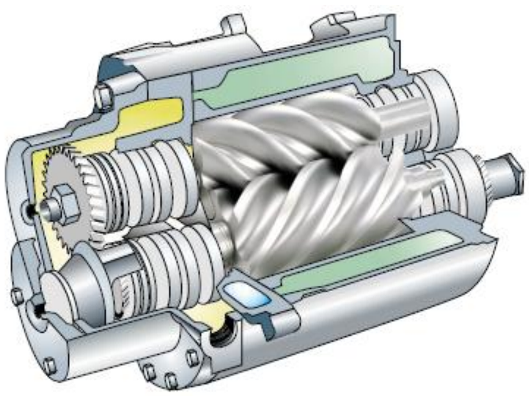
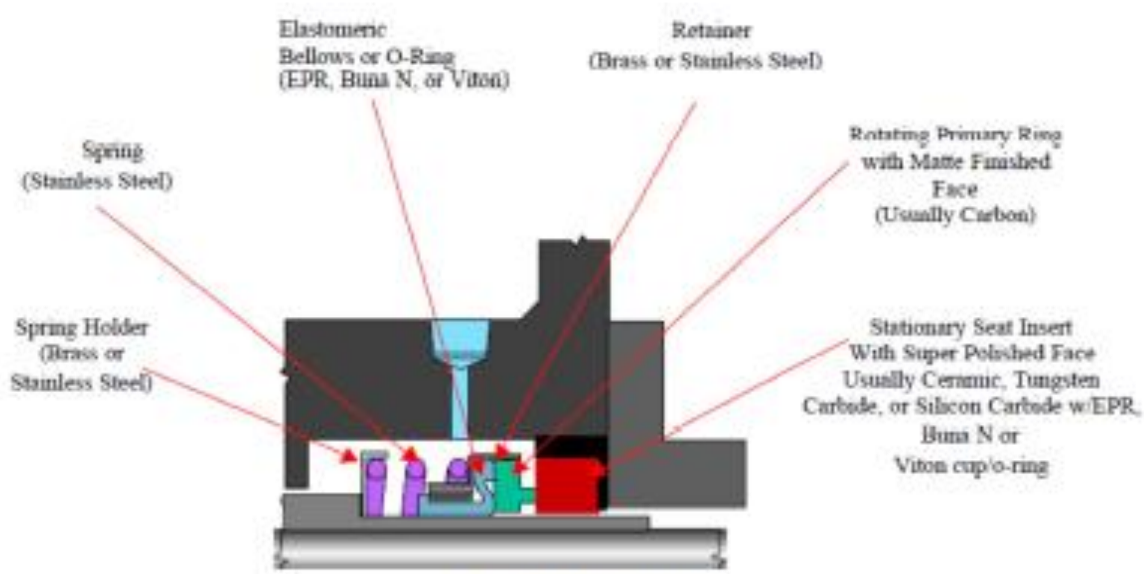
POWER MANAGEMENT INSTITUTE

Pumps, Compressors and Seals



Screw Compressor



Seal Elements

Contents

A. **Pumps**

1. Pumps - Introduction

1.1 Background

The most numerous types of fluid machineries are of the pump family (machines which *add* energy to the fluid), other important types are turbines (which *extract* energy from fluid). Both types are usually connected to a rotating shaft, hence also called *turbomachineries*. The prefix *turbo*- is a Latin word meaning “spin’’ or “whirl,’’ appropriate for rotating devices. The pump is the oldest fluid-energy-transfer device known. At least two designs date before Christ: (i) the undershot-bucket waterwheels, or *norias*, used in Asia and Africa (1000 B.C.) and (ii) Archimedes’ screw pump (250 B.C.), still being manufactured today to handle solid-liquid mixtures or to raise water from the hold of a ship. Paddlewheel turbines were used by the Romans in 70 B.C., and Babylonian windmills date back to 700 B.C. Since that time, many variations and applications of pumps have been developed. The power generating turbomachines (turbines) decrease the head or energy level of the working fluids passing through them and they are coupled to machines, such as electric generators, pumps, compressors etc.

1.2 Desired Function

A pump imparts energy into a liquid to lift the liquid to a higher level, to transport the liquid from one place to another, to pressurize the liquid for some useful purpose, or to circulate the liquid in a piping system by overcoming the frictional resistance of the piping system. The head or pressure producing machines increase the energy level (pressure or head) of the fluids passing through them. These machines are pumps, compressors, fans, blowers and propellers. They are driven by prime movers such as turbines and electric motors for supplying the power required to increase the energy level of the fluid.

1.3 The Operating Principle

In a pump energy is imparted to a liquid by following two basic methods: addition of kinetic energy, and volumetric or positive displacement. Kinetic energy can be added to a fluid by rotating the fluid at high speeds, in a device typically known as a centrifugal pump, or by providing an impulse in the direction of flow. Dynamic pumps simply add momentum to the fluid by means of fast-moving blades or vanes or certain special designs. There is no closed volume, the fluid increases momentum while moving through open passages and then converts its kinetic energy (due to high velocity) to a pressure increase by exiting into a diffuser section. Volumetric (positive) displacement of a fluid can be accomplished either mechanically by the action of a screw or a plunger, or by the use of another fluid. The fluid is admitted to a cavity through an inlet (valve opens), the valve then closes, and the fluid is squeezed through an outlet (the fluid is forced along by volume changes). The mammalian heart is a good example, and many mechanical designs are in wide use.

2. Pump Hydraulic Nomenclature and Definitions

Accurate terminology is essential to those who make pump applications and selections. Many pumping terms are confusing to both experienced and inexperienced pump designers and application engineers. One of the best sources of current terminology, nomenclature, and definitions is the *Hydraulic Institute Standards.* The definitions and terminology used herein are based on today's practice among pump manufacturers.

2.1 Pressure Measurements and Units

Energy imparted into a liquid in the pump increases the pressure of the liquid to lift, to transport, or to circulate the liquid in a system by overcoming the frictional resistances. Pressure is the normal force exerted by a system against unit area of its bounding surface. SI unit of the pressure is Pascal (N/m^2), which is very small. Instead the unit bar, kPa or MPa is used.

$$1 \text{ bar} = 10^5 \text{ N/m}^2 = 100 \text{ kN/m}^2 = 100 \text{ kPa}$$

Pressures are stated in mm or cm of Hg and in MKS system kgf/m^2. The unit kgf/m^2 is very large by comparison with many engineering values, and unit kgf/cm^2 or ata (atmosphere technical absolute) is used.

$$1 \text{ standard atmosphere} = 760 \text{ mmHg} = 1.01325 \text{ bar}$$

$$= 101.325 \text{ N/m}^2 = 1.0332 \text{ ata} = 1.0332 \text{ kgf/cm}^2.$$

Most instruments indicate pressure relative to the atmospheric pressure, whereas the pressure of a system is its pressure above zero or relative to a perfect vacuum. The pressure relative to the atmosphere is called gauge pressure. The pressure relative to the perfect vacuum is called absolute pressure. Figure 1 illustrates the relationship among absolute vacuum, atmospheric and gauge pressure**.**

Absolute pressure = Gauge pressure + Atmospheric pressure

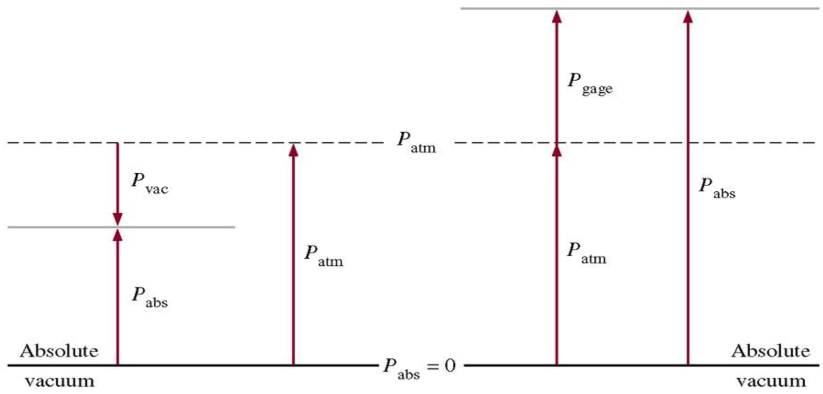


Fig.1: Absolute vacuum, atmosphere and gauge pressure

Figure 2 shows the basic barometer and Fig. 3 an open U-tube indicating system gauge pressure. One end close U-tube indicates absolute pressure. When the pressure in a system is less than atmospheric pressure, the gauge pressure becomes negative it is frequently designated by positive number and called vacuum. For example 700 mmHG vacuum will be (760-700)/760*1.013 = 0.08 bar (abs)

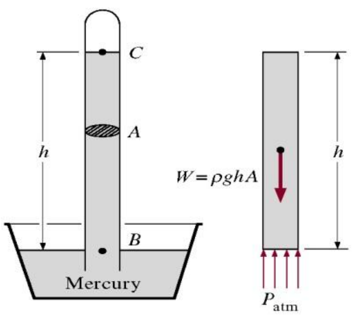


Fig. 2: The basic barometer and The atmospheric pressure

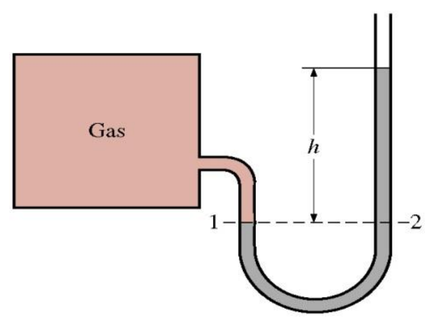


Fig. 3: The basic manometer and gauge pressure

If the height of the liquid (of density ρ kg/m^3) column is h m, the pressure exerted by the liquid column is given by $Pg = h\rho g$ [m.kg/m^3.m/s^2] $= h\rho g$ N/m^2. The pressure exerted by 1 m mercury column = 1X13616X9.81=1.3366 bar.

2.2 Pump Hydraulic Terminologies

Important terminologies that are used in relation to pump are shown in Fig. 4 and illustrated in subsequent paragraphs.

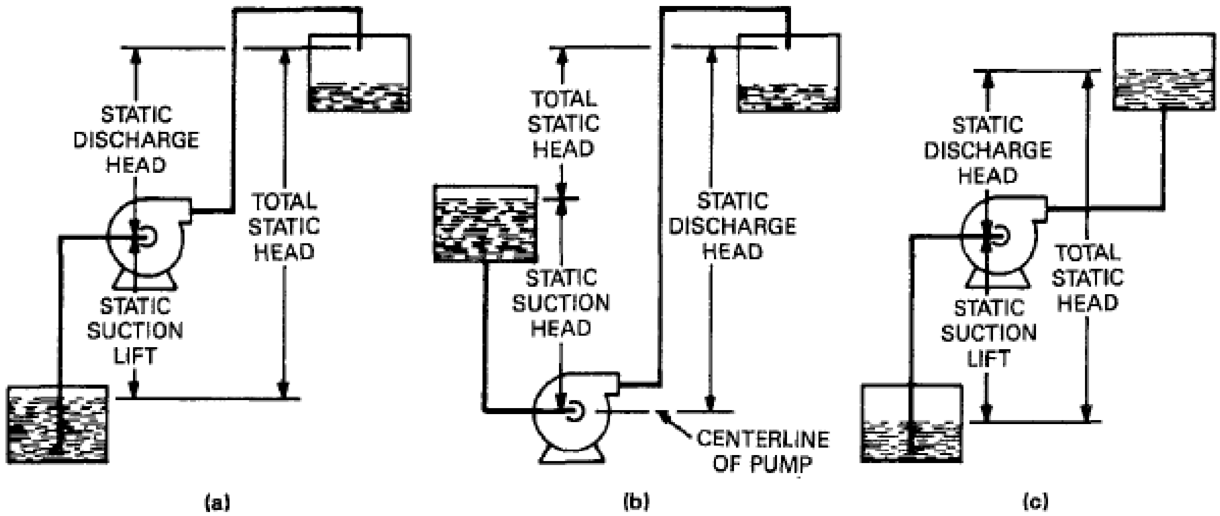


Fig.4: Different heads in relation to pump

2.2.1 Static discharge head

The vertical distance in meters between the pump center line and the point of free discharge on the surface of the liquid in the discharge tank is defined as static discharge head.

2.2.2 Total static head

The vertical distance in meters between the liquid level of the supply and the point of free discharge on the surface of the liquid in the discharge tank is defined as total static head.

2.2.3 Friction head

The head required to overcome the resistance to flow in the pipe and fittings is defined as friction head. It is dependent on the size and type of pipe, flow rate, and liquid properties.

2.2.4 Velocity head

The energy of a liquid as a result of its motion at some velocity is defined as velocity head. Velocity head is the equivalent head in meters through which a liquid would have to fall to acquire the velocity, or the head, necessary to accelerate the liquid.

Velocity head is calculated as, $h_v = \frac{V^2}{2g}$.

2.2.5 Pressure head

If a pumping system begins or ends in a tank in which internal pressure is above or below atmospheric pressure, the pressure head of the pumping system must be included in the head calculation of a pump. The pressure in such a tank must be first converted to meters of the liquid. The vacuum in a suction tank or a positive pressure in a discharge tank must be added to the pumping system head; on the other hand, a positive pressure in a suction tank and a vacuum in a discharge tank must be subtracted.

2.2.6 Total dynamic suction lift (h_s)

Total dynamic suction lift (h_s) is the static suction lift plus the velocity head at the pump suction flange plus the total friction head in the suction pipeline. Total dynamic suction head (h_s) is the static suction head minus the velocity head at the pump suction flange minus the total friction head in the suction pipeline.

2.2.7 Total dynamic discharge head (h_d)

Total dynamic discharge head (h_d) is the static discharge head plus the velocity head at the pump discharge flange plus the total friction head in the discharge line.

2.2.8 Total head or total dynamic or total developed head (TDH)

Total head or total dynamic head (total developed head, TDH) is the total dynamic discharge head minus the total dynamic suction head or plus the total dynamic suction lift.

$TDH = h_d - h_s$ (for a suction head)

$TDH = h_d + h_s$ (for a suction lift)

2.2.9 Capacity

Capacity is the throughput of the pump normally expressed in (cubic meters per hour, m^3/h). Since liquids are almost incompressible, there is a direct relationship between the capacity in a pipeline and the velocity of the flow, $Q = AXV$. Where Q = capacity, (m^3/sec); A = area of the pipeline, (m^2); and V = liquid velocity, (m/s).

3. Classification of Pumps and Different Applications

Because of the many and varied uses and applications for a pump, many types of pumps have evolved. There are two basic types of pumps: **positive-displacement pumps (PDPs)** and **dynamic or momentum change pumps**. There are several billion of each type in use in the world today.

3.1 Positive-Displacement Pumps

In this category of pumps, volumetric (positive) displacement of a fluid is accomplished either mechanically by the action of a screw or a plunger, or by the use of another fluid. The fluid is admitted to a cavity through an inlet (valve opens), the valve then closes, and the fluid is forced along by volume changes. The classifications of PDPs are as follows:

I. Reciprocating
   a. Piston or plunger
   b. Diaphragm

II. Rotary
   A. Single rotor
      a. Sliding vane
      b. Flexible tube or lining
      c. Screw
      d. Peristaltic (wave contraction)
   B. Multiple rotors
      a. Gear
      b. Lobe
      c. Screw
      d. Circumferential piston

All PDPs deliver a pulsating or periodic flow as the cavity volume opens, traps, and squeezes the fluid. Schematic of the operating principles of few of these types of pumps are shown in Fig.5. Since PDPs compress mechanically against a cavity filled with liquid, a common feature is that they develop immense pressures if the outlet is shut down for any reason. Sturdy construction is required, and complete shutoff would cause damage if pressure- relief valves were not used. Their great advantage is the delivery of any fluid regardless of its viscosity.

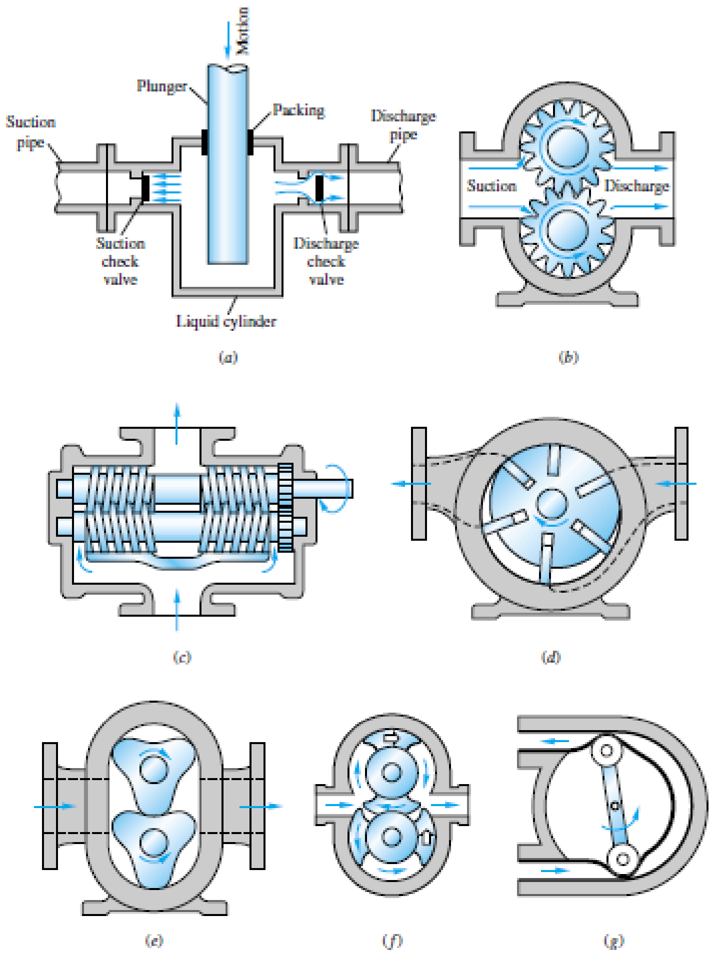


Fig. 5: Schematic design of positive-displacement pumps: (a) reciprocating piston or plunger, (b) external gear pump, (*c*)double-screw pump, (*d*) sliding vane, (*e*) three lobe pump, (*f*) double circumferential piston, (*g*) flexible-tube squeegee.

3.2 Dynamic Pumps

Dynamic pumps simply add momentum to the fluid by means of fast-moving blades or vanes or certain special designs. There is no closed volume: The fluid increases momentum while moving through open passages and then converts its high velocity to a pressure increase by exiting into a diffuser section. Kinetic pumps can be divided into two classes: centrifugal and regenerative. The most common type of kinetic pump used in a modern central power plant is the centrifugal pump. Although the first centrifugal pump was developed in the late 1600 ADs, most of the development of this pump type has occurred in the present century. Centrifugal pumps include radial, axial, and mixed flow types, with the radial flow volute type used for the bulk of power plant applications. Regenerative pumps are typically referred to as vortex, peripheral, or turbine pumps.

Dynamic pumps can be classified as follows:

I. Rotary
   a. Centrifugal or radial exit flow
   b. Axial flow
   c. Mixed flow (between radial and axial)

II. Special designs
   a. Jet pump or ejector
   b. Electromagnetic pumps for liquid metals
   c. Fluid-actuated: gas-lift or hydraulic-ram

Here, emphasis is given on the rotary designs, sometimes called rotodynamic pumps. Dynamic pumps generally provide a higher flow rate than PDPs and a much steadier discharge but are ineffective in handling high-viscosity liquids. Dynamic pumps also generally need priming; i.e., if they are filled with gas, they cannot suck up a liquid from below into their inlet. The PDP, on the other hand, is self-priming for most applications. A dynamic pump can provide very high flow rates (up to 300,000 gal/min) but usually with moderate pressure rises (a few atmospheres). In contrast, a PDP can operate up to very high pressures (300 atm) but typically produces low flow rates (100 gal/min). The relative *performance* (Δp versus Q) is quite different for the two types of pump. At constant shaft rotation speed, the PDP produces nearly constant flow rate and virtually unlimited pressure rise, with little effect of viscosity. The flow rate of a PDP cannot be varied except by changing the displacement or the speed. The reliable constant-speed discharge from PDPs has led to their wide use in metering flows. The dynamic pump, by contrast, provides a continuous constant-speed variation of performance, from near-maximum Δp at zero flow (shutoff conditions) to zero Δp at maximum flow rate. High-viscosity fluids sharply degrade the performance of a dynamic pump.

Let us begin our brief look at rotodynamic machines by examining the characteristics of the centrifugal pump. This pump consists of an impeller rotating within a casing. Fluid enters axially through the *eye* of the casing, is caught up in the impeller blades, and is whirled tangentially and radially outward until it leaves through all circumferential parts of the impeller into the diffuser part of the casing. The fluid gains both velocity and pressure while passing through the impeller. The doughnut-shaped diffuser, or *scroll*, section of the casing decelerates the flow and further increases the pressure. The impeller blades are usually *backward-curved*, but there are also radial and forward-curved blade designs, which slightly change the output pressure. The blades may be *open*, i.e., separated from the front casing only by a narrow clearance, or *closed*, i.e., shrouded from the casing on both sides by an impeller wall. The diffuser may be *vaneless*, or fitted with fixed vanes to help guide the flow toward the exit.

3.2.1 Volute Type Centrifugal Pumps

The operating principle of a volute pump is shown in Fig.6. Liquid enters the pump at the impeller eye and is thrown radially outward through an expanding pump casing. This action creates a low pressure at the impeller eye which draws more liquid into the impeller. The velocity head imparted to the liquid by the impeller is converted into static pressure by the widening spiral pump casing.

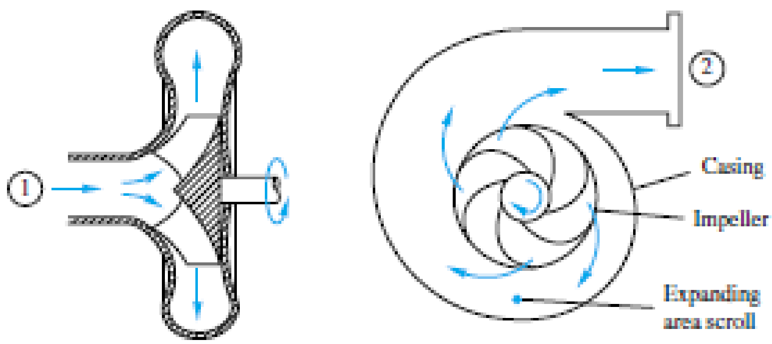


Fig.6: Volute centrifugal pump

3.2.2 Diffuser Type Centrifugal Pumps

Another type of radial flow centrifugal pump is the diffuser pump, shown in Fig. 7. As with the volute centrifugal pump, the liquid enters a diffuser type pump at the impeller eye and is thrown outward along the impeller vanes. After the liquid has left the impeller, it is passed through a ring of stationary guide vanes that surround the impeller and diffuse the liquid to provide a controlled flow and efficient conversion of velocity head into static pressure.

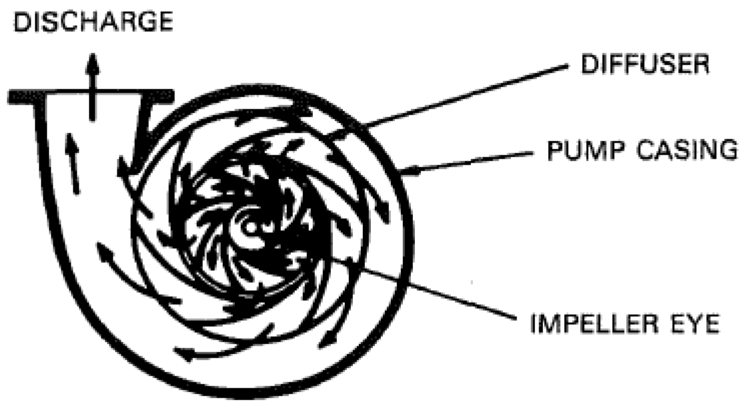


Fig.7: Diffuser Centrifugal Pump

3.2.3 Axial Flow and Mixed Flow Centrifugal Pumps

In an axial flow pump, the impeller acts as a propeller. Liquid flows parallel to the axis or shaft of the pump, as shown in Fig. 8. Pressure is generated by the propelling and lifting action of the impeller vanes on the liquid. Normally, diffusion vanes are located on the discharge side of the pump to eliminate rotation and radial velocity of the liquid imparted by the impeller. In mixed flow pumps, liquid is discharged both radially and axially into a volute type of casing. Pressure is developed both by centrifugal force and by the lift of the impeller vanes on the liquid. The action of a mixed flow impeller is shown in Fig. 9.

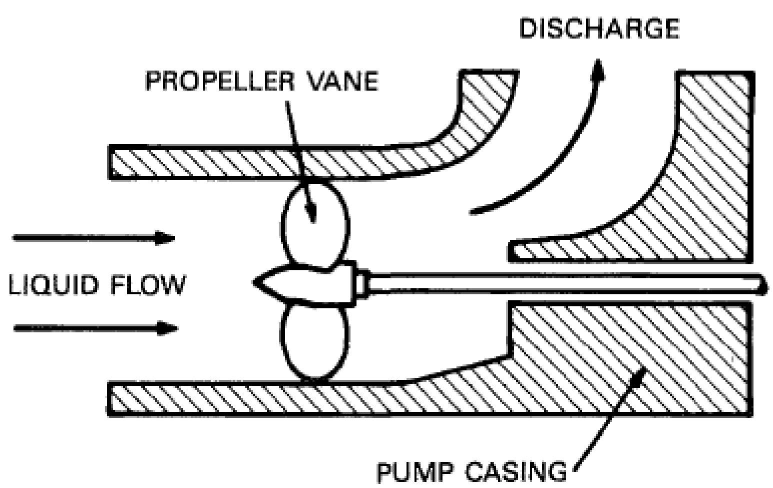


Fig.8: Axial flow pump

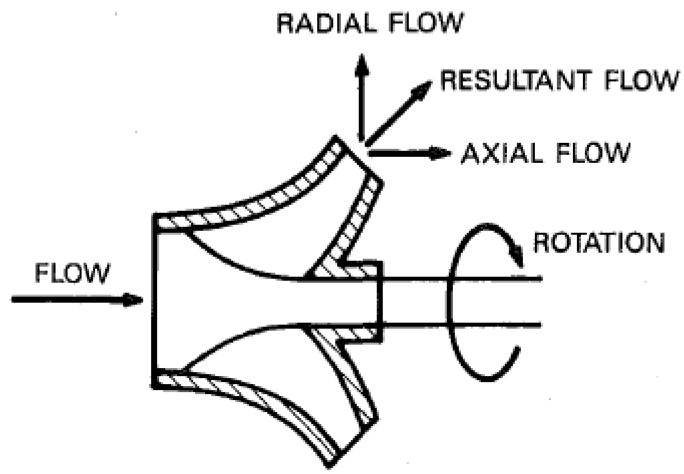


Fig.9: Mixed flow impeller

3.2.4 Regenerative pumps

A regenerative type of centrifugal pump has an impeller with vanes on both sides of a rim that rotates in a channel in the pump's casing, as shown in Fig. 9. Liquid enters through a nozzle into an impeller vane and is forced outward by centrifugal force. The liquid impacts the pump casing and is turned inward to reenter the impeller at a different vane. This cycle is repeated throughout the rotation of the impeller, generating pressure until the liquid is forced out of the pump at the discharge nozzle. Because of close running tolerances, the regenerative pump is suitable only for clear liquids with relatively low viscosities. These pumps are useful in pumping liquids containing vapors and gases because of their resistance to cavitation.

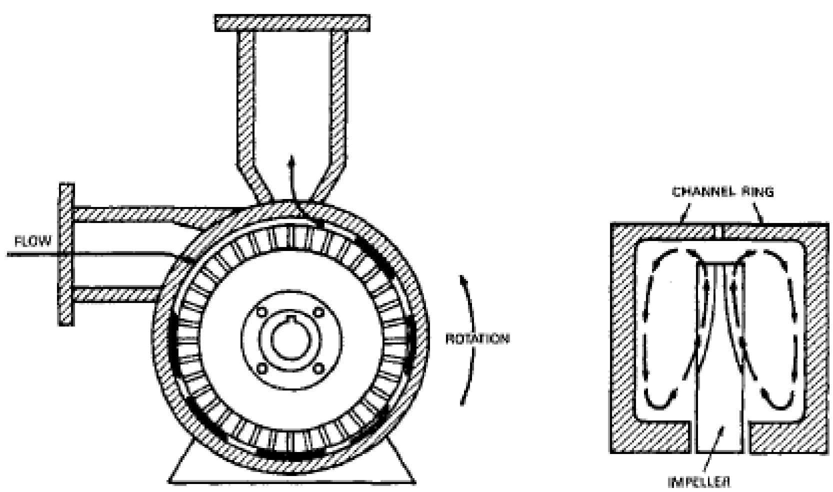


Fig.9: Regenerative turbine pump

4. Specific Speed and Impeller Configuration

A dimensionless index of pump type known as specific speed has been developed for pump design and selection to show the relationship between pump capacity, head, and impeller speed. Specific speed of an impeller is defined as the speed in revolutions per minute at which a geometrically similar impeller would operate to deliver unit flow rate at a developed head of 1 m. Specific speed is algebraically defined as $N_s = \frac{n\sqrt{Q}}{h^{\frac{3}{4}}}$.

Where, N_s = specific speed; n = pump speed, rpm; Q = pump flow at best efficiency point (BEP), and h = pump developed head at BEP.

Specific speed characterizes the shape and configuration of an impeller. Since the ratios of the major impeller dimensions vary uniformly with specific speed, specific speed is useful to the pump designer in determining impeller proportions and dimensions required, as well as to the application engineer in checking suction limitations of the pump.

It should be noted that the Eq. for specific speed is written for single-stage pump applications. For multistage pumps, the head per stage is used to calculate specific speed. Generalizing, the head per stage is found by dividing the total head of the pump by the number of stages of the pump. For power station centrifugal pump applications, the following specific speeds and impeller profiles are generally seen:

Application	Typical impeller profile	Specific speed, N_s
Boiler feed	Radial-vane area or Francis-vane area	1,000–1,800
Condensate	Francis-vane area	1,500–2,500
Circulating water	Mixed flow area	4,500–7,500

5. Affinity Laws

Another important tool used by pump designers and application engineers is Affinity Laws. These laws express the mathematical relationship and illustrate the effect of changes in pump operating conditions or pump performance variables such as pump head, flow, speed, horsepower, and pump impeller diameters. The affinity laws are summarized by the following three Eqs. where the subscript 1 refers to the original conditions and 2 refers to the new conditions.

$$\text{Flow } Q_2 = Q_1 \frac{n_2}{n_1} \frac{D_2}{D_1}$$

$$\text{Head } h_2 = h_1 \left(\frac{n_2}{n_1}\right)^2 \left(\frac{D_2}{D_1}\right)^2$$

$$\text{Horsepower } hp_2 = hp_1 \left(\frac{n_2}{n_1}\right)^3 \left(\frac{D_2}{D_1}\right)^3$$

where

Q = pump flow, gpm (m^3/h);

n = pump speed, rpm;

D = impeller diameter, in. (m);

h = pumphead, ft (m);

hp = pump brake horsepower, bhp (kw); and

$$\text{Efficiency } \varepsilon_{p2} = \varepsilon_{p1}$$

6. Pump Power and Efficiency

Assuming steady flow, the pump basically increases the Bernoulli head of the flow between point 1, the eye, and point 2, the exit. Neglecting viscous work and heat transfer, this change is denoted by *H*:

$$H = \left(\frac{p}{\rho g} + \frac{V^2}{2g} + z\right)_2 - \left(\frac{p}{\rho g} + \frac{V^2}{2g} + z\right)_1 = h_s - h_f$$

Where, h_s is the pump head supplied and h_f the losses. The net head *H* is a primary output parameter for any turbomachine. Usually V_2 and V_1 are about the same, z_2 - z_1 is no more than a meter or so, and the net pump head is essentially equal to the change in pressure head

$$H \approx \frac{p_2 - p_1}{\rho g} = \frac{\Delta p}{\rho g}$$

The power delivered to the fluid simply equals the specific weight times the discharge times the net head change

$$P_w = \rho g Q H$$

This is traditionally called the *water horsepower*. The power required to drive the pump is the *brake horsepower*

$$\text{bhp} = \omega T$$

Where, ω is the shaft angular velocity and T the shaft torque. If there were no losses, Pw and brake horsepower would be equal, but of course Pw is actually less, and the *efficiency* of the pump is defined as

$$\eta = \frac{P_w}{\text{bhp}} = \frac{\rho g Q H}{\omega T}$$

The chief aim of the pump designer is to make η as high as possible over as broad a range of discharge Q as possible.

The efficiency is basically composed of three parts: volumetric, hydraulic, and mechanical.

The volumetric efficiency is

$$\eta_v = \frac{Q}{Q + Q_L}$$

Where, Q_L is the loss of fluid due to leakage in the impeller-casing clearances. The hydraulic efficiency is

$$\eta_h = 1 - \frac{h_f}{h_s}$$

where h_f has three parts: (1) shock loss at the eye due to imperfect match between inlet flow and the blade entrances, (2) friction losses in the blade passages, and (3) circulation loss due to imperfect match at the exit side of the blades.

Finally, the mechanical efficiency is

$$\eta_m = 1 - \frac{P_f}{\text{bhp}}$$

where P_f is the power loss due to mechanical friction in the bearings, packing glands, and other contact points in the machine.

By definition, the total efficiency is simply the product of its three parts

$$\eta = \eta_v \eta_h \eta_m$$

The designer has to work in all three areas to improve the pump.

7. Pump Performance Curves

The measured shutoff head of centrifugal pumps is only about 60 percent of the theoretical value. With the advent of the laser-doppler anemometer, researchers can now make detailed three-dimensional flow measurements inside pumps and can even animate the data into a movie. The positive-slope condition in Fig. 10 can be unstable and can cause pump *surge*, an oscillatory condition where the pump "hunts'' for the proper operating point. Surge may cause only rough operation in a liquid pump, but it can be a major problem in gas compressor operation. For this reason a backward-curved or radial blade design is generally preferred.

Since the theory is rather qualitative, the only solid indicator of a pump's performance lies in extensive testing. For the moment let us discuss the centrifugal pump in particular. The general principles and the presentation of data are exactly the same for mixed-flow and axial-flow pumps and compressors.Performance charts are almost always plotted for constant shaft-rotation speed n (in r/min usually). The basic independent variable is taken to be discharge Q (in gal/min usually for liquids and ft3/min for gases). The dependent variables, or "output,'' are taken to be head H (pressure rise p for gases), brake horsepower (bhp), and efficiency. Figure 10 shows typical performance curves for a centrifugal pump. The head is approximately constant at low discharge and then drops to zero at $Q = Q_{max}$. At this speed and impeller size, the pump cannot deliver any more fluid than Q_{max}. The positive slope part of the head is shown dashed; as entioned earlier, this region can be unstable and can cause hunting for the operating point.

The efficiency is always zero at no flow and at Q_{max}, and it reaches a maximum, perhaps 80 to 90 percent, at about $0.6Q_{max}$. This is the *design flow rate* Q^* or *best efficiency point* (BEP). The head and horsepower at BEP will be termed H^* and P^* (or bhp*), respectively. It is desirable that the efficiency curve be flat near max efficiency, so that a wide range of efficient operation is achieved. However, some designs simply do not achieve flat efficiency curves. As shown in Fig. 10, the horsepower required to drive the pump typically rises monotonically with the flow rate. Sometimes there is a large power rise beyond the BEP, especially for radial-tipped and forward-curved blades. This is considered undesirable because a much larger motor is then needed to provide high flow rates. Backward- curved blades typically have their horsepower level off above BEP.

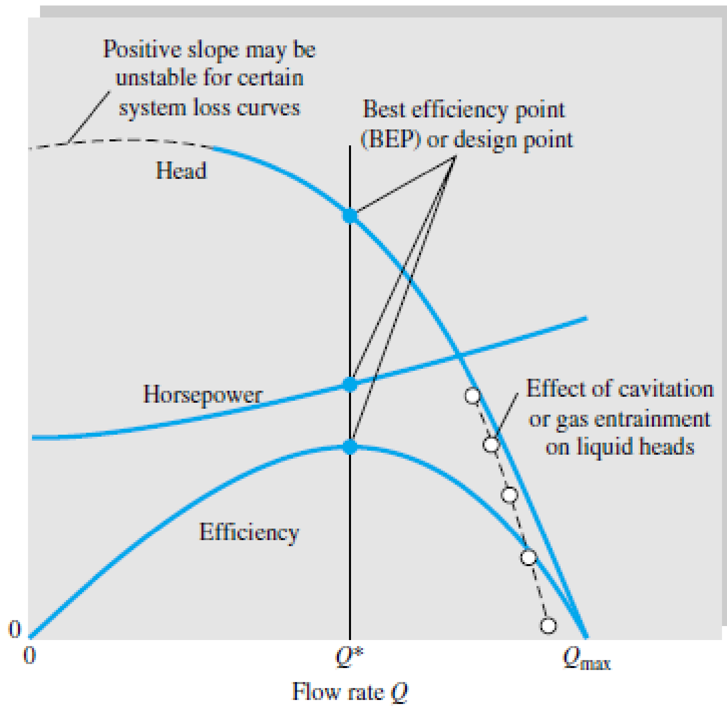


Fig.10: Pump characteristics curves

8. Cavitation

In the handling of incompressible liquids, all pumps have a tendency to cavitate when the pressure in the pump inlet regions approaches the vapor pressure of liquids. The total energy at any point in the flowing liquid consists of pressure and velocity (kinetic) heads. Therefore, until the impeller vanes begin to add energy to the liquid being pumped, the pressure decreases as the velocity increases. So, the suction pressure necessary to prevent cavitation increases with an increase in the pump output. Cavitation is a two-stage phenomenon consisting of the formation of vapor cavities resulting from low pressure and their collapse as they move out of the low-pressure into higher pressure regions. The higher pressure region causing the vapor cavity to collapse can be immediately following the formation of the vapor cavity or some distance downstream from the impeller inlet, depending on the downstream pressure conditions and the quantity of vapor formed. The formation and collapse of the vapor cavities occur in a very short time interval. It is at the point of collapse where the physical damage to the impeller occurs.

Therefore, when selecting a pump for a piping system, it is necessary to avoid cavitation in the pump operating range. The pump must be located so that there is sufficient pressure above the vapor pressure at the pump inlet to ensure no cavitation. The cavitation characteristics of each pump design are determined by testing on the basis of handling water at or near room temperature. However, the pump application may be for another fluid of varying viscosities or temperatures. In each case, the pump manufacturer should be consulted for accompanying changes in the cavitation characteristics of the pump design.

At the inception of cavitation there may not be a measurable decrease in pump performance although pump damage may be taking place. As cavitation increases, there is a reduction in pump output because the vapor cavities occupy some of the impeller water passage area. This reduces developed head and efficiency. The point at which suction lift begins to affect head and efficiency is the critical condition shown on the pump rating charts (characteristic curves).

9. Net Positive Suction Head (NPSH)

In the top of Fig. 11.7 is plotted the *net positive-suction head* (NPSH), which is the head required at the pump inlet to keep the liquid from cavitating or boiling. The pump inlet or suction side is the low-pressure point where cavitation will first occur. Net positive suction head (NPSH) is the term used by the pump industry for describing pump cavitation characteristics. NPSH is defined as the pressure (head) in excess of the saturation pressure of the fluid being pumped. NPSH is expressed as NPSH A (available) and NPSHR (required). NPSHA is the NPSH available or existing at the pump installed in the system. NPSH available for various pump suction arrangements are shown in Fig. 11. NPSHR is a performance characteristic of a pump and is established through closed loop or valve suppression tests conducted by the pump manufacturer. These tests consist of lowering the NPSHA provided to the test pump until the pump head, power, or efficiency noticeably decreases. At this point, the pump is cavitating. The NPSHR is then established based on a predefined percentage reduction in head, power, or efficiency. Usually NPSHR is established as 3% head reduction in single-stage pumps or 3% first-stage head reduction in multistaged pumps. NPSHR is defined for each flow rate shown on the pump rating chart and is the pumping system NPSH required to prevent cavitation at the pump impeller inlet. Typically, the NPSHR shown on the pump rating chart was determined by the pump manufacturer based on a 3% or greater reduction in pump discharge head. Piping system designers should keep this in mind when allowing for the pump suction conditions in a particular pumping system. First, the pump manufacturer should be consulted to determinethe basis of his stated values of NPSHR. Second, the pumping system designer should provide some margin above the stated NPSHR value when designing for the pump suction conditions. Typical margins over the published NPSHR values may run from 10% to 50% for a simple cold water pumping system to 50% to 100% for a complex boiler feed pumping system with transient suction condition operations. Particular attention to NPSH margin should be given to boiler feedwater systems where load rejection and system transients are expected.

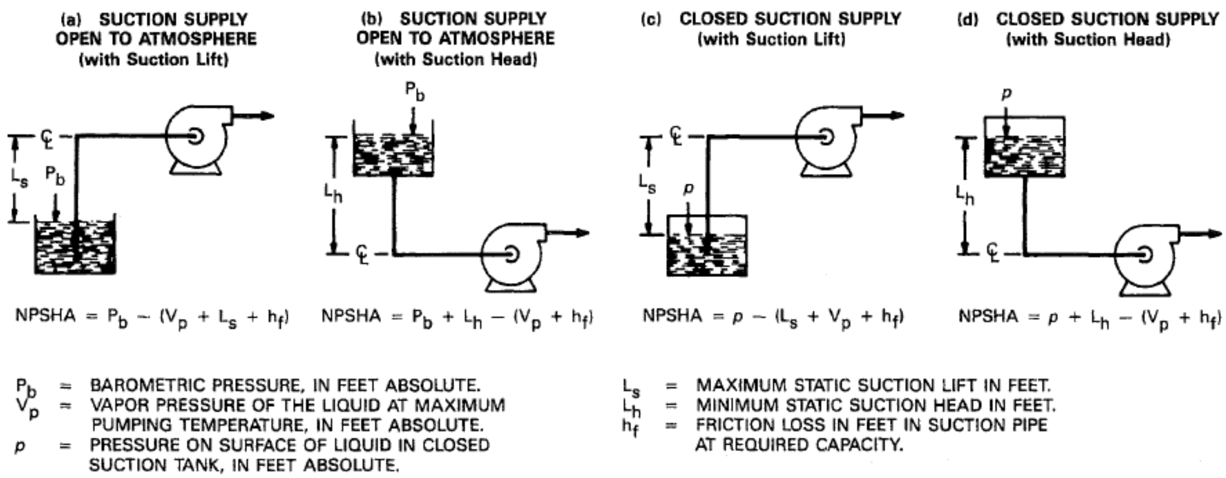


Fig.11: NPSH available for various pump suction arrangements

10. Some Typical Pumps in the Plant

10.1 Condensate Extraction Pump (CEP)

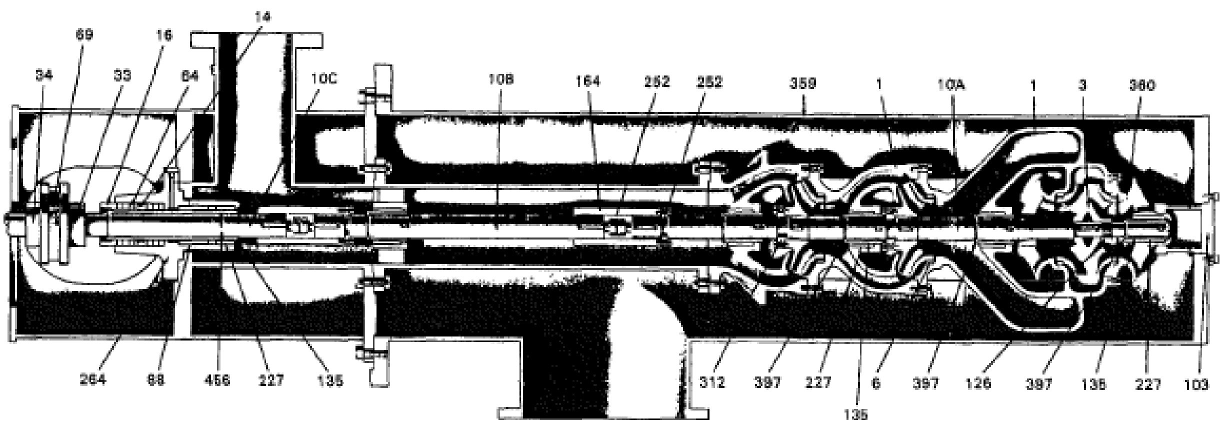
The selection of a condensate pump requires the proper application and evaluation of several important condensate pump operating parameters and general guidelines. These include the pump capacity and total head requirements, net positive suction head, pump/suction can length, suction specific speed, peripheral velocity of the outer tip of the impeller inlet vanes, and piping system layout.

10.1.1 Design Point

The design point for condensate pumps is specified by the design capacity and total developed head. The design capacity for each condensate pump is determined by summing the condensate flow requirements for the following items: Condensate flow requirements from the condenser hot well with the unit operation at maximum turbine heat balance conditions; typically 5% overpressure conditions with the turbine control valves wide open (VWO); Condensate cycle makeup flow, typically 1% to 3% of the maximum condensate flow.

10.1.2 CEP Components

CEPS are usually vertical multistaged units with below-floor suction cans and are located near the condenser. The two important condensate pump selection factors that remain are the number of condensate pumps required and the type of pump flow control to be used. These two factors are determined by the plant designer during a detailed investigation and evaluation of the condensate system. Information regarding the planned plant operating characteristics (base loaded or peaking), pumping flexibility, and operator requirements are considered as well as the plant economics, equipment capital cost, and operation and maintenance costs. Figure 12 shows the components of a vertical, multistage, wet suction can type double-suction first-stage impeller condensate pump.



PART NO.	PART NAME				
1	CASING	34	DRIVER COUPLING HALF	252	SPLIT RING
3	IMPELLER	64	STUFFING BOX PACKING	264	STUFFING BOX EXTENSION
6	CASING RING	69	ADJUSTING NUT	312	LOCK COLLAR
10A	PUMP ELEMENT SHAFT	88	STUFFING BOX BUSHING	359	SHELL
10B	INTERMEDIATE SHAFT	103	SELF-SEATING SNUBBER	360	SUCTION HEAD
10C	UPPER SHAFT	126	SHAFT NUT	361	DISCHARGE HEAD
14	SEAL CAGE	135	JOURNAL SLEEVE	397	RETAINING RING
16	SPLIT GLAND	164	SHAFT COUPLING	423	OUTER COLUMN
33	PUMP COUPLING HALF	227	BEARING	456	"O"-RING

Fig.12: Condensate pump - vertical, multistage, wet suction can type double-suction first-stage impeller.

10.2 Boiler Feed Pumps

At the heart of the power station preboiler systems is the boiler feed pump. This is a pump that must be reliable and rugged to withstand not only continuous normal high-pressure pumping operation but also transient system upset conditions and possible frequent starting and stopping to match the plant's load output requirements (cycling operation). In general terms, a boiler feed pump is any pump that supplies feedwater to a steam generator for the production of steam either for energy conversion (supply to a steam turbine) or for plant or other industrial uses. Boiler feed pumps are available in many pump configurations and sizes and supply the flow and head required for any application. However, this chapter describes only boiler feed pumps that are diffuser or volute, horizontal, double-case barrel, single- and double-suction first-stage impeller, multistage centrifugalpumps. This is the most common type of boiler feed pump used in central power station applications today. A typical boiler feed pump configuration is illustrated in Fig. 13.

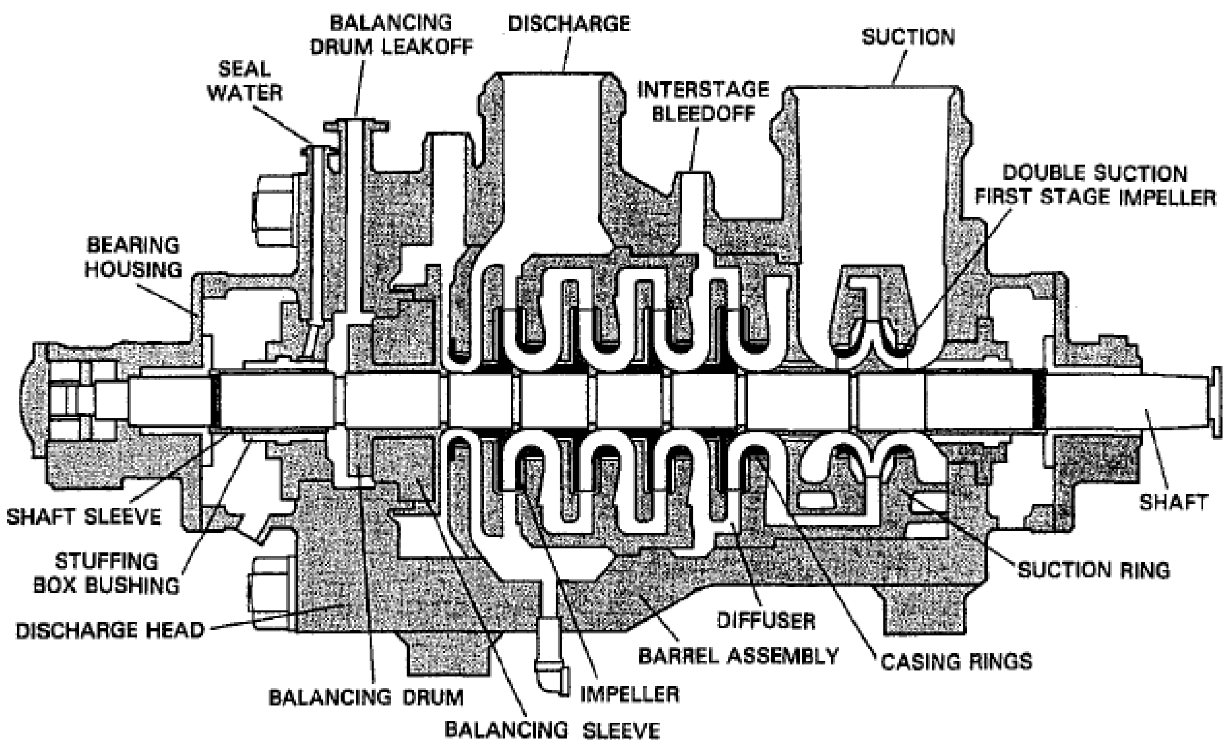


Fig. 13. Boiler feed pump—double case barrel pump, double-suction first-stage impeller, and multistage centrifugal type.

Pump Component	Boiler Feedwater Quality Typical Material Selections
Casing barrel	Forged carbon steel, ASTM A266, Class 1 or 2, or ASTM A105
Diffusers, volute cases, diaphragm, and stage pieces	12–14% chromium stainless steel, ASTM A743 or A487, Grade CA-15 or CA-6NM
Discharge head	Forged carbon steel, ASTM A266, Class 1 or 2, or ASTM A105, or ASTM A516 Pressure Vessel Plate
Shaft	12–14% chromium stainless steel, ASTM A276, Type 410, Condition T or ASTM A479, Type 410, Class 2
Impellers	12–14% chromium stainless steel, ASTM A743, Grade CA-15 or CA-6NM
Wearing rings	12–14% chromium stainless steel, ASTM A743, Grade CA-40, CA-6NM, or ASTM A276, Type 416 HT, AISI Type 410, 416, or 420

10.2.1 Boiler Feed Pump Systems

Many boiler feed pump system configurations have been used in power station applications. The basic system usually includes a deaerating heater and storage tank at some elevation above the suction of the boiler feed pump to provide a reservoir of heated, deaerated condensate to the boiler feed pump and available suction head for the pump. The suction pipeline to the boiler feed pump is amply sized to maintain the required suction velocity into the boiler feed pump, as well as limit the friction drop between the deaerator and the boiler feed pump. A booster boiler feed pump may be included to provide suction head to the main boiler feed pump. The booster pump may be motor-driven or driven by the main boiler feed pump driver through an extended shaft off the driver (usually a steam turbine) through reduction gearing. A single-suction first-stage impeller is usually used in the main boiler feed pump in this system. A conservative boiler feed pump system would also include a motor-driven startup boiler feed pump. This pump would be used primarily for plant startup activities, but could also be used to supplement the reliability and availability of the boiler feed system by operating in parallel with the main boiler pump at reduced plant loads or even at design load, depending on the capacity and head available from the startup boiler feed pump. The dischargE of the boiler feed pump should include pump recirculation instrumentation, valving, and piping back to the deaerator storage tank. Downstream, several stages of regenerative feedwater heating may be used, depending on the plant economics and cycle efficiency required. Some cycling plants also include a boiler feed system recirculation to the condenser for use during off-peak/standby operation to keep the boiler feed system warmed and poised for operation. A typical boiler feed pump systems are shown in Fig14.

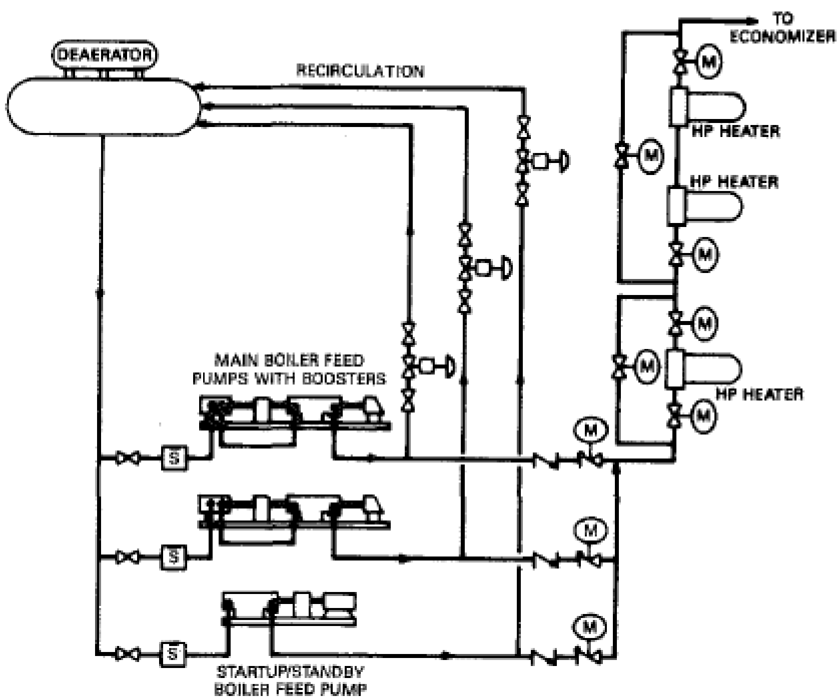


Fig. 14. Boiler feed pump system. Two half-capacity boiler feed pumps with boosters and startup/standby boiler feed pump.

10.2.2 Other Considerations of Boiler Feed Pump

Several references provide excellent recommendations and guidelines for boiler feed pumps and boiler feed pump installations. The reference of most significance for larger central power station boiler feed pump installations is *Recommended Design Guidelines for Feedwater Pumps in Large Power Generating Units* (1980), prepared by the Energy Research Consultants Corporation for EPRI. Another EPRI reference is *Survey of Feed Pump Outages* (1978), which also presents interesting boiler feed pump data that should be consulted by any boiler feed pump system designer or application engineer.

B. Compressors

1. Compressors - Introduction

1.1 Definitions and Objectives

Depending upon the pressure rise, three different terms are in use for the turbomachines that pressurize gases. If the pressure rise is very small (a few mm of water column) - it is termed a fan; up to 1 atm, it is usually called a blower; and above 1 atm it is commonly called a compressor. A compressor must operate within a system that is designed to acquire and compress a gas. Compressed air is used widely throughout industries. Almost every industrial plant, from a small machine shop to an immense pulp and paper mill, has some type of compressed air system. In many cases, the compressed air system is so vital that the facility cannot operate without it. In many industrial facilities, air compressors use more electricity than any other type of equipment. Inefficiencies in compressed air systems can therefore be significant. Energy savings from system improvements can range from 20 to 50 percent or more of electricity consumption. For many facilities this is equivalent to thousands, or even hundreds of thousands of dollars of potential annual savings, depending on use. A properly managed compressed air system can save energy, reduce maintenance, decrease downtime, increase production throughput, and improve product quality.

1.2 Need for Compressed Air System

The earliest compressors were bellows, used by blacksmiths to intensify the heat in their furnaces. The first industrial compressors were simple, reciprocating piston-driven machines powered by a water wheel. Compressed air systems consist of a supply side, which includes compressors and air treatment, and a demand side, which includes distribution and storage systems and end-use equipment. A properly managed supply side will result in clean, dry, stable air being delivered at the appropriate pressure in a dependable, cost-effective manner. A properly managed demand side minimizes wasted air and uses compressed air for appropriate applications. Improving and maintaining peak compressed air system performance requires addressing both the supply and demand sides of the system and how the two interact.

2. Components of an Industrial Compressed Air System

A modern industrial compressed air system is composed of several major sub-systems and many sub-components. Major sub-systems include the compressor, prime mover, controls, treatment equipment and accessories, and the distribution system. The compressor is the mechanical device that takes in ambient air and increases its pressure. The prime mover powers the compressor. Controls serve to regulate the amount of compressed air being produced. The treatment equipment removes contaminants from the compressed air, and accessories keep the system operating properly. Distribution systems are analogous to wiring in the electrical world—they transport compressed air to where it is needed. Compressed air storage can also serve to improve system performance and efficiency. Figure 1.1 shows a representative industrial compressed air system and its components.

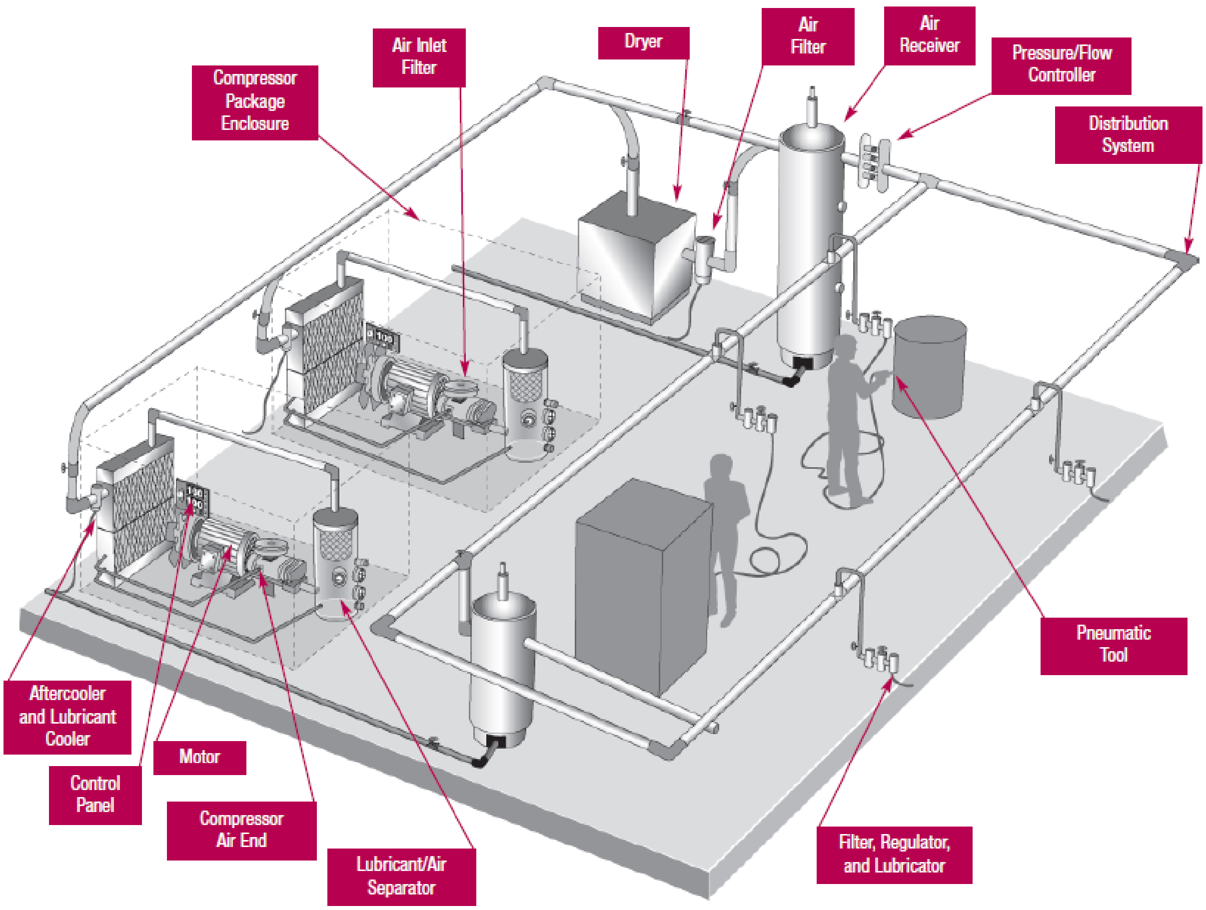


Fig. 1. Components of a typical industrial compressed air system

3. The Compression Process

There are two fundamentally different principles used to compress gases. Dynamic Compressors are continuous flow machines, they use rotating vanes or bladed discs to sequentially accelerate the gas (increasing its energy) then decelerate it (trading kinetic energy for increased pressure). This normally requires a number of stages, often within the same casing. Dynamic compressors always have an open gas route through the machine. Positive Displacement Compressors are discontinuous flow machines, they induce a fixed volume of gas into a pocket, chamber or cylinder for compression. The size of this pocket is then reduced mechanically, compressing the gas. At the end of the compression cycle the pocket opens, discharging the high-pressure gas. Figure 2 shows the compression process in a P-V diagram, while Fig. 3 explains working of a positive displacement compressor with the corresponding P-V diagram. Often only one or two stages of this compression process are required. There is never an open gas passage from delivery to suction (except for leakage through the clearances between moving parts.

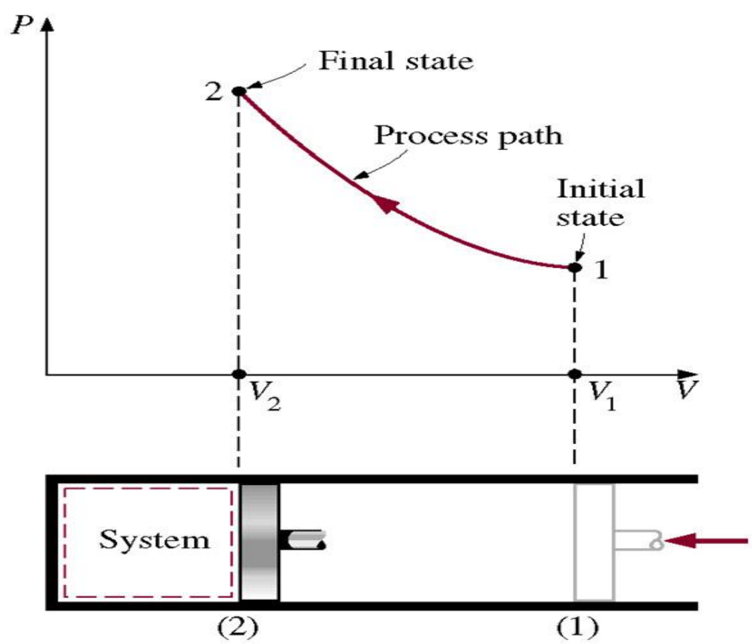


Fig 2. The P-V diagram of a compression process

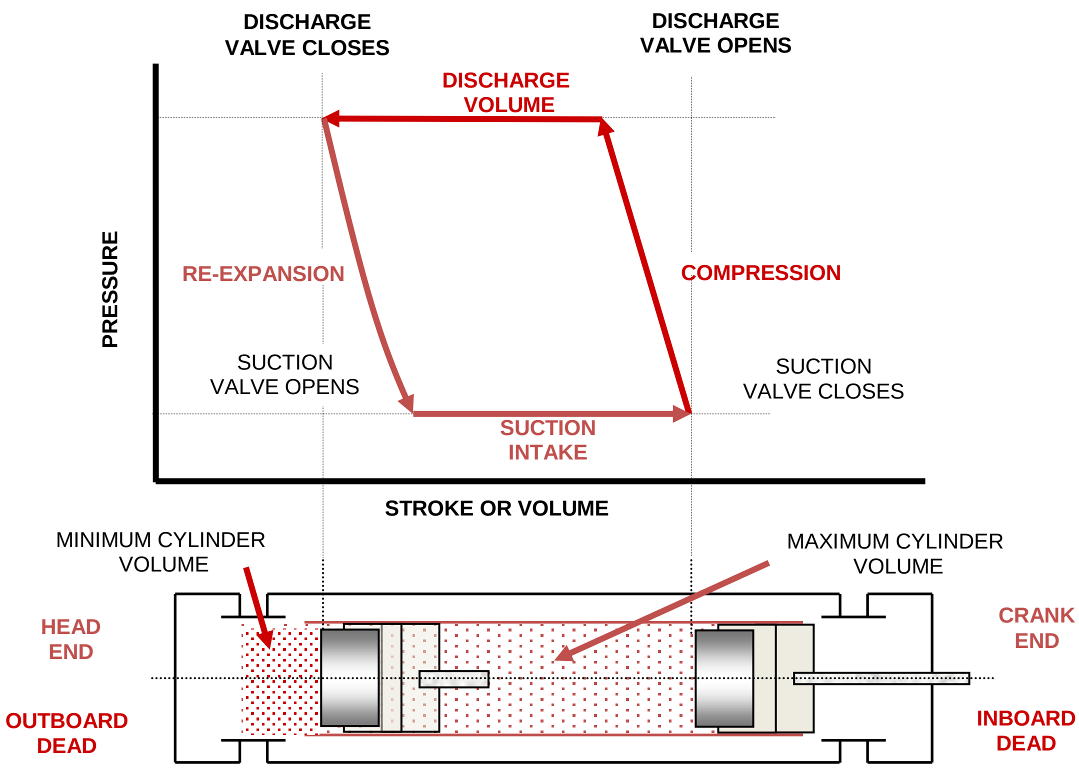


Fig 3. The working of a reciprocating compressor and the corresponding P-V diagram.

Dynamic compressor, for example centrifugal or axial compressors have relatively few moving parts, low vibration levels and thus high intrinsic reliability. Hence they are preferred over other compressor types where they can be used effectively. The centrifugal compressor (shown in Fig. 4) is characterized by the radial discharge flow. Air is drawn into the centre of a rotating impeller with radial blades and is thrown out towards the periphery of the impeller by centrifugal forces. Before the air is led to the centre of the next impeller, it passes a diffuser and a volute where the kinetic energy is converted to pressure.

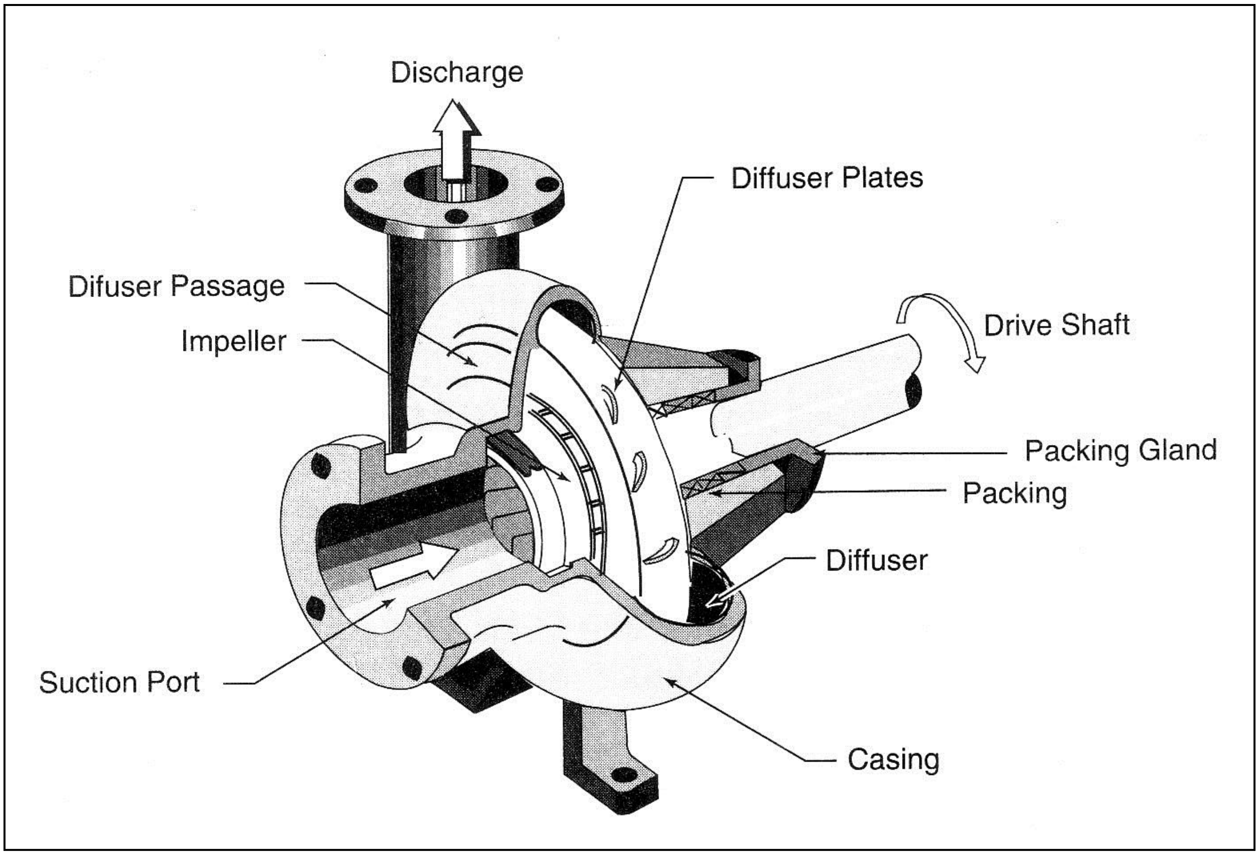


Fig. 4. Centrifugal compressor

4. Compressor Types

As shown in Fig. 5, there are two basic compressor types: positive-displacement and dynamic. In the positive-displacement type, a given quantity of air or gas is trapped in a compression chamber and the volume which it occupies is mechanically reduced, causing a corresponding rise in pressure prior to discharge. At constant speed, the air flow remains essentially constant with variations in discharge pressure. Dynamic compressors impart velocity energy to continuously flowing air or gas by means of impellers rotating at very high speeds. The velocity energy is changed into pressure energy both by the impellers and the discharge volutes or diffusers. In the centrifugal-type dynamic compressors, the shape of the impeller blades determines the relationship between air flow and the pressure (or head) generated.

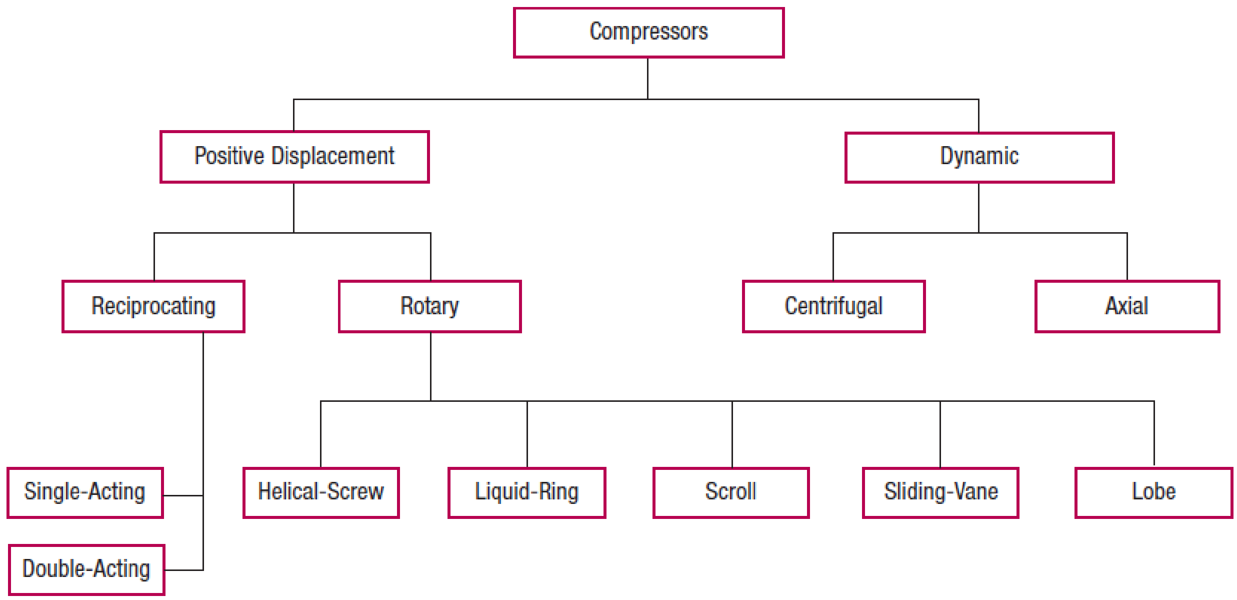


Fig. 5. Compressor family tree

5. Compressor Selection

Compressor selection is a complex and subjective process, with similar duties resulting in quite dis-similar compressors choices. The materials of construction must be able to take the mechanical loads; in addition those parts in contact with the process gas must be chemically compatible. Non-metallic materials are often used in seals and valves.

It is common practice, for dynamic compressors, to mount multiple compression stages on the same shaft within a common casing. For pressure ratios above perhaps 10: 1, and for discharge pressures above perhaps 20 bar g, a Barrel Type Multistage Centrifugal Compressor would be a reasonable selection.

Where a high-pressure ratio is required, different sizes of compressor, running at different speeds, may be linked to a single common driver. Gearboxes match the various shaft speeds.
To achieve reasonably practical shaft alignment and permit thermal expansion, flexible couplings are used between co-axial shafts.

Compressors require suitable piping, inter-stage vessels and coolers with associated control systems. Together with baseplate and driver this forms the "Compressor System". The vast majority of compressors are shaft driven by a separate electric motor, gas turbine or diesel engine. Thus the compressor will require at least one shaft seal, which may have to contain hazardous gas. The safety of compressors handling hazardous materials is dominated by their shaft sealing systems. These require appropriate design, maintenance and operator attention.

Air power compressors generally operate at pressures of 500 psig or lower, with the majority in the range of 125 psig or less. All major types of compressors (i.e. reciprocating, vane, helical lobe and dynamic, are used for this type of service. Choice is limited somewhat by capacity at 100 psig of about 10,500 ft3 per minute but can be built to approximately 28,000 cfm. The vane-type rotary has an upper listed size of 3700 cfm as a twin unit and the helical lobe rotary can be used to nearly 20,000 cfm. The centrifugal can be built to very large sizes. It is currently offered in the proven, moderate speed designs starting at a minimum of about 5000 cfm.

The following guidelines should be used for the selection process. While the criteria listed are not all inclusive, they will provide definition of the major considerations that should be used to select the best compressor for a specific application. *Application:* The mode of operation of a specific application should be the first consideration. The inherent design of each type of compressor defines the acceptable operating envelope or mode of operation that it can perform with reasonable reliability and life cycle costs. *Load factor:* is the ratio of actual compressed air output, while the compressor is operating, to the rated full-load output during the same period. It should never be 100 per cent, a good rule being to select an installation for from 50 to 80 per cent load factor, depending on the size, type and number of compressors involved. Proper use of load factor results in more uniform pressure, a cooling-off period, less maintenance, and ability to increase use of air without additional compressors. *Life cycle cost:* All capital equipment decisions should be based on the true or life cycle cost of the system. Life cycle cost includes all costs that will be incurred beginning with specification development before procurement to final decommissioning cost at the end of the compressor's useful life. In many cases, the only consideration is the actual procurement and installation cost of the compressor. While these costs are important, they represent less than 20 per cent of the life cycle cost of the compressor. The cost evaluation must include the recurring costs, such as power consumption, maintenance, etc. that are an integral part of day-to-day operation. Other costs that should be considered include training of operators and maintenance personnel who must maintain the compressor.

6. Reciprocating Compressors

The target duty is the production of oil-free air for distribution to instruments and other clean services. Reciprocating compressors are commonly used for small moderate air flow rates. Reciprocating compressors can achieve high pressure ratios per stage at low volume flows. They are used for smaller flows than screw and centrifugal compressors, and offer greater flexibility of duty than, in particular, centrifugal compressors. They are mechanically significantly more complicated than centrifugal compressors. Reciprocating compressors comprise sets of one or more compression cylinders, each with a matching piston. Process compressors are designated as Horizontal or Vertical design according to the orientation of the cylinder centre-lines. Service compressors (typically on instrument air duty) may have different orientations referred (as shown in Fig.6) to as V, W, L, but these complex options are not used on process units. Figure 7 shows Crankshaft, Connecting Rods, Crossheads on Typical Opposed Cylinder Machine. Compressors require suitable piping, inter-stage vessels and coolers with associated control systems. Together with baseplate and driver this forms the "Compressor System". The vast majority of compressors are shaft driven by a separate electric motor, gas turbine or diesel engine. A drive gearbox may be required to match the compressor and driver speeds. Reciprocating compressors are not normally variable speed as there are a number of ways to modify the output from such machines including: - reducing cylinder efficiency using clearance pockets, control on suction valve opening, and offloading cylinders.

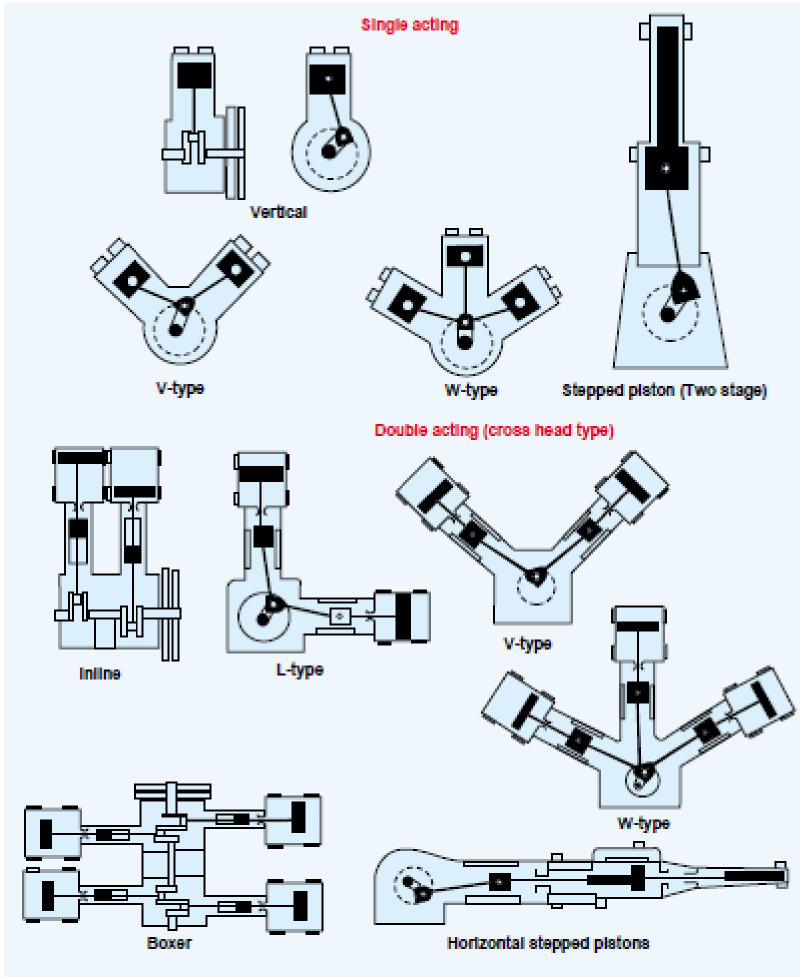


Fig.6. Examples of cylinder placement on piston compressors

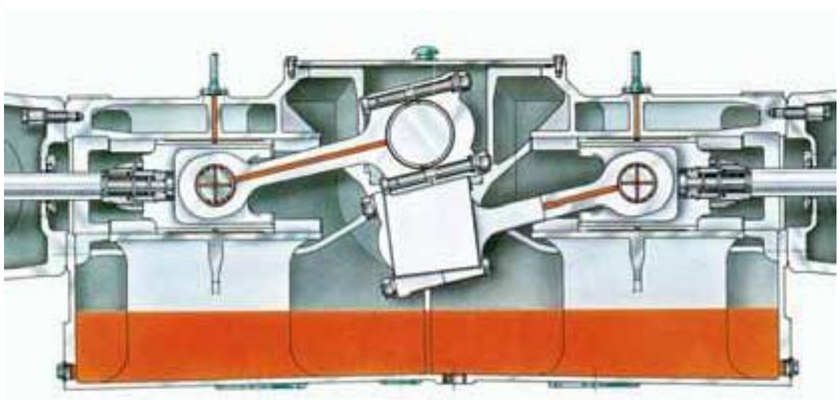


Fig. 7. Crankshaft, connecting rods, crossheads on typical opposed cylinder machine

Improvements in materials, seals and design principles have significantly improved the service life and reliability. Although centrifugal compressors can now reach the high pressures that previously required reciprocating compressors, low flow and flexible duties often require reciprocating compressors. Reciprocating compressors consist of a rotating crankshaft, linked to a number of piston rods by connecting rods. Each piston rod passes through a set of packed rod seals, into the compression cylinder. The rod carries the piston, which moves in the cylinder bore in a reciprocating manner. Pistons are normally double acting (compression on out and return strokes) Gas inlet and exhaust is via plate, ring or poppet type valves. The crankshaft runs inside an enclosed crankcase, in multiple bearings. The crankshaft end of the piston rod is also inside the crankcase, this is supported by a sliding "cross-head bearing". These bearings are oil lubricated, pressure fed by a pump. The crankcase atmosphere is air or, preferably, nitrogen to reduce fire risk. The trend for design of hydrocarbon compressors has been to a horizontal layout (to accommodate the long piston rod seal systems) with opposed pairs of cylinders (to optimize mechanical balance). Non-hydrocarbon compressors may be of vertical design, which has a much smaller footprint. A special type of vertical compressor is the "labyrinth piston" design, which has non-contacting piston seals. Improvements in materials of construction and seal design have permitted higher pressure and higher speed operation with reduced emissions. Typically, carbon steel and cast iron components are used for hydrocarbon duties, with non-metallic materials in seals and valves. Early machines were often direct driven at low speed by multi-pole electric motors. Sometimes the motor was directly mounted on the compressor shaft. These motors were very large, expensive, and not suitable for hazardous areas. Accordingly, modern compressors run faster, driven by standard motors. A drive gearbox is used, where necessary, to match speed.

6.1 Operation and Maintenance Issues

6.1.1 Equipment Hazards

The reciprocating action of the motion works imposes a variable torque on the crankshaft, the total effect is that most parts of a reciprocating compressor are subject to fatigue conditions at 1 or 2 cycles per revolution. A failure of the crankshaft, connecting rod or cross-head has the potential to breach the crankcase and eject major parts. This will release a quantity of lubricating oil and may permit back-flow of compressed air from the delivery manifold. Failure or partial failure of the crankcase could displace cylinders leading to very high vibration levels which can rupture pipework, flanged connections, and foundation bolting. Within the cylinder, loosening or failure of the piston nut or cross-head nut can cause the piston to be driven against the end of the cylinder, breaking the cylinder end or bending the piston rod. A bent or unsupported piston rod could then wreck the piston rod seals. Failure of a valve cover or valve retainer could cause ejection of a valve complete with cover.

Oil feed pipes to bearings are at relatively low pressures (3 - 5 bar g) but can drip or spray oil if damaged. The oil may pose personnel risk (toxicity, spray in eyes, or as a slipping hazard) or catch fire from a hot surface. Compressor metal temperatures are unlikely to be high enough, except as a result of ongoing bearing failure. A failed or failing crankshaft bearing can generate sufficient heat to ignite oil. Opening a crankcase cover can let fresh air contact hot, oily, metal, provoking a fire or explosion, procedures should allow cooling of the machine so that by the time the equipment has been isolated and covers removed the hazard is no longer present. Sets should not be run with crankcase covers loose or partly bolted, as these could be blown off. It is good practice to fit crankcase explosion relief valves. A compressor can be driven backwards e.g. by incorrect electrical connections to the motor. The compressor will run and will compress air, but the bearings and lubrication system are not intended for reverse running and will be damaged. The shaft driven oil pump, if fitted, may well not work if run backwards.

6.1.2 Operational aspects

Reciprocating compressor with oil lubricated cylinders either by direct injection of oil to the piston or lubrication from crankcase have a particular hazard associated with oil air explosions. The hazard is present when a combination of carbon contamination and the oil air mixture becomes too hot initiating an explosion in the machine or discharge system (pipes and vessels).

Stringent procedures to limit compression temperatures (normally less than 160 C), correct choice of oil (low carbon formation), and inspections for carbon build up are needed to avoid such events.

6.1.3 Maintenance issues

The standards of maintenance for reciprocating compressor must be maintained at the highest level. Reciprocating compressors are vulnerable to incorrect mechanical standards, these can lead to rapid deterioration and failure of components leading to potential for disruption of the machine. Examples of this are:

- Correct assembly and fitting of the self-actuating machine valves is crucial to machine operation. Though such valves are normally designed to be unique to a particular position and duty on the machine it may still be possible to confuse the installation. In such cases this can cause the machine to be gas locked with catastrophic results. Machines returning from external maintenance may require preservative actions to be reversed and the removal of dessicant bags from valve ports is also vital.
- Incorrect bearing clearances – on crankshaft this has led to failure of the crankshaft and major machine disruption.
- Incorrect set up for piston rod connections either for cylinder or crosshead can cause complete failure of piston rod.
- Wrong maintenance standards applied to recover wear on piston rods have caused rod failure.
- Wrong maintenance standards applied to recover wear on crankshafts have resulted in failure of crankshaft.

These compressors are maintained in situ, requiring effective lifting facilities and laydown areas. Compressor control panel and ancillaries may be located on the same skid, or adjacent, or in a control room.

- **Reciprocating air compressors are generally designed for periods of steady operation, typically with an annual overhaul. Valves and seals require service at perhaps 3 or 6 monthly intervals.**
- **Reciprocating compressors are designed to be maintained in situ. Seals and valves can be removed and replaced as assemblies. Major parts e.g. cylinders, crankshaft, require lifting facilities.**

Maintenance requirements for reciprocating compressors are a combination of preventative maintenance of the motion work and major components and essential predictive maintenance based on measured performance such as valve temperatures, vibration. Valves, rod seals, bearings can be accessed by removal of local covers. Removal / maintenance of seals and bearings require lifting equipment to support and position motion work as required. The pistons and rod are normally removed as an assembly by unscrewing the cross-head nut. Only then can the rod seal and the piston rings, be inspected and changed. Reciprocating compressors are thus relatively maintenance-intensive, reducing the availability compared to screw compressors.

7. Centrifugal Compressor

Dynamic compressors always have an open gas route through the machine unlike Positive Displacement Compressors which are discontinuous flow machines. Dynamic compressors have relatively few moving parts, low vibration levels and thus high intrinsic reliability. Hence they are preferred over other compressor types where they can be used effectively. It is common practice, for dynamic compressors, to mount multiple compression stages on the same shaft within a common casing. For pressure ratios above perhaps 10 : 1, and for discharge pressures above perhaps 20 bar g, a Barrel Type Multistage Centrifugal Compressor (shown in Fig 8) would be a reasonable selection. Where a high-pressure ratio is required, different sizes of compressor, running at different speeds, may be linked to a single common driver. Gearboxes match the various shaft speeds. To achieve reasonably practical shaft alignment and permit thermal expansion, flexible couplings are used between co-axial shafts. Compressors require robust base-plates to carry shaft torques and piping loads without excessive distortion.

Fig.8. Barrel type multistage centrifugal compressor

8. Screw Compressor

Screw compressors require very special machine tools to produce the complex shapes of the rotors to the required accuracy. Once these tools were available, screw compressors became commonly available as air and gas compressors. They are generally less efficient than reciprocating compressors, but are much easier to install and maintain. Oil flooded and oil-free ("dry") screw compressors are available to suit the different markets. Service air is normally provided by the cheaper and more rugged oil flooded designs. Screw compressors have been developed in a range of sizes to suit most Instrument Air demands, particularly as good practice often calls for several compressors working in parallel to provide flexibility and reliability. A stage in an oil-free screw compressor is shown in Fig. 9. Male and female rotors are journalled in the rotor housing, which here is water-cooled. The front rotor, with four lobes, is the male, this is connected to the gearbox. The distant rotor, with six lobes, is the female, this is held in place by the synchronizing gear to the left.

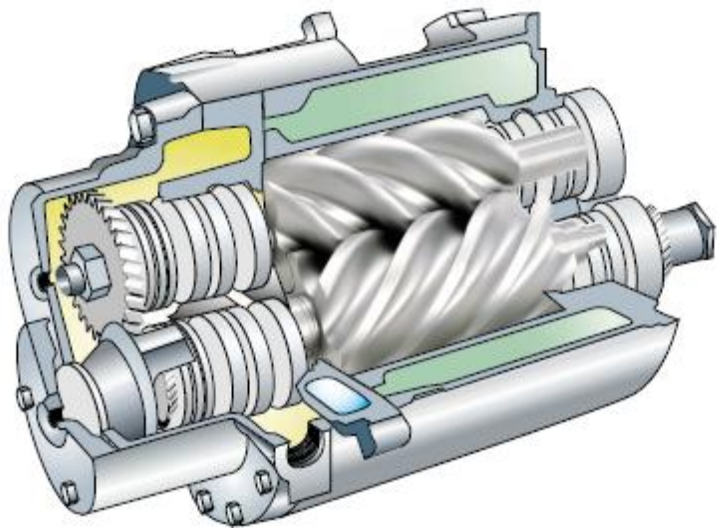


Fig. 9. Screw compressor

The first screw compressors had a symmetric profile and did not use liquid in the compression chamber, so-called oil-free or dry screw compressors. At the end of the 1960s a high speed, oil-free screw compressor was introduced with an asymmetric screw profile. The new rotor profile resulted in significantly improved efficiency, due to reduced internal leakage. An external gear is used in dry screw compressors to synchronize the counter rotating rotors. As the rotors neither come into contact with each other nor with the compressor housing, no particular lubrication is required in the compression chamber. Consequently the compressed air is completely oil-free. The rotors and housing are manufactured with great precision to minimize leakage from the pressure side to the inlet. The integrated pressure ratio is limited by the temperature difference between the intake and the discharge. This is why oil-free screw compressors are frequently built with several stages.

9. Ancillaries

Protective systems are fitted to protect against machine damage through loss of oil pressure, or excessive temperatures. Each compressor stage should have its own relief valve. Equally, the relief valves must be tested. Reciprocating air compressors have cooling jackets around the cylinders, oil cooler and inter / after-coolers. It is convenient to use a circulated fluid, normally closed circuit fresh water, as the coolant. Heat can then be exchanged against air or, more compactly, water. Air cooling requires a significant cooling air supply and a significant radiator area as the heat to be lost equates to the compressor motor power. Oil-flooded screw compressors lose most of their heat directly into the oil, which can then be cooled against air or water.

9.1 Lubrication system

The lubrication system has two basic functions: to lubricate the compressor's moving components and to cool the system by removing heat from the compressor's moving parts. While all compressors must have a lubrication system, the actual design and function of these systems will vary depending on compressor type. The lubrication system of a dry reciprocating compressor is fully self-contained and should not come into significant contact with the compressed air. Hence the oil should stay clean and can be sampled and checked for contamination e.g. from damaged bearings. An oil pump, filter and oil cooler are normally fitted, to supply clean cool oil to the bearings and crosshead. The main oil pump is often shaft driven, for simplicity. A small motor driven pre-lubrication pump may be fitted. Loss of oil pressure must trip the compressor.

The lubricating system for centrifugal or dynamic compressors is designed to provide bearing lubrication. In smaller compressors, the lubrication systems may consist of individual oil baths located at each of the main shaft bearings. In larger compressors, such as a bullgear design, a positive system is provided to inject oil into the internal, tilting-pad bearings located at each of the pinion shafts inside the main compressor housing. In positive lubrication systems, a gear-type pump is normally used to provide positive circulation of clean oilwithin the compressor. In some cases, the main compressor shaft directly drives this pump. In others, a separate motor-driven pump is used. Positive displacement compressors use their lubrication system to provide additional functions. The lubrication system must inject sufficient quantities of clean fluid to provide lubrication for the compressor's internal parts, such as pistons and lobes, and to provide a positive seal between moving and stationary parts. The main components of a positive displacement compressor's lubrication system consist of an oil pump, filter, and heat exchanger. The crankcase of the compressor acts as the oil sump. A lockable drain cock is installed at the lowest end of the crankcase to permit removal of any water accumulation that has resulted from sweating of the crankcase walls. The oil passes through a strainer into the pump. It then flows through the heat exchanger, where it is cooled. After the heat exchanger, the cooled oil flows directly to the moving parts of the compressor before returning to the crankcase sump. A small portion is diverted to the oil injector if one is installed. The oil that is injected into the cylinder seals the space between the cylinder wall and the piston rings. This prevents compressed air from leaking past the pistons, and thus improves the compressor's overall efficiency. The oil pump is usually gear driven from the crankshaft so that it will start pumping oil immediately on start-up of the compressor. In compressors that work in an oil-free system, oil injectors are not used. Oil separators are installed on the discharge side after leaving the after-cooler. The basic purpose of an oil separator is to clean the pressurized air of any oil contamination, which is highly detrimental to pneumatically controlled instrumentation. A separator consists of an inlet, a series of internal baffle plates, a wire mesh screen, a sump, and an outlet. The pressurized air enters the separator and immediately passes through the baffle plates. As the air impinges on the baffle plates it is forced into making sharp directional changes as it passes through each baffle section. As a result, the oil droplets separate from the air and collect on the baffles before dropping into the separator's sump. After the air clears the baffle section, it then passes through the wire mesh screen where any remaining oil is trapped. The relatively oil-free air continues to the air reservoir for storage. The air reservoir acts as a final separator where moisture and oil is eventually removed. The air reservoir has drain traps installed at its lowest point where any accumulated moisture/oil is automatically discharged. As a part of any routine maintenance procedure, these discharge traps should periodically be manually bypassed to ensure that the trap is functioning, and no excessive water accumulation is evident.

9.2 Air Receivers and Safety Valves

Air reservoirs are designed to receive and store pressurized air. Air receivers, being simple volume tanks, are not often thought of as highly engineered items, but the use of simple engineering with receivers can reduce equipment costs. A pertinent example not infrequent in industry is the intermittent requirement for fairly large volume of air at moderate pressure for a short period of time. Some boiler soot blowing systems are in this class. The analysis necessary to arrive at the most economical equipment often involves the storage of air at high pressure to supplement the compressor's output when the demand requires. Air reservoirs are classified as pressure vessels and have to conform to the ASME Pressure Vessel Codes. As such, the following attachments must be fitted:

- Safety valves
- Pressure gages
- Isolation valves
- Manhole or inspection ports
- Fusible plug

Pressure regulating devices are installed to maintain the pressure within operational limits. When the air reservoir is pressurized to the maximum pressure set point, the pressure regulator causes the air compressor to off-load compression by initiating an electrical solenoid valve to use lubricating oil to hydraulically hold open the low pressure suction valve on the compressor. As the compressed air is used, the pressure drops in the reservoir until the low-pressure set point is reached. At this point, the pressure regulated solenoid valve is de-energized. This causes the hydraulic force to drop off on the low-pressure suction valve, restoring it to the full compression cycle. This cycling process causes drastic variations in noise levels. These noises should not be regarded as problems, unless accompanied by severe knocking or squealing noises. All compressed air systems that use a positive displacement compressor must be fitted with a pressure relief or safety valve that will limit the discharge or inter-stage pressures to a safe maximum limit. Most dynamic compressors must have similar protection due to restrictions placed on casing pressure, power input and/or keeping out of surge range. Two types of pressure relief devices are available, *safety valves* and *relief valves*. Although these terms are often used interchangeably, there is a difference between the two. Safety valves are used with gases. The disk overhangs the seat to offer additional thrust area after the initial opening. This fully opens the valve immediately, giving maximum relief capacity. These are often called *pop-off safety valves*. With relief valves, the disk area exposed to overpressure is the same whether the valve is open or closed. There is a gradual opening, the amount depending upon the degree of over-pressure. Relief valves are used with liquids where a relatively small opening will provide pressure relief.

9.3 Cooling System – Intercooler and after-cooler

A water-based cooling system is expected to cool the circulating jacket water. The cooling water (clarified water) will be supplied from a utility manifold and returned to open drain. A pump will be required to circulate the jacket water. This pump should be interlocked to the main motor starter. An air cooling system will require a plentiful supply of clean air from a safe location, this will normally be the same as the air supply for the compressor air intake. A ventilation fan will be included in the compressor package, but additional fan(s) may be required if the air inlet and outlet are restricted by ducts. The amount of moisture that air can hold is inversely proportional to the pressure of the air. As the pressure of the air increases, the amount of moisture that air can hold decreases. The amount of moisture air can hold is also proportional to the temperature of the air. As the temperature of the air decreases, the amount of moisture it can hold decreases. The pressure change of compressed air is larger than the temperature change of the compressed air. This causes the moisture in the air to condense out of the compressed air. The moisture in compressed air systems can cause corrosion, water hammers, and freeze damage. Therefore, it is important to avoid moisture in compressed air systems. Coolers are used to address the problems by moisture in compressed air systems. Coolers are frequently used on the discharge of a compressor. These are called after-coolers, and their purpose is to remove the heat generated during the compression of the air. The decrease in temperature promotes the condensing of any moisture present in the compressed air. This moisture is collected in condensate traps that are either automatically or manually drained. If the compressor is of the multi-stage type, there may be an intercooler, which is located after the first stage discharge and second stage suction. The principle of the intercooler is the same as the principle of the after-coolers, The combinations of drier compressed air (which helps prevent corrosion) and cooler compressed air (which allows more air to be compressed for a set volume) is the reason the air coolers are worth the investment.

9.4 Dry Air Distribution System

Most plants are highly dependent upon their compressed air supply and it should be assured that the air is in at least reasonable condition at all times. It is possible that the line condensation would be so bad that some air applications would be handicapped or even shut down if there were no protection. A vital part of the entire endeavor to separate water in the conventional compressed air system is also the trapping of dirt, pipe scale and other contaminates. This is still necessary with a dried air system. As a minimum, all branch lines should be taken off the top of the main and all feeder lines off the top of branch lines.

9.5 Air Drying Plant - Absorption Drying

Air entering the first stage of any air compressor carries with it a certain amount of native moisture. This is unavoidable, although the quantity carried will vary widely with the ambient temperature and relative humidity. In any air–vapor mixture, each component has its own partial pressure and the air and the vapor are each indifferent to the existence of the other. It follows that the conditions of either component may be studied without reference to the other. In a certain volume of mixture, each component fills the full volume at its own partial pressure. The water vapor may saturate this space or it may be superheated. As this vapor is compressed, its volume is reduced while at the same time the temperature automatically increases. As a result, the vapor becomes superheated. More vapors are now contained in one cubic meter than when originally entering the compressor. Under the laws of vapor, the maximum quantity of a particular vapor a given space can contain is dependent solely upon the vapor temperature. As the compressed water vapor is cooled, it will eventually reach the temperature at which the space becomes saturated, now containing the maximum it can hold. Any further cooling will force part of the vapor to condense into its liquid form – water. It is evident that the lower the temperature and the greater the pressure of compressed air, the greater will be the amount of vapor condensed.

Absorption drying is a chemical process, where water vapour is bound to the absorption material. The absorption material can either be a solid or liquid. Sodium chloride, sulphuric acid or silica gel is frequently used, which means the possibility of corrosion must be taken into consideration. Figure shows the drying system and in the diagram the left tower dries the compressed air while the right tower regenerates. After cooling and pressure equalization the towers are automatically switched.

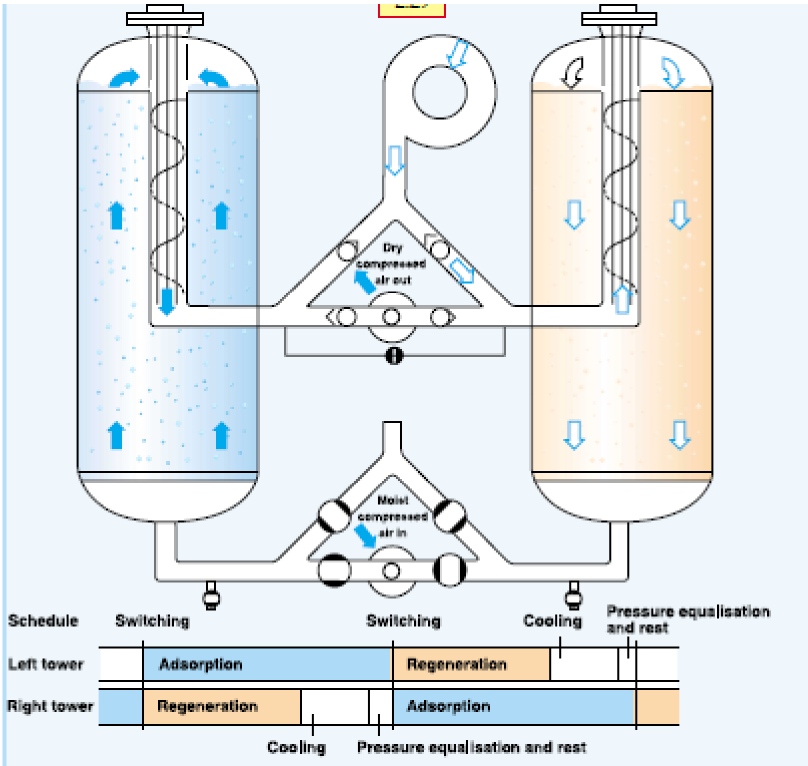


Fig. 10 Compressed air drying plant

10.Performance Analysis of the Compression Process

The actual compression and the ideal (isentropic) processes are indicated in h-s and T-s diagrams in Fig. 11 and Fig.12 respectively. The actual work done is always greater than the ideal work to irreversibilities involved in the practical process. The compressor efficiency is defined as ratio of isentropic to ideal work.

$$\eta_C = \frac{w_s}{w_a} \cong \frac{h_{2s} - h_1}{h_2 - h_1}$$

Well-designed compressors have isentropic efficiencies in the range from 75 to 85 percent. Figure 13 shows the compression process with intercooling in P-v and T-s diagram.

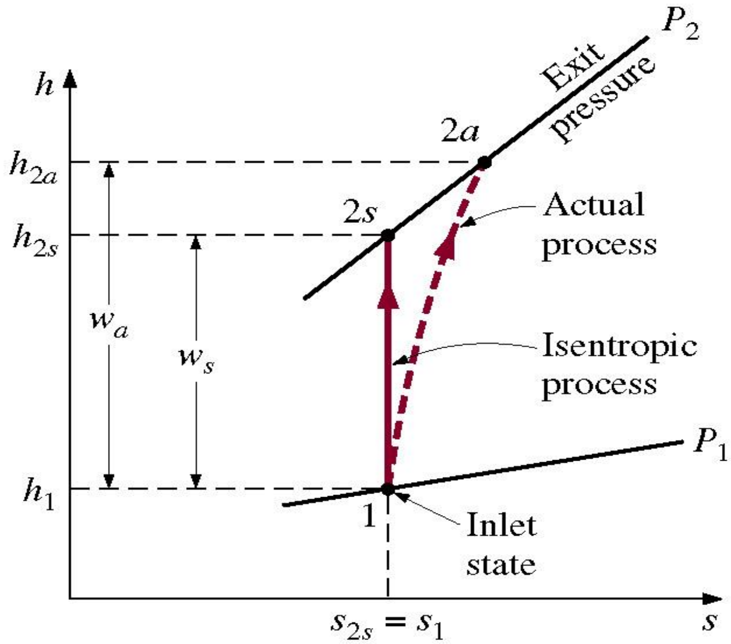


Fig. 11. The isentropic and actual compression process in h-s diagram

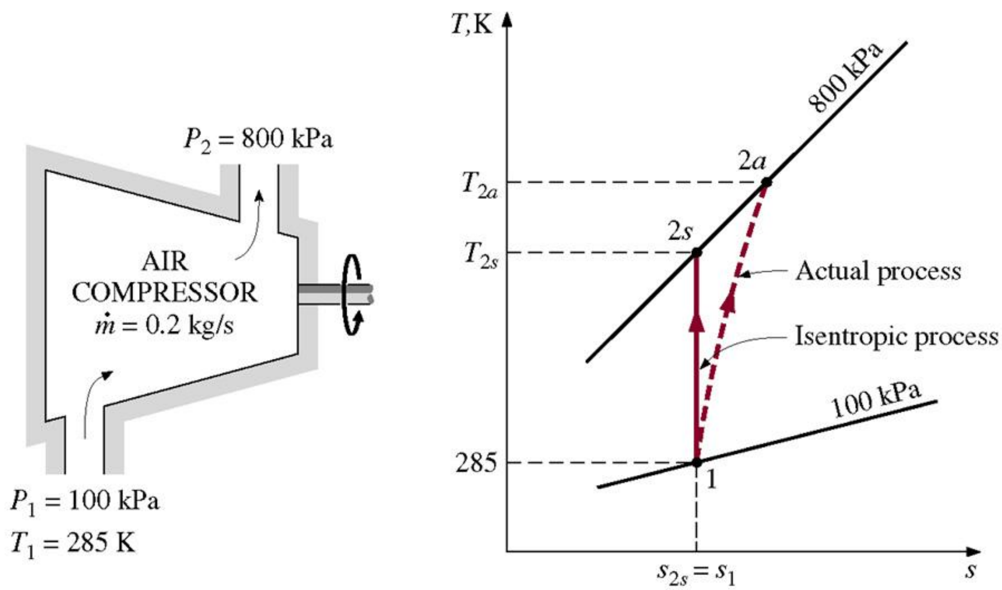


Fig. 12. The isentropic and actual compression process in T-s diagram

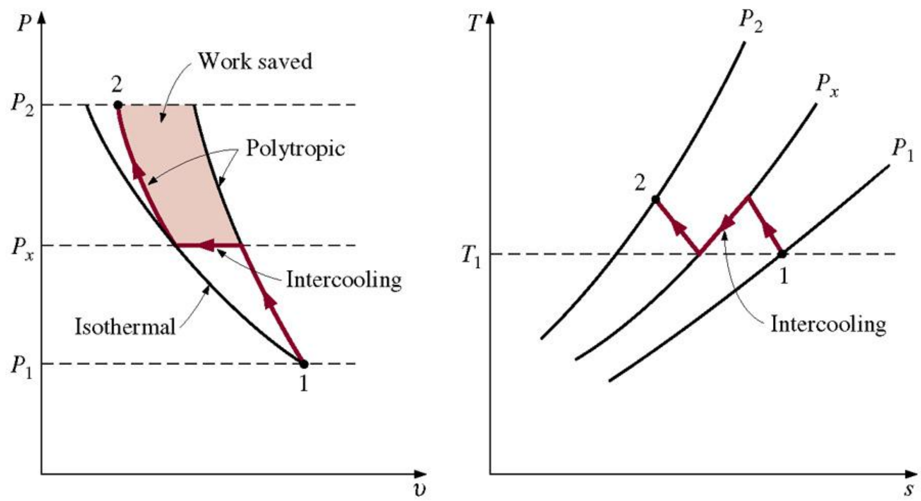


Fig.13. The compression process with intercooling.

The shaded region in the P-v diagram in Fig. 13 represent worked saved due to intercooling in between compression stages. This is due to the reason that the isothermal process requires minimum work and incorporating intercooling, the compression process is tending towards the isothermal one.

C. Seals

1. Seals – Introduction

A mechanical seal prevents leakage of pressurized fluid between a rotating shaft and a stationary housing. They are widely used for numerous plant equipment applications, of various sizes and pressure ratings. Problems with mechanical seals represent a significant reliability impact on fluid machineries. Even though they are capable of providing long term service, mechanical seals sometimes exhibit unsatisfactory performance, unpredictable failures, and a short life, which can directly affect plant reliability and performance, resulting in vital equipment downtime and plant outages.

2. Sealing Mechanism and Seal Elements

A mechanical face seal is a dynamic seal that prevents leakage of pressurized fluid. Mechanical face seals are available in a variety of configurations, and their selection depends on the application. However, no matter what the application is, all mechanical face seals operate on the same principle. Basically, the seal is comprised of two rings, either of which rotates relative to the other. One of the rings is usually mounted rigidly and the other is mounted so that it can flex and align axially and angularly with the rigidly mounted ring. Dynamic sealing is achieved at the interface between the two rings, the primary ring and the mating ring. The rings achieve a seal at the interface due to their very high face flatness. Typically, the two rings are made of dissimilar materials. The essential elements of a mechanical face seal are illustrated in Fig.1. These elements serve the functions of sealing dynamically and statically, loading the faces, and transmitting rotation to the ring.

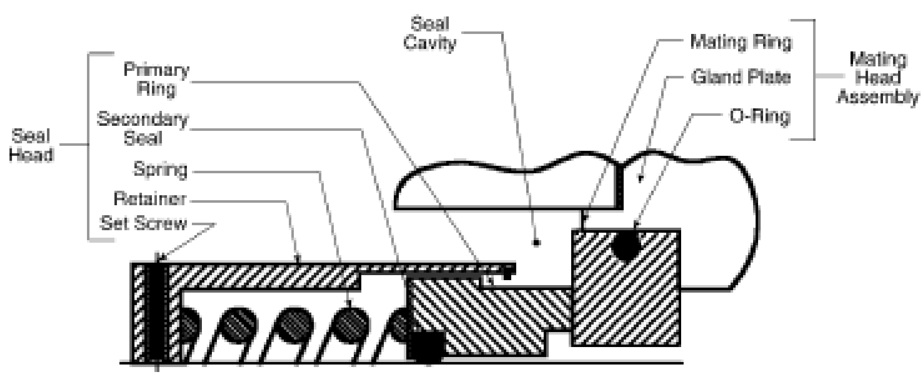
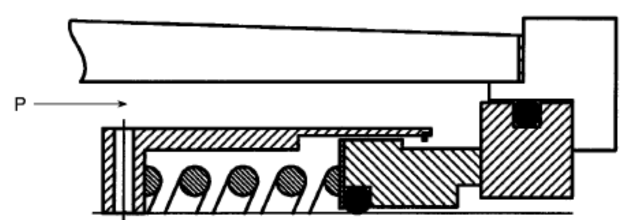


Fig.1. Components of a mechanical seal

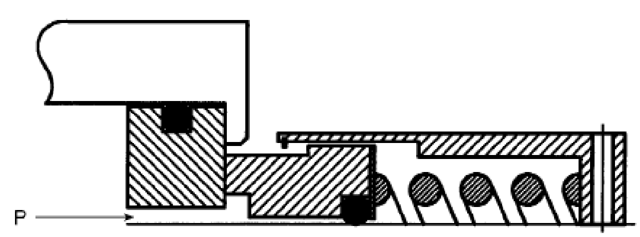
3. Various configurations of seal elements

Mechanical face seals come in a variety of configurations, materials, and designs for primary sealing faces, secondary seals, springs, drive mechanisms. Options also include unbalanced or balanced designs, whether the primary seal or the mating seal is rotating, and whether the fluid pressure is on the outside or the inside surface of the seal. Seal design for a given application should be selected after a careful evaluation of trade-offs.

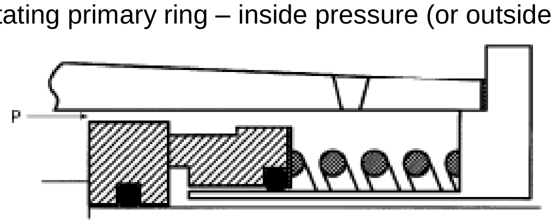
Primary Ring*:* The primary ring is also called a seal ring. The primary ring is the floating seal element that is usually spring-mounted and permits axial and angular alignment in the assembly. Depending on the application requirements, it can be either the rotating member as shown in Fig.2 a and b, or the stationary member as shown in Fig.2 c and d. The method in which the primary ring is mounted is dictated by the application requirements because each configuration offers both advantages and disadvantages. The mechanical face seal design or style is defined by the primary ring configuration, that is, rotating primary ring, stationary primary ring, double seal, bellows seal, and so on.



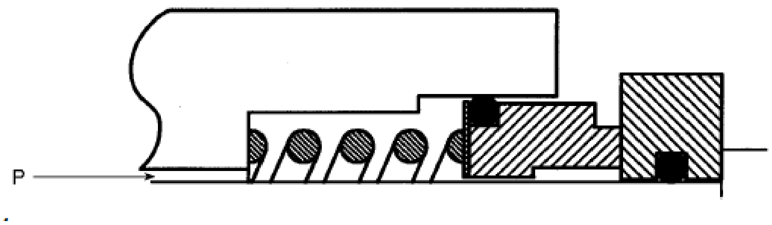
(a) Rotating primary ring – outside pressure (inside mounted)



(b) Rotating primary ring – inside pressure (or outside mounted)



(c) Stationary primary ring – outside pressure (or inside mounted)



(d) Stationary primary ring – inside pressure (or outside mounted)

Fig.2 Stationary and rotating primary rings with inside or outside pressure.

Mating Ring: The mating ring is also called a seat or seal seat. The mating ring is the rigidly mounted element and can be installed in the housing as shown in Fig.1 or on the shaft as shown in Fig.2. Where the mating ring is installed is dependent upon the application requirements and the preferred implementation of the primary ring.

Secondary Seal: Seals used to prevent leakage through paths alternative to that between the seal faces. The secondary seals can be static or dynamic. Static secondary seals prevent leakage between assembled parts that are not subject to relative motion in service, for example, between seal sleeve and shaft, between stationary seal member and housing. Dynamic secondary seals prevent leakage between the shaft or housing and the floating seal member. The type of secondary seal depends on the fluid type, service pressure, and service temperature.

Spring: Springs are used to develop the contact load between the primary ring and the mating ring in the absence of fluid pressure. The amount of face load generated can vary significantly depending on the type of spring selected. The choice includes a single coil spring, multiple coil springs, metal bellows, non-metal bellows, wave or Belleville washer, and magnets. In some cases, such as bellows, the spring can serve both the face-loading function and the secondary sealing function.

4. Drive Mechanism

All mechanical face seals require some kind of device to position the primary ring axially and to transmit the rotation of the shaft to the primary ring to ensure that relative motion occurs only at the seal faces. The drive mechanism is designed such that it is not rigidly attached to the primary ring so that it does not prevent self-alignment between the primary ring and the mating ring. The drive mechanism is typically a setscrew, locking collar, key, or wedge ring. In some designs, the secondary seal is used to transmit the torque to the primary ring when sufficient friction can be developed at the secondary seal interface. The drive mechanism is also used to provide torque restraint to the stationary seal if the static secondary seal does not develop sufficient friction to prevent the stationary seal from turning.

5. Seal/Flushing Chamber

An area around the seal is provided to permit heat transfer through the fluid and to allow flushing of contaminants such as abrasive particles or toxic media. In a single seal configuration, flushing is accomplished by injecting a liquid into the seal chamber at a higher pressure than the sealed product.

6. Multiple Seals

Some applications require the use of multiple seals to provide for flushing or barrier fluids, or pressure staging to deal with higher pressures. Flushing is used to remove contaminants, to cool the faces, or to provide for proper lubrication. This is achieved by installing the seals in a back to- back or face-to-face configuration, as illustrated in Fig. 3 and 4. For cooling and solids/abrasives removal, fluid can be re-circulated from the product side or provided by an external source. In applications where the product has a relatively low vapor pressure, for example, water or hydrocarbons, a barrier fluid with a higher vapor pressure is used to keep the product from vaporizing at the seal interface and to prevent the inboard seal from running dry. If the product is toxic or harmful, a clean barrier fluid is introduced at a higher pressure to minimize toxin release. The outboard seal also provides a back-up in case of failure of the product seal.

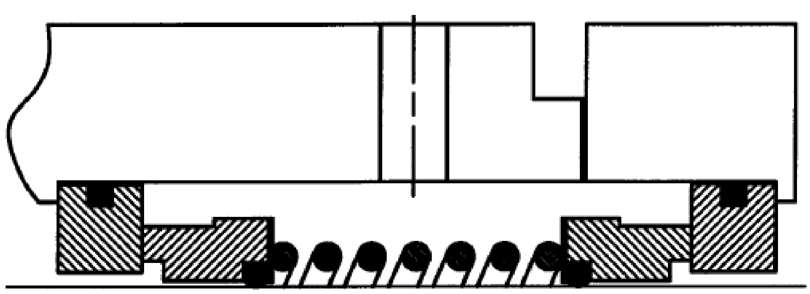


Fig.3. Back-to-Back Dual Seal

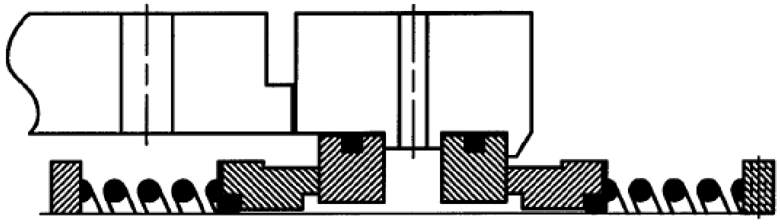


Fig.4. Face-to-Face Dual Seal

7. Seal Cartridge

Seal cartridges are pre-assembled mechanical face seal assemblies that contain all of the essential components. Cartridges are used to package mechanical face seals for ease of handling and installation. An example of a single seal cartridge is shown in Fig.5. In this arrangement, the primary ring and its associated devices are mounted on a sleeve temporarily attached to the enclosure that holds the mating ring. The assembly provides for proper spring loading and axial positioning of the primary ring and mating ring. After the cartridge is mounted on the housing and the sleeve is secured to the shaft, the temporary attachment device holding the sleeve to the mating ring enclosure is removed. Cartridges can be provided with either rotating primary rings or stationary primary rings and with single or multiple mechanical face seals. The schemes for assembling cartridges vary from design to design.

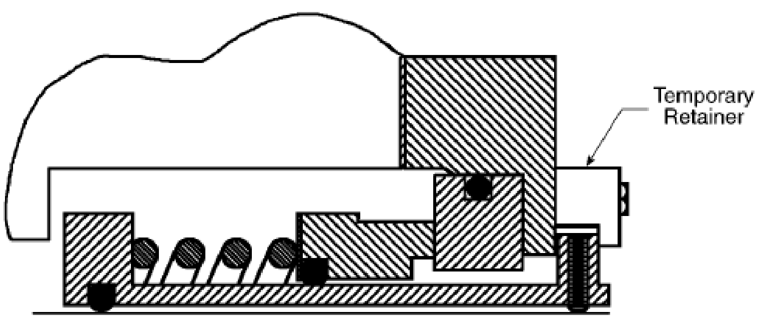


Fig.5. Single Seal Cartridge

8. Seal Arrangements for Abrasive Applications

Abrasives will generally cause rapid wear of the faces while excessive heat from the pumped fluid, or as a result of seal friction, will damage the elastomers and distort seal components, causing the seal to leak and fail. The seal should be provided with a clean, relatively cool, abrasive-free flush to lubricate and remove the heat generated by the seal faces and to prevent flashing at the seal faces. A clean liquid from an outside source can be used. However, the resulting contamination of the pumped product by an external source might make this type of flush undesirable. For this reason, a re-circulated or bypass fluid from the liquid being pumped is frequently used. If necessary, this re-circulated flush fluid can be cooled and any abrasive particles removed before it is injected into the seal. When multiple seals, as shown in Fig.3 and 4, are used, a combination of internal and/or external seal flush arrangements can be used. In severe abrasive duty applications (for example, clinker grinder in fossil plants and abrasive slurry handling pumps), mechanical face seals have a history of unreliable performance and short life, even when flushing arrangements are used. This is due to the fact that, in addition to exposure to harsh abrasive particles, seals are exposed to large shaft deflections (both static and dynamic), frequent starts/stops, transients, shock, and vibration, which exceed the capabilities of face seals. Similar sealing problems in downhole drilling applications have been solved by an alternative elastomeric seal design employing hydrodynamic lubrication. This design might be a potential solution to the fossil plant slurry handling equipment and sealing problems where application conditions are unsuitable for mechanical face seals.

9. Some Key Technical Points

Balancing: Mechanical face seals can be unbalanced, fully balanced, or partially balanced to reduce the face loading due to hydraulic pressure. The term balanced refers to the case where the average pressure load on the face is less than the sealed pressure. Most mechanical face seals have a balance ratio between 0.65 to 0.85. This range provides reduced face loading without potential concern of face parting.

Pressure distribution: Pressure distribution across the seal face width can be linear, concave, or convex and it can change with variations in pressure, temperature, and seal wear. This can affect seal performance (leakage, torque, temperature) during operation.

Temperature considerations: For satisfactory performance, the seal design and material selections should satisfy the PV limit and the 'T limit under all operating conditions to ensure that fluid film is maintained between the seal faces. Loss of film can lead to immediate seizure and seal failure.

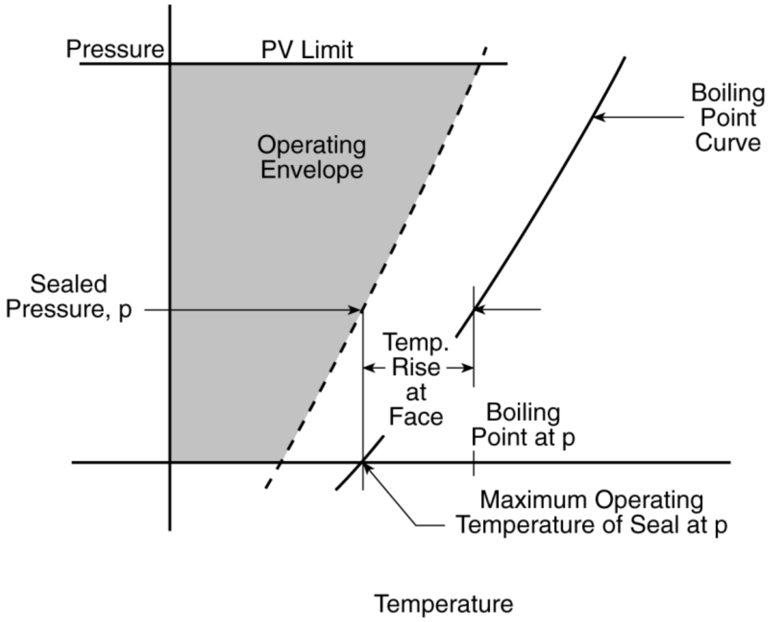
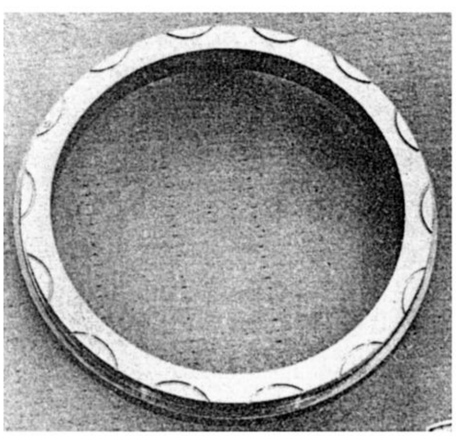
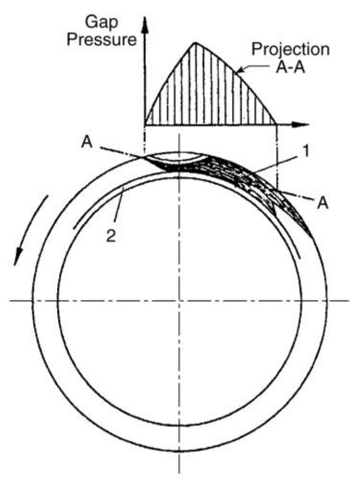


Fig.6. Pressure/Temperature Operating Envelope Showing ΔT Margin Required for Seal Operation

Improved Seal Design: Seal designs with special features to enhance lubrication at the sealing interface (for example, as shown in Fig.7, hydrodynamic grooves, recesses, or laser-textured surfaces) can extend the pressure, speed, and temperature limits. The tradeoff (for example, higher leakage rate versus increased reliability under transient conditions) should be carefully evaluated during seal selection.



(a) Face Geometry



(b) Hydrodynamic Action

Fig.7. Seal Face with Thermal Hydrodynamic Grooves for Positive Hydrodynamic Lubrication

Hydrostatic Seal Design : The hydrostatic seal design (as shown in Fig.8) is a non-contacting mechanical face seal that permits some controlled flow rate to pass between the faces. To prevent dry running, the seal requires that some pressure be applied to the tapered side prior to rotation.

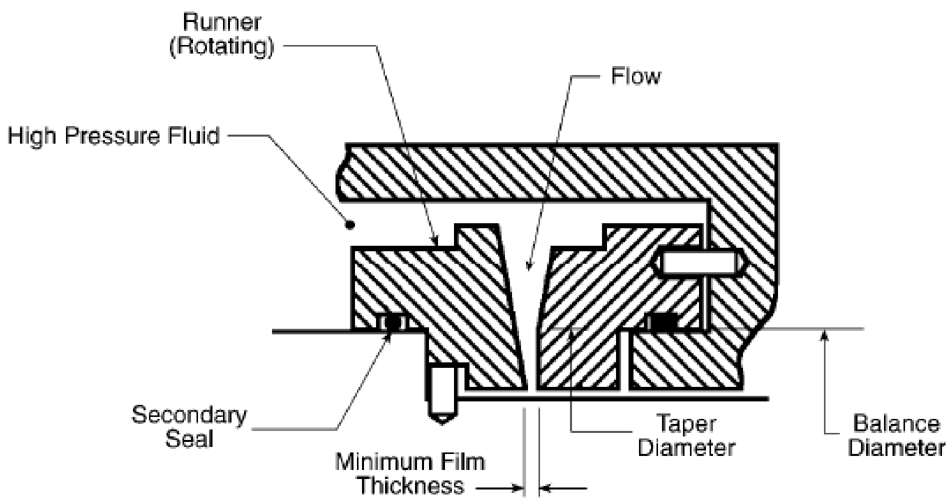


Fig.8. Hydrostatic Face Seal

10. Seal Failure, Causes and Remedial Strategies

The eventual failure mode of all mechanical face seals is leakage that is considered unacceptable for the seal design/configuration being used. Excessive leakage can cause unacceptable loss of fluid, reduction of pressure, or contamination of the system fluid by the barrier fluid in double seal installations. The level of acceptable leakage is dependent upon the application.

For satisfactory performance, the seal design and material selections should satisfy the PV limit and the ΔT limit under all operating conditions to ensure that fluid film is maintained between the seal faces. Loss of film can lead to immediate seizure and seal failure.

Mechanical seals are often installed in the same cavity that is designed to accept conventional packings. This limits the fluid circulation around the seal, leading to high seal temperatures and accumulation of solids. An enlarged seal chamber with tapered bore can dramatically improve fluid circulation, lowering seal temperature and eliminating accumulation of solids.

Operation away from Best Efficiency Point (BEP) is a frequent cause of short seal life/seal failures. Off BEP conditions cause large shaft deflections and vibrations resulting in premature degradation of mechanical seals.

Static and dynamic misalignment between seal faces can cause strong fluid pumping action across the faces causing either inward pumping or outward pumping of the product fluid and/or buffer fluid. Leakages under misaligned conditions can be several times the normal leak rate.

The most cost-effective maintenance program should be based on predicted seal performance and its expected life. The least cost-effective maintenance program is one based on reactions to failure. An effective preventative or periodic maintenance program, based on plant experience and manufacturer recommendations, should be implemented to improve plant reliability and prevent unplanned shutdowns.