance requirements. It is generally considered true that any additional costs spent improving the compressed air piping system will pay for them many times over the life of the system.

3.2 Piping materials

Common piping materials used in a compressed air system include copper, aluminum, stainless steel and carbon steel. Compressed air piping systems that are 2" or smaller utilize copper, aluminum or stainless steel. Pipe and fitting connections are typically threaded. Piping systems that are 4" or larger utilize carbon or stainless steel with flanged pipe and fittings. Plastic piping may be used on compressed air systems, however caution must used since many plastic materials are not compatible with all compressor lubricants. Ultraviolet light (sun light) may also reduce the useful service life of some plastic materials. Installation must follow the manufacturer's instructions.

Corrosion-resistant piping should be used with any compressed air piping system using oil-free compressors. A non-lubricated system will experience corrosion from the moisture in the warm air, contaminating products and control systems, if this type of piping is not used.

It is always better to oversize the compressed air piping system you choose to install. This reduces pressure drop, which will pay for itself, and it allows for expansion of the system.

3.3 Compressor Discharge Piping

The discharge piping from the compressor should be at least as large as compressor discharge connection and it should run directly to the after cooler. Discharge piping from a compressor without an integral after cooler can have very high temperatures. The pipe that is installed here must be able to handle these temperatures. The high temperatures can also cause thermal expansion of the pipe, which can add stress to the pipe. Check the compressor manufacturer's recommendations on discharge piping. Install a liquid filled pressure gauge, a thermometer, and a thermowell in the discharge airline before the aftercooler. Proper support and/or flexible discharge pipe can eliminate strain.

1. The main header pipe in the system should be sloped downward in the direction of the compressed air flow. A general rule of thumb is 1" per 10 feet of pipe. The reason for the slope is to direct the condensation to a low point in the compressed air piping system where it can be collected and removed.
2. Make sure that the piping following the after cooler slopes downward into the bottom connection of the air receiver. This helps with the condensate drainage, as well as if the water-cooled after cooler develops a water leak internally. It would drain toward the receiver and not the compressor.
3. Normally, the velocity of compressed air should not be allowed to exceed 6 m/s; lower velocities are recommended for long lines. Higher air velocities (up to 20 m/s) are acceptable where the distribution pipe-work does not exceed 8 meters in length. This would be the case where dedicated compressors are installed near to an associated large end user.

4. The air distribution should be designed with liberal pipe sizes so that the frictional pressure losses are very low; larger pipe sizes also help in facilitating system expansion at a later stage without changing header sizes or laying parallel headers.

3.4 Pressure Drop

Pressure drop in a compressed air system is a critical factor. Pressure drop is caused by friction of the compressed air flowing against the inside of the pipe and through valves, tees, elbows and other components that make up a complete compressed air piping system. Pressure drop can be affected by pipe size, type of pipes used, the number and type of valves, couplings, and bends in the system. Each header or main should be furnished with outlets as close as possible to the point of application. This avoids significant pressure drops through the hose and allows shorter hose lengths to be used. To avoid carryover of condensed moisture to tools, outlets should be taken from the top of the pipeline. Larger pipe sizes, shorter pipe and hose lengths, smooth wall pipe, long radius swept tees, and long radius elbows all help reduce pressure drop within a compressed air piping system.

The discharge pressure of the compressor is determined by the maximum pressure loss plus operating pressure value so that air is delivered at right pressure to the farthest equipment. For example, a 90 psig air grinder installed in the farthest drop from the compressor may require 92 psig in the branch line 93 psig in the sub-header and 94 psig at the main header. With a 6 psi drop in the filter/dryer, the discharge pressure at the after cooler should be 100 psig.

The following nomogram can be used to estimate pressure drop in a compressed air system. Draw a straight line starting at pipe internal diameter and through flow (m/s) to be extended to the reference line. From this point draw another line to meet the air pressure (bar) line. The point of intersection of this line with the pressure drop line gives the pressure drop in mbar/m.

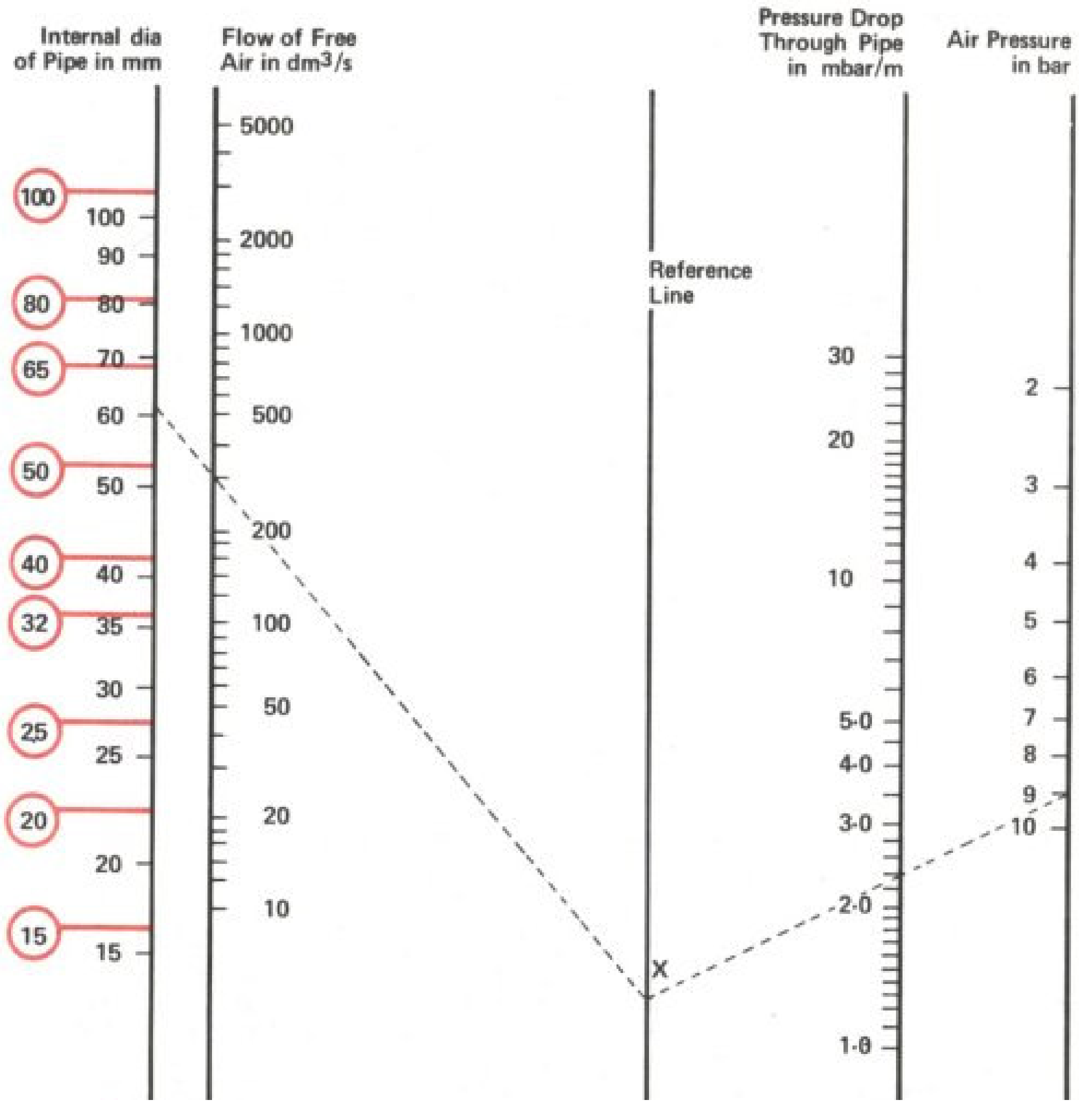


Figure 3-1: Pressure drop calculations

3.5 Piping system Design

There are two basic systems for distribution system.

1. A single line from the supply to the point(s) of usage, also known as radial system

2. Ring main system, where supply to the end use is taken from a closed loop header. The loop design allows airflow in two directions to a point of use. This can cut the overall pipe length to a point in half that reduces pressure drop. It also means that a large volume user of compressed air in a system may not starve users downstream since they can draw air from another direction. In many cases a balance line is also recommended which provides another source of air. Reducing the velocity of the airflow through the compressed air piping system is another benefit of the loop design. This reduces the velocity, which reduces the friction against the pipe walls and reduces pressure drop.

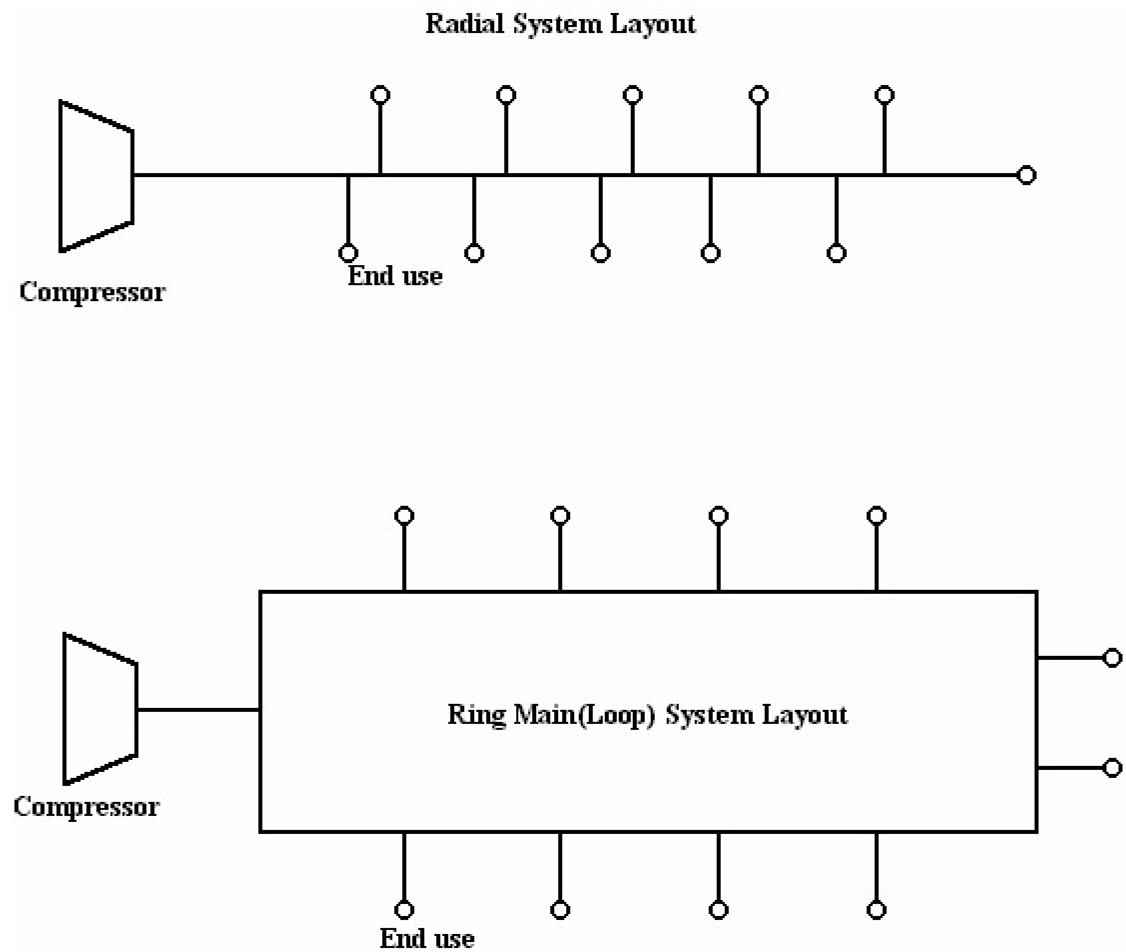


Figure 3-2: Types o piping layout

3.6 Compressed Air leakage

Leaks can be a significant source of wasted energy in an industrial compressed air system and may be costing you much more than you think. Audits typically find that leaks can be responsible for between 20-50% of a compressor's output making them the largest single waste of energy. In addition to being a source of wasted energy, leaks can also contribute to other operating losses:

- Leaks cause a drop in system pressure. This can decrease the efficiency of air tools and adversely affect production
- Leaks can force the equipment to cycle more frequently, shortening the life of almost all system equipment (including the compressor package itself)
- Leaks can increase running time that can lead to additional maintenance requirements and increased unscheduled downtime
- Leaks can lead to adding unnecessary compressor capacity

Observing the average compressor loading and unloading time, when there is no legitimate use of compressed air on the shop floor, can estimate the leakage level. In continuous process plants, this test can be conducted during the shutdown or during unexpected production stoppages.

$$\text{Air Leakage} = Q \times \frac{\text{On load time}}{\text{On load time} + \text{Off load time}}$$

Where Q = compressor capacity

3.7 Leakage reduction

Leakage tests can be conducted easily, but identifying leakage points and plugging them is laborious work; obvious leakage points can be identified from audible sound; for small leakage, ultrasonic leakage detectors can be used; soap solution can also be used to detect small leakage in accessible lines.

When looking for leaks you should investigate the following:

CONDENSATE TRAPS -Check if automatic traps are operating correctly and avoid bypassing.

PIPE WORK - Ageing or corroded pipe work.

FITTINGS AND FLANGES - Check joints and supports are adequate. Check for twisting.

MANIFOLDS - Check for worn connectors and poorly jointed pipe work.

FLEXIBLE HOSES - Check that the hose is moving freely and clear of abrasive surfaces. Check for deterioration and that the hose has a suitable coating for the environment e.g. oily conditions. Is the hose damaged due to being too long or too short?

INSTRUMENTATION - Check connections to pneumatic instruments such as regulators, lubricators, valve blocks and sensors. Check for worn diaphragms.

PNEUMATIC CYLINDERS Check for worn internal air seals.

FILTERS Check drainage points and contaminated bowls.
TOOLS Check hose connections and speed control valve. Check air tools are always switched off when not in use.

<u>The following points can help reduce compressed air leakage:</u>

- Reduce the line pressure to the minimum acceptable; this can be done by reducing the discharge pressure settings or by use of pressure regulators on major branch lines.
- Selection of good quality pipe fittings.
- Provide welded joints in place of threaded joints.
- Sealing of unused branch lines or tapings.
- Provide ball valves (for isolation) at the main branches at accessible points, so that these can be closed when air is not required in the entire section. Similarly, ball valves may be provided at all end use points for firm closure when pneumatic equipment is not in use.
- Install flow meters on major lines; abnormal increase in airflow may be an indicator of increased leakage or wastage.
- Avoid installation of underground pipelines; pipelines should be overhead or in trenches (which can be opened for inspection). Corroded underground lines can be a major source of leakage.

The following table 3.1 shows cost of compressed air leakage from holes at different pressures. It may be noted that, at 7 bar (100 psig), about 100 cfm air leakage is equivalent to a power loss of 17 kW i.e. about Rs.6.12 lakhs per annum.

Table 3-1: Cost of Compressed Air Leakage

Orifice Diameter	Air leakage Scfm	Power wasted KW	Cost of Wastage (for 8000 hrs/year) (@ Rs. 4.50/kWh
At 3 bar (45 psig) pressure			
1/32”	0.845	0.109	3924
1/16”	3.38	0.439	15804
1/8”	13.5	1.755	63180
¼”	54.1	7.03	253080
At 4 bar (60 psig) pressure			
1/32”	1.06	0.018	6487
1/16”	4.23	0.719	25887
1/8”	16.9	3.23	103428
¼”	164.6	14.57	395352
At 5.5 bar (80 psig) pressure			
1/32”	1.34	0.228	8201
1/16”	5.36	0.911	32803
1/8”	21.4	3.64	130968
¼”	85.7	14.57	524484
At 7 bar (100 psig) pressure			
1/32”	1.62	0.275	9915
1/16”	6.49	1.10	39719
1/8”	26	4.42	159120
¼”	104	17.68	636480

4 STEAM DISTRIBUTION

4.1 Introduction

The objective of the steam distribution system is to supply steam at the correct pressure to the point of use. It follows; therefore, that pressure drop through the distribution system is an important feature. One of the most important decisions in the design of a steam system is the selection of the generating, distribution, and utilization pressures. Considering investment cost, energy efficiency, and control stability, the pressure shall be held to the minimum values above atmospheric pressure that are practical to accomplish the required heating task, unless detailed economic analysis indicates advantages in higher pressure generation and distribution.

The piping system distributes the steam, returns the condensate, and removes air and non-condensable gases. In steam heating systems, it is important that the piping system distribute steam, not only at full design load, but also at partial loads and excess loads that can occur on system warm-up. When the system is warming up, the load on the steam mains and returns can exceed the maximum operating load for the coldest design day, even in moderate weather. This load comes from raising the temperature of the piping to the steam temperature and the building to the indoor design temperature.

4.2 Energy Considerations

Steam and condensate piping system have a great impact on energy usage. Proper sizing of system components such as traps, control valves, and pipes has a tremendous effect on the efficiencies of the system.

Condensate is a by-product of a steam system and must always be removed from the system as soon as it accumulates, because steam moves rapidly in mains and supply piping, and if condensate accumulates to the point where the steam can push a slug of it, serious damage can occur from the resulting water hammer. Pipe insulation also has a tremendous effect on system energy efficiency. All steam and condensate piping should be insulated. It may also be economically wise to save the sensible heat of the condensate for boiler water make-up systems operational efficiency

Oversized pipe work means:

- Pipes, valves, fittings, etc. will be more expensive than necessary.
- Higher installation costs will be incurred, including support work, insulation, etc.
- For steam pipes a greater volume of condensate will be formed due to the greater heat loss. This, in turn, means that either:
- More steam trapping is required, or wet steam is delivered to the point of use.

In a particular example:

- The cost of installing 80 mm steam pipe work was found to be 44% higher than the cost of 50 mm pipe work, which would have had adequate capacity.

- The heat lost by the insulated pipe work was some 21% higher from the 80 mm pipeline than it would have been from the 50 mm pipe work. Any non-insulated parts of the 80 mm pipe would lose 50% more heat than the 50 mm pipe, due to the extra heat transfer surface area.

Undersized pipe work means:

- A lower pressure may only be available at the point of use. This may hinder equipment performance due to only lower pressure steam being available.

- There is a risk of steam starvation.

- There is a greater risk of erosion, water hammer and noise due to the inherent increase in steam velocity.

The allowance for pipe fittings:

The length of travel from the boiler to the unit heater is known, but an allowance must be included for the additional frictional resistance of the fittings. This is generally expressed in terms of 'equivalent pipe length'. If the size of the pipe is known, the resistance of the fittings can be calculated. As the pipe size is not yet known in this example, an addition to the equivalent length can be used based on experience.

- If the pipe is less than 50 metres long, add an allowance for fittings of 5%.

- If the pipe is over 100 metres long and is a fairly straight run with few fittings, an allowance for fittings of 10% would be made.

- A similar pipe length, but with more fittings, would increase the allowance towards 20%.

4.3 Selection of pipe size

There are numerous graphs, tables and slide rules available for relating steam pipe sizes to flow rates and pressure drops.

To begin the process of determining required pipe size, it is usual to assume a velocity of flow. For saturated steam from a boiler, 20 - 30 m/s is accepted general practice for short pipe runs. For major lengths of distribution pipe work, pressure drop becomes the major consideration and velocities may be slightly less. With dry steam, velocities of 40 metres/sec can be contemplated -but remember that many steam meters suffer wear and tear under such conditions. There is also a risk of noise from pipes.

Draw a horizontal line from the saturation temperature line (Point **A**) on the pressure scale to the steam mass flow rate (Point **B**).

- From point **B**, draw a vertical line to the steam velocity of 25 m/s (Point **C**). From point **C**, draw a horizontal line across the pipe diameter scale (Point **D**).

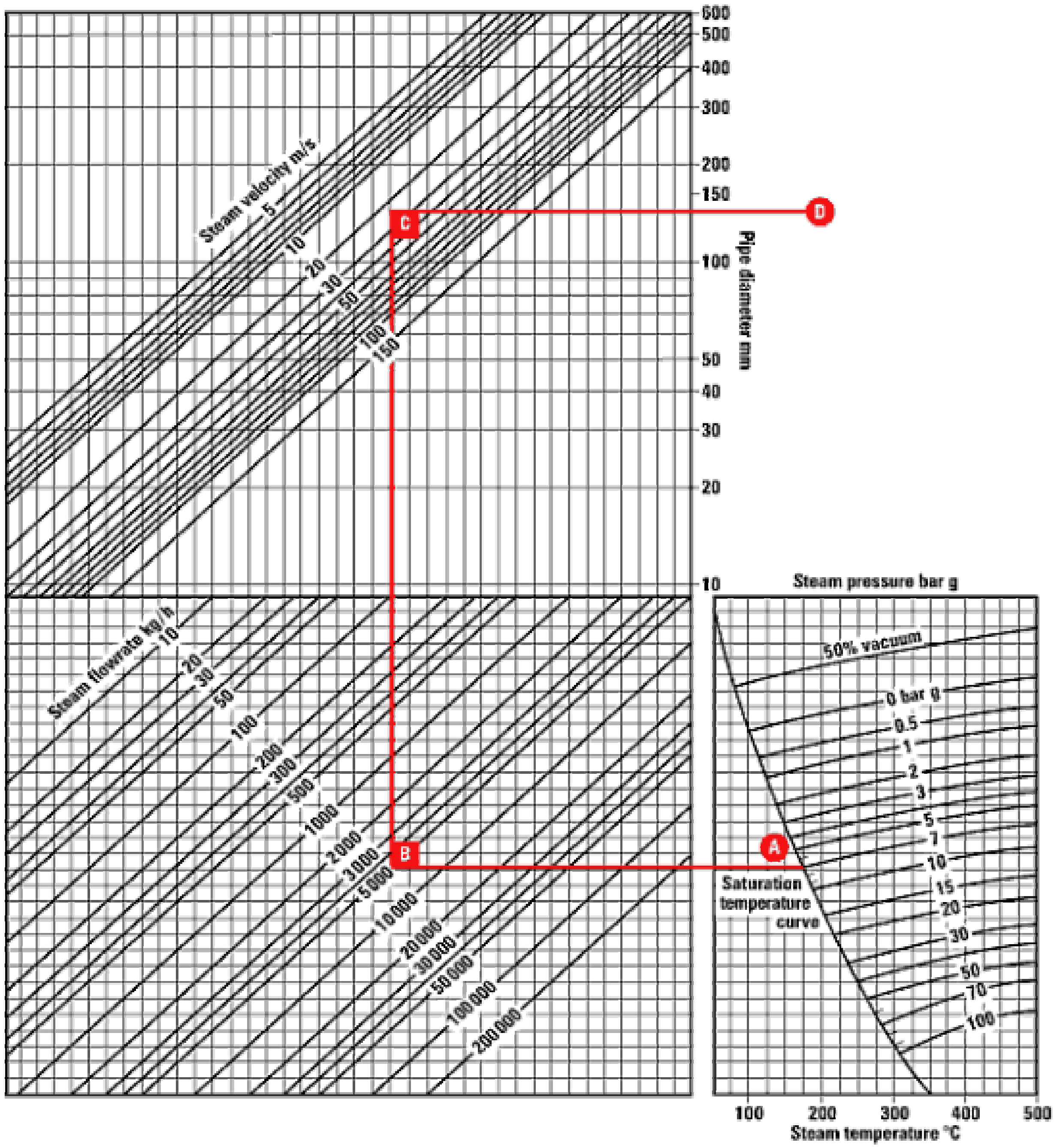


Figure 4-1: Steam pipe sizing

The following table also summarises the recommended pipe sizes for steam at various pressure and mass flow rate.

Table 4-1: Recommended pipe sizes for steam

Capacity (kg/hour)		Pipe Size (mm)													
Pressure (bar)	Steam Speed (m/s)	15	20	25	32	40	50	65	80	100	125	150	200	250	300
0.4	15	7	14	24	37	52	99	145	213	394	648	917	1606	2590	368
	25	10	25	40	62	92	162	265	384	675	972	1457	2806	4101	5936
	40	17	35	64	102	142	265	403	576	1037	1670	2303	4318	6909	9500
0.7	15	7	16	25	40	59	109	166	250	431	680	1006	1708	2791	3852
	25	12	25	45	72	100	182	287	430	716	1145	1575	2816	4629	6204
	40	18	37	68	106	167	298	428	630	1108	1715	2417	4532	7251	10323
1	15	8	17	29	43	65	112	182	260	470	694	1020	1864	2814	4045
	25	12	26	48	72	100	193	300	445	730	1160	1660	3099	4869	6751
	40	19	39	71	112	172	311	465	640	1150	1800	2500	4815	7333	10370
2	15	12	25	45	70	100	182	280	410	715	1125	1580	2814	4545	6277
	25	19	43	70	112	162	195	428	656	1215	1755	2520	4815	7425	10575
	40	30	64	115	178	275	475	745	1010	1895	2925	4175	7678	11997	16796
3	15	16	37	60	93	127	245	385	535	925	1505	2040	3983	6217	8743
	25	26	56	100	152	225	425	632	910	1580	2480	3440	6779	10269	14316
	40	41	87	157	250	357	595	1025	1460	2540	4050	5940	10479	16470	22950
4	15	19	42	70	108	156	281	432	635	1166	1685	2460	4618	7121	10358
	25	30	63	115	180	270	450	742	1080	1980	2925	4225	7866	12225	17304
	40	49	116	197	295	456	796	1247	1825	3120	4940	7050	12661	1963	27816
5	15	22	49	87	128	187	352	526	770	1295	2105	2835	5548	8586	11947
	25	36	81	135	211	308	548	885	1265	2110	3540	5150	8865	14268	20051
	40	59	131	225	338	495	855	1350	1890	3510	5400	7870	13761	23205	32244
6	15	26	59	105	153	225	425	632	925	1555	2525	3400	6654	10297	14328
	25	43	97	162	253	370	658	1065	1520	2530	4250	6175	10629	17108	24042
	40	71	157	270	405	595	1025	1620	2270	4210	6475	9445	16515	27849	38697
7	15	29	63	110	165	260	445	705	952	1815	2765	3990	7390	12015	16096
	25	49	114	190	288	450	785	1205	1750	3025	4815	6900	12288	19377	27080
	40	76	177	303	455	690	1210	1865	2520	4585	7560	10880	19141	30978	43470
8	15	32	70	126	190	285	475	800	1125	1990	3025	4540	8042	12625	17728
	25	54	122	205	320	465	810	1260	1870	3240	5220	7120	13140	21600	33210
	40	84	192	327	510	730	1370	2065	3120	5135	8395	12470	21247	33669	46858
10	15	41	95	155	250	372	626	1012	1465	2495	3995	5860	9994	16172	22713
	25	66	145	257	405	562	990	1530	2205	3825	6295	8995	15966	25860	35890
	40	104	216	408	615	910	1635	2545	3600	6230	9880	14390	26621	41011	57560
14	15	50	121	205	310	465	810	1270	1870	3220	5215	7390	12921	20538	29016
	25	85	195	331	520	740	1375	2080	3120	5200	8500	12560	21720	34139	47128
	40	126	305	555	825	1210	2195	3425	4735	8510	13050	18630	35548	54883	76534

4.4 Piping Installation

1. All underground steam systems shall be installed a minimum of 10 feet from plastic piping and chilled water systems. All plastic underground piping must be kept at a 10 foot distance from steam/condensate lines.
2. Install piping free of sags or bends and with ample space between piping to permit proper insulation applications.
3. Install steam supply piping at a minimum, uniform grade of 1/4 inch in 10 feet downward in the direction of flow.
4. Install condensate return piping sloped downward in the direction of steam supply. Provide condensate return pump at the building to discharge condensate back to the Campus collection system.
5. Install drip legs at intervals not exceeding 200 feet where pipe is pitched down in the direction of the steam flow. Size drip legs at vertical risers full size and extend beyond the rise. Size drip legs at other locations same diameter as the main. Provide an 18-inch drip leg for steam mains smaller than 6 inches. In steam mains 6 inches and larger, provide drip legs sized 2 pipe sizes smaller than the main, but not less than 4 inches.
6. Drip legs, dirt pockets, and strainer blow downs shall be equipped with gate valves to allow removal of dirt and scale.
7. Install steam traps close to drip legs.

Following are some of the hard facts regarding steam losses in various components of Steam distribution system.

Leakage:

Steam Pressure	Steam kg/year	FO kg/year	Rs./Year
Hole Dia of 1/10 Inch			
7 kg/cm$^{2)}$	50,880	4,070.4	42,780
21 kg/cm^{2}	1,20,000	9,600	1,00,896
Hole Dia of 1/8 Inch			
7 kg/cm$^{2)}$	2,03,636	16,291	1,71,217
21 kg/cm^{2}	4,80,000	38,400	4,03,584
Hole Dia of 3/16 Inch			
7 kg/cm$^{2)}$	4,58,182	36,655	3,85,244
21 kg/cm^{2}	10,80,000	86,400	6,84,874
Hole Dia of 1/4 Inch			
7 kg/cm$^{2)}$	8,14,545	65,164	9,08,064
21 kg/cm^{2}	19,20,000	1,53,600	16,14,336

Basis :	F. O. Price	=	Rs. 10.5 per kg
	Operating hrs	=	8,000 hrs per year
	Steam Ratio	=	12.5 kg of steam per kg FO

5 WATER DISTRIBUTION SYSTEM

5.1 Recommended Velocities

As a rule of thumb, the following velocities are used in design of piping and pumping systems for water transport:

Table 5-1: Recommended velocities

Pipe Dimension		Velocity
Inches	mm	m/s
1	25	1
2	50	1.1
3	75	1.15
4	100	1.25
6	150	1.5
8	200	1.75
10	250	2
12	300	2.65

If you want to pump 14.5 m^3/h of water for a cooling application where pipe length is 100 metres, the following table shows why you should be choosing a 3" pipe instead of a 2" pipe.

Table 5-2:Calculation of System Head Requirement for a Cooling Application (for different pipe sizes)

Description	units	Header diameter, inches		
		2.0	3.0	6.0
Water flow required	m^3/hr	14.5	14.5	14.5
Water velocity	m/s	2.1	0.9	0.2
Size of pipe line (diameter)	mm	50	75	150
Pressure drop in pipe line/metre	m	0.1690	0.0235	0.0008
Length of cooling water pipe line	m	100.0	100.0	100.0
Equivalent pipe length for 10 nos. bends	m	15.0	22.5	45.0
Equivalent pipe length for 4 nos. valves	m	2.6	3.9	7.8
Total equivalent length of pipe	m	117.6	126.4	152.8
Total frictional head loss in pipes/fittings	m	19.9	3.0	0.1
Pressure drop across heat exchanger, assumed	m	5	5	5
Static head requirement, assumed	m	5	5	5
Total head required by the pump	m	29.9	13.0	10.1
Likely motor input power	kW	2.2	1.0	0.9

If a 2" pipe were used, the power consumption would have been more than double compared to the 3" pipe. Looking at the velocities, it should be noted that for smaller pipelines, lower design velocities are recommended. For a 12” pipe, the velocity can be 2.6 m/s without any or notable energy penalty, but for a 2” to 6” line this can be very lossy.

To avoid pressure losses in these systems:

1. First, decide the flow
2. Calculate the pressure drops for different pipe sizes and estimate total head and power requirement
3. Finally, select the pump.

5.2 Recommended water flow velocity on suction side of pump

Capacity problems, cavitation and high power consumption in a pump, is often the result of the conditions on the suction side. In general - a rule of thumb - is to keep the suction fluid flow speed below the following values:

Table 5-3: Recommended suction velocities

Pipe bore		Water velocity	
inches	mm	m/s	ft/s
1	25	0.5	1.5
2	50	0.5	1.6
3	75	0.5	1.7
4	100	0.55	1.8
6	150	0.6	2
8	200	0.75	2.5
10	250	0.9	3
12	300	1.4	4.5

6 THERMAL INSULATION

6.1 Introduction

There are many reasons for insulating a pipeline, most important being the energy cost of not insulating the pipe. Adequate thermal insulation is essential for preventing both heat loss from hot surfaces of ovens/furnaces/piping and heat gain in refrigeration systems. Inadequate thickness of insulation or deterioration of existing insulation can have a significant impact on the energy consumption. The material of insulation is also important to achieve low thermal conductivity and also low thermal inertia. Development of superior insulating materials and their availability at reasonable prices have made retrofitting or re-insulation a very attractive energy saving option.

The simplest method of analysing whether you should use 1" or 2" or 3" insulation is by comparing the cost of energy losses with the cost of insulating the pipe. The insulation thickness for which the total cost is minimum is termed as economic thickness. Refer fig 6.1. The curve representing the total cost reduces initially and after reaching the economic thickness corresponding to the minimum cost, it increases.

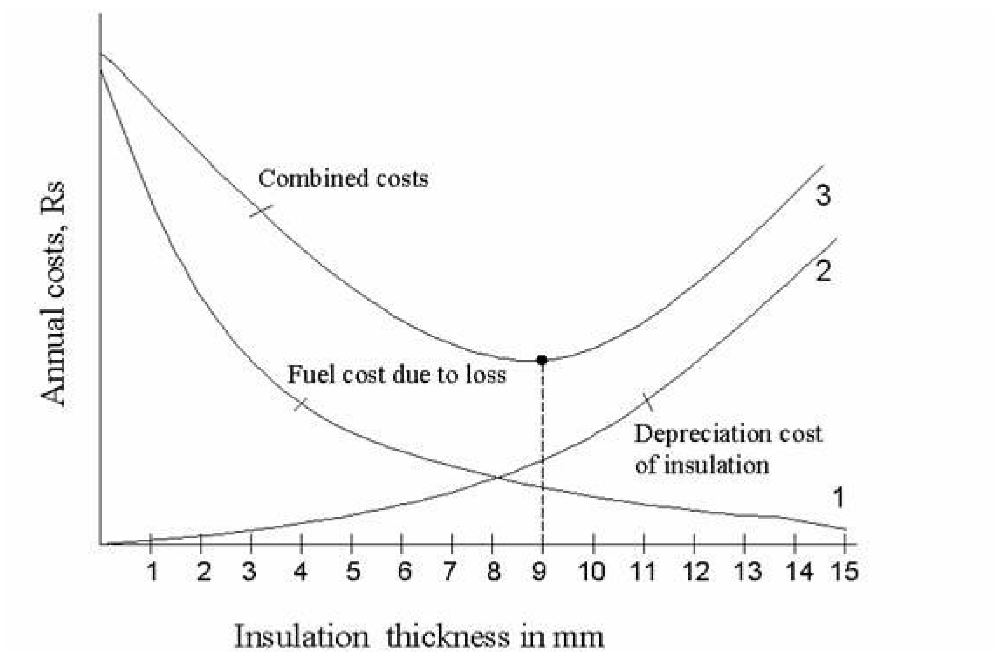


Figure 6-1: Economic insulation thickness

However, in plants, there are some limitations for using the results of economic thickness calculations. Due to space limitations, it is sometimes not possible to accommodate larger diameter of insulated pipes.

A detailed calculation on economic thickness is given in section 6.5.

6.2 Heat Losses from Pipe surfaces

Heat loss from 1/2" to 12" steel pipes at various temperature differences between pipe and air can be found in the table below.

Table 6-1: Heat loss from Fluid inside Pipe (W/m)

Nominal bore		Temperature Difference (^oC)											
(mm)	(inch)	50	60	75	100	110	125	140	150	165	195	225	280
15	1/2	30	40	60	90	130	155	180	205	235	280	375	575
20	3/4	35	50	70	110	160	190	220	255	290	370	465	660
25	1	40	60	90	130	200	235	275	305	355	455	565	815
32	1 1/4	50	70	110	160	240	290	330	375	435	555	700	1000
40	1 1/2	55	80	120	180	270	320	375	420	485	625	790	1120
50	2	65	95	150	220	330	395	465	520	600	770	975	1390
65	2 1/2	80	120	170	260	390	465	540	615	715	910	1150	1650
80	3	100	140	210	300	470	560	650	740	860	1090	1380	1980
100	4	120	170	260	380	5850	700	820	925	1065	1370	1740	2520
150	6	170	250	370	540	815	970	1130	1290	1470	1910	2430	3500
200	8	220	320	470	690	1040	1240	1440	1650	1900	2440	3100	4430
250	10	270	390	570	835	1250	1510	1750	1995	2300	2980	3780	5600
300	12	315	460	670	980	1470	1760	2060	2340	2690	3370	4430	6450

The heat loss value must be corrected by the correction factor for certain applications:

Application	Correction factor
Single pipe freely exposed	1.1
More than one pipe freely exposed	1.0
More than one pipe along the ceiling	0.65
Single pipe along skirting or riser	1.0
More than one pipe along skirting or riser	0.90
Single pipe along ceiling	0.75

6.3 Calculation of Insulation Thickness

The most basic model for insulation on a pipe is shown below. r_1 show the outside radius of the pipe r_2 shows the radius of the Pipe+ insulation.

Heat loss from a surface is expressed as

H = h X A x (T_h-T_a) ---(4)

Where
h = Heat transfer coefficient, W/m^2-K
H = Heat loss, Watts
T_a = Average ambient temperature, K
T_s = Desired/actual insulation surface temperature, ºC

T_h = Hot surface temperature (for hot fluid piping), ºC & Cold surface temperature for cold fluids piping)

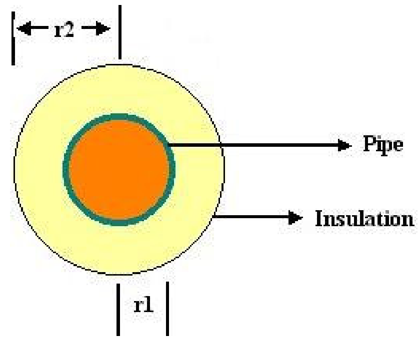


Figure 6-2: Insulated pipe section

For horizontal pipes, heat transfer coefficient can be calculated by:

$h = (A + 0.005\,(T_h - T_a)) \times 10$ W/m^2-K

For vertical pipes,

$h = (B + 0.009\,(T_h - T_a)) \times 10$ W/m^2-K

Using the coefficients A, B as given below.

Table 6-2: Coefficients A, B for estimating 'h' (in W/m^2-K)

Surface	ε	A	B
Aluminium , bright rolled	0.05	0.25	0.27
Aluminium, oxidized	0.13	0.31	0.33
Steel	0.15	0.32	0.34
Galvanised sheet metal, dusty	0.44	0.53	0.55
Non metallic surfaces	0.95	0.85	0.87

$$T_m = \frac{(T_h + T_s)}{2}$$

k = Thermal conductivity of insulation at mean temperature of T_m, W/m-C
t_k = Thickness of insulation, mm
r_1 = Actual outer radius of pipe, mm
$r_2 = (r_1 + t_k)$

R_s = Surface thermal resistance = $\frac{1}{h}$ ºC-m^2/W

R_l = Thermal resistance of insulation = $\frac{t_k}{k}$ ºC-m^2/W

The heat flow from the pipe surface and the ambient can be expressed as follows

H = Heat flow, Watts

$$= \frac{(T_h - T_a)}{(R_l + R_s)} = \frac{(T_s - T_a)}{R_s} \text{ ---(5)}$$

From the above equation, and for a desired T_s, R_l can be calculated. From R_l and known value of thermal conductivity k, thickness of insulation can be calculated.

Equivalent thickness of insulation for pipe, E_{tk}.= $(r_1 + t_k) \times \ln\left(\frac{(r_1 + t_k)}{r_1}\right)$

6.4 Insulation material

Insulation materials are classified into organic and inorganic types. Organic insulations are based on hydrocarbon polymers, which can be expanded to obtain high void structures. Examples are thermocol (Expanded Polystyrene) and Poly Urethane Form (PUF). Inorganic insulation is based on Siliceous/Aluminous/Calcium materials in fibrous, granular or powder forms. Examples are Mineral wool, Calcium silicate etc.

Properties of common insulating materials are as under:

Calcium Silicate: Used in industrial process plant piping where high service temperature and compressive strength are needed. Temperature ranges varies from 40 C to 950 C.

Glass mineral wool: These are available in flexible forms, rigid slabs and preformed pipe work sections. Good for thermal and acoustic insulation for heating and chilling system pipelines. Temperature range of application is –10 to 500 C

Thermocol: These are mainly used as cold insulation for piping and cold storage construction.

Expanded nitrile rubber: This is a flexible material that forms a closed cell integral vapour barrier. Originally developed for condensation control in refrigeration pipe work and chilled water lines; now-a-days also used for ducting insulation for air conditioning.

Rock mineral wool: This is available in a range of forms from light weight rolled products to heavy rigid slabs including preformed pipe sections. In addition to good thermal insulation properties, it can also provide acoustic insulation and is fire retardant.

The *thermal conductivity* of a material is the heat loss per unit area per unit insulation thickness per unit temperature difference. The unit of measurement is W-m^2/m°C or W-m/°C. The thermal conductivity of materials increases with temperature. So thermal conductivity is always specified at the mean temperature (mean of hot and cold face temperatures) of the insulation material.

Thermal conductivities of typical hot and cold insulation materials are given below.

Table 6-3: Thermal conductivity of hot insulation

Mean Temperature °C	Calcium Silicate	Resin bonded Mineral wool	Ceramic Fiber Blankets
100	-	0.04	-
200	0.07	0.06	0.06
300	0.08	0.08	0.07
400	0.08	0.11	0.09
700	-	-	0.17
1000	-	-	0.26

Specific heat(kJ/kg/°C)	0.96	0.921	1.07
	(at 40°C)	(at 20°C)	(at 980°C)
Service temp, (°C).	950	700	1425
Density kg/m^3	260	48 to144	64 to 128

Specific Thermal Conductivity of Materials for Cold Insulation

MATERIALS	Thermal Conductivity W/m-°C
Mineral Or Glass Fiber Blanket	0.039
Board or Slab	
Cellular Glass	0.058
Cork Board	0.043
Glass Fiber	0.036
Expanded Polystyrene (smooth) - Thermocole	0.029
Expanded Polystyrene (Cut Cell) - Thermocole	0.036
Expanded Polyurethane	0.017
Phenotherm (Trade Name)	0.018
Loose Fill	
Paper or Wood Pulp	0.039
Sawdust or Shavings	0.065
Minerals Wool (Rock, Glass, Slag)	0.039
Wood Fiber (Soft)	0.043

6.5 Recommended values of cold and hot insulation

Refer table 6.3. Insulation thickness is given in mm for refrigeration systems with fluid temperatures varying from 10 to –20° C is given below. The emissivity of surface (typically cement, gypsum etc) is high at about 0.9. Ambient temperature is 25° C and 80% RH.

Table 6-3: Insulation thickness for refrigeration systems

Nominal Dia of pipe	Temperature of contents														
	10			5			0			-10			-20		
	Thermal conductivity at mean temperature														
	0.02	0.03	0.04	0.02	0.03	0.04	0.02	0.03	0.04	0.02	0.03	0.04	0.02	0.03	0.04
1”	10	14	17	14	18	23	17	23	29	23	32	41	29	41	53
1.5”	11	16	20	15	21	27	19	27	33	26	37	47	33	47	62
2”	13	18	23	17	25	31	22	31	40	30	44	57	38	57	77
4”	14	20	27	20	30	38	25	38	51	37	57	73	49	92	92
6”	15	23	31	22	35	45	30	45	57	43	62	79	55	99	99
10”	17	26	34	25	37	48	33	48	61	47	67	86	60	110	110

Recommended thickness of insulation for high temperature systems is given in Table 6.4.

Table 6-4Recommended Thickness of Insulation (inches)

Nominal Pipe Size NPS (inches)	Temperature Range (^{o}C)					
	Below 200	200– 300	300-370	370–500	500 – 600	600 – 650
< 1	1	1	1.5	2	2	2.5
1.5	1	1.5	1.5	2	2	2.5
2	1	1.5	1.5	2	2.5	3
3	1	1.5	1.5	2.5	2.5	3
4	1	1.5	1.5	2.5	2.5	3.5
6	1	1.5	1.5	2.5	3	3.5
8	1.5	1.5	2	2.5	3	3.5
10	1.5	1.5	2	2.5	3	4
12	1.5	2	2	2.5	3	4
14	1.5	2	2	3	3	4
16	2	2	2	3	3.5	4
18	2	2	2	3	3.5	4
20	2	2	2	3	3.5	4
24	2	2	2	3	3.5	4

6.6 Economic thickness of insulation

To explain the concept of economic thickness of insulation, we will use an example. Consider an 8 bar steam pipeline of 6” dia having 50-meter length. We will evaluate the cost of energy losses when we use 1”, 2” and 3” insulation to find out the most economic thickness.

A step-by-step procedure is given below.

1. Establish the bare pipe surface temperature, by measurement.
2. Note the dimensions such as diameter, length & surface area of the pipe section under consideration.
3. Assume an average ambient temperature. Here, we have taken 30° C.
4. Since we are doing the calculations for commercially available insulation thickness, some trial and error calculations will be required for deciding the surface temperature after putting insulation. To begin with assume a value between 55 & 65° C, which is a safe, touch temperature.
5. Select an insulation material, with known thermal conductivity values in the mean insulation temperature range. Here the mean temperature is 111° C. and the value of k = 0.044 W/m2-°C for mineral wool.
6. Calculate surface heat transfer coefficients of bare and insulated surfaces, using equations discussed previously. Calculate the thermal resistance and thickness of insulation.
7. Select r2 such that the equivalent thickness of insulation of pipe equals to the insulation thickness estimated in step 6. From this value, calculate the radial thickness of pipe insulation = r2-r1

8. Adjust the desired surface temperature values so that the thickness of insulation is close to the standard value of 1" (25.4 mm).
9. Estimate the surface area of the pipe with different insulation thickness and calculate the total heat loss from the surfaces using heat transfer coefficient, temperature difference between pipe surface and ambient.
10. Estimate the cost of energy losses in the 3 scenarios. Calculate the Net Present Value of the future energy costs during an insulation life of typically 5 years.
11. Find out the total cost of putting insulation on the pipe (material + labor cost)
12. Calculate the total cost of energy costs and insulation for 3 situations.
13. Insulation thickness corresponding to the lowest total cost will be the economic thickness of insulation.

Table 6-5: Economic insulation thickness calculations

		Insulation thickness		
Description	Unit	1"	2"	3"
Length of pipe, L	m	50	50	50
Bare Pipe outer diameter, d1	mm	168	168	168
Bare pipe surface area, A	m^2	26.38	26.38	26.38
Ambient Temperature, Ta :	℃	30	30	30
Bare Pipe Wall Temperature, Th:	℃	160	160	160
Desired Wall Temperature With Insulation, Tc :	℃	62	48	43
Material of Insulation :		Mineral Wool		
Mean Temperature of Insulation, Tm = (Th+Tc)/2 :	℃	111	104	101.5
Sp.Conductivity of Insulation Material, k (from catalogue) :	W/m℃	0.044	0.042	0.04
Surface Emissivity of bare pipe:		0.95	0.95	0.95
Surface emissivity of insulation cladding(typically Al)		0.13	0.13	0.13

Calculations

Surface Heat Transfer Coefficient of Hot Bare Surface, h :(0.85+ 0.005 (Th – Ta)) x 10	W/m^2℃	15	15	15
Surface Heat Transfer Coefficient After Insulation, h' = (0.31+ 0.005 (Tc – Ta)) x 10	W/m^2℃	4.7	4	3.75
Thermal Resistance, R_{th} = (Th-Tc)/[h'x (Tc-Ta)] :	℃-m^2/W	0.7	1.6	2.4
Thickness of Insulation, t = k x R_{th} :(if surface was flat)	mm	28.7	65.3	96.0
r_1=outer diameter/2 =	mm	84	84	84
$t_{eq} = r_2 \times \ln(r_2/r_1)$ = (select r_2 so that t_{eq} = t)	mm	28.7	65.3	106.3
Outer radius of insulation , r_2=	mm	109.2	135.9	161.9
Thickness of insulation	mm	**25.2**	**51.9**	**77.9**
Insulated pipe Area , A :	m^2	34.29	42.66	50.85
Total Losses From Bare Surface, Q = h x A x (Th-Ta) :	kW	51.4	51.4	51.4
Total Loss From Insulated Surface, Q' = h' x A' x (Tc-Ta) :	kW	5.16	3.07	2.48
Power Saved by Providing Insulation, P = Q - Q' :	kW	46.3	48.4	49.0
Annual Working Hours, n :	Hrs	8000	8000	8000
Energy Saving After Providing Insulation, E = P x n :	kWh/year	370203	386892	391634

Economics

Steam cost,	Rs/kg	0.70	0.70	0.70
Heat Energy Cost, p :	Rs./kWh	1.11	1.11	1.11
Annual Monetary Saving, S = E x p :	Rs.	412708	431313	436599
Discount factor for calculating NPV of cost of energy loss	%	15%	15%	15%
Cost of insulation (material + labor)	Rs/m	450	700	1100
Total cost of insulation	Rs/m	22500	35000	55000
Annual Cost of energy loss	Rs/year	46000	27395	22109
NPV of annual cost of energy losses for 5 years	Rs	154198	91832	74112
Total cost (insulation & NPV of heat loss)	Rs	**176698**	**126832**	**129112**

Note that the total cost in lower when using 2” insulation, hence is the economic insulation thickness.

7 CASE STUDIES

7.1 Pressure drop reduction in water pumping

The Pharmaceutical plant had a 4" pipeline main header for distributing chilled water from the chilling plant to the end uses. The number of end uses of chilled water has increased over the years; however, the main header size remained the same at 4".

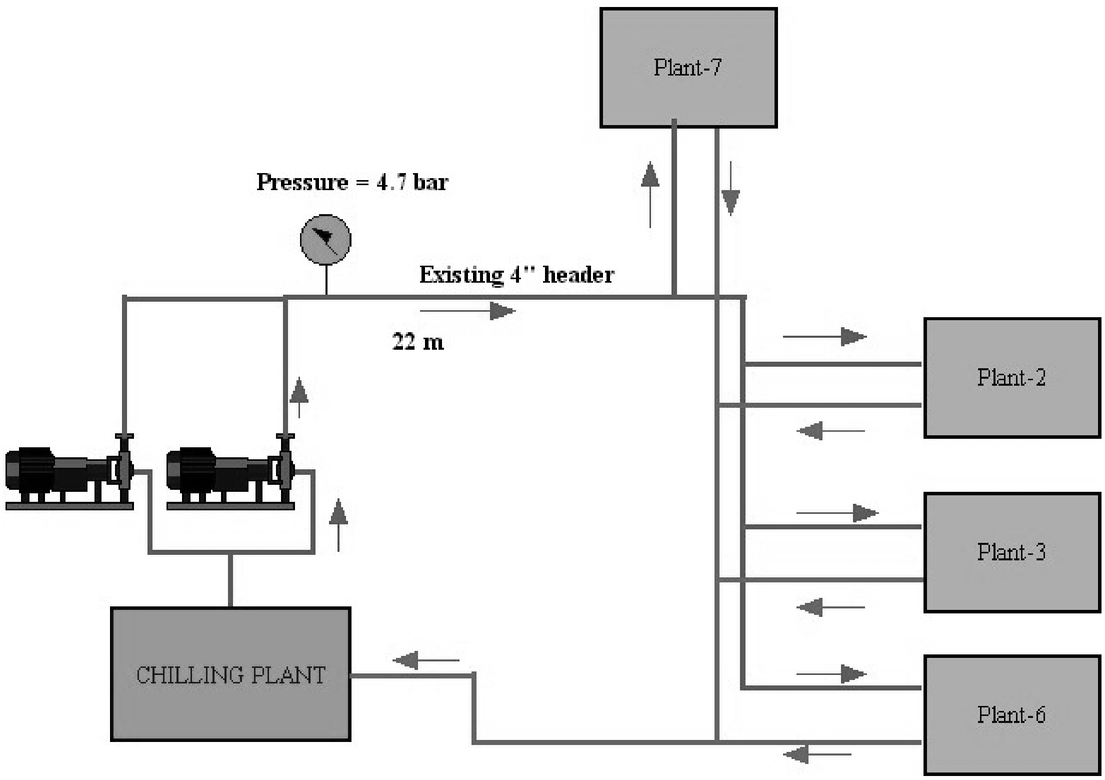


Figure 7-1: Chilled water system piping schematic

Flow measured was varying from 120 to 180 m3/h. It was observed that the line pressure at the main header at the inlet to plant-2 was 2.2 bars only when the pump discharge pressure was 4.7 bars. At plat-6, the line pressure was 2.0 bar. The pressure drop was about 2.5 bar!

It was clear that the pressure drop in the main header section having 22 meter length (refer figure above) was very high. Usually, a 4" line is used for carrying a maximum flow of 60 m3/hr and for very short distances, it can carry 80 m3/h. The pump power consumption was 35 kW.

Modifications:

An additional 4" line was laid parallel to the main header up to plant –7 supply point. The existing pressure drop of 2.5 bar reduced to 0.5 bar. Along with this, the existing pump impeller was trimmed properly so that the new discharge pressure was 3.0 bar. The power consumption after modification was 21.0 kW.

Power saving of 14 kW has resulted by this measure. Annual energy saving was 1,12,000 kWh. I.e. Rs 4.8 lakhs/annum. Investment for the piping modifications was Rs 80,000/- with a payback period of 2 months.

7.2 Pressure drop reduction in Compressed air system

In this synthetic yarn manufacturing plant, compressed air is generated at 12 bar for supplying air to FDY plant. The central compressor station located at about 250 metres from the FDY plant consists of reciprocating compressors, dryers, receivers etc. The average airflow requirement is 760 Nm3/h. Compressed air to some other plants are also supplied from the same station. These sections, though supplied by 12 bar compressed air use air at 8.0 bar. The total compressed air generation at 12 bar was 3800 Nm3/h.

For satisfactory operation of FDY machines, the pressure required at the machine is 9.0 bar. Refer fig 4.2. There were 2 rows of FDY machines, one consisting of old FDY machines and the other having new machines in large numbers. Originally, the old FDY machines were supplied air through a 2” line from the compressor. For the new FDY machines, a 6” header was installed. The 2” and 6” lines were independently operated, and there was no interconnection between them.

During a pressure optimisation study, it was seen that the air pressure at old FDY machines was 9.5 bar; at the same time pressure at new FDY machines was 11.0 bar. While investigating the reasons for the difference in pressure it was found that due to small size of old FDY header, the pressure drop was significant.

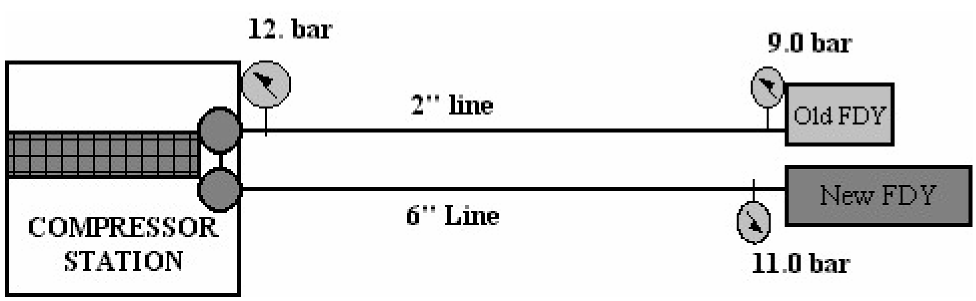


Figure 7-2: Compressed air system piping schematic

Modification:

It was decided to interconnect the 2” and 6” line near the FDY plant so that the air requirement at FDY plant is shared by both lines and hence less pressure drop in the 2” line. Measurements after the modifications indicated that the pressure at old FDY machines were 10.5 bar when the supply pressure was 12 bar.

Interestingly, the 2.5 bar pressure drop in the 2” line was the sole reason for keeping the air pressure at a higher margin. The pressure setting for the entire station was reduced to 10.5 bar after the modification. I.e a reduction of 1.5 bar. The total power consumption of 500 kW for 3800 Nm3/h reduced to 455 kW after the modifications. Minor piping cost was incurred for the modifications. Annual saving was 3,60,000 kWh/annum. I.e Rs 8.0 lakhs per annum.

7.3 Replacement of Globe Valves with Butterfly Valves

An often overlooked opportunity to reduce waste energy-particularly during retrofit applications-is the type of throttling valve used. The ISA handbook of control valves states that "In a pumped circuit, the pressure drop allocated to the control valve should be equal to 33% of the dynamic loss in the system at the rated flow, or 15 psi, whichever is greater."

An inherent result of this guideline is that high-loss valves, such as globe valves, are frequently used for control purposes. These valves result in significant losses even when they are full open. Figure 4.3 illustrates the frictional head loss for three styles of full-open 12-inch valves as a function of flow rate. (The "K" value is the valve loss coefficient at full-open position.). Even at relatively low flow rates, the power losses can be significant in high-loss valves. For instance, at 1500 gpm (for which the fluid velocity in a 12-inch line is only about 4.3 ft/sec), about 3.3 hp is lost to valve friction in the reduced trim globe valve.

Assuming the combined pump and motor efficiency is 70%, the cost of electricity is 10¢/kWh, and continuous system operation, the annual cost of friction can be estimated. About $ 3000/annum is saved by replacing the globe valve with k=30 by a butterfly valve of k=0.35.

A 250-lb pressure class butterfly valve can be purchased and installed for less than $1,000. The simple return on investment period would range from only 4 months to a year at 1500 gpm flow.

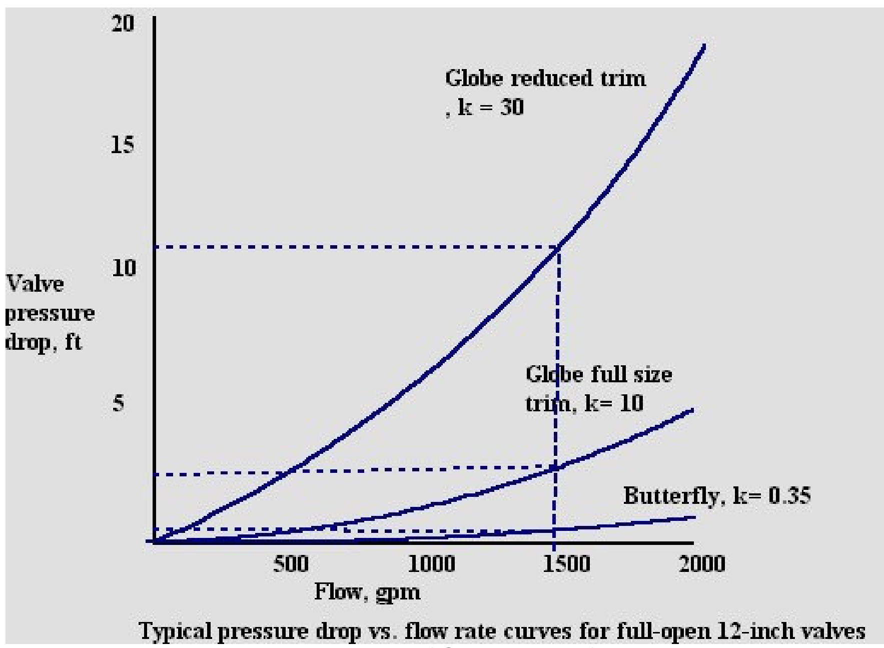


Figure 7-3: Pressure drop of Globe & Butterfly Valves

7.4 Reduction in pressure drop in the compressed air network

A leading bulk drug company has three reciprocating compressors located in a centralized compressor house. During the normal operation only one compressor is operated. The peak compressed air consumption in the plant is about 280 cfm and the corresponding power consumption was 58 kW (4.83 cfm /kW @ 7.5 kg/cm2). The pressure requirement at the user end was only 6 kg/cm2.

The compressor main line size is of is 2” inch. The main line air pressure near the receiver located next to the compressor house varies from 6.8 – 8 kg/cm2g. Pressure drop survey was carried out to evaluate the distribution system. The survey revealed that pressure drop in the system is as high as 1.5 kg/cm2g. The pressure drop in the distribution network (from the compressor house to entry to the user divisions) should not have been more than 0.6 kg/cm2, whereas in this case, the pressure drop is much higher than the optimum values. **High pressure drop in the system was due to under sizing of the piping.**

Moreover, at lower pressures and high volume flow rates, the air velocity and pressure drop is quite high. In order to maintain the required pressure at user ends, the generating air pressure was always kept higher than the compressor rated pressure of 7.03 kg/cm2. Maintaining higher generating pressure than rated, results in higher power consumption at the compressor and increased stress on the compressor leading to heating of the machine. The latter can be sensed by difference in water temperatures across the inter and after coolers.

Suggestion:
Existing pipe was replaced with 3" line reduced pressure drop by 1.0-1.5 kg/cm2. There by the generating pressure settings were reduced to 6.0- 6.5 kg/cm2g.

Cost Benefit Analysis:
- Type of Measure: Medium investment
- Annual Energy Savings: 0.35 lakh kWh
- Actual cost savings: Rs. 1.23 lakh
- Actual investment : Rs.2.50 lakh
- Payback: Two years

7.5 Thermal insulation in Steam distribution system

A leading pharmaceutical company has one 4 tph boiler to meet the steam requirement of the plant. The boiler uses furnace oil and consumes about 900 kL of furnace oil per year, which accounts for about Rs. 60 Lakh. The steam generation pressure at the common header varied from 7-9 kg/cm2-g. Steam is supplied to various sections of the plant. Detailed survey indicated that the insulation of the steam lines was completely damaged. The surface temperatures measured in the range of 68-80 oC, which were on higher side. The steam insulation was damaged from the top and it was also observed that the water was entrapped in the insulation and causing huge steam losses.

Estimated surface heat losses indicated that about 16-17 lph of furnace oil was consumed to compensate the losses. Plant has taken immediate measure to replace the entire insulation and replaced with 2-3" of insulation

Details of techno-economics:

Surface temperature before replacing the insulation	=68-80 oC
Surface temperature after replacing the insulation	= 35-37 oC
Estimated FO oil loss – before modification	= 16.7 lph
Estimated FO loss after the insulation	= 2.8 lph
FO savings	= 13.9 lph = 100 KL/year
Cost savings	= 12.7 Rs Lakh/year
Investment	= 3.0 Rs Lakh
Payback period	= 3 months

7.6 Compressed Air Leakage Reduction at Heavy Engineering Plant

This large engineering plant manufactures boilers and other heat exchangers. Use of compressed air was extensive for a number of machines and pneumatic tools. The overall housekeeping of the plant was very good; a walk through of the plant on a holiday with compressor distribution energised was done and very few leakages were seen at the end uses.

The compressed air leakage was observed to be extremely low, keeping in view the vastness of the plant where production activities are spread over a dozen bays. The leakage levels were very low in all bays (in the range of 6 to 33 cfm), except in the case bays nos. 5 and 5A, where it was as high as 196 cfm. Inspection of the plant pipeline, joints and end use points showed virtually no leakage. This was surprising because a leakage of 196 cfm would generally create sufficient hissing sounds to help in its detection.

Then it was conjectured that the leakage was possibly in the main header from the compressor room to the bays, which has a short run underground. Since part of the main header was buried in the foundation of a large machine, we presumed that the sound of leakage was being muffled. Inspection of the foundation showed mild drafts of air leaking from some crevices. Though there was no conclusive proof, a decision was taken to replace the short underground line with an overhead line.

The leakage test after the replacement of the line clearly indicated that the leakage had dropped from 196 cfm to about 15 cfm. The estimated energy savings are 1,80,000 kWh/annum i.e. Rs 5.4 lacs/annum.

The investment for replacing the compressed air line was Rs.30,000/-. It may be noted that the investment was paid back in only 21 days.

7.7 Reducing Steam Header Pressure

In any steam system, reducing unnecessary steam flow will reduce energy consumption and, in many cases, lower overall operating costs. This flow reduction can be achieved in many steam systems by lowering normal operating pressure in the steam header. To determine if such a cost saving opportunity is feasible, industrial facilities should evaluate the end use requirements of their steam system.

By evaluating its steam system and end-use equipment, Nalco Chemicals, USA realized that a lower header pressure could still meet system needs. The services performed by high-level steam jets were no longer required for the products manufactured at this plant. Instead, the steam system only needed to serve process heating and low-level steam jets, which require lower steam pressure.

The following benefits were expected from this measure.

- Decreased friction losses resulting from lower steam and condensate flow rates. Because the head loss due to friction in a piping system is proportional to the square of the flow rate, a 20% reduction in flow rates results in a 36% reduction in friction loss.

- Lower piping surface energy losses due to lower steam temperatures.

- Reduced steam losses from leaks.

- Less flash steam in the condensate recovery system, this reduces the chance of water hammer and stress on the system.

To minimize the risk of unexpected problems, the steam header pressure was first reduced from 125 psig to 115 psig. Changes in system operating conditions should be implemented carefully to avoid adverse affects on product quality. The participation of system operators is essential in both planning the change and subsequently monitoring the effects on system performance. At Nalco, after no problems were observed from the first reduction in header pressure, the pressure was stepped down further to 100 psig. Encouraged by the success of their efforts, Nalco is evaluating the feasibility of reducing the pressure even more.

Results
Overall, reducing steam header pressure was successful. This project did not require a capital investment and minimal downtime was necessary. The only costs associated with this project were for labor resources to analyze project feasibility, to recalibrate the flow meter (which receives periodic calibration anyway), and to monitor system response to the operating change. Nalco realized annual energy savings of 56,900 million Btu, cutting costs by $142,000 annually. On a per pound of product basis, the amount of energy was reduced by 8%, from 2,035 Btu/lb to 1,873 Btu/lb. The decreased fuel consumption translates into an annual 3,300-ton decrease in CO2 emissions. Additionally, by operating at lower energy levels and flow velocities, the steam and condensate systems experience less erosion and valve wear.

7.8 Insulation of steam pipelines

Present Scenario

Boiler Capacity	:	850 kg/hr (non-IBR)
Fuel consumption (LDO)	:	50 liters per hour (900 liters per day)
Boiler operating hours	:	18 per day
Plants to which boiler is attached	:	Reactor and dryers both indirect heating applications

No moisture separator installed in the line and only TD traps for drain points. After the Moisture separator was installed in the pipeline:

Fuel consumption	:	45 liters per hour (630 liters per day)
Boiler operating hours	:	14 per day

<u>Case Study to elaborate the effect of insulation of flanges:</u>
100 ft of 6 Inch pipe 12 Flanges of 6 Inch = 5 ft of pipe length Heat loss in following 3 cases:

Case (I) – Bare pipe (Bare Flanges)
Case (II) – Pipe with 2 inch insulation aluminum cladding and bare flanges
Case (III) – Insulated pipe and Flanges

		Case (I)	Case (II)	Case (III)
Heat Loss	Kcal/year	36,300	4,100	2,490
Steam Loss	Kg/Year/100ft	68	3.2	–
Fuel Loss	Kg/Year/100ft	55	0.26	–
Energy Saving Potential	Rs. Per Year/100 ft	60	2.8	–

Energy Conservation Potential:

Daily fuel saving	:	270 liters
Annual reduction in fuel bill	:	Rs. 10 Lacs
Investment Required	:	Rs. 4,500/-

Additionally the production capacity increased due to availability of the production equipment for longer durations.

7.9 Cooling water piping system modification to increase productivity

The plant, located in Gujarat, manufactures benzene derivatives. Two cooling tower pumps operating in parallel supply water to condensers in new hydrogenation plant. Specification of pumps is given below.

Pump-1
Make: KSB, Model: MEGA G 32/180
Speed: 2900 rpm, Head: 32 m, Flow = 180 m3/h,
Efficiency : 81%
Motor: 30 HP, 2900 rpm

During the energy audit, measurements were taken on these pumps as summarised below.

Pump	No.1	No.2
Power input, kW	17.9	17.7
Flow, m3/h	120	122
Head, mWC	38	38
Operating efficiency, %	82	82

The discharge pressure of the pumps was found to be 38 mWC(3.8 kg/cm2). Observing the piping and end use equipments, it was found that all the valves are fully open and the 8" header was properly sized to handle a flow of 180 m3/h. The pressure drop across the heat exchanger was also low, of the order of 0.5 kg/cm2. The reason for higher discharge pressure was still elusive. The process which is capable of 7 using condensers at a time, had to be operated with only 3 condensers on line at a time.

Further observations of pressure at various points in the system indicated that the NRV (non return valve) at the pump discharge is jammed. Pressure before the NRV, same as the pump discharge pressure, was 3.8 kg/cm2 and after the NRV were 2.0 kg/cm2. Hence it was decided to install new valves.

After replacing the existing NRVs with new valve, the system flow increased to 180 m3/h per pump, an increase of about 50% in flow. Power consumption of the pumps also increased to 22.5 kW each. However, the increase in productivity has also been 50% more resulting in higher throughput. Increased energy cost of 9.4 kW was equivalent to Rs 1.8 lakhs per year, where as the value of increase in production was roughly 10 times (Rs 20.0 lakhs per year).

7.10 Excessive pressure drop due to inadequate piping-chilled water system

The plant located in Ankleshwar, Gujarat manufactures Pesticides products. Chilling is a major end use of energy, roughly about 15% the plant energy consumption. There are two ammonia based vapour compression system to produce chilled water at 8 C. The specifications of the plant are as follows.

Compressor make: Kirloskar
Model: KC6
Capacity (at 0ºC SST) = 120 TR
Rated specific power consumption at above SST = 0.72 kW/TR
Rated primary flow = 61 m^3/h
Rated condenser flow = 105 m^3/h
The chilled water system has primary and secondary pumping arrangements. The primary pump specifications and measurements are given below.

Make: Kirloskar
Model: DB 65/13
Head: 30 mWC
Flow: 130 m^3/h
Motor: 20 HP

Measured values:
Discharge Head: 37.5 mWC
Suction head: 4 mWC
Flow: 112.2 m^3/h
Power input: 14.0 kW
Operating efficiency: 80%

The pressure at pump discharge was 37.5 mWC and the pressure at chiller inlet was 28 mWC. This indicated that the pressure drop in the 4" supply piping from pump to the chiller was 9.5 mWC. This is very high. Similar pressure drop was observed in return line also.

The pipe sizing of 4" in generally adequate for a rated primary flow of 61 m3/h. However, due to improper selection of pump, the pump is giving about 112.2 m3/h; almost double that of the required flow rate. This flow through a 4: pipe is expected to produce a pressure drop of about 10 mWC.

The solution suggested was to reduce the primary flow, by reducing the impeller diameter. The plant personnel wanted to know if reducing primary flow would effect the chilling capacity. A trial was taken by reducing the flow by controlling valves to evaluate chiller performance. Flow reduced from 112 m^3/h to 75 m^3/h. Total reduction in pressure in the supply and return was 1.5 kg/cm2.

The following graph shows the variations in chiller inlet/outlet temperature and capacity before and after reducing the flow.

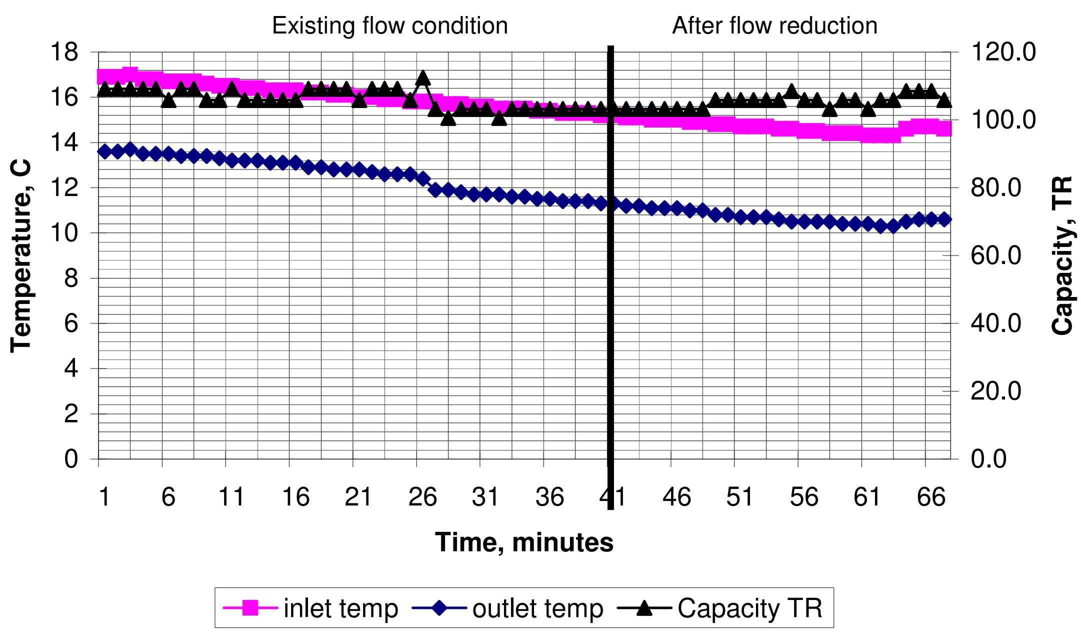


Figure 7-4: Chiller performance

Note that, with reduction in primary flow, inlet and outlet temperatures found to be reduced. Most importantly, the capacity of the machine remained unaffected.

After gaining confidence from the above exercise, the impeller diameter of primary pump was reduced from existing 174 mm to 145 mm. Since the existing impeller diameter and the new impeller diameter were very different, a new 145 mm impeller was purchased, instead of trimming.

After installing the new impeller, the performance is as follows.

Head: 23 mWC
Flow: 75 m^3/h
Power input: 8.2 kW

Energy saving for 9 months,10 hours/day operation is found to be 15,660 kWh/annum. i.e. Rs 70,000/- per annum. Investment for a new impeller was Rs 4000/- with a payback period of one month.

ANNEXURE-1: REFERENCES

1. Fuel efficiency Booklet- FEB 002- Steam: ETSU, BRESCU-UK
2. Fuel efficiency Booklet- FEB 008-Economic Thickness of Insulation of Hot Pipes-ETSU, BRESCU-UK
3. Fuel efficiency Booklet- FEB 019-Process Plant Insulation and fuel efficiency-ETSU, BRESCU-UK
4. Steam System Survey Guide- ORNL & US Department of Energy, USA
5. Pipes & Pipe Sizing- Spirax Sarco

Websites:
1. www.cheresources.com
2. www.energymanagertraining.com
3. www.oit.doe.gov
4. www.plantsupport.com
5. www.ecompressedair.com
6. www.spiraxsarco.com
7. www.engineeringtoolbox.com