Pumps

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Introduction

Pump Designs and Operation

Learning Outcome

When you complete this learning material, you will be able to:

Describe the designs, principles, components and operating procedures for common industrial pumps.

Learning Objectives

You will specifically be able to complete the following tasks:

1. Explain the principle of operation and describe the components of typical piston and plunger reciprocating pumps.
2. Explain the designs and operating principles of the external gear, internal gear, sliding vane, lobe, and screw type rotary pumps.
3. Explain the designs and operating principles of volute and diffuser centrifugal pumps, including impeller designs.
4. Describe centrifugal pump arrangements, including vertical, horizontal, single and double suction, opposed impellers, multi-staging, split and barrel casings.
5. Describe the design and applications of axial and mixed flow pumps.
6. Describe the design and components of a multistage centrifugal pump, clearly stating the purpose and general design of: wearing rings, shaft sleeves, seals, bearings and lubrication components, vents and drains.
7. Explain design features that eliminate thrust in large centrifugal pumps.
8. Describe a complete piping system for a multistage centrifugal pump, including the purpose and design of the automatic recirculation (minimum flow) valve.
9. Explain proper priming, start-up procedures and operating considerations for pumps

Introduction:

Pumps are used in oil field to transfer the fluids (Liquids) form a place to another.

It is a mechanical device to increase the pressure energy of a fluid.

Generally pumps are used tom lift the fluid from low level to high level.

There are Two Main Types:

1- Reciprocating pumps (which are used for developing very high pressure) or head at low or medium flow.

2- Centrifugal pumps, which are used for supplying very large quantities at low or medium pressure.

Objective One

When you complete this objective you will be able to...

Explain the principle of operation and describe the components of typical piston and plunger reciprocating pumps.

Learning Material

Positive Displacement Pumps (Principle of operation and design)

1- Principle Of Operation

1.1 **The positive displacement pump** works in the same way as putting a block in a bucket of water. The block will' displace a volume of liquid equal to the volume of the block. As a plunger moves into a cylinder, it displaces a volume of liquid. The plunger's displacement is positive. A pump that displaces a constant volume of liquid is called a positive displacement pump. This pump first traps a volume of liquid- in the cylinder. The piston or plunger then displaced a volume of liquid from the . cylinder by moving forward. If liquid is displaced on the forward stroke only, the pump is single acting. If the pump displaces liquid on both the forward stroke and the return stroke, it is a double-acting pump.

1.2 **Reciprocating pumps** are classified by the number of cylinders Simplex means one ; duplex means two; triplex means three. A multiplex pump is any pump with more then one cylinder. Positive displacement pumps are required when the pump must operate at low flow rate and high head or when the material being handled is 'viscous.

1.3 **The principle of the reciprocating pump** depends Bon valves. Liquid enters the cylinder through suction valves on the back stroke and is displaced through discharge valves on the' forward stroke. The cylinder valves act as check valves, permitting flow only in one direction. Pressure differences control the action of the valves.

At the beginning of the forward stroke, both cylinder valves are closed. As the piston moves forward, cylinder pressure forces the discharge valve open and liquid is displaced from the cylinder. As the piston moves back, it leaves a partial vacuum in the cylinder. This low cylinder pressure permits the 'suction valve to open and the discharge valve closes. A volume of liquid enters the cylinder on the backstroke. If . the cylinder fills completely, this intake volume will equal the volume that was displaced on the pistons forward stroke.

1.4 **The same mechanics** are employed in a double-reciprocating pump. On the backstroke, liquid is displaced from the crank end of the cylinder. At the Sam time, a volume of liquid enters the head end of the cylinder. The rod reverses direction at each end of each stroke. The flow of liquid from a reciprocating pump is a **pulsating flow**, Discharge pressure is more constant if the pump is double-acting. The discharge pressure is even more constant if the pump is multiplex.

Table 1

Pump Selection Guide

Feature	Horizontal	Vertical
Space Requirements	Less Head Room	Less floor area, more head room
NPSH	Requires more	Requires less
Priming	Required*	Required*
Flexibility (Relative to future changes)	Less	More
Maintenance	More Accessible	Major work project
Corrosion and Abrasion	No great problem	Can be considerable problem
Cost	Less	More (requires more alloy to handle corrosive Fluid)

- For some conditions

Table 2

Type Selection Based on Liquid Handled

Liquid	Basic Pump Type	Type Impellers
Water and other clear non-corrosive liquids at cold or moderate temperatures	Single or double suction	Closed except for very small capacities
Water above 250º F	Single or Double suction. This is usually boiler feed service at high pressure requiring multi-stage pump	Closed except for very small capacities
Hydrocarbons, hot	Single suction, often of the special type called refinery pumps, designed particularly for high temperature service	Closed with large inlets
Corrosives: Mildly acid or alkaline Strongly acid or alkaline Hot corrosives	Single or double suction Single or double suction with single suction probably less expensive if available for the rating Single suction with many refinery pump types also used here because of high temperatures and corresponding suction pressures.	 Closed except for very small capacities or where liquid tem\ns to form scale on surfaces of moving parts
Water with solids in suspension: Fine abrasives	Single suction with end clearance wearing fits. If all particles pass through 1/8 “ mesh screen, rubber lined pumps are available which will give many times the life of metal pumps. providing no chemical action or excessive tem pressure will deteriorate the rubber. Special rubber compounds can be applied to	Open, which allows better application of the rubber, except in larger sizes. Also made in closed type

Coarse abrasives	improve resistance to certain chemicals. Single suction, Not available for full range of ratings, that is, small capacities not too easily obtained . Often have very large impellers operated at slow for used when solids larger than 1" diameter at the standard diet. This would be of the type calied dredge pumps handling sizable rocks.	Closed
Pulpy solids such as paper stock	Single suction, Double suction only used on very slight solids concentrations and then with special end clearance wearing fits.	Closed . Open type used to be standard but change to end clearance wearing fits made closed impellers better suited.

Table 3
Pump Materials of Construction
Table materials are for general use, specific service experience is preferred when available

Liquid	Casing & Wear Rings	Impeller Wear Ring	Shaft	Shaft Sleeves	Types of Seal	Seal Case	Gland	Remarks
Ammonia, Anhydrous & Aqua	Cast Iron	Cast Iron	Carbon Steel	Carbon Steel	Mechanical		Mall, Iron	NOTE: Materials of Construction shown will be revised for some jobs.
Benzene	Cast Iron	Cast Iron	Carbon Steel	Nickel Moly. Steel	Ring Packing		Mall, Iron	
Brine (Sodiu	Ni-Resist*	Ni-Resist	K Monel	K Monel	Ring Packing		Ring Packing	* Cost Iron

m Chloride)		*					**	acceptable. ** Malleable Iron acceptable
Butadiene	Casing :C Steel- Rings C.I	Impeller: C.I- Rings: C. Steel	Carbon Steel	13% Chrome Steel	Mechanical		Carbon Steel	
Carbon Tetrachloride	Cast Iron	Cast Iron	Carbon Steel	Carbon Steel	Carbon Steel	Mechanical	Mall, Iron	
Caustic, 50% (Max. Temp. 200º F)	Micro C	Micro C	18-8 Stainless Steel	Micro C	Ring Packing		Carbon Steel	Micro C manufactured by Michigan Steel Casting Company 29-Cr-9 Ni-Stan lesssteel or equal
Caustic , 50% (Over 200º F) & 73%	Nickel	Nickel	Nickel or 18-8 Stanless steel	Nickel	Ring Packing		Nickel	
Caustic 10% (with some sodium chloride)	Cast Iron	23% Cr. 52% Ni Stanless Steel	23% Cr. 52% Ni Stanless Steel	23% Cr. 52% Ni Stanless Steel	Ring Packing	Cast Iron	----------	Specifications for 50% Caustic (Maximum Temperature 200ºF) also used.

Ethylene	Cast Steel	Carbon Steel	Carbon Steel	Carbon Steel	Mechanical	Cast Iron	Mall, Iron	
Ethylene Dichloride	Cast Iron	Cast Iron	Steel	K Monel	Mechanical	-----------------	K Monel	
Ethylene Glycol	Bronze	Bronze	18-8 Stanless steel	18-8 Stanless steel	Ring Packing	----------------	Bronze	
Hydrocarbon Acid 32% (Alternate)	Rubber Lined C. Iron	Hard Rubber	Carbon Steel	Rubber or Plastic	Ring Packing	Rubber	Rubber	
Methyl Chloride	Cast Iron	Cast Iron	18-8 Stanless steel	18-8 Stanless steel	Mechanical		Mall, Iron	
Propylene	Casing :C Steel-Rings C.I	Impeller: C.I-Rings: C. Steel	Carbon Steel	Carbon Steel	Mechanical	Cast Iron	Mall, Iron	

Table 3 (contd)
Pump Materials of Construction
Table materials are for general use, specific service experience

Liquid	Casing & Wear Rings	Impeller Wear Ring	Shaft	Shaft Sleeves	Types of Seal	Seal Case	Gland	Remarks
Sulfuric Acid, Below 55%	Hard Rubber	Special Rubber	Carbon Steel	Hastelloy C	Ring Packing	Special Rubber	Special Rubber	
Sulfuric Acid, Below 55 to 95 %	Cast Si-Iron	Si-Iron	Type 316 Stn-Stl	Si-Iron	Ring Packing	Teflon	Si-Iron	
Sulfuric Acid,	Cast Iron	Cast Iron	Carbon Steel	13 % Cr	Mechanical	------------	Mall, Iron	

Above 95%						---		
Styrene	Cast Iron	Cast Iron	Carbon Steel	13 % Chrome Steel	Mechanical	Cast Iron	Mall, Iron	
Water River	Cast Iron	Bronze	18-8 Stanless steel	Bronze	Mechanical	Cast Iron	Mall, Iron	
Water, Sea	Casing 1-2% Ni Cr 3-0.5 % Cast Iron Rings Ni-Resist 2B	Impeller: Monel Rings : S-Monel	K Monel (Aged)	K Monel or Alloy 20 SS	Ring Packing	K Monel or Alloy 20 SS	K Monel or Alloy 20 SS	

is preferred when available

2- Plunger Pumps

The basic design of the plunger type of reciprocating pump is shown in Fig. 1. When the plunger starts moving from right to left, the pressure in the cylinder drops below that in the suction line and the liquid is drawn into the cylinder via the suction ball check. The high pressure in the discharge line keeps the discharge ball check firmly on its seat. At the end of its travel the plunger reverses direction and starts moving from left to right. This causes the pressure in the cylinder to rise above that in the discharge line and the liquid is forced out via the discharge ball check, while the suction ball check is forced to close.

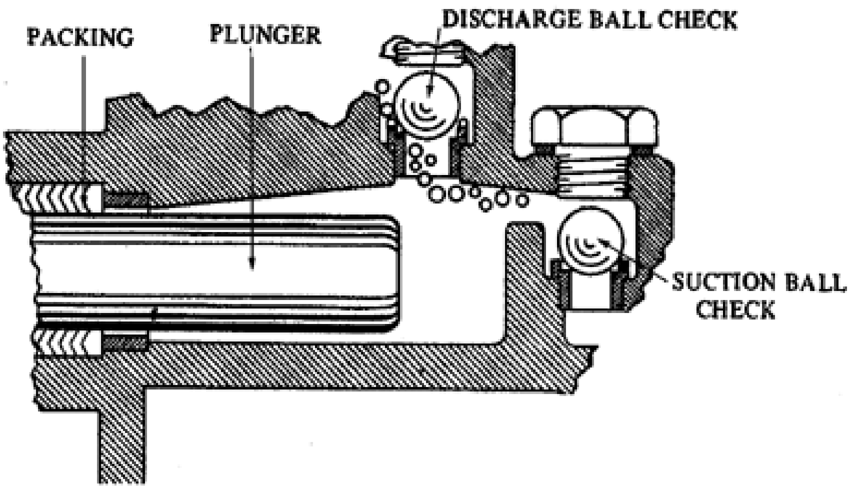

Figure 1
Plunger Pump (Single-Acting)

The movement of the plunger in one direction is called the stroke of the plunger. The distance the plunger moves in and out of the cylinder is the *length of the stroke*. As can be seen in the sketch, only one side of the plunger takes part in the pumping action and liquid is only discharged during one out of every two strokes. Hence, the pump is called single acting Because of its design a plunger pump cannot be *double-acting*.

Fig. 2 shows a cross-sectional view of a power driven plunger pump. This pump is widely used as a chemical feed pump. It has an adjustable stroke for volume control and a double set of suction and discharge ball checks in step arrangement for high-pressure applications.

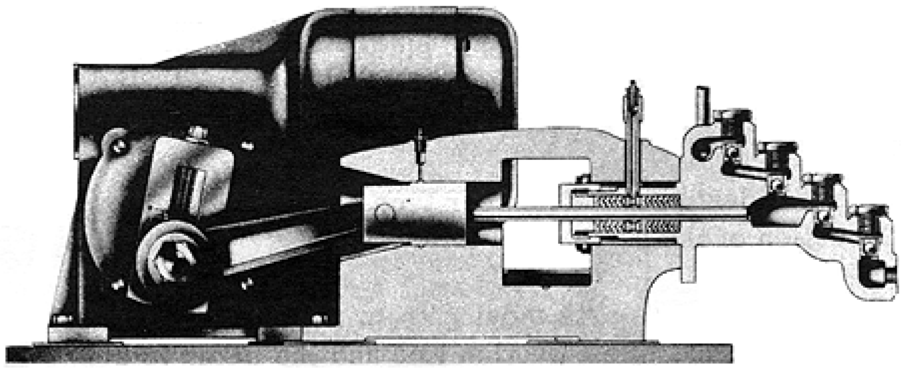

Figure 2
Power Driven Plunger Pump Cross-Section

Fig. 3 shows a large capacity triplex plunger pump. As the name implies, it has three cylinders. The pump is single acting and a crankshaft drives its plungers with three cranks placed 120 degrees apart.

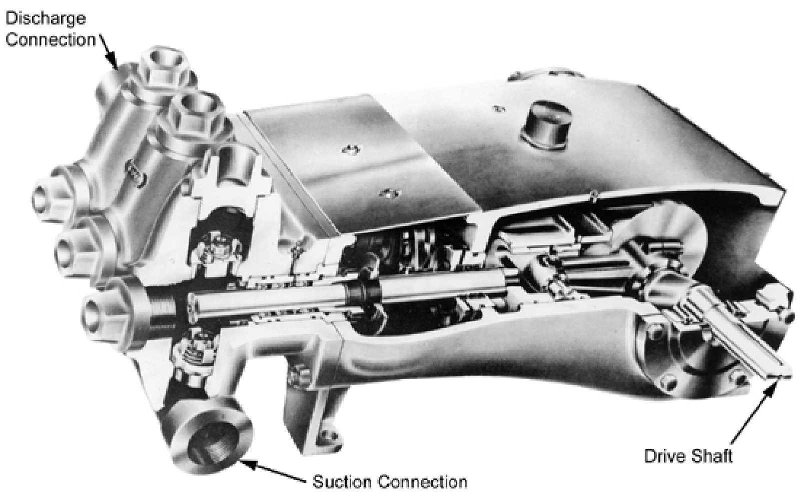

Figure 3
Power Driven Triplex Pump

A plunger has almost the same diameter as its cylinder and it extends beyond the cylinder. A piston has almost the same diameter as its cylinder but does not extend beyond the cylinder. Consequently, if a piston rod is used rather than a connecting rod, the drive end of the cylinder can be closed and work can then be done on both sides of the piston.

3- Piston Pumps

Piston pumps may be either single acting or double acting. The basic operating principle of a double-acting piston pump is shown in Fig. 4. The pump has two discharge valves D A, and D B, and two suction valves, S A and S B. When the piston moves from left to right, (a) the liquid is drawn into the left side of the cylinder via suction valve S A. At the same time the piston forces liquid out of the right side via discharge valve D B.

Discharge Outlet Discharge Outlet

Suction Inlet

Suction Inlet

(A) (B)

Figure 4
Double-Acting Piston Pump

When the piston reverses direction and moves from right to left, (b) liquid is drawn into the right side of the cylinder via suction valve S B and discharged from the left side via discharge valve DA. With the above arrangement, both sides of the piston take part in the pumping action and liquid is discharged when the piston moves in either direction, hence the name “double-acting”

4- Diaphragm Pump

The diaphragm pump differs from the piston or plunger types of pumps in that the pumped fluid is completely isolated by the diaphragm from the reciprocating mechanism thereby eliminating leakage along piston rod, plunger, and/or piston. The diaphragm is a flexible membrane, which acts as the liquid displacing component. The diaphragm can be made of flexible metal or non-metallic materials such as plastic, rubber or neoprene, depending on the fluid being pumped.

A cross-sectional view of a mechanically actuated diaphragm pump is shown in Fig. 5. The diaphragm D is attached to the piston guide P by the disc B. An eccentric is used to produce reciprocating motion of the guide P, causing the diaphragm to move to and fro, resulting in pumping action.

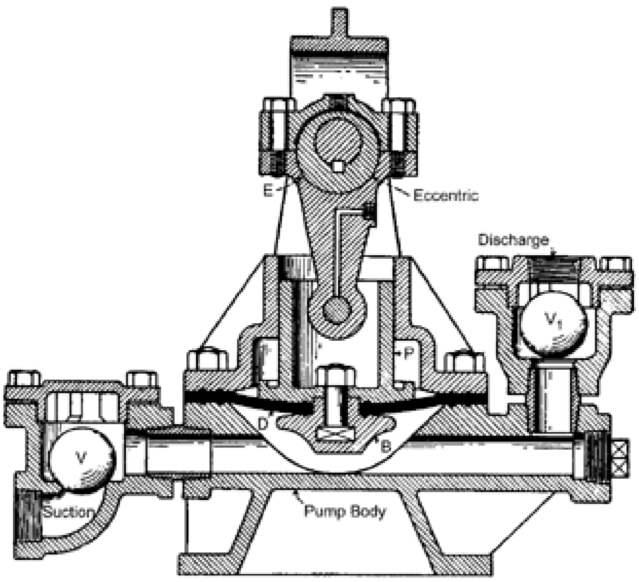

Figure 5
Mechanically Actuated Diaphragm

The amount of liquid pumped per stroke by a reciprocating pump is determined by the area of the piston, plunger or diaphragm and the length of the stroke, or in other words, by the volume displaced per stroke. This amount remains constant regardless of discharge pressure. For this reason, the reciprocating pump is classified as a *positive displacement pump*. The pump capacity depends on the number of pumping strokes per minute.

4- Cryogenic Pumps

BNI manufactures a complete line of cryogenic pumps that can be modified to handle any cryogen a most operating conditions. Our cryogenic pumps provide quiet, reliable performance at low NPSH and most utilize a hermetic design to eliminate shaft seals and fluid leakage. WE can design and built cryogenic pumps for low, high head application, high flow, and low head application.

Design Features

Hermetic

Maintenance problems in many manufacturers' pumps are due to bearing and seal failures. Most BNI pumps eliminate the troublesome mechanical seal by utilizing a hermetic design that incorporates the motor and pump within the same housing. The picture at right shows a sectional view of a typical BNI hermetic cryogenic pump that incorporates the long shaft feature. We also have an extensive background in bearing application in severe environments utilizing both fluid film or rolling element bearings. head/flow conditions.

Long Shaft

In cryogenic applications it is often critical that heat dissipated into the cryogen be kept at an absolute minimum. BNI has designed a complete line of hermetic pumps that utilize a long shaft between the motor and pump to minimize motor and ambient heat from transferring into the pumped cryogen.

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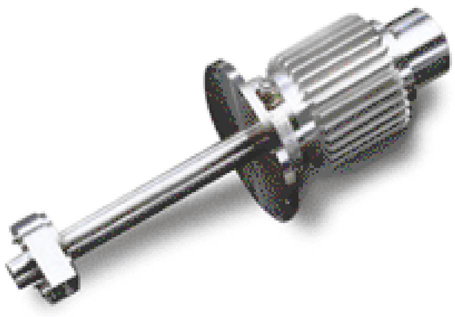


Figure 6 Long Shaft

Cold Box Vacuum Sustaining Design

One of the most unique design features BNI has developed provides for maintenance of all pump components while cold box vacuum is maintained. The picture at right shows the motor, mounting flange and rotating assembly for one of these designs that can be removed from the system while the lower casing and connections remain in line. This feature can result in substantial savings to the user in many cryogenic applications.

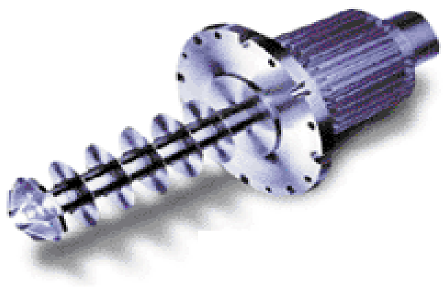


Figure 7 Cold Box Vacuum

Completely Submersible Units

Utilizing a completely submersible pump in which the motor is immersed in the cryogen is very advantageous in many applications. Barber-Nichols manufactures cryogenic submersible pumps that perform well at temperatures as low as 4 K and will endure the most rigorous start-stop conditions. These designs are inherently small in size and highly

Figure 8 Submersible

Applications

1. Cryogenic cooling loops
2. Superconductor cooling loops
3. De-stratification
4. Liquid transfer
5. Aerospace
6. LNG

Objective Two

When you complete this objective you will be able to…

Explain the designs and operating principles of the external gear, internal gear, sliding vane, lobe, and screw type rotary pumps.

Learning Material

1- Rotary Pumps

The name “rotary pump” is given to those pumps, which consist of a closed casing in which gears; lobes, vanes or screws rotate with a minimum of clearance. These rotating parts trap the liquid and push it around the casing from suction to discharge. Unlike in the reciprocating pump, the flow of the liquid through the rotary pump is continuous and the discharge is smooth with minimal pressure fluctuations.

Rotary pumps are used for fuel, lubricating and hydraulic oil systems but also for many other liquids of various viscosities such as liquefied gases like propane, butane, ammonia, and Freon. Rotary pumps are available in many different designs. Five common types are shown in Fig. 9 and a description of these pumps is given below.

The external gear pump (a) has two gears, which rotate in opposite directions inside the casing. Liquid drawn in through the inlet is trapped between the teeth of the gears and the casing wall and is carried to the discharge side. The meshing teeth in the centre prevent liquid from flowing back to the inlet side.

The internal gear pump (b) has an externally-cut gear which meshes with an internally-cut gear on one side and is separated from this gear on the other side by a crescent-shaped partition which prevents liquid from passing back from discharge to suction side. Liquid from the suction fills the spaces between the teeth of both gears when they un-mesh and it is forced out of these spaces into the discharge when the gears mesh again.

The sliding-vane pump (c) has a rotor containing vanes within slots. The rotor is mounted off-centre in the casing with minimal clearance on one side. The vanes are forced out against the casing wall by the centrifugal force. The liquid trapped between the vanes is carried around from suction to the discharge. If pump speed is low the vanes can be forced out of the rotor by the use of springs installed between the blade and the rotor.

The three-lobe pump (d) has two rotors each with three lobes. The rotors are driven by external gears, which synchronize the lobes. The liquid trapped in the pockets formed by the lobes and the casing is carried to the discharge.

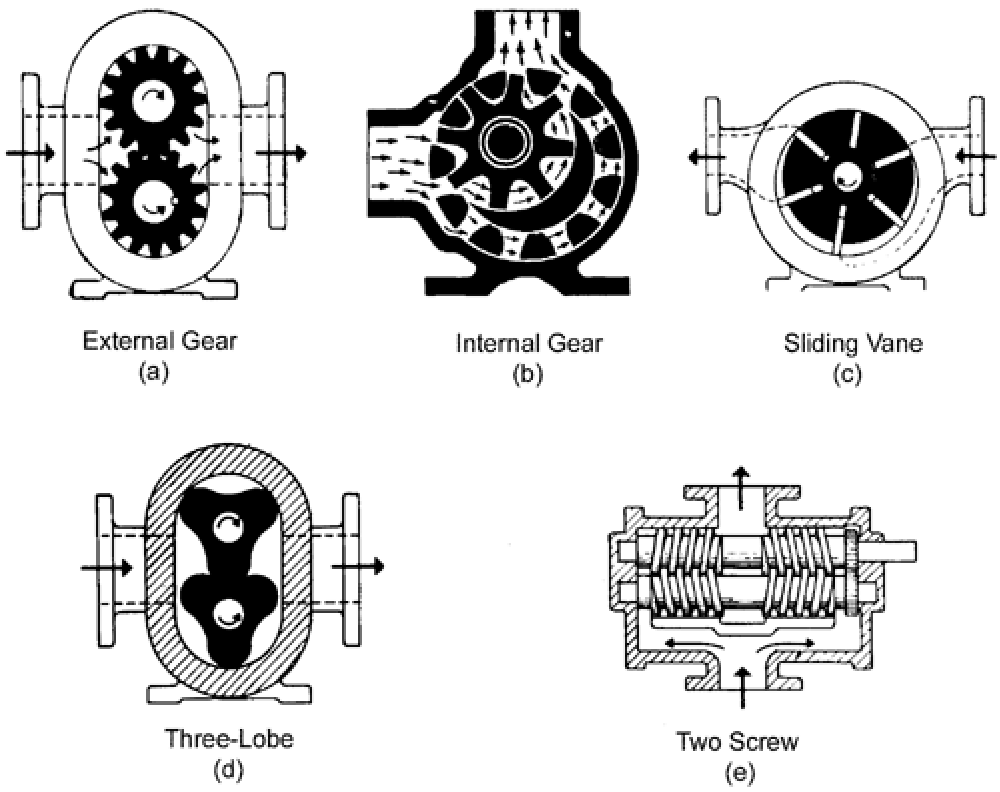


Figure 9 Rotary Pumps

The three-lobe pump (d) has two rotors each with three lobes. The rotors are driven by external gears, which synchronize the lobes. The liquid trapped in the pockets formed by the lobes and the casing is carried to the discharge.

The two-screw pump (e) has two rotors each with two opposing spirals or screw threads. The liquid is carried between the screw threads of the rotors and is forced axially towards the discharge as the screws rotate and mesh.

Rotary pumps are positive displacement pumps and, therefore, should be protected against excessive pressures by a relief valve on the discharge side of the pump.

2- **The gear pump** is a positive displacement pump using two gears rotating in opposite directions. This pump has a drive gear and an idler gear. The gear teeth make a good trap for the liquid, and can be built For very high-pressure service.

3- **The spur gear pump** is a low-speed pump and is suitable for speeds up to 600RPM. For higher speeds, up to 1750 RPM, the herring bone gear pump should be . used.

NOTE Positive displacement pumps must not be run without first making sure that discharge and suction valves are opened first. Operating a 'positive "displacement pump with the discharge closed could create dangerously high pressures in the discharge of the pump. Running with the suction closed could cause pump damage because of a lack of lubrication provided by the pumped fluid.

Objective Three

When you complete this objective you will be able to...

Explain the designs and operating principles of volute and diffuser centrifugal pumps, including impeller designs.

Centrifugal Pumps

The Principle of Operation:

1.1 Centrifugal pumps are classified into two main types: horizontal and vertical according to the shaft centre Line.

1.2 The centrifugal pump may be defined as any pump, which uses centrifugal force to move the liquid, which it pumps. This type of pump. Literally throws the Liquid out Liquid enters the eye of the impeller, at a designed suction pressure the rotation of the impeller then throw the Liquid radically out of the edge of the impeller where it is collected in the case of the pump which is called volute

1.3. The volute is the spiral - shaped casing surrounding the impeller. It collects the Liquid discharged by the pump and convert, velocity energy to pressure energy.

The volute increases in area from its initial point until it en compasses the full 360° around the impeller and then flares out the final discharge opening of the single stage pumps.

1.4 As the liquid produces pressure in the volute case, it also creates an unbalanced load on the shaft bearing (radial thrust), to reduce this, a flow splitter is added, so this called a double volute Casing.

1.5 The diffuser type case is another method. to over come radial thrust .The diffuser. Is a stationary vane ring around the impeller which converts the velocity to pressure all around the impeller without having any high unbalanced redial thrust at anyone point?

The purpose of the diffuser is to collect the high velocity liquid leaving one impeller and direct them into the eye of the next impeller in multi stage pumps.

1.6 Impeller classification according to the mechanical construction:

1. Enclosed impeller, with shrouds or sidewalls enclosing the waterways from both sides.

2. Open impeller, with no shrouds
3. Semi-open or semi-closed impeller has one side shroud
4. Impeller Classification

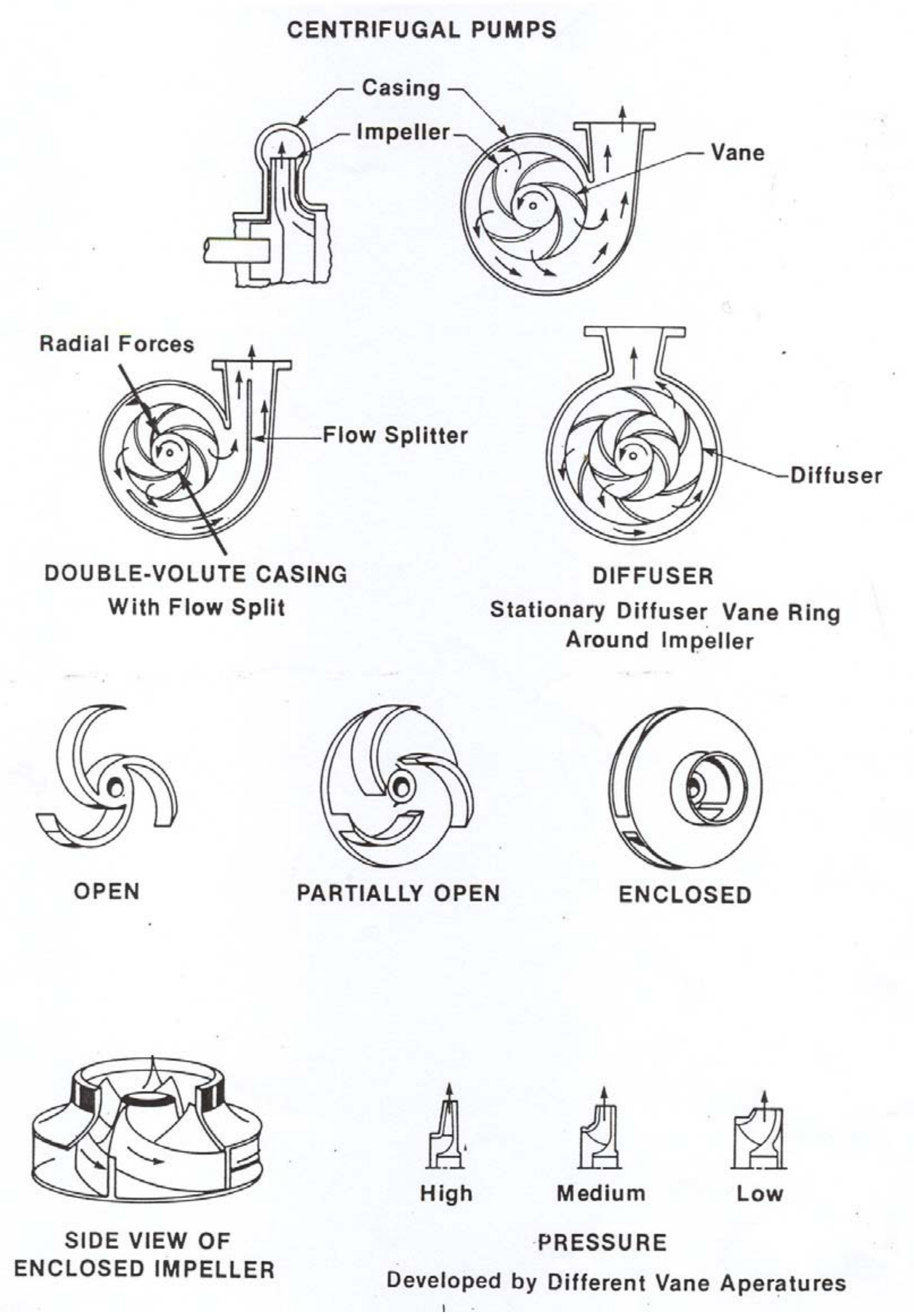


Figure 10 Impellers

A centrifugal pump may be defined as a pump, which uses centrifugal force to impart energy to a fluid by giving the fluid high velocity and then converting this velocity into pressure. Because there are many different designs of centrifugal pumps, they can be divided into a number of types according to specific characteristics. The main division according to the method of imparting energy to the fluid gives us the following types: volute, diffuser, mixed flow, axial flow and regenerative.

They can also be divided according to the:

- Number of stages - single or multi-stage
- Suction inlet - single or double suction
- Position of shaft - horizontal or vertical
- Type of casing - horizontal split or vertical split
- Mounting - in-line or base-mounted

Centrifugal pumps can also be classified according to the application such as: boiler feed pump, general-purpose pump, vacuum pump, circulating pump, etc. Each application requires a different design of pump.

Volute Pump

Basically, the volute centrifugal pump consists of an impeller made up of a number of vanes, which rotates in a volute stationary casing, Fig. 11. The term “volute” refers to the gradually increasing cross-sectional area of the spiral casing.

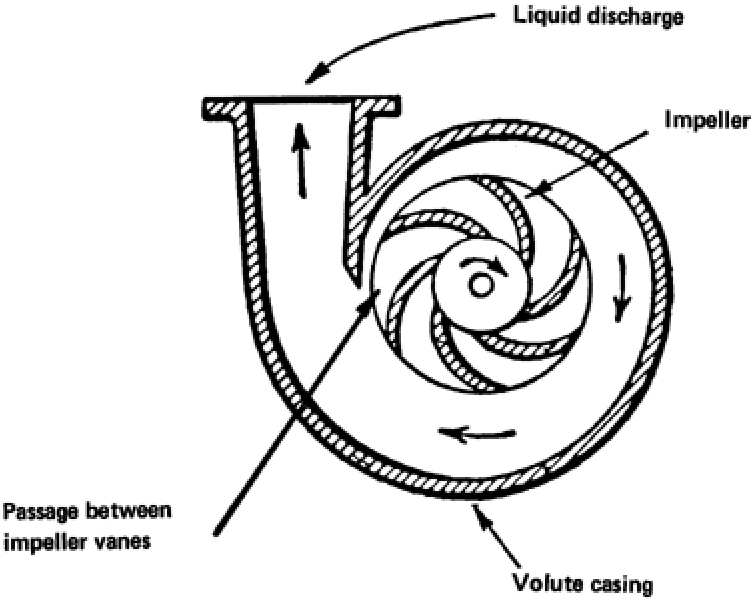


Figure 11
Volute Centrifugal Pump

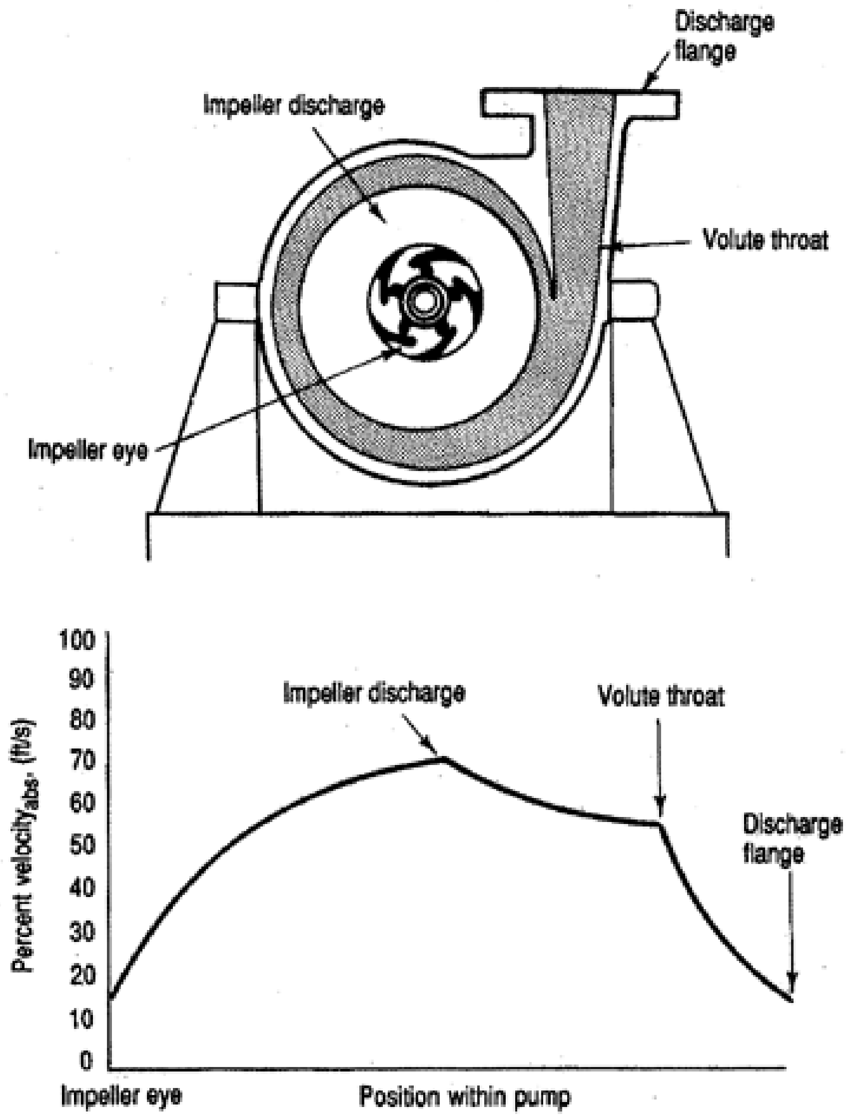


Figure 12
Fluid Velocity Within A Volute Pump

The graph in Fig. 12 illustrates the fluid velocity within a volute pump casing. The liquid being pumped is drawn into the centre or eye of the impeller. It is picked up by the vanes and accelerated to a high velocity. The fluid now contains kinetic energy, and is discharged into the casing by centrifugal force. As the liquid travels through the volute casing to the discharge, its kinetic energy is converted into pressure or potential energy.

Since the liquid between the vanes is forced outward, an area of low pressure is created in the eye of the impeller. More liquid is drawn in through the suction inlet. As a result, the flow of liquid through the pump is constant. A cross-sectional view of the volute pump is shown in Fig. 13.

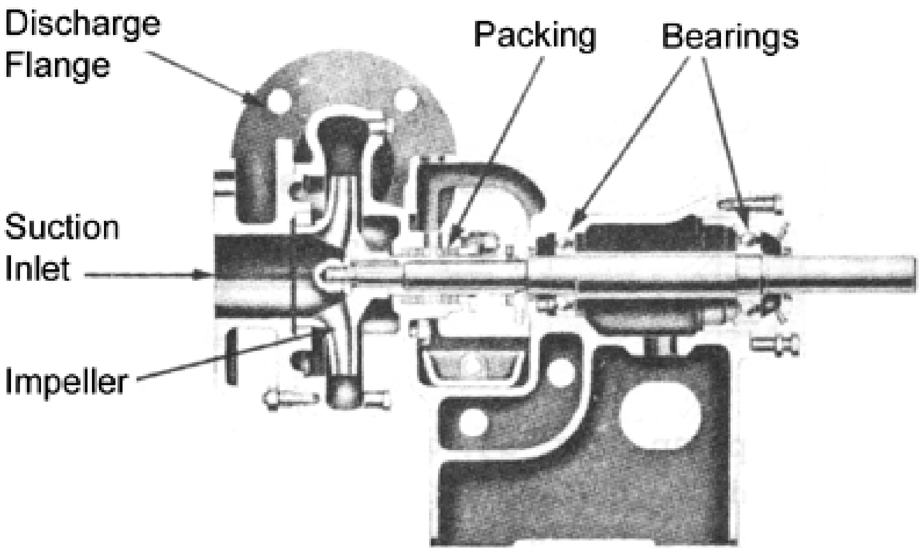


Figure 13
Single-Stage, Single-Inlet Volute Pump

Diffuser Pump

In the diffuser centrifugal pump, the high velocity liquid leaving the impeller passes between vanes in a stationary diffuser ring. These vanes are shaped in such a way that the channels between them gradually increase in area, Fig. 14. As the liquid passes through these channels, its velocity energy is converted into pressure energy. The liquid is then discharged either into a concentric casing (A) or into a volute casing (B) where further velocity-pressure conversion takes place. Since the flow of liquid in volute and diffuser pumps is away from the centre, these pumps are often classified as radial flow centrifugal pumps.

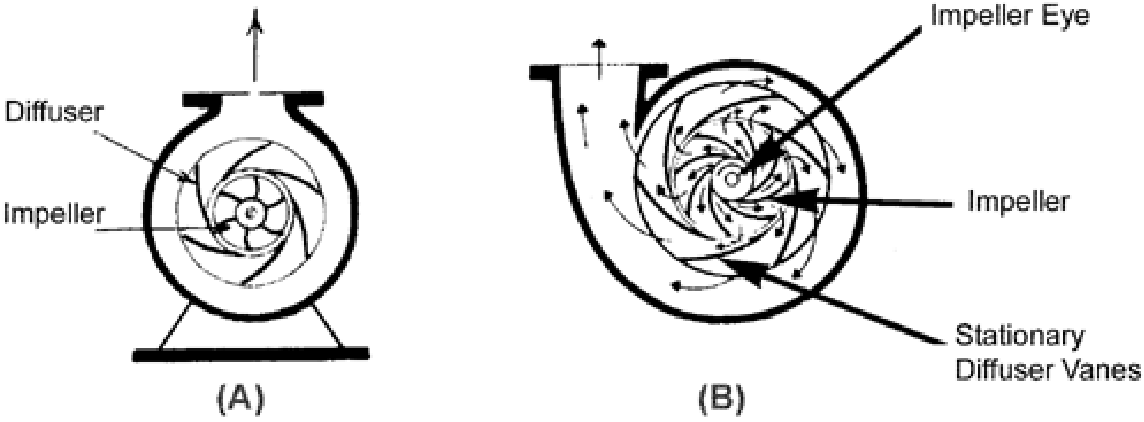


Figure 14
Diffuser Pumps

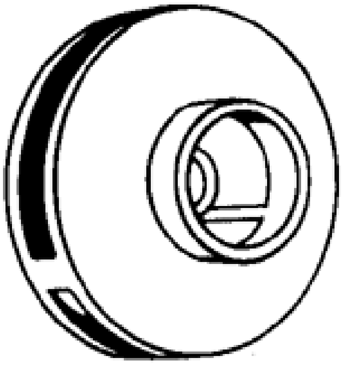
Impeller Designs

Centrifugal pumps are also classified according to their impeller design. There are three basic configurations, which are common. The three basic styles are shown in Fig. 15. They are (a)open, (b) semi-closed and (c) closed.

- Open impeller: From the impeller hub, the vanes extend radially, without shrouds or cover plates.

(a)

(b)



(c)

Figure 15
Impeller Designs

- Semi-open (or Semi-closed) impeller: Vanes extend radially from the hub, but one side has a cover plate.

- Closed impeller: Vanes extend from the hub, and both sides are protected by shrouds or cover plates.

The open and semi-open impellers are normally used on small centrifugal pumps, where the liquid being pumped contains some particles or solids. Such services could be sump pumps, slurry pumps, or even sludge pumps. They allow larger particles to pass through the impeller, which would become lodged in a fully closed design. Due to internal leakage and recirculation, these impeller designs are often less efficient than closed impeller designs. In larger pumps where the area between the vanes is larger, closed impeller designs are used, because they are more efficient. Multistage pumps are typically closed impeller designs.

Objective Four

Characteristics of Centrifugal Pumps)
Centrifugal Pump Performance

Before a centrifugal pump is shipped from the factory where it was made, it is tested. at capacities from zero to maximum for that pump. The results of this test are then plotted to become what is known as the "performance" 'or "characteristic curve" of the pump. An example is shown in Figure 16 A. The characteristic curve shows how the head developed by the pump, and the efficiency of the pump vary with the capacity when the pump is operating at constant speed. Note that horsepower increases as the capacity increases. In other words, if a pump is running at constant speed and the valve in the discharge line is opened wider, the capacity will increase and the horsepower required of the driver will also increase. This is true even though the discharge pressure will usually fall. Centrifugal pumps should be started with the discharge valve almost closed because the horsepower requirements are least at zero capacity. This permits the driver to be brought up to speed before full load is applied. Also, it is likely to be more difficult to make a pump take suction when the discharge valve is open. The head-capacity curve. is independent of the specific gravity of the liquid being pumped. Head measured in feet of liquid will be the same for two identical pumps operating at the same capacity and running at the same speed if one is pumping a light liquid like gasoline and the other is pumping water. Since the two impellers are revolving at the same speed, both liquids leave the impellers with the same velocity. They will be thrown the same distance or height. Therefore, the head in feet will be the same for each.

The pressure developed by a pump and the power required to operate it vary directly with the specific gravity of the liquid being pumped. A change in specific gravity will cause a corresponding change in pressure and horsepower. The heavier the liquid the higher the pressure for the same velocity. In other words, for the example above of gasoline and water developing the same head at the same velocity, the impeller discharging the water would develop a higher discharge pressure than the one handlling gasoline. When more pressure is developed, more work can be done.

Occasionally two or more centrifugal pumps are operated in series or in parallel. The combined head capacity curve for two centrifugal pumps operating in series is shown in Figure 16 D. The total head at any given capacity is that of the individual heads. The combined head capacity curve for two centrifugal pumps operating in parallel is shown in Figure 16 B. Here the total capacity at any given head is the sum of the individual capacities.

Centrifugal pumps in the refinery are driven by induction motors, and steam turbines. Diesel engine driven pumps operate essentially at constant speed. The speed of turbines is variable. The effect of speed on the characteristic head-capacity curve is shown in Figure 16 C. A similar effect may be obtained, within limits, by increasing or decreasing the diameter of the impeller. As could be expected, increases in operating speed of centrifugal pumps causes increases in the horsepower required from the drivers.

The efficiency of a centrifugal pump decreases rapidly with large increases in viscosity of the fluid being pumped. The horsepower required increases rapidly. Viscous materials are handled with reciprocating or rotary pumps.

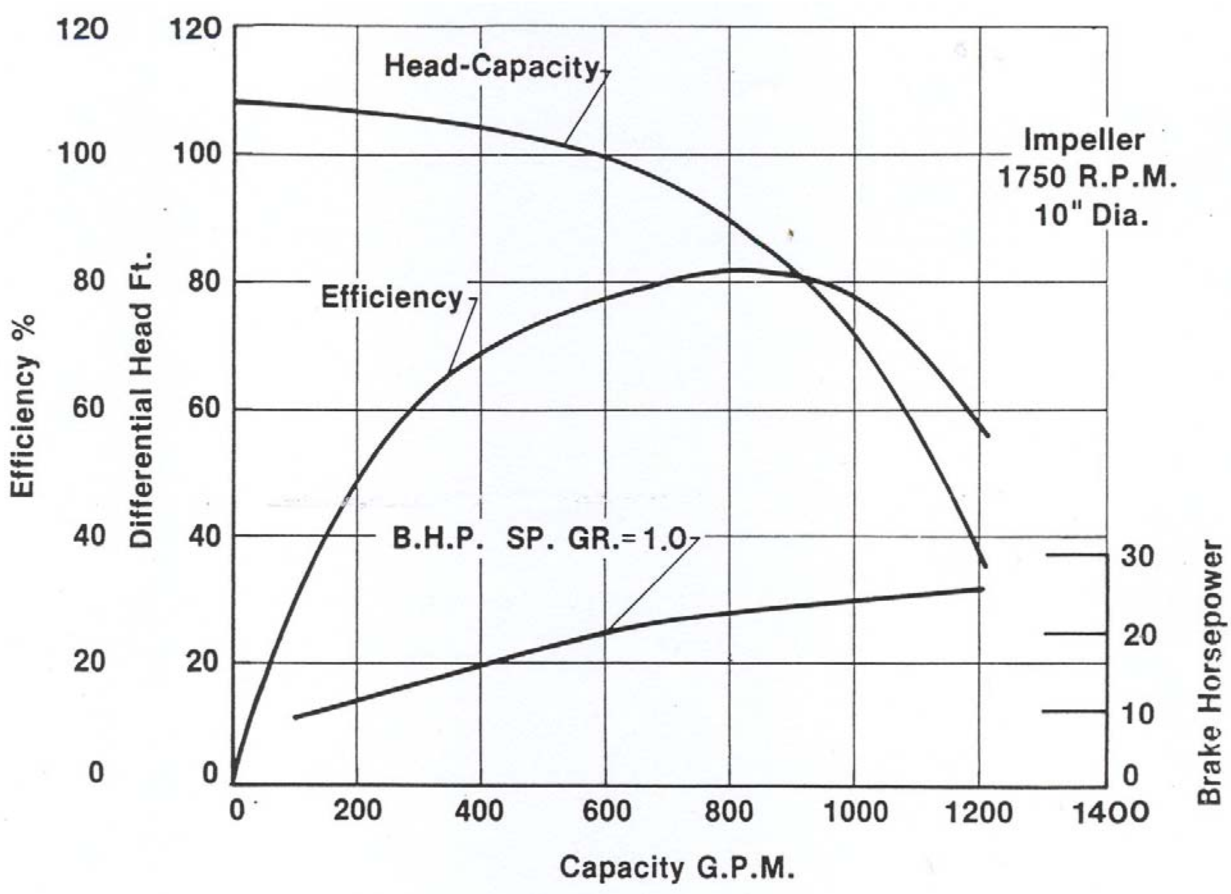


Figure 16 A
Characteristics Curves At Constant Speed

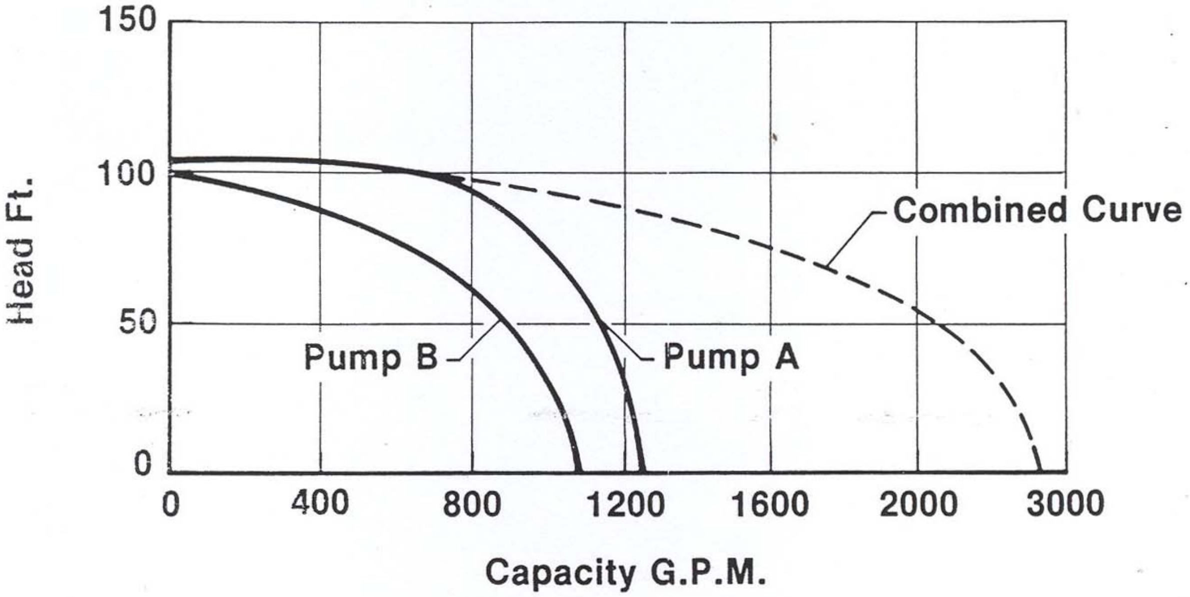


Figure 16 B
Two Centrifugal Pumps in Parallel

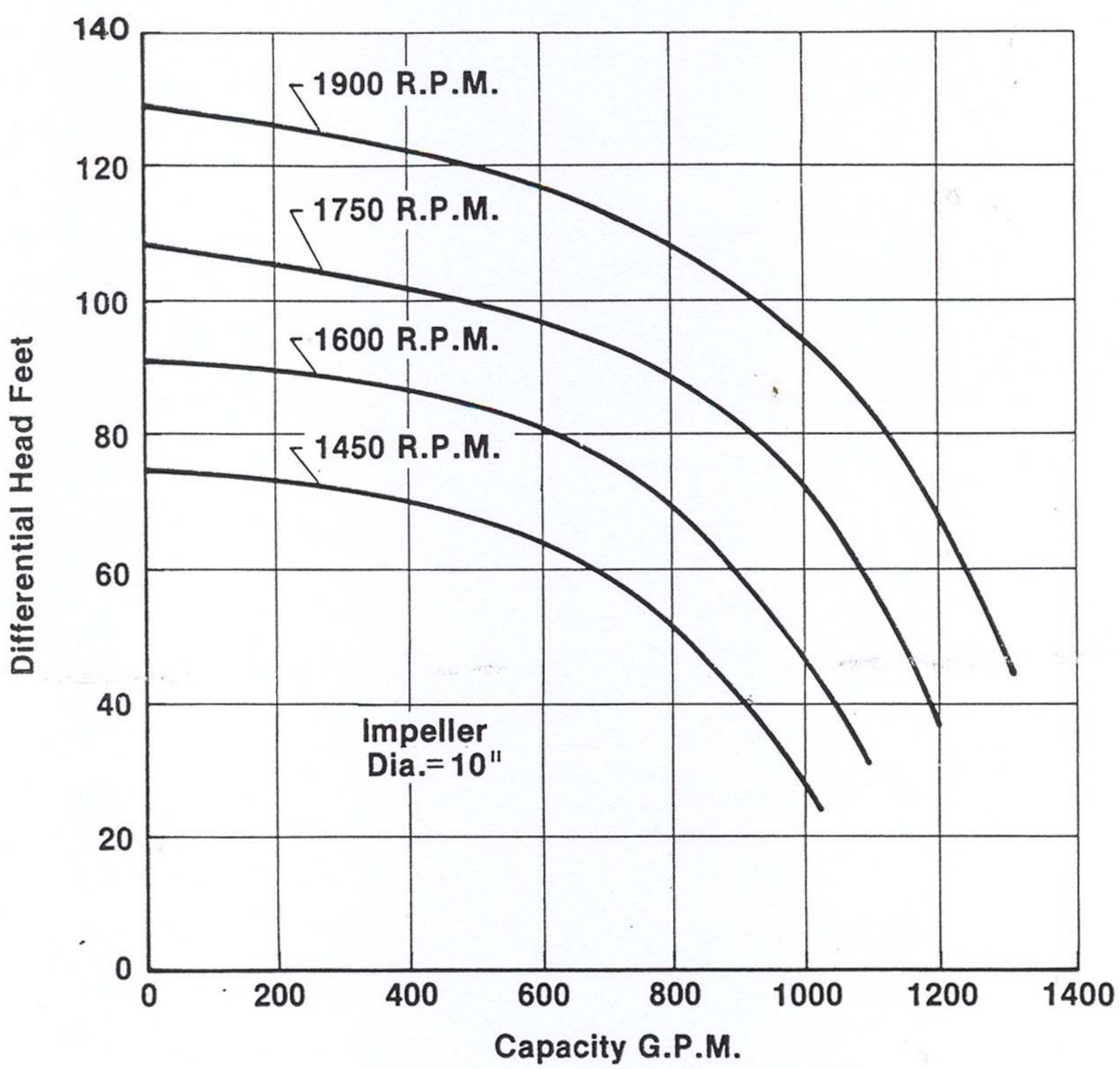


Figure 16 C

Speed Variation

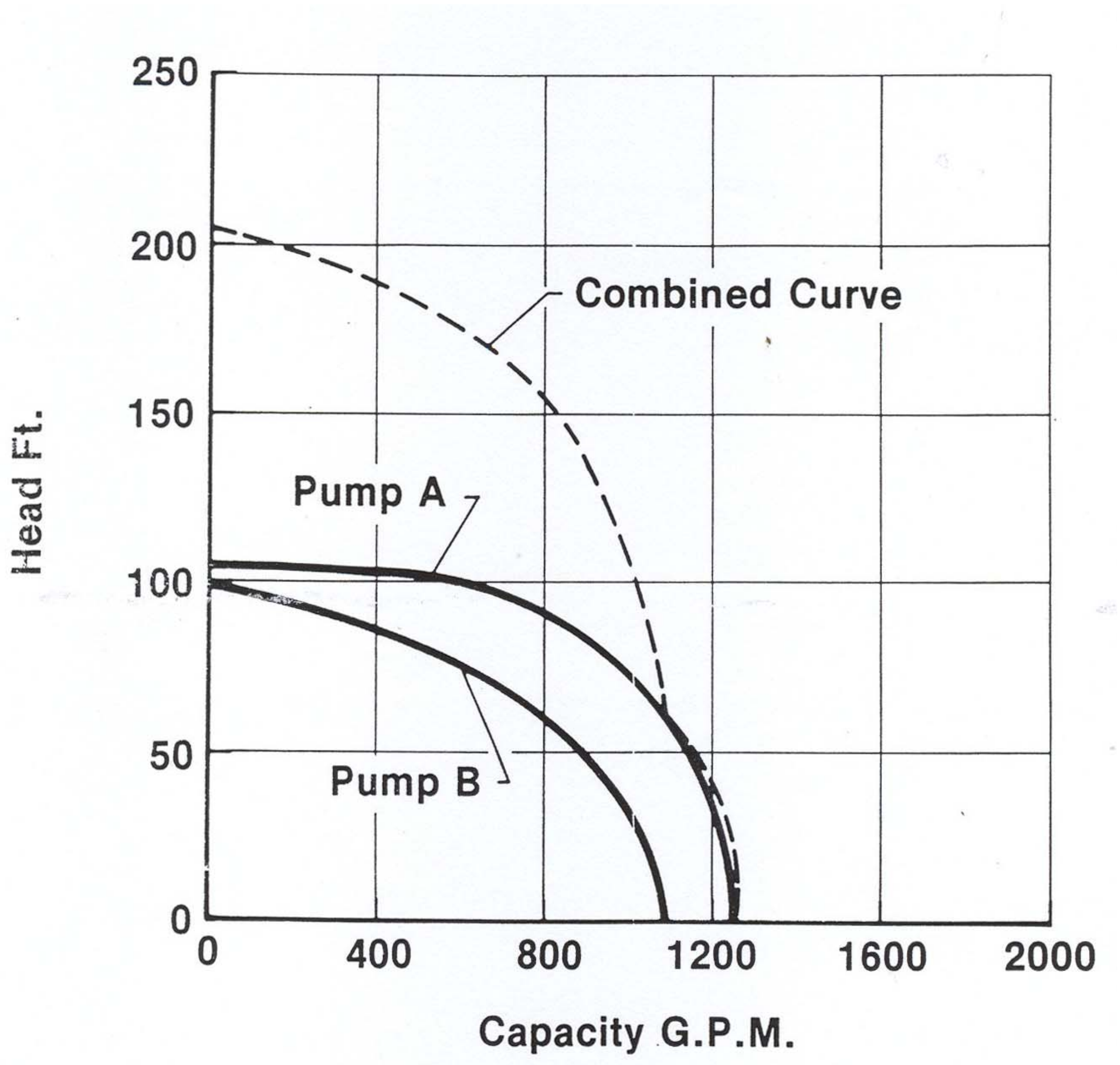


Figure 16 D

Two Centrifugal Pumps in Series

Objective Five

When you complete this objective you will be able to...

Describe centrifugal pump arrangements, including vertical, horizontal, single and double suction, opposed impellers, multi-staging, split and barrel casings.

1. Submersible Centrifugal Pumps

1.1 General Information

The length of satisfactory service obtained from the equipment will, in part, depend on proper installation and maintenance. This instruction manual is provided to present the basic information for operating and management personnel. Due to the many variations and custom designed units It is impossible to cover every design variation *or* contingency which may arise, however, the basic information - contained herein will cover' most questions.

1.2 Identification

Should questions arise concerning the pump; the factory will require the complete serial number to be of assistance. The serial number is stamped on a metal nameplate affixed to the discharge head assembly. The driver will have a separate nameplate attached to it when requesting information on the driver both the driver serial number and pump serial number will be required.

1.3 General Description

The basic components of Booster and Process Pumps are the driver, discharge head assembly, column assembly (when used), bowl assembly, and suction vessel. The pumps are normally shipped assembled and ready, for installation. The drivers, couplings and strainers (when used) are shipped un-assembled to prevent damage.

1.4 Drivers

A variety of drivers may be used, however, electric motors are most common. For the purposes of this manual all types of drivers can be grouped into two categories:

1. Hollow shaft drivers where the pump shaft extends thru a tube in the center *of* the rotor and is connected to the driver by a clutch assembly at the top of

the driver.

2. Solid shaft drivers where the rotor shaft is solid and projects below the driver mounting base. This type driver requires an adjustable coupling for connecting to the pump.

2. Vertical Pumps

Centrifugal pumps are also classed according to the arrangement of their mechanical parts. The most common arrangements are vertical and horizontal pumps. Vertical pumps have a vertical shaft. The motor or turbine is usually on the top, with the pump being below. The pump assembly often is immersed in the fluid being pumped. They can have one or multiple stages. The vertical pump in Fig. 17 is a large mixed flow type.

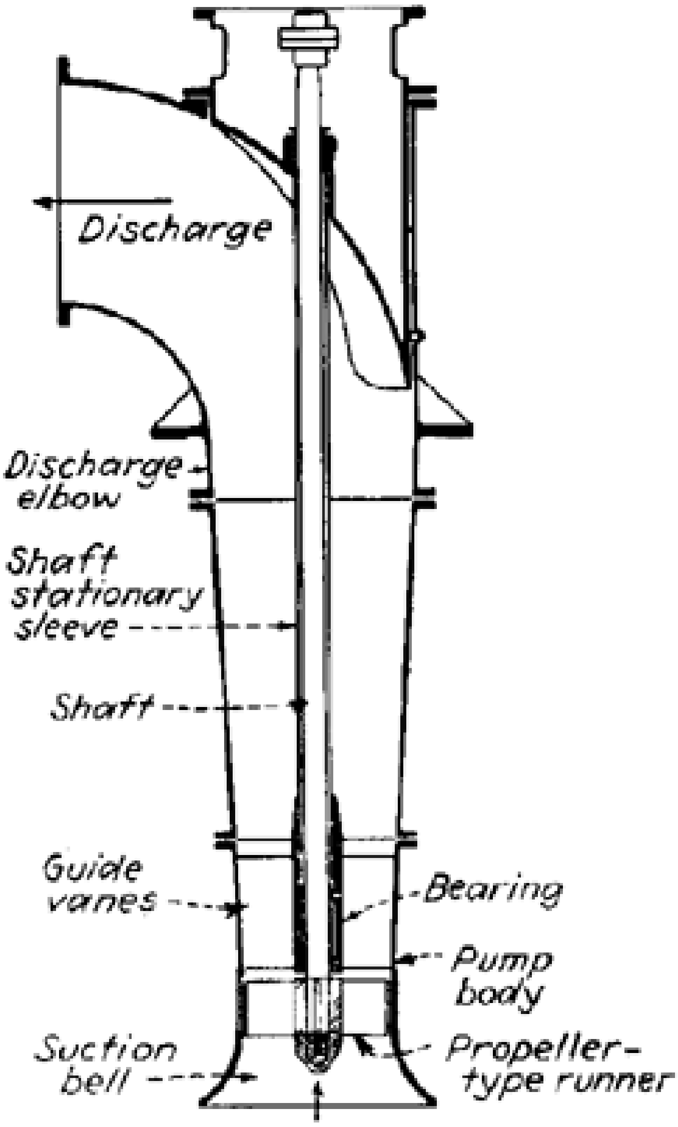


Figure 17 Vertical Pumps

3. Horizontal Pumps

Horizontal pumps are the most common, with the motor, shaft and pump running horizontal to the mounting surface. Fig. 18 shows a typical horizontal pump arrangement.

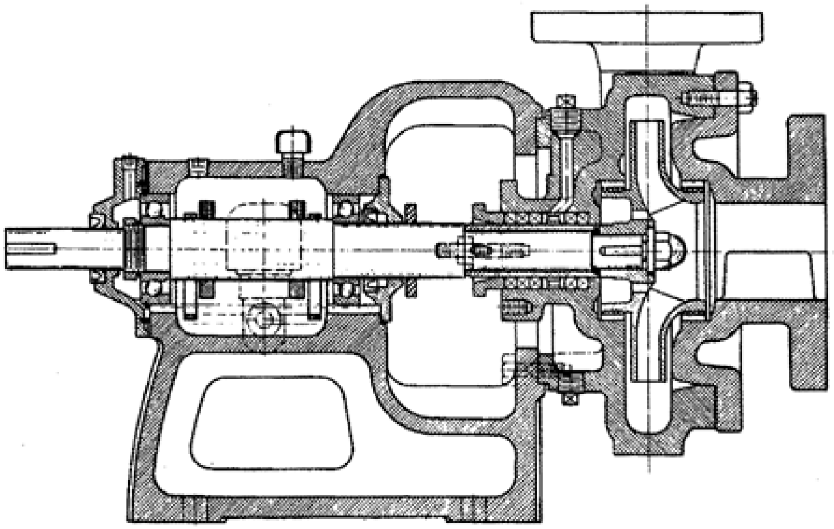


Figure 18 Horizontal Single Stage Centrifugal with Ball Bearings

Single and Double Suction

The single suction pump as the name implies has a single suction as in the pump above in Fig. 18. Any imbalance of thrust is taken care of by mechanical means, such as a thrust bearing. A double suction pump has two suctions, as in Fig.19. The symmetry of suction and discharge passages balances out most axial forces. A small thrust bearing absorbs the remaining imbalance in thrust.

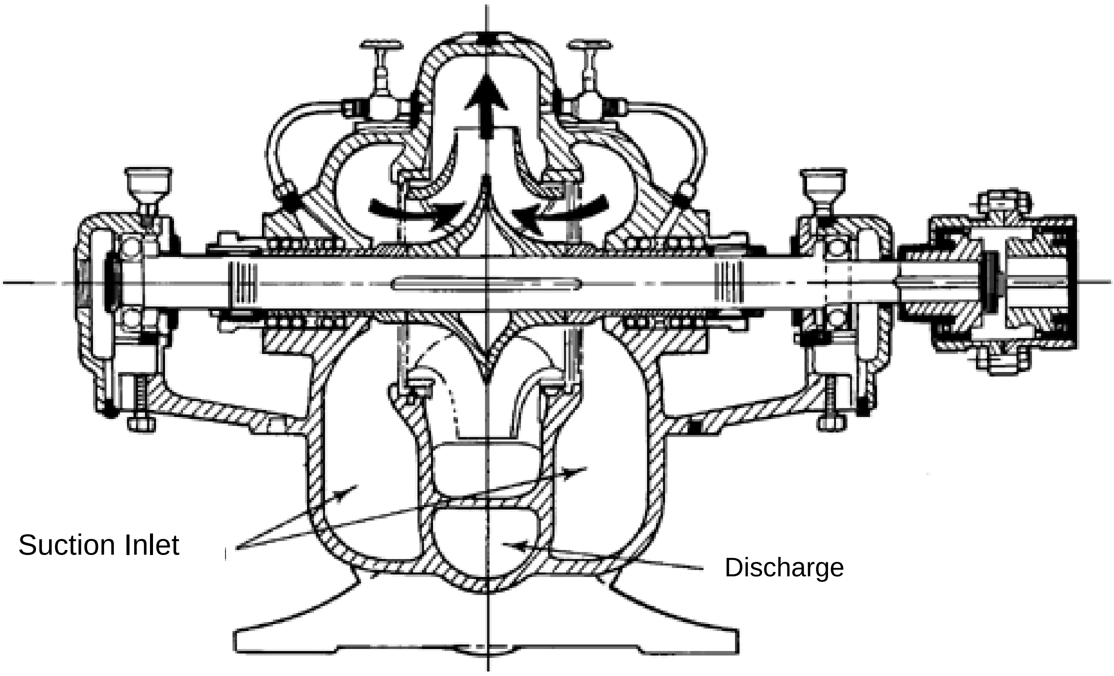


Figure 19 Double Suction Pump

Opposed Impellers

Pumps such as the two stage volute pump in Fig. 20 can have impellers mounted opposed to each other to balance out thrust forces.

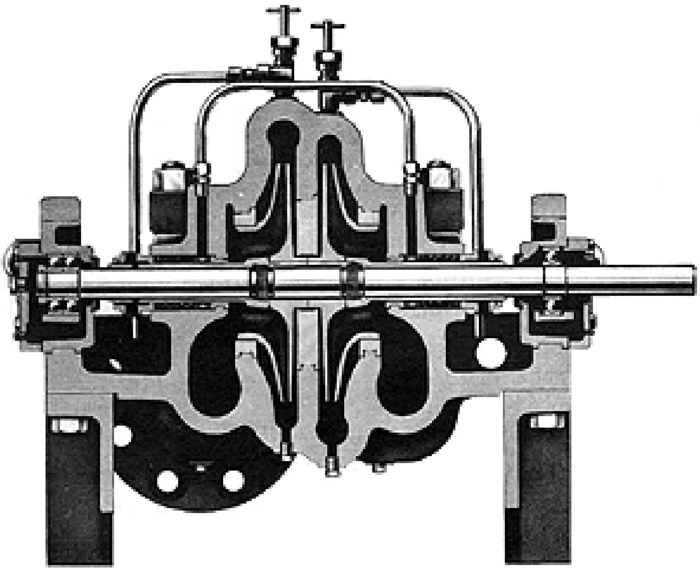


Figure 20 Two-Stage Volute Pump with Opposed Impellers

Multi-Stage Pumps

The pressure developed by a centrifugal pump with a single impeller is limited to about 1000 kPa. Many pumps, such as boiler feedwater pumps, are required to deliver much higher discharge pressures. To obtain the higher pressures, centrifugal pumps are equipped with two or more impellers operating in series. The discharge of one impeller is connected to the suction of the next impeller. These pumps are classed as multi-stage pumps. The unit in Fig. 21 is a five-stage volute pump.

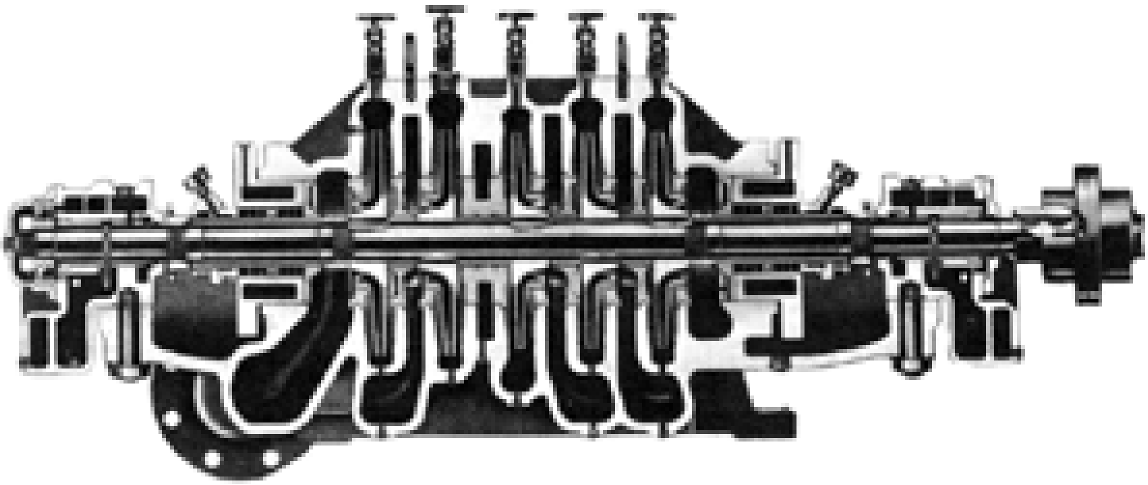


Figure 21
Five-Stage Volute Pump

Split Casings

Centrifugal pump casings may be split horizontally, vertically or diagonally. A horizontally split casing, also called an axially split casing, is shown in Fig. 22. The suction and discharge nozzles are usually in the lower half of the casing. The upper half can be easily lifted for inspection.

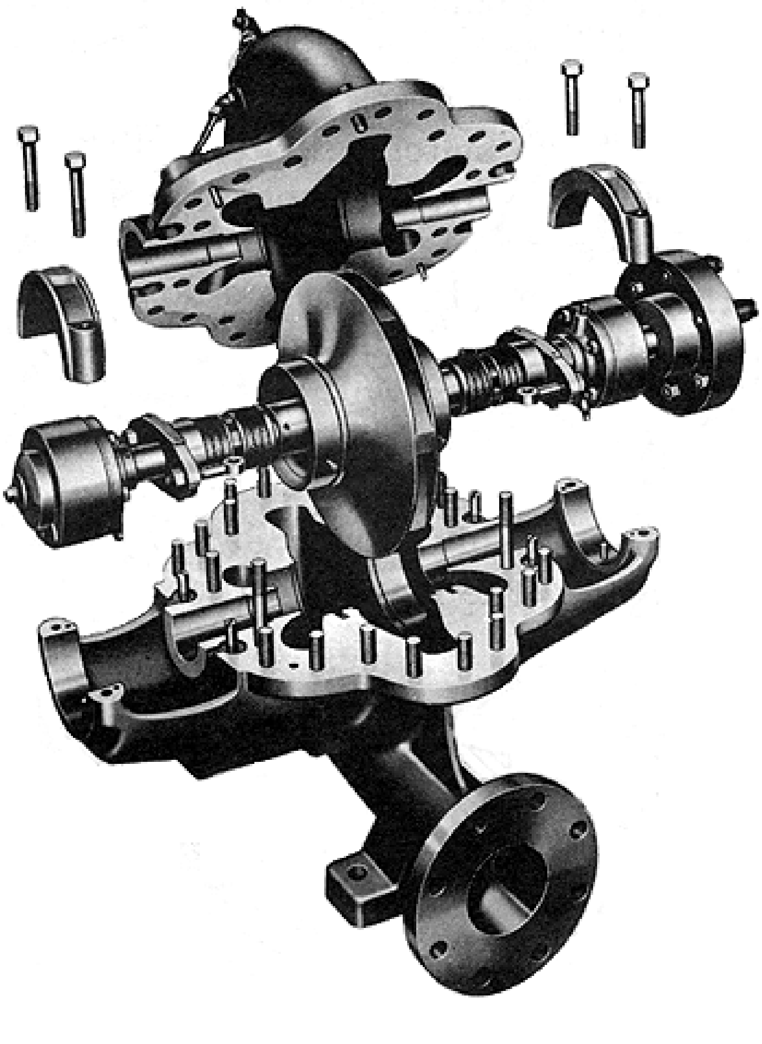


Figure 22 Volute Pump With Horizontally Split Casing

Barrel Casings

For multi-stage pumps of the volute or diffuser type with discharge pressures above 10 000 kPa barrel casings are used to avoid the difficulty in maintaining a tight joint between the halves of a horizontally split casing and the sections of radially split casings.

The barrel casing consists of an inner casing fitted in an outer casing. The space between the two casings is subjected to discharge pressure, which tends to hold the sections of the inner casing together. The inner casing may be made up of two halves, horizontally joined, or a number of sections with circumferential joints. The outer casing, the barrel, has no horizontal joints. It has removable heads at each end.

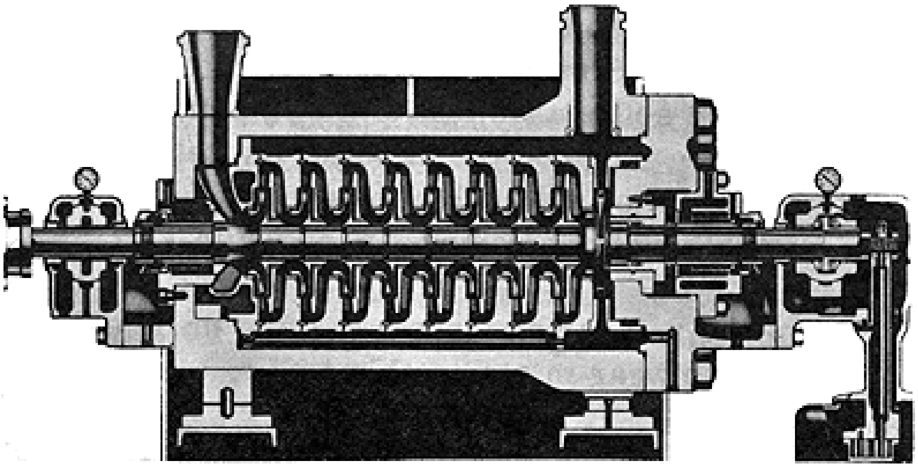


Figure 23
High Pressure Nine-Stage Barrel-Type Feed Pump

A cross-sectional view of a barrel type boiler feed pump is shown in Fig. 23. The inner casing is made up of circumferentially joined ring sections. It is fitted into the outer casing through one end, which is then closed by a head.

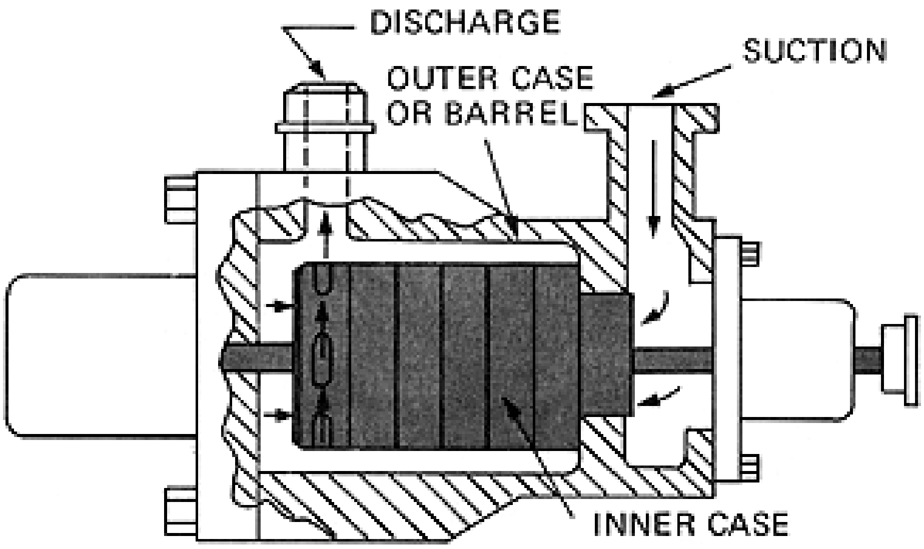


Figure 24 Barrel Pump Casing Arrangement

The general arrangement of the two casings is shown in Fig. 24. It has an inner casing contained within an outer barrel casing, which has suction and discharge nozzles welded into place. The inner casing, which contains the rotor and diffusers, is made up of casing rings. The rings are held together by means of tie rods and can be removed from the outer barrel as a unit without disturbing the driver or the piping. Barrel pumps are well suited to multistage, high- pressure applications.

Objective Six

When you complete this objective you will be able to…

Describe the design and applications of axial and mixed flow pumps.

Learning Material

Axial Flow Pump

The axial flow pump, Fig. 25, often called a propeller pump, uses an impeller with vanes similar to a ship's propeller. The pump develops its head by the propelling or lifting action of the vanes on the liquid and the flow of the liquid is through the casing, parallel to the shaft. It is usually of vertical design but horizontal units are available. As compared to a radial flow pump, the axial pump has a low suction lift and develops a relatively low discharge head, but it has a large flow capacity.

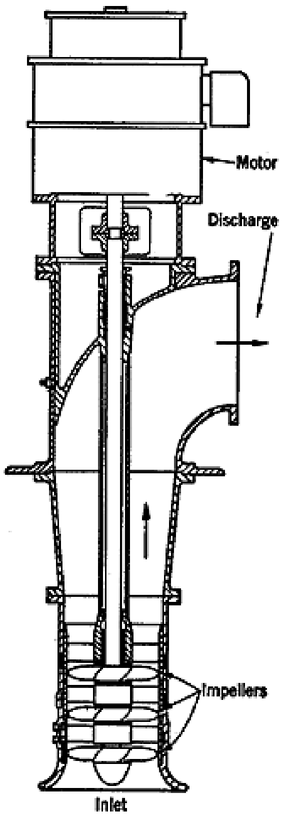


Figure 25

Vertical Axial Flow Pump

Axial flow pumps have the advantages of compact size and the ability to operate at high speeds. Their disadvantages include low suction lift capacity and a relatively low discharge head capability. They are used mainly for low head, high capacity applications and are available in the single stage design or the multistage design as shown in Fig. 25.

Mixed Flow Pump

The mixed flow pump, Fig. 26 combines some of the characteristics of the radial flow and axial flow pumps. It develops its discharge head by using both centrifugal force and lift of the vanes on the liquid.

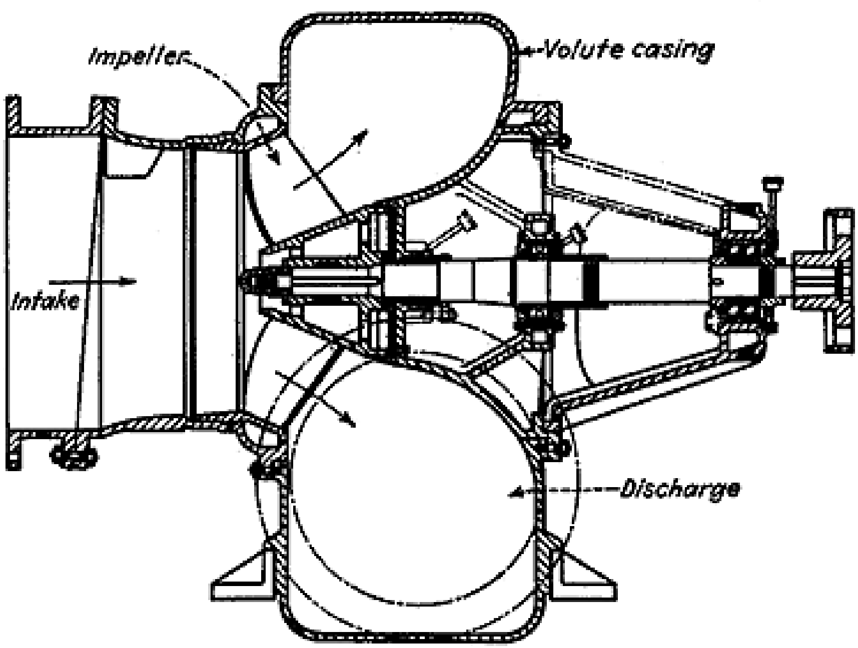


Figure 26 Mixed Flow Pump

The mixed flow pump shown in Fig. 26 has a single inlet impeller with the flow entering the pump in an axial direction and leaving the pump in a direction somewhere between axial and radial. Although the mixed flow pump in Fig. 26 is arranged horizontally, this type of pump, like the axial flow type, is frequently arranged for vertical operation. With the vertical arrangement the pump can be placed directly in the suction well and thus be primed at all times.

Like the axial flow pump, the mixed flow type is used mainly on low head, high capacity service and like the axial flow pump it may be fitted with variable pitch impeller vanes.

Objective Seven

When you complete this objective you will be able to…

Describe the design and components of a multistage centrifugal pump, clearly stating the purpose and general design of: wearing rings, shaft sleeves, seals, bearings and lubrication components, vents and drains.

Learning Material

Wearing Rings

The rotating impeller of the centrifugal pump must be sealed in the stationary casing with a minimum of clearance in order to keep leakage from discharge to suction as small as possible. This seal is provided by the flat joint formed by the rim around the impeller eye and a matching flat circular surface in the casing, as shown in Fig. 27 (a). However, during operation, the continuous leakage of the liquid through the joint will slowly wear away the surfaces of this joint and pump efficiency will drop off. When the clearance becomes too large, restoration of the original clearance will be necessary, either by building up the worn surfaces or by replacing the casing and impeller. This, however, will be quite costly for all but the smallest of pumps.

The cost of restoration can be considerably reduced by installation of wearing rings on the impeller, the casing, or both. The wearing rings are renewable and can be replaced at a relatively low cost.

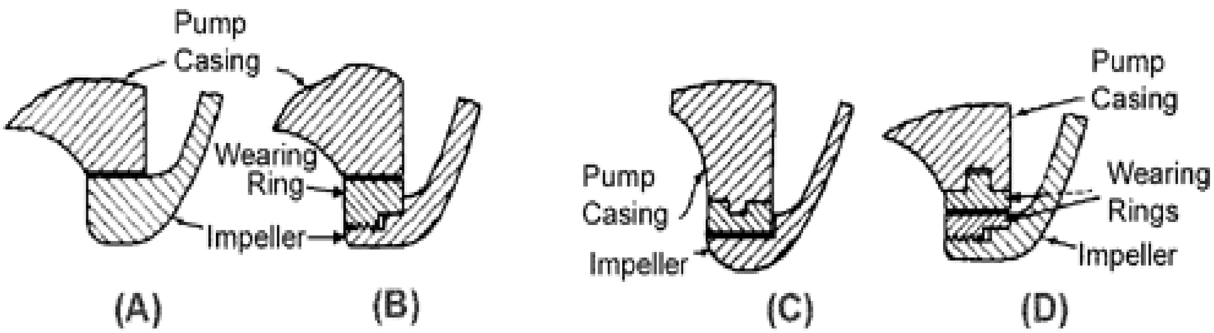


Figure 27 Wearing Rings

When a wearing ring is mounted only on the impeller, Fig. 27 (b), it is made of a softer material than that of the casing so that practically all wear is on the ring. Similarly, when a ring is mounted in the casing only (c), its material will be softer than that of the impeller. Most large pumps are equipped with wearing rings on the impeller as well as in the casing (d). A cross-sectional view of a centrifugal pump with a double inlet impeller having wearing rings on the impeller and in the casing is shown in Fig. 28

Wearing rings are often made of bronze or cast iron since these materials tend to wear in a smooth manner. They are installed on the rim of the impeller by either threading or shrinking. Setscrews are used to prevent them from working loose. Casing wearing rings consist of either a continuous ring used in vertically split casings, or of two half rings for horizontally split casings. Split rings can be fitted onto a ridge or into a groove of the casing which will prevent any axial movement should the ring work loose.

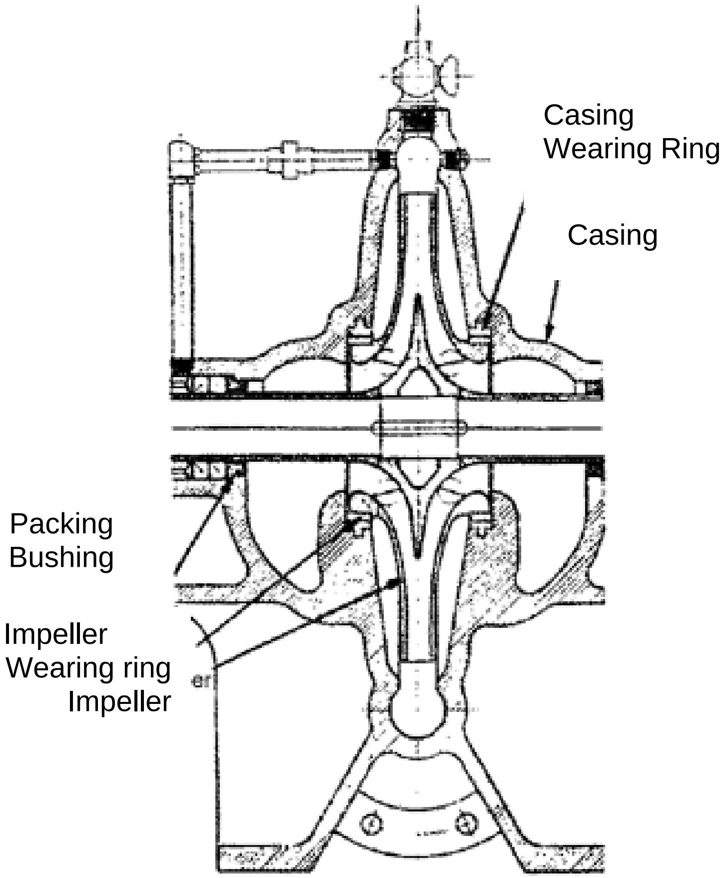


Figure 28
Impeller and Casing Wearing Rings (Allis-Chalmers Mfg. Co.)

Pump Shaft Sealing

In order to minimize leakage around the pump shaft where it passes through the casing, the following means are used:

1. Stuffing boxes
2. Mechanical seals

Stuffing Boxes

A stuffing box consists of a cylindrical recess around the shaft that holds a number of rings of packing, which provide a seal between the casing and shaft. The packing is held in place by a gland that can be adjusted to compress the rings. The desired fit is obtained by tightening the adjusting nuts. The bottom or inside end of the stuffing box may be formed by either the pump casing itself or by a bottom bushing. The basic construction of a stuffing box with five rings of packing is shown in Fig. 29.

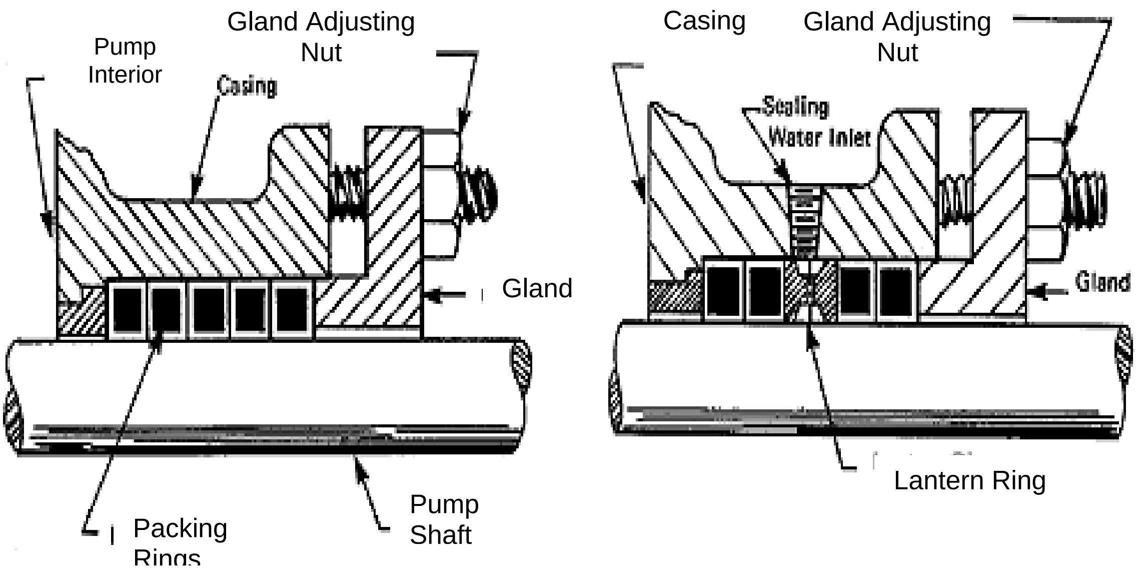


Figure 29

(a) Stuffing Box with Packing (b) Stuffing Box with Lantern Ring

Packing rings are made of pliable, yet durable materials such as nylon, flax or Teflon. Metals such as lead, copper or aluminum are also used. They are wound as a foil around an asbestos or plastic core. The packing is usually impregnated with a lubricant, which makes the packing self-lubricating during the startup period. Packing is supplied in continuous coils of square cross-section, or in preformed die-moulded rings.

Packing in stuffing boxes should never be compressed so tightly that leakage is completely stopped. The resulting friction would cause excessive heat build-up, the packing would burn up and the shaft could be severely damaged by scoring. Instead, a slight leakage of liquid should be allowed to provide lubrication between packing and shaft.

When a pump operates with a negative suction pressure, a fully-packed stuffing box as in Fig. 29 (a) will not provide proper sealing since air would be drawn into the casing along the shaft stopping the required leakage of liquid. To provide proper sealing, the stuffing box is then fitted with a lantern ring and a sealing water connection as shown in Fig. 29 (b).

The lantern ring (also called seal cage) is a metal ring with channels machined in its inside and outside perimeter, connected by radially drilled holes. It serves to distribute sealing liquid under pressure to the packing thus preventing air infiltration and providing lubrication. This sealing liquid is usually provided by the high-pressure section of the pump casing, either through an external connection, Fig. 30, or through an internally drilled passage in the casing.

Lantern rings are also used on pumps handling liquids containing sand, grit or other abrasive particles, which could damage the shaft and shorten the life of the packing when allowed to enter the stuffing box. Clean sealing liquid provided by a separate source or by the discharge side of the pump via a filter or separator will then keep the gritty substances out of the stuffing box.

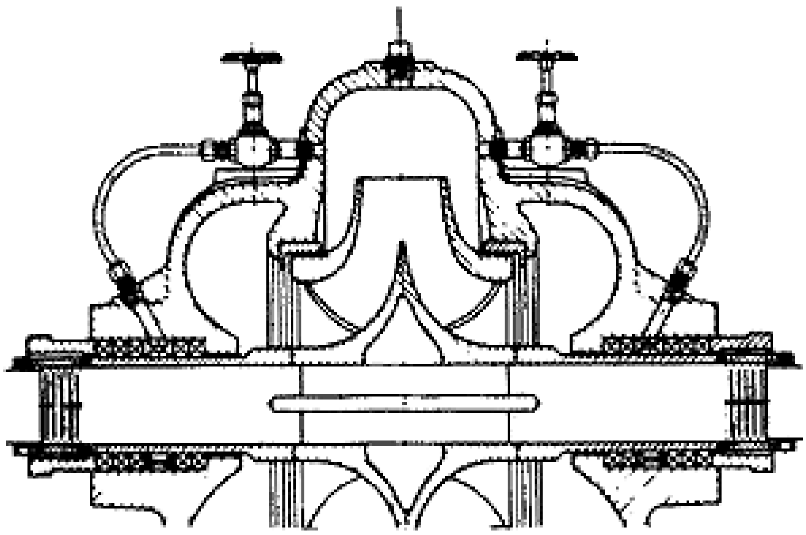


Figure 30
Pumped Liquid to Seals

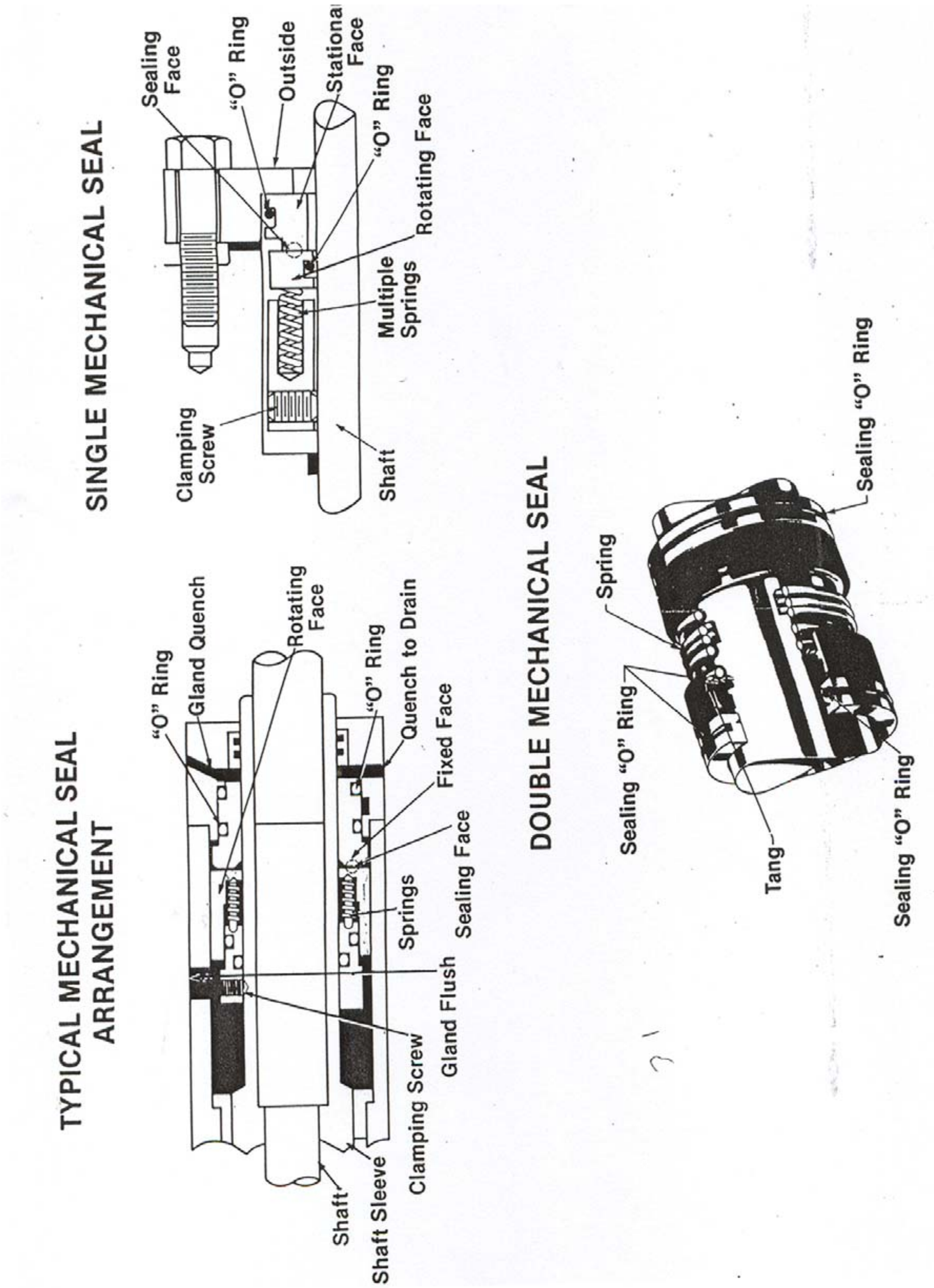

SINGLE MECHANICAL SEAL
Sealing Face
"O" Ring
Outside
Stationa Face
"O" Ring
Rotating Face
Multiple Springs
Clamping Screw
Shaft
TYPICAL MECHANICAL SEAL ARRANGEMENT
"O" Ring
Gland Quench
Rotating Face
"O" Ring
Quench to Drain
Fixed Face
Sealing Face
Springs
Gland Flush
Clamping Screw
Shaft Sleeve
Shaft
DOUBLE MECHANICAL SEAL
Spring
Sealing "O" Ring
Sealing "O" Ring
Tang
Sealing "O" Ring

Figure 31

PACKED GLANDS

1. To stop leakage of pumped liquid along the shaft, a gland or seal is fitted. This might be a packed gland or a mechanical seal.

2. To save wear on the pump shaft, a shaft sleeve is fitted. With both mechanical seals and packed glands, wear takes place along the sealing area. This is called fretting. Some form of seal is fitted between the shaft sleeve and shaft and is normally an "0" ring type seal.

3. With a packed gland, rings of packing material are fitted in a packing box. The rings of packing fit around the shaft sleeve. They may be preformed rings, or sections of a length of packing. The packing may consist of graphited asbestos, or soft metal-like ribbons of babbitt. Metal twisted and folded around an asbestos core.

4. In the center of the packing box, there is normally a lantern ring. The purpose of the lantern ring is to make a place to provide coolant and flushing to the packing box. This liquid has to be compatible with the pumped fluid. For example: on the reduced. crude pump for the bottoms of a crude tower, if the stuffing box flush is a material with a low flash point and there is excessive leakage into the pump along the shaft, it can affect the quality of the final product.

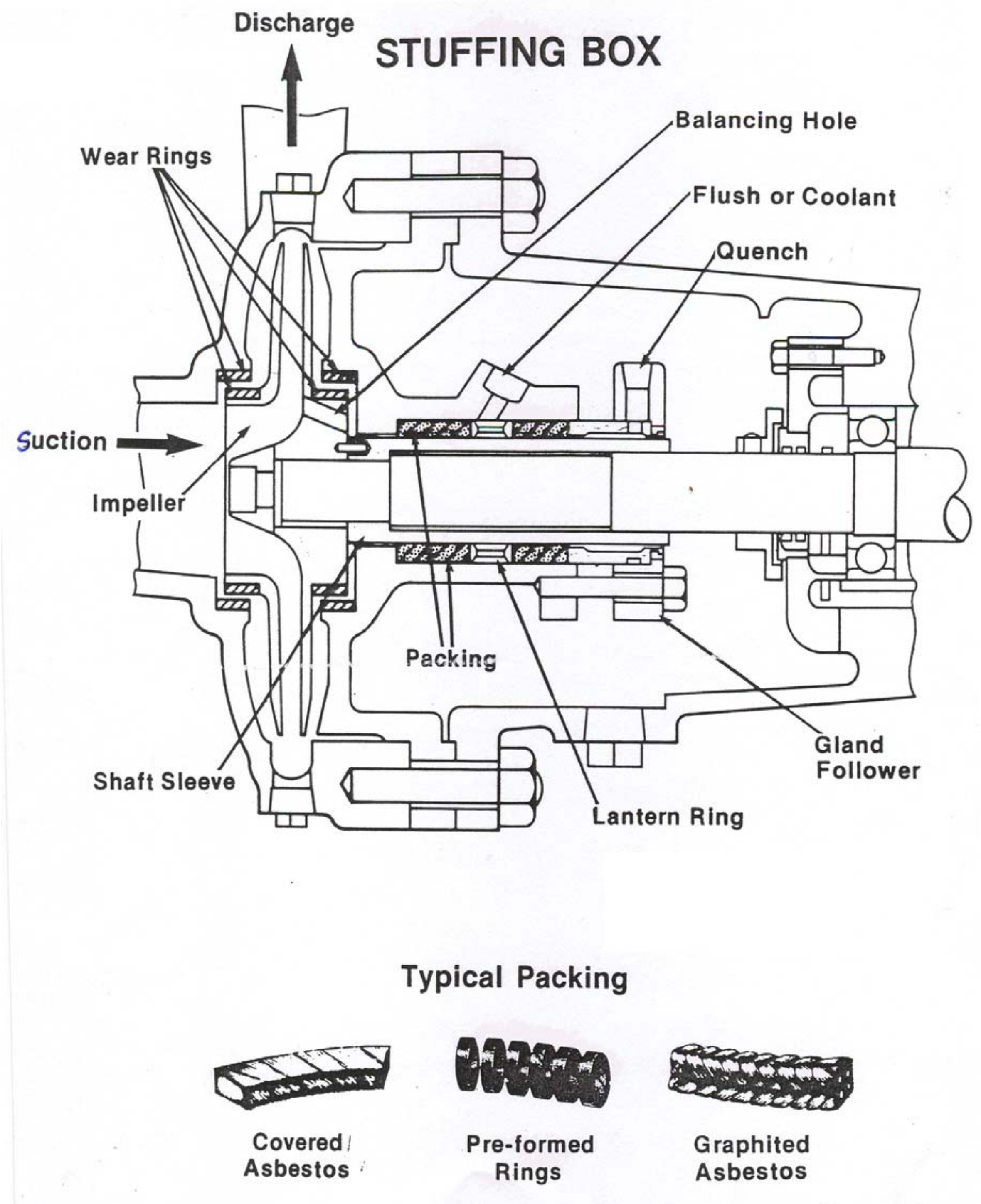

Discharge
STUFFING BOX
Balancing Hole
Wear Rings
Flush or Coolant
Quench
Suction
Impeller
Packing
Gland
Follower
Shaft Sleeve
Lantern Ring
Typical Packing
Covered Asbestos
Pre-formed Rings
Graphited Asbestos

Figure 32

MECHANICAL SEALS

In general, centrifugal pumps have mechanical seals. This is to prevent the escape of process liquid along the shaft. A mechanical seal operates by having one sealing face rotating with the shaft held by spring pressure against another sealing face which is stationary. The rotating face and the stationary faces are mechanically smooth and matched so they fit together perfectly. The stationary face is normally carbon and is fixed in the seal flange with an “O “ ring to seal it.

The rotating face is normally made out of stainless steel. It is not joined to the shaft or sleeve, but is driven by a spring or spring loaded pins. It rotates with the shaft and is held against the stationary face by spring pressure. Again, the rotating face assembly is sealed against the shaft with an “O “ ring or u-cup. This is what is known as a single seal. There are double seals for special jobs.

Improvements in mechanical seals have' occurred through developments in the face materials, these being satellite, tungsten and ceramics, together with an improved carbon wearing face. Therefore, the life of mechanical seals has been improved.

The “O “ rings have become more. durable by being made from material Such as Viton, Teflon and Silicone, depending upon the service. Still, the squareness of the faces is the most important' feature for a good seal. To give a constant pressure all over the seal face, multiple springs are being used instead of single springs.

Seal Flush is taken from the discharge of the pump via a cyclone to ensure the flush liquid enters the seal clean. The seal flush also keeps the seal spring!:. clean so they can operate efficiently.

Seal Quench is usually water which circulates around the seal flange to condense any vapor that might have' leaked past the seal and become a hazard. The condensate is piped away. to a safe location.

Shaft Sleeves

Shafts are subjected to corrosion, erosion and wear at the stuffing boxes, which will affect their strength and make effective sealing with packing rings difficult. Shafts of smaller pumps are usually made of corrosion and wear-resistant materials for longer life. Renewable sleeves usually protect large pump shafts, as shown in Fig. 33. The sleeve is secured on the shaft by the shaft nut. Rotation of the sleeve is prevented by a key, which joins the shaft and the sleeve.

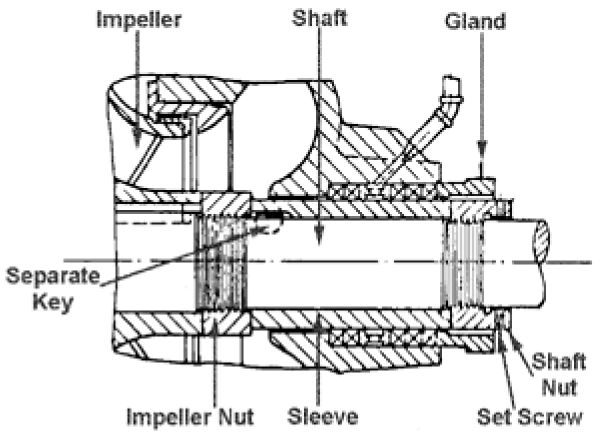


Figure 33
Shaft Sleeve

Mechanical Seals

Leakage from stuffing boxes is objectionable on pumps handling liquids such as gasoline, acids, and ammonia. Instead, these pumps are equipped with mechanical seals, which reduce leakage to a minute amount. They are also used on pumps where stuffing boxes cannot offer adequate leak protection such as high-pressure pumps.

Basically a mechanical seal consists of two flat rings each with a polished flat sealing surface. The rings are perpendicular to the pump shaft and the sealing faces rotate on each other. One of the rings is called the sealing ring. It is made of a carbonaceous (carbon based) material such as graphite, and it is held in position by a spring. The other ring, its face in contact with that of the sealing ring, is called the mating ring. It is made of hard material such as stellite, ceramic, etc. Mechanical seals may be divided into two general types: the rotating seal has a rotating seal ring and the stationary seal has a stationary seal ring.

Rotating Mechanical Seal

A cut-away view of a rotating mechanical seal is shown in Fig. 34. In this seal, leakage between sealing ring and shaft is prevented by a Teflon wedge ring; between mating ring and seal housing coved by a flat Teflon ring. The seal housing is provided with a quenching liquid inlet as required on pumps operating with a negative suction pressure. The liquid supplied to the seal prevents air infiltration and provides lubrication and cooling. If the liquid pumped is clear, the quenching liquid can be drawn directly from the pump discharge but if the liquid contains particles of foreign matter, a separator should be installed in the quenching line.

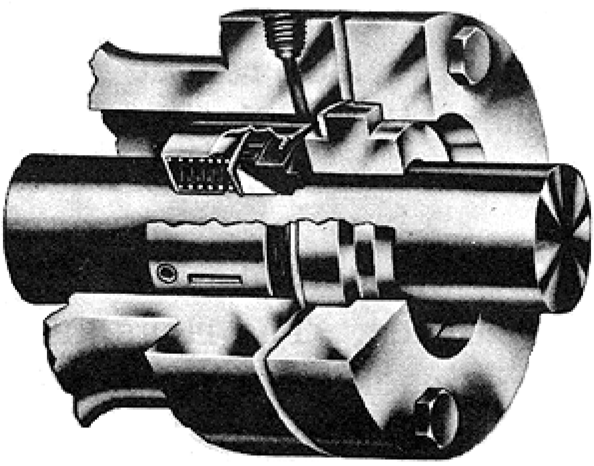


Figure 34
Rotating Seal

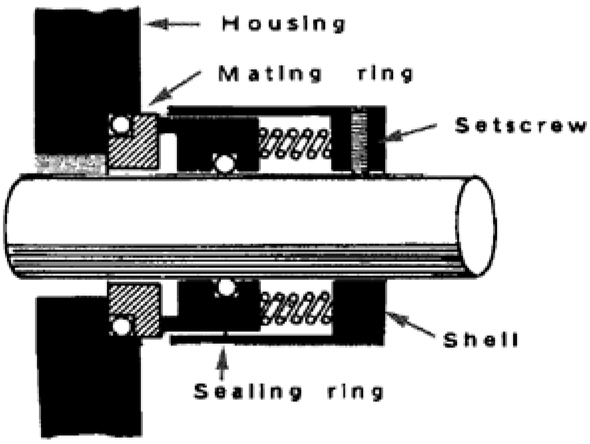


Figure 35
Rotating Seal

In the rotating seal sketched in Fig. 35, the sealing ring and spring are held in place by a shell, which is fastened to the pump shaft with a setscrew. Therefore the sealing ring will turn with the shaft. The mating ring, however, is held stationary within the pump casing.

As the pump shaft turns, the rotating sealing ring is forced against the mating ring, thus preventing leakage between the faces. "O" ring type seals prevent leakage between the casing and the mating ring and between the shaft and the sealing ring.

Stationary Mechanical Seal

In this type of seal, the shell containing the springs and sealing ring is held stationary in the annular space of the pump housing, as sketched in Fig. 36. The mating ring is fastened rigidly to the shaft, usually against a shoulder, so that it rotates with the shaft. The springs force the sealing ring against the mating ring so leakage between the faces is prevented. "O" rings are used to prevent leakage between the sealing ring and the shell and between the mating ring and the shaft.

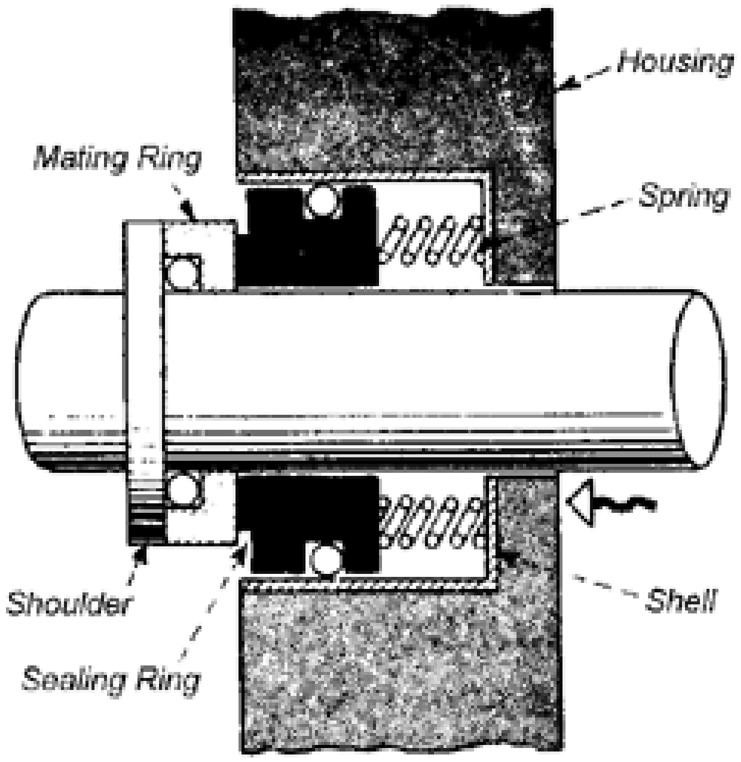


Figure 36 Stationary Seal

The choice of materials used for sealing and mating rings depends on many factors such as type of liquid pumped, temperature, pump speed and seal design. The friction between the faces of these rings should be kept as small as possible. Materials commonly used are bronze, carbon graphite, ceramics, stellite and tungsten carbide.

When a pump is equipped with mechanical seals, the following precautions should be taken before and during operation:

1. Never run the pump unless it is completely filled with liquid.

2. Vent all air out of the seal housings before start up.

3. Make sure an adequate flow of quenching or cooling liquid is flowing to the seals.

It is extremely important that the seals never run in a dry condition because this causes the faces to score and become grooved. Dry running seal faces are often indicated by a squealing sound. Absence of this sound should not be interpreted as an indication that sufficient liquid is supplied to the seals. A leaking seal may be caused by:

1. Seal faces that are scored or grooved.

2. Distortion of the rings due to unevenly tightened bolts of the seal housing gland.

3. “O” ring seals or other gaskets that are cut or nicked during installation.

4. Misalignment of piping resulting in distortion of pump parts.

5. Excessive pump shaft vibration.

Pump Bearings

The functions of the bearings of a centrifugal pump are: (1) to support the shaft carrying one or more impellers, (2) to allow the shaft to rotate with a minimum of friction, and (3) to keep the rotating shaft and impellers in correct position within the stationary parts of the pump. Pump bearings can be divided into two basic types: sleeve and shell bearings, and ball and roller bearings.

Sleeve and Shell Bearings

The bearings of small pumps usually consist of bronze bushings or sleeves fitted around the shaft with a small clearance, and held in place by the bearing brackets attached to the casing. On larger pumps the sleeve-type bearing consists of two half-shells made of cast-iron or steel and lined with babbitt, shown in Fig. 37This bearing is usually self-aligning so that it adjusts itself automatically to small changes in shaft position.

Sleeve and shell bearings are usually oil-lubricated. On small pumps drip lubricators supply the oil to the bearings. Medium-sized pumps use the lower part of the bearing housing as an oil reservoir and the oil is supplied to the bearing by endless chains or rings riding on the shaft, as shown in Fig. 39 Large pumps are usually equipped with a shaft-driven oil pump, which supplies the bearings with oil under pressure. The system usually has an oil tank as well as oil filters and coolers. A standby pump may also be supplied.

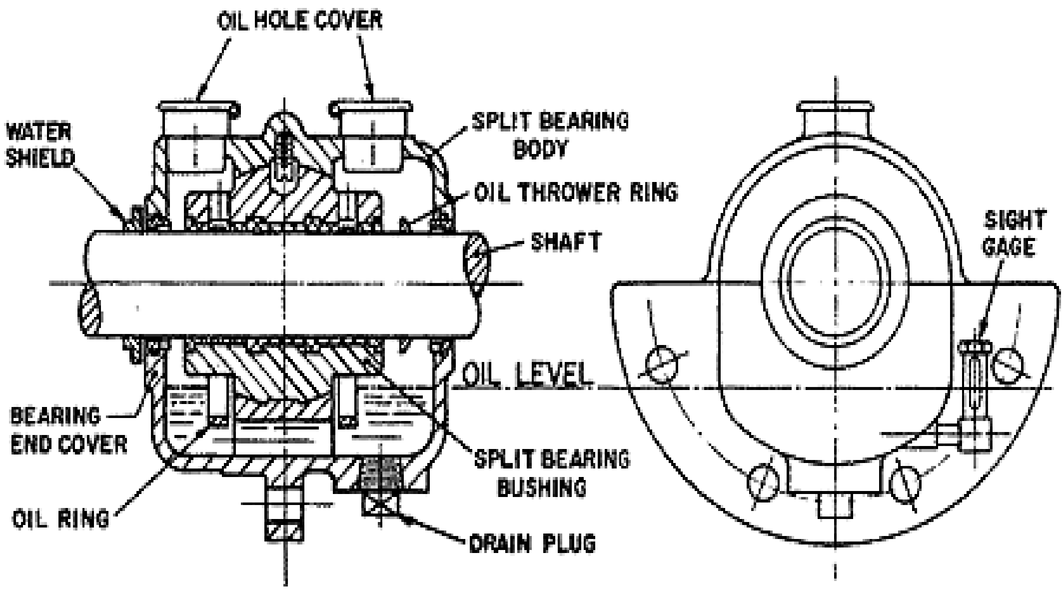


Figure 37 Self-Aligning Shell Bearing

Ball and Roller Bearings

Ball and roller bearings, also called anti-friction bearings, have replaced sleeve bearings in many modern pump designs. Ball bearings of single- or double-row design are mostly used on small and medium sized shafts; roller bearings are widely used on larger shafts. Anti-friction bearings may be lubricated either by oil or grease. Fig. 38 shows a single-stage centrifugal pump fitted with oil lubricated, single-row ball bearings.

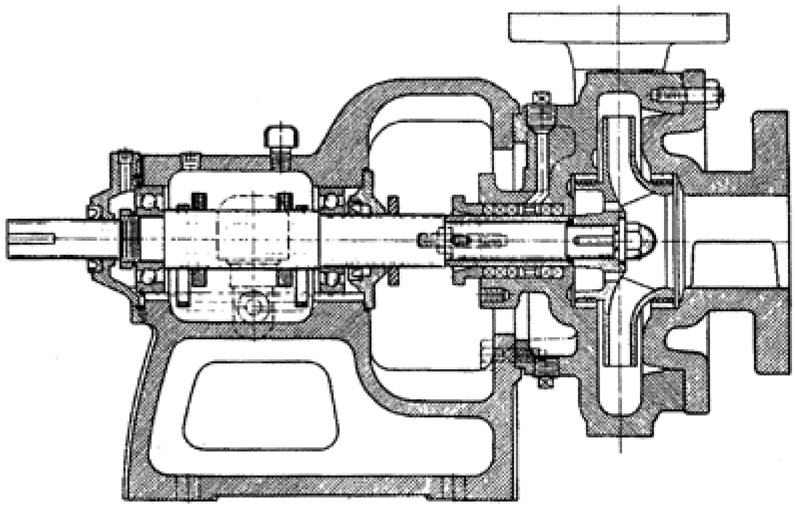


Figure 38
Single-Stage Centrifugal Pump with Ball Bearings

Vents and Drains

All centrifugal pumps are fitted with a drain line to allow the casing to be drained of the operating fluid. The drain line is fitted with an isolation valve as near the pump casing as is practical. A vent line is also connected to the top of the casing, also with an isolation valve. The vent valve can be opened when the drain valve is open to enable the pump casing to completely drain. The drain valve can be closed and the suction valve opened, when filling the pump again. The vent valve is closed when liquid reaches that point. The vent and drain piping is a very necessary part of any pumping system.

Objective Eight

When you complete this objective you will be able to...

Explain design features that eliminate thrust in large centrifugal pumps.

Learning Material

Axial Thrust

During operation a single-inlet impeller is subjected to hydraulic forces, which create an axial unbalance. This is illustrated in Fig. 39, which shows that the area of the eye of the impeller is subjected to suction pressure while the partial shroud on that side and the full shroud on the opposite side are subjected to discharge pressure. The resulting imbalance causes an axial thrust towards the suction that tends to move the impeller out of its proper position in the casing.

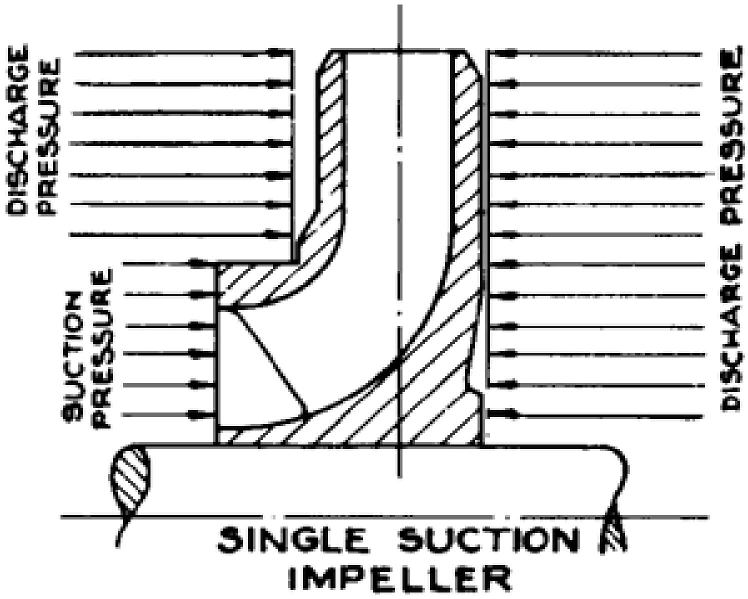


Figure 39
Unbalanced Axial Thrust

On low capacity, single-stage pumps, axial movement of the impeller is usually prevented by installation of a thrust bearing on the shaft. On larger capacity, single-stage pumps, however, the axial imbalance is eliminated by one of the following methods.

Single-Inlet Impeller with Backside Wearing Ring and Balancing Holes

As shown in Fig. 39, the single-inlet impeller is equipped with a wearing ring on its backside with the same diameter as the one on the suction side. By connecting the space inside this ring to the suction side by means of balancing holes, axial balance is achieved.

Double-Inlet Impeller

By using a double-inlet impeller the forces on the impeller are theoretically balanced, as shown in Fig. 34 In practice, the flow to each eye is not always equal so a light thrust bearing is still required.

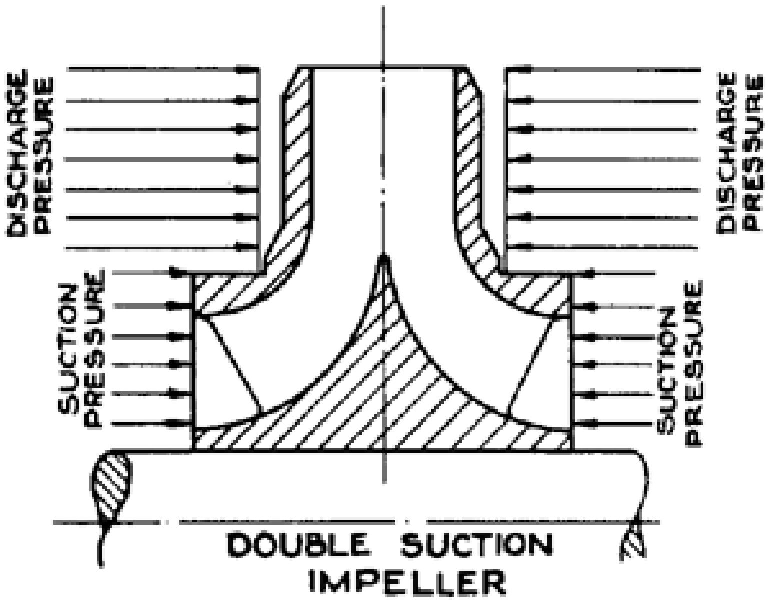


Figure 40
Balanced Axial Thrust

Opposing Single-inlet Impellers

On multi-stage pumps with single-inlet impellers, the axial thrust can be eliminated by the use of opposed impellers. The inlets of one half of the impellers face in one direction, the other half in opposite direction, Fig. 22 and 23. With this arrangement, axial thrust on the first half of the impellers is counter-acted by the opposing axial thrust on the second half.

On multi-stage pumps having the single-inlet impellers all facing in one direction, the axial thrust toward the suction end of the pump will be theoretically equal to the sum of the individual impeller thrusts. This total thrust is partially or fully counter-acted by one of the following hydraulic balancing devices: a balancing drum, a balancing disc or a combination of drum and disc.

Balancing Drum

A balancing drum, illustrated in Fig. 41, is installed on the shaft between the last impeller and the balancing chamber, which is connected to the suction side of the pump. The drum rotates inside the stationary member of the balancing device, the balancing drumhead. The drum and the head are separated by a small clearance allowing some leakage from the high-pressure side of the pump to the low-pressure chamber.

The drum is subjected to two forces: the discharge pressure acting on area B and the suction pressure acting on area C. Since the first force is greater than the second, an axial thrust is produced toward the discharge side of the pump. This thrust counter-acts the axial thrust exerted on the impellers.

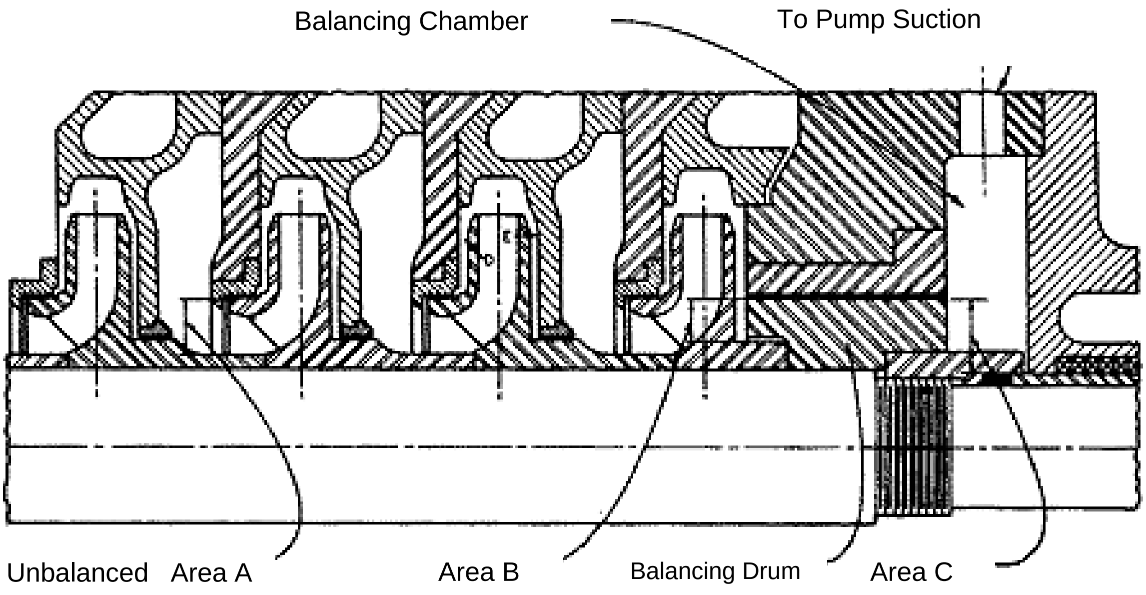


Figure 41 Balancing Drum

Balancing Disc

The simple balancing disc consists of a disc mounted on, and rotating with the shaft. It is separated from the balancing disc head attached to the casing by a small axial clearance. The leakage from the high pressure side of the pump flows through this clearance into the low pressure balancing chamber and from there to the suction of the pump via a restricting orifice that normally keeps the pressure in the balancing chamber well above the suction pressure. A simple balancing disc is shown in Fig. 42.

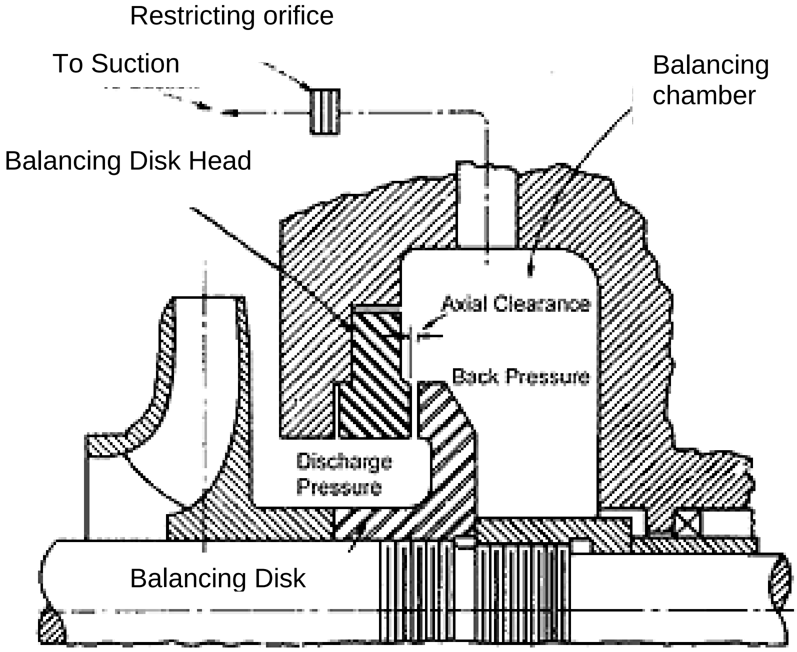


Figure 42 Simple Balancing Disc

The back of the disc is subjected to the balancing chamber back pressure while the centre part of the front of the disc is subjected to full discharge pressure and the ring area facing the head, to a pressure gradually dropping from discharge pressure to balancing chamber back pressure. The difference in forces acting on the front and back of the disc produces an axial thrust, which balances the axial thrust on the single-inlet impellers. When the thrusts are balanced, the clearance between disc and head will be a specific amount and the backpressure will be maintained at a specific value.

Should, during operation, the axial thrust on the impellers increase and exceed the thrust acting on the disc, the shaft would move slightly over toward the suction side of the pump causing the clearance between disc and head to be reduced. This will reduce the liquid leakage resulting in a drop in back pressure on the disc, which, in turn, causes the thrust on the disc to increase so that it moves away from the head, increasing the clearance again. The increased leakage builds up the backpressure again until equilibrium in thrusts is reached. The opposite will happen when the thrust on the impellers decreases below the thrust on the disc.

While the balancing drum provides only counter-thrust, the balancing disc provides not only counter-thrust but it also restores automatically, the position of shaft and impellers if it changes position due to variations in axial thrust. The use of a simple balancing disc, however, has certain disadvantages and it is therefore seldom used. Most multi-stage pumps are now equipped with a combination of balancing drum, and disc. This combination has all the advantages of both hydraulic devices without their disadvantages.

Objective Nine

Pump Start Up and Shut Down

1. Pump start up

After the pump has been installed and coupling alignment completed, an appropriate checklist may be consulted and these steps should be followed for a successful start up:

1. Check for sufficient proper lubrication for pump and driver.
2. Drive should be checked for correct rotation.
3. Pump suction valve should be fully opened.
4. Check pump and piping for leaks.
5. Pump casing should be vented. Open vent at top of pump casing until all air is expelled.
6. If product is hot, ample time should be allowed for pump casing to heat up. Pump casing and rotating assembly could distort from uneven heat transfer.
7. Before starting, rotate shaft by hand. It should be free with no rubbing.
8. Crack open discharge valve, don't fully open. A centrifugal pump uses less power at start-up with the discharge valve nearly closed.
   This practice will also prevent initial cavitation.

9. Start pump, watch discharge pressure gage. As soon as the pump pressure stabilises, open discharge valve slowly. Watch discharge gage, discharge pressure will fall off for few turns of the discharge valve until it is fully open.

Important! Never allow pump to run too long with discharge valve closed.

2. Pump shut down

The discharge valve on a centrifugal pump should be partially closed before the drive is stopped in order to prevent reversed flow. Usually, there is a check valve in the discharge line to prevent such reversed flow.

Stop motor, and then open the warm up valve if the pump is to be kept at the operating temperature. Close the valve of the cooling water supply to the bearings and of water cooling stuffing box.

Objective Ten

When you complete this objective you will be able to…

Describe a complete piping system for a multistage centrifugal pump, including the purpose and design of the automatic recirculation (minimum flow) valve.

LUBRICATION

The objection of lubrication is to reduce friction, wear and heating of mechanical parts which move relative to each other.

A lubrication is any substance, which, when inserted between the moving surface, accomplishes these proposes:

1- To reduce the wear and friction between the contact surfaces
2- To carry away the heat
3- To act as a seal for contamination .
4- To keep the surface clean
5- To carry and absorb heavy loads.

A. Lubrication System

"The lube oil pump draws oil from the power frames lube oil sump, through a suction strainer and supplies pressurized to the inlet of the power frames integral oil supply header. The lube oil supply header distributes the oil to the crossheads, Wrist pins and the main bearings through ports in the frame. A portion of the oil supplied to the
wrist pins passes through radial holes in the pins and is direct toward the crankpin . bearings through the rifle-drilled connecting rod.

Oil requires periodic replenishment at normal temperatures and very frequent replenishment at high temperatures. Oil is always subject to gradual deterioration from use and contamination from dirt and moisture. In time, the accumulated sludge will be harmful to the bearing and cause premature wear. For this reason, oil should be checked for contamination weekly and draining and flushing are necessary at regular intervals (at least every six months).

Lube-oil Systems. Figure 43 shows a. typical forced-feed lube-oil system for a horizontal multistage centrifugal pump. It consists of an integral gear-type oil pump mounted within the thrust-bearing housing and taking its suction from an oil reservoir mounted below the pump shaft, as shown, a: tubular oil cooler, piping, and pressure and level

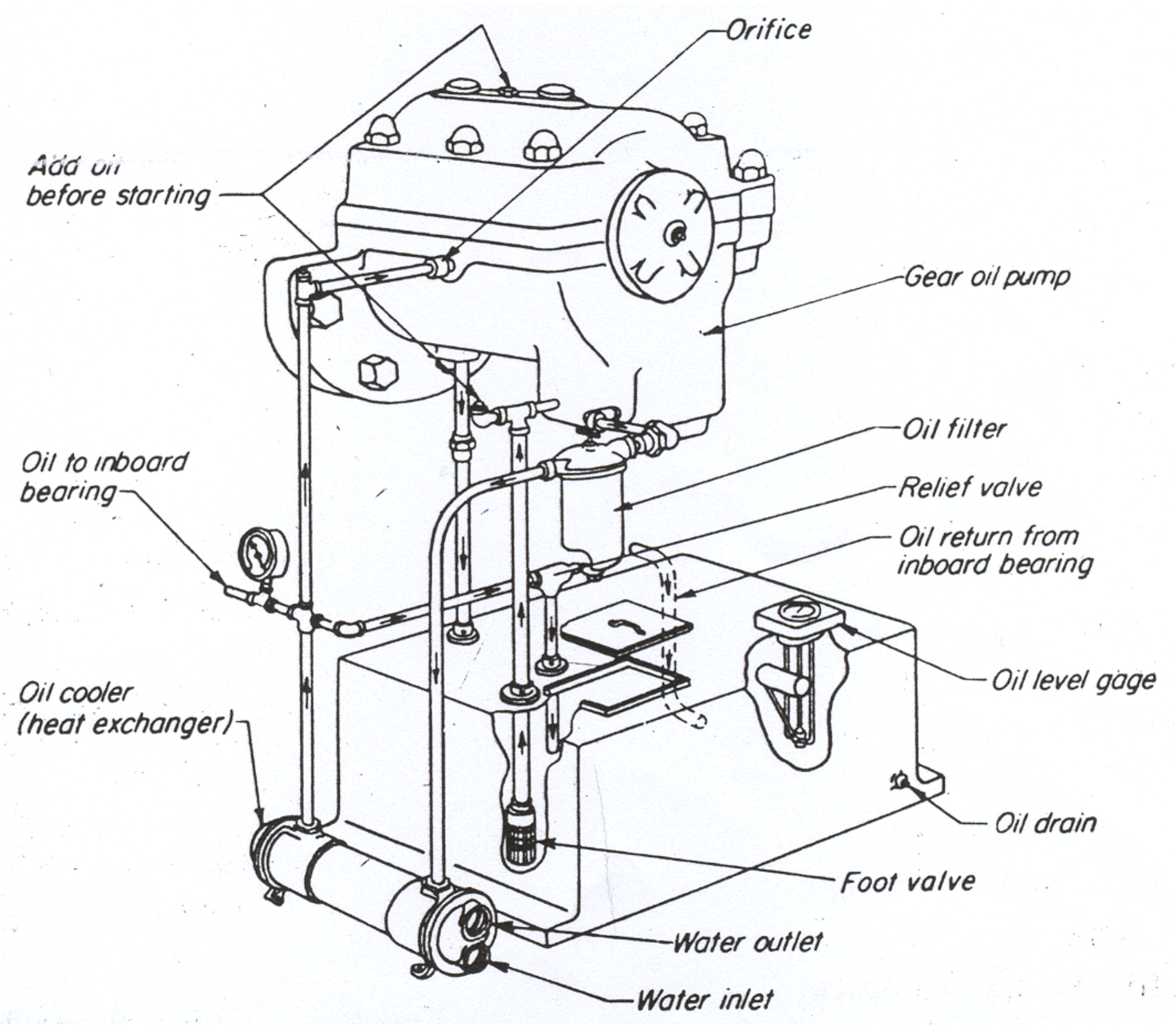


Figure 43 Typical forced lube-oil system. (Courtesy of Ingersoll – Rand Co.)

With this type of system, which is often used with pumps having sleeve bearings for the main shaft, clean and flush the bearings as outlined above, including the sump-tank reservoir and the oil cooler. Pour oil into each bearing bushing and fill the sump tank until it is three-quarters full, or more. *Important:* Fill the supply line to the earings so. it will be full when the pump starts and there will be no .

Delay in the oil reaching the bearings. The system supplies oil to both the bearings of the main pump and its driver. Some pumps "have a centrifugal-type impeller on the end of their shaft, instead of a gear

Control Valve System

It is imperative that the flow thorough a multistage pump be kept above a minimum flow rate. The manufacturer of the pump sets the flow rate. Flows below this minimum flow result in overheating of the fluid being pumped and eventually overheating and damage to the pump components. As it is critical to maintain this minimum flow, piping designs incorporate valve arrangements to guarantee the minimum flow will be met.

The standard method is to have a separate recirculation control valve, which allows for liquid to flow back to the pump suction drum. It is called a control valve or control loop system. For a boiler feed pump it means a recirculation line returning to the deaerator. A typical arrangement is shown in Fig. 44. The flow-sensing orifice on the suction of the pump sends a signal to the FIC (flow controller). The FIC will open the regulation control valve if the flow drops below the FIC set point. The control valve allows liquid to be returned to the deaerator, or recirculated. When the flow is above the set point of the FIC the control valve will be closed.

The total flow will be sent to the boiler.

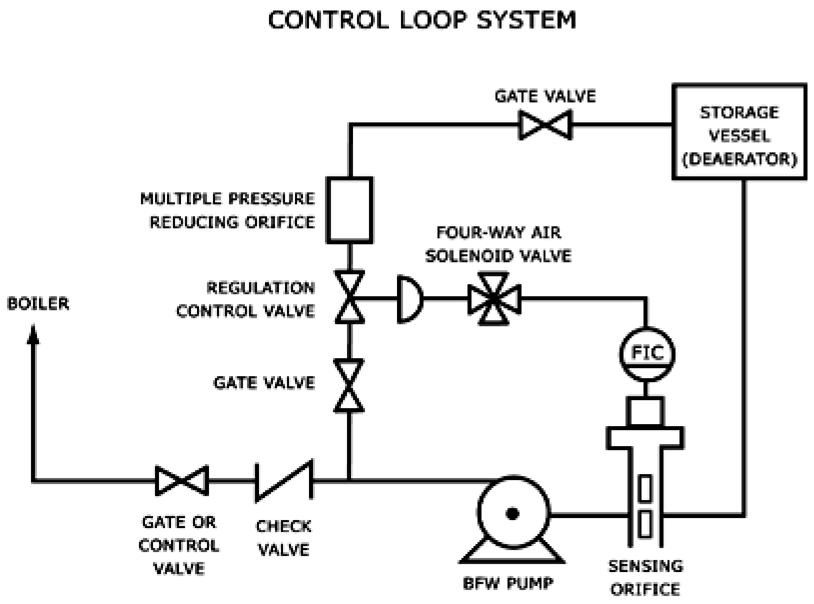


Figure 44
Pump With Check Valve and Separate Recirculation

Control Valve

Automatic Recirculation Valve System

Another way of accomplishing the same thing is shown in Fig. 45. It uses an automatic recirculation valve. This is a combination check valve and minimum flow valve in one unit. Below the pump minimum flow, the check valve (disc) is closed and the bypass valve is open. When the check valve opens, the bypass element gradually shuts. Thus the pump's minimum flow is always maintained. The ARC valve is sized specifically to match the flow rate of the pump.

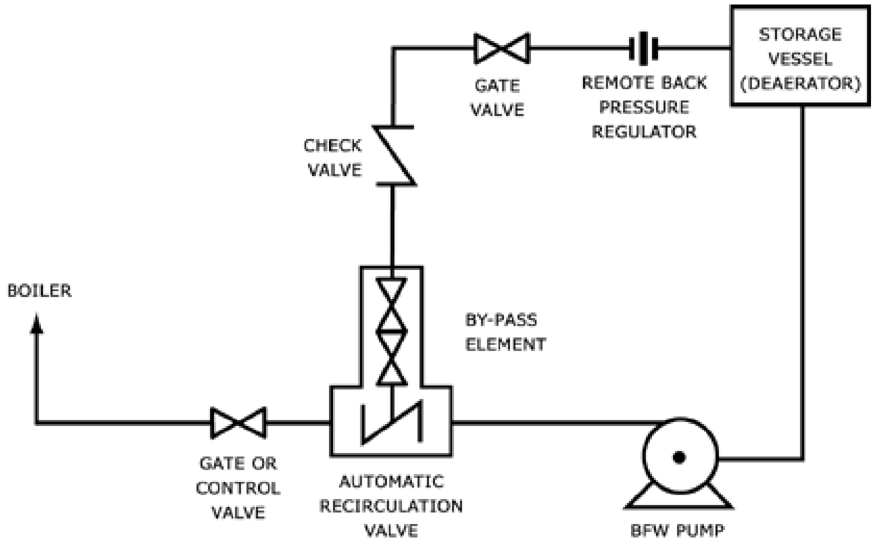


Figure 45
Pump With Automatic Recirculation Valve (Yarway ARC Valve)

Fig. 46 shows a cutaway of an automatic recirculation valve built by Yarway. This view shows the valve in the recircuation position, with the main flow disc closed and the recirculation or bypass valve open.

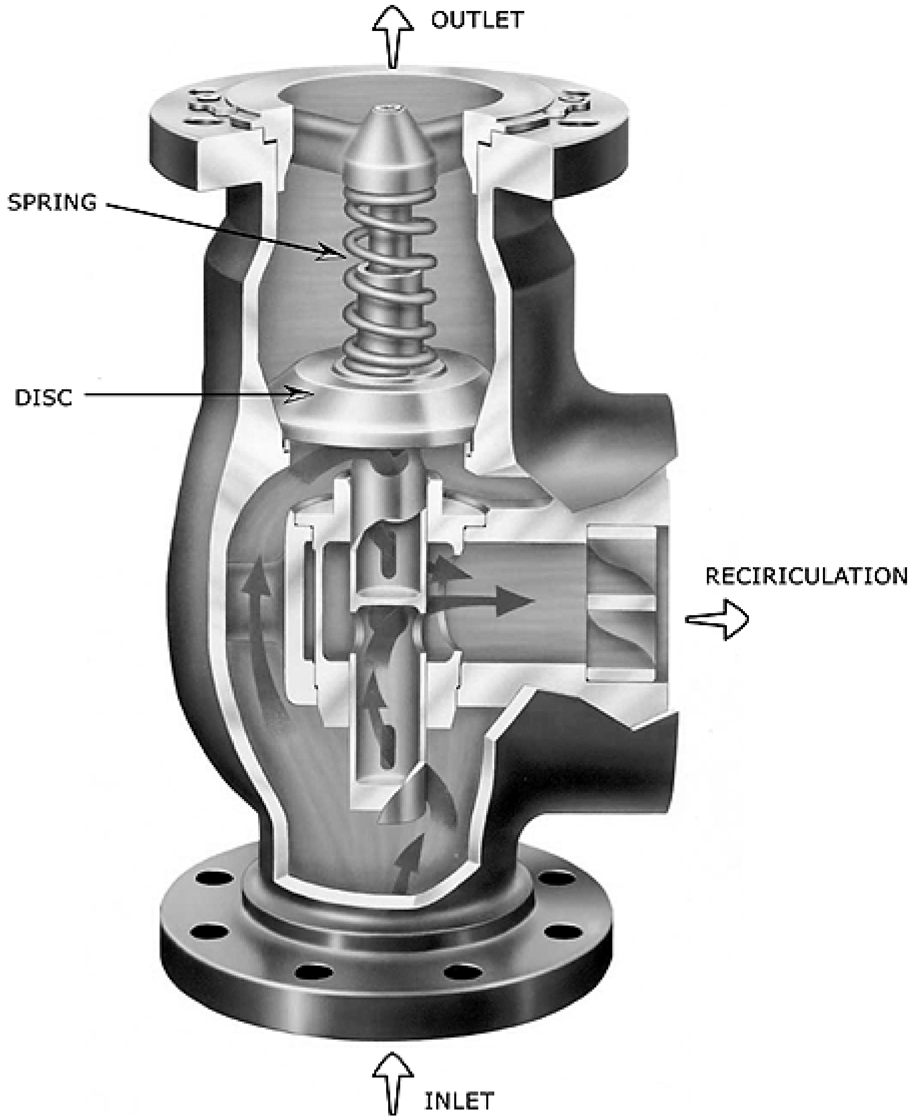


Figure 46
A.R.C. Valve Cutaway (Yarway Valve Co.)

Objective Eleven

When you complete this objective you will be able to…

Explain proper priming and start-up procedures and considerations for pumps.

Learning Material

Priming of Pumps

The term "priming," as used in connection with pumps, simply means the filling of pump casing and suction line with the liquid to be pumped, before the pump is started. Positive displacement pumps, reciprocating and rotary, are self-priming for total suction lifts up to about eight metres at sea level when in perfect condition. But with long suction lines, high lifts, or poor mechanical condition, they must be primed.

Centrifugal pumps are not self-priming. They must be primed before start-up otherwise the impeller will simply churn air and no suction will be produced. Also, when the pump is started without proper priming, the mechanical seals will run dry causing the seal faces to score or, if stuffing boxes are used, shaft and packing rings may wear. Some of the methods used to prime a centrifugal pump are shown in Fig. 47.

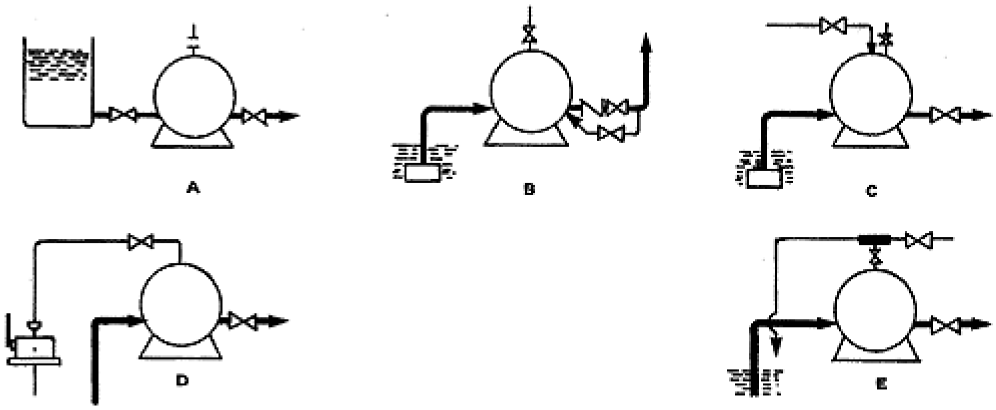


Figure 47 Pump Priming Methods

During priming of a centrifugal pump, the discharge valve is closed. When the pump is below the source of supply, as in Fig. 47 A, the pump has positive suction head. It is primed by opening the air vent valves on the pump casing, then slowly opening the suction valve. The incoming liquid forces the air out of the casing. When the liquid flows through the vents, they can be closed. The pump is then primed and ready to be started.

When the pump is located above the source of supply, (pump has suction lift), various methods of priming can be used. The suction line should be equipped with a foot valve, a flap-type valve attached to the lowest part of the suction line. The foot valve acts as a check valve. The foot valve allows the liquid to enter the line but prevents liquid from draining out of the line.

Fig. 47 B shows how the pump can be primed by filling suction line and casing with liquid supplied through a bypass around the discharge valve. Vents are kept open until the liquid escapes. In C, the suction line and casing are filled by liquid supplied through an auxiliary line. In D, a separate priming pump is used to draw the air from the casing, creating a vacuum, which draws the liquid in through the foot valve. The same result can be achieved by the use of an ejector as shown in E.

Capacity Regulation of Pumps

Reciprocating and rotary pumps are classified as positive displacement pumps, which mean that, at a constant speed, they move a specific amount of liquid regardless of pump head. Varying their speed usually regulates the capacity or flow rate through these pumps. The capacity of centrifugal pumps, however, changes when the pump head is changes. Hence, this pump is not a positive displacement pump. When the head is increased, the capacity of the pump decreases and when the head is lowered, the capacity increases. When the head is increased so much that it exceeds the design head of the pump, the output drops to zero.

Varying the speed can regulate the capacity of a centrifugal pump but this requires a pump driver capable of varying its speed (steam turbine, internal combustion engine, variable speed electric motor (VFD), etc. Most centrifugal pumps are driven by a constant speed electric motor.

When driven by a constant speed electric motor, the capacity of volute and diffuser pumps (the radial flow type) can be regulated by adjusting the discharge valve. Throttling the discharge valve increases the flow resistance, thus enlarging the friction head, and reducing the flow. The discharge pressure of the pump will increase moderately, but not enough to endanger the pump as with positive displacement pumps. Even with the discharge valve completely closed, the pressure increase will be well within safe limits.

When the flow is throttled, the power requirement of the radial flow centrifugal pump is also reduced, even with the resulting pressure increase. Advantage is taken of this fact by starting large, electrical motor-driven centrifugal pumps with closed or nearly closed discharge valve. Since the no-flow power requirement is relatively small, excessive power surging during start-up can be avoided.

The power requirement of axial and mixed flow centrifugal pumps, when operated at low capacity, is actually higher than at full capacity. These pumps should always be started with the discharge valve wide open.

Caution

Never run a centrifugal pump continuously with the discharge valve completely closed. The mechanical power applied to the impeller is dispelled as friction to the water trapped and churned about in the casing. This friction causes overheating of the water to the point where it turns into steam, which may result in damage to the pump. This is known as cavitation due to friction.

Always operate a centrifugal pump with its suction valve wide open. Never use the suction valve for flow control. Throttling or closing of the suction valve starves the impeller of its supply of water. The casing becomes partially empty resulting in excessive vibrations, which may ruin the bearings. The lack of liquid may also damage mechanical seals and stuffing boxes. If the water temperature is high enough, this reduction in pressure can cause cavitation to occur.

Assignment

1. using a sketch, describe how a diaphragm pump differs from a piston pump.

2. Sketch a sliding vane pump and describe its principle of operation.

3. Name four ways in which centrifugal pumps may be classified.

4. What is the purpose of a diffuser in a centrifugal pump?

5. Sketch and describe a centrifugal pump with opposed impellers. What is the advantage of this arrangement?

6. using a simple sketch, describe a stationary mechanical seal.

7. What are three ways of balancing thrust in a centrifugal pump?

8. using a simple sketch, describe a pumping system with an automatic recirculation valve.

9. What types of pumps require priming? Which types of pumps are self-priming?

10. How is the flow from a centrifugal pump controlled or regulated? Is this different than a positive displacement pump? Explain.

Pumps Heat Calculation

Table of Contents:

Pump Head Calculations

Learning Outcome

When you complete this learning material, you will be able to:

Define terms associated with pumping and perform pump head calculations.

Learning Objectives

You will specifically be able to complete the following tasks:

1. Explain the relationship between the height of a liquid, the density of the liquid and the pressure exerted at the bottom of the liquid. Perform simple calculations involving this relationship.
2. Define equivalent head and calculate equivalent heads for water and other liquids.
3. Define static suction head, static suction lift, static discharge head, total static head, pressure head, and calculate each of these for a given pump arrangement.
4. Define and calculate friction head and velocity head.
5. Define dynamic suction head, dynamic suction lift, dynamic discharge head, total dynamic head, and calculate each of these for a given pump arrangement.
6. Explain vapour pressure, cavitation, and net positive suction head. Calculate the required suction pressure for a water pump, given the manufacturers required NPSH.

Objective One

When you complete this objective you will be able to...

Explain the relationship between the height of a liquid, the density of the liquid and the pressure exerted at the bottom of the liquid. Perform simple calculations involving this relationship.

Learning Material

Pressure Due to Liquid Height

When a liquid exists at a certain level in a container, the liquid creates a downward force, due to its mass, on the bottom of the container. Because this force is distributed evenly over the base, a pressure results and is given by:

$$\text{Pressure} = \frac{\text{Force}}{\text{Area}}$$

The amount of force at the base, and hence the amount of pressure, is dependent on the height of the liquid level above the base. If the height increases, then the pressure also increases.

In Fig. 1, a square tank is filled to a level, ‘h’ meters, with liquid. The area of the tank is ‘A’ square meters and a pressure gauge is connected to measure the pressure, ‘P’, in kilopascals (kPa), at the base of the tank.

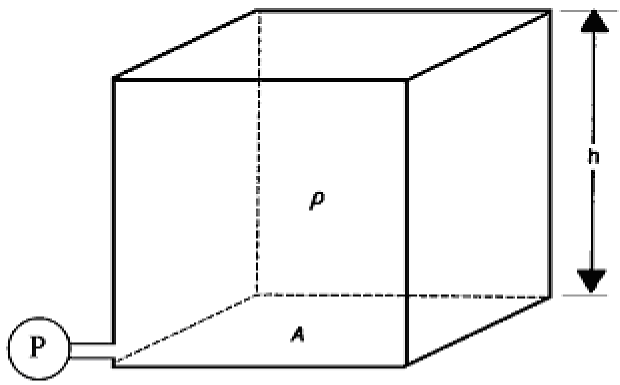


Figure 1
Liquid Head and Pressure in a Tank

Using the formula, Pressure (P) = Force (F) / Area (A), we can determine the pressure at the base of the tank, in relation to the height of liquid, as follows:

Force due to gravity acting on the liquid equals the mass times the acceleration due to gravity:

Force = mass x acceleration

or F = m g

by substitution, then:

$$\text{Pressure} = \frac{m\,g}{A}$$

The mass of the liquid can be found from its known density, where the density is the mass per unit volume, or

$$\text{Density } (\rho) = \frac{\text{mass } (m)}{\text{volume } (V)}$$

therefore,

$$\text{mass } (m) = \text{volume } (V) \times \text{density } (\rho)$$

Substituting this for mass in the pressure formula gives:

$$\text{Pressure } (P) = \frac{V\,\rho\,g}{A}$$

But, volume = area x height, or V = Ah

$$\text{so:} \quad P = \frac{A\,h\,\rho\,g}{A}$$

$$\text{which simplifies to:} \quad P = h\,\rho\,g$$

The pressure at the bottom of the liquid is therefore equal to the height of the liquid, times the density of the liquid, times acceleration due to gravity.

The units of measurement for these factors are as follows:

Pressure (P)	-	N/m^2 or Pa
Height (h)	-	m
Density (ρ)	-	kg/m^3
Gravity (g)	-	9.81 m/s^2 or 9.81 N/kg (see note that follows)

Note: It may be easier for the student to understand the gravity factor in this case by realizing that we are really interested in the force due to gravity, not the acceleration. Since the force exerted, due to gravity, by a one kilogram mass is 9.81 Newtons, then the units for ‘g’ can here be expressed as N/kg. Thus the 9.81 factor remains the same, but the units are more easily dealt with.

Example 1:

An open, cylindrical tank contains a liquid to a height of 7.5 meters. If the density of the liquid is 4.5×10^3 kg/m^3, what is the pressure exerted by the liquid at the bottom of the tank?

Solution:

$$P = h \rho g$$

$$= 7.5 \text{ m} \times 4.5 \times 10^3 \text{ kg/m}^3 \times 9.81 \text{ N/kg}$$

$$= 331\,100 \text{ N/m}^2 = 331\,100 \text{ Pa}$$

$$= \mathbf{331.1 \text{ kPa} \quad (Ans)}$$

Example 2:

A pressure gauge is located in the side of a tank at 2 m up from the bottom, which is the minimum tank level to ensure suction to the pumps. If the gauge reads 260 kPa and the level in the tank is 8 m, what is the density of the liquid in the tank?

Solution:

$$P = h \rho g$$

or:

$$\rho = \frac{P}{h\,g}$$

$$= \frac{260 \times 10^3 \ N/m^2}{(8-2)m \ \times \ 9.81 \ N/kg}$$

$$= \mathbf{4.417 \times 10^3 \ kg/m^3} \quad \textbf{(Ans)}$$

Self Test Problem

1. A process solution has a density of 3.4×10^3 kg/m^3 and is held in a rubber-lined tank that is 9 meters high. If the tank is exactly 2/3 full, what reading, in kPa, would the operator see on a level-indicating pressure gauge, located at the base of the tank?

(Ans…..200 kPa)

Objective Two

When you complete this objective you will be able to...

Define ‘equivalent head’ and calculate equivalent heads for water and other liquids.

Learning Material

Equivalent Head

In the previous discussion we used the word ‘height’ to refer to the level of the liquid. In practice, however, particularly when dealing with pumps, it is common to refer to this liquid height as the head of the liquid.

In Fig. 1, the pressure, P, at the bottom of the tank is a result of the height, or head, of the liquid surface above the location of the pressure gauge. While the pressure gauge will normally indicate a pressure, in kPa, it would be possible, and quite correct, to convert this pressure to an equivalent head measurement. The equivalent head is the height of liquid that produced the indicated pressure. In a simple system this can be found by rearranging the formula, P = h r g, as follows:

$$\text{Head (h)} = \frac{\text{pressure}}{\text{density x gravity}}$$

or:

$$h = \frac{P}{\rho\, g}$$

Yes, this is the same equation we established in Objective 1, but here we are simply changing the terminology to indicate head pressure.

Equivalent Head for Water

The density of water at 4 °C is 1000 kg/m^3. Therefore, a column of water 1 m. high, at this temperature, would exert a pressure of:

$$P = h\,\rho\,g$$

$$= 1\text{ m x } 1000\text{ kg/m}^3 \text{ x } 9.81\text{ N/kg}$$

$$= 9810\text{ Pa} = 9.81\text{ kPa}$$

It follows that, since 9.81 kPa is equal to 1 m. of head

1 kPa = 1 / 9.81 m = 0.102 m of head

That is, the equivalent head of 1 kPa pressure, for water at 4 °C, is 0.102 meters.

Note: When considering water at any temperature other than 4°C, it is necessary to realize that the density of the water will not be exactly 1000 kg/m^3. The correct density may be found using the value of specific volume, v_f, listed in Table II of the Steam Tables, and the formula:

$$\rho = \frac{1.0}{v_f} \ gm/cm^3 \quad or \quad \frac{1000}{v_f} \ kg/m^3$$

For example, if the water temperature is 80 °C, the density will be:

$$\rho = \frac{1000 \ kg/m^3}{1.0291} = 971.7 \ kg/m^3$$

Example 3:

What height of water is necessary to exert a pressure of 1 kPa at a temperature of 150°C? (Use Steam Tables to find v_f.)

Solution:

$$P = h \rho g$$

or: $$h = \frac{P}{\rho g}$$

we know: $$P = 1 \text{ kPa or } 1000 \text{ Pa} = 1000 \text{ N/m}^2$$
$$g = 9.81 \text{ N/kg}$$

and: $$\rho = \frac{1.0}{V_f \text{ at } 150°C}$$

$$= \frac{1.0}{1.0905 \text{ cm}^3/\text{g}}$$

$$= 0.917 \text{ g/cm}^3 \text{ or } 917 \text{ kg/m}^3$$

therefore: $$h = \frac{1000 \text{ N/m}^2}{917 \text{ kg/m}^3 \times 9.81 \text{ N/kg}}$$

$$= \mathbf{0.1112 \text{ m (Ans.)}}$$

Example 4:

The pressure at the discharge of a water pump is 150 kPa and the temperature of the water is 80°C. Find the equivalent head in meters.

Solution:

$$h = \frac{P}{\rho g}$$

$$P = 150 \text{ kPa or } 150\,000 \text{ Pa}$$

$$g = 9.81 \text{ N/kg}$$

$$\rho = \frac{1.0}{1.0291 \times 10^{-3} \text{ m}^3\text{/kg}}$$

$$= 972 \text{ kg/m}^3$$

therefore: $$h = \frac{150\,000}{972 \times 9.81}$$

$$= \mathbf{15.73\ m} \quad \mathbf{(Ans.)}$$

Equivalent Head for Liquids Other Than Water

If the liquid is anything other than water, then the liquid's relative density must be known in order to determine heads and pressures. Remember that the relative density of a substance is the density of the substance divided by the density of water, or:

$$\text{Relative density} = \frac{\text{density} \ (\text{kg/m}^3)}{1000 \text{ kg/m}^3}$$

therefore:

$$\text{Density } (\rho) = \text{Relative density} \times 1000 \text{ kg/m}^3$$

Example 5:

If a column of mercury has an equivalent head of 50 cm. calculate the pressure produced at the bottom of the column. Relative density of mercury is 13.6.

Solution:

$$P = h \rho g$$

where: $h = 0.5\ m$
$g = 9.81\ N/kg$
$\rho = 13.6 \times 1000\ kg/m^3$

$$\therefore P = 0.5\ m \times 13\,600\ kg/m^3 \times 9.81\ N/kg$$

$$= \mathbf{66\,708\ Pa\ or\ 66.7\ kPa} \quad \textbf{(Ans.)}$$

Note: It is interesting to realize that the pressures exerted by equal heights of different liquids differ by a factor of their densities. The greater the relative density of the liquid, then the greater will be the pressure exerted by the column. For example, if the column of mercury in Example 5 were replaced with water, the pressure would be:

$$P = h \rho g = .5 \times 1000 \times 9.81 = 4905\ Pa \text{ or } 4.905\ kPa$$

Notice that the pressure due to the mercury is equal to 13.6 times the pressure due to the equal height of water.

Example 6: Two open tanks are filled to the same level, but with different liquids. In tank A the liquid has a relative density of 4.32 and the pressure at the base of the tank is 58 kPa. What is the pressure at the base of tank B if the liquid in it has a relative density of 1.45?

Solution:

$$\text{Pressure at tank B} = \frac{\text{relative density of B}}{\text{relative density of A}} \times \text{pressure at tank A}$$

$$P = \frac{1.45}{4.32} \times 58\ kPa$$

$$= \mathbf{19.47\ kPa} \quad \textbf{(Ans.)}$$

Self Test Problems

2. Water in a treated water storage tank is at 90 °C. If the pressure at the bottom of the tank is 42 kPa, what is the water level in the tank. (given that V_f at 90 °C = 1.036)
(Ans. = 4.46 m)

3. A raw water supply pump is sitting idle with its discharge valve open. The equivalent discharge head is known to be 35 meters and the water is at 70 °C. What pressure, in kPa, should be indicating on the pump discharge gauge ? (given: v_f at 70 °C = 1.0228)
(Ans. = 336 kPa)

4. A chemical in a storage tank has a relative density of 7.60 . What is the equivalent head at the outlet of the tank where a pressure gauge reads 250 kPa ?
(Ans = 3.35 m)

5. Two tanks sit side by, with equal levels in the two tanks. However, pressure gauges (assumed accurate) at the bottom of the tanks read differently. The gauge on tank A reads 120 kPa, while the gauge on tank B reads 210 kPa. Compare the densities of the liquids in the two tanks.
(Ans. Density of tank B = 1.7 times the density of tank A)

Objective Three

When you complete this objective you will be able to…

Define static suction head, static suction lift, static discharge head, total static head, and pressure head, and calculate each of these for a given pump arrangement.

Learning Material

Introduction to Pump Head

In general, a pump is designed to move a liquid through a certain maximum head. That is, the size, speed and design of the pump are chosen so it is capable of pumping the liquid through an equivalent height of the liquid. This total height (or head) of liquid becomes a measure of the pump's ability to move the liquid.

For example, if two pumps in a system are taking a liquid from the same suction source, but one pump is delivering the liquid to a height of 50 feet and the other to a height of only 30 feet, then the pump discharging to 50 feet must be designed for a greater head.

There are many factors that determine the equivalent head through which a pump must deliver a liquid and for which the pump must be designed. These factors include liquid levels, vessel pressures, pipe friction, and liquid velocity.

Static Heads

The word "static" suggests zero motion. In a pumping system, the pump is required to move a liquid from a source location, on the suction side of the pump, to a destination location, on the discharge side of the pump. In most cases, both the source and the destination contain a level of liquid in some reservoir, or vessel. The levels of the liquid in the suction and discharge sides of the pump create respective head pressures that are equivalent to the height of the liquid above the pump. Since these heads exist, even if the pump is not in operation, they are referred to as static heads.

Fig. 2 shows the two most common arrangements of a pump in relation to the suction and discharge reservoirs. In both cases we will assume that the reservoirs are open-top tanks, exposed only to atmospheric pressure.

Explanations of static head terms follow the figure.

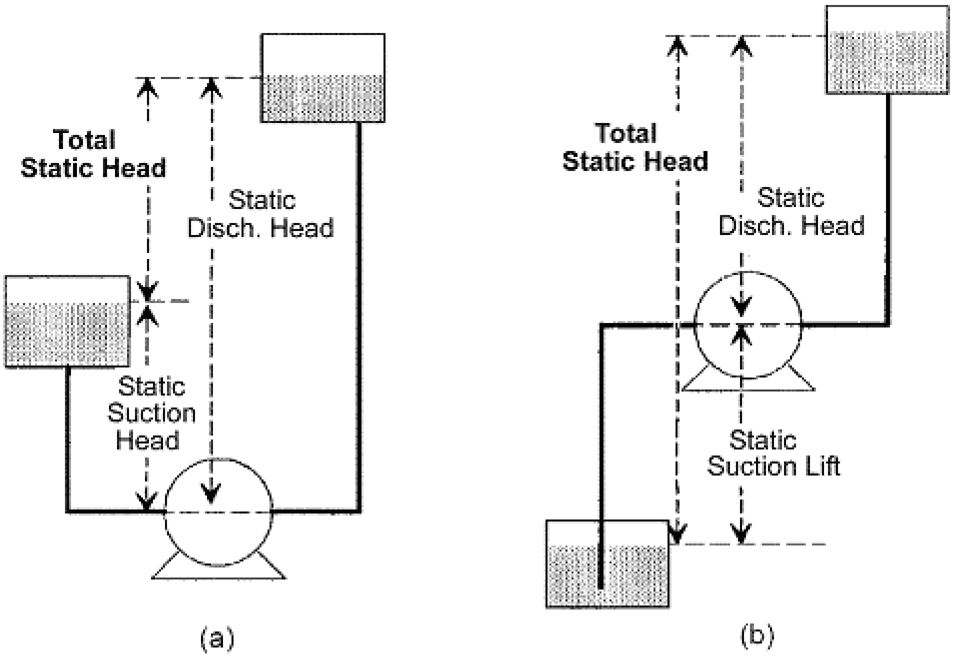


Figure 2
Pump Static Head Arrangements

Static Suction Head

In Fig. 2(a), the pump is located below the suction source and the height of the liquid creates a positive head on the suction side of the pump. This head is termed the static suction head. It is defined as the vertical distance, in meters, from the centerline of the pump UP to the surface of the liquid in the suction source above the pump. Static suction head assists the pump by forcing water down into the suction.

Static Suction Lift

In Fig. 2(b), the pump is located above the suction source. In this case, the pump is required to “lift” the liquid from the source to the pump inlet before it can deliver the water into the discharge. The equivalent head required to lift the liquid is called the static suction lift. It is defined as the distance, in meters, from the centreline of the pump DOWN to the surface of the suction liquid below the pump.

Static Discharge Head

In both (a) and (b) of Fig. 2, the pump is discharging to a tank that is located above the pump. The pump is required to overcome the head created by the height of the liquid, in order to pump liquid up to the tank. This head is called the static discharge head. It is defined as the distance, in meters, from the centerline of the pump UP to the surface of the liquid in the discharge tank, located above the pump, or to the height of free discharge.

Note: Sometimes the discharge may have a "down spout" and this will decrease the Static Discharge Head due to syphon action.

‘Free discharge’ refers to the open end of a pump discharge line. Fig. 3 demonstrates this with a pump discharging into the side of a tank at a point that is above the liquid level in the tank. Here the static discharge head is the height, above the pump, at which the line enters the tank, since this is the height to which the liquid must be pumped.

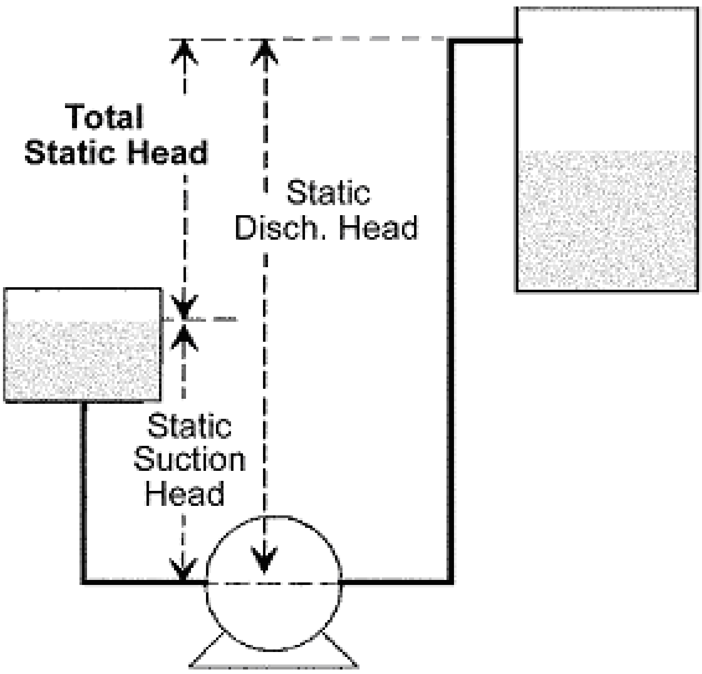


Figure 3
Static Discharge Head above liquid level

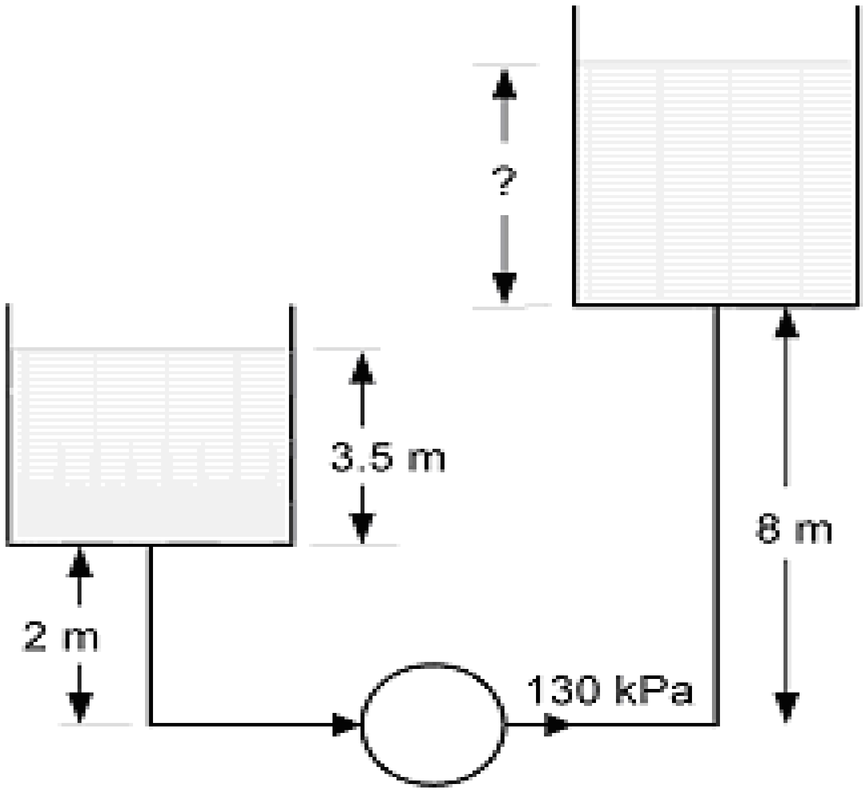
Total Static Head

The total head effect of liquid levels on a pump is referred to as the total static head. By definition, the total static head is the distance, in meters, from the surface of the suction source to the surface of the discharge tank (or to the point of free discharge). Referring back to Fig. 2, one can see that:

Example 7:

A pump sits in a sump and takes suction from an open tank, located 2 meters above it and having a water level of 3.5 meters. The pump discharges into the bottom of an open tank that is 8 meters above the pump. When the pump is shut down and its suction valve (located at the pump) is closed, the discharge pressure gauge indicates 130 kPa. Assuming the water is at 4oC, what is the level in the discharge tank and what is the total static head for the pump?

Solution:



Sketch 7.0

For water at 4 °C, 1 kpa = .102 m of head

therefore: 130 kPa = 130 x .102 = 13.26 m

and the tank level = 13.26 m - 8 m = **5.26 m** (Ans)

Total static head = static discharge head - static suction head

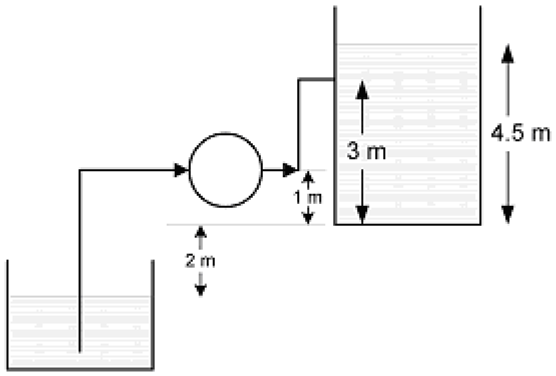
= 13.26 m - (2 + 3.5) m

= **7.76 m (Ans.)**

Example 8:

A rotary pump draws oil from a waste collection sump and delivers it into a storage tank. The level in the sump is 2 m below the base of the storage tank and the level in the tank is 4.5 m. If the pump sits 1m above the tank base and discharges into the side of the tank at a height of 3 m, what is the total static head on the pump?

Solution:



Sketch 8.0

Static suction lift = 2 m + 1 m = 3 m

Static discharge head = 4.5 m - 1 m = 3.5 m

Total static head = 3 m + 3.5 m = **6.5 m (Ans)**

Pressure Head

So far we have considered suction and discharge reservoirs that are open to atmosphere. However, in many process situations a pump is required to take suction from one pressure vessel and transfer it to another. The pump must move a liquid against the internal pressure of the vessel into which it is discharging. This pressure creates an equivalent head on the discharge side of the pump. Also, if the suction source is pressurized, then this suction pressure creates an equivalent suction head on the pump.

Pressure head is defined as the equivalent head, in meters of liquid, exerted on a pump by the internal pressure of a closed vessel into which the pump discharges or from which it takes suction.

Fig. 4 shows a similar arrangement to Fig. 2, except that the suction and discharge vessels are pressurized.

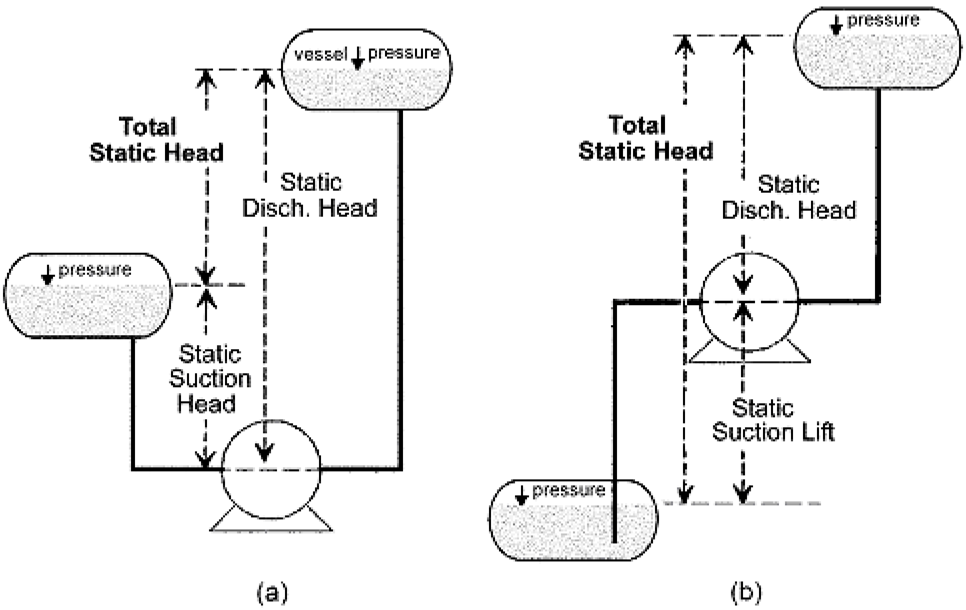


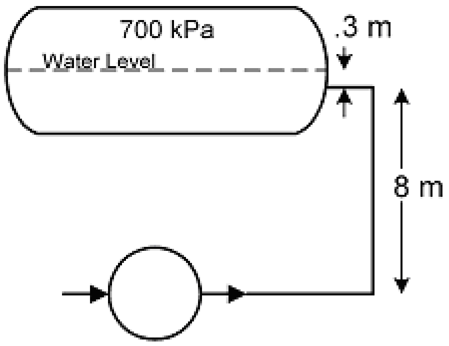
Figure 4
Pressure Heads on a Pump

Notice in Fig. 4 that the static suction and discharge heads still exist, due to the levels of the liquids in the vessels. However, the total head on each side of the pump is increased by the addition of the internal pressure. In the case of suction lift in Fig. 4(b), the pressure in the suction vessel aids in the lift of the liquid. If the pressure head in the suction vessel is greater than the static suction lift, there will be a positive suction head on the pump.

Example 9:

A feedwater pump supplies water to a boiler, which is operating at 700 kPa. If the water enters the boiler at a height of 8 m above the pump and 30 cm below the boiler water level, what is the discharge head of the pump? Ignore the effects of water temperature.

Solution:



Sketch 9.0

1 kPa = 0.102 m head

therefore: pressure head = 700 x 0.102 = 71.4 m

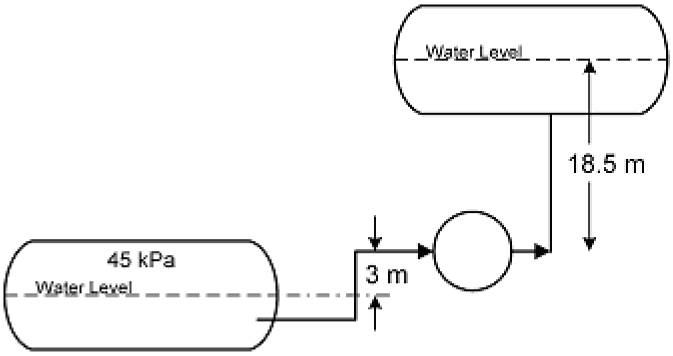
static disch. head = 8 m + .30 m = 8.30 m

and: total disch. head = 71.4 m + 8.30 m = **79.7 m (Ans.)**

Example 10:

Calculate the pressure differential across the pump in a pressurized water system if the liquid level in the suction vessel is 3 m below the pump and the vessel has a gas blanket of 45 kPa, the discharge pressure head is 150 m and the discharge level is 18.5 m above the pump.

Solution:



Sketch 10.0

Suction pressure head	=	45 x 0.102 m	=	4.59 m
Total suction head	=	4.59 - 3 m	=	1.59 m
so: suction pressure	=	1.59 / .102	=	15.58 kPa
Total disch. head	=	150 m + 18.5 m	=	168.5 m
Discharge pressure	=	158.5 x 9.81		
	=	1653 kPa		
Pressure differential	=	1653 - 15.58		
	=	**1637.42 kPa (Ans.)**		

Self Test Problems

6. A pump is sitting at rest with its discharge valve open and the pressure gauge showing 210 kPa. The total static head of the pump in this situation is 11 m. If the pump takes suction from a transfer tank located 4 m above it, what is the current level in the transfer tank? The liquid is water at 42 °C.

(Ans. = 6.42 m)

7. A pump sits in a pit at the bottom of a chemical sump, which has a level of 2.6 m. At rest, the pump discharge indicates a pressure of 30 kPa. The pump delivers to a storage tank located above it. If the relative density of the chemical is 1.73, find a) the static discharge head of the pump, and b) the total static head

(Ans. a) = 17.68 m, b) = 15.08 m)

8. A deaerator has a pressure of 100 kPa and supplies feedwater to a boiler, which operates at 1200 kPa. If the standby feedwater pump is located 7m below the water level in the deaerator and 6 m below the water level in the boiler, what is the total head on the pump, ignoring water temperature.

(Ans a) = 111.2 m)

Objective Four

When you complete this objective you will be able to…

Define and calculate friction head and velocity head.

Learning Material

Friction Head

When a liquid flows through a system it encounters resistance to flow due to physical contact with the surfaces of pipes, valves and fittings. This resistance is commonly referred to as friction. The frictional resistance causes a pressure drop in the liquid, in the direction of flow.

The pump must overcome the pressure drop and, therefore, it creates an equivalent head on the pump. This **friction head** can be defined as the equivalent head, in meters of the liquid being pumped, due to the resistance to flow caused by pipes, valves and fittings.

In general, the friction head will increase if the length of the piping (ie. the distance pumped) is increased, the diameter of the piping is reduced, flow velocity is increased, or more fittings or valves are added to the system.

For design purposes, there are tables available that specify the friction losses for liquids in terms of the piping material and diameters. The loss is given in terms of head, or meters of liquid, per 25 meters of pipe length. For example, the following excerpt is from a table for water in a steel, schedule 40 system.

Diameter (mm)	Flow (L/sec)	Friction loss (m. water / 25 m. of pipe)
50.8	25	36.5
101.6	25	1.22
101.6	75	12.6
152.4	75	1.54
203.2	75	0.39

Table 1
Friction losses in steel, sched. 40 pipe (excerpt)

Similarly, tables are available that indicate the resistance to flow for various pipe fittings. In this case the flow resistance is given in terms of the length of pipe of the same diameter that would offer the same resistance as the fitting. One can then use the piping table to convert the fitting resistance to head (in m. of the liquid).

Diameter (mm)	Standard Elbow	Tee	Open Globe Valve	Open Gate Valve
50.8	1.68	3.35	17.4	0.37
101.6	3.35	6.7	33.5	0.7
152.4	4.88	10.1	48.8	1.07
203.2	6.4	13.1	67.0	1.37

Table 2
Friction losses in steel pipe fittings (excerpt)

By adding up all the fittings and the length of pipe in a system and knowing the pipe diameter and flow rate, one can estimate the friction loss (or friction head).

Example 11:

Schedule 40, steel pipe, 50.8 mm diameter. is used to construct a piping system. The length of piping is 60 m and there are 4 standard elbows, 1 tee and 1 globe valve. If the flow is 25 L/sec, what is the friction head of the system?

Solution:

Equivalent resistance in elbows = 4 x 1.68 = 6.72 m of pipe
Equivalent resistance in tee = 1 x 3.35 = 3.35 m of pipe
Equivalent resistance of valve = 1 x 17.4 = 17.4 m of pipe
Meters of piping = 60
Total equivalent meters of pipe = 87.47 m

From Table 1, at 25 L/sec, friction head = 36.5 m per 25 m of pipe
therefore: Friction head = (87.47 / 25) x 36.5
= **127.7 m (Ans.)**

Velocity Head

A pump must impart motion to a liquid in order to transfer the liquid from one location to another. When a substance is in motion it contains energy, more specifically kinetic energy. Therefore, a pump must create this kinetic energy in a liquid. Remember that the kinetic energy of a mass is given by the formula:

$$KE = \frac{1}{2} m v^2$$

where: m = mass, in kg
v = velocity, in m/sec.

Also, if a substance is at a height it contains potential energy, which is given by the formula:

$$PE = m\ g\ h$$

where: m = mass, in kg
g = gravity or 9.81 m/sec^2
h = height, in m

In a pumping system, the kinetic energy of the liquid must equal the potential energy at the height of the liquid. That is:

$$KE = PE$$

or: $$\frac{1}{2} m v^2 = mgh$$

and, solved for height: $$h = \frac{v^2}{2g}$$

This formula defines the equivalent head, in meters, that a liquid exerts as a result of its velocity in a piping system. It is called the velocity head of a pump. In other words, the velocity head is the pressure (in m of liquid) required giving a liquid its motion through a system at a given velocity.

Normally a pump has two velocity heads, one on the suction side and one on the discharge side. This is because the suction piping is usually larger in diameter, so that for a given flow the velocity is lower in the suction line than in the discharge line. The suction velocity head is required to move the liquid into the suction of the pump. The velocity head required on the discharge side is then only equal to the head required to increase the velocity to that of the discharge line.

Example 12:

If a centrifugal pump is required to move 20,000 kg of water per hour at a velocity of 12 m/sec through a 4 cm diameter line, what velocity head must be considered in the system design?

Solution:

$$h = \frac{v^2}{2g}$$

$$= \frac{144}{2 \times 9.81}$$

$$= \mathbf{7.34\ m} \quad \mathbf{(Ans.)}$$

Example 13:

What is the velocity in the discharge of a centrifugal pump if the velocity head is 12 m?

Solution:

$$h = \frac{v^2}{2g}$$

therefore: $v^2 = 2\,g\,h$

$$v = \sqrt{2\,g\,h}$$

$$= \sqrt{2 \times 9.81 \times 12}$$

$$= \mathbf{15.34\ m/sec} \quad \mathbf{(Ans.)}$$

Self Test Problems

9. A 152.4 mm dia. piping system is 50 m long and contains 3 elbows, 2 gate valves, one globe valve, and 2 tees. If flow in the system is 75 L/sec., calculate the friction head.
(Ans. = 11.37 m)

10. A pump has a discharge velocity head of 12 m and a suction velocity head of 7.5 m. How does the suction velocity compare to the discharge velocity.
(Ans. = discharge velocity is 1.26 times the suction velocity)

Objective Five

When you complete this objective you will be able to…

Define dynamic suction head, dynamic suction lift, dynamic discharge head, total dynamic head, and calculate each of these for a given pump arrangement.

Learning Material

Dynamic Suction Head

"Dynamic" generally refers to a system in motion. All the pump terms so far discussed collectively play a role in the dynamics of a pump, when the pump is not in service. The following terms are specifically used to define a dynamic pumping system.

When a pump is located below the suction source, the dynamic suction head is the numerical result, in meters, of all the equivalent heads acting on the suction side of the pump. Therefore:

```
Dynamic suction head  =  static suction head
                         PLUS  suction pressure head
                         MINUS  suction friction head
                         MINUS  suction velocity head
```

Notice that the static suction and pressure heads contribute to the dynamic suction head, while the suction friction and velocity heads reduce the dynamic suction head

Dynamic Suction Lift

When a pump is located above the suction source, the dynamic suction lift is the numerical result, in meters, of all the equivalent heads and lifts acting on the suction side of a pump.

```
Dynamic suction lift  =  static suction lift
                         PLUS  suction friction head
                         PLUS  suction velocity head
                         MINUS  suction pressure head
```

Notice that only the suction pressure head acts to reduce the dynamic suction lift

Dynamic Discharge Head

The dynamic discharge head is the numerical sum, in meters, of all the equivalent heads acting on the discharge side of the pump.

Dynamic discharge head = static discharge head
PLUS discharge pressure head
PLUS discharge friction head
PLUS discharge velocity heads

Total Head (or Total Dynamic Head)

The total head on an operating pump (also called the total dynamic head) can be calculated as follows.

- For a pump with suction head:

Total head = dynamic discharge head - dynamic suction head

- For a pump with suction lift:

Total head = dynamic discharge head + dynamic suction lift

Example 14:

A vacuum evaporator has an absolute pressure of 25 kPa. The level in the evaporator is 10 m above the centerline of the pump. The pump discharges into a storage tank, located 20 m above the pump. This tank has a level of 6 m and a pressure blanket of 100 kPa (gage). Liquid velocities are 3 m/s in the suction piping and 4 m/s in the discharge line. Friction loss in the suction line is 0.5 m and in the discharge line, 1.8 m. Calculate the total dynamic head for the pump.

Solution:

Total head = dynamic discharge head - dynamic suction head

First, Find Dynamic Suction Head:

Dynamic suction head = static suction head - suction friction head - suction velocity head + pressure head

static suction head = 10 m

suction friction head = 0.5 m

$$\text{suction velocity head} = \frac{v^2}{2g} = \frac{(3\ m/s)^2}{2 \times 9.81\ m/s^2} = 0.459$$

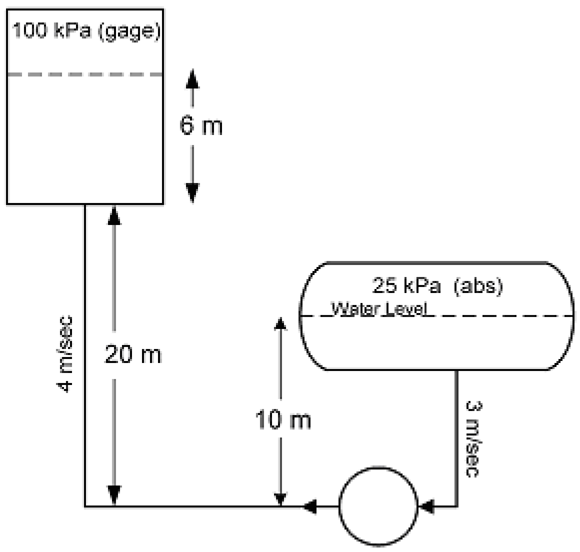
Pressure head on the suction side of the pump is less than atmospheric:

(25 - 101.3) kPa = -76.3 kPa

and 1 kPa = 0.102 m

-76.3 x 0.102 m = <u>-7.783 m</u> = suction pressure head

so, Dynamic suction head = 10 m - 0.5 m - 0.459 m + (- 7.783 m) = <u>1.258 m</u>



Sketch 14.0

Then, Find Dynamic Discharge Head:

Dynamic discharge head = static head + friction head + discharge velocity head + pressure head.

discharge Static head = 20 m + 6m = 26m

discharge Friction head = 1.8 m

discharge velocity head = $\frac{v^2}{2g}$

$= \frac{(4 \text{ m/s})^2}{2 \times 9.81 \text{ m/s}^2}$

= 0.815 m

discharge pressure head = 100 kPa

= 100 x 0.102 m

= 10.2 m

Dynamic discharge head = 26 m + 1.8 m + 0.815 m + 10.2 m = 38.815 m

therefore: Total Head = dynamic disch. head - dynamic suct. head

= 38.815 m - 1.258 m

= **37.557 m (Ans.)**

Self Test Problems

11. What is the dynamic suction head for a pump that has static suction head = 8 m, suction pressure head = 12 m, suction friction head = 2 m, and suction velocity head = 3 m ?

(Ans. = 15 m)

12. A water pump is located 4 m above it's source, which is an open tank. The pump discharges to a level 15 m above the pump. Given the following data, calculate the total dynamic head of the pump:

    suction friction head = 1.5 m
    suction velocity head = 3 m
    discharge pressure head = 16 m
    discharge friction head = 3.7 m
    discharge velocity head = 7 m

(Ans. = 50.2 m)

13. A chemical pump discharges into the side of a process vessel at a point 15 m above the pump. The operating level in the vessel is 12 m above the pump and operating pressure in the vessel is 220 kPa. The discharge velocity of the pump is 4.2 m/sec, friction losses are negligible, and the relative density of the chemical is 1.40. If the total dynamic head is 27.0 m, what is the dynamic suction head of the pump ?

(Ans. = 2.09 m)

Objective Six

When you complete this objective you will be able to…

Explain vapour pressure, cavitation, and net positive suction head. Calculate the required suction pressure for a water pump, given the manufacturers required NPSH.

Learning Material

Vapour Pressure

All liquids form vapours at their free surface, creating a pressure that is known as vapour pressure. This pressure increases in value as the temperature of the liquid increases.

In a closed system completely filled with liquid, no vapours will form as long as the liquid is subjected to a pressure that is greater than the vapour pressure of the liquid, at the existing temperature. However, should the pressure exerted on the liquid be allowed to drop below the vapour pressure, some of the liquid will flash into vapour.

This is an important factor to consider in a pumping system where the pressure on the suction side of the pump could drop below the vapour pressure of the liquid. This drop may be caused by insufficient head, high suction lift, excessive friction head, or high liquid temperature. If the liquid flashes, the formed vapour can partially or completely stop liquid flow into the pump. The pump is then said to be vapour-bound or vapour-locked.

Cavitation

When the pressure, at any point inside a centrifugal pump, drops below the vapour pressure of the liquid, vapour bubbles will form, creating cavities in the liquid flow. These bubbles are carried along with the flow until they reach a region of higher pressure where they collapse, producing a shock wave. This phenomenon is called cavitation.

When the bubbles are carried onto the surface of the impeller and collapse there, the impact of the liquid suddenly filling the void and hitting the metal will damage the surface by gouging out small pieces. When this action is repeated in rapid succession, it produces noisy operation and vibration. Prolonged operation under these conditions may result in mechanical damage to the pump impeller and/or casing.

Cavitation should not be confused with the possible inclusion of air bubbles in the liquid flow. Although air bubbles are not desirable in liquid flow, they do not have the same destructive characteristics as do vapour voids. Air bubbles become smaller when they are compressed, but vapour bubbles totally collapse, producing the shock wave referred to earlier.

Net Positive Suction Head

To prevent cavitation and vapour-binding and to ensure maximum flow through a pump, it is necessary to provide sufficient head on the pump suction so that the suction pressure will always be greater than the vapour pressure of the liquid.

This available pressure, in excess of the vapour pressure, at the pump suction is expressed in meters of liquid head and is called the net positive suction head (N.P.S.H.).
Available N.P.S.H. is the head at the pump suction under operating conditions in the system.
Required N.P.S.H. is the head necessary at the pump suction for the pump to perform properly. Pump manufacturers will specify the required N.P.S.H. for their pumps, as determined by testing the pump under varying conditions of speed, load, etc.

When considering the suitability of a pump to perform the duties required in a system, the 'Available N.P.S.H'. must be calculated and checked to ensure that this is at least equal to (and preferably greater than) the required N.P.S.H. specified by the manufacturer.

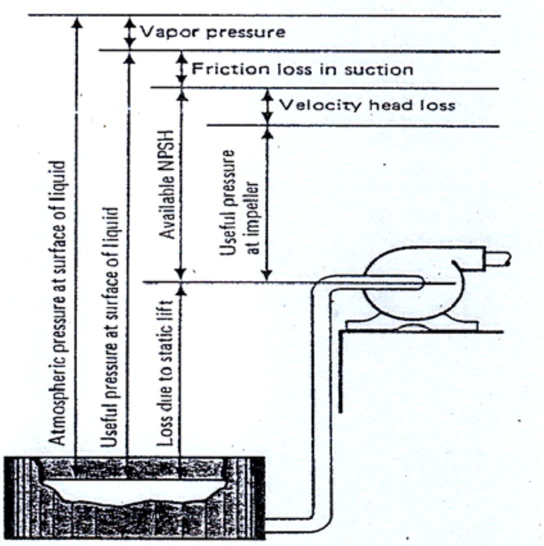


Figure 5 Available NPSH – suction lift condition

Example 15:

A manufacturer specifies a required N.P.S.H. of 5m for a certain pump. To satisfy this requirement, what must the minimum pressure be at the suction of the pump if the liquid being moved is water at 120 °C?

Solution:

From Steam Tables, at 120 °C:

$$\text{Vapour Pressure} = 198.53 \text{ kPa}$$

$$V_f = 1.0603$$

$$\rho = \frac{1000}{v_f} = \frac{1000}{1.0603} = 943 \text{ kg/m}^3$$

$$\text{Pressure due to 5m of N.P.S.H.} = P$$

$$= h \rho g$$

$$= 5 \times 943 \times 9.81$$

$$= 46\,254 \text{ Pa}$$

$$= 46.254 \text{ kPa}$$

$$\therefore \text{Pressure required} = 198.53 + 46.254$$

$$= \mathbf{244.8 \text{ kPa} \text{ (Ans.)}}$$

Self Test Problems

14. A boiler feedwater pump is to supply water from a boiler at 110 °C. According to the manufacturer, this pump must have a minimum suction pressure of 210 kPa to ensure the required NPSH is maintained. What value of NPSH has been specified by the manufacturer ?

(Ans = 7.15 m)

15. A liquid with a vapour pressure of 110 kPa and density of 840 kg/m^3 is moved by a pump that has a NPSH of 6 m. How much pressure must exist at the pump suction ?

(Ans = 159.44 kPa)

Pumping Terms and Definitions

Pumping is the addition of' energy to a liquid to move it from one point to another.

Reciprocating Pumps use pistons, plungers, diaphragms or other devices to displace a given volume of liquid during each stroke of the unit.

Uquid Piston or Plunger of a reciprocating pump is the moving member that contacts the liquid and imparts energy to it.

Simplex reciprocating pumps are those which are equipped with only one liquid piston or plunger.

Duplex or Triplex reciprocating pumps are equipped!, respectively, with two or three liquid. Pistons or plungers.

Single-Acting reciprocating pumps produce only one suction and one discharge stroke per cycle.

Double-Acting reciprocating pumps produce two suction and two discharge strokes per cycle.

Surge Chambers are containers built into a reciprocating pump or attached to its adjacent piping to cushion the shock of the reciprocating action and therefore smoothing the liquid flow.

Centrifugal Pumps employ centrifugal force to develop a pressure rise for moving a fluid.

Impeller is the rotating element in a centrifugal pump through which liquid passes and by means of which energy is imparted to the liquid.

Casing of a centrifugal pump is the housing surrounding the impeller. It contains the bearings for supporting the shaft on which the impeller mounts.

Single-Stage centrifugal pump is one in which total head is developed by one impeller.

Multistage centrifugal pump is one having two or more impellers acting in series in one casing.

Critcal Speed of a centrifugal pump is that speed at which the rotating shaft corresponds to its natural frequency. At this speed any minor unbalance of the shaft is magnified, and excessive vibration will occur.

Rotary Pumps use gears, vanes, pistons, screws, cams, etc., in a fixed casing to produce positive displacement of a liquid.

Packing is any material used to control leakage between a moving and stationary part in the pump.

Mechanical Seals are devices mounted on the shaft of centrifugal pumps to seal the liquid in the casing. These are frequently used in preference to packing because of their longer life and minimized leakage.

Cavitation is the phenomenon caused by vaporization of a liquid inside a pump. When the pressure at any point drops below the vapor pressure corresponding to the temperature of the liquid being pumped, vaporization of the liquid will occur. Small cavities of vapor thus formed move with the flow through the primp until a region of higher pressure is reached. The higher pressure causes the vapor cavities to collapse with tremendous shock on the surrounding metal.

Viscosity is that property of a liquid that resists any force tending to produce flow.

Specific Gravity of a liquid is that number which denotes the ratio of the weight of the liquid to the weight of an equal volume of water.

Suction Lift Exists when the source of supply is below the center line of the pump.

Static Suction UFT is the vertical distance in feet from the center line of the pump to the free level of the liquid to be pumped.

Total Dynamic Suction UFT is the vertical distance in feet from the center line of the pump to the free level of the liquid to be pumped plus all friction losses in the suction pipe and fittings.

Suction Head exists when the source of supply is above the center line of the pump.

Static Suction Head is the vertical distance in feet from the center line of the pump to the free level of the liquid to be pumped.

TOTAL DYNAMIC SUCTION HEAD is the vertical distance in feet from the center line of the pump to the free level of the liquid to be. Pumped minus all friction losses in suction pipe and fittings.

NET POSITIVE SUCTION HEAD *required* is the energy needed on the suction side of a pump to fill the pump to the discharge valves during operation.

Assignment

1. (a) Explain how liquid flow is created and maintained into the suction of an operating pump.
   (b) Suggest at least three ways in which the flow into a pump suction can be assisted.

2. (a) Why do equal columns of different liquids exert different pressures at their bases?
   (b) Calculate the pressure exerted by a 15 m column of a liquid with a relative density of3.7.

3. Assuming friction and velocity heads are the same for both pumps, which of the following pumps must operate against the greatest total head and how much greater is this head than the other pump?

   Pump A -liquid = water at 4°C; pump is 2 m above suction surface; suction tank is open to atmosphere; discharge vessel has a 180 kPa. gas blanket and level 5 m above the pump.

   Pump B -liquid has relative density of 2.6; suction is from a vessel with 250 kPa pressure and liquid level 3 m above pump; discharge tank is open and located 18 m above the tank with a level of 4.5 m.

4. (a) Explain how cavitation can occur in a pump and describe its potential effects.
   (b) Define net positive suction head and explain how it prevents cavitation.
   (c) If a liquid at operating conditions has a vapour pressure of 320 kPa and a density of $1078\ kgm/^3$, what must the minimum operating pressure be at the suction of the pump if the manufacturer requires an available NPSH of 6 m?

Seal & Oil Systems

Table of Contents :

Objective One:

TECHNICAL ASPECTS OF MECHANICAL SEALS

1.1 Principles of Operation

Flexibox mechanical seals are the answer to the problem of preventing leakage between rotating shafts and their housings.

The old method of using soft packing has many disadvantages, for example, the packing must be regularly inspected and tightened down to take up wear, and it must also be replaced frequently. In time, the pump shaft or sleeve becomes worn, necessitating expensive renovation. Since the packed gland relies for its successful operation on lubrication of the packing material by the fluid being pumped, it can never entirely stop leakage, unless a costly liquid sealing system is introduced.

Flexibox face-type seals overcome these disadvantages. There is a wide range of seals available, differing in design and application, but all applying certain basic design considerations.

1.1.1 How Do They Work?

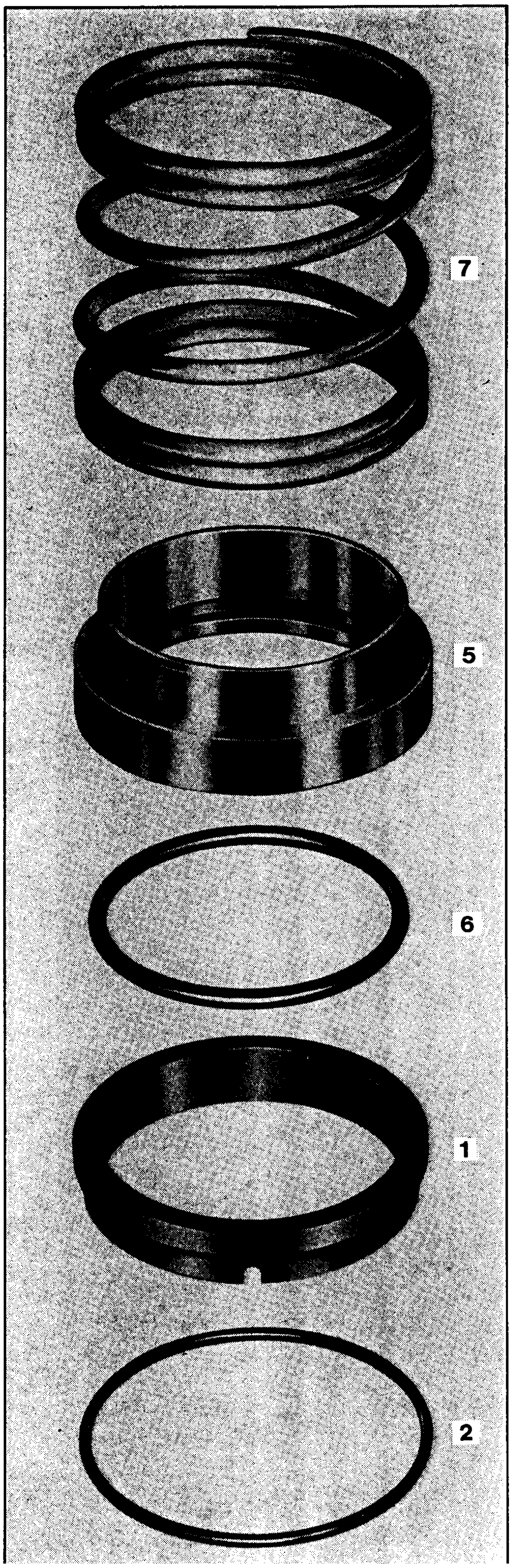
To illustrate the principle of operation of the mechanical seal, the simplest type, five components, are illustrated .The five basic parts are:

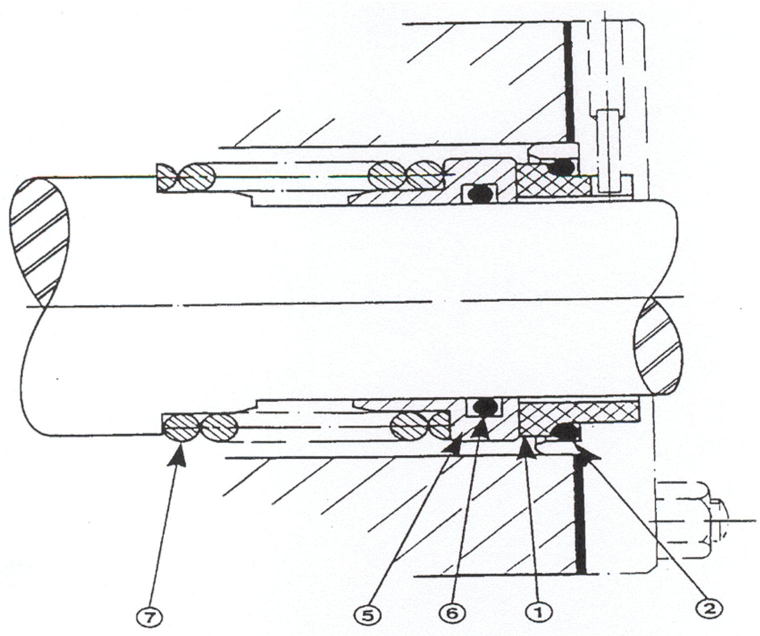
- 1 Stationary seal ring
- 2 Stationary seal ring packing
- 5 Rotary seal ring
- 6 Rotary seal ring packing
- 7 Spring

Sealing action is obtained by intimate contact between the opposing faces of two rings, one of which is held resiliently (stationary seal ring) in the gland housing while the other (rotary seal ring) rotates with the shaft As the rubbing surfaces are extremely flat- they are lapped within two light bands-leakage of the fluid being handled is prevented.

Rotary Seal Ring

The rotary seal ring rotates with, and is driven by, the shaft In Flexibox mechanical seals, the rotary seal ring is normally the hard face. A flexible 'O' ring (rotary seal ring packing) effectively prevents leakage between the rotary seal ring and the shaft while allowing the ring sufficient freedom of movement to maintain full face contact with the stationary seal ring. The shaft must be smooth and free from flaws.


7
5
6
1
2



Stationary Seal Ring

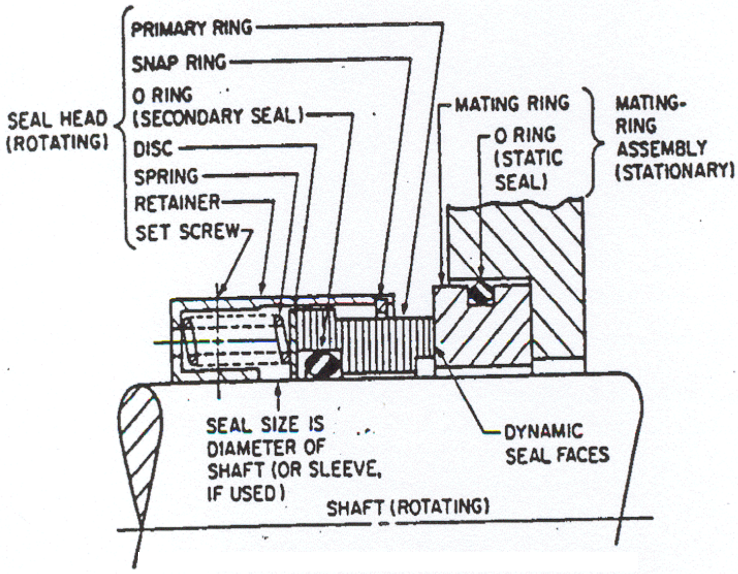
The stationary seal ring, usually made from a specially compounded grade of carbon, is resiliently mounted on a second flexible 'O' ring (stationary seal ring packing). This prevents leakage between the stationary seal ring and the gland housing and, at the same time, provides a cushion for the former, to allow it to take up a fair degree of 'out-of-squareness.

The stationary seal ring is provided with a slot to enable the ring to be positively located in the seal housing, thus preventing its rotation.

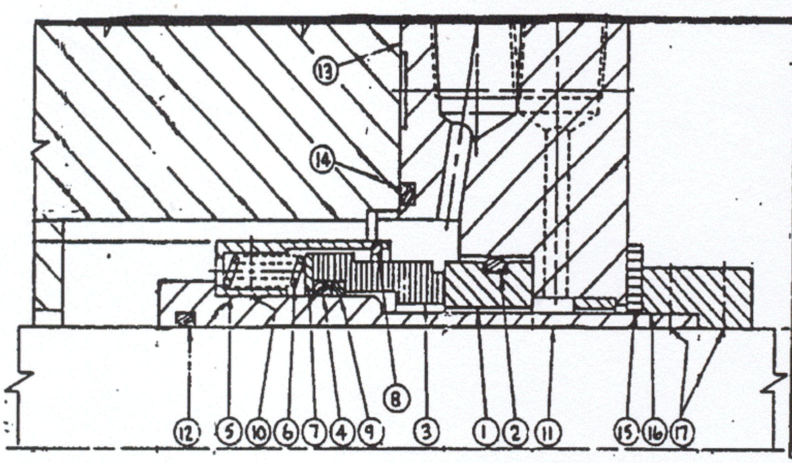
Spring Drive

Initial seal face contact is maintained by a fifth component the spring. Interference fit between the spring and the shaft at one end and the neck of the rotary seal ring at the other provides a positive resilient drive. The spring is available with either a right hand or left hand coil.
The handing of the coil is arranged so that in operation, the spring will tend to increase its grip. Spring drive also helps to make the fitting of a Flexibox seal a comparatively simple matter, since access to the back of the seal, to locate and fix it is unnecessary. In addition, the use of heavy gauge wire, normally stainless steel, eliminates what can be serious operational drawback when a number of smaller springs of lighter gauge are employed.

Single Mechanical Construction



Mechanical Seals



ITEM	DESCRIPTION	MATERIAL	QTY
1	MATING RING	TUNG. CARB.	1
2	O-RING	VITON OR EQ.	1
3	PRIMARY RING	CARBON	1
4	O-RING	VITON OR EQ.	1
5	RETAINER	316 S.S.	1
6	SPRING	316 S.S.	
7	DISC	316 S.S.	1
8	SNAP RING	316 S.S.	1
9	ANTI-X-RING	CHEMLON	1
10	SET SCREW	316 S.S	
11	SLEEVE	316 S.S.	1
12	O-RING	VITON OR EQ	
13	GL. PLT. ASS'Y	316 S.S. /BRONZE	1
14	O-RING	VITON OR EQ.	
15	SPACER	555	3
16	COLLAR	316 S.S.	1
17	SET SCREW	316 S.S.	8

1.1.2 Hydraulic Balance

The term hydraulic balance, when applied to mechanical seals, is used to denote the relationship between the pressure being sealed and the seal face contact pressure. Seals are classified as balanced or unbalanced but within each classification there is a variation of degree of balance. In both types of seal, the initial face load is provided by the seal spring. The fluid pressure then acts on radially disposed areas in such a way as to increase face pressure in proportion to fluid pressure. In the case of an unbalanced seal, the face contact pressure may be a hundred per cent or more of hydraulic pressure. In the case of balanced seal, close control of the radially disposed areas enables the face contact pressure to be maintained at a value less than pressure being sealed. This lowering of seal face pressure allows a thicker fluid film between the faces, thus reducing the friction and consequent heat generation. Because of this, a balanced seal is able to handle more arduous duties and higher fluid pressures than an unbalanced seal. Normally, a balanced seal is designed to operate with the lowest face pressures that will effectively prevent leakage between the faces.

Variations of seal balance are illustrated in Fig. 1.1.2. Here S, the diameter of the seal sleeve shoulder, represents the effective sealing diameter, which may also be referred to as the hydraulic or sliding diameter since all thrust loads would be computed about this surface 'A' represents the hydraulic piston area of the sliding seal ring and 'B' represents the contact face of the sealing faces.

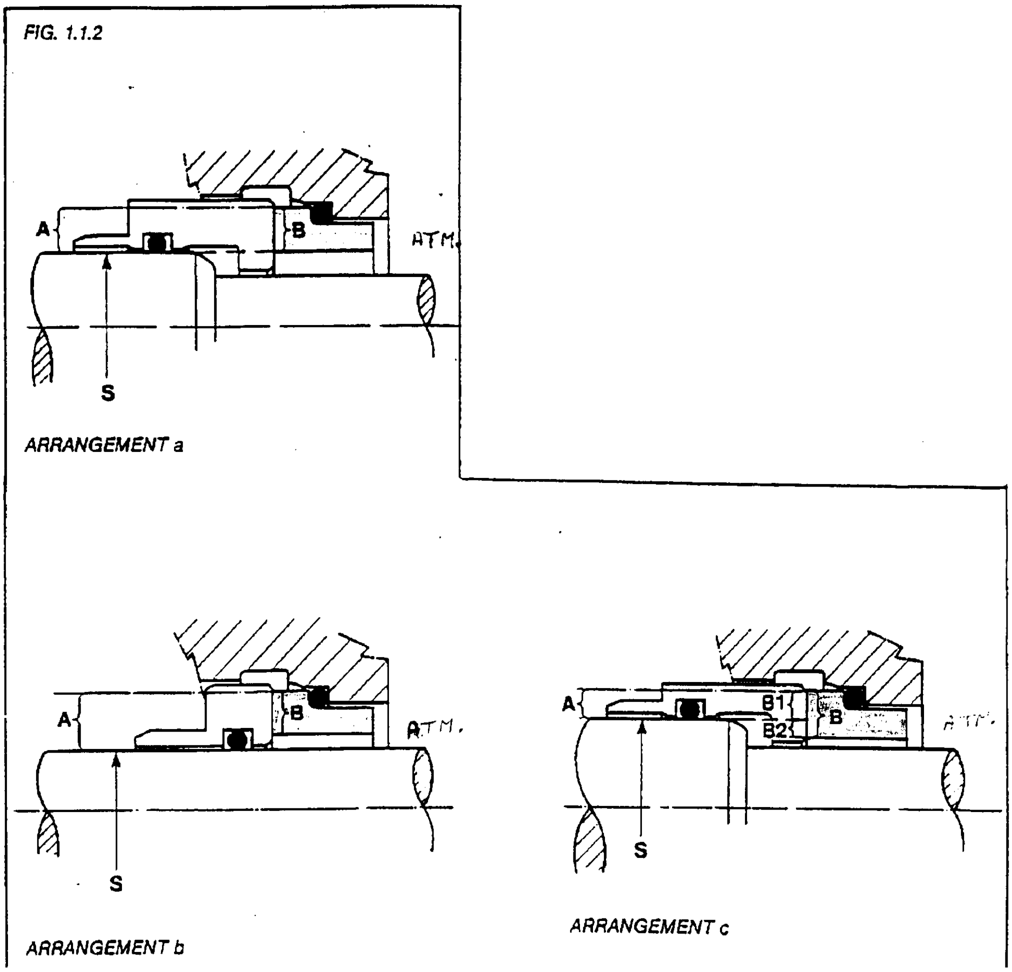
- In arrangement a, all the contact face area B is disposed outside the effective sealing diameter S and the hydraulic piston area A is equal to the contact face area B. This, therefore, represents a condition of one hundred per cent balance, which also indicates that the average face contact pressure will be exactly one hundred per cent of the hydraulic pressure sealed.

- In arrangement b, all contact face area B is disposed outside the effective sealing diameter S and the hydraulic piston area A is greater than the contact face area B. In this instance, the seal is more that one hundred per cent balanced, in accordance with the ratio of area A to B. The unit contact face loading will be higher, by an equivalent percentage, than the hydraulic pressure being sealed. This is a condition existing in most unbalanced seals.

- Arrangement c, shows the relationship of most balanced seals, here part of the contact face area B, designated as B1, is disposed outside the effective sealing diameter S. Area B1 is, therefore, equal to the hydraulic piston area A Since the rest of the face area B2 is located inside the effective sealing diameter S, the total seal face area B will equal the sum of B1 + B2 and the unit face loading will be less than the pressure sealed in accordance with the ratio of the two areas, ie

$$\frac{B1}{B1 + B2} \qquad \frac{A}{B}$$

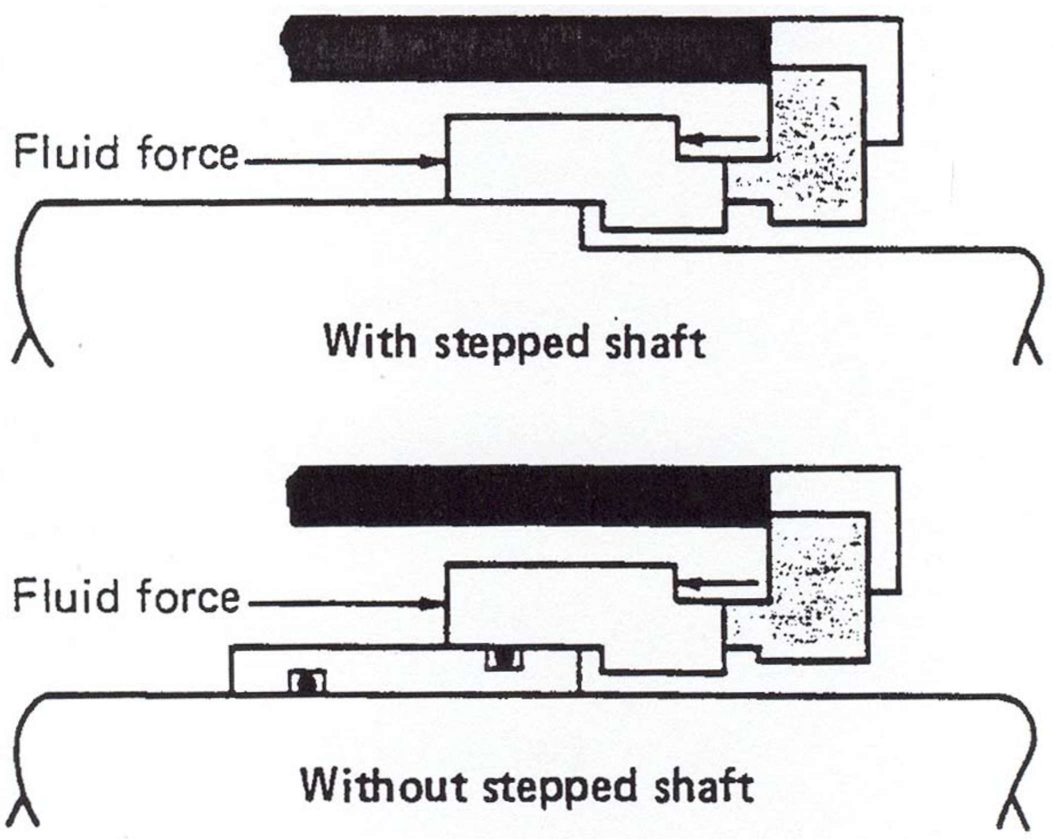
This value, expressed as a percentage, indicates the degree of balance of the seal.



Step design balanced seals, shown in figure below, where the rotating member has a step design on its front edge. This step allows a certain amount of pumped fluid to counter act the force on the back of the seal element.

This force relieves some of the pressure on the seal, thus producing lighter contact pressure between the rotating and stationary elements. This contact pressure increase seal life and ensure adequate lubrication on the contact surface, by increasing oil film thickness.

Balanced Seals



1.2 Importance Of Fluid Film Stability

It is known that for the seal to work efficiently, it is necessary for a stable fluid film to exist between the faces. In the majority of cases this film is a liquid, although in certain special applications a gas film may be induced between the faces. If this film stability is destroyed, excessive wear takes place leading to rapid seal failure.

Vaporisation Due to Excessive Heat or Dry Running:

The most common cause of premature seal failure arises from the loss of the liquid film between the seal faces. Just before this occurs, the seal emits a puff of vapour every few seconds and there is local boiling off of the liquid film. This causes the seal to open and tilt temporarily. This allows more liquid to enter between the seal faces, giving a temporary cooling effect When the frictional heat generated at the seal faces has built up sufficiently to vaporise more liquid, another puff of vapour is emitted and the cycle is repeated. At this stage, the only damage suffered by the seal is the chipping of the edges of the carbon seal face, caused by the tilting of the rotary seal ring.

This is the prelude to complete loss of the liquid component between the seal faces and dry running with resultant heavy face wear occurring. Carbon and metal seal ring surfaces are then heavily grooved, somewhat like a gramophone record.

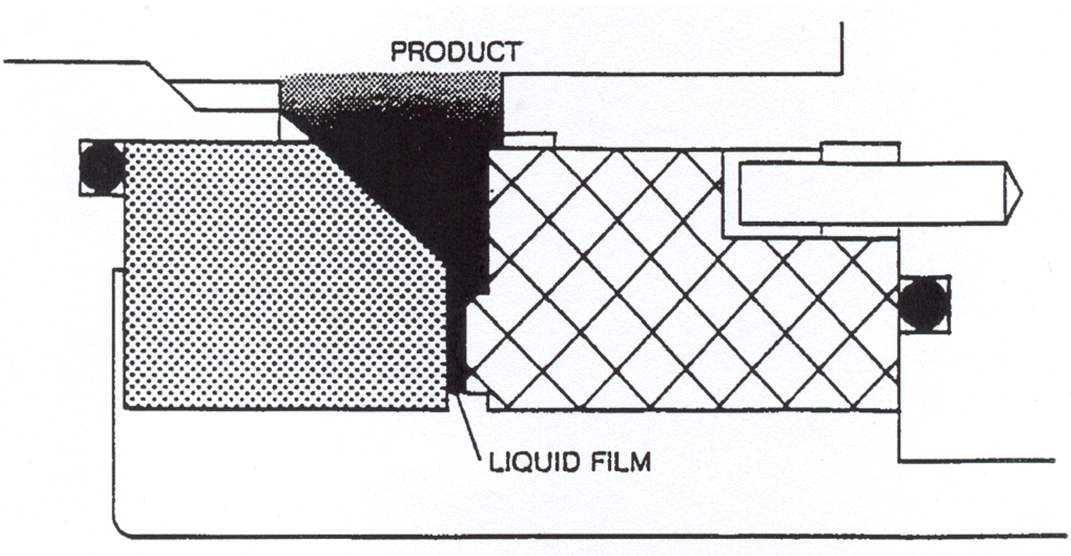


Figure 1.2 Seal Face Liquid film (Exaggerated)

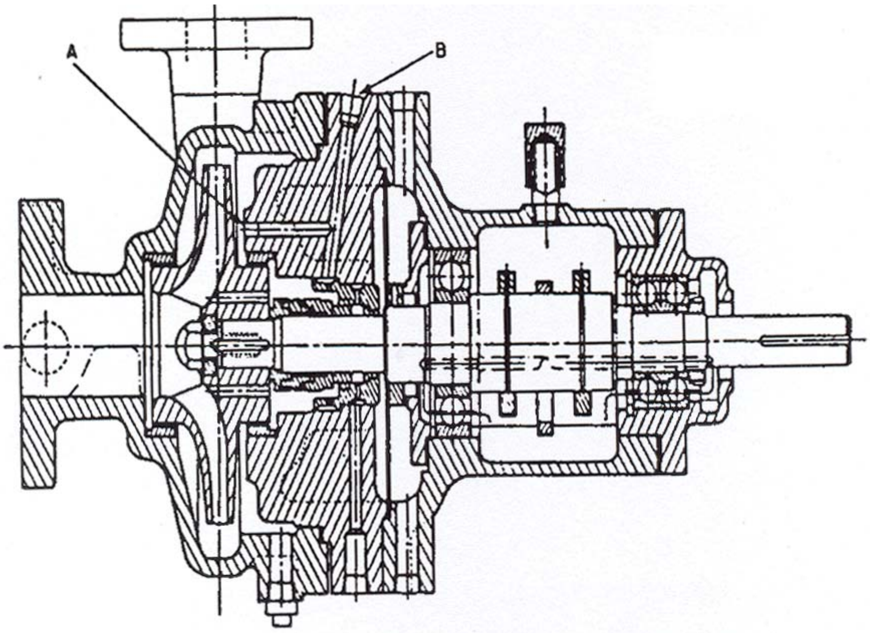
1.3 Circulation

As already discussed, the use of mechanical seals inevitably involves the removal of generated heat from the immediate seal area. The most practical and efficient method of achieving this is to arrange for a flow, of the product being pumped, to pass over the seal. This transfer the local heat generated to the main flow in the equipment. It is this arrangement of flow which is covered by the term circulation.

The main purpose of circulation is the removal of heat from the seal area. Other benefits include the prevention of sediment deposits; control of the flow temperature by means of coolers or heaters and the facility of cleaning the flow by the use of separators or strainers. By full use of these facilities the conditions at the seal can be modified to provide longer seal life.

1.3.1 Circulation in Single Seals

The main requirements of heat removal by the circulation flow can only be achieved efficiently if the flow passes around or over the seal faces. With most of the standard range of seals the circulation flow is directed onto the seal faces, efficiently removing the heat from its source. With seals in the high duty range, the circulation flow can be passed through the spring pockets, in order to prevent sediment deposition, and then over the seal faces.



Cooling Circulation to Mechanical Seal: (A) Internal Circulation Plug Port, (B) External Circulation Plug Port

1.3.2 Circulation Systems for Double And Tandem Seals

A double or tandem seal circulation system should possess each of the following characteristics:

- A means of circulating the sealant
- A means of recharging the system to make up leakage.
- A means of cooling to remove the heat generated by the seals.
- Systems for double seals, and occasionally tandem seals, also need a means of pressurising the system.

1.3.2.1 Circulation of Sealant

A By using a pump working against a relief valve, the circulation generally being returned to an open tank on the suction side of the pump.

B. Thermo-syphon Circulation, which, by means of height and differential temperature in the two vertical legs, causes a circulating flow between the seal chamber and a header vessel.

C. By using a small pump in a pressurised circulation system.

D. By using an impeller or pumping ring, fixed to the rotating sealing ring or pump shaft inside the seal housing. In this case, the impeller is generally made integral with the seal and is known as a W feature.

1.3.2.2 Charging Means

A charging means is necessary in order to make up for any barrier liquid loss, to change contaminated sealants, or to ensure completely automatic operation.

1.3.2.3 Cooling Means

There are four principal ways of cooling:

A By using a water cooler in the system, passing the sealant through the cooler coil.

B. By using an air cooler.

C. By cooling a closed header tank by means of an integral cooling coil.

1.3.2.4 Pressurising the Seal Unit

There are four ways of pressurising.

A. By vapour from the sealed product which means vaporising the product by means of ambient temperature.
B. By pressurising a small header vessel by a neutral gas from an outside source.
C. By using a pressure controller, which will pressurise the sealant by using product pressure.
D. By using a pump both to pressurise and circulate the sealant.

1.3.2.5 Barrier Liquid Conditions

There are two basic considerations affecting the choice of barrier liquid.

1. The barrier liquid, or sealant between the seals must be a stable liquid and should preferably have lubricating properties. The following liquids are preferred as sealants.

Temperature Range	Sealant
-120 deg C to;-90 deg C	Propanol.
- 90 deg C to -30 deg C	Methanol or Propanol.
-30 deg C to 20 deg C	Kerosene or Hydraulic Oils
20 deg C to 200 deg C	Light Oil or Gas Oil.

2. In either the double or tandem seal arrangement there will be mixing of the product and barrier liquid; these must, therefore, be compatible.

- For double seal systems, the barrier liquid pressure should always be above the liquid being sealed (ie product pressure at the inner seal), thus ensuring that a film of the sealant, essential for the correct operation of the seal, is introduced at the inner seal faces. This film replaces the major part of the product film which would normally be present. The pressure to produce, this should be at least 2 bar greater than the sealed pressure.

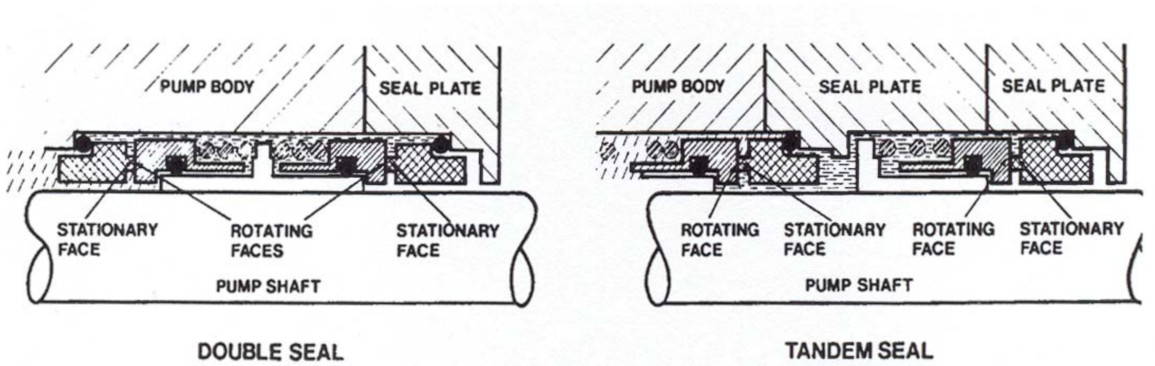
- For tandem seals, the barrier liquid pressure should be lower than the product at the inner seal. This ensures that there will be no barrier liquid contamination of the product

1.3.2.6 Testing Double and Tandem Seals

It is essential that double seals are assembled correctly and that they are leak free from startup; otherwise a pressure drop and loss of barrier fluid could be the result Because the design of a double seal is contained in a housing or stuffing box, it should be possible to put a static pressure on the seal before proceeding with the complete assembly of the seal into the equipment

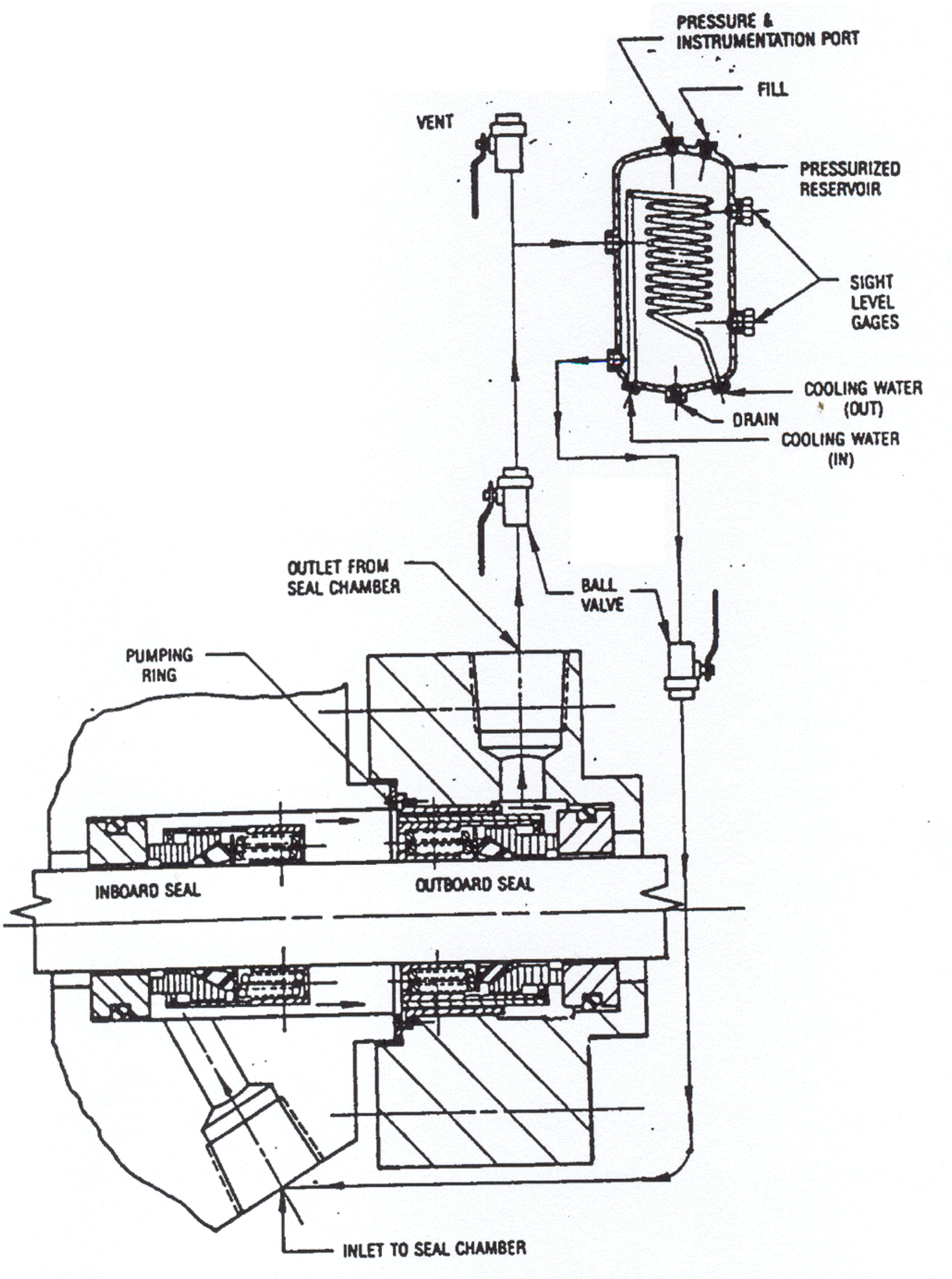
The pressure between the seals should hold for at least 30 minutes; any pressure drop over this period suggests a possible leak path from one or both of the seals and should be investigated.

Because the inner seal operates against the pumped liquid, it is not usually practical to test tandem seals until the pump is pressurised, when the pump is installed. Tandem seals should be tested by pressurising both the pump and the seal chamber after fitting the seal. The pressure should be at least that of the barrier fluid pressure.

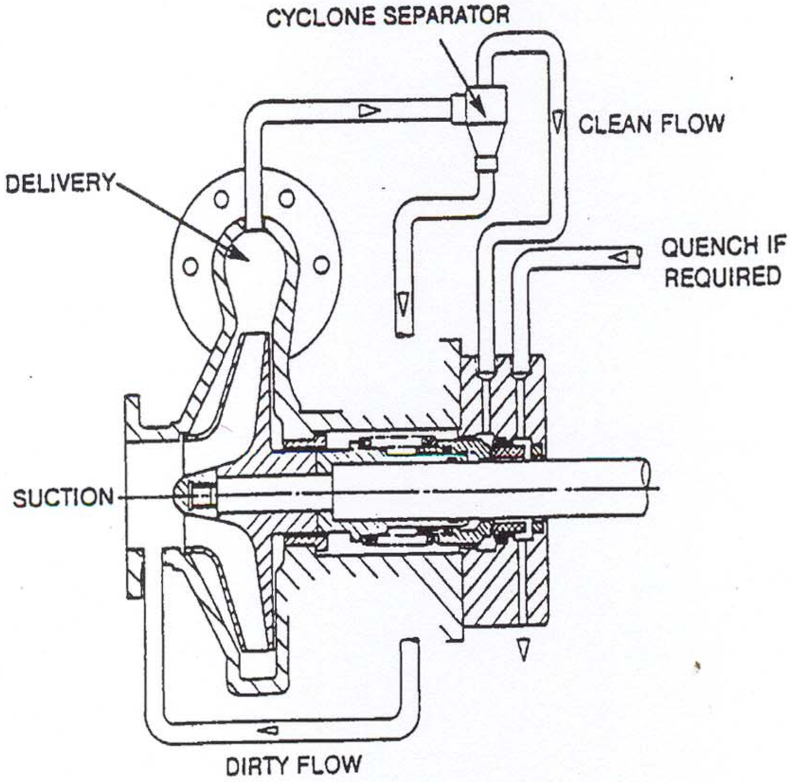


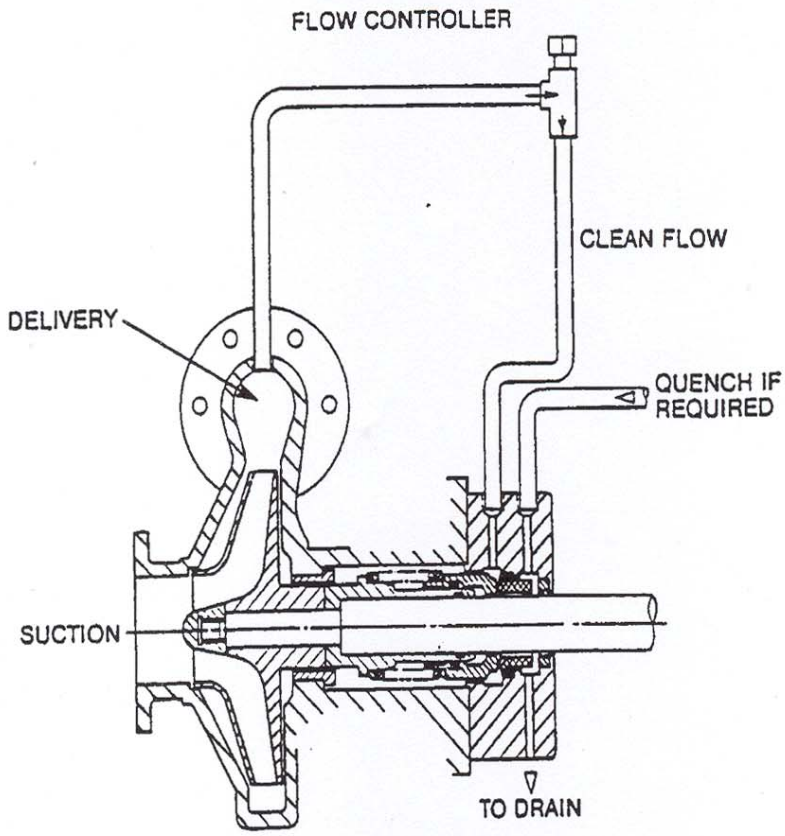
PUMP FLOW (PRODUCT

BARRIER FLUID



Double Seal Installation with axial- flow pumping ring (Johan Crane- Houdialle, inc.)


CYCLONE SEPARATOR
CLEAN FLOW
DELIVERY
QUENCH IF REQUIRED
SUCTION
DIRTY FLOW


FLOW CONTROLLER
CLEAN FLOW
DELIVERY
QUENCH IF REQUIRED
SUCTION
TO DRAIN

Optional features

1.4.1 'B' Feature (Safety Bush)

Balanced or unbalanced seals can be fitted with a throttle bush to conform to the requirements of specification API 610. This gives extra protection in the event of a sudden seal failure and takes the form of a close clearance safety bush. The bush is pressed into the seal plate from the inside to prevent it being blown out and functions by considerably reducing the pressure of the escaping product and directing the leakage to drain. In the event of a pump bearing failure, whereupon the shaft could run eccentric and to eliminate any possibility of sparking, the bush is manufactured from a nonferrous material and connections are provided for quenching and draining the space contained by the bush.

In the event of the bush having to be replaced, the diametrical clearance between the bush and shaft or sleeve should be 0.025 inch/0.63mm as specified by API 610.

1.4.2 'P' Feature (Auxiliary Sealing Device)

To prevent or control the leakage of the quenching medium from between the outer end of the seal plate and the pump shaft, it is necessary to incorporate an additional 'P' feature in the form of a special lip seal or a labyrinth bush, or even an auxiliary packed gland. Controlling the quench medium leakage could be for safety reasons, or to aid the efficiency of the seal and to prevent the leakage from entering the pump bearing housing.

1.4.3. ‘Q’ Feature (Quench)

The quench feature is standard on all balanced seals and may be specified on unbalanced seals. The feature provides for the admission of a quenching medium e.g. water, LP steam or methanol, through a threaded connection to the outer or atmospheric side of the seal, flowing freely around the annulus to drain.

The feature is mainly used to improve the function of the seal by preventing a product build up on the atmospheric side of the rotary or sliding packing which causes the seal to hang up or stick on the sleeve, thus preventing face contact. There could be occasions where the quench could be used to neutralise seal leakage for safety reasons or to satisfy environmental controls.
To improve the efficiency of the quench feature and to reduce or prevent the quench medium from leaking between the safety bush and sleeve, lip seals or auxiliary packings can be incorporated in the design. This also prevents the medium from entering the pump bearing housing.

NOTE: The pressure of the quench medium entering the seal plate, should not exceed 1 bar (15 psi) and should enter the plate from above the centre line and be drained away from the bottom to prevent a build up of liquid; also it is essential that the drain hole is clear to prevent a blockage.

1.4.4'C' Feature (Circulation)

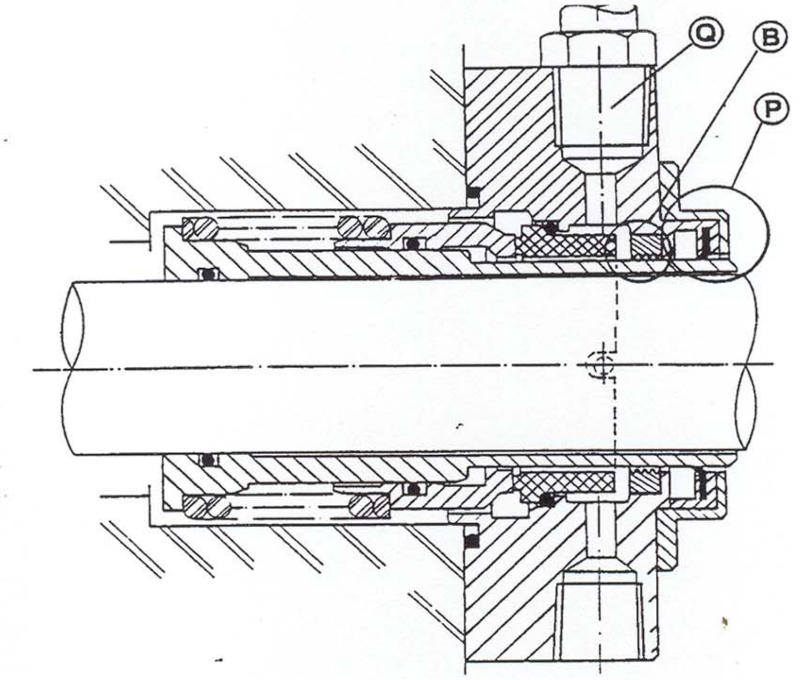
This is a feature on standard Flexibox balanced seals and of unbalanced seals which are supplied with a seal plate. They are provided with threaded connections into which a calculated or recommended flow of the product is directed from a point of higher pressure, around the stationary and rotary seal rings, back to a point of lower pressure in order (a) to remove generated frictional heat; (b) to prevent the settling of sediments, abrasives or polymers; (c) to keep seals handling hot products cooler in the line, if necessary.
This method of providing circulation is generally applicable to single stage pumps. In the case of multistage pumps, the circulation would come from first or second stage because of the high differential across a group of stages.
A close clearance neck bush may be used in a general purpose process pump to restrict the circulant, and build up of a pressure in the stuffing box on light hydrocarbon duties
Where the application demands an external injection of less aggressive product, for example on hot, abrasive or corrosive duties, the neck bush can isolate the seal faces and products from the pumped fluid. The low volume, higher pressure, injection flow under the neck bush, from the stuffing box to the pump impeller, prevents this from happening.

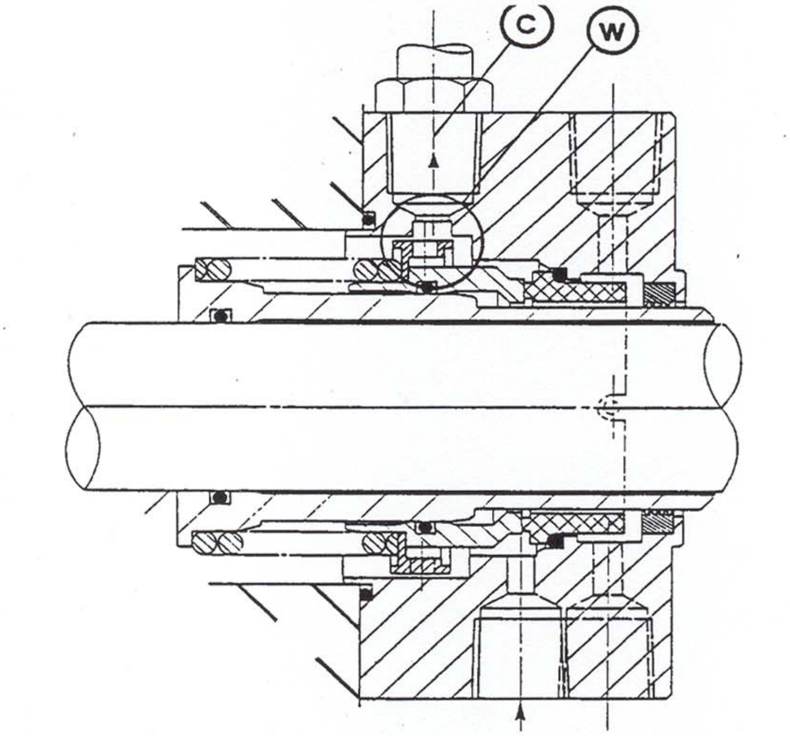
1.4.5 ‘W’ Feature (Impeller)

An impeller is added to a seal so that it can generate its own circulation flow, providing a means of heat removal which is more thermally efficient than normal circulation cooling. Another case where the 'W' feature can be used is where a seal is operating at pump discharge pressure, and the seal requires cooling below the product temperature. Further applications are to generate circulation in a double or tandem seal system or on a single seal where there is little or no pressure differential across the seal.
The Flexibox 'W' feature essentially consists of an open impeller mounted on one of the rotating parts of the seal, generally the rotary seal ring.

1.4.6 'M' Feature (Multi-Point Injection)

The circulation system for seals on liquified gases is of paramount importance. The usual injection of product at a single point will sometimes cause localised hot spots resulting in unequal thermal expansion, or contraction of the seal rings and deformation of their faces. With light hydrocarbons, vapour bubbles may form an insulating blanket on the seal rings, greatly reducing the heat transfer to the circulation liquid. The Flexibox multipoint injection feature supplies the circulation evenly around the seal faces to provide symmetrical cooling and break up the bubble blanket, thereby enhancing heat dissipation.


Q
B
P


C
W

1.4.7 Flushing

The flushing of the mechanical seal is the fluid flow from the stuffing box through the neck bush to the pump impeller.

The flushing is used when the pumped fluid is abrasive or corrosive product, to prevent the settling of sediments, abrasive or polymers on the seal faces.

The flushing fluid should be of low volume and high pressure.

Procedure For Flushing.

1. Bring clean liquid (not steam or gas), from an outside source (not the pump discharge) into the stuffing box at a rate of 1-2 gallons (4-8 liters) per hour.

2. The liquid is brought in 15 Ibs. (1Bar) higher than stuffing box pressure.

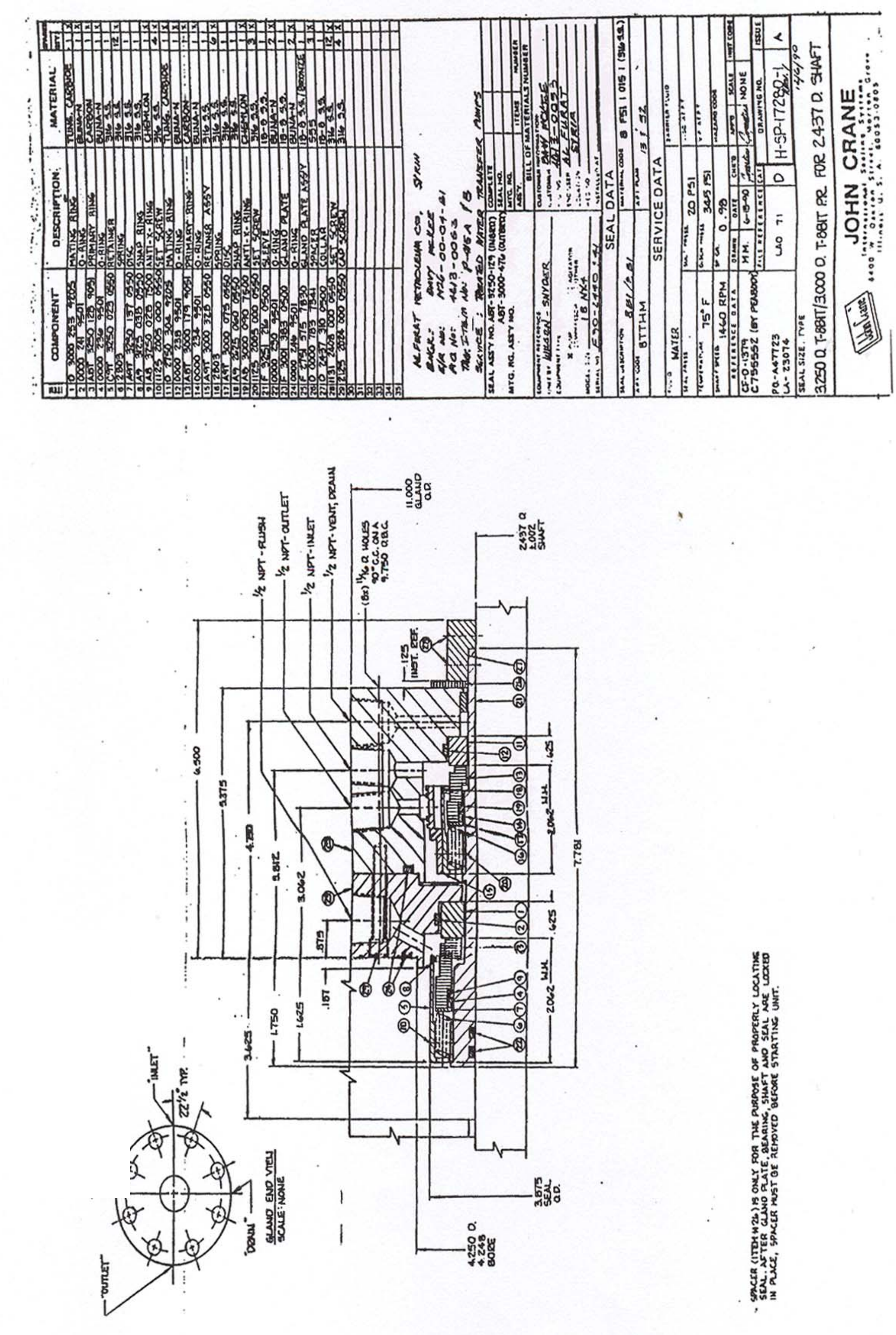
3. Regulate the flow with a meter.

4. A restrictive bushing (usually carbon) is placed into the bottom of the stuffing box to restrict the flow and keep the liquid clean in the stuffing box.
   Bushing length -At least 3/8: (10 mm)
   Bushing Clearance-Close as possible - .008" (0, 25 mm) on the diameter

This bushing must be chemically compatible with both the product and flushing fluid. (Refer to Feature" circulation")

5. The flushing fluid can be any clean compatible liquid:

   a. Finishing product.
   b. Solvent.
   c. An additive normally added downstream.
   d. Water.
   e. Compatible liquids.


JOHN CRANE
SEAL DATA
SERVICE DATA
COMPONENT
DESCRIPTION
MATERIAL
BILL OF MATERIALS NUMBER
GLAND END VIEW
SCALE: NONE
"INLET"
"OUTLET"
"DRAIN"
22½° TYP.
½ NPT-FLUSH
½ NPT-OUTLET
½ NPT-INLET
½ NPT-VENT, DRAIN
11.000 GLAND O.D.
2.437 Ø 1.002 SHAFT
3.875 SEAL O.D.
4.250 Ø 4.248 BORE
H-SP-17280-1
3250 Ø T-8BIT/3000 Ø T-8BIT P.R. FOR 2.437 Ø SHAFT
SPACER (ITEM #26) IS ONLY FOR THE PURPOSE OF PROPERLY LOCATING SEAL. AFTER GLAND PLATE, BEARING, SHAFT AND SEAL ARE LOCKED IN PLACE, SPACER MUST BE REMOVED BEFORE STARTING UNIT.

1.5 Use of ancillaries

Many ancillary devices are used in seal circulation systems to modify the product conditions and to give the best seal performance. Ail ancillaries will have an influence on the flow to the seal, and this must be given due consideration to ensure adequate circulation rates are maintained.

AIR OR WATER COOLERS

Coolers are used in circulation lines to modify the temperature of the product and ensure correct seal performance. Careful selection of a cooler will often eliminate the need for a more sophisticated shaft sealing arrangement.

FLOW CONTROLLERS

A flow controller is an adjustable orifice and is used to regulate the product circulation to the seal. Flow controllers must be designed so that circulation cannot be completely cut off.

STRAINERS

Strainers are generally fitted in new lines to protect the seal from pipescale, dirt or other solids that may be initially present. They are usually removed, or the element withdrawn, when the plant is fully operational and considered clean.

CYCLONE SEPARATORS

Cyclone separators are fitted as permanent devices in circulation circuits to remove solids continuously. The product entres the separator, where centrifugal force extracts the denser solids and flushes these away through the dirty outlet. The dirty outlet is usually piped to pump suction. The product carried over through the clean outlet is supplied to the seal, through the circulation connection. As this is only a portion of the original flow to the separator, this must be taken into account when considering the circulation requirements of the seal.

MAGNETIC FILTERS

Magnetic filters are used to remove ferritic solids continuously from circulation lines. They are used principally on boiler feed applications where the problem is due to abrasive black iron oxide particles.

Objective Two

FLEXIBOX SEAL TYPES

2.1 Single Seals –Unbalanced

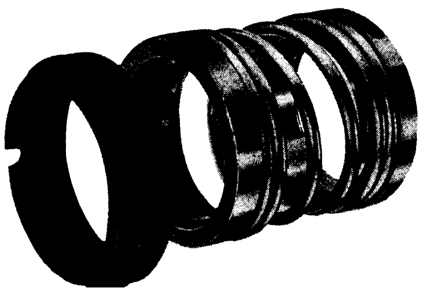
2.1.1 Single Spring Unbalanced Seals

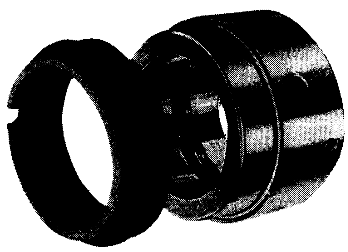
The unique single spring feature gives a flexible drive whilst maintaining intimate contact of the two seal faces. The drive is achieved by shaft rotation which makes the spring grip both the shaft and rotary seal ring, thereby eliminating the need for extra keys or pins. In the reverse direction, the spring releases its grip on the components, easing assembly and dismantling. Secondary seals are generally 'O' rings selected for product compatibility and they provide resilient mountings for the seal rings.

2.1.2. Multispring Unbalanced Seals

The multi- spring unbalanced seals are designed for applications where seal length is critical and for dual rotation.

The multi- spring seal design include a set screwed spring sleeve.



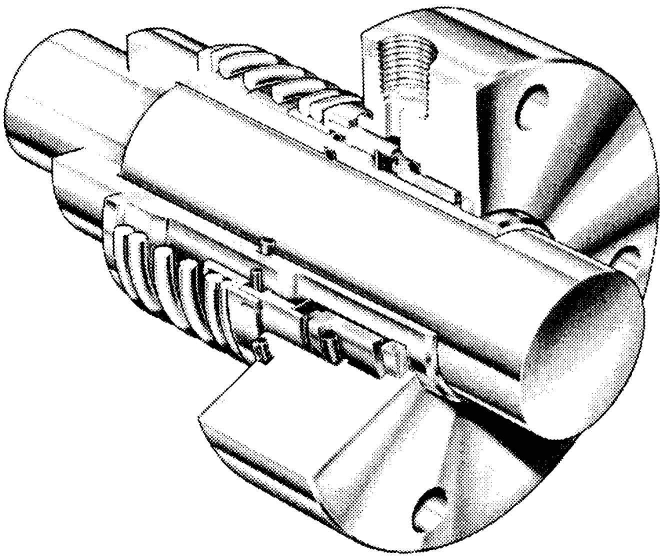


A Single Spring Unbalanced Seal *Multi- Spring Unbalanced Seals*

2.2 Single Seals-Balanced

2.2.1 Single Spring Balanced Seals

In this design, a single robust spring provides a flexible drive from the shaft sleeve to the rotary seal ring, whilst maintaining contact of the two seal faces. Shaft rotation makes the spring grip the shaft and the rotary seal ring, eliminating the need for extra keys or pins. In the reverse direction the spring releases its grip on the components, easing assembly and dismantling. Secondary seals are generally '0' rings, selected for product compatibility and they provide resilient mountings for the seal rings.



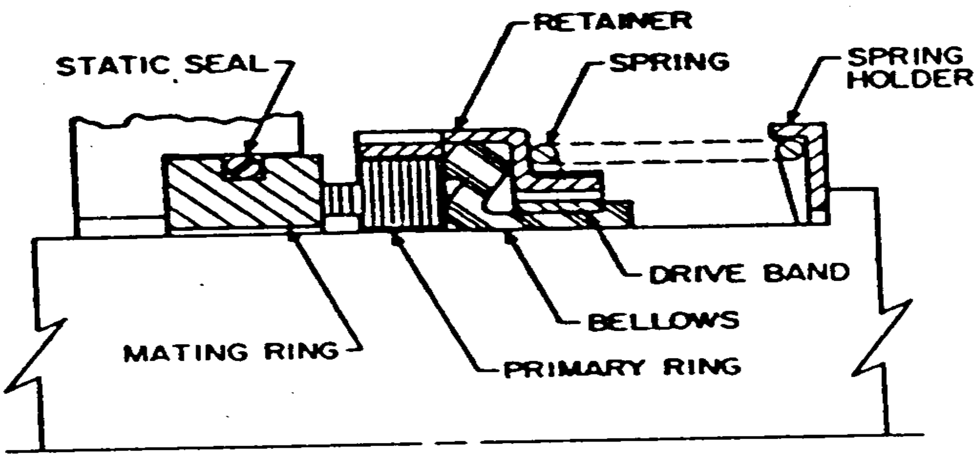
2.2.2 Multi-Spring Balanced Seals

A multi-spring version of the balanced seal range, which includes a set screwed spring sleeve. The multispring design makes the seal bi-directionai for applications where the reverse rotation is anticipated.

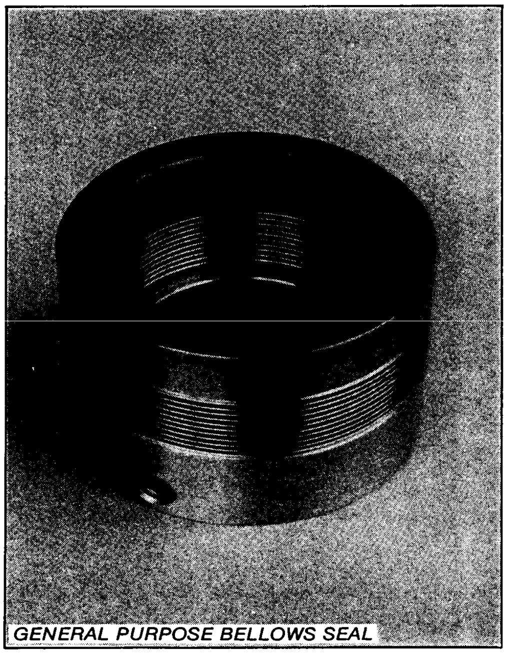
2.3 Bellow Seals

General Purpose Bellows Seal

The general purpose metal bellows seal is designed with a very spring rate giving very low specific face and excellent misalignment capabilities. The seal is available to offer a wide range of chemical and mechanical operating capability.



Bellow Seal



2.4 Multiple Seals

Double Seals

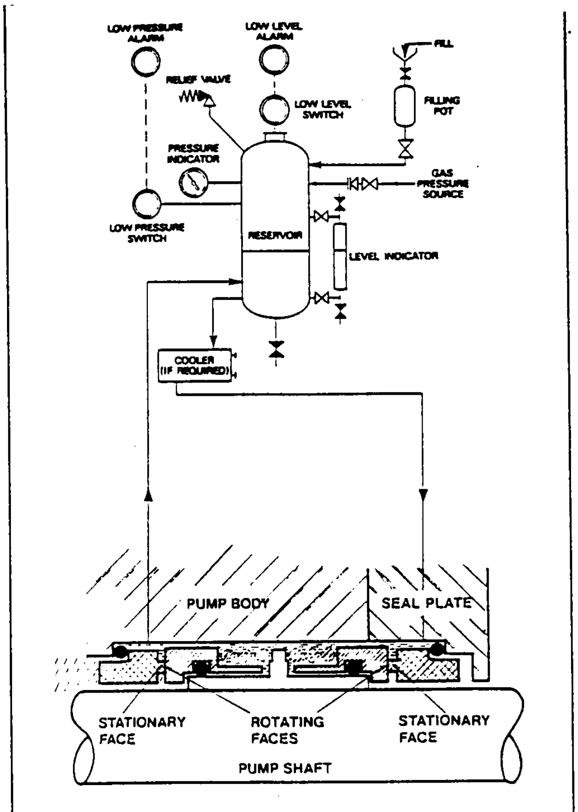
Double seals are often used when a single seal cannot give satisfactory performance on the pumped products, or where any product leakage is unacceptable for economic or hazardous reasons.

In the double seal configuration two single seals are arranged back to back to maintain a liquid barrier between the pumped product and barrier fluid. The inner seal operates on the barrier fluid which is kept at a pressure higher than the product, and therefore any
leakage is of the barrier fluid and not the pumped product. The product and pumped liquid must consequently be compatible. Most Flexibox single, mechanical seals have been successfully used in double seal configurations and Fexibox also have specific ranges of integral double seals available. In designing these double seal ranges a short axial length has been achieved by using components which are common to both inner and outer seals.

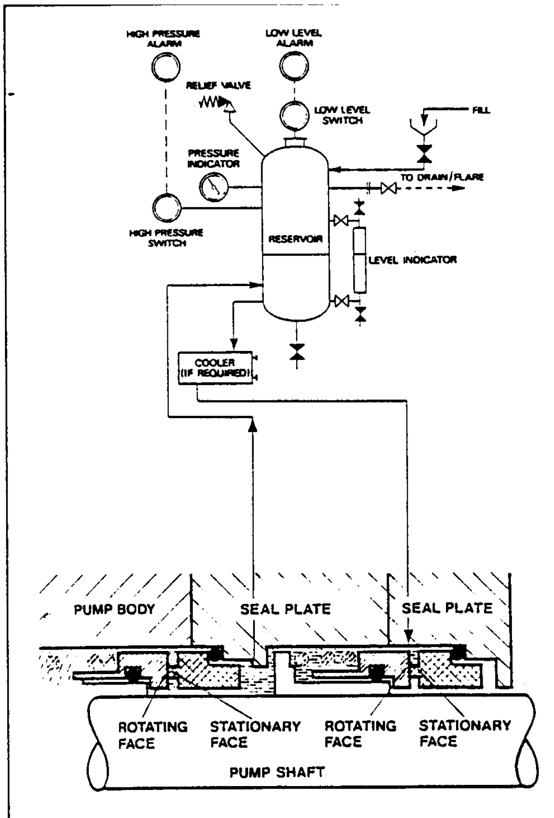
Tandem Seals

A tandem seal is defined as an arrangement, using two seals, where the pressure between the seals, i.e. the barrier fluid pressure, is less than that of the product pressure at the inner seal.

Tandem seal arrangements are used to solve a variety of sealing problems and are particularly useful when contamination of the product by a barrier liquid must be avoided. In a tandem arrangement, single seals are arranged in line and the inner seal operates on the
pumped product. Any leakage which may occur will be from the pumped product into the barrier fluid, thereby avoiding contamination of the product.



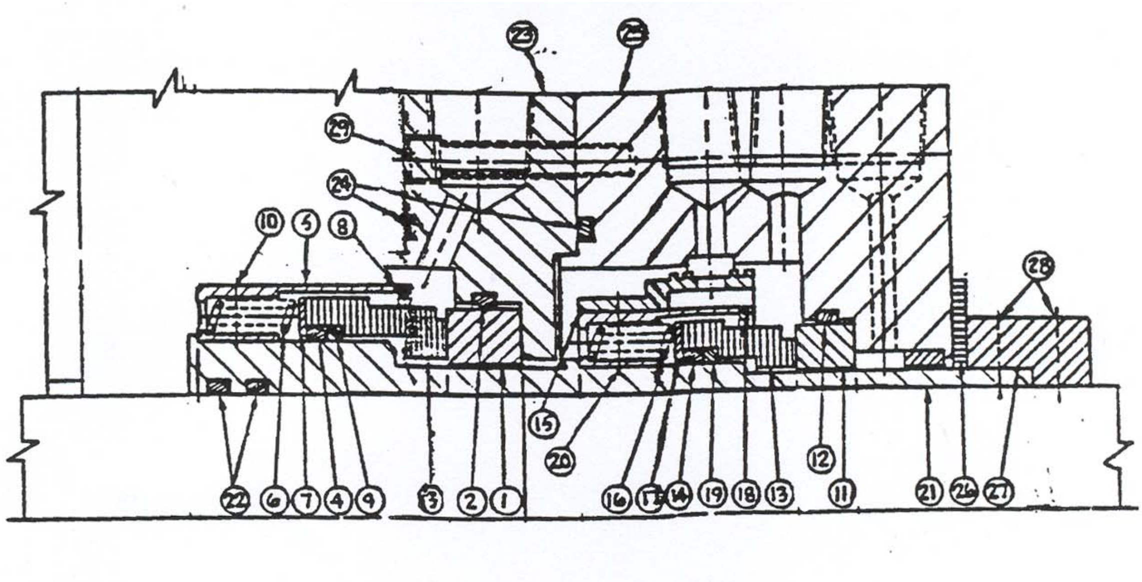
DOUBLE SEAL



TANDEM SEAL

TANDEM SEALS

15	RETAINER ASSY	316 – S.S.	1
16	SPRING	316 – S.S.	6
17	DISC	316 – S.S.	1
18	SNAP RING	316 – S.S.	1
19	ANTI – X – RING	CHEMLON	1
20	SET SCREW	316 – S.S.	3
21	SLEEVE	18-8 S.S.	1
22	O-RING	BUNA – N	2
23	GLAND PLATE	18-8 S.S.	1
24	O-RING	BUNA – N	2
25	GLAND PLATE ASSY	18-8 S.S./ BRONZE	1
26	SPACER	555	3
27	COLLAR	18-8 S.S.	1
28	SET SCREW	316 – S.S.	12
29	CAP SCREW	316 – S.S.	4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Objective Three

INSTALLATION

An analysis of seal failures reports show that a large proportion of failures are caused by pumping or equipment problems, i.e. operational, vibration and eccentricity. The next section deals with equipment checks that should be car-ried out to ensure long and continuous service from your mechanical seals.

Misalignment

Alignment between equipment and driving unit is very important in order to obtain maximum running efficiency, to reduce maintenance and ultimately break-downs. Although normally referred to as coupling alignment, it is actually the alignment of the shaft centre lines. Correct alignment between the driving and driven shaft is equally important to obtain the maximum life from your mechanical seals. Bad alignment puts lateral, angular and axial load on the shafts which excessive errors which in turn puts extra load on the bearings. This results in excessive wear and bearing failures allowing the shaft to move in an orbital fashion or to become eccentric, causing swash and track run out at the seal faces.

Eccentricity

Here again, swash and tracking at the seal face can be caused by eccentricity. Another serious fault arising from eccentricity, which fortunately does not apply to Flexibox seals using a spring drive, is wear on drive pins and slots. The conclusion to be drawn is that the seal with the maximum amount of flexibility built into it must be considered the best, hence the name 'Flex-in-box' or' FLEXIBOX'.

Hot Alignment Check

After the initial alignment check, equipment handling hot Products should be given a further alignment check; this is normally carried out ofter several hours running at working temperature. Due to the conditions, the job is hot and arduous and should be carried out as soon as possible after shut down. If it is found from readings taken that the driving or driven unit has expanded outside the permitted limits then the alignment shims should be adjusted to allow for the amount of movement.

Some equipment manufacturers give an expansion growth to a given temperature, e.g. 0.016 inch at 400 degrees centigrade, but to be safe the above method is recommended.

Pipe Strain

Where repeated alignment has to be carried out to correct a fault it may be that there is a strain on the equipment from the pipework attached to it. The suggested method to correct this is to unbolt the pipework and carry out the alignment in the usual manner leaving the dial indicators in position whilst the pipe flanges are re-connected, observe any variation in the dial that the pipe strain is severe and other corrective methods must be carried out, i.e. cutting pipework and re0welding.

Limits

A recommended allowable limit of 0.05 mm between the coupling faces and 0.10 mm TIR on the peripery may be worked to when aligning couplings on medium speed applications. Other applications are treated on their own merits i.e. turbine and high speed applications could be as low as 0.02 mm.

Pump and Equipment Checks

To ensure maximum service life, we advise that the following dial indicator checks to be carried out on equipment to pin- point any excessive errors which may have prevented the seals from running concentric (within limits). These checks are also good maintenance practice and can only contributed to reduce down- time and running costs.

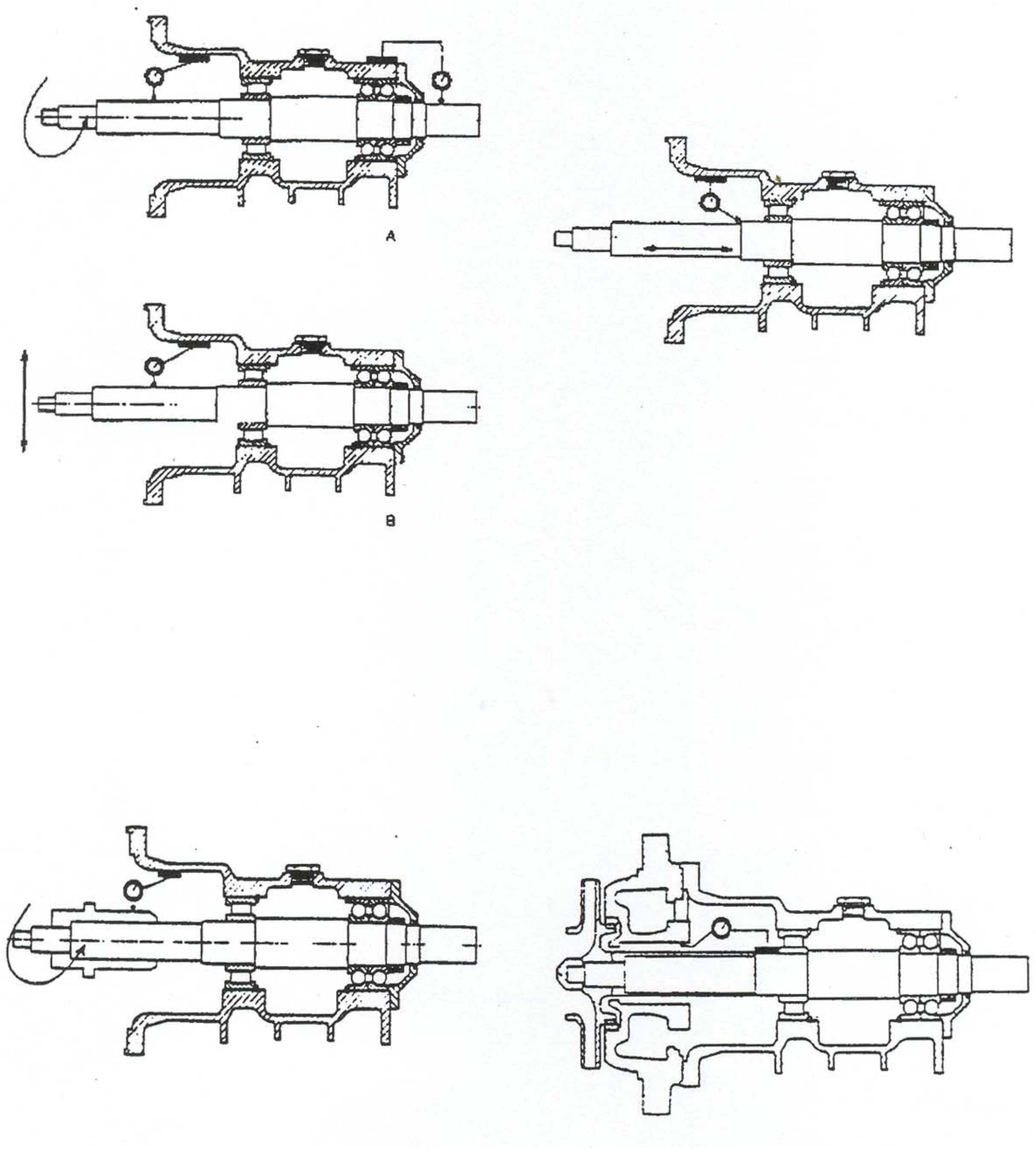
3.1 Shaft Run Out, Deflection Lift

If possible, mount two indicators locating the stems as shown in Fig. A rotation of the shaft could show radial run out at the seal end whilst observation of both
Indicators would show whether or not the shaft was bent. Lightly lifting the shaft may show a greater reading than shaft run out, which would indicate wear in the bearings. These faults can result in fretting and wear of seal components and variation in the fluid film, which shortens seal life, also eccentricity which can result in bearing failure.
On multi-stage and split casing pumps the shaft should be removed from the pump and taken to the workshop where it should be checked between centres, taking readings at several points. This will show whether or not the shaft is bent. The maximum TIR at any point should not exceed 0.05mm (0.002")

3.2 End Play Or Shaft Float

With the dial indicator mounted on the pump housing and the stem located against the shoulder of the shaft, sleeve or deflector ring, attempts should be made to move the shaft from end to end.

This is normally done by tapping it lightly with a hide mallet. End play should normally be less than 0.05mm (0.002"), although with some types of bearings, for instance 'Michell' type, there could be as much as 0.20mm.
The excessive end play causes fretting and wears between the point of contact of the rotary seal ring packing and the shaft or sleeve.



3.3 Concentricity of Sleeve

Having checked the shaft for run out it now remains for the sleeve to be checked, locating and locking it into position before doing so. Allowing for tolerances the total run-out of the sleeve should not exceed 0.05mirn (0.002") TIR. If found to be more than this, then the fault should be corrected. Here again, the effects are similar swash and tracking at the seal faces, variation of the fluid film thickness (which is very important on light hydrocarbons) setting up vibration and possibly causing bearing failure.

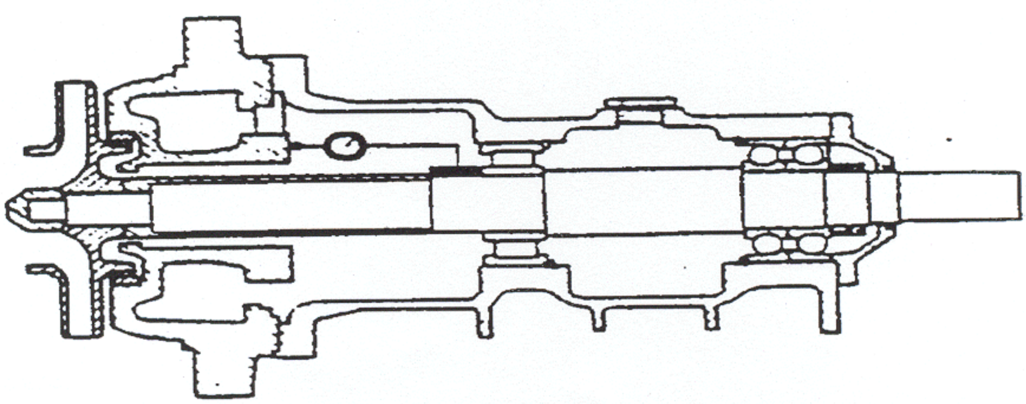
3.4 Concentricity of Stuffing Box

To carry out this check it may be necessary to make available an adaptor to fit onto the indicator stem to enable readings to be taken between the box bore and shaft TIR should not exceed 0.1 Omm (0.004"). Effects of this fault are track-ing and swash at the seal face resulting in uneven wear on the carbon face and eventually breakdown.

3.5 Squareness of the Stuffing Box

With the pump completely assembled, including the thrust bearing but without the seals, mount the dial indicator on the shaft with the stem located on the stuffing box face. (it may be necessary to make a special adaptor if space is restricted.). When the dial indicator is rotated TIR should not exceed 0.08mm (0.0OW). If the stuffing box face is not normal to the centre line of the shaft the stationary seal face will also not be square, causing the problems associated with washing or wobbling of the rotary faces. If the run-out is excessive contact may not be maintained between the

Seal faces and fretting may occur under the rotary seal ring packing.



Objective Four

CAUSES OF SEAL FAILURES:

An indication of the type of failure can be gained by a visual inspection of the running characteristics and nature of the leakage etc. Precise identification of a fault can only be achieved by close examination of the seal components during the removal of the seal, from which it may possible to determine the leakage paths.

4.1 Vaporisation

Occurs when heat generated at the faces is not removed effectively and local boiling of the interface film takes place, indicated by a popping or puffing noise.
Occasionally (nearly always on water) a seal will blow open and remain open.

4.2 Dry Running

Dry running occurs when no,or insufficient, liquid exists between the two seal faces.

4.3 Abrasives in Product

If the product being pumped contains any abrasive matter then this will tend to penetrate the seal faces leading to rapid wear and seal failure.

4.4 Sludging/ Bonding

Sludging is associated with the sealing of high viscosity liquids. The wear stresses between the seal faces can exceed the ruptive strength of the carbon and particles are pulled from the face of the stationary seal ring. The problem can be particularly acute on pumps sealing hydrocarbon liquids at temperatures above ambient. When shut down the viscosity of the liquid and interface film increases as the temperature drops and problems may arise on re-starting the pump. Another possible cause of sludging occurs when the interface film partially carbonises due to overheating.

Bonding

A similar type of phenomenon to sludging is bonding. In this case a bond is formed between the two seal faces usually crystallisation after the pump has been standing for a long period. On starting particles are pulled from the carbon face and leakage occurs.

4.5 Coking

Coking is a type of failure that frequently occurs when the product is hydrocarbons at high temperatures.
Minute quantities of film ;leakage tend to carbonise on the atmosphere side of the seal, causing the sliding member (the rotary seal ring) to jam up and preventing it from following up any face wear. This type of seal failure would be indicated, on stripping down for inspection, by the rotary seal ring having no sliding action after removing the seal plate.

4.6 Carbon Ring Erosion

Occurs when the differential of the circulation to the seal between the point of take off and entry to the seal is too great, or when circulation flow contains abrasive matter.

4.7 Face Distortion

In some cases leakage can be due to face distortion.

4.8 Broken Carbon Seal Rings

This problem mainly arises on seals fitted with PTFE 'O' rings when the pin sleeve has been omitted on re-assembly. Due to the low coefficient of friction on of the PTFE 'O' rings the carbon ring may on occasions tend to spin, allowing the side of the slot in the carbon ring to come into sudden contact with the pin.
This can also happen occasionally on seals fitted with synthetic rubber 'O' rings on high viscosity duties.

4.9 'O' Ring Extrusion

Extrusion occurs when parts of the 'O' ring is forced through close clearance gaps.
Extrusion can be caused by the use of excessive force when fitting and assembling components. Operational extrusion is generally caused by excessive pressure combined with overheating and incompatibility. It can also be caused when seal parts have been reconditioned on site and size are beyond their limits, creating larger clearance between the components.

4. 10 'O' Ring Overheating

Overheating of the packing is generally caused by adverse conditions at the seal faces causing excessive heat generation.

4.11 Sleeve Damage, Preventing Follow Up Of Rotary Seal Ring

Vibration

Severe vibration of the shaft or pump will cause the close clearance of the landings, on either side of the 'O' ring grooves in the rotary seal ring, to come into contact with the nose of the sleeve, resulting in fretting and marking into which foreign matter lodges, thus preventing the sliding member from moving.

4.12 Spring Distortion or Breakage

In most single spring seals, the drive is unidirectional. In operation the spring should always grip the sleeve and rotary seal ring. If, for any reason, the wrong hand spring is fitted, or the pump shaft is rotated in the wrong direction, e.g. turbined backwards, the spring will tend to uncoil, slip and distort or crack the spring may even break. However, this type of failure occurs more frequently when springs are fitted incorrectly on high viscosity duties, where excessible torque at the seal faces may be caused by sludging or boding. On multi-spring seals, a build-up of solids around the springs can make some of them inneffective and overload the remaining springs, causing failure.

4. 13 Checklist of Identifying Causes of Seal leakage

Symptom	*Possible causes*	*Corrective procedures*
Seal spits and sputters (face popping) in operation	*Seal fluid vaporising at seal interfaces*	*Increase cooling of seal faces* *Add bypass flush line if not in use* *Enlarge bypass flush line and/or orifices in gland plate*
Seal drips steadily	*Faces not flat* *Carbon graphite seal faces blistered* *Seal faces thermally distorted*	*Check for incorrect installation dimensions* *Improve cooling flush lines* *Check for gland plate plate distortion due to overtorquing of gland bolts* *Check gland gasket for proper compression* *Clean out foreign particles between seal faces; relap faces if necessary* *Check for cracks and chips at seal faces; replace primary and mating rings.*
	Secondary seals nicked or scratched during installation *O rings overaged* *Secondary seals hard and brittle from compression set* *Secondary seals soft and sticky from chemical attack*	*Replace secondary seals* *Check for proper lead in chamfers, burrs, etc.*
	Spring failure *Drive mechanism corroded*	*Replace parts*
Seal squeals during operation	*Amount of liquid inadequate to lubricate seal faces*	*Add bypass flush line if not in use* *Enlarge bypass flush line and/or orifices in gland plate.*

Carbon dust accumulates on outside of gland ring	*Amount of liquid inadequate to lubricate seal faces* *Liquid film evaporating between seal faces*	*Add by pass flush line if not in use* *Enlarge bypass flush line and/or orifices in gland plate* *Check for proper seal design with seal manufacturer if pressure in stuffing box is excessively high*
Seal leaks	*Nothing appears to be wrong*	*Refer to list under "seal drips steadily"* *Check for squareness of stuffing box to shaft* *Align shaft, impeller, bearing, etc., to prevent shaft vibration and/or distortion of gland plate and/or mating ring*
Seal life is short	*Abrasive fluid*	*Prevent abrasives from accumulating at seal faces* *Add bypass flush line if not in use* *Use abrasive separator or filter*
	Seal running too hot	*Increase cooling of seal faces* *Increase bypass flush line flow* *Check for obstructed flow in cooling lines*
	Equipment mechanically out of line	*Align* *Check for rubbing of seal on shaft*

**POSSIBLE LEAKAGE PATHS
FROM A STANDARD BALANCED SEAL**