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# Practical Centrifugal Pumps

Paresh Girdhar
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# Practical Centrifugal Pumps

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# Practical Centrifugal Pumps

## Design, Operation and Maintenance

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# Contents

# Preface

This books covers the essentials of pump construction, design applications, operations, maintenance and management issues and the authors have tried to provide you with the most up-to-date information and best practice in dealing with the subject. Key topics which the book homes in on are: the various types of centrifugal pumps; relevant pump terminology; pump characteristics and pump curves; pump calculations; auxiliary equipment associated with pumping circuits; operating pump systems – drafting the correct operations, controls and procedures; pump reliability definition in terms of availability, criticality and wear characteristics; pump efficiency – capital, maintenance and life cycle costs.

From the reader's perspective the following is offered:

- If you are an engineer or technician you will learn the inside information on why and how pumps are designed. No longer will you be specifying pumps you don't understand.
- If you are working in the plant and maintenance area you will learn how pumps work, what the main causes of pump problems are and how to fix them quickly and effectively.
- Also if you are a design engineer or technician, you will gain a global picture in designing pumps from the authors' many years of experience.

We would hope that you will gain the following knowledge from this book:

- Pump terminology
- Real pump classifications, types and criteria for selection
- How to read pump curves and cross referencing issues
- Pump efficiency determination and cost analysis
- Critical elements in pump system design
- Shaft seal selection and failure determination
- How to install and commission a pump
- Condition monitoring and trouble-shooting of pumps
- What makes up a pump's total discharge head requirement
- How to install pumps
- How to look after pump bearings
- Precautions when starting up a new pump or after strip-down for maintenance.

Typical people who will find this book useful include:

- Plant Operations & Maintenance Personnel
- Plant Engineer, Managers & Supervisors
- Process Control Engineers & Supervisors
- Consulting Engineers
- Maintenance Engineers & Technicians
- Pump Sales and Applications Personnel
- Pump Users
- Pump Service Contractors.

You should have a modicum of mechanical knowledge and some exposure to pumping systems to derive maximum benefit from this book.

# 1

# Introduction

The transfer of liquids against gravity existed from time immemorial. A pump is one such device that expends energy to raise, transport, or compress liquids. The earliest known pump devices go back a few thousand years. One such early pump device was called 'Noria', similar to the Persian and the Roman water wheels. Noria was used for irrigating fields (Figure 1.1).


**Figure 1.1**
*Noria water wheel (From the Ripley's believe it not)*

The ancient Egyptians invented water wheels with buckets mounted on them to transfer water for irrigation. More than 2000 years ago, a Greek inventor, *Ctesibius*, made a similar type of pump for pumping water (Figure 1.2).

During the same period, Archimedes, a Greek mathematician, invented what is now known as the 'Archimedes' screw' – a pump designed like a screw rotating within a cylinder (Figure 1.3). The spiraled tube was set at an incline and was hand operated. This type of pump was used to drain and irrigate the Nile valley.

In 4th century Rome, Archimedes' screw was used for the Roman water supply systems, highly advanced for that time. The Romans also used screw pumps for irrigation and drainage work.

Screw pumps can also be traced to the ore mines of Spain. These early units were all driven by either man or animal power.

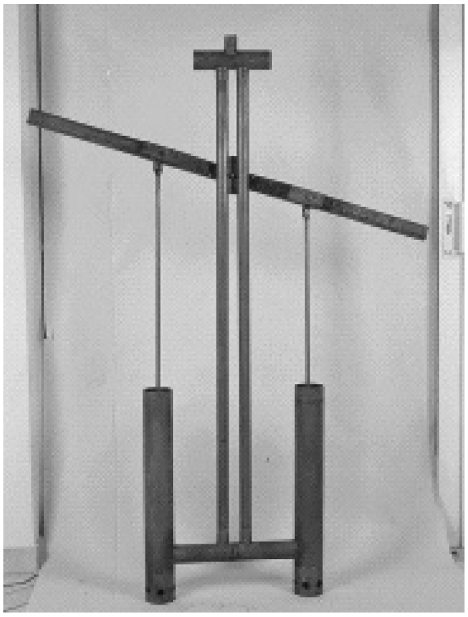


**Figure 1.2**
*Model of a piston pump made by Ctesbius*

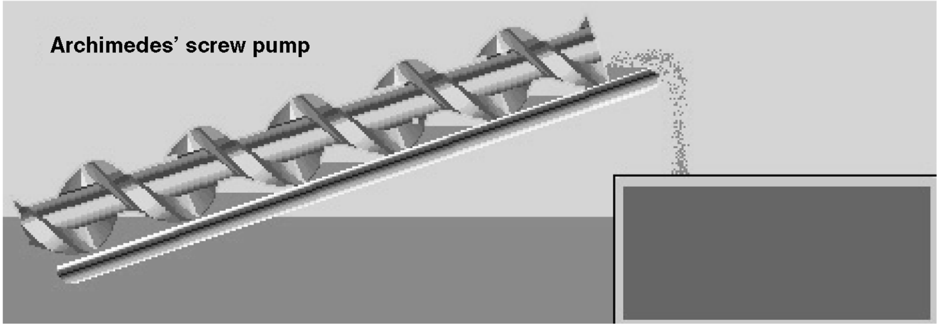



**Figure 1.3**
*Archimedes' screw pump*

The mining operations of the Middle Ages led to the development of the suction (piston) pump, types of which are described by Georgius Agricola in De re metallica (1556). Force pumps, utilizing a piston-and-cylinder combination, were used in Greece to raise water from wells (Figure 1.4).

Adopting a similar principle, air pumps operated spectacular musical devices in Greek temples and amphitheaters, such as the water organ.

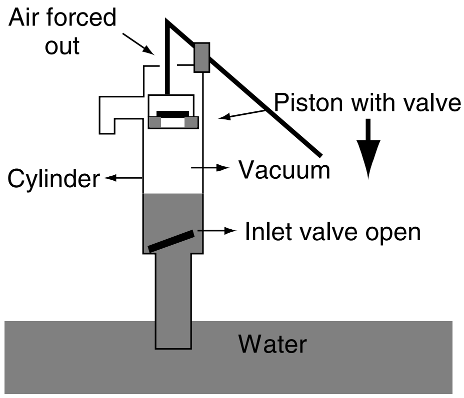



**Figure 1.4**
*Reciprocating hand pump in suction stroke*

## 1.1 Applications

Times have changed, but pumps still operate on the same fundamental principle – expend energy to raise, transport, or compress liquids. Over time, the application of pumps in the agricultural domain has expanded to cover other domains as well. The following are a few main domains that use pumps extensively:

- *Water supply*: To supply water to inhabited areas.
- *Drainage*: To control the level of water in a protected area.
- *Sewage*: To collect and treat sewage.
- *Irrigation*: To make dry lands agriculturally productive.
- *Chemical industry*: To transport fluids to and from various sites in the chemical plant.
- *Petroleum industry*: Used in every phase of petroleum production, transportation, and refinery.
- *Pharmaceutical and medical field*: To transfer of chemicals in drug manufacture; pump fluids in and out of the body.
- *Steel mills*: To transport cooling water.
- *Construction*: Bypass pumping, well-point dewatering, remediation, and general site pumping applications.
- *Mining*: Heavy-duty construction, wash water, dust control fines and tailings pumping, site dewatering, groundwater control, and water runoff.

Pumps are also used for diverse applications like in transfer of potatoes, to peel the skin of hazelnuts in chocolate manufacture, and to cut metal sheets in areas that are too hazardous to allow cutting by a gas flame torch. The artificial heart is also a mechanical pump. The smallest pump ever made is no bigger than the tip of a finger. It moves between 10 and 30 nl of liquid in one cycle (10- to 30-thousandths of a drop of water). It was not found to have any practical use so maybe it was created just for the records!

## 1.2 Pump types

Pumps can be classified on various bases. For example, a typical classification of rotating shaft (kinetic) pumps is given in Appendix.

Pumps based on their principle of operation are primarily classified into:

- Positive displacement pumps (reciprocating, rotary pumps)
- Roto-dynamic pumps (centrifugal pumps)
- Others.

### 1.2.1 Positive displacement pumps

Positive displacement pumps, which lift a given volume for each cycle of operation, can be divided into two main classes, reciprocating and rotary.

Reciprocating pumps include piston, plunger, and diaphragm types. The rotary pumps include gear, lobe, screw, vane, regenerative (peripheral), and progressive cavity pumps.

### 1.2.2 Roto-dynamic pumps

Roto-dynamic pumps raise the pressure of the liquid by first imparting velocity energy to it and then converting this to pressure energy. These are also called centrifugal pumps. Centrifugal pumps include radial, axial, and mixed flow units.

A radial flow pump is commonly referred to as a straight centrifugal pump; the most common type is the volute pump. Fluid enters the pump through the eye of impeller, which rotates at high speed. The fluid is accelerated *radially* outward from the pump casing. A partial vacuum is created that continuously draws more fluid into the pump if properly primed.

In the axial flow centrifugal pumps, the rotor is a propeller. Fluid flows parallel to the axis of the shaft. The mixed flow, the direction of liquid from the impeller acts as an in-between that of the radial and axial flow pumps.

### 1.2.3 Other types

The other types include electromagnetic pumps, jet pumps, gas lift pumps, and hydraulic ram pumps.

## 1.3 Reciprocating pumps

Reciprocating pumps are positive displacement pumps and are based on the principle of the 2000-year-old pump made by the Greek inventor, Ctesibius.

### 1.3.1 Plunger pumps

Plunger pumps comprise of a cylinder with a reciprocating plunger in it (Figure 1.5). The head of the cylinder houses the suction and the discharge valves.

In the suction stroke, as the plunger retracts, the suction valve opens causing suction of the liquid within the cylinder.

In the forward stroke, the plunger then pushes the liquid out into the discharge header. The pressure built in the cylinder is marginally over the pressure in the discharge.

The gland packings help to contain the pressurized fluid within the cylinder. The plungers are operated using the slider-crank mechanism. Usually, two or three cylinders are placed alongside and their plungers reciprocate from the same crankshaft. These are called as duplex or triplex plunger pumps.

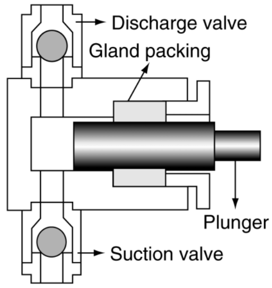



**Figure 1.5**
*Plunger pump*

### 1.3.2 Diaphragm pumps

Diaphragm pumps are inherently plunger pumps. The plunger, however, pressurizes the hydraulic oil and this pressurized oil is used to flex the diaphragm and cause the pumping of the process liquid.

Diaphragm pumps are primarily used when the liquids to be pumped are hazardous or toxic. Thus, these pumps are often provided with diaphragm rupture indicators.

Diaphragm pumps that are designed to pump hazardous fluids usually have a double diaphragm which is separated by a thin film of water (for example, see Figure 1.6). A pressure sensor senses the pressure of this water. In a normal condition, the pressure on the process and oil sides of the diaphragms is always the same and the pressure between the diaphragms is zero.

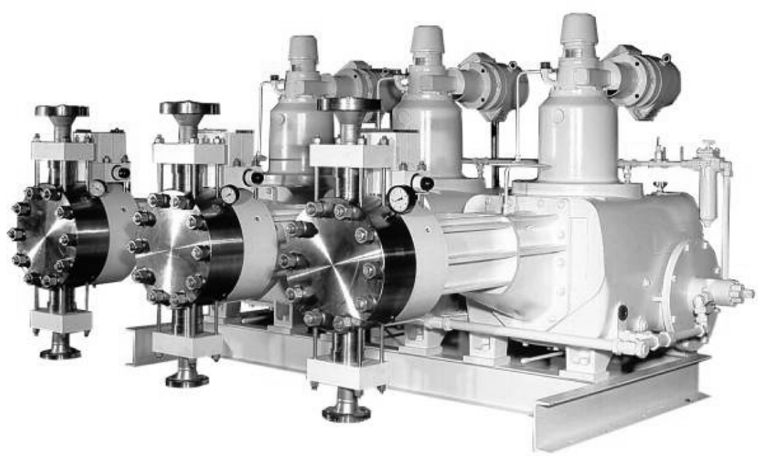


**Figure 1.6**
*Double diaphragm pumps (Lewa pumps)*

However, no sooner does one of them ruptures than the pressure sensor records a maximum of process discharge pressure. The rising of this pressure is an indicator of the diaphragm rupture (Figure 1.7).

Even with the rupture of just one diaphragm, the process liquid does not come into contact with the atmosphere.

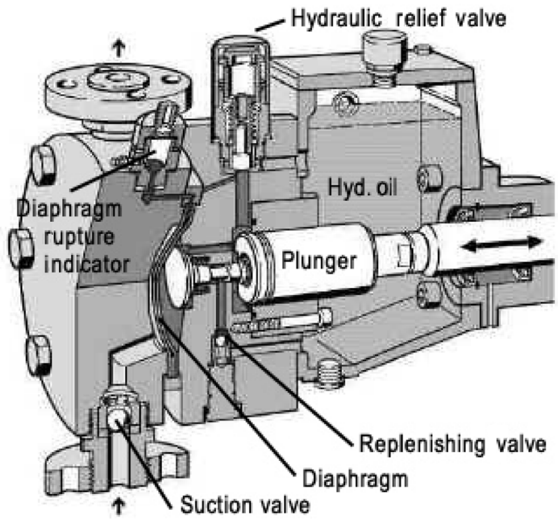



**Figure 1.7**
*Diaphragm pump*

# 1.4 Rotary pumps

## 1.4.1 Gear pump

Gear pumps are of two types:

1. External gear pump
2. Internal gear pump.

### External gear pump

In external gear pumps, two identical gears rotate against each other. The motor provides the drive for one gear. This gear in turn drives the other gear. A separate shaft supports each gear, which contains bearings on both of its sides (Figure 1.8).

As the gears come out of the mesh, they create expanding volume on the inlet side of the pump. Liquid flows into the cavity and is trapped by the gear teeth while they rotate.

Liquid travels around the interior of the casing in the pockets between the teeth and the casing. The fine side clearances between the gear and the casing allow recirculation of the liquid between the gears.

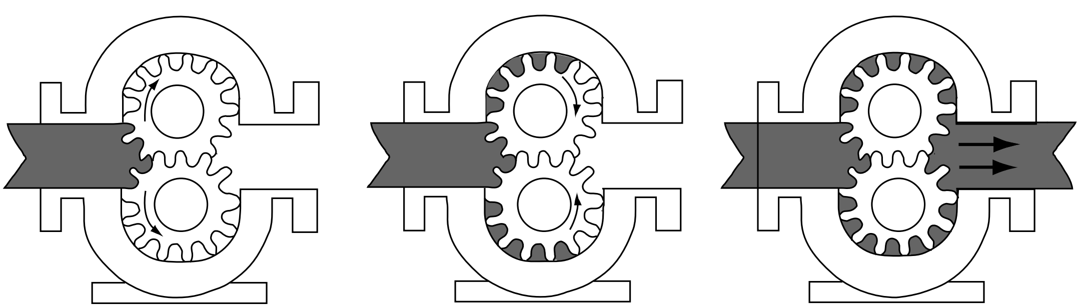


**Figure 1.8**
*External gear pump*

Finally, the meshing of the gears forces liquid through the outlet port under pressure. As the gears are supported on both sides, the noise levels of these pumps are lower and are typically used for high-pressure applications such as the hydraulic applications.

### Internal gear pump

Internal gear pumps have only two moving parts (Figure 1.9). They can operate in either direction, which allows for maximum utility with a variety of application requirements.

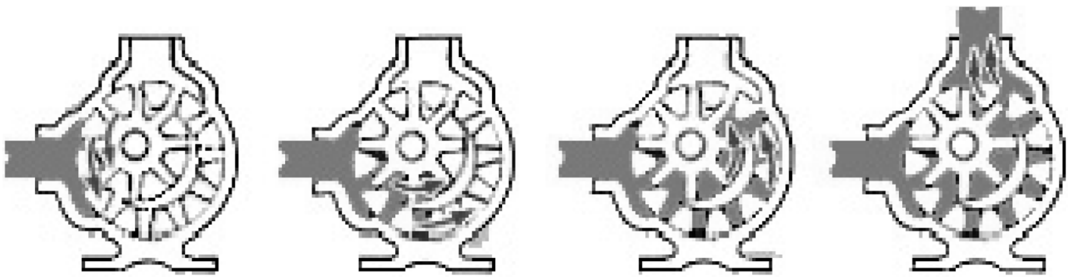


**Figure 1.9**
*Internal gear pump*

In these pumps, liquid enters the suction port between the large exterior gears, rotor, and the smaller interior gear teeth, idler. The arrows indicate the direction of the pump and the liquid.

Liquid travels through the pump between the teeth of the 'gear-within-a-gear' principle. The crescent shape divides the liquid and acts as a seal between the suction and the discharge ports.

The pump head is now nearly flooded as it forces the liquid out of the discharge port.

Rotor and idler teeth mesh completely to form a seal equidistant from the discharge and suction ports. This seal forces the liquid out of the discharge port.

The internal gear pumps are capable of handling liquid from very low to very high viscosities. In addition to superior high-viscosity handling capabilities, internal gear pumps offer a smooth, nonpulsating flow. Internal gear pumps are self-priming and can run dry.

## 1.4.2 Lobe pump

The operation of the lobe pumps is similar to the operation of the external gear pumps (Figure 1.10). Here, each of the lobes is driven by external timing gears. As a result, the lobes do not make contact.

Pump shaft support bearings are located in the gearbox, and since the bearings are not within the pumped liquid, pressure is limited by the location of the bearing and shaft deflection.

As the lobes come out of mesh, they create expanding volume on the inlet side of the pump. The liquid then flows into the cavity and is trapped by the lobes as they rotate.

The liquid travels around the interior of the casing in the pockets between the lobes and the casing and it does not pass between the lobes.

Finally, the meshing of the lobes forces the liquid through the outlet port under pressure. Lobe pumps are frequently used in food applications because they can handle solids without damaging the product. The particle size pumped can be much larger in lobe pumps than in any other of the PD types.

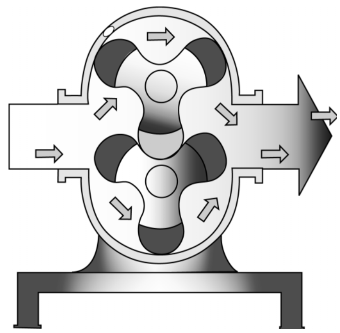


**Figure 1.10**
*Lobe pump*

### 1.4.3 Vane pump

A vane pump too traps the liquid by forming a compartment comprising of vanes and the casing (Figure 1.11). As the rotor turns, the trapped liquid is traversed from the suction port to the discharge port.

A slotted rotor or impeller is eccentrically supported in a cycloidal cam. The rotor is located close to the wall of the cam so a crescent-shaped cavity is formed. The rotor is sealed in the cam by two side plates. Vanes or blades fit within the slots of the impeller. As the impeller rotates and fluid enters the pump, centrifugal force, hydraulic pressure, and/or pushrods push the vanes to the walls of the housing. The tight seal among the vanes, rotor, cam, and side plate is the key to the good suction characteristics common to the Vane pumping principle.

The housing and cam force fluid into the pumping chamber through the holes in the cam. Fluid enters the pockets created by the vanes, rotor, cam, and side plate.

As the impeller continues around, the vanes sweep the fluid to the opposite side of the crescent where it is squeezed through the discharge holes of the cam as the vane approaches the point of the crescent. Fluid then exits the discharge port.

Vane pumps are ideally suited for low-viscosity, nonlubricating liquids.

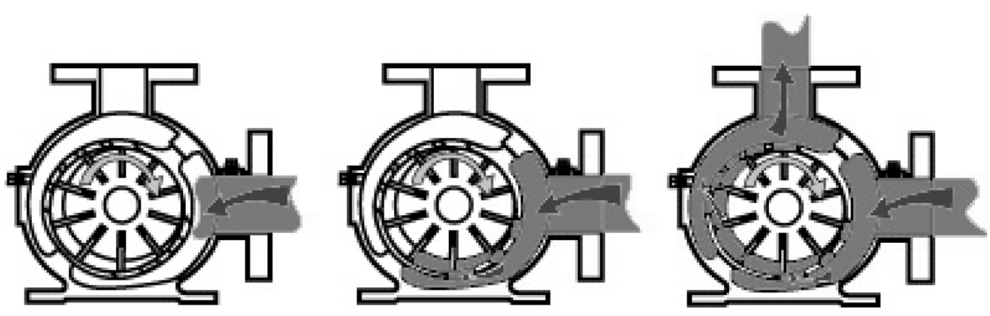


**Figure 1.11**
*Vane pump*

### 1.4.4 Progressive cavity pump

A progressive cavity pump consists of only one basic moving part, which is the driven metal rotor rotating within an elastomer-lined (elastic) stator (Figure 1.12).

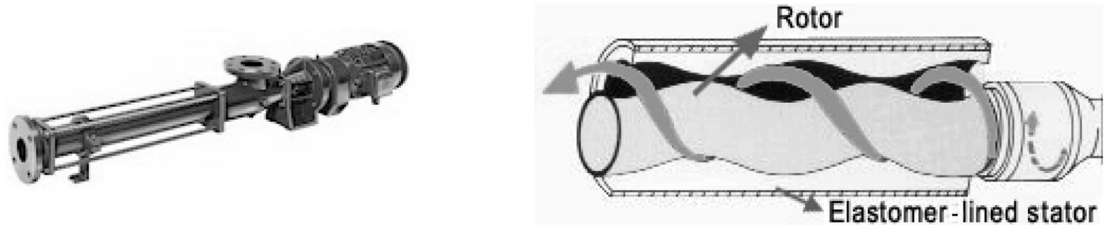



**Figure 1.12**
*Vane pump progressive cavity pump*

As the rotor turns, chambers are formed between the rotor and stator. These chambers progress axially from the suction to the discharge end, moving the fluid. By increasing the pitch of the rotor and stator, additional chambers or stages are formed.

The Vane pumps are solutions to the special pumping problems of municipal and industrial wastewater and waste processing operations. Industries, such as, chemical, petrochemical, food, paper and pulp, construction, mining, cosmetic, and industrial finishing, find these pumps are ideally suited for pumping fluids with nonabrasive material inclusion.

### 1.4.5 Peripheral pump

As shown in Figure 1.13, the impeller has a large number of small radial vanes on both of its sides. The impeller runs in a concentric circular casing. Interaction between the casing and the vanes creates a vortex in the spaces between the vanes and the casing, and the mechanical energy is transmitted to the pumped liquid.

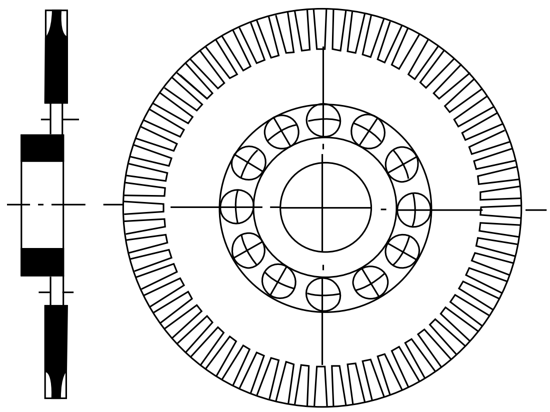


**Figure 1.13**
*Peripheral pump impeller*

Peripheral pumps are relatively inefficient and have poor self-priming capability. They can handle large amounts of entrained gas. They are suitable to low flow and high-pressure applications with clean liquids.

### 1.4.6 Screw pump

In addition to the previously described pumps based on the Archimedes' screw, there are pumps fitted with two or three spindles crews housed in a casing.

Three-spindle screw pumps, as shown in Figure 1.14, are ideally suited for a variety of marine and offshore applications such as fuel-injection, oil burners, boosting, hydraulics, fuel, lubrication, circulating, feed, and many more. The pumps deliver pulsation free flow and operate with low noise levels. These pumps are self-priming with good efficiency. These pumps are also ideal for highly viscous liquids.

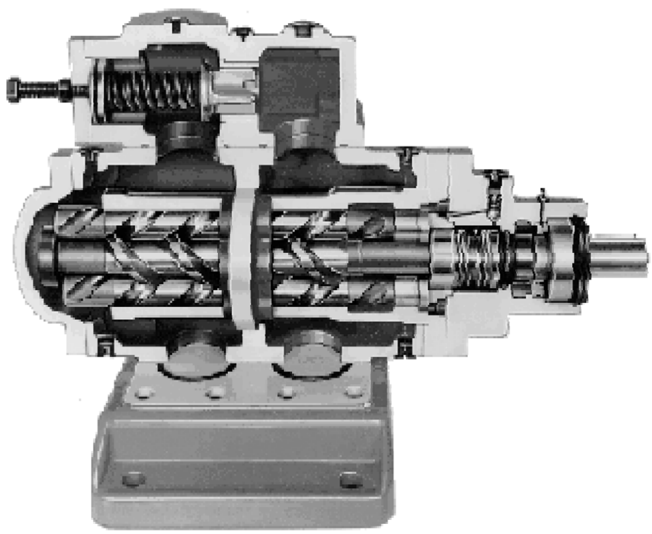


**Figure 1.14**
*Three-spindle screw pump – Alweiller pumps*

## 1.5 Centrifugal pumps

The centrifugal pumps are by far the most commonly used of the pump types. Among all the installed pumps in a typical petroleum plant, almost 80–90% are centrifugal pumps. Centrifugal pumps are widely used because of their design simplicity, high efficiency, wide range of capacity, head, smooth flow rate, and ease of operation and maintenance.

The 'modern' era pumps began during the late 17th and early 18th centuries AD. British engineer Thomas Savery, French physicist Denis Papin, and British blacksmith and inventor Thomas Newcomen contributed to the development of a water pump that used steam to power the pump's piston. The steam-powered water pump's first wide use was in pumping water out of mines.

However, the origin of the centrifugal impeller is attributed to the French physicist and inventor Denis Papin in 1689 (Figure 1.15).

Papin's contribution lies in his understanding of the concept of creating a forced vortex within a circular or spiral casing by means of vanes. The pump made by him had straight vanes.

Following Papin's theory, Combs presented a paper in 1838 on curved vanes and the effect of curvature, which subsequently proved to be an important factor in the development of the centrifugal impeller. In 1839, W.H. Andrews introduced the proper volute casing and in 1846, he used a fully shrouded impeller.

In addition, in 1846, W.H. Johnson constructed the first three-stage centrifugal pump, and in 1849, James S. Gwynne constructed a multistage centrifugal pump and began the first systematic examination of these pumps.

Around the same time, British inventor, John Appold conducted an exhaustive series of empirically directed experiments to determine the best shape of the impeller, which culminated in his discovery that efficiency depends on blade curvature. Appold's pump of 1851 with curved blades showed an efficiency of 68%, thus improving pump efficiency three-fold.

**Figure 1.15**
*Denis Papin*

The subsequent development of centrifugal pumps was very rapid due to its relatively inexpensive manufacturing and its ability to handle voluminous amounts of fluid. However, it has to be noted that the popularity of the centrifugal pumps has been made possible by major developments in the fields of electric motors, steam turbines, and internal combustion (IC) engines. Prior to this, the positive displacement type pumps were more widely used.

The centrifugal pump has a simple construction, essentially comprising a volute (1) and an impeller (2) (refer to Figure 1.16). The impeller is mounted on a shaft (5), which is supported by bearings (7) assembled in a bearing housing (6). A drive coupling is mounted on the free end of the shaft.

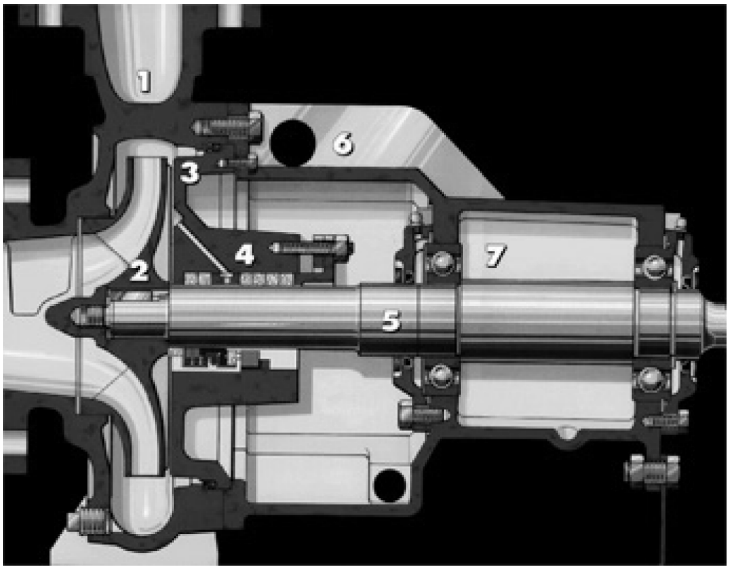



**Figure 1.16**
*Centrifugal pump – basic construction*

The prime mover, which is usually an electrical motor, steam turbine, or an IC engine, transmits the torque through the coupling.

As the impeller rotates, accelerates, and displaces the fluid within itself, more fluid is drawn into the impeller to take its place; if the pump is properly primed. The impeller thus, impacts kinetic or velocity energy to the fluid through mechanical action. This velocity energy is then converted to pressure energy by the volute. The pressure of the fluid formed in the casing has to be contained and this is achieved by an appropriate sealing arrangement (4). The seals are installed in the seal housing (3).

The normal operating speed of pumps is 1500 rpm (1800 rpm) and 3000 rpm (3600 rpm). However, there are certain designs of pumps that can operate at speeds in the range of 5000–25 000 rpm.

## 1.5.1 Types of centrifugal pumps

Centrifugal pumps can be categorized in various ways. Some of the main types are on the following basis:

### Orientation of the pump shaft axis

This refers to the plane on which the shaft axis of the pump is placed. It is either horizontal or vertical as shown in Figure 1.17.

**Figure 1.17**
*Vertical pump and horizontal pump*

### Number of stages

This refers to the number of sets of impellers and diffusers in a pump. A set forms a stage and it is usually single, dual, or multiple (more than two) stages (Figure 1.18).

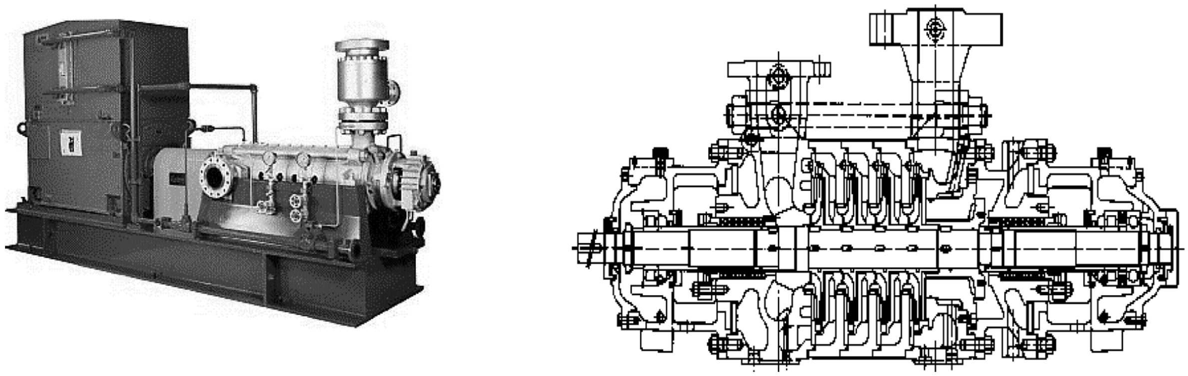


**Figure 1.18**
*Multistage pump*

### Suction flange orientation

This is based on the orientation of the pump suction flange. This orientation could be horizontal (also known as End) or vertical (also known as Top) (Figure 1.19).

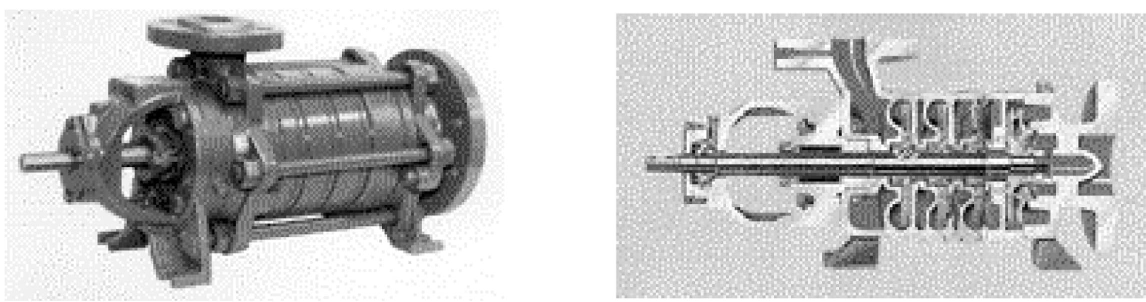


**Figure 1.19**
*Multistage pump with end suction*

### Casing split

This classification is based on the casing split. It is either Radial (perpendicular to shaft axis) or Axial (plane of the shaft axis) (Figure 1.20).

### Bearing support

This is judged based on the location of the bearings supporting the rotor. If the rotor is supported in the form of a cantilever (Figure 1.22), it is called as an Overhung type of pump. When the impellers on the rotor are supported with bearings on either side, the pump is called as an in-between bearings pump.

### Pump support

This refers to how the pump is supported on the base frame. It could be a center-line (Figure 1.21a) support or foot-mounted support (Figure 1.21b).

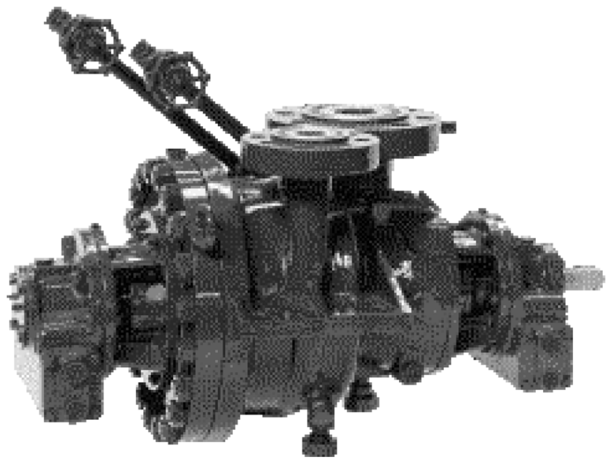


**Figure 1.20**
*Axial split casing*

(a) (b)

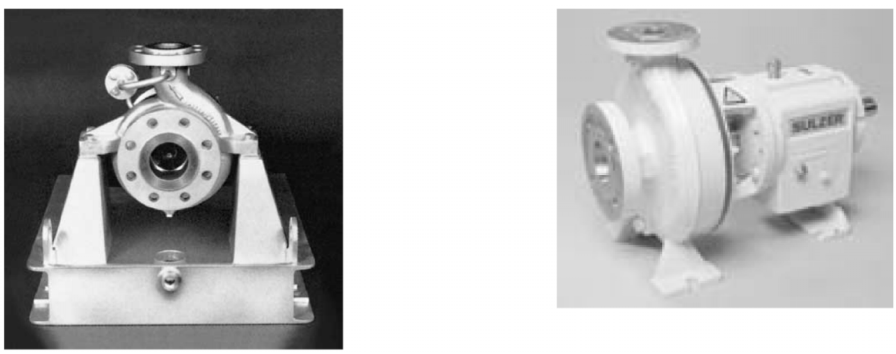



**Figure 1.21**
*Models of pump supports*

## Shaft connection

The closed coupled pumps are characterized by the absence of a coupling between the motor and the pump. The motor shaft has an extended length and the impeller is mounted on one end (Figure 1.22).

The vertical monobloc pumps have the suction and discharge flanges along one axis and can be mounted between pipelines. They are also termed as ‘in-line pumps’.

## Sealless pumps

Pumps are used to build the pressure in a liquid and if necessary to contain it within the casing. At the interface of the rotating shaft and the pump casing, mechanical seals are installed to do the job of product containment. However, seals are prone to leakages and this maybe unacceptable in certain critical applications. To address this issue, sealless pumps have been designed and manufactured.

**Figure 1.22**
*Closed coupled monobloc pumps with end suction*

These are of two types – canned and magnetic drive pumps:

1. *Canned pumps*: In the construction of this second type of sealless pump, the rotor comprises of an impeller, shaft, and the rotor of the motor. These are housed within the pump casing and a containment shell (Figure 1.23). The hazardous or the toxic liquid is confined within this shell and casing.

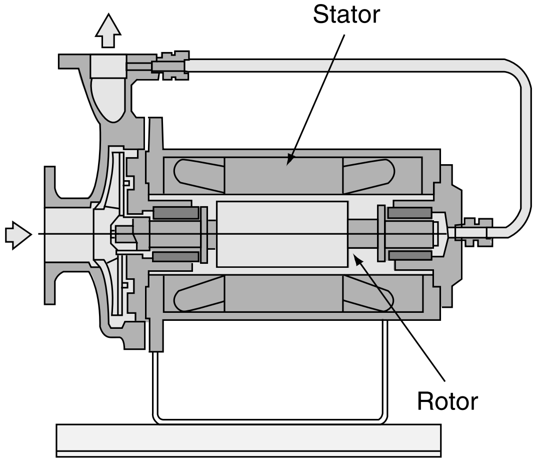



**Figure 1.23**
*Canned pump*

The rotating flux generated by the stator passes through the containment shell and drives the rotor and the impeller.

2. *Magnetic drive pumps*: In magnetic drive pumps, the rotor comprises of an impeller, shaft, and driven magnets. These housed within the pump casing and the containment shell ensures that the usually hazardous/toxic liquid is contained within a metal shell (Figure 1.24).

   The driven magnets take their drive from the rotating drive magnets, which are assembled on a different shaft that is coupled to the prime mover.

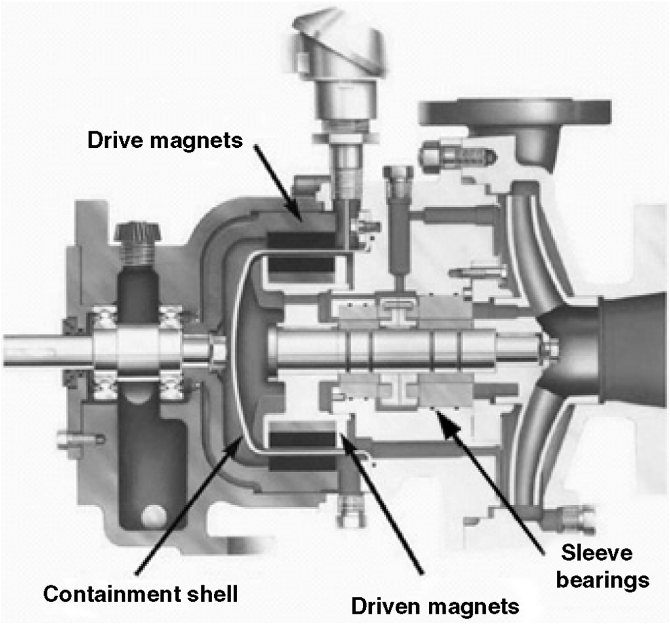



**Figure 1.24**
*Magnetic drive pump*

## 1.5.2 Pump standards

In order to bring about uniformity and minimum standards of design and dimensional specifications for centrifugal pumps, a number of centrifugal pump standards have been developed. These include the API (American Petroleum Institute), ISO (International Standards Organization), ANSI (American National Standards Institute), DIN (German), NFPA (Nation Fire Protection Agency), and AS-NZ (Australia–New Zealand).

Some of the famous standards, which are used in the development and manufacture of centrifugal pumps are API 610, ISO 5199, 2858, ANSI B73.1, DIN 24256, NFPA-21.

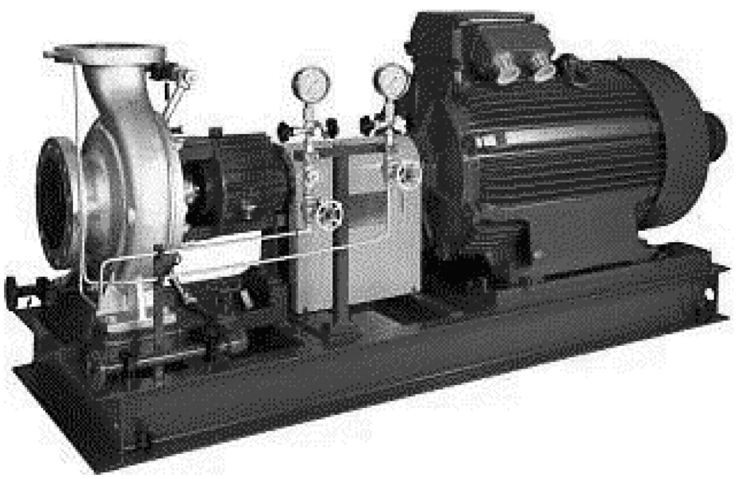


**Figure 1.25**
*Pump built to API 610 standard*

In addition to the above, there are many National Standards. Some of these are:

- *France*: NF E 44.121
- *United Kingdom*: BS 5257
- *German*: DIN 24256
- *Australia & New Zealand*: AS 2417-2001, grades 1 and 2.

Usually, the service criticality or application of the pump forms the deciding factor for a choice of standard. A critical refinery pump handling hazardous hydrocarbons would be in all probability built as per standard API 610 (Figure 1.25).

However, ordinary applications do not require the entire API-specified features and so the premium that comes with an API pump is not justified. Such pumps can be purchased built to lesser demanding standards like the ANSI B73.1. One big advantage of ANSI pumps is the outline dimensional interchangeability of same size pumps regardless of brand or manufacturer, something that is not available in the API pumps.

In a similar way, pumps meant for firewater applications are usually built to the design specifications laid out in NFPA-21.

There are some standards like the ISO 2858, which are primarily meant as dimensional standards. This does not provide any requirement for the pump's construction. The standard from ISO that addresses the design aspects of pumps is ISO 5199.

For a good comparative study of the API, ANSI, and ISO standards, it is recommended to read the technical paper called, 'ISO-5199 Standard Addresses Today's Reliability Requirements For Chemical Process Pumps', by Pierre H. Fabeck, Product Manager, Durco Europe, Brussels, Belgium and R. Barry Erickson, Manager of Engineering, The Duriron Company, Incorporated, Dayton, Ohio. This paper was presented at the 7th (1990) Pumps Symposium at the Texas A&M University.

### 1.5.3 Pump applications

The classification of pumps in the above sections is based on the construction of the pump and its components. However, on the basis of the applications for which they are designed, pumps tend to be built differently.

Some of the applications where typical pumps can be found are:

- Petroleum and chemical process pumps
- Electric, nuclear power pumps
- Waste/wastewater, cooling tower pumps
- Pulp and paper
- Slurry
- Pipeline, water-flood (injection) pumps
- High-speed pumps.

As this needs an introduction to the components/construction of the pump, these are covered in detail in subsequent topics.

# 2

# Centrifugal pump design and construction

In the previous chapter, we have studied different types of centrifugal and positive displacement pumps and their various distinguishing external features. In this chapter, we will learn about various types of internal components of centrifugal pumps.

The diversity among pumps does not only limit itself to the external features of the machines but also extends to its internal components. This is especially true in the case of centrifugal pumps. The basic components are essentially the same in almost every design but depending on the design and its applications, the construction features of the internal components differ to meet various requirements.

## 2.1 Impellers

The impeller of the centrifugal pump converts the mechanical rotation to the velocity of the liquid. The impeller acts as the spinning wheel in the pump.

It has an inlet eye through which the liquid suction occurs. The liquid is then guided from the inlet to the outlet of the impeller by vanes. The angle and shape of the vanes are designed based on flow rate. The guide vanes are usually cast with a back plate, termed *shroud* or *back cover*, and a front plate, termed *front cover*.

Impellers are generally made in castings and very rarely do come across fabricated and welded impellers.

Impellers can have many features on them like balancing holes and back vanes. These help in reducing the axial thrust generated by the hydraulic pressure. This is covered in Chapter 4.

In order to reduce recirculation losses and to enhance the volumetric efficiency of the impellers, they are provided with wearing rings. These rings maybe either on the front side or on both the front and backsides of the impeller. It is also possible to have an impeller without any wearing rings.

The casting process, as mentioned above, is the primary method of impeller manufacture. Smaller size impellers for clean water maybe cast in brass or bronze due to small section thickness of shrouds and blades. Recently, plastic has also been introduced as casting material.

For larger impellers and in most of the applications, cast iron is the first choice of the material. The grade used is ASTM A-48-40 (minimum tensile strength is 40 000 psi or 2720 kgs/cm$^2$).

This is used for a maximum peripheral speed of 55 m/s and a maximum temperature of 200 °C. When the temperature exceeds 200 °C, carbon steel castings of the grade A-216 WCA/WCC are recommended.

The adequacy of cast steel is dependent on its usage in handling of abrasives like ash, sand, or clinker. In such cases, the impellers could also be cast in 12% Cr steels (A-743 CA15). Stainless steel castings (A-744 CF8M) are used for their high corrosion resistance and for low-temperature applications. In case of low-temperature applications (not lower than 100 °C), ferritic steel castings containing 3.5% nickel can be used (A-352 LC3). For temperatures until 200 °C, A-276-Type 304 castings are used.

Marine applications may demand castings made from aluminum bronze (B-148 – Alloy C 95 800). Copper bronze casting grade adopted is B 150-Alloy 63 200. Caustic Acid solutions and other corrosive liquids may demand special materials. For example, Sulphuric acid (concentration 67% and at a temperature of 60–70 °C) needs hi-silicon cast iron (15% Si).

During the casting process, it is important to keep the liquid contact surfaces of the impeller as smooth as possible. Thus, the composition of the core sand mixture and the finish of the core play an important part in the casting process. Largely, the relative smoothness of the liquid path determines the efficiency of the pump.

In a closed impeller design, the contact surface area of the metal with the liquid is higher which results in high friction losses. When the impeller's diameter is large, the problem becomes more acute and so there is a higher demand for smoother surface. Friction losses are related to 5th power of the diameter.

Subsequent to the casting and surface finishing operations, the impellers are dynamically balanced. The limits of residual unbalance are generally specified in ISO 1940, or even in API, which has a stricter limit. The balance of impellers alone is insufficient. Once the pump rotor components are ready, these should be mounted assembly wise on the balancing machine and balanced to stated limits.

### 2.1.1 Construction of impellers

There are three types of construction seen in an impeller. These are based on the presence or absence of the impeller covers and shrouds.

The three types (Figure 2.1) are:

1. Closed
2. Semi-open
3. Open.

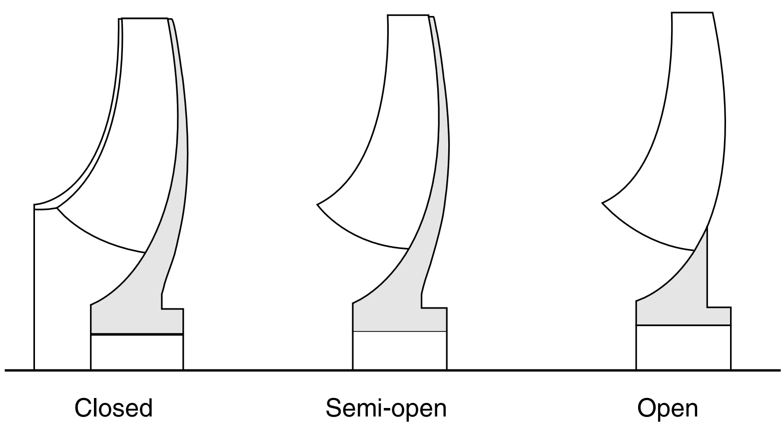



**Figure 2.1**
*Types of impellers*

## Closed impellers

The closed impeller consists of radial vanes (typically 3–7 in number), which are enclosed from both sides by two discs termed 'shrouds'. These have a wear ring on the suction eye and may or may not have one on the back shroud. Impellers that do not have a wear ring at the back typically have back vanes. Pumps with closed type impellers and wear rings on both sides have a higher efficiency.

## Semi-open impellers

The semi-open type impellers are more efficient due to the elimination of disk friction from the front shroud and are preferred when the liquid used may contain suspended particles or fibers. The axial thrust generated in semi-open impellers is usually higher than closed impellers.

## Open impellers

There are three types of back shroud configurations. The first one is a fully scalloped open impeller as shown in Figure 2.2.

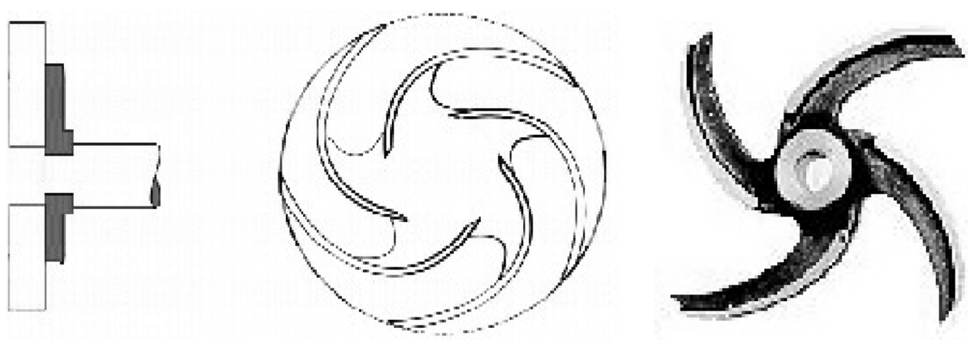


**Figure 2.2**
*Fully scalloped open impeller*

The back shroud is almost taken out and thus the axial thrust caused by the hydraulic pressure is almost eliminated.

The second type is known as the partially scalloped open type of impeller as shown in Figure 2.3. It experiences a greater axial thrust than the fully scalloped open impeller. However, this has higher efficiency and head characteristics.

The third type is known as the fully back shroud open impeller (Figure 2.4) where there is an open impeller with a full back shroud. It normally has almost 5% higher efficiency than a fully scalloped impeller, though it has diminished head generation capabilities.

The fully shrouded open impellers experience the maximum axial thrust among the open impeller types. To reduce this effect, back vanes are provided to relieve the hydraulic pressure that generates the axial thrust.

The vortex or non-clog impellers (Figure 2.5) are the fully shrouded open type of impellers. These are used in applications where the suspended solid's size maybe large or the solid's maybe of crystals and fibers type. The vortex impeller does not impart energy directly to the liquid. Instead it creates a whirlpool, best described as a vortex. The vortex in turn imparts energy to the liquid or pumpage. The location of the impeller is usually above the volute, so it experiences hardly any radial forces. This allows extended operation of the pump even at closed discharge conditions.

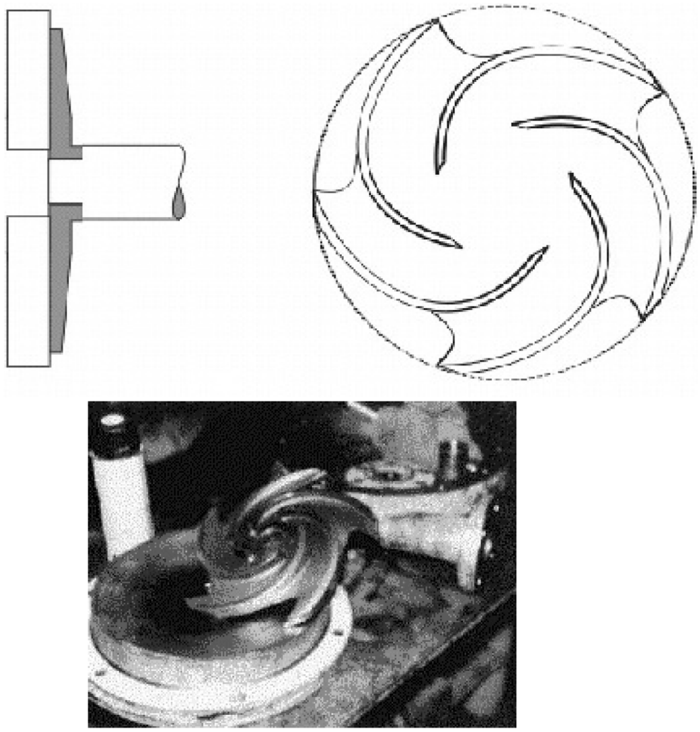


**Figure 2.3**
*Partially scalloped open impeller*

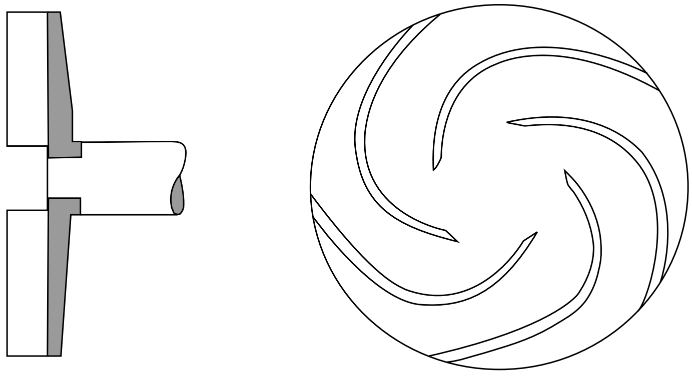


**Figure 2.4**
*Fully back shroud open impeller*

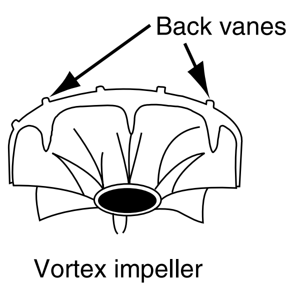



**Figure 2.5**
*Vortex impeller*

Some of the other non-clogging designs of impellers in the closed and semi-open types are shown in Figures 2.6 and 2.7.

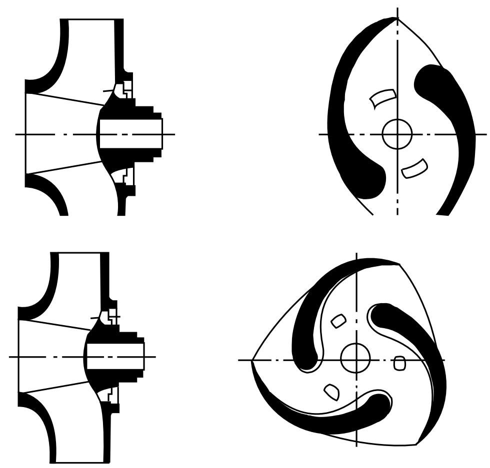


**Figure 2.6**
*2 & 3 Passage closed non-clog impellers*

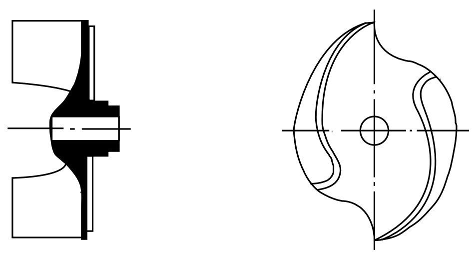


**Figure 2.7**
*Semi-open 2-passage non-clog S-shaped impeller*

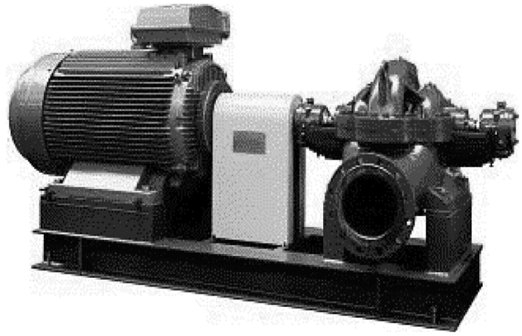
In general, most of the open impellers are of the partially scalloped and fully shroud types. Fully open impellers are rarely used because of its lower efficiency and the bending load on the vanes.

## 2.1.2 Impeller suction

In general, an impeller has one eye or a single opening through which liquid suction occurs. Such impellers are called as single-suction impellers. Pumps with a single-suction impeller (impeller having suction cavity on one side only) are of a simple design but the impeller is subjected to higher axial thrust imbalance due to the flow on one side of the impeller only.

In certain pumps, the flow rate is quite high. This can be managed by having one impeller with two suction eyes. Pumps with double-suction impeller (impeller having suction cavities on both sides) has lower NPSH-r than single-suction impeller. Such a pump is considered hydraulically balanced but is susceptible to an uneven flow on both sides if the suction piping is improper.

Generally, flows that are more than 550 $m^3/h$ (or 153 l/s) may necessitate a double suction impeller (Figure 2.8).



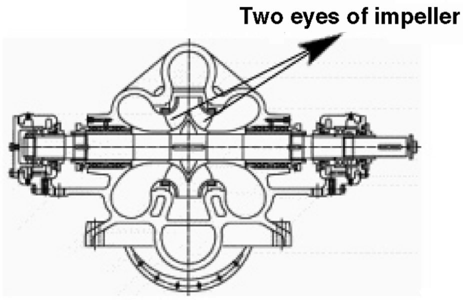



**Figure 2.8**
*Pump with double suction impeller*

## 2.1.3 Flow outlet from impeller

The flow direction of the liquid at the outlet of the impeller can be:

- Radial (perpendicular to inlet flow direction)
- Mixed
- Axial (parallel to inlet flow direction).

The flow outlet is determined by an important parameter called as the specific speed of the pump. As the specific speed of a pump design increases, it becomes necessary to change the construction of the impeller from a radial type to an axial type (Figure 2.9, and Figure 2.10 for mixed flow type). Generally, it can be said that for low specific speeds (low flows and high heads) radial impellers are used whereas for high specific speeds (high flows and low heads) axial (propeller) impellers are used (refer Figure 2.11).

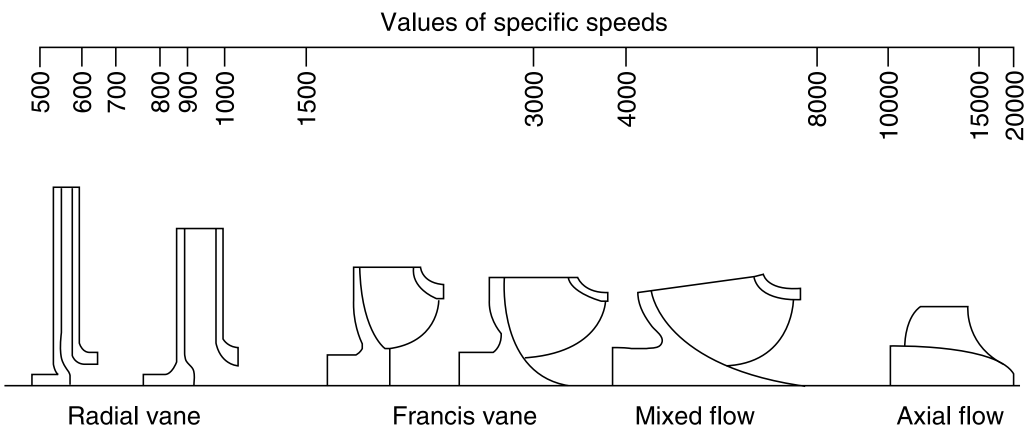



**Figure 2.9**
*Shapes of impellers according to their specific speeds*

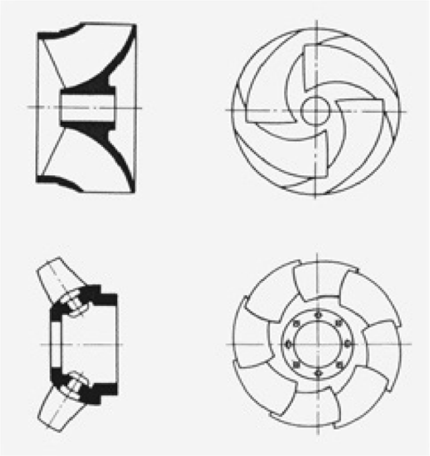


**Figure 2.10**
*Mixed flow impeller and propeller vaned mixed flow type impeller*

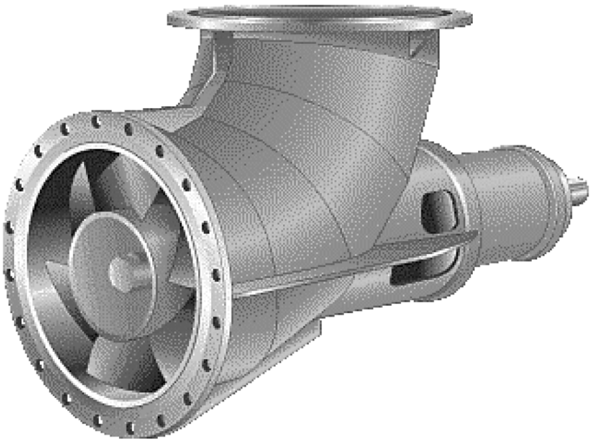


**Figure 2.11**
*Pump with axial flow impeller*

## 2.2 Pump casings

At the impeller outlet, the velocity of the liquid can be as high as 30–40 m/s. This velocity has to be reduced within a range of 3–7 m/s in the discharge pipe.

Velocity reduction is carried out in the pump casing by recuperators. The kinetic energy in the liquid at the outlet is converted to pressure energy by the recuperators.

Here, energy conversion has to be undertaken with a minimal loss to have an insignificant effect on pump efficiency.

Some of the recuperators are:

- Vaneless guide ring
- Concentric casing
- Volute casing
- Diffuser ring vanes
- Diagonal diffuser vanes
- Axial diffuser vanes.

### 2.2.1 Vaneless guide ring

A vaneless guide ring consists of two smooth discs (Figure 2.12). The distance between the two guide rings is either constant or is increased toward the outlet.

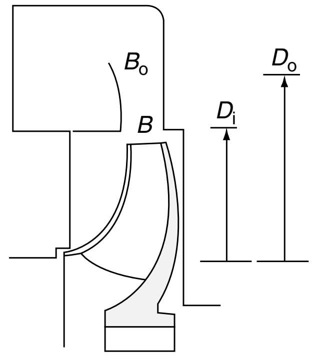



**Figure 2.12**
*Vaneless guide ring*

It follows that the conversion of kinetic energy of the liquid to pressure energy is entirely proportional to the ratio of the outlet diameter ($D_o$) of the ring to the inlet diameter ($D_i$).

The breadth of the ring has little role in the generation of liquid head, though it is observed that rings of constant breadth are more efficient than those with higher breadths at the outlet diameter ($B_o$).

Due to the above, the vaneless guide ring is used in pumps where liquid velocities are lower. It is thus found in pumps developing low heads. For larger heads, the outlet diameter of the ring would become larger and this maybe unpractical.

Vaneless guide rings are usually used in mixed flow impeller pumps of higher specific speeds along with an annular delivery passage of constant cross section. These may also be found in lower specific speed pumps handling liquid with solid matter.

### 2.2.2 Concentric casing

Concentric casings are usually found in single-stage centrifugal pumps and in the last stage of multistage pumps (Figure 2.13).

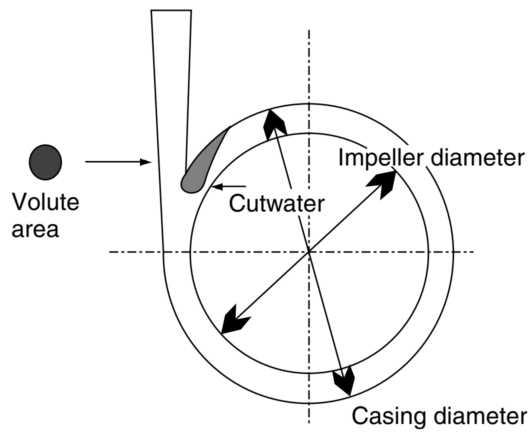



**Figure 2.13**
*Concentric casing pump*

In some of the earlier designs of a single-stage centrifugal pump for larger heads, an annular delivery passage is used in conjunction with a *diffuser ring*. The liquid outlet is through a conical diffuser.

The ratio of the impeller diameter to the diameter of the casing is not less than 1.15 and not more than a ratio of 1.2.

The volute width is designed to accommodate the maximum width of the impeller. The capacity at the most efficient point of operation is controlled by the volute diameter ($d$).

To minimize the recirculation in the volute, a cutwater tongue is used. In addition, this helps in significantly reducing the radial loads on the shaft.

In pumps with a specific speed of less than 600 (US-gpm, feet, rpm), the concentric casing provides higher efficiency than a conventional volute casing. Above the specific speed, $N_s$ of 600, the efficiency progressively drops.

The concentric casings are used:

- For less flow and higher head; low specific speeds $N_s$ is in the range of 500–600
- Where the pump casing has to accommodate several impeller sizes
- Where pump has to use a fabricated casing
- Where volute passage has to be machined from a casting
- Where foundry limitations result in higher impeller width.

### 2.2.3 Volute casing

Volute casings when manufactured with smooth surfaces offer insignificant hydraulic losses. In pumps with volute casings, it is possible to trim down impeller vanes and shrouds with minimal effect on efficiency.

In volute casings, the kinetic energy is converted into pressure only in the diffusion chamber immediately after the volute throat. The divergence angle is between 7° and 13°.

The volutes encountered can be of various cross-sections and these are shown in Figure 2.14.

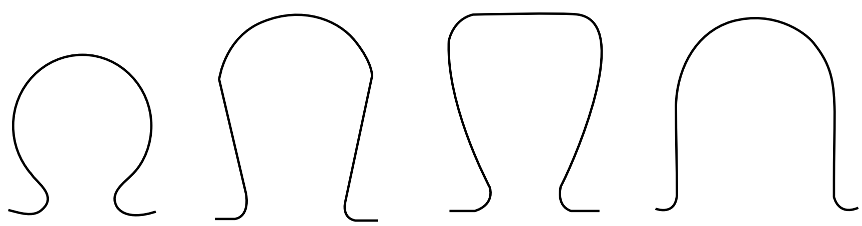


**Figure 2.14**
*Different volute cross-section shapes*

The first two profiles are of circular cross-section; the third is called as the trapezoidal cross-section, which is typically found in single-stage pumps. The last profile is the rectangular cross-section.

The rectangular section is used in small single-stage pumps and in multistage pumps. It is economical to manufacture due to its low pattern cost and production time. The hydraulic losses are minimal in the specific speed range of less than 1100.

Volute casings are manufactured in various designs and these are:

- Single volute casing
- Double volute casing.

## Single volute casing

Single volute designs are the most commonly found designs and those designed on the basis of constant velocity are the most efficient among all types. They are easy to cast and less expensive to manufacture.

In a single volute casing, the pressure distribution is balanced only at the Best Efficiency Point (BEP) of the pump. At other operating points, this leads to a residual radial load on the shaft, which is maximum at shut-off conditions and almost zero at the BEP.

At low flow rates, the pressure distribution is such that the surfaces of the impeller closest to the discharge are acted upon by high pressures. Those on the other side of the cutwater are acted upon by comparatively low pressures (Figure 2.15).

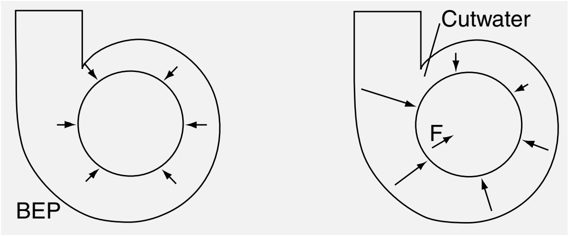



**Figure 2.15**
*Forces as generated in a single volute*

The resulting unbalanced forces can be assumed to be acting at a point 240° from the cutwater and acting in a direction which points to the center of the impeller.

Theoretically, these casings can be used over the entire range of specific speed pumps; however, these are used mainly on low capacity, low specific speed pumps. They can also be used in pumps handling slurries and solids.

## Double volute casing

A double volute casing design is actually two single volute designs combined in an opposed arrangement (Figure 2.16). The total throat area of the two volutes is identical to that which would be used on a comparable single volute design.

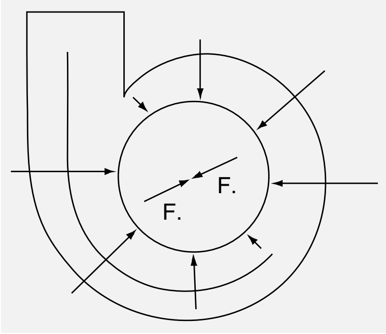



**Figure 2.16**
*Balance of forces in double volute*

Single volute designs inherently generate a radial load on the shaft. The double volute designs limit this radial force to a greater extent.

In this design, the volute is symmetrical about its centerline; however, the two passages carrying the liquid to the discharge flange are not symmetrical. As a result, the pressure forces around the impeller periphery do not cancel and this leads to some radial force.

The hydraulic performance of the double volute is on a par with the single volute design. At the BEP, the efficiency is marginally lower but is higher at operating points; lower and higher than BEP. Thus, for flows over the entire range, the double volute design is preferred.

Therefore, flow rate is the basic criterion that determines the selection of one design over another. For flows under 125 $m^3/h$, double volute designs are not used since it becomes difficult to manufacture and clean them in smaller casing. In larger pumps, double volutes are invariably used.

### 2.2.4 Vaned diffuser ring

The vaned diffuser ring has a series of symmetrically placed vanes forming gradually widening passages (Figure 2.17). This ring comprises of a series of vanes set around the impeller. The flow from the vaned diffuser is collected in a volute or circular casing and is discharged through the discharge pipe.

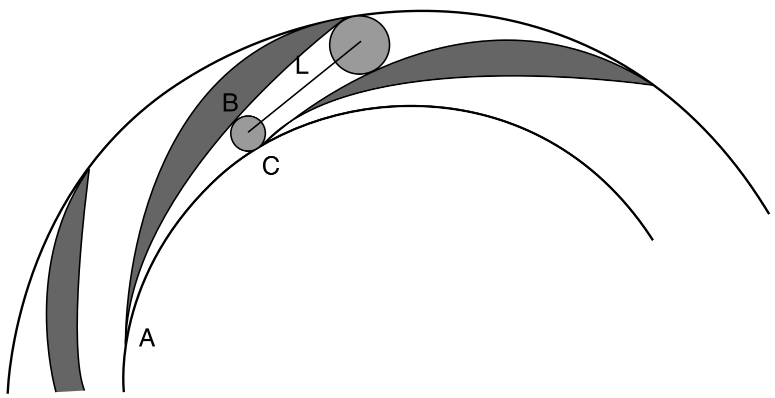



**Figure 2.17**
*Vaned diffuser*

In these passages, the velocity head is converted to pressure energy. The distance BC shown in Figure 2.17 is called as the throat.

The design of the vaned diffuser is similar to the volute except that there are many throats in a vaned diffuser compared to just one continuous expanding section in the volute.

From the throat onward, the area of the vane channel increases progressively so that, further, a slight increase in pressure takes place. The centerline of the vane channel after the throat maybe straight or curved. The straight diffusing channel is slightly more efficient but results in a larger casing.

The vane surface from the vane inlet to the outlet can be shaped like a volute but even a circular arc works fine.

The number of diffuser vanes is usually one more than the impeller vanes, as it is found that the number of diffuser vanes should not be much larger than the number of impeller vanes.

With just one vane more than the impeller, it insures that one impeller passage does not extend over several diffuser passages.

### 2.2.5 Diagonal diffuser vanes

Diagonal diffuser vanes are recuperators for the mixed-flow impeller pumps.

The functions of the diagonal diffuser vanes are:

- To change the direction of flow of the liquid leaving the impeller and direct it along the axis of the pump
- To reduce the velocity of liquid and convert it to pressure.

The vanes are disposed in the axial direction forming channels with no sudden changes in cross-section. They make it possible to use impellers of different diameters and breadths so as to extend the range of application of the given model of diffuser.

As the specific speed increases, the profiles of impellers and diffusers change and approximate to the shapes of impellers and diffusers of propeller pumps.

### 2.2.6 Axial diffuser vanes

Axial diffuser vanes are vanes placed behind the impeller of an axial flow pump (Figure 2.18). The functions of these vanes are similar to those of a mixed-flow pump.

The vanes usually number 5–8. The lower number is found in pumps with a lower specific speed (diffuser type-1).

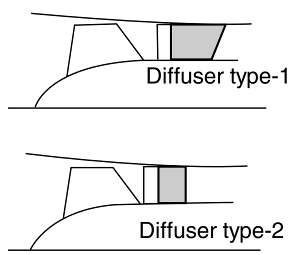



**Figure 2.18**
*Axial flow pumps with diffuser behind the propeller*

The efficiency is influenced to a certain extent by the shape of the diffuser passage. This depends on the number of vanes and their axial length and the distance between the impeller blades and the diffuser vanes.

Shorter and higher number of vanes (diffuser type-2) for the same flow and head give better efficiency.

When specific speeds are higher, these vanes are superfluous and a simple conical diffuser is constructed in their place.

## 2.3 Wearing rings

The impeller is a rotating component and it is housed within the pump casing. To prevent frictional contact, a gap between these two parts is essential.

So there exists a gap between the periphery of an impeller intake and the pump casing. In addition, there is a pressure difference between them, which results in the recirculation of the pumped liquid. This leakage reduces the efficiency of the pump.

The other advantages of lower clearance is that reduced leakage prevents erosion due to suction recirculation and also provides a much better rotor dynamic stability to the pump. As a result, the vibration of the pump operates with lesser vibrations.

Thus, it is essential to keep this gap or clearance at an optimum value. When this clearance is kept at a lower value, the efficiency improves but there is always a risk of contact of the impeller with the casing.

Such a frictional contact may render the impeller or the casing useless which would be a loss since these are expensive parts. Therefore, in the areas of the impeller intake, metallic rings are fitted on the impeller eye as well as on the pump casing.

Accordingly, the wearing ring on the impeller is called as impeller wearing rings and the one fitted on the casing is called as the case wearing ring (Figure 2.19).

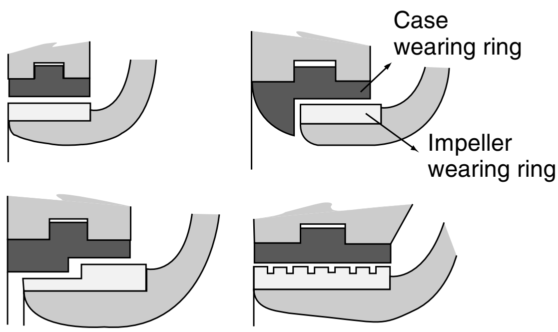



**Figure 2.19**
*Wearing rings of different types*

The cross-section of wearing rings shown in Figure 2.20 is fitted with an impeller eye and is called as the front wearing ring. However, in some cases, wearing rings are installed even at the back shroud of the impeller.

Usually, these are required when impellers are provided with balancing holes in order to minimize the axial thrust coming onto the pump impeller and consequently onto the bearings. The arrangement of the wearing rings on the back of the impeller is shown in Figure 2.20.

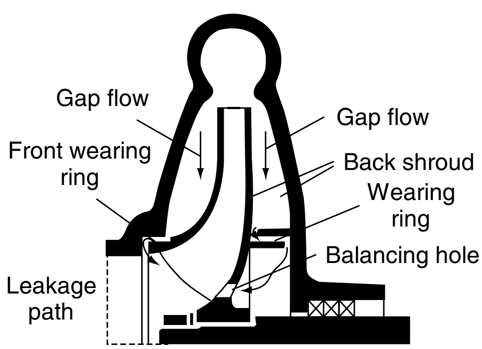



**Figure 2.20**
*Locations of wearing rings*

The material of the wearing rings is selected to prevent seizure on frictional contact. As a result, materials like SS-316 which have galling tendencies are not considered for this application.

The other materials considered favorably are:

- Austenitic Gray Iron Castings – ASTM A-436, Type-1
- Austenitic Ductile Iron Castings – ASTM A-439, Type-D2
- 12% Chrome Steels – AISI 420 (hardenable)
- 18 Cr – 8 Ni Steel Castings – AISI 304
- Copper Alloy Sand Castings (Bronze) – B-584, Alloy C 90 500
- Aluminum Bronze Sand Castings – B-148, Alloy C 95 800
- Monel – K 500
- Nickel 200.

The hardness range of the case wearing ring is in the region of 225–275 BHN, whereas the corresponding impeller wearing ring is kept harder by about 50–100 BHN. The range of hardness varies from 325 to 375 BHN.

API 610 standard for centrifugal pumps provides guidelines on the minimum recommended wearing ring clearances for metallic wearing rings. However, these clearances have to be in line with the pumping temperatures, thermal expansion, and galling tendencies of the ring material and the efficiency of the pump.

For materials that have galling tendencies and pumps operating at temperatures above 260 °C diameters are provided with an additional clearance of 5 mils (0.127 mm) over and above those recommended in Table 2.1.

For 26 in. and above, the diametrical clearance is recommended to be 0.037 in. plus 0.001 in. clearance for every additional inch of impeller diameter.

For example, a 30 in. impeller wearing ring diameter will have a minimum recommended clearance of 0.037 in. + 0.004 in. = 0.041 in.

API is also quite particular in the way the rings need to be fitted to the impellers. API 610 does not recommend tack welding of rings to impellers. They should be pressed with locking pins or threaded dowels, in the radial or axial direction.

Thermoplastic composite materials are also now being considered as ideal wearing ring materials. They can be applied to the stationary wear part or with the mating component remaining in steel. The use of thermoplastic composite material provides for greater hardness differential between wear parts, the thermoplastic serving as a sacrificial component.

Thermoplastics too have their limitations, however in some cases, they provide the best alternative.

Thermoplastic composite materials are non-galling and have a lower coefficient of friction. They demonstrate excellent wear resistance in clean liquids. Some of these plastics contain reinforced carbon fibers, which greatly enhance the mechanical properties of these plastics. As a result, they can be a direct replacement of the metal wearing rings.

Due to the reduced friction and low galling tendencies, it is possible to almost have half of the clearances that would be considered as optimum with metal wearing rings.

This possibility allows improving pump efficiency especially in low specific speed pumps.

However, the limitations of such materials are that:

- Maximum life is obtained in clean fluids.
- They do not have a wide compatibility with various chemicals.

| Diameter of Rotating Member at Clearance Inches | | Minimum Diametrical Clearance | |
|---|---|---|---|
| From | To | Inches | mm |
| <2 | | 0.01 | 0.254 |
| 2.000 | 2.499 | 0.011 | 0.28 |
| 2.500 | 2.999 | 0.012 | 0.30 |
| 3.000 | 3.499 | 0.014 | 0.36 |
| 3.500 | 3.999 | 0.016 | 0.41 |
| 4.000 | 4.499 | 0.016 | 0.41 |
| 4.500 | 4.999 | 0.016 | 0.41 |
| 5.000 | 5.999 | 0.017 | 0.43 |
| 6.000 | 6.999 | 0.018 | 0.46 |
| 7.000 | 7.999 | 0.019 | 0.48 |
| 8.000 | 8.999 | 0.02 | 0.51 |
| 9.000 | 9.999 | 0.021 | 0.53 |
| 10.000 | 10.999 | 0.022 | 0.56 |
| 11.000 | 11.999 | 0.023 | 0.58 |
| 12.000 | 12.999 | 0.024 | 0.61 |
| 13.000 | 13.999 | 0.025 | 0.64 |
| 14.000 | 14.999 | 0.026 | 0.66 |
| 15.000 | 15.999 | 0.027 | 0.69 |
| 16.000 | 16.999 | 0.028 | 0.71 |
| 17.000 | 17.999 | 0.029 | 0.74 |
| 18.000 | 18.999 | 0.03 | 0.76 |
| 19.000 | 19.999 | 0.031 | 0.79 |
| 20.000 | 20.999 | 0.032 | 0.81 |
| 21.000 | 21.999 | 0.033 | 0.84 |
| 22.000 | 22.999 | 0.034 | 0.86 |
| 23.000 | 23.999 | 0.035 | 0.89 |
| 24.000 | 24.999 | 0.036 | 0.91 |
| 25.000 | 25.999 | 0.037 | 0.94 |

**Table 2.1**
*Minimum diametrical clearance*

## 2.4 Shaft

The pump rotor assembly comprises of the shaft, impeller, sleeves, seals (rotating element), bearings or bearing surfaces, and coupling halves. The shaft, however, is the key element of the rotor.

The prime mover drives the impeller and displaces the fluid in the impeller and pump casing through the shaft.

The pump shaft is a stressed member for during operation it can be in tension, compression, bending, and torsion. As these loads are cyclic in nature, the shaft failure is likely due to fatigue.

The shaft design depends on the evaluations of either the torsion shear stress at the smallest diameter of the shaft or a comprehensive fatigue evaluation taking into consideration the combined loads, the number of cycles, and the stress concentration factors. The design at all times involves sophisticated finite-element computer evaluations.

The shaft design is limited not only to the stress evaluation but is also dependent on other factors such as:

- Shaft deflection
- Key stresses
- Mounted components
- Critical speeds (rotordynamics).

The most common pump shaft material is plain carbon steel, typically BS-970-En 8.

Higher grades include BS-970-En19 or AISI 4140, ASTMA-322, Grade-4140 (quenched and tempered).

Austenitic steel shafts may also be used of grade ASTM A-276, Type 316 and AISI 304. Some applications like sour water with pH less than 7, drain water or slightly acidic non-aerated liquids, and hydrocarbons containing corrosive aqueous phase may demand shafts made from aluminum bronze material. The recommended grade is B-150-Alloy C 63 200. Special applications may call for Monel or even Hastalloy C shafts.

The mechanical seals or gland packing, in contact with the shaft, can cause excessive wear due to frictional contact or fretting corrosion. As a sacrificial component, shaft sleeves are used. These are fitted closely onto the shaft; and seals and gland packing are exposed to the sleeve rather than the shaft. It is far less expensive to replace a sleeve than the complete shaft.

The material of construction of the pump sleeves is similar to that of the shaft but the standardization favors the use of SS-316. The portion of the sleeve that is exposed to the secondary seal of the mechanical seals such as an O-Ring or a Teflon wedge is hard coated. The plasma sprayed, hard coating can be of Chrome-oxide, tungsten carbide, or alumina. This offers hardness around 70–72 Rc. The surface is then provided with a ground finish.

## 2.5 Stuffing boxes

The stuffing box is a chamber or a housing that serves to seal the shaft where it passes through the pump casing (Figure 2.21).

In a stuffing box, 4–6 suitable packing rings are placed and a gland (end plate) for squeezing and pressing them down the shaft.

The narrow passage, between the shaft and the packing housed in the stuffing box, provides a restrictive path to the liquid, which is at a high pressure within the pump casing.

The restrictive path causes a pressure drop, prevents leakage resulting in considerable friction between the shaft and the packing, and causes the former to heat up. It is thus good practice to tighten the gland just enough to allow for a minimal leak through the packing. This slight leakage of the liquid acts as a lubricant as well as a coolant. Obviously, this cannot be allowed for hazardous and toxic liquids, but then gland packings are also not used in such applications.

When pumps are handling dirty or high-pressure liquid, lantern rings are used. These are rings with holes drilled along its circumference.

A lantern ring substitutes one of the packing rings in the stuffing box and is situated at the pump end or midway between the packings.

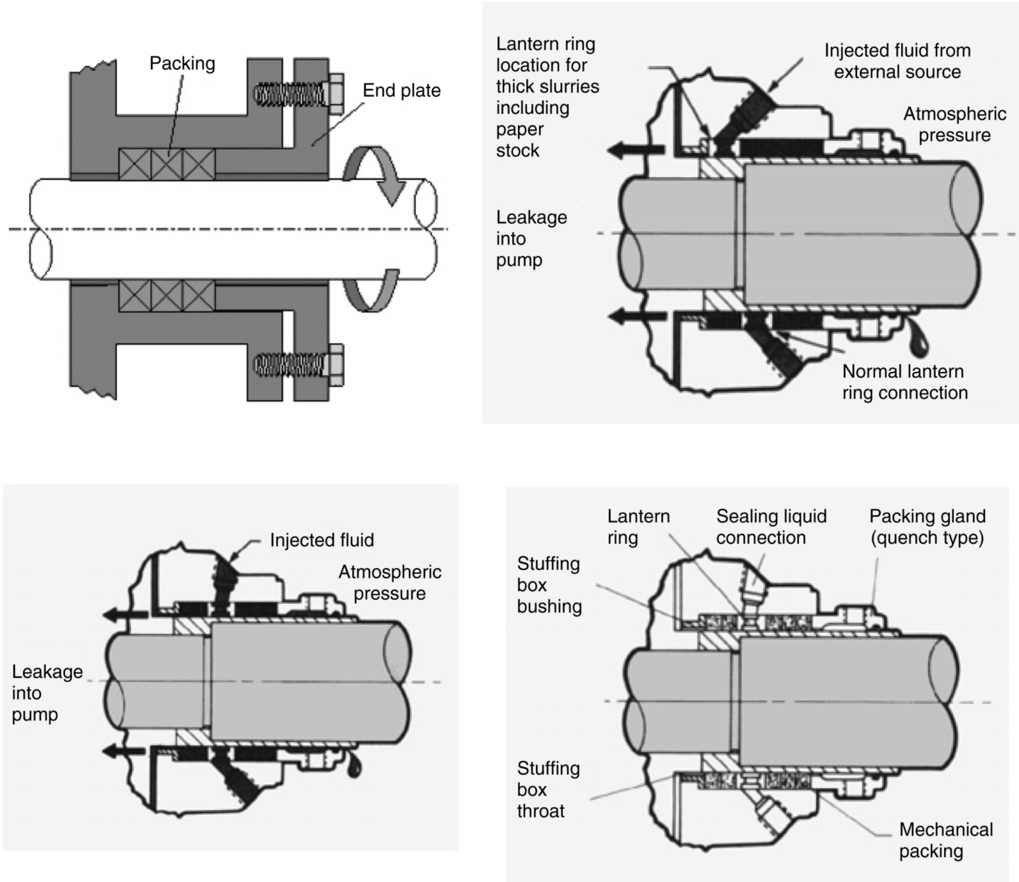



**Figure 2.21**
*Stuffing box*

In applications where the discharge pressure of the pump is higher, a restrictive bush is placed at the throat of the stuffing box.

When the liquid being handled is at a higher temperature (above 120 °C), the stuffing box has an integrally cast water jacket housing. This allows for water circulation and keeps the packings at a lower temperature.

When toxic or corrosive liquids are handled, it is necessary to insure complete sealing of the stuffing box. Leakage of such liquids is a hazard to the plant personnel and can also be detrimental to the outer surface of the pump and foundation. It can also result in the loss of a valuable product.

This is achieved by first reducing the pressure in front of the packings. The reduction in pressure is brought about by having radial blades at the back shroud of an auxiliary impeller (see Figure 2.22). This auxiliary impeller is also called as a repeller.

As the repeller rotates with a shaft, it throws the liquid outwards thus reducing the pressure in front of the packings. The pressure generated by the repeller is dependent on the length of the blades and its clearance with the casing.

Another common design is to cast back vanes of the main impeller itself. The back vanes help reduce the pressure acting on the packing.

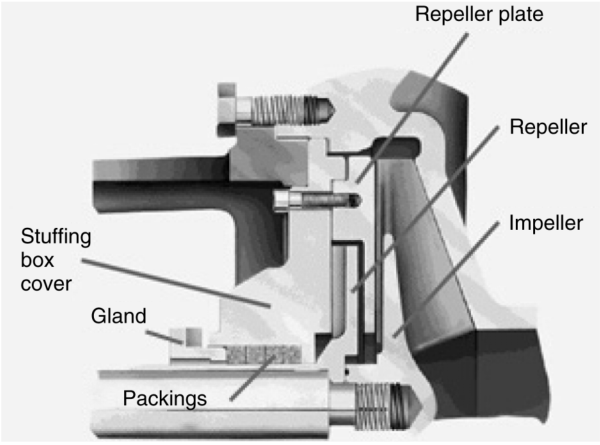



**Figure 2.22**
*Stuffing box for a corrosive liquid*

The magnitude of the work done due to the friction between the shaft and the packings is influenced by:

- Kind of packing quality
- Length of the gland
- Diameter of the shaft
- Speed of rotation
- Pressure acting on gland
- Volume of liquid passing through the packing.

In a properly operating stuffing box, the friction losses are usually of the order of 1% of the total pump power. This is independent of the size and kind of pump.

The present day packing used in pumps are predominantly made of PTFE (Teflon)/Graphite filaments. These are braided and formed into square shapes. They offer heat dissipation and low friction qualities. For example, see Figure 2.23 for a high-performance filament packing.

**Figure 2.23**
*A high-performance filament packing (EGK®** Filament Packing – Style 2070)*

Packing properties can be enhanced by including other special materials during braiding of the packings, to handle contaminants, acids, alkalis, temperature, speed, and other factors. For example, a material called as Aramid can enhance the mechanical properties of the packing. This helps in prevention of extrusion of packing and withstands slurries and abrasives in liquids.

## 2.6 Mechanical seals and seal housings

The stuffing boxes described above have many disadvantages and these include:

- A persistent leakage and loss of product if the shaft surface is not smooth.
- If the gland is too tightened, the shaft/sleeve gets hot and there can be rapid wear of the surface as shown in Figure 2.24.
- They require constant supervision.

**Figure 2.24**
*Wear on shaft/sleeve due to tight packing*

As a result, the use of gland packing is being phased out but is still used in noncritical and low-power applications. In most of the applications, mechanical seals are used. Most of the disadvantages of packing are eliminated by the use of mechanical seals.

From its origins in 1930s, the technology of mechanical seals continues to evolve at a rapid pace. This is, especially, in regard to the enhancement of the reliability of seals.

Until 1950s, packing in the stuffing box was a standard method of shaft sealing. As operating conditions became more demanding and pumps were used on a greater variety of fluids, mechanical seals were designed to handle these changing conditions.

Mechanical seals comprise of two perfectly lapped mating faces. One face is stationary and the other is rotating. The leakage resistance in gland packing is along the axis of the shaft but in seals, it is orthogonal.

The seal faces cannot run mating with each other without any lubricant (Figure 2.25). This can lead to an early wear and seal damage results in leakage. Usually, the sealant fluid is injected in the seal housing at a specified pressure, which lubricates and cools the faces.

The fluid between the faces can escape into the atmosphere and this is called as fugitive emissions. In some applications, fugitive emissions are unacceptable and in such cases, multiple seal arrangements are used.

However, due to its precise design, mechanical seal demands careful attention to precision during pump assembly.

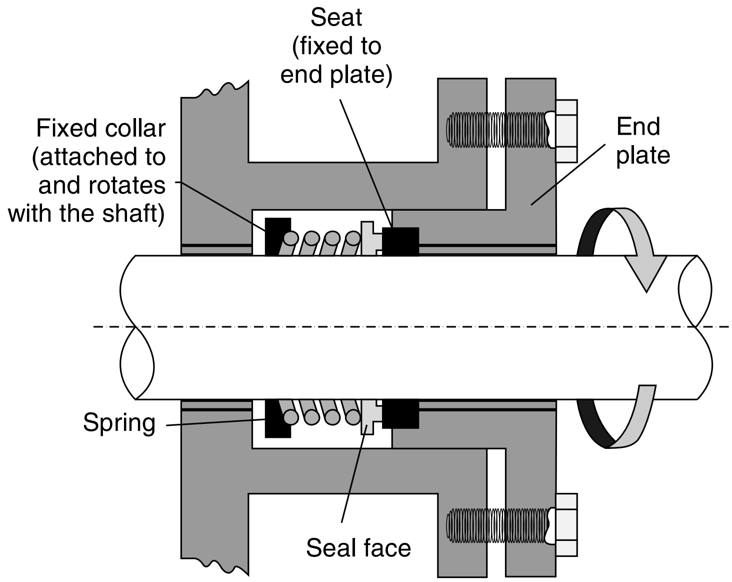



**Figure 2.25**
*Mechanical seal*

There are three points of sealing as shown in Figure 2.26, common to all mechanical seal installations:

1. At the mating surfaces of the primary and mating rings
2. Between the rotating component and the shaft or sleeve
3. Between the stationary component and the gland plate.

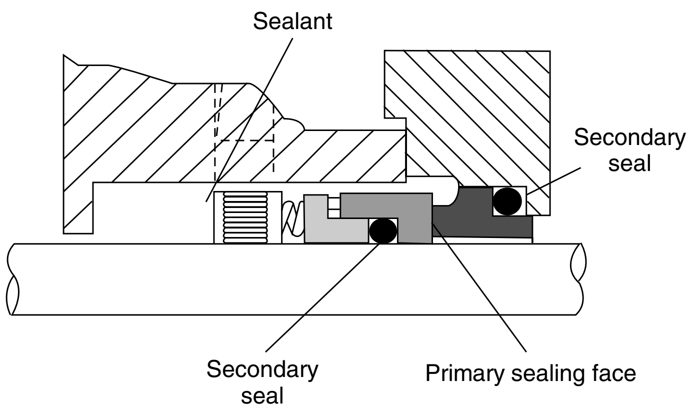



**Figure 2.26**
*Points of sealing in a mechanical seal (Pusher type unbalanced seal)*

When a seal is installed on a sleeve, there is an additional point of sealing between the shaft and sleeve. Certain mating ring designs may also require an additional seal between the gland plate and stuffing box.

Normally, the mating surfaces of the seal faces are made of dissimilar materials and held in contact with a spring. Preload from the spring pressure holds the primary and mating rings together during shutdown or when there is a lack of liquid pressure.

The secondary seal between the shaft and sleeve must be partially dynamic. As the seal faces wear, the primary ring must move slightly forward. Because of vibration from the machinery, shaft run out, and due to thermal expansion of the shaft to the pump casing, the secondary seal must move along the shaft. Flexibility in sealing is achieved from secondary seal forms such as an O-ring, wedge, V-ring, or bellows (see Figure 2.27 for bellow seal). Most seal designs fix the seal head to the sleeve or shaft and provide for a positive drive to the primary ring.

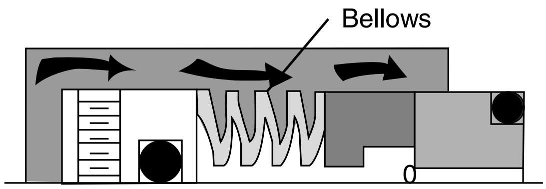



**Figure 2.27**
*Bellow seal*

Although mechanical seals may differ in various physical aspects, they are fundamentally the same in principle. The wide variation in design is the result of the many methods used to provide flexibility, ease of installation, and economy.

A seal arrangement is used to describe the design of a particular seal system and the number of seals used on a pump.

The most common sealing arrangements may be defined as:

- Single seal installations
- Internally mounted
- Externally mounted
- Dual seal installation
- Tandem seals
- Double seals
- Externally pressurized
- Internally pressurized.

A single seal mounted inside the seal chamber represents at least 75% of all installations. It is the most economical sealing system available to the industry.

Just as gland packings are housed in stuffing boxes, the mechanical seals are housed in seal housings. Research has indicated that providing an enlarged bore can provide distinctive advantages depending on the applications. Some of the seal housings suggested are given in Figure 2.28.

Studies demonstrate that the fluid flow within the enlarged chamber is increased. This aids in the removal of seal generated heat.

Optimum selection of enlarged chambers can help deal with:

- Gases in seal housing under start stop conditions
- Light hydrocarbons with low boiling points
- Liquids with solid particles.

These designs help in improving the reliability of the seals.

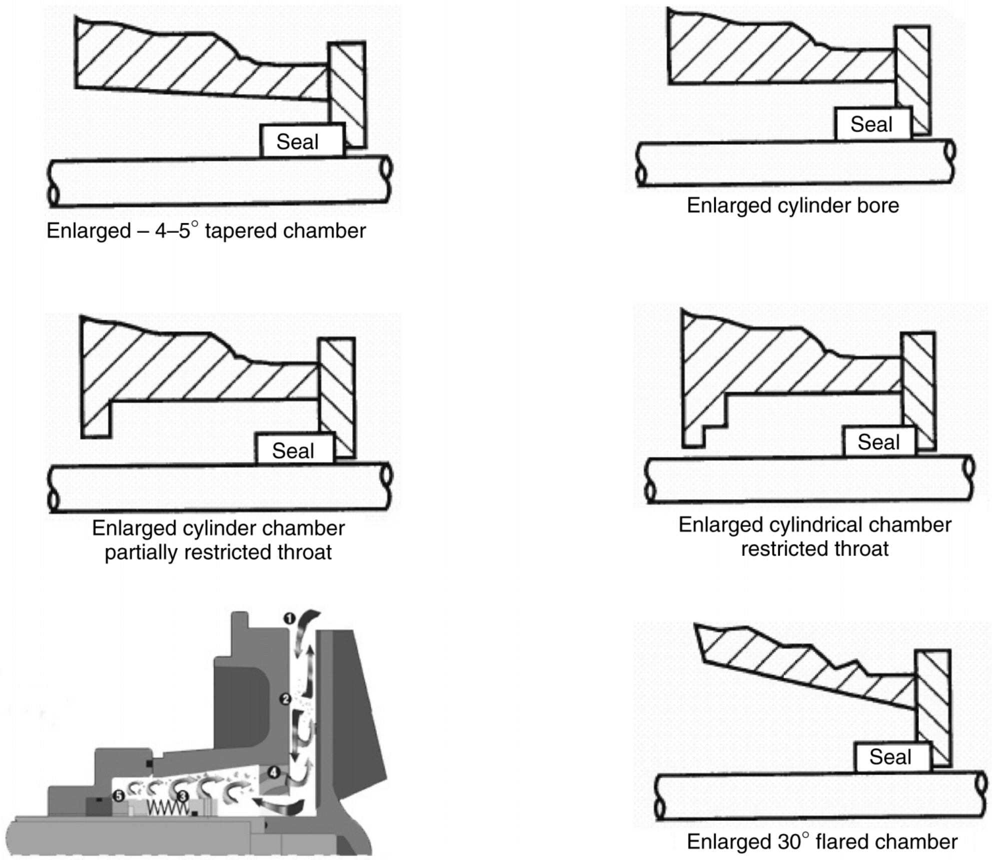



**Figure 2.28**
*Types of seal housings*

# 2.7 Bearing housing/bearing isolators

## 2.7.1 Cantilevers or overhung impeller pumps

Overhung impeller pumps usually employ anti-friction bearings only. In a typical bearing housing arrangement, the radial ball or cylindrical roller bearing is located adjacent to the impeller or inboard position. It is arranged to take only radial loads.

The thrust bearing is located closest to the coupling and usually consists of a duplex pair of angular contact bearings or double row angular contact bearings.

Typically, when a ball bearing is mounted on the inboard side, the coupling side or the outboard side is provided with a duplex angular contact bearing in a face-to-face arrangement or a double row angular contact bearing.

With a cylindrical roller bearing on the inboard side, the outboard bearings are mounted back-to-back so that the axial thrust load can be carried in either direction. This duplex bearing pair carries both the unbalanced axial thrust loading and the radial load.

## 2.7.2 In-between bearing or fully supported shaft pumps

In-between bearing pumps, the ball radial bearing and the ball thrust bearing combination have individual bearing housings.

The radial bearing is normally located at the coupling end of the pump. The ball thrust bearing is located at the outboard pump end.

The thrust bearing must be secured axially on the shaft to transmit the axial thrust load to the bearing housing through the bearing.

The bearing is usually located against a shoulder on the shaft and locked in place by a bearing nut. This means that the shaft diameter under the thrust bearing is less than the shaft diameter under the radial bearing.

Thus, by mounting the radial bearing on the inboard (or coupling) end of the pump shaft, a larger shaft diameter is available to transmit pump torque from the coupling to the impeller.

The thrust bearing, on the other hand, is locked axially in the thrust bearing housing; the radial bearing is axially loose in its housing to allow for axial thermal growth.

A popular combination for in-between bearing double suction pumps consists of journal type radial bearings and a ball thrust bearing.

In such an arrangement, all radial pump loads are handled by the journal radial bearing.

The ball thrust bearing is mounted in the thrust bearing housing such that the thrust bearing carries only axial loads. The housing around the ball thrust bearing is radially loose. A metallic strap is employed on the outer rings of the thrust bearing. This strap locks into the bearing housing to prevent rotation of the outer rings.

Such a bearing arrangement is useful in higher horsepower and higher speed applications where ball radial bearings would be impractical due to speed, load, and lubrication limitations.

Due to the location of the ball thrust bearing on the outboard end of the shaft, the shaft diameter under the ball thrust bearing can be relatively small since no torque is transmitted from this end of the shaft.

### 2.7.3 Vertical pumps

In vertical pumps, the unbalanced axial hydraulic forces as well as the static weight of the rotating element (i.e. pump shaft and impeller(s)) is taken up by the thrust bearing, which by design, may be located within the driver or normally at the head end within the pump casing.

These bearings could be ordinary ball bearings, angular contact bearings, split inner race angular contact bearings, and even the spherical roller thrust bearings in larger pumps like the vertical deep well pump.

Typically, these bearings are rated to handle at least thrice the maximum thrust load. This is due to number of varying factors in the determination of the thrust loads generated by such pumps. The most severe thrust loads are generated at the time of shutting down the pump and or from impact loading, as a result of water hammer.

Some of these are mentioned below:

- The calculation of pump thrust is not highly accurate.
- Pump thrust increases as internal clearances increase.
- The thrust load varies with the vertical position of the impellers with the casing(s).
- The thrust load varies with flow. (In some cases, it may even reverse direction.)

A reasonable margin should be provided between the driver thrust bearing rating and the maximum calculated pump thrust.

### 2.7.4 Bearing housing protection devices

There is a close relationship between the life of rolling element bearings and mechanical seals in pumps.

Liquid leakage from a mechanical seal may cause the bearings to fail, while a rolling element bearing in poor condition can reduce seal life. Only about 10% of rolling element bearings achieves their 3–5-year design life.

Rain, product leakage, debris, and wash-down water entering the bearing housing contaminate the bearing lubricant and have a catastrophic effect on bearing life.

A contamination level of only 0.002% water in the lubricating oil can reduce bearing life by as much as 48%. A level of 0.10% water will reduce bearing life by as much as 90%.

To improve the conditions inside a bearing housing, various types of end seals are used.

In almost every case, the normal operating life and quality of the end seal is not nearly as good as that of the rolling element bearings. Improving the quality of the end seals will increase the life of rolling element bearings.

## Felt and lip seals

One of the earliest, bearing housing isolators was the 'felt' (Figure 2.29). The bearing covers are provided with a groove in which a felt strip is cut and inserted. The felt acts as a barrier for oil and dust from the atmosphere.

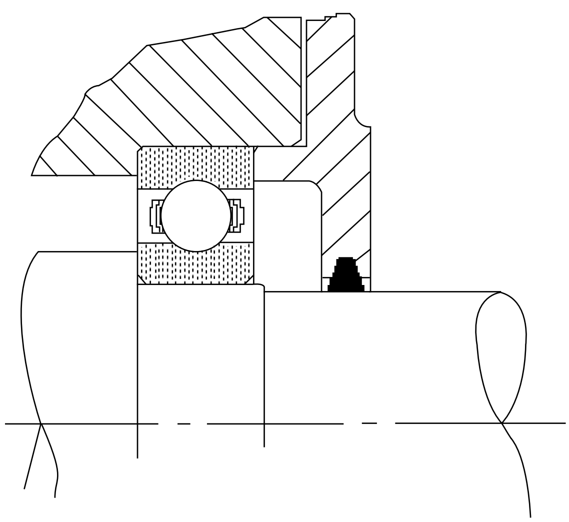


**Figure 2.29**
*Felt seal*

The lip or the oil seals (Figure 2.30) have low initial cost, availability, and are common. New lip seals provide protection in both static and dynamic modes. Their major disadvantage is short protection life due to wear of the elastomer.

Life expectancy of a common single lip seal can be as low as 3000 h, or 3–4 months. Thus, while a bearing is designed to last from 3 to 5 years of continuous operation, the lip seal will provide protection for only a few months.

The temperature limits of lip seals are –40 to 400 ºF (–42 to 203 ºC) for Viton.

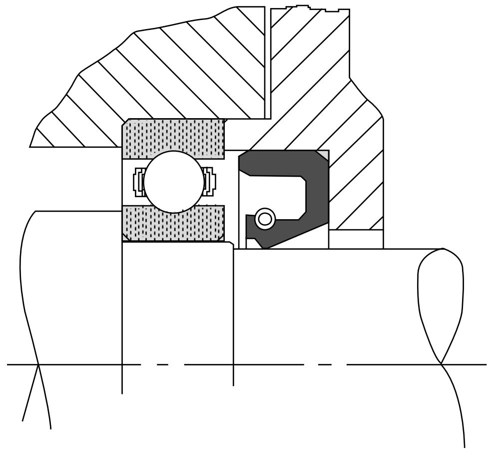


**Figure 2.30**
*Lip seal*

## Labyrinths

Labyrinths are devices that contain a tortuous path, making it difficult for contaminants to enter the bearing housing (Figure 2.31). Labyrinths are devices that contain a tortuous path, which in turn discourages and hence minimizes leakage of fluid without there being any physical contact between the Stationery and moving elements that make up the seal. Labyrinth seal design may vary hence selection must be based on its suitability for the application and purpose.

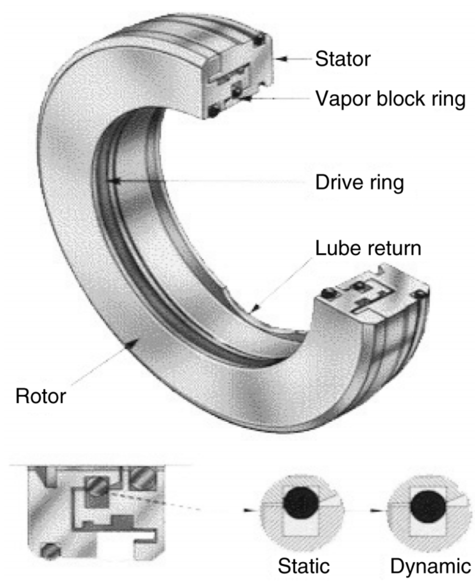



**Figure 2.31**
*A special type of labyrinth seal (Inpro seals)*

The advantages of labyrinths are their non-wearing and self-venting features. With no contacting parts to wear out, a labyrinth can be reused for a number of equipment rebuilds. Because the labyrinth provides an open, however difficult, path to the atmosphere, the bearing housing vent can be removed and the tapped hole can be plugged with a temperature gage.

The disadvantages of labyrinths include a higher initial cost than lip seals and the existence of an open path to the atmosphere, which can enable contamination of the lubricant by atmospheric condensate as the housing chamber 'breathes' during temperature fluctuations in humid environments. Also, they do not work as well in a static mode as in a dynamic, rotating mode.

The temperature limits of labyrinths are determined by the elastomer driving the rotor and holding the stator in place, the same as for the lip seal.

### Magnetic seals

Magnetic seals use a two-piece end face mechanical seal with optically flat seal faces held together by magnetic attraction (Figure 2.32).

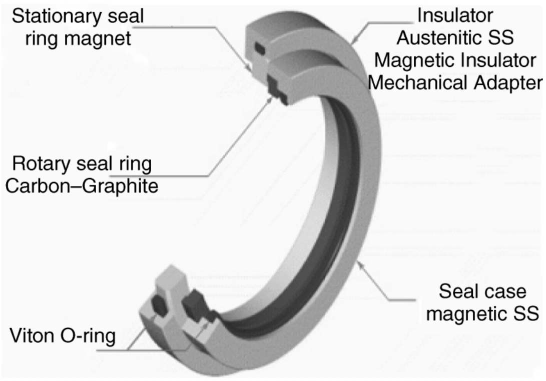



**Figure 2.32**
*Magnetic seal*

They have a design life equivalent to mechanical seals and rolling element bearings and can be repaired.

The major advantage of magnetic seals is the hermetic seal they provide for the bearing housing. Because of the positive seal, other arrangements must be made to allow for the 'breathing' that results from expansion and contraction of the air pocket above the lubricant during normal temperature changes.

Disadvantages of magnetic seals include higher initial cost and a shorter life than the almost infinite life of a labyrinth.

Magnetic seals are generally not recommended with dry sump, oil mist lubricated bearing housings, or grease-lubricated bearings. The upper operating temperature limit of magnetic seals is lower than that of labyrinth seals, in the range of 250 ºF (121 ºC).

## 2.8 Couplings

Couplings for pumps usually fall in the category of general-purpose couplings. General-purpose couplings are standardized and are less sophisticated in design. The cost of such coupling is also on the lower side. In addition, there are special purpose couplings that can be used on turbo machines and are covered by the API 671 specification.

In these couplings, the flexible element can be easily inspected and replaced. The alignment demands are not very stringent. The couplings fitted on pumps usually fall in any of the five types mentioned below.

These are:

1. Gear coupling
2. Grid coupling
3. Disk coupling
4. Elastomeric compression type
5. Elastomeric shear type.

### 2.8.1 Gear coupling

Gear couplings comprise of two hubs with external teeth that engage the internal teeth on a two or one piece flanged shroud or sleeve (Figure 2.33).

One hub with shroud is mounted on the pump shaft and the other on the shaft of the prime mover. The flanges are bolted after packing grease between the meshing gears.

To contain the grease within an enclosed space, the flange faces and the shroud are provided with suitable static seals (like O-rings).

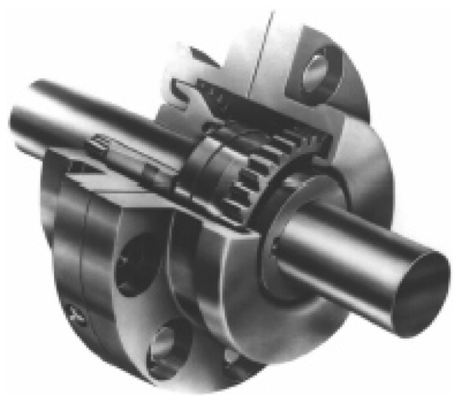


**Figure 2.33**
*Gear coupling*

Some of the couplings may have a spacer between the flanged shrouds. In case of any misalignment between the shafts, sliding occurs between the external gears on the hub and its corresponding internal gears on the shroud.

Gear couplings are usually deployed on pumps above a rating of 75 kW.

### 2.8.2 Grid coupling

The grid coupling in some ways is similar to the gear coupling. It also consists of two hubs mounted on the driver and driven shafts. The hubs are slotted and house a flexible grid member. Grease is applied to lubricate any sliding that may occur between the grid member and the slots of the hub.

A cover contains the lubrication. Grid couplings are usually not used in pumps with a power rating of 750 kW.

A grid coupling is shown in Figure 2.34.

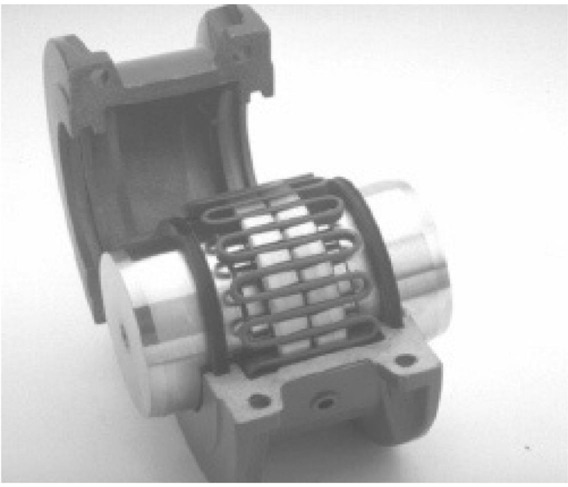


**Figure 2.34**
*Grid coupling*

### 2.8.3 Disk coupling

A disk coupling shown in Figure 2.35 comes under the category of Metallic Element Coupling. Diaphragm couplings used on turbo machines also belong to this category.

**Figure 2.35**
*Metallic disk coupling*

Metallic disk coupling comprises of two hubs mounted on the driver and driven shafts. A set of flexible shims or metallic element is placed between the spacer and the hub.

The torque is transmitted by simple tensile force between alternate driving and driven bolts on a common bolt circle diameter.

Such couplings are used on pumps with a power rating of over 75 kW.

### 2.8.4 Elastomeric shear type coupling

All elastomeric couplings are classified according to how their elastomeric elements transmit torque between driving and driven hubs. The elements could be either in compression or in shear.

In shear-type couplings, the driving and driven hubs operate in separate planes, while the driving hub pulls the driven hub through an elastomeric element suspended between them. Here, the element transmits and cushions the force between the hubs by being stretched between them (Figure 2.36).

**Figure 2.36**
*Elastomeric shear type coupling*

Among all couplings, this type can probably take the maximum amount of parallel misalignment.

These coupling are used in pumps below a rating of 75 kW.

### 2.8.5 Elastomeric compression type coupling

In jaw couplings, the element (called as a spider) is loaded in compression between the jaws of mating hubs.

These jaws operate in the same plane, with the driving hub jaws pushing toward the driven hub jaws. Legs of the elastomeric spider transmit and cushion the force between the driving and driven jaws by being compressed between them.

Compression type couplings offer some advantages over the shear type of coupling (Figure 2.37). These include:

- Higher load capacity
- Greater torsional stiffness
- More safety
- Easier installation.

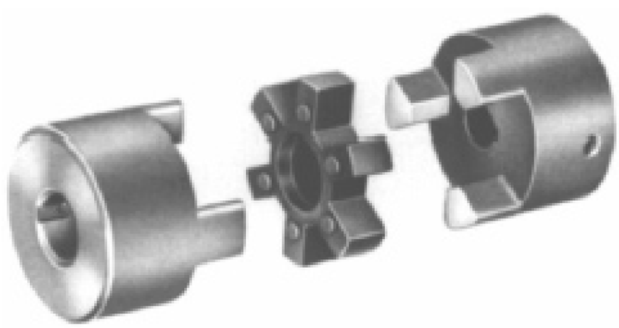


**Figure 2.37**
*Elastomeric compression type coupling*

The design of the coupling allows it transmit the torque even if the spider breaks. The driving jaws simply rotate until they contact the driven jaws directly, and the coupling continues to function, though it is accompanied by considerable noise and accelerated wear. In some cases, it can prevent an expensive downtime.

They typically accommodate angular shaft misalignment up to 1° and parallel misalignment up to 0.015 in.

The elastomeric material in the above two types of couplings is mostly NBR (Nitrile Butadiene Rubber), sometimes called Buna N. It is the most economical and widely used standard coupling element material. It resembles natural rubber in resilience and elasticity, and is resistant to oil, hydraulic fluid, and most chemicals.

The operating temperature ranges from –40 to +100 °C. With hardness of 80 Shore A, NBR provides the best damping capability among elastomeric elements.

Another material used is Urethane. It has 1.5 times the torque capacity of NBR with very good chemical and oil resistance. But has less damping capability (90 Shore A hardness) and narrower operating temperature range from –39 to +71 °C. Urethane spiders are good choices when the application calls for greater torque in a confined space, or for resistance to atmospheric effects such as ozone, sunlight, and hydrolysis in tropical conditions.

Hytrel (*registered trademark of E.I. DuPont de Nemours & Co*) is designed for high operating temperature range from –51 to +121 °C. It offers excellent resistance to oils and chemicals and can transmit 3 times the torque of standard NBR.

It also provides resistance to ozone, sunlight, and hydrolysis in tropical conditions. With hardness of 55 Shore D, however, Hytrel cuts angular misalignment ratings in half, and damping capacity is low.

The spiders come in the following four types:

1. Standard solid center spider
2. Open center type
3. *Snap wrap (with or without retainer ring)*: This flat-strip, open-end design connects the spider legs around the perimeter of the coupling rather than at the center. This allows for easy removal or installation without disturbing the alignment of either coupling hub. With no center connections, this design does not overlap into the bore, and therefore it allows shaft ends to extend at maximum bore diameter to a minimal distance between the shaft ends. This element is radially ‘wrapped’ around the jaws and needs to be held in place by either a ring or a collar. When retained by a ring, it has a maximum RPM limit of 1750. The collar configuration, on the other hand, achieves the same RPM rating as the standard coupling because the collar is attached to one hub.
4. *Load cushions (separate blocks)*: As small, separate blocks, these cushions can be installed easily and removed radially, which can be very helpful for maintenance in heavy-duty applications. In certain models of coupling, load cushions must be held in place by a collar.

# 3

# Pump hydraulics

In the previous chapters, we have seen the evolution of pumps, their construction, and their wide applicability. The many requirements of pressurized liquid lead to a large variety of pumps each designed and suited to a required application.

In spite of many different types of components, the basic mechanics and the principle of operation of the centrifugal pumps are similar.

Centrifugal Pumps are hydraulic machines that are used to energise and transfer a fluid within a system, at a flow that is dependent upon system needs. In order to understand how the pumps perform this function, it is essential to get familiar with some of the hydraulic terms associated with centrifugal pumps and of the liquids that they handle.

## 3.1 Specific gravity

The term 'specific gravity' refers to the ratio of the density of liquid to the density of water at 4 °C (the density of water at this temperature is 1.000 kg/l). Specific gravity is a ratio and is hence a dimensionless quantity and is not expressed in any units.

$$\text{Specific gravity} = \frac{\rho_{\text{liquid}}}{\rho_{\text{water}} \ \text{at } 4^{\circ}\text{C}}$$

To find the specific gravity of a liquid, we must know its density in kilograms per meter cubed ($kg/m^3$) or in grams per millimeter cubed ($g/mm^3$). Then, divide this density by the density of pure water in the same units. If you use $kg/m^3$, divide by 1000. If we use $g/mm^3$, divide by 1 (that is, leave the number alone). It is important to use the same number of units in the numerator and denominator.

Materials with a specific gravity of less than 1 are less dense than water and therefore will float on it. Substances with a specific gravity of more than 1 are denser than water and will sink.

An object with a density of 100 $kg/m^3$ has a specific gravity of 0.1, and can float on the surface of the body of water. An object with a density of 10 $g/mm^3$ has a specific gravity of 10 and can sink rapidly.

## 3.2 Viscosity

Viscosity is best understood by imagining a styrofoam cup with a hole in the bottom. If honey is poured in this glass, it is noticed that the cup drains very slowly because the viscosity of honey is large compared to other liquids. If the same cup is filled with water, the cup will drain much more quickly. Viscosity is a measure of a fluid's resistance to flow.

It describes the internal friction of a moving fluid. A fluid with large viscosity resists motion because its molecular makeup gives it a lot of internal friction. A fluid with low viscosity flows easily because its molecular makeup results in very little friction when it is in motion.

In certain fluids called Newtonian fluids, the shear stress that causes the flow is directly proportional to the shear strain (rate of deformation).

The ratio of this shear stress to the shear strain is constant for a given fluid at a fixed temperature.

This constant is called the dynamic or absolute viscosity ($\mu$) and often simply the viscosity. The viscosity of liquids decreases rapidly with an increase in temperature. Thus, upon heating, liquids flow more easily.

The dimensions of dynamic viscosity are force times time divided by area. The unit of viscosity, accordingly, is newton-second per meter square (N-s/$m^2$).

For some applications, the kinematic viscosity is more useful than the absolute or dynamic viscosity.

Kinematic viscosity is obtained by dividing the absolute viscosity of a fluid by its mass density. (Mass density is the mass of a substance divided by its volume.)

$$\text{Kinematic viscosity} \quad \nu = \frac{\mu}{\rho}$$

The dimensions of kinematic viscosity are area divided by time. Its units are meter squared per second ($m^2$/s).

Kinematic viscosity ($\nu$) is often expressed in stokes, St, where $10^4$ St = 1 $m^2$/s. However, a more common unit of measure is centistokes (cSt).

Some other common viscosity units and conversion factor is listed below:

| **Kinematic Viscosity** | × | **Specific Gravity** | **Absolute Viscosity** |
|---|---|---|---|
| Centistokes | × | S.G. | Centipoise |
| SSU × 0.2198 – * | × | S.G. | Centipoise |

| | | | | |
|---|---|---|---|---|
| SSU – * | × | 0.2198 | = | Centistokes |
| Degree Engler – * | × | 7.45 | = | Centistokes |
| Seconds Redwood – * | × | 0.2469 | = | Centistokes |

* For centistokes greater than 50.

## 3.3 Vapor pressure

The vapor pressure of a liquid, pure or mixed, is defined as the pressure exerted by those molecules that escape from the liquid to form a separate vapor phase above the liquid.

If a quantity of liquid is placed in an evacuated, closed container, the volume of which is slightly larger than that of the liquid, most of the container is filled with the liquid. After a period, a vapor phase forms in the space above the liquid surface. This space consists of molecules that have passed through the liquid surface from liquid to gas. The pressure exerted by that vapor phase is called the *vapor* (or *saturation*) *pressure*. For a pure liquid, this pressure depends only on the temperature.

Following are some examples of vapor pressures for a few common liquids. The vapor pressure is 1 atm at 100 °C for water, at 78.5 °C for ethyl alcohol, and at 125.7 °C for octane.

Similarly, at 20 °C, water has a vapor pressure of 0.023 atm (17.5 mm Hg). Isopropyl alcohol (rubbing alcohol) has a vapor pressure of 0.043 atm (33 mm Hg) at 20 °C.

In a liquid solution, the component with the higher vapor pressure is called the light component (tendency to vaporize quicker), and that with the lower vapor pressure is called the heavy component.

## 3.4 Flow

The first and most important point to consider is that centrifugal pumps are volumetric machines. The liquid pumped is measured in terms of the volume flow rate. The units used are $m^3/h$ or gpm (US or Imperial).

It is worthy to note that any pump, for a single point of operation would always give the same volumetric flow rate for any liquid, be it hydrocarbon, water, or any other. Depending on the density of the liquid, the mass flow rate changes.

If we have a pump whose capacity is 20 $m^3/h$, then the mass flow rate would pump 20 tph of water. However, the same pump when handling a hydrocarbon with a specific gravity of 0.8 would pump only 16 tph.

## 3.5 Head

The pressure of the liquid can be stated in terms of meters (feet) of head of the liquid column (mlc). As in case of volumetric flow rate, the head generated by the pump in mlc for a single point of operation is the same for any liquid. Depending on the density of the liquid what changes is the reading on the pressure gage.

Any pump raises the liquid from one gradient (head) to another. Thus, the difference between the discharge head and the suction head is termed as 'differential head'.

The differential head developed by a pump is expressed in 'm' of liquid:

$$H_{\mathrm{m}} = \frac{(P_{\mathrm{d}} - P_{\mathrm{s}}) \times 10}{\rho}$$

$P_{\mathrm{d}}$ = Discharge pressure (kg/cm$^2$)

$P_{\mathrm{s}}$ = Suction pressure (kg/cm$^2$)

$\rho$ = Specific gravity of the liquid.

## 3.6 System resistance

The flow rate delivered by the centrifugal pump is dependent on the total *frictional and static* head that it has to overcome.

The required head comprises of two components. These are:

1. *A static component*: $h_{\mathrm{s}}$ in meters 'm', which is independent of the flow through the pump. For example, if the liquid has to be raised from one height to another, then it is the difference of the two heights.
2. *A friction head loss component*: $H_{\mathrm{f}}$ in meters 'm'. This head is proportional to the square of the flow rate – '$Q$' in l/s.

The friction component is the summation of losses that occur as the liquid flows through the pipes and various equipment like heat exchangers.

To account for the losses, the entire flow path from the suction vessel to the discharge vessel is considered.

If this path has a large number of fittings, such as elbows (more bends), reducers, valves, and orifices, the losses are higher. To ease the calculations, nomogram of equivalent length of valves and fittings is used. An equivalent length is the length of pipe that would offer the same losses for a flow rate as offered by the fitting.

Thus, the suction and discharge paths, which may have a few fittings, are converted to an equivalent length. Using another nomogram, the friction loss due to the flow rate can be estimated.

Usually the total losses (static + frictional) on the suction side of the pump are calculated separately to establish the fact that there is adequate NPSH-a available compared to NPSH-r required, for the pump to operate satisfactorily. If NPSH-a is less than NPSH-r, the pump will operate under conditions of cavitation which is undesirable. The next step is to evaluate the total system resistance $H_t$, which is the summation of losses on both the suction and discharge sides and includes the static lift $H_{st}$.

To compute the system resistance, consider the system as shown in Figure 3.1 where the pump flow rate is 100 $m^3$/h (27.8 l/s) of water, through steel pipes.

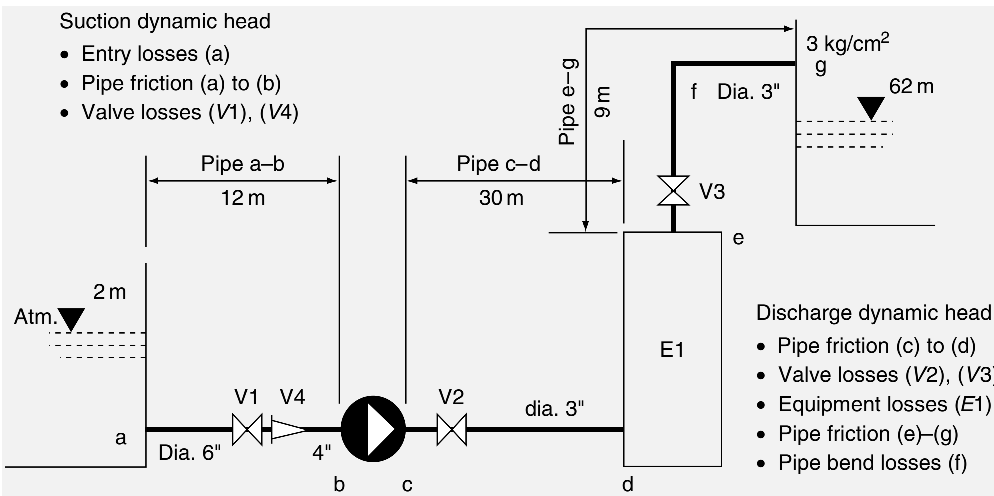



**Figure 3.1**
*A typical pumping system*

## 3.6.1 Evaluate the suction side

### Step 1: Calculate the velocity in the suction pipe – 6 in. (152.4 mm)

$$\text{Velocity} = \frac{\text{Flow}}{\text{Area}}$$

$$= \frac{(100/3600)}{[(\pi/4) \times (0.1524)^2]}$$

$$= 1.52 \text{ m/s}$$

### Step 2: Compute the suction head

The suction is from atmospheric vessel – $h_a = 10.34$ m
Suction height – $H_s = 2$ m

#### Step 3: Compute the equivalent pipe length

Pipe length = 12 m (assume almost all length is 6″ or 150 mm nom. bore)
6″ Gate valve (fully open) $V1$ – equivalent length = 1 m
6″ × 4″ eccentric reducer $V4$ – equivalent length = 1.4 m
Entry losses – equivalent length = 6 m
Total equivalent pipe length = 14.4 m.

#### Step 4: Compute friction loss from pipe friction tables

Friction loss for pipe length of 100 m = 1.52 m/100 m
Friction loss for pipe length of 20.4 m = 0.31 m/100 m

#### Step 5: Compute total suction head

$$\begin{aligned} h_a + H_s - P_1 &= 10.34 + 2 - 0.31 \\ &= 12.03 \text{ m} \end{aligned}$$

### 3.6.2 Evaluate the discharge side

#### Step 1: Calculate the velocity in the discharge pipe – 3.5 in. (90.1 mm)

$$\begin{aligned} \text{Velocity} &= \frac{\text{Flow}}{\text{Area}} \\ &= \frac{(100/3600)}{\left[(\pi/4) \times (0.0901)^2\right]} \\ &= 4.35 \text{ m/s} \end{aligned}$$

#### Step 2: Compute the discharge head

The discharge head in the vessel – $h_d$ = 30 m
Discharge height – $H_d$ = 62 m

#### Step 3: Compute the equivalent pipe length

Pipe length = 39 m (assume almost all length is 3.5″ or 90 mm nom. bore)
2 nos of 3.5″ Gate Valve (fully open) $V3$ and $V4$ – Equivalent Length = 0.5 × 2 = 1 m
One 90° bend – Equivalent length = 1 m
Total Equivalent pipe length = 40 m.

#### Step 4: Compute friction loss from pipe friction tables

Total Friction loss:

$$\begin{aligned} & 20.4 \times (1.33/100) && \text{for 150 mm nom. Bore pipe} \\ & + 41 \times (20.06/100) && \text{for 90 mm nom. Bore pipe} \\ & = 0.271 + 8.23 = 8.5 \text{ m} \end{aligned}$$

#### Step 5: Compute total discharge head

$$\begin{aligned} h_d + H_d + P_1 &= 30 + 62 + 8.5 \\ &= 100.5 \text{ m} \end{aligned}$$

Thus, the pump sees a differential head of:
Discharge head – suction head = 100.5 – 12.03 = 88.47 m.
The system resistance which the pump has to overcome is 88.47 m.

## 3.7 Pump efficiency

The pump does not completely convert kinetic energy to pressure energy since some of the kinetic energy is lost in this process. Primarily, there are three areas where this energy is dissipated and not converted to useful work. Pump efficiency is a factor that accounts for these losses. Pump efficiency is a product of the following three efficiencies:

1. Hydraulic efficiency (primarily, disk friction, which is the friction of the liquid with the impeller shrouds. This is a function of speed and impeller geometry. Other losses are shock losses during rapid changes in direction along the impeller and volute)
2. Volumetric efficiency (recirculation losses at wear rings, interstage bushes and other)
3. Mechanical efficiency (friction at seals or gland packing and bearings).

Some texts call the product of the first two efficiencies as internal efficiency of the pump.

Every pump is designed for a specific flow and a corresponding differential head, though it is possible to operate at certain percentage points away from the designed values.

However, the efficiency of the pump at the designed point is maximum and is called as the BEP. Efficiency at flows lower or higher than this design point is lower.

The efficiency of the pump has a close relationship to an important pump number called as the specific speed. This we shall cover in Section 3.11.

## 3.8 Hydraulic power

If a pump were an ideal machine, the required input power to drive the pump would entirely lift the mass flow rate from one elevation to another. This power is called as the hydraulic power.

$$P_{\mathrm{H(kW)}} = \frac{Q \times \rho \times g \times H}{3.6 \times 10^6}$$

Where

$Q$ = capacity in $m^3/h$
$\rho$ = liquid density in $kg/m^3$ at pumping temperature
$H$ = differential head in m (meters of liquid column)
$g$ = gravitational acceleration in $m/s^2$.

When this hydraulic power is divided by pump efficiency, we get the shaft power.

$$P_{\mathrm{S(kW)}} = \frac{P_{\mathrm{H}}}{\eta_{\mathrm{p}}}$$

## 3.9 Pump characteristic curve

With every pump, the manufacturer provides a curve depicting the performance or the behavior of the pump under various conditions. This is called as a characteristic curve of the pump.

Characteristic curve essentially comprises of four curves and these are:

1. Q *vs* H: Capacity vs differential head
2. Q *vs efficiency*: Capacity vs pump efficiency
3. Q *vs power*: Capacity vs shaft power
4. Q *vs NPSH*-r: Capacity vs Net Positive Suction Head – required.

### 3.9.1 Flow rate ($Q$) vs differential head ($H$) curve

The $Q$ vs $H$ curve is a continuously drooping curve from shut-off (no flow) condition to BEP. API recommends that the curve from BEP to shut-off should rise by at least 10% for single-stage, single pump operation.

Any pump model can be assembled with trimmed impellers (usually not smaller than 20% of the maximum possible diameter).

The $Q$ vs $H$ curve of trimmed impellers in the operating range are parallel and below the $Q$ vs $H$ curve of the maximum diameter impeller.

The characteristic curves encompassing performance of all the possible impeller diameters for that model have efficiency depicted as iso-efficiency curves on the $Q$ vs $H$ curve.

On every $Q$–$H$ curve, a small triangle is plotted to indicate the *rated* point of operation. The pump manufacturer guarantees this flow and the corresponding differential head.

Usually, the flow at rated point is in excess by 5 or 10% of the flow at which the pump will operate at most of the times or as specified by process demands. This operating point is called as the *normal* operating point.

Centrifugal pumps with radial impellers are started with discharge valves closed. At this point, there is no flow supplied by the pump and the entire liquid keeps churning in the casing. This point of operation is termed shut-off head and run time in this mode must be minimized.

### 3.9.2 Flow rate ($Q$) vs pump efficiency ($\eta_p$)

The $Q$ vs pump efficiency of the pump is an inverted 'U' shaped curve. At no flow, the efficiency is zero and then rises to a maximum value at a flow rate, which is termed as the BEP. Beyond this, the curve again drops.

The pumps operate in a range of flows but it has to be kept in mind that they are designed only for one flow rate point. Flow rates above and below this value result in higher hydraulic losses and hence lesser efficiency. The design point is the BEP.

### 3.9.3 Flow rate ($Q$) vs power ($P_s$)

The pump trial is carried using cold water as the liquid. As volumetric flow in $m^3/h$, differential head in $m$, and pump efficiency are independent of the liquid pumped, the results obtained are valid for all service liquids.

Power obtained is for water and can be easily extrapolated for the liquid by multiplying it with the specific gravity of the service liquid.

### 3.9.4 Flow rate ($Q$) vs NPSH-r

NPSH-r is covered in detail in Section 3.12. However to initiate, it is the Net Positive Suction Head required by a pump to avoid a phenomenon called as cavitation.

NPSH-r on the characteristic curves is the measured suction head obtained while throttling the suction flow until a 3% drop in the differential head is observed at any particular flow rate (see Figures 3.2 and 3.3).

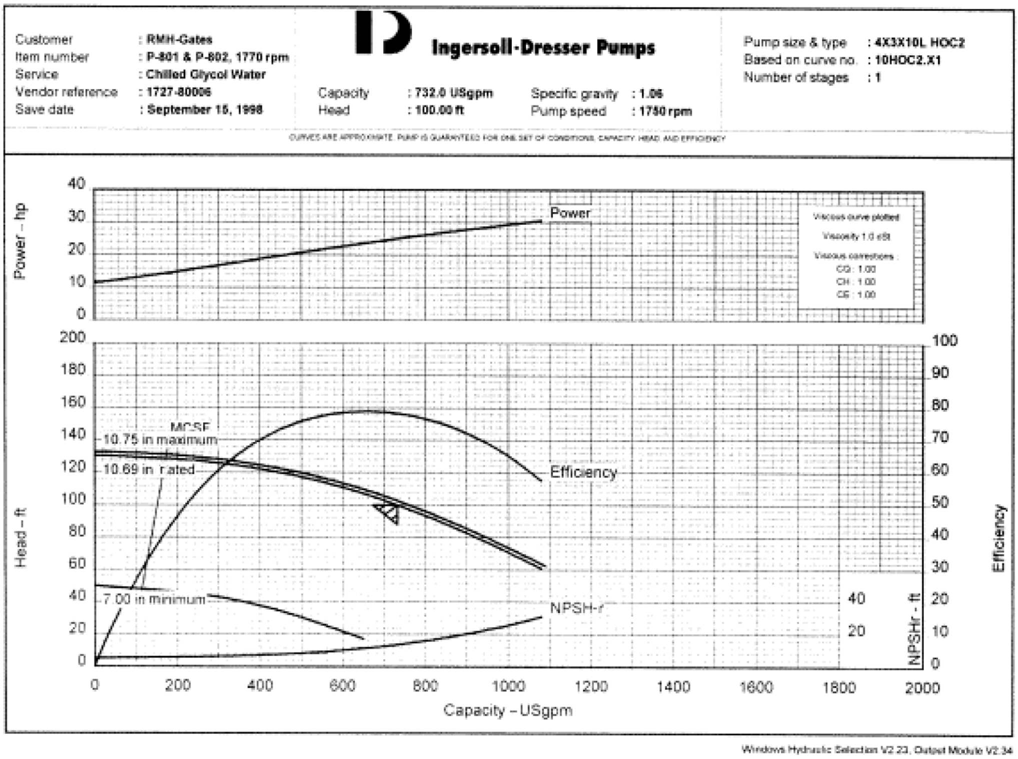



**Figure 3.2**
*A typical pump characteristic curve*

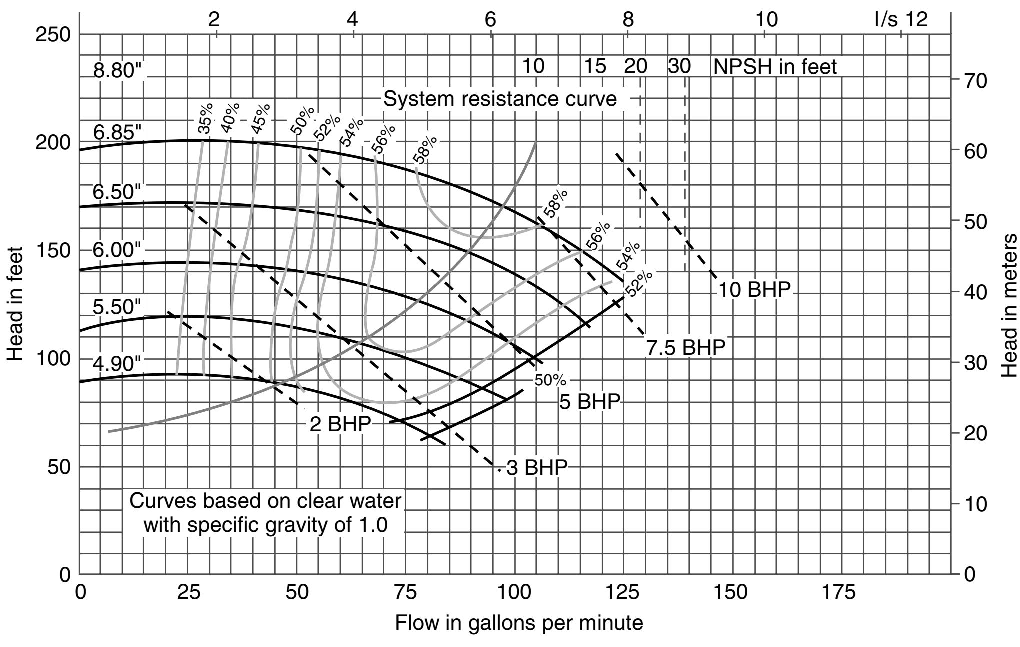



**Figure 3.3**
*Pump curve at various diameters with system resistance and iso-efficiency curves*

NPSH-r is dependant on the service liquid but it is known that cavitation resulting from cold water is most damaging as compared with most commonly pumped liquids (hydrocarbons, hot water) and so NPSH-r results obtained with cold water can be safely applied to other service liquids as well.

## 3.10 Curve corrections

The pump curves are generated while testing the pump using cold water as the liquid. The curve is fixed for a particular speed, impeller diameter, and water.

It is not necessary that the pumps' actual operation throughout its life will be for the same speed or impeller diameter and service. When any of these change, the pump flow and head generated will differ.

In certain cases, it is possible to predict the flow and head for alternate conditions using factors.

Thus, with the help of these factors, the curves can be corrected to obtain a performance map without retesting pump with modified conditions.

### 3.10.1 Affinity laws

The 'Affinity laws' are mathematical expressions that best define changes in pump capacity, head, and power absorbed by the pump when a change is made to pump speed, with all else remaining constant.

#### According to affinity laws

Capacity $Q$ changes in direct proportion to the change in pump speed $N$ ratio:

$$Q_2 = Q_1 \times \left(\frac{N_2}{N_1}\right)$$

Head $H$ changes in direct proportion to the square of the speed $N$ ratio:

$$H_2 = H_1 \times \left(\frac{N_2}{N_1}\right)^2$$

Power $P$ changes in direct proportion to the cube of the speed $N$ ratio:

$$P_2 = P_1 \times \left(\frac{N_2}{N_1}\right)^3$$

Where the subscript 1 refers to initial condition and 2 refers to new condition.

*Important*: The Affinity laws are valid only under conditions of constant efficiency.

The pump affinity laws mentioned above maybe utilized to determine the relationship between flow '$Q$' and impeller diameter as well as to predict Head '$H$' and Power '$P$' values with change in impeller diameter, whilst speed is kept constant.

The results obtained however are approximate as these formulae are analogous to the centrifugal pump affinity laws. Hence we have;

$$Q_2 = Q_1 \times \left(\frac{D_2}{D_1}\right) \quad H_2 = H_1 \times \left(\frac{D_2}{D_1}\right)^2 \quad P_2 = P_1 \times \left(\frac{N_2}{N_1}\right)^3$$

The affinity laws described above require correction when performance is to be predicted following a change in impeller diameter. Due to the above mentioned constant efficiency factor, there is a discrepancy between the calculated impeller diameter and the achieved performance. This error becomes larger with the increase in cut of the impeller.

If, $C$ is the calculated required percentage of impeller diameter and $A$ is the actual required diameter percentage of the original diameter then:

$$A = 16.2 + 0.838 \times C$$

Thus, by calculation using affinity laws, it is computed that the impeller has to be trimmed to 84% of the original diameter then the actual trimming should be limited to 86.6% of the original diameter.

*Note*: There are a number of recommended empirical formulae to calculate impeller diameters to match reduced or increased pump flow rates. KSB – Centrifugal Pump Design recommends the following approximate formula for KSB pump impeller trim.

$$\left(\frac{D_2}{D_1}\right)^2 \text{approx.} = \frac{Q_2}{Q_1} \text{approx.} = \frac{H_2}{H_1}$$

In all cases, however, proceed with caution.

### 3.10.2 Viscosity corrections

Under Section 3.2, we have discussed viscosity as a property of any fluid that is measure of its resistance to flow.

As the liquid flows through the pump, hydrodynamic losses are increased due to higher viscosity, as a result it is observed that when a viscous fluid is handled by a centrifugal pump:

- The brake horsepower requirement increases.
- There is a reduction in the head generated by the pump.
- Capacity reduction occurs with moderate and high viscosities.
- There is a decrease in the pump efficiency.

This is more evident in smaller pumps. For higher viscosities, larger pumps are used.

A viscosity correction chart from the Hydraulic Institute (as shown in Figure 3.4) provides coefficients for flow $C_q$, head $C_h$, and efficiency $C_\eta$.

These coefficients are used to modify the values of flow, head, and efficiency from the original curve. The new flow, head, and efficiency are obtained using the equations mentioned below.

$$Q_{vis} = C_q \times Q_w$$
$$H_{vis} = C_h \times H_w$$
$$\eta_{vis} = C_\eta \times \eta_w$$

Usually fluids more than 2 centipoise should be considered for viscosity correction.
e.g. – $Q$ = 500 US-gpm, $H$ = 80 ft, Viscosity = 1000 SSU.

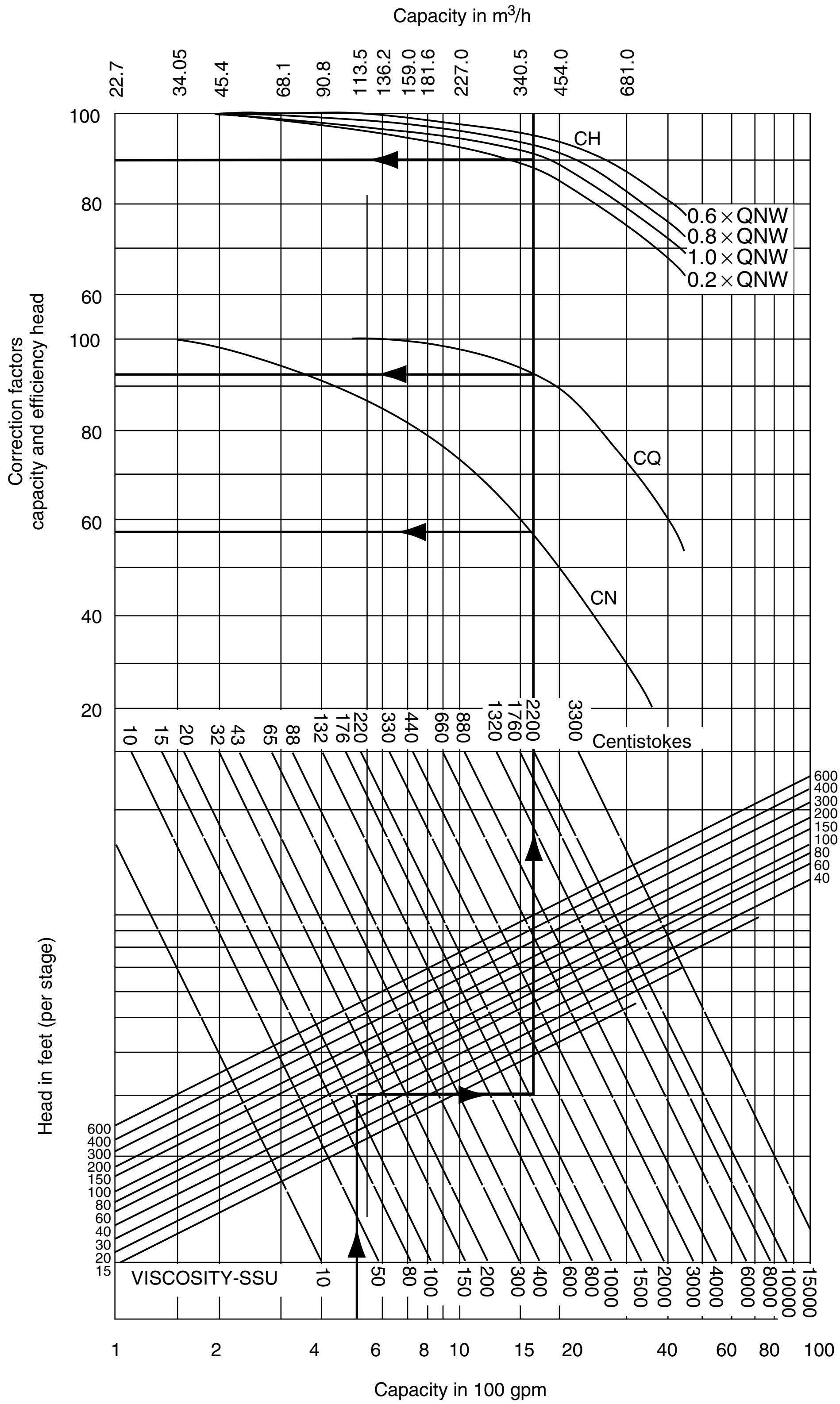



**Figure 3.4**
*Viscosity correction charts – correction factors, Hydraulic Institute*

## 3.11 Specific speed

Specific speed is a number characterizing the type of impeller in a unique and coherent manner.

Specific speed is defined by the equation,

$$N_s = \frac{N\sqrt{Q}}{(H)^{3/4}}$$

Where

$N$ = pump speed

$Q$ = flow at BEP at maximum impeller diameter (no corrections even if it is a double suction impeller)

$H$ = head per stage at BEP at maximum impeller diameter.

It states that $N_s$ is the speed in rpm at which a pump, if sufficiently reduced in size, would deliver (in US units) 1 gpm at a head of 1 ft. This definition is of little practical utility.

Specific speed is for an impeller, hence for multistage pumps only the first impeller is considered (in the equation, $H = H_{(total)}$/number of stages).

An index identifies the geometric similarity of pumps. Pumps of the same $N_s$ but have different sizes are considered geometrically similar, one pump being a size-factor of the other.

However, many critical parameters used for impeller design and geometry, volute design, pumps efficiency, layout of pump model performance charts as required by pump manufacturers are based on the specific speed.

Though, in principle, this text uses SI units as a base, an exception has been made in the case of specific speed. There is so much empirical work done with specific speed in US gpm, ft, and rpm units that it is considered prudent to get acquainted with FPS units than to persist with SI units and create confusion and errors.

To assist in getting familiar, the conversion from FPS to metric units is given below:

1 US-gpm = 0.2271 $m^3$/h → 1 British gpm = 0.2728 $m^3$/h
1 ft = 0.3048 m
US $N_s$ = 1.63 $N_s$ (metric $N_s$)
Metric $N_s$ = 0.614 US $N_s$ → British $N_s$ = 1.5 metric $N_s$

Since the 1900s, it was known that there existed a correlation between efficiency and the pump flow, head, and speed. In 1947, George Wislicenus generated curves (shown in Figure 3.5) of specific speed vs the pump efficiency. This was a statistical average of the data made available from the commercial pumps in those times.

A more detailed study was generated taking into account various factors such as dimensional tolerance, surface roughness of wet parts, specified wearing ring clearances, and designs not exceeding specified suction-specific speed limits and similar curves.

The curves shown in Figure 3.6 is based on the work conducted by Eugene P. Sabini and Warren H. Fraser in their paper, 'The Effect of Specific Speed on the Efficiency of Single Stage Pumps' presented at the 1986 Pumps Users Symposium.

They considered the following in the preparation of the curves:

- Single-stage pumps only
- Finish and dimensional tolerance to within ±1% for vanes and hydraulic passages

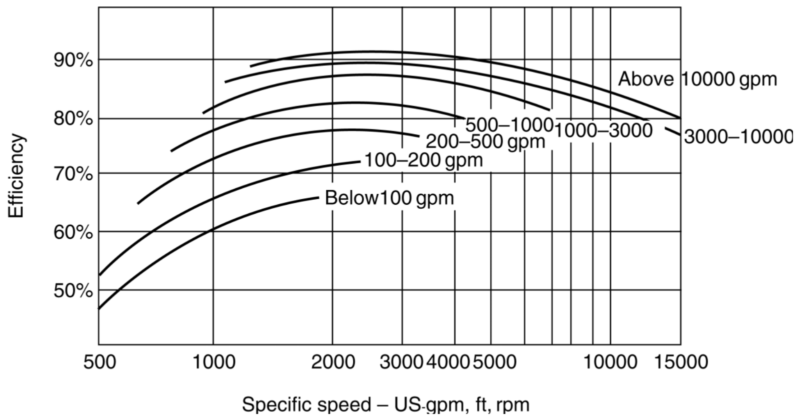



**Figure 3.5**
*An early chart – relating specific speed with single stage pump efficiency (Original by George Wislicenus – 1947)*

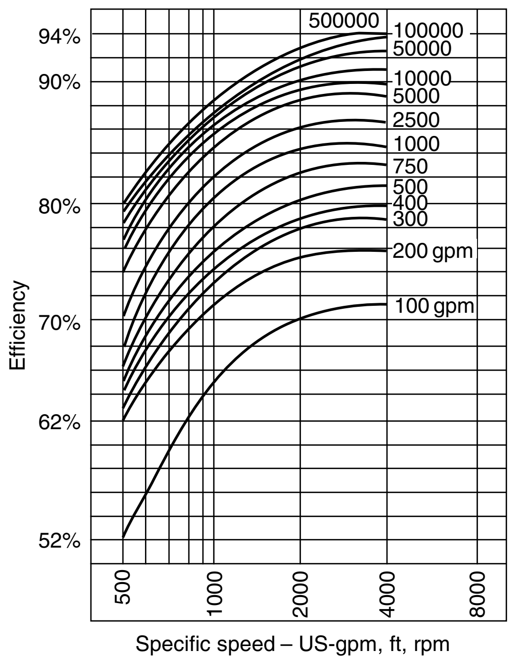



**Figure 3.6**
*Efficiencies of single-stage end-suction and double suction impeller pumps*

- Surface finish of wet surfaces to be $2 \times 10^{-6}$ per inch of impeller diameter or better
- Wearing ring clearances to be 0.0015 in. of ring diameter
- Suction-specific speed not exceeding 8500 (refer Section 3.13 – Units – US-gpm, ft, rpm)
- Discharge recirculation within a range of 80–90%
- A uniform velocity profile at impeller inlet
- Fluid used was clean water at a temperature of 150 °F or less
- Efficiencies were based on maximum impeller diameter
- Wet pit pump efficiencies were based on impellers with no back rings or balancing holes (Figure 3.7).

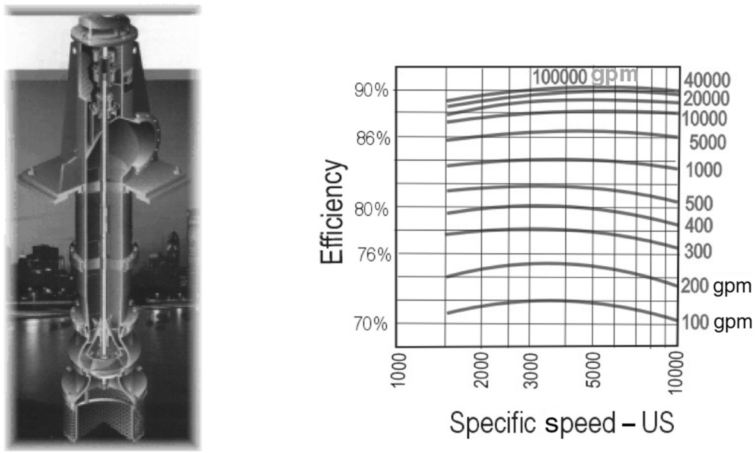



**Figure 3.7**
*Bowl efficiencies of wet pit centrifugal pumps*

A similar study was conducted by Lobanoff and Ross on pumps having six stages or less and operating at 3560 rpm. The study indicated that the efficiency for multistage pumps increases very rapidly to a specific speed of 2000 (US-gpm, ft, rpm) and stays constant until 3500 rpm. Then it begins to taper off a bit.

This is explained on the basis that hydraulic friction and shock losses for high specific speed pumps contribute greater percentage of total head than for low specific speed pumps.

The drop at low specific speeds is attributed to the fact that mechanical losses do not vary much over the range of specific speeds and are therefore a greater percentage of the total power consumption at the lower specific speeds.

Specific speed is a reference number that describes the hydraulic features of a pump, whether radial, semi-axial, or propeller type.

Another index related to the specific speed of the pump is the modeling law. It is usually applied to very large pumps in hydroelectric applications. It states that two geometrically similar pumps working against the same head will have similar flow conditions (same velocities at all sections) if they run at speeds inversely proportional to their size. In this case, the capacity will vary the square of the size.

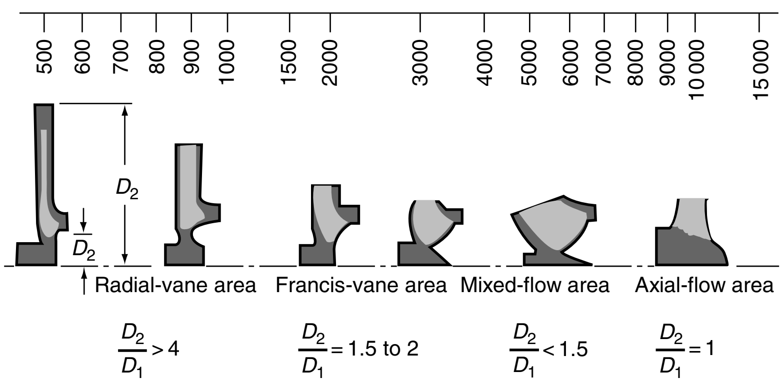



**Figure 3.8**
*Flow area chart*

The optimum laying out of the performance chart or the family curves of a pump model is based on the specific speed (Figure 3.8). The BEPs of the family of pumps are usually lined up along the same specific speed. Their size is factored upward for higher flows and heads.

By far, it is the most important number of any pump model.

## 3.12 Cavitation, recirculation, and Net Positive Suction Head (NPSH)

### 3.12.1 Cavitation

Gases under pressure can dissolve in a liquid. When the pressure is reduced, they bubble out. Opening of a soda-water bottle is a good example.

In a somewhat similar way, when the liquid is sucked in the pump inlet, the pressure acting on the liquid surface drops. Under conditions, when the reduced pressure approaches the vapor pressure of the liquid (at that temperature), it causes the liquid to vaporize (see Figure 3.9). As these vapor bubbles travel further into the impeller, the pressure rises again causing the bubbles to collapse or implode.

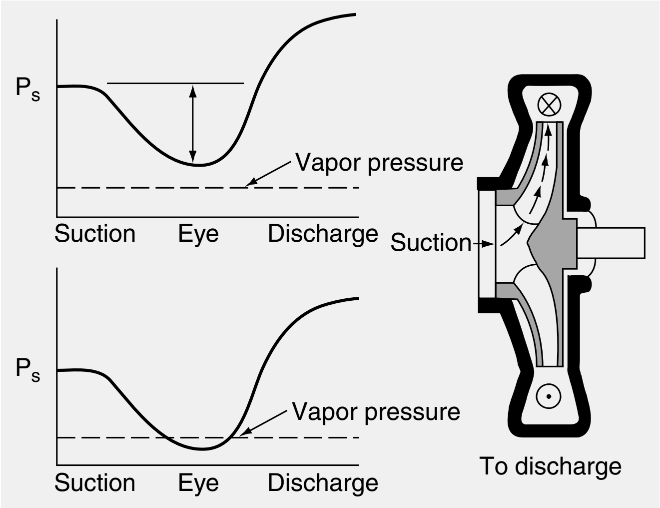



**Figure 3.9**
*Suction pressure falling below vapor pressure causes bubble formation*

This implosion adversely affects pump performance and could cause severe damage to pump internals. This phenomenon is called as 'cavitation'.

Cavitation damage to a centrifugal pump may range from minor pitting to catastrophic failure and depends on the pumped fluid characteristics, energy levels, and duration of cavitation.

Most of the damage usually occurs within the impeller; specifically, to the leading face of the non-pressure side of the vanes. This is the area where the bubbles normally begin to collapse and release energy on the vane. The net effect observed on the impeller vane will be a pockmarked, rough surface and severe thinning of the vanes from metal erosion.

### 3.12.2 Recirculation

Another type of cavitation seen in pumps is due to a phenomenon called as recirculation.

One of the complex problems associated with operation of pumps is that of recirculation. Recirculation is defined as flow reversal either at the inlet or at the outlet tips of the impeller vanes.

It is well established that cavitation type of damage seen on the inlet vanes and not associated with inadequate NPSH can be directly linked to the pump operation in the suction recirculation zone. Similar damage seen on the discharge vane tips too can be associated with pump operation in the discharge recirculation zone.

The suction and discharge recirculation may occur at different points as shown in Figure 3.10.

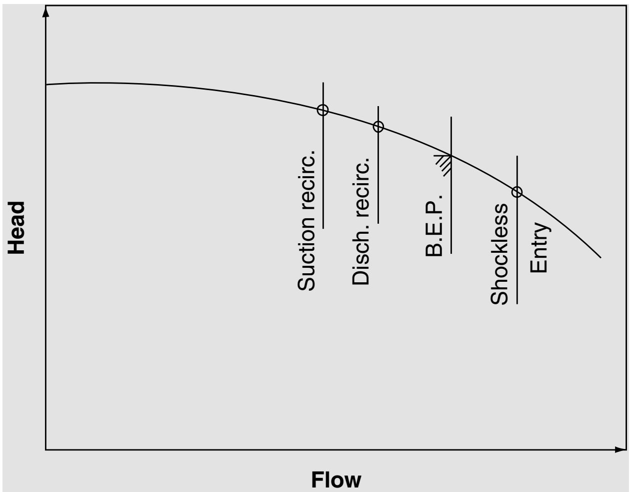



**Figure 3.10**
*Points on curve where recirculation can be expected*

The capacities at which the suction and discharge recirculation occurs are dependent on the design of the impeller at the inlet and outlet respectively. The casing has an influence on the intensity of the discharge recirculation but not on its inception.

Another observation made during extensive research indicated that when the inlet to outlet diameter ratio of the impeller equals or exceeds 0.5, the suction recirculation is in effect the capacity at which discharge recirculation occurs.

There are many explanations put forth to explain the phenomena of recirculation.

Recirculation can occur at the suction as well as in the discharge as shown in Figure 3.11.

One theory suggests that *recirculation cavitation* (rotating stall or separation) is the formation of vapor-filled pockets. This type of cavitation is different from the classical cavitation described earlier.

In suction recirculation, as the pump is operated to the left of the BEP, eddy currents begin to form at the eye of the impeller.

At this point of operation on the curve, there is no reduction in the flow rate through the pump. The eddy currents at the eye effectively reduce the flow channel size. As the flow rate is the same, the area is effectively reduced; it leads to an increase in the velocity of the liquid.

As the velocity increases, the pressure drops due to friction also increases. When there is a large drop in pressure below the liquid's vapor pressure, the pump experiences classical cavitation because of the initiating action of recirculation cavitation.

Another explanation provided for recirculation is that as the fluid flows over an impeller vane, the pressure near the surface is lowered and the flow tends to separate.

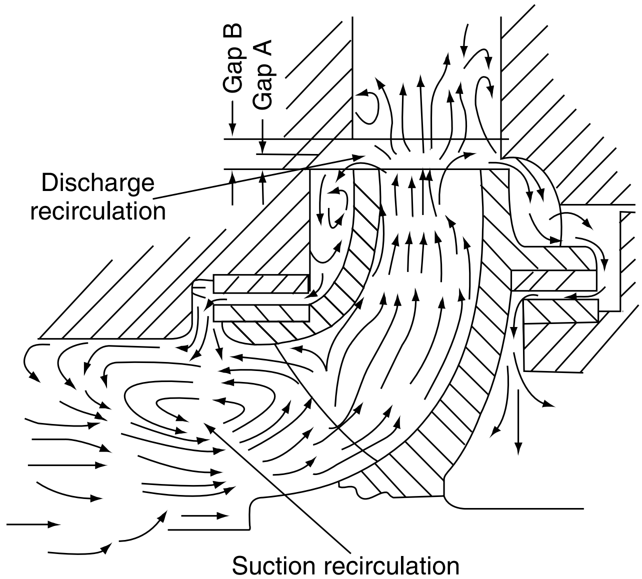



**Figure 3.11**
*Suction and discharge recirculation*

This separated region occurs when the incidence angle (see Figure 3.12), which is the difference between flow angle and pump impeller vane inlet angle, increases above a specific critical value.

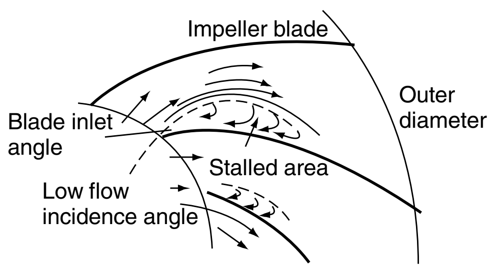



**Figure 3.12**
*Region of stall in the impeller*

The stalled area eventually washes but as the rotation continues, it is reformed. The area contains a vapor surrounded by a turbulent flowing liquid at a higher pressure than the vapor pressure. This separated region will then fill with liquid from the downstream end.

The vapor pocket collapses, which causes damage to the surface of the impeller vane. This may occur up to 200–300 times per second.

The damage due to recirculation occurs on the opposite side of the vane where classical cavitation occurs.

This continuous recycling results in noise, vibration, and pressure pulsations. These results imitate classical cavitation, and thus recirculation is often incorrectly diagnosed as cavitation. Figure 3.13 shows regions within an impeller that are affected by cavitation and recirculation.

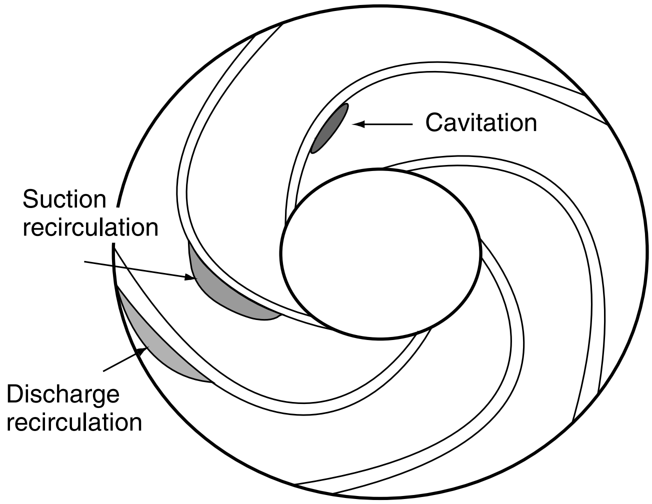



**Figure 3.13**
*Regions within an impeller that are affected by cavitation and recirculation*

It is often believed that only high-energy pumps (as per API 610 – sixth edition states: High-energy pumps are defined as pumping to a head greater than 650 ft (198 m) and more than 300 HP (224 kW) per stage) are affected by recirculation cavitation. However, an impeller constructed of cast iron or bronze can erode badly at much lower energy levels.

As the flow at the eye of the impeller recirculates, severe vortexing occurs. These vortices can pass through the impeller liquid channels and can initiate discharge recirculation, as shown in the figure on recirculation. The further the pump operates from its BEP, the greater the amount of vortexing.

Most pumps operate in either continuous or intermittent suction or discharge recirculation, especially those designed with high $N_{ss}$ – suction-specific speeds (above about 11 000). (Suction-specific speed is covered in Section 3.13.)

Impeller internal circulation usually shows up as cavitation noise and erosion damage, rotor oscillation, shaft breakage or surging in varying degrees depending on the pump design and application. Many of these problems can be avoided by designing the pump for lower suction-specific speed values and limiting the range of operation to capacities above the point of recirculation.

### 3.12.3 Net Positive Suction Head (NPSH)

The concept of NPSH involves two terms:

1. NPSH-r, called as the Net Positive Suction Head as required by the pump in order to prevent the inception of cavitation and for safe and reliable operation of pump.
2. NPSH-a, called as the Net Positive Suction Head as made available by the suction system of the pump.

NPSH and its correlation to inception of cavitation has been a matter of great research and many theories.

This field is still misunderstood, misapplied, and misused that results in costly over design of new systems or unreliable operation of existing installations of pumps.

## Net Positive Suction Head (NPSH) – required

The Hydraulic Institute defines NPSH-r of a pump as the NPSH that causes the total head (first stage head of multistage pumps) to be reduced by 3% due to flow blockage from cavitation vapor in the impeller vanes.

The above term is a practical method of exactly determining the point of minimum suction head for a pump.

The rated pump head is not achieved when the NPSH-a equals the NPSH-r of the pump. The head will be 3% less than the fully developed head value as shown in Figure 3.14.

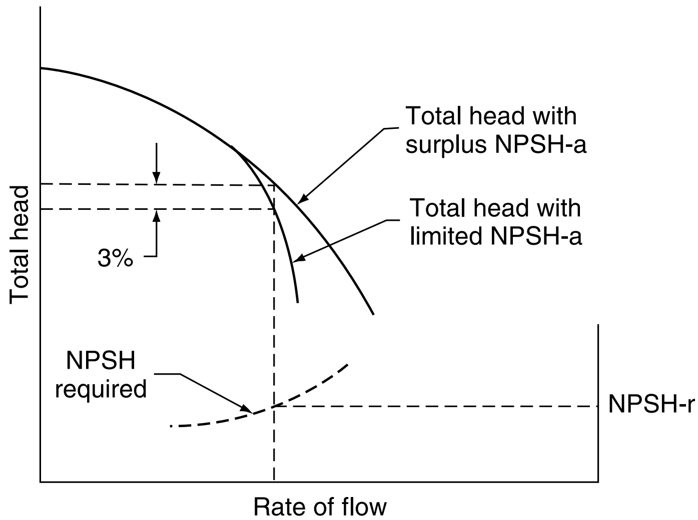



**Figure 3.14**
*When the margin of NPSH-a and NPSH-r lowers, the differential head drops*

Strange it may sound but NPSH-r by the above definition does not necessarily imply that this is the point at which cavitation starts; that level is referred to as incipient cavitation.

The NPSH at incipient cavitation can be from 2 to 20 times the 3% NPSH-r value, depending on pump design. The higher ratios are normally associated with high-suction energy pumps or pumps with large impeller inlet areas.

The suction energy level of a pump increases with:

- The casing suction nozzle size
- The pump speed
- The suction-specific speed
- Specific gravity of the pumped liquid.

Anything that increases the velocity in the pump impeller eye, the rate of flow of the pump, or the specific gravity, increases the suction energy of the pump (Figure 3.15).

Most standard low-suction energy pumps can operate with little or no margin above the NPSH-r value, without seriously affecting the service life of the pump.

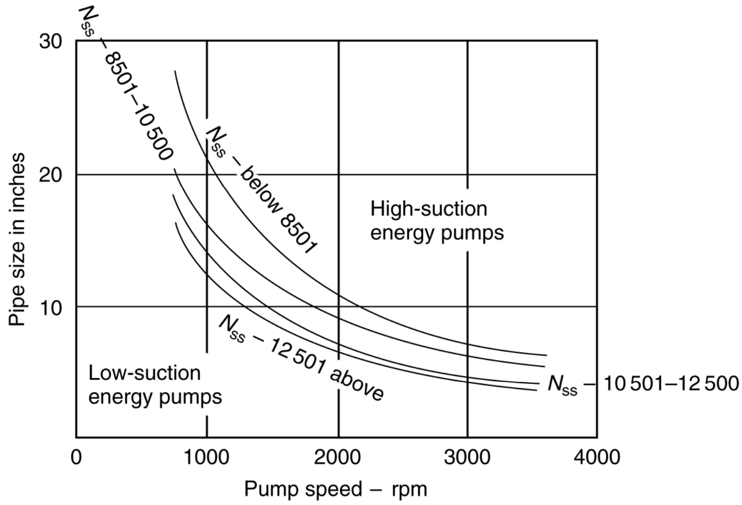



**Figure 3.15**
*Flow rate chart*

Thus, we see that NPSH-r as per Hydraulic Institute's definition of 3% head drop is not an indicator of inception of cavitation and consequent pump damage.

It then becomes necessary to come up with a theoretical NPSH-r that can indicate the inception of cavitation that can then ensure a cavitation-free operation.

The theoretical derivation of NPSH-r or 'Cavitation-Free NPSH' is based on factors such as:

- Head loss due to friction
- Head drop due to fluid acceleration
- Head loss due to improper fluid entry into the impeller blade.

As the liquid in the suction pipe approaches the impeller eye, it has velocity and acceleration. In addition, it has to change its direction to enter the impeller. Losses in terms of liquid head occur due to each of the above and because of friction.

The pump inlet nozzle and impeller inlet vane geometry are designed to minimize the losses largely but cannot be eliminated.

Other factors like higher flow rates and recirculation due to higher clearance at wear rings and use of smaller diameter impellers in volutes can increase the losses.

The summation of the above losses is termed as net positive suction head as required by the pump or NPSH-r. In other words, NPSH-r is the summation of losses in the critical area between the suction nozzle and the leading edge of the first stage impeller blades.

Mathematically, the NPSH-r is expressed in the following equation:

$$\text{NPSH-r} = \frac{K_1 \times C_{M_1}}{2g} + \frac{K_2 \times W^2}{2g}$$

The first term represents the friction and acceleration losses and the second term represents the blade entry losses.

To understand this equation, we need to learn the inlet flow velocity triangle of the pump impeller.

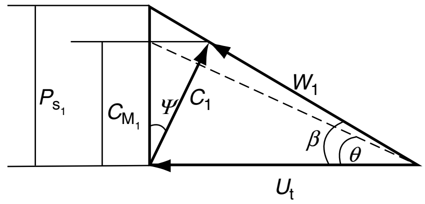



**Figure 3.16**
*Impeller inlet velocity triangle*

$C_{M_1}$ = Average meridional (plane passing through shaft axis) velocity at blade inlet

$$= \frac{Q \text{ m}^3/\text{gec}}{A_{\text{m}^2}} \text{ (m/s)}$$

$U_t$ = Peripheral blade velocity (m/s)

$= (\pi \times D \times N)/60$

$D$ = Impeller eye diameter in meters

$N$ = speed in rpm

$W_1$ = Relative velocity (m/s)

$C_1$ = Absolute velocity of flow (m/s)

$P_{s_1} = C_{M_1}/R_1$ ($R_1$ is a factor used to determine the vane outlet angle)

$\theta$ = Angle of flow approaching the impeller

= Blade angle at outer radius of impeller eye

$\Psi$ = Pre-rotation angle (usually not more than 30°)

$\alpha$ = Incidence angle = $(\beta - \theta)$.

The angle of the inlet edge of the impeller vane, at the point where the vane joins the front shroud, measured in a plane tangent to the shroud surface, is $\beta$.

$U_t$ is the peripheral velocity of that same point.

$W_1$ is the relative velocity (relative to the impeller) of the liquid just before entering the vanes.

$C_{M_1}$ is the meridional velocity of the liquid just before entering the vanes. The meridional velocity $C_{M_1}$ is the velocity relative to the casing. It lies in the meridional plane (the plane that passes through the shaft centerline).

(If you were standing in the suction nozzle of the casing, facing the impeller, $C_{M_1}$ velocity would hit you squarely in the back.) The capacity is such that these three vectors create a right triangle.

If there is 'no pre-rotation' or 'no pre-swirl' or 'shock less-entry', then the included angle $\theta = \beta$.

Going back to the NPSH-required equation, the first term of '$K_1 C_{M_1}^2/2g$' represents the friction and acceleration losses and the second term '$K_2 W_1^2/2g$' represents blade entry losses.

In low-suction energy pumps, the first term is the prime factor and in the high-suction energy pumps the latter term gains prominence.

The constant $K_1$ is influenced largely by the pump suction nozzle approaching the impeller eye.

The angle of incidence $\alpha$ that influences $K_2$ is the difference between the inlet angle $\beta$ and the flow angle $\theta$. The angle $\beta$ is determined from $C_{M_1}$ multiplied by the factor $R_1$ that allows for the effects of recirculated flow $Q_L$ and non-uniform velocity distribution.

The flow $Q_L$ may vary depending on the wear that has occurred causing additional leakages from wearing ring clearances and balance lines of a multistage pump. In lower specific speed pumps, the percentage of leakage flow is a higher percentage of the total flow and hence there is a greater impact on the NPSH-r of such pumps.

All the discussions on cavitation and recirculation were based on phenomena that lead to vapor formation and implosion.

The boiling of the liquid or vapor formation during cavitation or recirculation is a thermal process and is dependent on the properties of the liquid. These properties include pressure, temperature, specific and latent heats of vaporization. If liquid has to boil, the latent heat of vaporization has to be derived from the liquid flow.

The extent of cavitation depends on the proportion of vapor released, the rapidity of liberation, and the vapor specific volume.

This is accounted by a gas to liquid ratio factor $C_b$. In a paper by D.J. Vlaming, ‘A Method of Estimating the Net Positive Suction Head Required by Centrifugal Pumps’, ASME 81-WA/FE-32 has provided values of $C_b$ that indicate that cold water has the potential for causing most damage by way of cavitation.

Taking all these into consideration, it is found that cold water is the most damaging of the commonly pumped liquids. Similarly, this difference applies to water at different temperatures.

A review of the properties of water and its vapor at several temperatures shows that the specific volume of vapor decreases rapidly as pressure and temperature increase. Hence, due to this, cold water is again more damaging than hot water.

Even field experience tends to corroborate the above study. Pumps handling certain hydrocarbons or hot water operate satisfactorily with lower NPSH-a than does cold water.

Thus, pumps tested with cold water to detect NPSH-r are found to be good enough for the above-mentioned services.

A considerable research material is available on this topic and Terry L. Henshaw in his paper; ‘Predicting NPSH for Centrifugal Pumps’ has compiled the works of many people on this subject. Their research findings are listed below. It makes an interesting study into this complex number called as the NPSH-r.

- The current industry standard to define the NPSH requirement of a centrifugal pump as NPSH available with cool water, which creates cavitation in the eye of the impeller sufficient to cause the (one stage) head of the pump to drop 3%.
- Because of the 3% head-drop definition, a pump with a larger-diameter impeller will require less NPSH than the same pump with a smaller-diameter impeller.
- The amount of NPSH required to achieve 100% head is typically 1.05–2.5 times the NPSH-r for the 3% head drop.
- The amount of NPSH required to suppress all cavitation is typically 4–5 times the NPSH-r for the 3% head drop, although this ratio can vary from 2 to 20.
- ‘Incipient’ cavitation causes minimal damage to the impeller.
- The peak cavitation erosion rate occurs at an NPSH-a value above that of the 3% NPSH-r and below that coincident with incipient cavitation.
- Cool water among the many liquids is the most damaging to a cavitating pump.
- For cool-water services, the 3% head-drop NPSH is not sufficient to prevent cavitation erosion to the impeller.
- For most pumps, at best efficiency flow rate (BEP), the NPSH-r, based on the 3% head drop, does not vary with speed to the exponent of 2. The exponent is more typically 1.5. Therefore, suction-specific speed at BEP, $N_{SS}$-$_{BEP}$ increases as speed increases.

- The shape of the NPSH-r curve varies with the percentage head drop. The NPSH-$r_3$ decreases as flow rate decreases, reaching a minimum value, normally at or below 40% of the BEP. The NPSH-r curves for 1 and 0% head drop increase as the flow rate decreases below BEP.
- Field experience has revealed that an increase in pump failure rates occur when suction-specific speeds (calculated at BEP and in US units) exceed around 10 000, with a pronounced increase at 11 000.
- The Hydraulic Institute Standards uses $N_{ss}$ value of 8500 (US) as the basis for their maximum speed recommendations.
- At no-prerotation flow rate, the NPSH-a for incipient cavitation is equal to the NPSH-a for the 3% head drop plus 'peripheral velocity head' ($U_1^2/2g$).
- To reduce the NPSH-$r_3$ at BEP, impellers are typically designed, at the BEP flow rate, such that $P_{s_1} > C_{M_1}$ (refer Figure 3.16). The $P_{s_1} > C_{M_1}$ ratio is typically about 1.25. Therefore, the flow rate coincident with no prerotation is typically about 25% larger than the BEP flow rate.

## Net Positive Suction Head (NPSH) – available

Every pump has an associated inlet system comprising of vessel, pipes, valves, strainers, and other fittings. The liquid, which has a certain suction pressure, experiences losses as it travels through the inlet system.

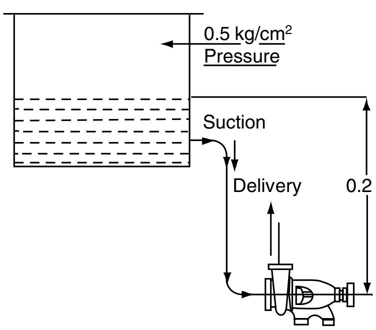
Thus, the net inlet pressure (in absolute terms) of the pipe and fitting losses is what is available at pump inlet and this is called as the Net Positive Suction Head – available or NPSH-a.

Now knowing what is NPSH-a and NPSH-r, it becomes clear that their difference has to be greater than the vapor pressure of liquid at that temperature to avoid vaporization of liquid.

As a convention, a mathematical simplification is done. The vapor pressure of the liquid is subtracted from the NPSH-a. In pump terminology, NPSH-a includes vapor pressure correction.

Thus, all we need to insure is that the NPSH-a is greater than NPSH-r. Their difference in such a case would then be in the true sense of the word *Net Positive Suction Head*. Some common examples of NPSH calculations are provided below.

### Calculating NPSH-available: pressurized flooded suction




$$\text{Vapor pressure} - P_{vap} = 0.45\,\text{kg/cm}^2$$

$$\text{Pipe losses} - H_f = 1.5\,\text{m}$$

$$\text{Specific gravity} = 0.8$$

$$P_g - \text{gage pr.} = 0.5\,\text{kg/cm}^2$$

$$H_g \text{ in meters 'm'} = 0.5 \times 10.2/0.8 = 6.4\text{ m}$$

$$H_{st} \text{ in meters 'm'} = +\,0.2\,\text{m}$$

$$H_f = 1.5\,\text{m}$$

$$H_a - \text{atmospheric pressure} = 10.325/0.8 = 12.9\,\text{m}$$

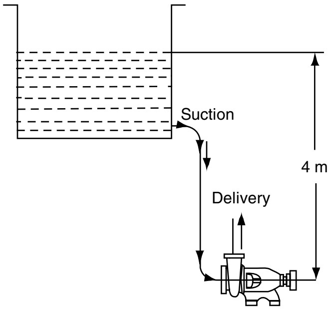
$$H_{vap} \text{ in meters 'm'} = 0.45 \times 10/0.8 = 5.7\,\text{m}$$

$$H_a + H_s + H_{st} - H_f - H_{vap}$$

$$\text{NPSH-a in meters 'm'} = 12.9 + 6.4 + 0.2 - 1.5 - 5.7$$

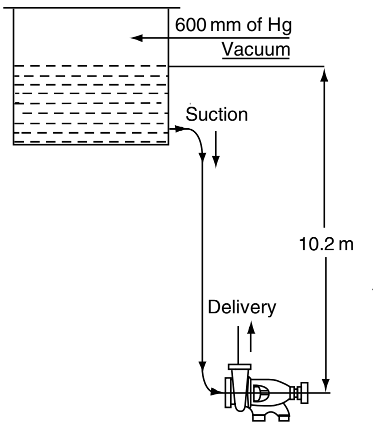
$$= 12.3\text{ m}$$

**Calculating NPSH-available: atmospheric flooded suction**



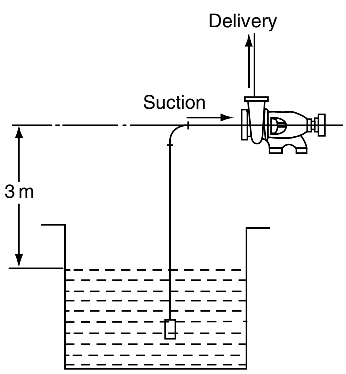

$$\text{Vapor pressure} - P_{\text{vap}} = 0.45\,\text{kg/cm}^2$$
$$\text{Pipe losses} - H_{\text{f}} = 1.5\,\text{m};$$
$$\text{Specific gravity} = 0.8$$
$$P_{\text{g}} - \text{gage pressure} = 0\,\text{kg/cm}^2$$
$$H_{\text{g}} \text{ in meters 'm'} = 0 \times 10.2/0.8 = 0\,\text{m}$$
$$H_{\text{st}} \text{ in meters 'm'} = +\,4\,\text{m}$$
$$H_{\text{a}} - \text{atmospheric pressure} = 10.325/0.8 = 12.9\text{ m}$$
$$H_{\text{f}} \text{ in meters 'm'} = 1.5\text{ m}$$
$$H_{\text{vap}} \text{ in meters 'm'} = 0.45 \times 10/0.8 = 5.7\,\text{m}$$
$$= H_{\text{a}} + H_{\text{g}} + H_{\text{st}} - H_{\text{f}} - H_{\text{vap}}$$
$$\text{NPSH-a in meters 'm'} = 12.9 + 0 + 4 - 1.5 - 5.7 = 9.7\,\text{m}$$

**Calculating NPSH-available: vacuum flooded suction**




$$\text{Vapor pressure} - P_{\text{vap}} = 0.45\,\text{kg/cm}^2$$
$$\text{Pipe losses} - H_{\text{f}} = 1.5\,\text{m};$$
$$\text{Specific gravity} = 0.9$$
$$P_{\text{g}} - \text{gage pressure} = -\,600\,\text{mm-Hg}$$
$$H_{\text{g}} \text{ in meters 'm'} = -(600/1000) \times 13.6/0.9 = -9.1\,\text{m}$$
$$H_{\text{st}} \text{ in meters 'm'} = +10.2\,\text{m}$$
$$H_{\text{a}} - \text{atmospheric pressure} = 1.033 \times 10/0.9 = 11.5\,\text{m}$$
$$H_{\text{f}} \text{ in meters 'm'} = 1.5\,\text{m}$$
$$H_{\text{vap}} = 0.45 \times 10/0.9 = 5.1\,\text{m}$$
$$= H_{\text{a}} + H_{\text{g}} + H_{\text{st}} - H_{\text{f}} - H_{\text{vap}}$$
$$\text{NPSH-a} = 11.5 - 9.1 + 10.2 - 1.5 - 5.1 = 6.0\,\text{m}$$

**Calculating NPSH-available: negative suction**




$$\text{Vapor pressure} - P_{\text{vap}} = 0.45\,\text{kg/cm}^2$$
$$\text{Pipe losses} - H_{\text{f}} = 1.5\,\text{m};$$
$$\text{Specific gravity} = 0.8$$
$$P_{\text{g}} - \text{gage pressure} = 0\,\text{kg/cm}^2$$
$$H_{\text{g}} \text{ in meters 'm'} = 0 \times 10.2/0.8 = 0\,\text{m}$$
$$H_{\text{a}} - \text{atmospheric pressure} = 10.325/0.8 = 12.9\,\text{m}$$
$$H_{\text{st}} \text{ in meters 'm'} = -3\,\text{m}$$
$$H_{\text{f}} \text{ in meters 'm'} = 1.5\text{ m}$$
$$H_{\text{vap}} \text{ in meters 'm'} = 0.45 \times 10.2/0.8 = 5.7\,\text{m}$$
$$= H_{\text{a}} + H_{\text{g}} + H_{\text{st}} - H_{\text{f}} - H_{\text{vap}}$$
$$\text{NPSH-a in meters 'm'} = 12.9 + 0 + (-3) - 1.5 - 5.7$$
$$= 2.7\,\text{m (Satisfactory)}$$

## Net Positive Suction Head (NPSH) – margin

The simple approach considered in regard to NPSH margin is the net between the available and required NPSH.

It is a requirement that the NPSH-a available must be equal to or greater than the NPSH-r stipulated by the pump manufacturer. Most pump specifications quote a margin of not less than 1 to 1.5 m over the entire range of pump operation.

When the difference between NPSH-a and NPSH-r is less than the stated margin, it calls for an exact determination of the NPSH-r by carrying out the NPSH-r test.

Another approach adopted to define the margin is by taking the ratio of NPSH-a and NPSH-r.

The table given below offers suggested minimum NPSH margin ratio guidelines (NPSH-a/NPSH-r), within the allowable operating region of the pump (with standard materials of construction). It is based on the experience of the many pump manufacturers with many different pump applications.

| | **Minimum NPSH Margin Ratio Guidelines (NPSH-a/NPSH-r)** | | |
|---|---|---|---|
| | **Suction Energy Levels** | | |
| **Application** | **Low** | **Medium** | **High** |
| Petroleum | 1.1-a | 1.3-c | |
| Chemical | 1.1-a | 1.3-c | |
| Electrical Power | 1.1-a | 1.5-c | 2.0-c |
| Nuclear Power | 1.5-b | 2.0-c | 2.5-c |
| Cooling Towers | 1.3-b | 1.5-c | 2.0-c |
| Water/Waste water | 1.1-a | 1.3-c | 2.0-c |
| General Industry | 1.1-a | 1.2-b | |
| Pulp and Paper | 1.1-a | 1.3-c | |
| Building Services | 1.1-a | 1.3-c | |
| Slurry | 1.1-a | | |
| Pipeline | 1.3-b | 1.7-c | 2.0-c |
| Water Flood | 1.2-b | 1.5-c | 2.0-c |

*Source* – www.pumps.org:

‘a’ – Or 0.6 m (2 ft) whichever is greater
‘b’ – Or 0.9 m (3 ft) whichever is greater
‘c’ – Or 1.5 m (5 ft) whichever is greater.

Vertical turbine (misnomer) pumps often operate without NPSH margin without damage, but with slightly reduced discharge head. Such pumps generally have low-suction energy, and cavitation noise is normally not an issue. NPSH-a has to be equal to or larger than the NPSH-r over the allowable operating region of the pump, including a low water level.

High and very high suction energy pumps operating with the margins specified in the table will have acceptable bearing and seal life. However, they may still be susceptible to impeller erosion and higher noise levels.

In addition to the margins specified in the table additional requirements of suction head arise due to:

- Increase in wearing ring clearances due to wear. This increases the leakage flow to the impeller eye and disturbs the inlet flow pattern
- Gas content in the liquid
- Improperly designed inlet piping and pump casing that cause non-uniform suction flow or turbulence
- Operation of the pump on the farther right hand side of BEP. In this region, the NPSH-a reduces and NPSH-r increases.

## 3.13 Suction-specific speed

Suction-specific speed is defined by the equation;

$$N_{ss} = \frac{N\sqrt{Q}}{(\text{NPSH-r})^{3/4}}$$

Where
$N$ = pump speed
$Q$ = capacity at BEP at maximum impeller diameter (it gets halved for a double suction impeller)
NPSH-r = Net Positive Suction Head (required) at BEP at maximum impeller diameter.

Studies carried out have empirically established that pump models with $N_{ss}$ less than 11 000 (US units: $Q$ – US gpm, $N$ – rpm, NPSH-r – feet) have a more stable operation and are more reliable.

Therefore, it is commonly used as a basis for estimating the safe operating range of capacity for a pump. The higher the $N_{ss}$ is, the narrower is its safe operating range from its BEP. Most users prefer that their pumps have $N_{ss}$ in the range of 8000–11 000 for optimum and trouble-free operation.

It is usually recommended that such pumps ($N_{ss}$ > 11 000) should not be operated at flow rates below 60–70% of the BEP. When the pump is operated below this range, it may experience:

- Impeller and casing erosion
- Shaft deflection and stress
- Radial and thrust bearing failures
- Seal problems.

The above are attributed to the recirculation of liquid at the impeller inlet, which has been covered in the earlier section.

For a smooth operation, the liquid enters the impeller at a particular designed angle. This inlet angle is meant for flows at the BEP, however, at lower flows, the liquid enters the impeller at a much different angle and is unable to make an entry into the impeller.

As a result, it is forced back into the pump suction pipe. The liquid keeps recirculating in front of the impeller.

Evidence of recirculation at impeller inlet is:

- Suction pressure gage fluctuations
- Noisy operation
- High vibrations at low flow rates.

If a higher $N_{ss}$ pump model is thus encountered, users prefer to buy a lower speed pump even though it may cost more.

## 3.14 Performance calculation procedure

For a centrifugal pump, the performance calculation's aim is to determine the pump efficiency. This value can be read on the characteristic curves provided by the manufacturer. The deviation of the calculated efficiency from the rated efficiency indicates the performance degradation of the pump.

### 3.14.1 Flow measurement

Flow measurement can be taken from a flow measuring device, if fitted. In cases where flow measurement devices are not installed, non-invasive ultrasonic flow meters can be used to measure the flow from the pump.

The flows are usually indicated as mass flow in kg/h. It is recommended to convert this to volume flow in $m^3/h$.

$M$ = Mass flow (kg/h)
$Q$ = Volumetric flow ($m^3/h$)
$\delta$ = Density at pumping temperature ($kg/m^3$)

$$Q = \frac{M}{\delta}$$

### 3.14.2 Differential head

$h_s$ = Suction head (m)
$P_s$ = Suction pressure ($kg/cm^2$)
$\rho$ = Specific gravity at pumping temperature

$$h_s = \frac{10 \times P_s}{\rho}$$

Discharge head $h_d$ in m is calculated in a similar manner.
Differential head – $h$ in m is further calculated as

$$H = h_d - h_s$$

### 3.14.3 Hydraulic power

The next step in this process is to calculate the hydraulic power. This is calculated in the following manner.

$g$ = gravitational acceleration – 9.81 $m/s^2$.

$$P_{H(kW)} = \frac{Q \times \delta \times g \times H}{3.6 \times 10^6}$$

The hydraulic power is minimum power required to pump the fluid. This will be the power if the pump had an efficiency of 100%. However, this is not possible in practice. To obtain the actual pump efficiency we need to go to the next step of calculating the energy being provided to the pump by the prime mover. Let us assume that in this case the prime mover is an electrical motor.

### 3.14.4 Motor power

The electrical power is fed to the motor to its terminals. However, we are interested in the power that is delivered by the motor at its coupling with the pump. Thus, we need to consider the efficiency of the motor too.

Efficiency of the motor is not a fixed number but changes with the load on the motor. A part load efficiency of the motor is much lower than its efficiency at full load. This table of motor efficiency with respect to its load is provided by the motor manufacturer.

$V$ = Measured voltage in volts
$I$ = Measured current in ampere
$\cos\varphi$ = Measured power factor
$\eta_e$ = Motor efficiency

Motor Power, $P_M$ in kW at its coupling is:

$$P_M = \frac{\sqrt{3} \times V \times I \times \cos\phi \times \eta_e}{1000}$$

### 3.14.5 Pump efficiency

Having performed the above calculations, we are now in a position to derive the pump efficiency. The ratio of pump hydraulic power to the motor power gives the pump efficiency.

$\eta_p$ = Pumping efficiency (hydraulic efficiency)

$$\eta_p = \frac{P_H}{P_M}$$

# 4

# Forces in centrifugal pumps

As we have seen in Chapter 3, a centrifugal pump can deliver a wide range of flows and heads of liquid column. The operating point on the curve; the internal pressures generated in the pump result in radial and axial forces on the rotor of the pump.

In this chapter, we shall discuss these forces in qualitative and quantitative terms and their impact on the design of the pumps.

## 4.1 Axial thrust

The axial forces of thrust generated in a centrifugal pump results from the internal pressures acting on the exposed areas of the rotating element.

It may appear as simple as a product of the net of discharge and suction pressure and the exposed area of the impeller. Though this is truly the basis, there are many uncertainties that are not covered by this simple approach.

The other variables that may affect the evaluation of axial loads are:

- Location of the impeller relative to the casing wall
- Impeller shroud symmetry
- Surface roughness of the walls
- Wearing ring clearance
- Geometry of the balancing holes.

Many of these variables are indeterminate and hence the axial forces calculated are at best approximate values.

In critical applications and during the testing of prototypes or to trouble-shoot repeated thrust bearing failures, it maybe essential to arrive at accurate thrust values. In such cases, thrust-measuring devices are installed. The internal pressures may also be recorded to get an idea of the hydraulic forces.

If the results are found unacceptable then design modifications maybe necessary.

The approximate method of axial thrust calculations is explained below. The following assumptions are made:

- Impeller profile is symmetrical.
- Impeller centerline coincides reasonably with the volute centerline (within 0.8 mm).

- Pressure on the front cover and back shroud is equal and constant from the impeller outside diameter to the wearing ring diameter.
- The pressure reduction, which is parabolic in nature, is ignored and average pressure acting on the shrouds is considered to be 3/4th of the pump discharge pressure.

### 4.1.1 Axial thrust in single-stage overhung impeller pumps

Axial thrust calculation in these simple pumps is made complex due to many variants in the design of the impellers and mating components. Axial thrust in some of the typical construction is discussed below.

#### Overhung impeller pumps: closed impeller with front wearing ring

Consider perfect impeller symmetry (Figure 4.1), the axial forces in the region above the diameter $D_1$ is balanced. The pressure on either side is assumed to be the same.

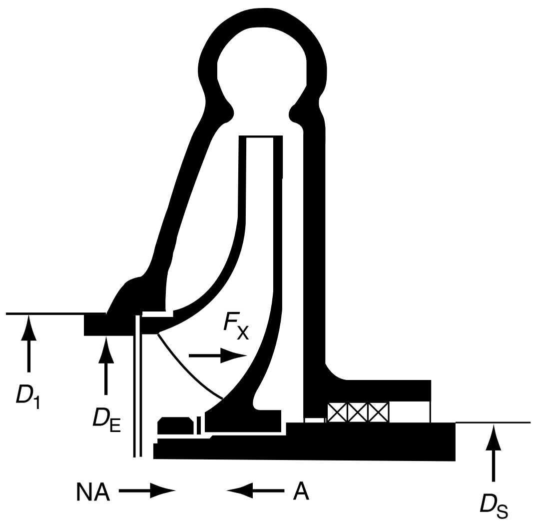



**Figure 4.1**
*Closed impeller with only front wearing ring*

The net axial thrust is due to the pressure difference acting on the area between the front wearing ring and the shaft diameter.

$$\text{NA Thrust} = A_S \times P_S + F_X$$

$$\text{A Thrust} = \frac{3}{4} \times P_D \times (A_1 - A_S)$$

Net Axial Thrust is the difference of the two.

$F_X$ – This is the axial force due to change in momentum. It is computed using the equation,

$$\frac{\left[\text{Density} \times Q^2 \text{ (Flow rate)}\right]}{A_E}$$

$Q$ is m$^3$/s, $A_E$ is in m$^2$ and Density in kg/m$^3$, and $F_X$ is in newton.

Areas, $A_1$, $A_S$ are obtained by the standard formula using their respective diameters.

$$\text{Area} = \frac{\pi}{4} \times D^2$$

### Overhung impeller pumps: closed impeller with wearing rings on both sides and provided with balancing holes

Consider a closed impeller with wearing rings on both sides as shown in Figure 4.2.
$D_1$ and $D_2$ – Wearing diameters on front cover and back shroud of impeller respectively
$D_E$ – Impeller eye diameter
$D_S$ – Pump shaft diameter.

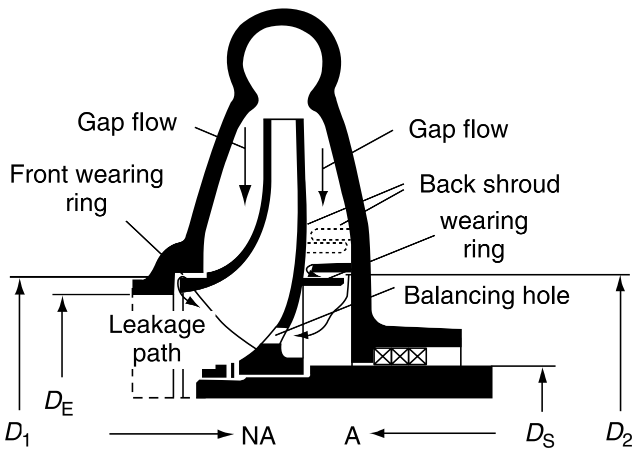



**Figure 4.2**
*Closed impeller with wearing rings on both sides and provided with balancing holes*

In this construction, the balancing holes drilled on the back shroud of the impeller play an important role in reducing the axial thrust. Due to these holes, the pressure in the region under the back wearing ring is reduced to suction pressure. However, some studies have indicated that the pressure in this region is 2–10% of the differential pressure depending on the diameter of the balancing holes.

The following equations are used to compute the axial thrust; the total area of balancing holes is considered to be eight times the area between the wearing rings.

When the wearing ring diameters, $D_1$ and $D_2$ are equal

$$\text{Axial Thrust} = A_S \times P_S - [0.03 \times P_D \times (A_1 - A_S)] + F_X$$

Where $F_X$ is the thrust due to momentum change.

When the wearing ring diameters, $D_1$ and $D_2$ are *not* equal

$$\text{NA Thrust} = A_S \times P_S - \left[0.03 \times P_D \times \left(A_1 - A_S\right)\right] + F_X$$

$$\text{A Thrust} = \frac{3}{4} \times P_D \times \left(A_1 - A_2\right)$$

Net Axial Thrust is the difference between the two.

### Overhung impeller pumps – closed impeller with back vanes

$$\text{NA Thrust} = A_S \times P_S + \left[0.00253 \times \left(A_R - A_S\right) \times \left(U_R^2 - U_S^2\right) \times \rho\right] + F_X$$

$$\text{A Thrust} = \frac{3}{4} \times P_D \times \left(A_I - A_S\right)$$

Net Axial Thrust is the difference of the two.

$U_R$ and $U_S$ are the peripheral velocity and can be calculated by $(\pi \times D_R \times N/60)$ and $(\pi \times D_S \times N/60)$ respectively

$$\text{All areas} - \text{A} = \left(\frac{\pi}{4} \times D^2\right)$$

In many cases, it is not preferable to install a wearing ring on the back shroud. In such cases, back vanes are integrally cast on the back shroud of the impeller.

The clearance between the back vane (in Figure 4.3: '$a-h$') and the casing is very important and is usually kept between 0.4 and 1 mm.

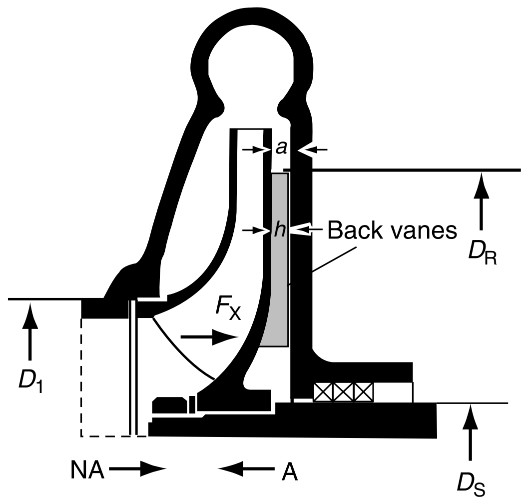



**Figure 4.3**
*Closed impeller with back vanes and no back wearing rings*

As the back vanes rotate with the impeller, they themselves act like impellers expelling liquid from the inner to the outer diameter. This helps in reducing the pressure at the back of the impeller and the axial force.

However, this method is unreliable, as it is dependent on the clearance $(a-h)$. If this gap increases, the pressure acting on the back shroud increases. This results in a higher axial thrust.

This design is less efficient than described earlier with back wearing rings. It is estimated that the back vanes can result in as much as 3% drop in efficiency of the pump.

## Overhung impeller pumps – open impeller with front wearing ring

Consider perfect impeller symmetry, the axial forces in the region above the diameter $D_1$ is balanced. The pressure on either side is assumed to be the same.

The net axial thrust is due to the pressure difference acting on the area between the front wearing ring and the shaft diameter (Figure 4.4).

$$\text{NA Thrust} = A_S \times P_S + \left[\left(A_3 - A_E\right) \times \left(\frac{P_D}{2}\right)\right] + F_X$$

$$\text{A Thrust} = \frac{3}{4} \times P_D \times \left(A_3 - A_S\right)$$

Net Axial Thrust is the difference of the two.

A semi-open impeller maybe designed with or without the back vanes based on the level of axial thrust generated in a pump.

Due to the absence of the front cover, the clearance between the impeller vanes and the pump casing is important. When back vanes are present, even the back clearance is important (Figure 4.5). This compounds the assembly problems.

$$\text{NA Thrust} = A_S \times P_S + \left[0.00253 \times (A_R - A_S) \times (U_R^2 - U_S^2) \times \rho\right] + \left[\frac{P_D}{2} \times (A_3 - A_E)\right] + F_X$$

$$\text{A Thrust} = \frac{3}{4} \times P_D \times (A_3 - A_S)$$

Net Axial Thrust is the difference of the two, where $A_3$ is the area of the impeller.

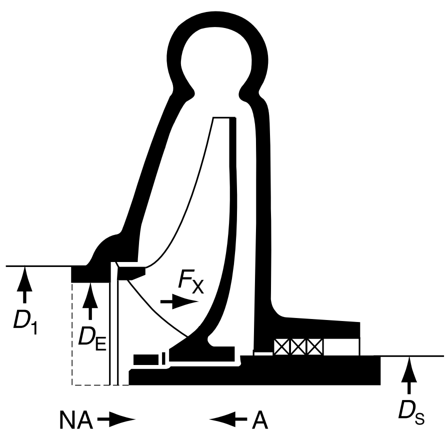



**Figure 4.4**
*Semi-open impeller with front wearing ring*

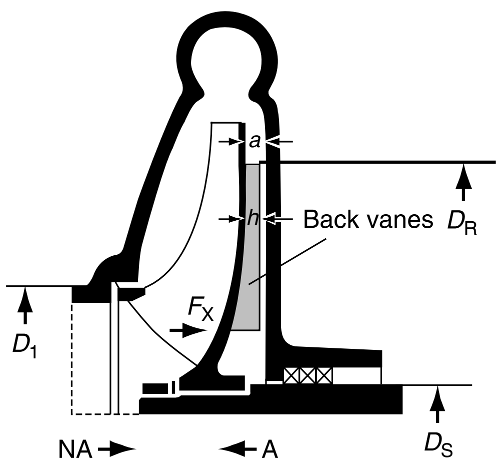



**Figure 4.5**
*Semi-open impeller with back vanes*

## 4.1.2 Double suction impeller

The double suction pumps equipped with double suction impellers, theoretically, develop little axial thrust due to the symmetry of their design.

However, the construction of such pumps can never be in perfect symmetry and therefore there is always some axial thrust generated which needs to be absorbed by a simple thrust bearing.

The symmetry of the design makes it necessary for the alignment of the impeller outlet and the volute center to be maintained. When this is violated, substantial thrust can be generated. This thrust is proportional to the axial displacement.

## Multistage pumps – stacked impellers

In this design of pumps, all impellers are stacked in one direction. As a result, the axial thrust pushing the rotor toward the pump suction can be quite high (Figure 4.6).

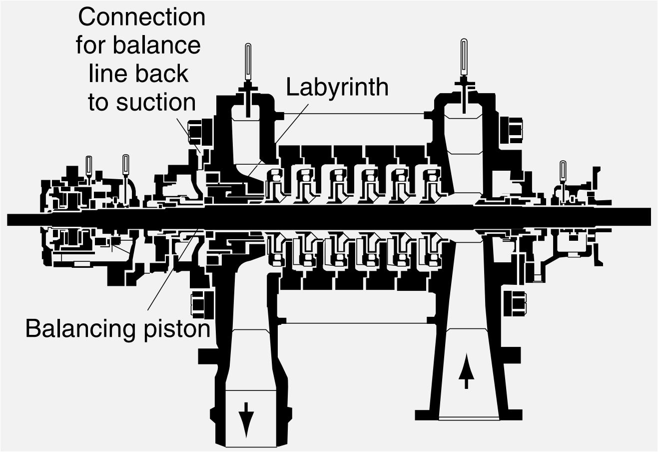



**Figure 4.6**
*Axial thrust balancing in a multistage stacked impeller design*

In these designs, a balancing drum or piston is provided. This is a rotating circular disk, which runs against a stationary corresponding disk or ring with a close clearance. The pressure of the liquid is dropped across this labyrinth passage. This leak is connected back to the suction of the pump.

The pressure difference across the drum is same as the net of the discharge and suction pressures. The force generated due to the drum is in a direction opposite to that acting on the rotor. In this way, the axial thrust is reduced.

This arrangement also helps to reduce the pressure acting on the stuffing box.

However, there is a loss of efficiency due to recirculation of the liquid through the balancing line.

## Multistage pumps – back-to-back impellers

In another arrangement in multistage pumps, impellers are arranged back-to-back. This simplifies the axial balance for an odd or even number of stages. In this design, there are two locations on the rotor that break down half total pump head. One is the center bushing between the center back-to-back impellers and the other is the throat bush in the high-pressure stuffing box. Sleeve diameters at these locations can be sized to provide axial balance for any number of stages.

It is preferable to have some residual axial thrust to keep the bearing lightly loaded and thus keep the rotor in one place rather than to allow it to float.

An increase in wearing ring or bush clearances does not affect axial thrust. It is recommended that all impeller rings should be of the same diameter and on even stage pumps, the center sleeve diameter should be the same as the sleeve diameters in each stuffing box.

To balance an odd number of stages, the simplest solution is only to change the diameter of the center sleeve.

## 4.2 Radial loads

In an end-suction centrifugal pump with an overhung impeller, the hydraulic radial load is due to the non-uniform velocity of the liquid within the pump casing or a volute. This unequal liquid velocity leads to a non-uniform pressure distribution of the pressure acting on the circumference of the pump impeller.

The radial load is most influenced by the design of the volute. The volute is designed to channel the liquid from the impeller to the discharge piping. The design of the volute is based on the flow at BEP. At this flow rate, the distribution of velocity and pressure of the liquid in the pump casing is uniform and there is negligible radial thrust on the impeller.

The minimal radial load at BEP is due to the cutwater of the volute. The cutwater causes a non-uniform pressure distribution on the circumference of the impeller that results in a net radial load on the impeller. This radial force is directed toward the cutwater.

At flows greater and less than the flow at BEP, the magnitude of the radial force increases. The direction of the radial force also changes as shown in Figure 4.7.

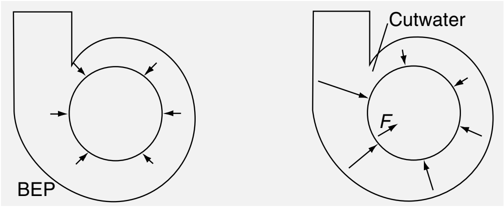



**Figure 4.7**
*Radial forces in a single volute for flows at BEP and greater than BEP*

Double volute casing as shown in Figure 4.8.

Double volutes are observed in large pumps. An additional rib is cast in the volute and this construction provides for two cutwaters. This can be seen in Figure 4.8.

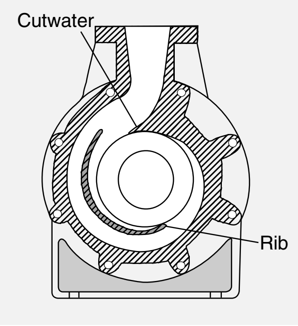



**Figure 4.8**
*Double volute pump*

The symmetry of the volute around the impeller due to rib results in an equal distribution of the forces and pressures acting on the impeller and thus the radial loads in this construction are reduced to a large extent.

The radial forces in such a construction over the range of operation is shown in Figure 4.9.

In concentric casing pump (covered under Section 2.2), uniform velocity and pressure distribution occur only at zero flow. The uniformity is progressively lost as the flow rate increases. At BEP, the non-uniformity is much higher than at no-flow situation.

This is quite the opposite of what happens in a conventional volute casing.

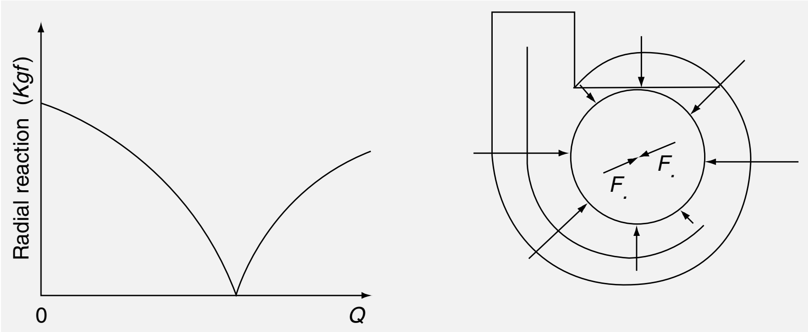



**Figure 4.9**
*Radial forces over the flow range of the pump with a double volute*

In a vaned diffuser pump, covered in Section 2.2, the velocity distribution around the impeller is more uniform and therefore has a much lower radial loads on the impeller.

For an end-suction single volute type of pump, the radial forces at shut-off conditions (the pump is running and the discharge valve is shut or almost shut) can be calculated using the following formula:

$$R_{SO} = K_{SO} \times \left[ \frac{(H_{SO} \times \rho)}{2.31} \right] \times D_2 \times B_2$$

At other points of operation, the following formula can be used to obtain an approximate value of the radial load on the impeller.

$$R = \left( \frac{K}{K_{SO}} \right) \times \left( \frac{H}{H_{SO}} \right) \times R_{SO}$$

Where

$K$ $= K_{so} \times [1 - (Q / Q_{BEP})^x]$
$R_{so}$ = radial thrust (in pounds) at shut-off
$R$ = radial thrust (in pounds) at operating conditions
$K_{so}$ = thrust factor at shut-off (obtained from Figure 4.10)
$K$ = thrust factor at operating conditions (see above formula)
$H_{so}$ = total head at shut-off (in feet)
$H$ = total head at operating conditions (in feet)
$\rho$ = specific gravity of the liquid
$D_2$ = impeller diameter (in inches)
$B_2$ = impeller width at the discharge including shrouds (in inches)
$Q$ = capacity at operating conditions (in gpm)
$Q_{BEP}$ = capacity at the best efficiency point (in gpm).

$x$ = An exponent varying between 0.7 and 3.3 established by testing. It is safe to assume a linear relationship between 0.7 at a specific speed of 500 and 3.3 at a specific speed of 3500.

Another approximate variation of the formula available for calculating the radial load is as given below:

$$R = K \times H \times D_2 \times \frac{B_2}{2.31}$$

Where

$K = 0.36 \times \left[1 - (Q / Q_{BEP})^2\right]$

$R$ = radial thrust (in pounds) at operating conditions

$D_2$ = impeller diameter (in inches)

$B_2$ = impeller width at the discharge including shrouds (in inches)

$H$ = total head at operating conditions (in feet)

$Q$ = capacity at operating conditions (in gpm)

$Q_{BEP}$ = capacity at the best efficiency point (in gpm).

In addition to the hydraulic radial loads generated mainly due to the pump construction and the operating point, there are also other factors that exert radial loads on the impeller.

A fluctuating radial load is sometimes due to the reduced clearance between the impeller vanes and the cutwater of the volute. The resulting vibration has a frequency, which is the product of the number of vanes on the impeller and the pump speed. This is called as the vane pass frequency.

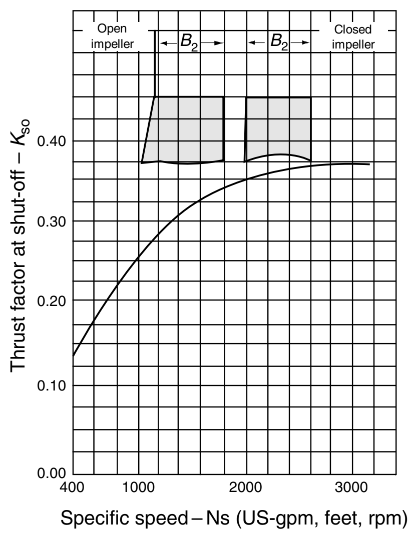



**Figure 4.10**
$K_{so}$ – *thrust factor at shut-off – (first formula)*

The mechanical imbalance of the pump rotor also generates a radial force on the pump shaft. Uneven flow through the impeller can also cause imbalance of the impeller.

## 4.2.1 Shaft deflection due to radial loads

A radial load such as that from hydraulic origin or due to mechanical imbalance causes the pump shaft to bend downward when it is in one position. When the shaft is rotated by

180°, it still bends downward in a similar way. This bending of shaft due to a constant load in one direction is called as deflection (Figure 4.11).

The shaft that is deflected rotates on its own centerline even though the centerline may not be straight.

This produces a reversal of stresses in the shaft in each revolution that could lead to fatigue cracking and an eventual breakage of shaft.

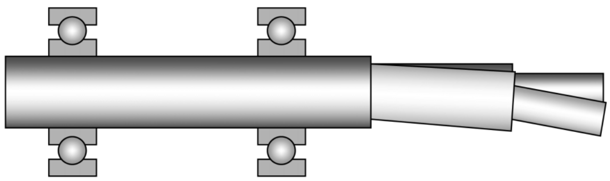


**Figure 4.11**
*Shaft deflection*

In addition to the above, there are several reasons why it is important to limit the distance that a shaft can deflect. The most important reason is that some of the rotating parts maybe exposed to the stationary parts. Stainless steel parts will gall as soon as rubbing contact is made that results in motor overload. Galling also tends to become progressively worse and may result in a complete seizure of the rotating elements.

Usually the surfaces which come into contact are the pump wearing rings or the throat bushing fitted to the seal housing and the shaft. The clearance between the wearing rings must be kept to a minimum in order to have a high-pressure drop across them. These surfaces are therefore the first to make contact. On pumps without the wearing rings, the first point of contact is usually of the shaft with the throat bush of the stuffing box.

The second effect of excessive deflection occurs in the stuffing box. Conventional pump packing when adjusted for the deflection under one operating condition will not readjust itself when the deflection is changed. For example, if the packing is set for operation at the high-capacity end of the curve and the discharge valve is throttled, the shaft would move from one extreme position to another, leaving a gap between the packing and the shaft. This gap coupled with the higher stuffing box pressure present under close to shut-off conditions would lead to excessive leakage.

Similarly, in pumps installed with mechanical seals, a shaft deflection can cause facial misalignment of the mechanical faces. This leads to an opening of the seal faces or an uneven mating of the seal faces, which then leads to an uneven and early wear of the seal faces leading to leakage.

There is often confusion between shaft deflection and another motion of the shaft termed ‘shaft whip’. In a shaft whip, the shaft end rotates in a manner to generate a cone (Figure 4.12).

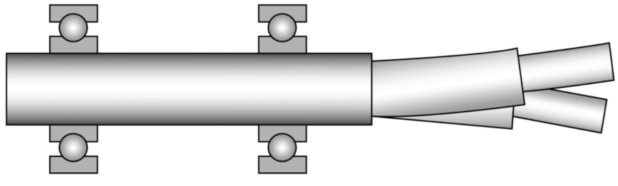


**Figure 4.12**
*Shaft whip*

In this case, the shaft centerline changes 180° from every 180° turn of the shaft. Shaft whip occurs due to rotor dynamic problems and the radial loads have little contribution to this phenomenon.

### 4.2.2 Calculating shaft deflection

Shaft deflection is calculated by treating the shaft as a cantilever beam using an expanded version of the beam formula (Figure 4.13). The formula for the deflection of a shaft with three major diameters in the overhung section is as given below.

$$\delta = \frac{P}{3E}\left[\frac{C^3}{I_C} + \frac{B^3 - C^3}{I_B} + \frac{A^3 - B^3}{I_A} + \frac{A^2 \times L}{I_L}\right]$$

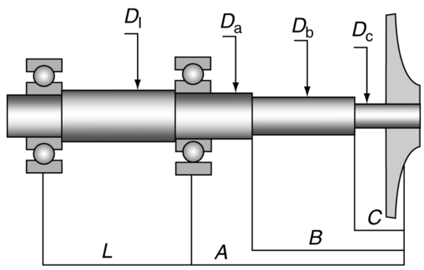



**Figure 4.13**
*Calculating shaft deflection*

Where

$\delta$ = shaft deflection

$P$ = radial resultant force

$E$ = Young's modulus − modulus of elasticity of the shaft material

$A, B, C, L$ = distances from impeller as shown in the figure above

$I_A, I_B, I_C, I_L$ = moments of Inertia at various diameters.

Consistent units have to be used.

The above formula can be reduced to state that shaft deflection $\delta$ is proportional to $L^3/D^4$, where $L$ is the length of the shaft and $D$ is the diameter of the shaft.

Pumps with lower $L^3/D^4$ would exhibit higher shaft deflections than pump shafts with higher $L^3/D^4$ ratio.

The shaft deflection at the mechanical seal faces should not exceed 0.05 mm. The reliability of the seal is affected in case the deflections are higher.

However, it is recommended that the $L^3/D^4$ ratio should not be used as a yardstick to compare the reliability of the pump vis-à-vis another pump. Instead, the actual shaft deflection values as obtained by the OEM for the most severe flow conditions should be used as a basis for comparison.

Thus, we have seen that the hydraulic loads are dependent on the type and size of the impeller, casing, the operating point of the pump, the suction, and the discharge pressures. The magnitude and direction vary greatly with the change in any of the above mentioned factors.

The variation of radial forces as a function of the operating point is shown in Figure 4.14. The comparison is between the radial forces generated in an ordinary volute and in a double volute.

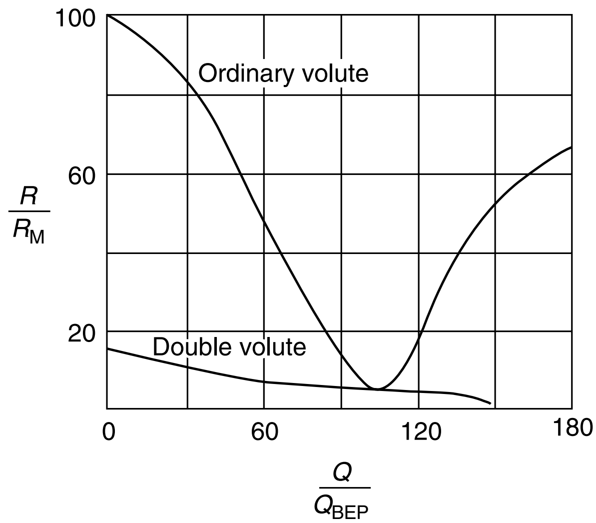



**Figure 4.14**
*Ordinary volute vs double volute*

These curves are for particular specific speed pumps. Changes in specific speed can modify the shape of these curves.

The $X$-axis is the percentage ratio of the flow rate and flow rate at BEP. The $Y$-axis is the percentage ratio of the radial force to the maximum radial force at shut-off conditions.

In addition to the radial forces due to hydraulic factors, cavitation, misaligned belts, and couplings also contribute to the radial loads on the pump shaft.

A complete shaft analysis is done considering the axial as well as the radial forces. These are then used to compute the reaction forces at the bearing locations.

Once having known the forces that would be acting on the bearings, a proper selection of bearings can be made to obtain maximum availability from the pump (Figure 4.15).

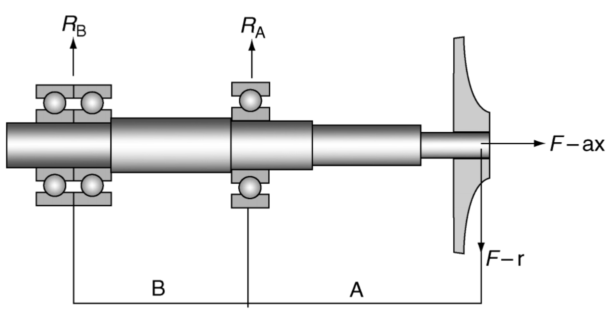



**Figure 4.15**
*The forces and reactions on a pump shaft*

The method to evaluate the bearing reactions is as shown below.

$$R_A = \frac{F_r \times (A + B)}{B}$$

$$R_B = F_r - R_A$$

At location A, there is only a radial bearing, thus it will withstand the force

$$P_A = R_A$$

At location B, there are double row angular contact bearings, which are meant to take the radial as well as the axial loads. Thus, they will take the combined load given by:

$$P_B = XR_B + YF_{\alpha x}$$

# 5

# Centrifugal pump operation and characteristics

One of the most attractive features of a centrifugal pump is its ability to perform in a system under a wide range of operating conditions.

Pumps are able to operate with satisfactorily under varying operating conditions and with a range of fluid types. These could be variations in:

- Flow rate
- Suction and discharge heads
- Speed
- Type of fluids handled and its many properties
- Fluids with varying properties.

The pump is designed for a discrete value of flow rate, differential head, and speed. This is the best efficiency point of operation or 'BEP'. However, in practical applications, the pumps are rarely operated at the operating parameters for which the pump has been designed.

Among the parameters stated, the flow rate $Q$ and differential head $H$ of the pump vary a great deal during normal operation.

Consider this case where a pump discharges into a delivery pipe that is connected to the bottom of a vessel situated at a certain height. As liquid is discharged into the vessel, the height of the liquid in the vessel increases and this increases the differential head that the pump has to generate. As the differential head $H$ increases, the flow rate of the pump $Q$ decreases.

If in the above case, the power requirement increases then the load on the motor could result in a possible drop in speed of the prime mover.

Other cases include where process and utility requirements to cover a greater range of flow rate and differential head may demand a simultaneous operation of multiple pumps within a single system. The performance of one pump in a common system is certainly affected by the behavior of the other pumps.

Even though a pump may perform with wide variations in the parameters and conditions, the pump performance is not unaffected by these changes.

It is therefore important to determine the behavior of the pump as it responds to the variation in the parameters under operating conditions that are different from parameters that were considered during the design of the pump.

The basics of the pump hydraulics have been covered in chapter 3 and this chapter would cover the behavior of the pump in response to the changes in its environment.

We may revisit some of the fundamentals covered in the earlier chapters to understand the variation in the behavior of the pump.

## 5.1 Behavior of hydraulic properties of pumps

The hydraulic properties of any centrifugal pump are studied taking the shaft speed of the pump $N$ as a constant.

After the speed is considered constant, the behavior of the differential head $H$ with respect to the flow rate $Q$ is obtained by throttling the discharge valve of the pump. The various openings of the discharge valve result in different flow rates and corresponding heads.

This experiment provides the relationship of $Q$ with $H$ that can be represented as Differential Head:

$$H = f(Q)$$

This is the fundamental characteristic of any centrifugal pump.

During this experiment, simultaneous readings of Power are noted and the efficiency values are also computed.

Even these can be represented as

Hydraulic power:

$$P = f(Q)$$

Pump efficiency:

$$\eta = f(Q)$$

The curves generated from the above functions are called 'performance curves' or the centrifugal pump characteristic curves.

The fourth characteristic representing the function of NPSH with respect to the flow rate $Q$ is a supplementary characteristic.

In the first case, the values of the flow rate $Q$ is plotted along the $X$-axis (absicissae). The units that maybe used are $m^3$/h, $m^3$/s, l/s, and l/min, US-gpm, Imp-gpm.

The values of differential head $H$ (meters, feet), Power $P$ (kW, HP), and Efficiency (as percentage, decimal fractions) are plotted on the $Y$-axis (coordinates).

The performance of most centrifugal pumps is given in terms of its capacity, discharge head, efficiency, and input power. Because these quantities are directly interdependent, a series of curves are used to express pump performance. Pump manufacturers provide the design characteristic curves for their individual pumps. These curves are then used as a basis for pump acceptance tests.

In Figure 5.1, the above-mentioned curves for some representative pumps are shown.

As the three graphs indicate, the specific speeds of the pump have quite an impact on the nature of these characteristic curves. The shape of the curves for each characteristic may appear different but the trends are similar. We shall discuss these after having a look at the curves.

### 5.1.1 Head–flow characteristics

One of the most important characteristics of a pump is its capacity; that is, the amount of fluid it moves per unit time. The capacity of a centrifugal pump decreases as the pressure at the pump discharge increases.

When the discharge valve is completely closed, the head developed by the pump is called the shut-off head. At this point, the pump will obviously not deliver any liquid. This is also the maximum head any pump can develop. The shut-off head is represented as $H_{so}$.

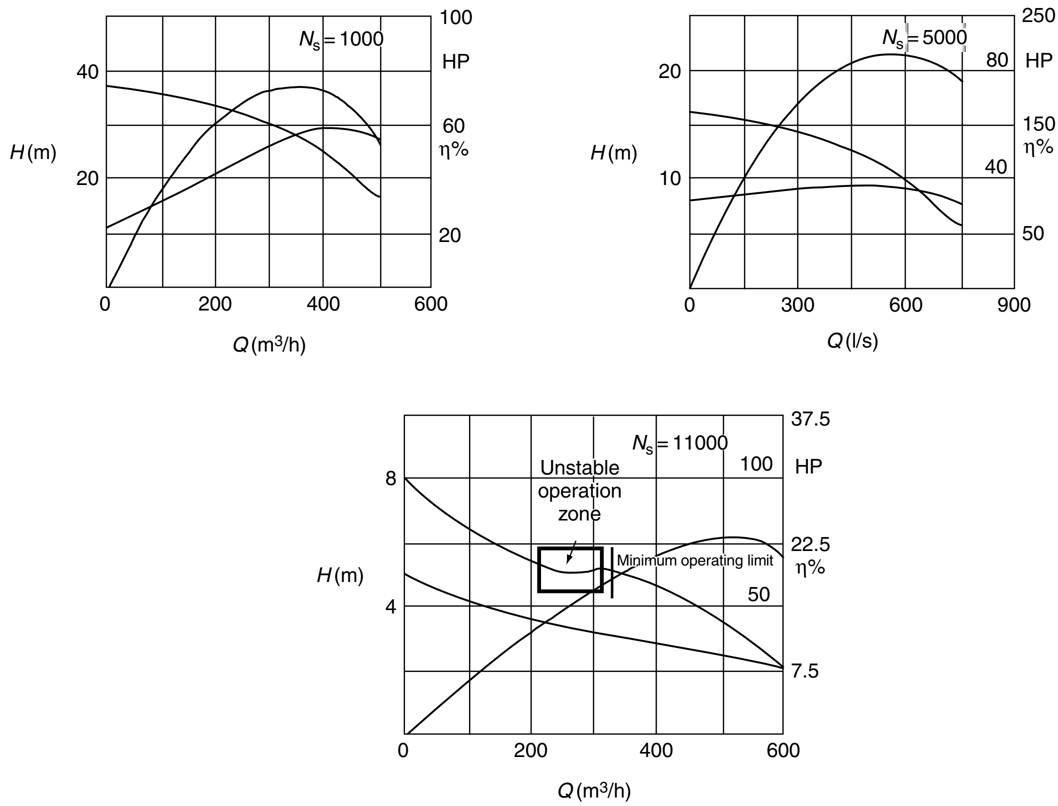



**Figure 5.1**
*Pump design characteristic curves*

Now, if the discharge valve is opened completely and the flow is directed into the atmosphere, the pump discharges at maximum capacity.

The shape of the *H–Q* curve as seen in the graphs in Figure 5.1 is dependent on the specific speed or the shape of the blades of the impeller.

The other factors that have a bearing on the shape of this curve are:

- Number of blades or impeller vanes
- The type of the pump casing (volute casing, diffuser ring, vane less guide ring, or concentric casing).

The *H–Q* curves of any pump are of two types:

1. Stable *H–Q* curve
2. Unstable *H–Q* curve.

## 5.1.2 Stable head–flow characteristics

Stable *H–Q* curves is one in which the differential head *H* progressively falls with the increase in flow rate *Q*.

This is shown in Figure 5.2.

It can be seen from the curve that $H_{so}$ is the maximum head that the pump will develop when it has stable $H$–$Q$ characteristics.

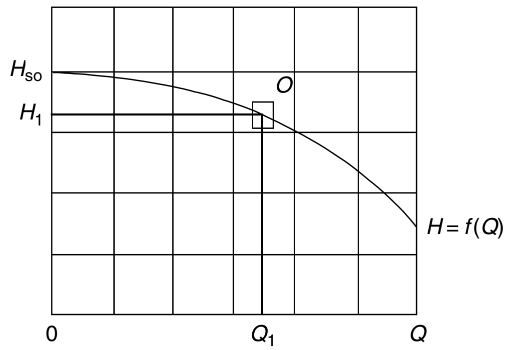



**Figure 5.2**
*Stable* H–Q *curves*

## 5.1.3 Unstable head–flow characteristics

In this type of curve, the $H$ generated by the curve starting from $H_{so}$ rises to a maximum value at point A and then falls progressively.

The drop in the curve indicates that for a single system resistance requirement of $H_1$, there can be two possible flow rates $Q_1$ and $Q_2$. This is not a desirable characteristic of a pump.

In actual operation, the pump with unstable characteristics and head requirement, that is greater than $H_{so}$, has a pump flow that oscillates between $Q_1$ and $Q_2$, leading to a modulating flow rate and possible vibrations in the pipeline (Figure 5.3).

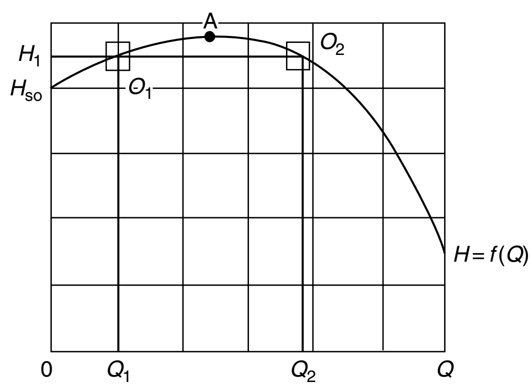



**Figure 5.3**
*Unstable* H–Q *curve*

In single-stage centrifugal pumps, the stability of the curve can be obtained by:

- Reducing the number of vanes/blades
- Modification of the vane geometry
- Reducing the vane outlet angle
- Modifications of the diffuser shape and return vanes in a multistage pump.

API 610, the design standard for centrifugal pumps recommends that all pumps shall have stable *H–Q* curves, which implies that the head, at rated capacity should continuously rise until it reaches the shut-off head.

In case of parallel operation of pumps, the head rise from the rated capacity to shut-off as in the case of single-/double-stage pumps shall be 10–20%. In multistage pumps, a lower percentage is recommended.

### 5.1.4 Steepness of the head–flow characteristics

The Steepness of the *H–Q* curve is usually associated with a stable *H–Q* curve. It is defined as percentage fall in head from shut-off conditions to BEP of the pump (Figure 5.4).

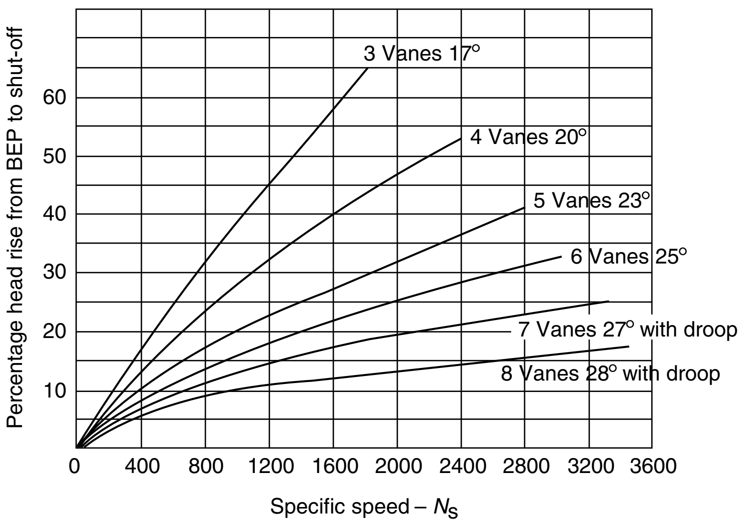



**Figure 5.4**
*Steepness of the* H–Q *curve*

This is represented by the formula given below:

$$\text{Steepness Percentage} = \frac{H_{\text{so}} - H_{\text{BEP}}}{H_{\text{BEP}}} \times 100$$

The number of vanes and blade outlet angle determine the steepness of the *H–Q* curve. The relationship of the above factors with the specific speed of the pump is shown in Figure 5.4. The figure represents some of the practically used combinations of vane number and outlet blade angle in the industry.

Another way of expressing steepness of the *H–Q* curve is by using the ratio of the change in differential head $H$ to the change in flow rate $Q$, $\Delta H/\Delta Q$.

In *H–Q* curves, where $\Delta H/\Delta Q$ is smaller, the curves are said to have flat characteristics and while those with a higher value of $\Delta H/\Delta Q$ are called as steep characteristics.

In most cases, the application of the pump determines whether the curve should be of the flat or steep type.

For example, pumps used in the firewater service need to have very flat characteristic curves.

Consider this scenario, where a pump is feeding water to a header system that is being used to extinguish fires. One or many fire hydrants maybe opened simultaneously to fight the fire and the required flow rate may increase sharply. As the flow from the pump increases it is important that the pressure from the pump should not drop substantially. A drop in head delivered may affect the height at which the water has to be sprayed.

It is observed that as the specific speed of the pump goes on increasing the *H–Q* curves keep getting steeper.

In addition, in high specific speed pumps, there is a relatively steep slope toward shut-off and beyond BEP.

The procedure for selecting an axial flow pump is much the same as for any other process pump; however, the duty point is limited to a range within 75–115% of the pump's BEP. This is more critical with the axial flow designs than with radial vane designs because the hydraulics become unstable at 70–75% of BEP (shown in the Figure 5.4).

This phenomenon may also be seen in mixed-flow impeller pumps. As a result, this flow from such pumps is rarely throttled. They are also not started with a closed discharge valve. It will be seen in the subsequent section that the power curves too favor the starting of such pumps with an open discharge valve.

## 5.1.5 Power–flow characteristics

The nature/shape of power–flow rate characteristic curves are also dependent on the specific speed of the pump.

Centrifugal pumps having low to medium specific speeds have *P–Q* curves that rise upward. For higher specific speeds, the *P–Q* curves maybe approximately flat and horizontal (Figure 5.5). In case of propeller (axial flow) pumps, which have very high specific speeds, the power for the pump falls as the flow rate increases.

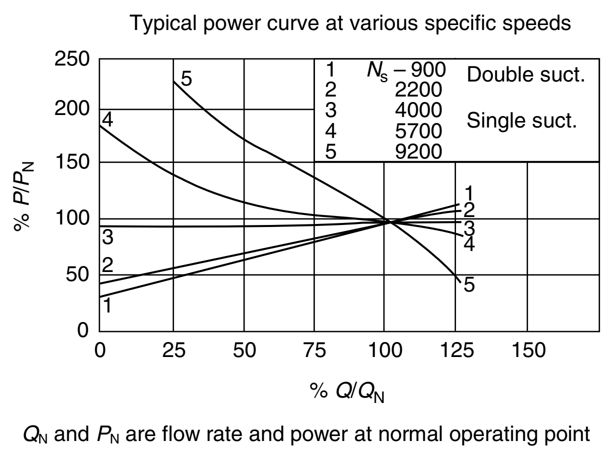



**Figure 5.5**
*The* P–Q *curves as a function of specific speed of pumps*

The pumps that are driven by AC induction motors should be started with minimum load on them so as to limit the starting current.

The above characteristics indicate as to how the pumps should be started so the above condition is fulfilled.

Therefore, it is recommended that radial impeller pumps be started with discharge valves closed whereas propeller pumps be started with discharge valves fully open.

The pump *P–Q* curves are of two types:

1. Non-overloading
2. Overloading.

These terms are derived from the fact that the demand of power by the pump increases or decreases with the increase in flow rate.

In mixed flow and axial flow pumps, it is observed that the power curve remains flat or tapers downward. Thus, it is rare that overloading of the motor occurs due to increase in flow rate by the pump. These curves are called 'Non-overloading' *P–Q* curves.

In centrifugal pumps with radial impellers, the power curves rises with an increase in flow rate. Thus, when the system resistance on the pump drops, there is a tendency for the motor to trip on account of motor overload.

The *P–Q* curves of such pumps are called 'Overloading' *P–Q* curves.

### 5.1.6 Efficiency–flow characteristics

The efficiency vs power curve $\eta - Q$ curve initially rises to a peak, which as mentioned earlier is called as the BEP. Subsequently, as the flow rate is increased the efficiency drops.

It is observed that this drop in efficiency occurs after BEP gets sharper with the increase in specific speed.

Thus, a radial impeller curve will show gradual drop in efficiency post BEP whereas it would be quite steep in the case of an axial flow pump.

## 5.2 Non-dimensional characteristics

In the world of pumps, there are numerous varieties of pump types and sizes. The various applications and requirements call for a large variety of constructional designs, and a wide range flow rates, heads, powers, and efficiencies.

Changes in design results in differences in pump characteristics and gives rise to a wide range of hydraulic properties; it often becomes difficult to compare one type of pump with another.

To overcome this problem, non-dimensional characteristics are defined.

In order to determine the dimensional characteristics of a geometrically similar pump based on a known dimensional characteristic of another pump, it is necessary to transform the known characteristic into a non-dimensional characteristic.

To work out a non-dimensional characteristic, the normal operating point (the point at which the pump is supposed to operate at most of the times) is considered. The Flow rate, differential head, power, and efficiency are represented as $Q_n$, $H_n$, $P_n$, and $\eta_{n.}$

These values are considered as 100% and the values deviating from the normal are expressed as percentages of the normal values.

Once the $Q_n$, $H_n$, $P_n$, and $\eta_n$ are known of the desired characteristics, it is easy to calculate the other values necessary for the drawing of the characteristic curves.

The power–flow rate curve shown under Section 5.1 is an example of the use of non-dimensional characteristics.

There are basically two types of non-dimensional characteristics that are commonly used. These are:

1. Individual characteristics
2. Universal characteristics.

The Individual characteristics are derived from the dimensional characteristics and these are that have been discussed above.

The Universal characteristics are basically derived from the similarity of centrifugal pumps.

These are the:

- Head coefficient
- Flow coefficient.

Head Coefficient is represent by $\psi$ and is obtained by the formula,

$$\varphi = \frac{H}{u_2^2/2g}$$

Flow Coefficient is represented by

$$\phi = \frac{Q}{u_2 A}$$

Where

$u_2$ is the peripheral velocity at impeller outlet

$A = \pi/4 \times d_2^2$ – Area of impeller.

Universal non-dimensional characteristics $\psi = f(\phi)$ makes it possible not only to compare different quantities of the same type but also enables to test pump using others fluids like air.

## 5.3 The cause of the *H–Q* curve

To understand the nature of the *H–Q* curve of a radial impeller pump, we have to learn the velocity triangles of the liquid flow at the outlet of the impeller.

The three cases of flow rate $Q$ less than, equal to and greater than $Q$ at BEP are shown in Figure 5.6. The changes in the corresponding vectors are also indicated.

The main velocities are:

- *Peripheral velocity*: $U_2 = \pi \times N \times d_2 / 60$ – As speed is constant, $U_2$ is constant
- *Absolute velocity*: $C_2$
- *Relative velocity*: $W_2$ – Direction is along the vane outlet angle
- *Radial velocity at impeller discharge*: $C_{m_2} = Q \times$ Area.

When the flow rate $Q$ changes, the following do not change:

- Peripheral velocity at the outlet $U_2$, as the speed $N$ is constant
- Blade outlet angle $\beta_2$, contained between the direction of the relative velocity $W_2$ and the vector $U_2$.

At the same time, the relative velocities of flow $W$ through the impeller passages shall increase or decrease based on variations in the flow rate. As a result, the magnitudes of the vectors forming the velocity triangle change. The shape of the parallelogram of velocities changes, and this changes the peripheral component of the absolute velocity $C_{u_2}$. This produces a change in the theoretical differential head $H$ given by

$$H = U_2 \times \frac{C_{u_2}}{g}$$

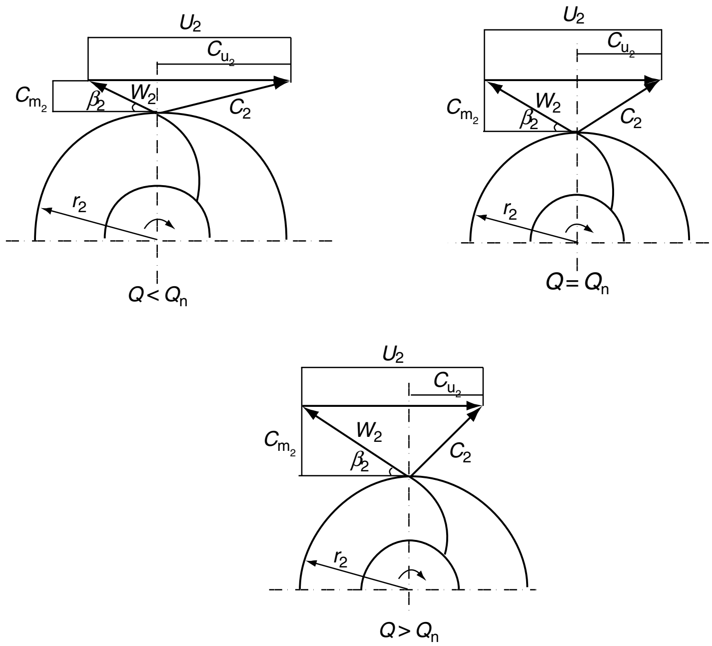



**Figure 5.6**
*Velocity triangles at outlet of impellers*

From the comparison of the outlet velocity triangles, it follows directly that when the flow rate is reduced the circumferential component of the absolute velocity $C_{u_2}$ increases, and the meridional component $C_{m_2}$ decreases. Increasing the flow rate $Q$ reduces the component $C_{u_2}$ and thus this reduces the differential head $H$ as developed by the pump.

In this manner, it is explained that when the flow rate increases the head developed by the pump decreases, hence the reason for the $H$–$Q$ curve.

The power absorbed to pump the liquid is directly proportional to the product of flow rate $Q$ and the head developed $H$. The changes in $Q$ and $H$ described determine the power of self-regulation of impeller pumps.

This is an essential and valuable feature in the exploitation of impeller pumps, since if the total head of the pump increases during operation; the pump automatically reacts by reducing the discharge so that the impeller can overcome the increased resistance.

Conversely, a reduction in the resistance in the delivery pipe stimulates the pump to increase the discharge.

## 5.4 The inlet velocity triangle

When the flow rate of a pump changes there are changes in the liquid flow direction even at the inlet of the impeller (Figure 5.7).

The three main velocities are:

1. *Peripheral velocity*: $U_1 = \pi \times N \times d_1 / 60$ – As speed is constant, $U_2$
2. *Absolute (Meridional) velocity*: $C_{m_1} = Q \times \text{Area}$
3. *Relative velocity*: $W_1$.

When the pump flow rate changes, there is a change in the angle of entry of the liquid, $\beta_{1-\text{en}}$. It is smaller than the blade inlet angle $\beta_1$, when the flow rate is less than the flow at BEP.

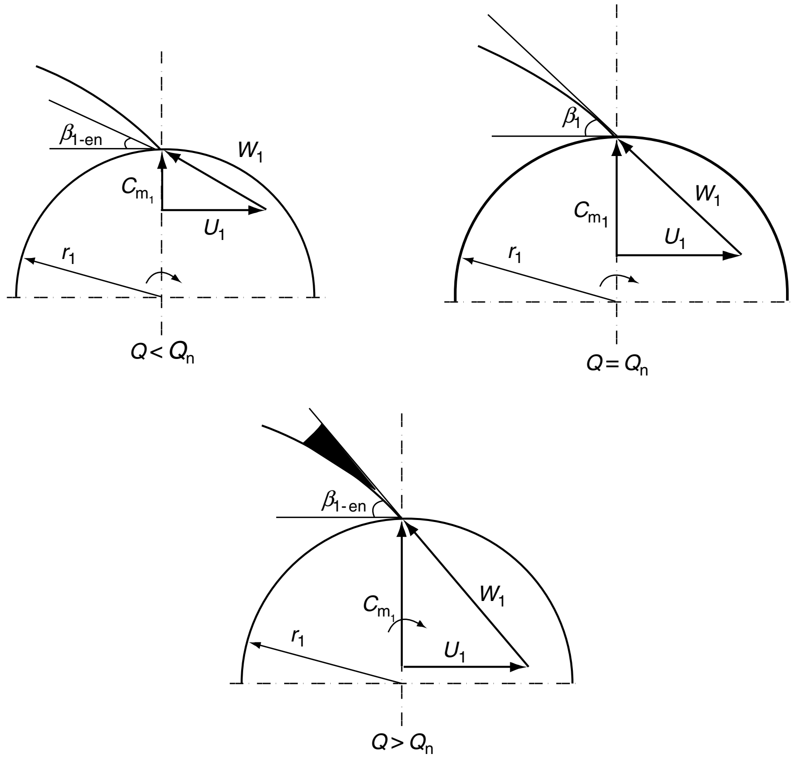



**Figure 5.7**
*Inlet velocity triangle*

At BEP, the flow is such that the liquid inlet angle is the same as the blade inlet angle $\beta_1$.

Similarly, when the flow rate $Q$ is more than the specified flow rate at BEP, the liquid inlet angle $\beta_{1-\text{en}}$ becomes greater than the blade inlet angle $\beta_1$.

During conditions of flow rate less than or greater than BEP flow rate, flow separation occurs and leads to the formation of eddies.

At $Q < Q_{\text{BEP}}$, eddies are formed on the inner surface of the vane; the converse happens in case of $Q > Q_{\text{BEP}}$, when eddies are formed on the outer surface of the vane.

In practical cases, a certain pre-whirl is present and this tends to reduce the difference between $\beta_{1-\text{en}}$ and $\beta_1$ at flow rates less than or greater than flow at BEP. Thus, the losses are less than that assumed by a straight flow into the impeller.

The velocity triangles at inlet provide a good insight to the cause of the losses that occur at flow rates other than that at BEP.

These flow rates other than BEP also cause disturbances and hydraulic losses in the diffuser ring and the volute casing.

It is described that such hydraulic losses are proportional to the cube of the flow rate.

## 5.5 The cause of the *P–Q* curve

As described earlier, hydraulic losses occur for every flow rate other than the flow rate at the BEP. As the flow rate moves further from the BEP in either direction the losses increase. These losses as mentioned are proportional to the square of the flow rate.

The other two losses that contribute to the pump inefficiency are recirculation losses and mechanical losses. None of these losses is grossly affected with the change in flow rate.

Thus, there is a certain value of the flow rate of the pump when sum of the losses are minimum. Thus, efficiency is at a maximum when the losses are minimal. At every other flow rate the efficiency is lower.

This explains the inverted U shape of the *P–Q* curve of a centrifugal pump.

## 5.6 The effect of speed changes on characteristic curves

All the discussions in the preceding chapters have been based on the prime assumption that the speed of the pump is constant. In many practical applications, pumps operate with variable speed drives and in this chapter, we will see the effect of change in speed on the characteristic curves on pumps.

In Figure 5.8 the *Q–H* curve-b at the normal speed *N* is a typical *Q–H* curve of a centrifugal pump.

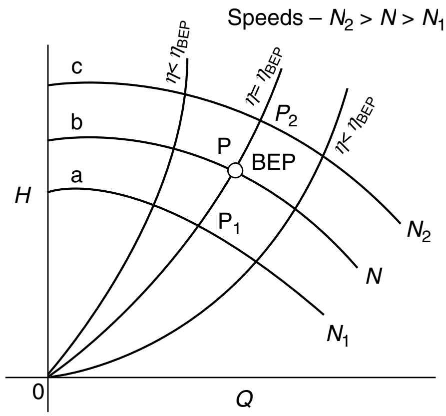



**Figure 5.8**
*Speed changes effects on characteristic curves*

When the speed of the pump is changed, the flow rate *Q* and the head developed *H* also changes.

If we consider the velocity triangles at the inlet and outlet, they remain similar and all the velocities vary in proportion to the speed of the pump.

The flow rate is direction proportional to the speed *N*.

$$\frac{Q_1}{Q_2} = \frac{N_1}{N_2}$$

The total heads is proportional to $u_2^2$, and hence it is proportional to $N^2$.

$$\frac{H_1}{H_2} = \left(\frac{N_1}{N_2}\right)^2$$

The power P of a pump is proportional to the product of flow rate *Q* and head developed *H*. Therefore, P is proportional to $N^3$.

$$\frac{P_1}{P_2} = \left(\frac{N_1}{N_2}\right)^3$$

The relationships derived above help to establish the characteristics of the pump at different speeds, once the *Q–H* curve at the normal speed *N* is known.

The relationships derived are based on a major assumption that the efficiency remains constant when transferring a point on one characteristic to the homologous point on another characteristic.

The results obtained by these equations often differ from the values obtained in practical application. The difference becomes larger when the speed change is more than ±25%.

Let us consider a point $P_2$ on the *Q–H* curve-c in Figure 5.8. The homologous points P and $P_1$ on the *Q–H* curve-b and *Q–H* curve-c are given by

$$Q_1 = \frac{N_1}{N} \times Q \qquad Q_2 = \frac{N_2}{N} \times Q$$

$$H_1 = \left(\frac{N_1}{N}\right)^2 \times H \qquad H_2 = \left(\frac{N_2}{N}\right)^2 \times H$$

Resolving the above equations, we get:

$$H_x = \left(\frac{Q_x}{Q}\right)^2 \times H$$

The above equation is of the form of a parabola passing through the point P and the origin 0. These are shown in Figure 5.8.

The operating P corresponds to a certain specific speed given by:

$$N_s = \frac{N \times \sqrt{Q}}{H^{\frac{3}{4}}}$$

If we substitute the values of *Q* and *H* by their proportional speeds we get a constant as shown below.

$$N_s = \frac{N \times c_1 \sqrt{N}}{c_2 \times N^{2\frac{3}{4}}}$$

$$N_s = \frac{c_1}{c_2}$$

We thus obtain the relationship that

$N_s$ = constant

Thus, the parabola passing through homologous points $P_1$, P, and $P_2$, bound by the affinity relationships, have the same specific speed.

These are called as Constant Specific Speed Curves.

As efficiency of the pump is a direct function of specific speed, the parabola containing homologous points have the same specific speed and same efficiency.

## 5.7 The complete characteristic curve

Thus, if we are to integrate all the characteristics in a single three-dimensional figure, it would look in the manner shown in Figure 5.9. The coordinates are *H*, *Q*, and $\eta$.

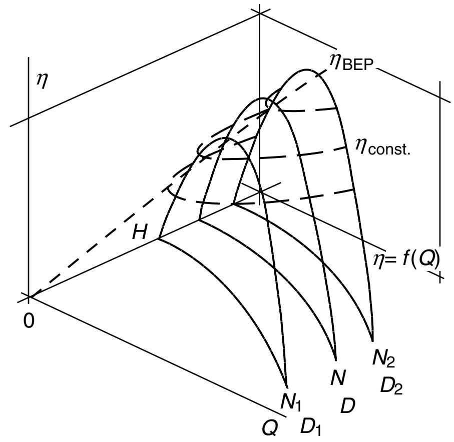



**Figure 5.9**
H–Q*, and efficiency curves*

The curves represent the function $H = f(Q)$ and $\eta = f(Q)$ are plotted along their respective axes. The *Q–H* curve varying as a function of impeller diameter or pump speed is also represented in this plot.

The new curves as seen in Figure 5.9 are the $\eta_{const}$ curves called as the iso-efficiency curves. However, we could even flatten these into a two-dimension figure as shown in Figure 5.10.

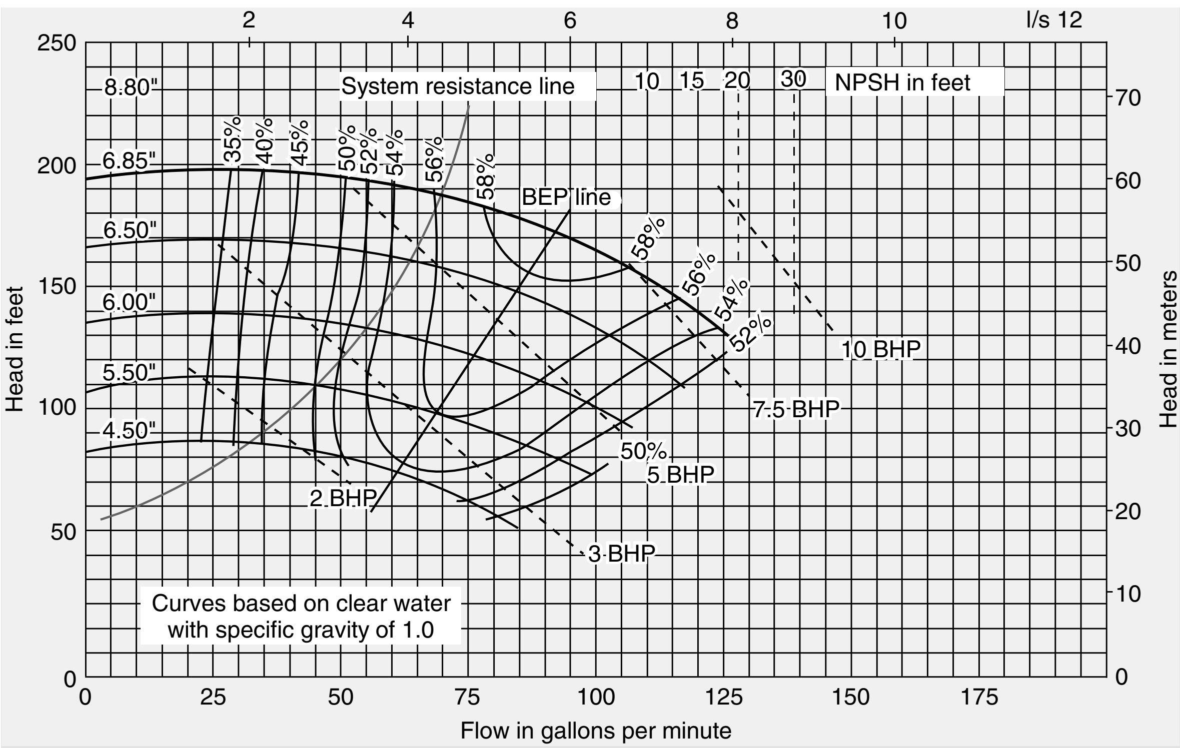



**Figure 5.10**
*Iso-efficiency curves*

## 5.8 Multiple pump operation

A process application may require operation of more than one in a single system. There could be a case where several pumps would discharge their flows in a common delivery pipe. The configuration of such pumps is called as Parallel Operation.

There could also be a case when a single pump cannot meet the total head requirement of the system and it has to be shared by various pumps. The configuration of such pumps is called as Series Operation.

When pumps are operated in parallel, they work against a common pressure. In this case, the flow rates are added. For a pump in series, the heads are added for a common total head.

Multiple pump operations are sensitive to the individual characteristics of the pump. The total flow rate or the total head developed may not be a simple addition of the individual flow rates and heads developed. This is especially in cases where the pumps do not have similar characteristic curves.

Before we take up the various cases of operation, it is recommended to the reader to revisit the topic on System Resistance covered under Section 3.6.

### 5.8.1 Parallel operation with similar *Q–H* curves

When several pumps with similar *Q–H* characteristic curves feed into a common discharge, the combined *Q–H* characteristic curve is obtained by simply adding up the abscissae corresponding to all pumps.

If two pumps with the same *Q–H* curve are operated in parallel, the combined curve is obtained by doubling the abscissa for each point on the individual pump characteristic, while keeping the total value of the head as constant.

When a single pump operates, the flow rate is say $Q_1$. The combine *Q–H* curve is obtained by doubling the abscissa for the same head, as shown in Figure 5.11.

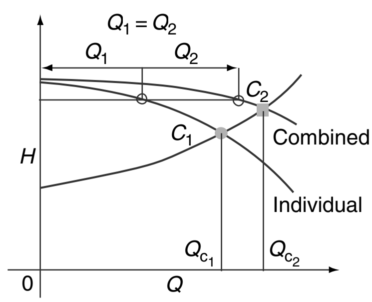



**Figure 5.11**
*Parallel operation of two pumps with similar characteristics*

It would be normal to assume that this would be the operating point of the pump when these are operated in parallel. However, this is not the case.

The point of operation is dependent on the system resistance.

In a parallel operation with a system resistance curve as shown in the figure, a single pump would deliver a flow rate of $Q_{c_1}$ When the next pump is also started, the system resistance takes a turn upwards and intersects the *Q–H* curve of the combined flows at a point, $Q_{c_2}$ This is due to the square relationship of the system resistance with the flow rate.

As this can be seen from the figure, $Q_{c_1}$ is not exactly half the flow rate of $Q_{c_2}$ .

In case there is a need to regulate the flow of the system, it is more efficient to throttle only one pump.

Pumps in parallel are used whenever a flat type of characteristic curve (small d$H$/d$Q$) is required. That is, the pump discharge head decreases gently with an increase in flow rate.

The combined efficiency for two pumps in parallel is given:

$$\eta = \frac{H \times (Q_1 + Q_2)}{K \times (P_1 + P_2)} \times 100$$

Where

$\eta$ = combined efficiency [%]
$H$ = total head
$Q_1$ = flow rate for pump 1
$Q_2$ = flow rate for pump 2
$K$ = unit conversion constant
$P_1$ = brake power input for pump 1
$P_2$ = brake power input for pump 2.

The combined NPSH-r for pumps in parallel is equal to that for the most limiting pump, the largest NPSH-r in this case.

### 5.8.2 Parallel operation with different *Q–H* curves

Figure 5.12 shows how the two pumps with dissimilar *Q–H* characteristic curves. These are represented by curves 1 and 2.

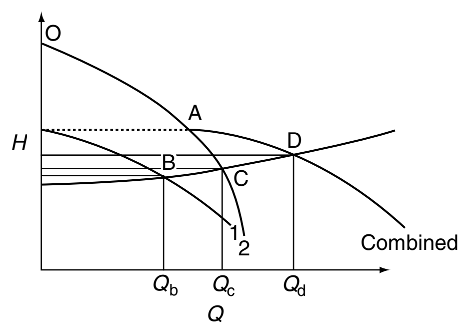



**Figure 5.12**
*Parallel operation with different* Q–H *curves*

The combined curve of such a combination is represented by OAD. However, until the operating point reaches A, the first pump does not deliver any flow as it sees for itself shut-off conditions.

It is only after point A that pump 1 can begin to contribute to the total flow.

It is for this reason that in such a case, the pump 2 should be started first and then pump 1.

### 5.8.3 Parallel operation with flat and steep *Q–H* curves

There are three possible combinations of pump operations with the flat and steep type of characteristics.

These are:

1. Both pumps have flat characteristics
2. Both pumps have steep characteristics
3. One flat and steep curves.

The Figures 5.13 and 5.14 are plotted for pumps operating in parallel and having similar flat and steep characteristics respectively.

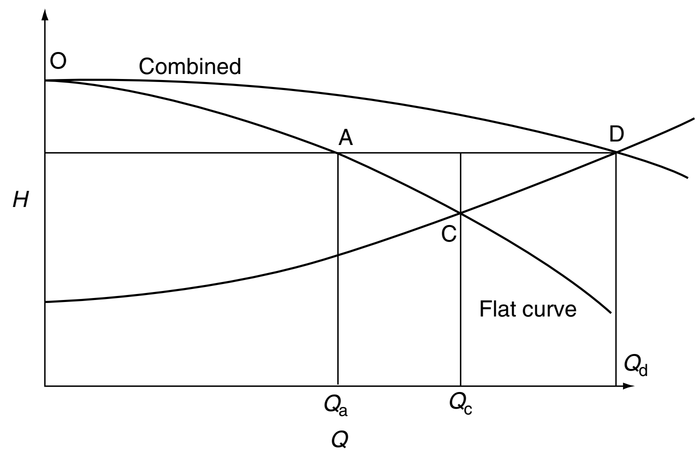



**Figure 5.13**
*Pumps with flat curves*

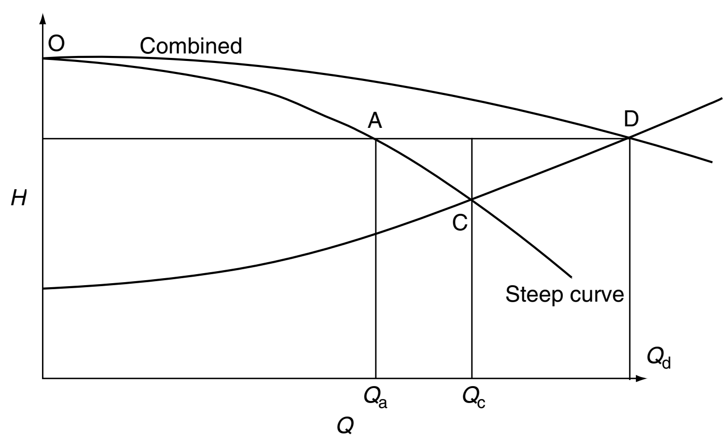



**Figure 5.14**
*Pumps with steep curves*

As shown in the figures above the combined $Q$–$H$ characteristics intersect the system resistance curve at point D. For the same differential head, the individual pump characteristic would intersect at point A.

It can be observed that in case of a flat curve, the difference ($Q_c$–$Q_a$) would be greater than ($Q_c$–$Q_a$) of a pump with steep characteristics.

The last combination is already covered in the previous section.

### 5.8.4 Series operation of pumps

When two pumps with similar characteristics are operated in series, the combined curve is formed by doubling the head ordinate for each point on the individual pump characteristic, while keeping the same value of flow rate *Q* (Figure 5.15).

For pumps with different characteristics, the method is similar.

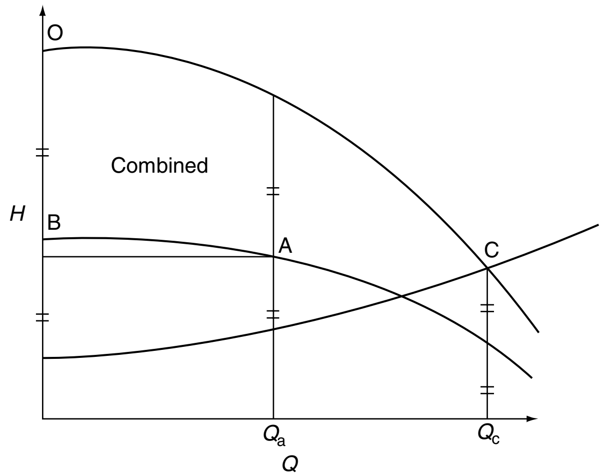



**Figure 5.15**
*Series operation of pumps*

When two pumps are connected in series, it follows that at any given instant the rates of flow through the two pumps must be same. The total head of the two pumps on the other hand, are only equal if the individual head discharge characteristics of the pumps are identical.

Series operation of single stage pumps is seldom encountered; more often, multistage centrifugal pumps are used.

## 5.9 Pump characteristics – viscous liquids, liquids with considerable solids

In Section 3.10, 'corrections', we have discussed as to how the behavior of the pump is affected while handling liquids with higher viscosity.

We had also seen the Viscosity Correction Chart provided by the Hydraulic Institute to look up the values of coefficients of flow, head, and efficiency. The coefficients corrected the pump characteristics obtained with clear liquid to the derated performance as expected while pumping viscous fluids.

An example of performance deterioration with increase in liquid viscosity is shown in Figure 5.16.

When solids are suspended in liquids, the characteristics of a pump begin to differ from those obtained for pumping clear liquids.

Solids in liquids tend to increase the hydraulic resistance in proportion to the concentration of the solid in the liquid.

As a result, both the pump flow rate and the head developed are lower. The power required is higher and consequently, the efficiency of the pump is lower.

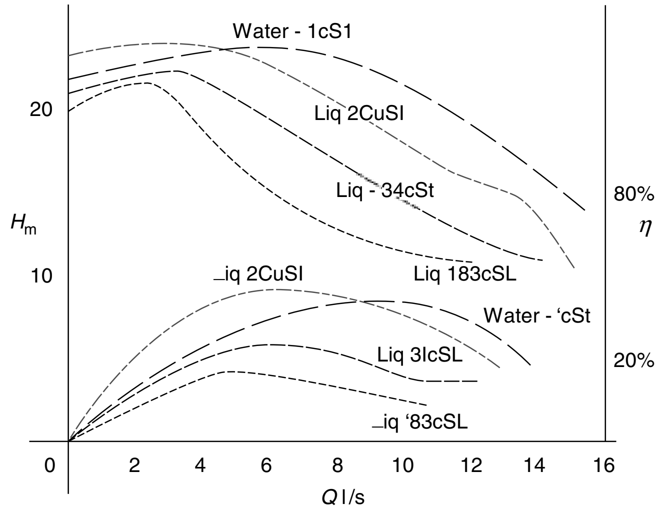



**Figure 5.16**
Q–H *and* Q–η *characteristic comparison of liquid with increasing viscosities*

If the solid particles are fine in nature, they tend to form a homogenous mixture with the liquid, the shape of the pump characteristics begin to seem like a pump handling a liquid with higher viscosity.

The *Q–H* curves for pumps transporting mixtures of water and solids such as sand, slag, sugar beet, potatoes, fishes etc. can only be determined by experiments.

The *Q–H* curves while pumping various mixtures formed by various percentages of solids in the liquids can be compared with that of clear water and nomogram can be created. This nomogram can then be used to predict the pump characteristics for a particular mixture.

## 5.10 Pump characteristics – abnormal operation

The normal operation of a pump is considered when direction of rotation of the pump in accordance with the backward vanes of the impeller. The flow of the pump is from the suction (lower head) to the discharge (higher head).

The pump characteristics considered in the preceding sections are based on the normal operation of the pump as described above and this shown in the right corner of Figure 5.17.

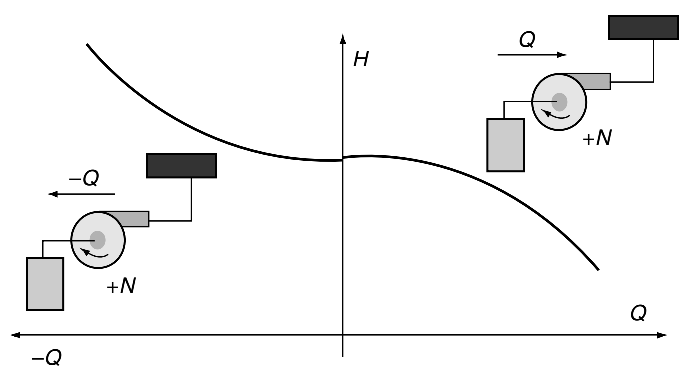



**Figure 5.17**
*Characteristics of pump in normal direction of rotation* (+N) *and flow rate* (+Q)

However, in practical applications there exist other possibilities in which the pump maybe operated in a manner different from the normal case. These cases can be witnessed during start, stop, or during operation of multiple pumps in a single system.

There are four possible cases whose characteristics when plotted form the basis of the Complete Characteristics of the pump.

1. *Normal direction of rotation (*+N*) and flow rate (*+Q*).*
2. *Normal direction of rotation (*+N*) and reverse flow rate (*–Q*)*: As the discharge head of the pump increases, the flow rates continues to drop until it reaches a stage of no flow of shut-off conditions. If the discharge head continues to rise (possible in parallel operation of pumps), it results in a backflow of liquid. Thus, though the shaft rotates in the normal direction, the flow rate is in the reverse direction.

   This is shown in the left corner of Figure 5.17. It can be seen that the discharge head above shut-off increases the reverse flow rate also rises.
3. *Reverse direction of rotation (*–N*) and normal direction of flow rate*: This case is quite common when the phase terminals of an induction motor are changed. The direction of rotation of the motor reverses and in such a case so does the pump. Even with reverse direction of rotation, the pump discharges the flow from the suction to the discharge. However, the head developed and the flow rate is substantially less. This is because of the poor pump efficiency in the reverse direction of rotation. This is shown in the right corner of Figure 5.18.
4. *Reverse direction of rotation (*–N*) and reverse direction of flow rate (*–Q*)*: As the discharge head increases, the liquid begins to flow from the discharge to the suction at a steadily increasing rate. This is the case of a hydraulic turbine, where the liquid head drop across the impeller converts into mechanical work. Therefore in such a case, the shaft should have a brake else the pump can acquire very high speeds. This is often seen in a system with no or passing non-return valves in the pump discharge and the pump motor trips. The liquid begins to flow in the reverse direction and the impeller too spins reverse. This case is shown in the left corner of Figure 5.18.

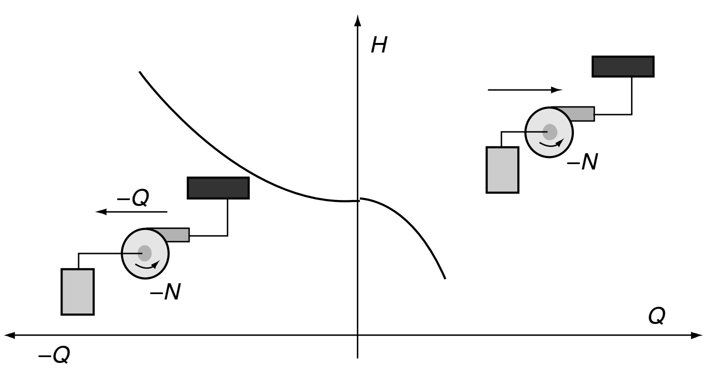



**Figure 5.18**
*Characteristics of pump in normal direction of rotation (*+N*) and reverse flow rate (*–Q*)*

## 5.11 Pump characteristics – speed–torque curve

Centrifugal pumps are driven machines. They need to be coupled to prime movers like electrical motors, IC engines, steam turbines, or gas turbines.

It is essential to rate and match the prime movers to the centrifugal pumps to insure proper startup and operation of the train.

To match these two machines it is essential to know the speed–torque (*N–T*) or the mechanical characteristics of the pump.

The speed–torque characteristic of a pump is defined as the relation between the torque on the shaft and the speed of rotation. These basically indicate the torque requirements of a pump during its startup. A startup of a pump maybe associated with many factors like an open or closed valve, the static head on the pump, the construction of the pump and any other.

The speed–torque characteristic of a centrifugal pump is represented as a graph with the torque *T* expressed as a percentage of nominal torque on the ordinate and the speed *N* as a percentage of nominal speed as the abscissa.

The theoretical *N–T* characteristic of a centrifugal pump is a parabola starting from the origin and proportional to the square of the speed.

Thus $T = kN^2$

However, in practical case, a certain torque is required at zero speed to overcome the mechanical losses in bearings, seals, packing, and inertia of the rotor and accelerating them.

The shape of a typical *N–T* curve is shown in Figure 5.19:

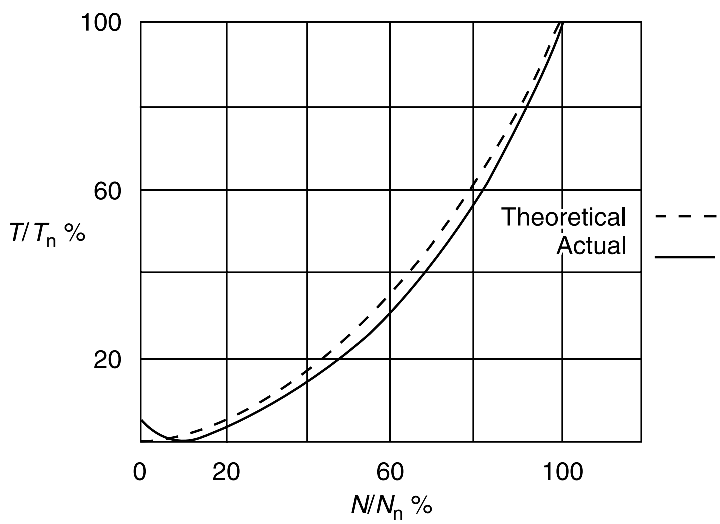



**Figure 5.19**
*Speed–torque curve*

The shape of the *N–T* curve depends on the type of the pump and whether the discharge valve is open or closed while starting the pump.

When centrifugal pumps are started with the discharge valve closed, it amounts to 30–50% of the nominal torque after attaining full speed. This value is again a function of the specific speed of the pump and the magnitude of the rotating masses.

The mixed flow pumps have more steeply falling *N–Q* characteristics than pumps with lower specific speed pumps.

In axial flow-propeller pumps, when the pump is started with a closed discharge valve, the torque attains a normal value before the full speed is obtained. Thus, this is another reason why such pumps are started with the discharge valves fully open.

### 5.11.1 Torque from the prime mover

The driver must be capable of providing more torque at successive speed from zero to full load speed. This excess torque is essential to accelerate the pump shaft to reach its rated speed.

As the pump typically has a slow rising *N–T* characteristic, most of the prime movers are able to meet this requirement.

The pump *N–T* curve is parabolic in nature and the full load torque requirement is given by the formula:

$$T = \frac{30 \times P}{(\pi \times N)}$$

$T$ is in kN-m
$P$ = kW
$N$ = rpm.

The torque requirement varies as the square of the speed, thus to obtain the torque at:

- 75% speed multiply the full load torque by 0.563
- 50% speed multiply the full load torque by 0.25
- 25% speed multiply the full load torque by 0.063.

#### Accelerating torque

The torque provided in excess of the pump requirement accelerates the pump shaft to its rated speed. This excess torque is called as accelerating torque.

The torque required to accelerate a body is equal to the $Wk^2$ of the body, times the change in rpm, divided by the time interval (in seconds) in which this acceleration takes place.

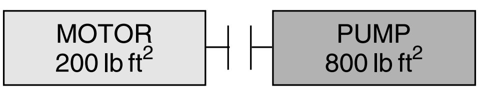
In FPS units, it is given by:

$$T_{\text{acc}} = \frac{Wk^2}{308} \times \frac{4N}{4T}$$

Where

$W$ = weight of rotor – lbs
$K$ = radius of gyration – feet
$\Delta N$ = change in speed in rpm
$\Delta T$ = time Interval for acceleration – seconds
$Wk^2$ = equivalent moment of inertia.

If, for example, we have simply a prime mover and a load with no speed adjustment:




The $Wk^2_{\text{EQ}}$ is the summation of the two moment of inertias

$$Wk^2_{\text{EQ}} = 1000 \text{ lb ft}^2$$

If we wish to accelerate this load to 1800 rpm in 1 min, the amount of torque necessary to accelerate the load would be as per the formula stated above.

$$T_{\text{acc}} = \left(\frac{1000}{308}\right) \times \left(\frac{1800}{60}\right)$$
$$= 97.4 \text{ lb ft}$$

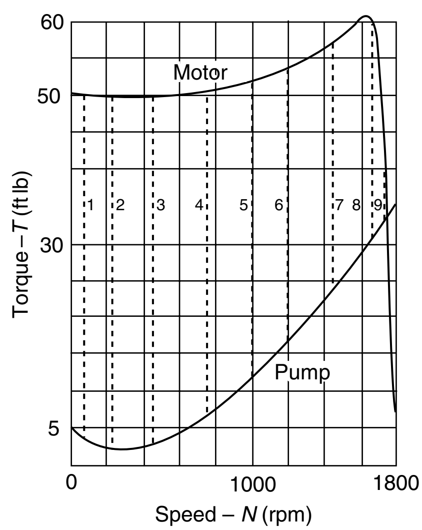
Thus, this amount of excess torque would raise the pump speed from 0 to 1800 rpm in 60 s.

The $T_{\text{acc}}$ is an average value of accelerating torque during the speed change under consideration. If a more accurate calculation is desired, the following method is adopted.

The time that it takes to accelerate an induction motor from one speed to another maybe found from the following equation:

$$t = \frac{Wk^2}{308} \times \left[\frac{N_1}{T_1} + \frac{N_2}{T_2} + \cdots\right]$$

The application of the above formula is explained by means of an example. Figure 5.20 is the speed–torque curves of a squirrel-cage induction motor driving a centrifugal pump.




Accelerating torques from Figure 5.20

$T_1$ = 46 lb ft  $T_4$ = 43.8 lb ft  $T_7$ = 32.8 lb ft

$T_2$ = 48 lb ft  $T_5$ = 39.8 lb ft  $T_8$ = 29.6 lb ft

$T_3$ = 47 lb ft  $T_6$ = 36.4 lb ft  $T_9$ = 11 lb ft

**Figure 5.20**
*Accelerating torque vs speed*

At any speed of the pump, the difference between the torque, which the motor can deliver at its shaft, and the torque required by the pump is the torque available for acceleration.

As seen in Figure 5.20, the accelerating torque may vary greatly with speed.

When the speed–torque curves for the motor and pump intersect there is no torque available for acceleration. The motor then drives the pump at constant speed and just delivers the torque required by the load.

In order to find the total time required to accelerate the motor and pump, the area between the motor speed–torque curve and the pump speed–torque curve is divided into strips, the ends of which approximate as straight lines.

Each strip corresponds to a speed increment, which takes place within a definite time interval. The vertical lines in the figure represent the boundaries of strips. In each of these bands or speed interval, the average $T_{acc}$ is worked out.

In order to calculate the total acceleration time for the motor and the direct-coupled pump, it is necessary to find the time required to accelerate the motor from the beginning of one speed interval to the beginning of the next interval and add up the incremental times for all intervals to arrive at the total acceleration time.

If the $Wk^2$ of the motor whose speed–torque curve is given in Figure 3.26 lb ft$^2$ and the $Wk^2$ of the pump referred to the motor shaft is 15 lb ft$^2$, the total $Wk^2$ is 18.26 lb ft$^2$.

The total time of acceleration is worked out in the following manner:

$$t = \frac{Wk^2}{308} \times \left[ \frac{N_1}{T_1} + \frac{N_2}{T_2} + \cdots \right]$$

$$t = \frac{18.26}{308} \times \left[ \frac{150}{46} + \frac{150}{48} + \frac{300}{47} + \frac{300}{43.8} + \frac{200}{39.8} + \frac{200}{36.4} + \frac{300}{32.8} + \frac{100}{29.6} + \frac{40}{11} \right]$$

Thus the time to accelerate the pump computes to 2.75 s.

## 5.12 Discharge regulation of pumps

A pump has to adapt to the temporary and permanent changes in the process demand. This variation in flow is called as regulation.

The discharge of the pump is regulated either at constant speed or by variation of speed. At constant speed, this variation in flow can be carried out in many ways and these include:

- Throttling of the discharge valve
- Bypass the flow
- Change in impeller diameter
- Adjustable guide vanes
- Modifying the impeller.

In some cases where within a system multiple pumps are in operation, the regulation maybe carried out by stoppage of a few pumps instead of regulating the discharge of one or many pumps.

This is usually done in boiler-feed water, cooling tower and other similar type of pumps typically found in the Utilities.

### 5.12.1 Regulation at constant speed by throttling

Regulation of the pump discharge is mostly carried out by the opening or closing of the discharge valve. This is called as regulation by throttling.

However, the deployment of this technique is a function pump's specific speed. We have seen in the characteristics of pumps that as the specific speeds of the pump increase, the *P–Q* curves tend to become flat or become drooping as in the case of axial flow pumps.

In such pumps, it is not economical to reduce the flow rate by throttling the discharge valve.

Thus, the regulation of the pump by throttling is adopted mostly in the radial impeller, low specific speed pumps. This can be carried out manually or even automatically using control valves.

Though this is the simplest method of regulation, it is an inefficient method.

The inefficiency is due to the following reasons:

- Increased pressure drop caused by the control valve closing is wasted energy.
- The introduction of an open control valve causes a dynamic head loss. It is typically taken to be 10% of the other losses.
- Throttling causes the system curve to shift and the operating point moves on the *Q–H* curve. Moving the operating point either to the left or right of BEP reduces the pump efficiency.
- The piping involved with control valves involves bypass piping and isolation valves to allow for valve repair.

Thus, this method is used only when the regulation required is temporary or continuously changing during an operation.

When permanent regulation is desired other regulation techniques like modifications to the impeller maybe used.

### 5.12.2 Regulation at constant speed by bypass of flow rate

Bypass regulation of pumps is a method in which a portion of the total discharge flow is diverted back to the suction of the pump or to some other system.

In this system the pump flow is diverted into two systems with different resistances.

This regulation technique is usually adopted for high specific speed pump and in case of boiler feed water pumps with demand variations that maybe on the higher side.

### 5.12.3 Regulation at constant speed by turning down the impeller

Trimming of the impeller brings about a permanent change in the flow and the head developed by the pump.

When the outside diameter $d_2$ is reduced to a new diameter $d_2'$, the outlet velocity triangle undergoes a change. The peripheral velocity $U_2$ reduces and the outlet angle $\beta_2$ increases.

Changes also occur in the width of the impeller $b_2$, the length of vanes and the overlap (shown as the shaded area in Figure 5.21).

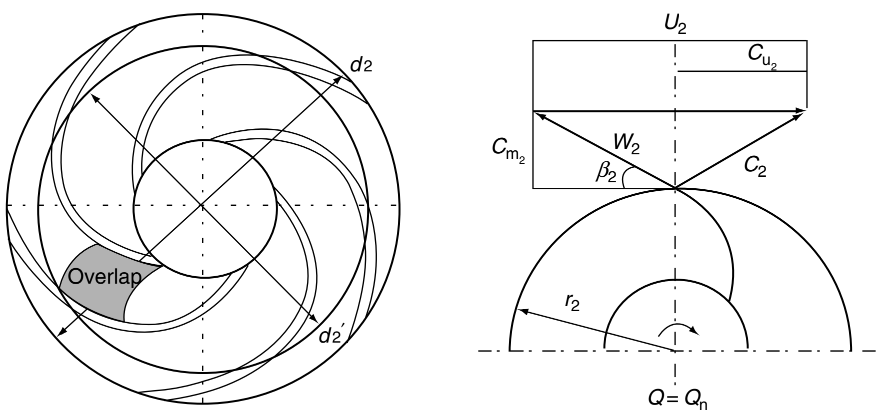



**Figure 5.21**
*Impeller trimming*

The ratio of the trim, that is $d_2'/d_2$ is determined by the specific speed of the pumps. In lower specific speed pumps (less than 2500), this ratio should be limited to 80%. For pumps with a specific speed in the range of 2500–4000, this ratio should be restricted to 90%.

In pumps with higher specific speeds, there is a considerable drop in efficiency even with a slight trimming of the impeller diameters.

In pumps, impellers can be paired with volutes or diffuser rings. When paired with volutes, the impellers can be trimmed along with the shrouds. However, in case of diffuser rings, the gap between the impeller and the diffuser should not be increased. Therefore, only the vanes should be trimmed and no cut should be made on the shrouds.

As mentioned earlier, when the diameter of the impeller is reduced, both the flow rate and the head also are reduced.

The relationship is stated in the section on Pump Hydraulics, Section 3.10 on Affinity Laws.

The basis of selecting a lower diameter $d_2'$ or say in this case as $d_B$ is determined by the Affinity laws or the *Q–H* curve as shown in Figure 5.22.

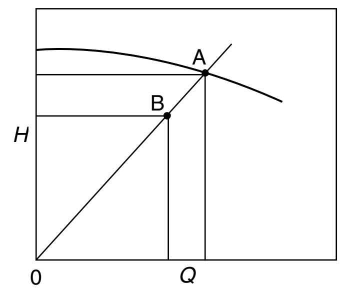



**Figure 5.22**
*New operating point on* Q–H *curve*

The new required operating point B is plotted on the *Q–H* curve. The line 0B is drawn and extended to point A on the original *Q–H* curve.

Projecting the points A and B, we get the points $Q_A$, $H_A$, $Q_B$, $H_B$.

When points A and B are close, a constant efficiency relationship is assumed.

As it has been discussed in Section 3.10 that when the percentage of trim is large, the values obtained by the affinity laws tend to give a value that needs to be modified by a correction factor.

In high specific speed pumps with an inclined outlet edge as shown in Figure 5.23, the calculation is based on the central streamline, constituting the centerline of the impeller passage.

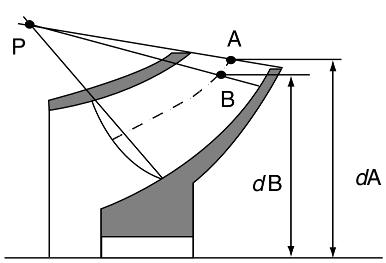



**Figure 5.23**
*High specific speed pumps with an inclined outlet edge*

The outlet edge of the impeller after trimming should pass through the point of intersection P. Point P is the intersection of the original outlet edge and the chord of the inlet edge.

### 5.12.4 Regulation by adjustable guide vanes

Some of the pumps maybe provided with inlet guide vanes. These are very similar to those observed in centrifugal fans, and the principle of operation too is similar.

Adjusting the guide vanes gives rise to pre-whirl in the liquid entering the impeller that brings about a change in the flow rate and total head developed by the pump.

It is found that positive pre-whirl (causing the liquid to rotate in the direction of the impeller) is suitable for the purpose of regulation. This causes the flow rate and discharge head to reduce as the pre-whirl is increased.

Negative pre-whirl is not found suitable as it causes reduction in pump efficiency.

Experiments carried out to observe the effect of pre-whirl regulation on specific speed of pumps indicate that regulation by means of pre-whirl is not very effective in low specific speed pumps.

As the specific speed increases, the effectiveness of regulation using a pre-whirl improves.

There have also been pumps designed with adjustable guide vanes in the outlet.

### 5.12.5 Regulation by adjustable impeller blades

In propeller pumps, the flow rate and the head developed can be changed with the help of having variable pitch blades.

The angle inclination of the blades given by $\beta+\alpha$ is altered without changing the diameter of the blades (Figure 5.24).

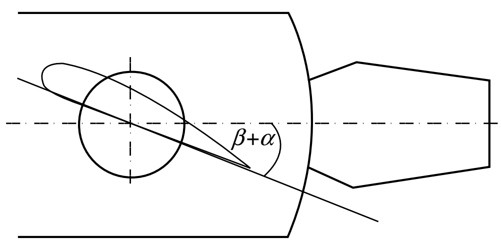



**Figure 5.24**
*Angle of inclination of blades*

When the angle of inclination is reduced, there is a reduction in the discharge without an appreciable drop in the efficiency of the pump. The change in the head developed is also not substantial.

### 5.12.6 Regulation by varying speed

The method of discharge regulation by varying the speed of the pump is one of the most economical since there are no losses due to the throttling of the valve and deviation from rated efficiency is minimal.

Figure 5.25 brings out the key difference between the regulation methods of valve throttling and speed variation.

As is shown, in the throttling method the system resistance curve is shifted whereas in the speed variation method, the pump *Q–H* curve moves along the system resistance curve with decreasing speed to alter the operating point.

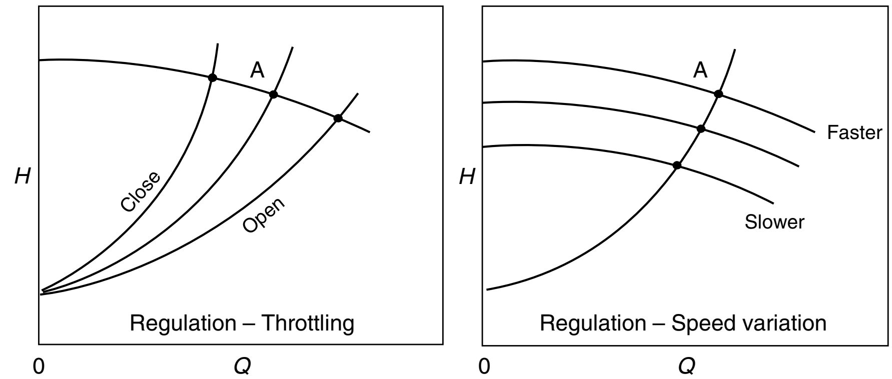



**Figure 5.25**
*Speed regulation*

With this method of control, the energy to the pump is reduced with the decrease in speed as compared to throttling and bypass where full energy is supplied even for a lower operating point.

However, there are some drawbacks with this technique and these are:

- The effectiveness of this control is much more dependent on the shape of the pump curve.
- Not all pump designs can be effectively controlled with speed variation, particularly pumps which require recycle control.
- There is a loss of pump efficiency, although it is not as great a loss as with discharge throttling.

Speed variation can be brought by steam turbines, gas engines, and variable frequency drives for asynchronous (induction) motors, DC motors, multiple pole motors, and some other electrical systems.

## Regulation by varying speed – hydraulic/hydrostatic drive

A hydraulic coupling is mounted between the pump and an induction motor. It permits speed reduction down to 20% of the normal speed.

The output speed of a hydraulic coupling is determined by the amount of slip between the input and output shafts. The output shaft speed cannot exceed the input shaft speed while the motor is operating. In the fluid coupling, the input shaft drives a vane impeller, while a vane runner drives the load.

Speed is controlled by adjusting the volume of oil in the working circuit, achieving a typical speed ratio of 5 to 1. For an output speed of 50%, the average overall efficiency of the hydraulic coupling is about 40%.

Although hydraulic couplings may operate with applications ranging from a few horsepower to tens of thousands of horsepower, they function well only when most of the duty cycle is in the upper speed range.

At lower speeds, the losses are simply too high to justify the use of a hydraulic system. In new applications, they are normally installed only in low-horsepower applications where they maybe less expensive than AC variable speed drives.

When ongoing maintenance and energy costs are included in the analysis, it is often still more cost-effective to retrofit a shaft-applied drive with an AC variable speed drive.

### Regulation by varying speed – mechanical drive

This is a variable belt and sheave drive. Some additional friction and windage losses are created, in addition to maintenance issues.

### Regulation by varying speed – eddy current drive/clutch

This drive uses a magnetic coupling to transfer load torque at a different speed. The slip losses in the clutch can become appreciable as the speed is decreased.

The eddy current drive couples an eddy current clutch to an AC induction motor. A rotating drum connected to the induction motor surrounds a cylinder attached to the output shaft.

The concentric cylinder and drum are coupled by a magnetic field, whose strength determines the amount of slip. A low-power solid-state device controls the speed that shifts the current in the magnetic field winding. This field excitation typically consumes 2% of the drive's rated power.

The eddy current clutch is a slip device like the hydraulic coupling, but much more efficient. Waste heat, generated by the motion of the drum and cylinder, is the main source of power loss and this is absorbed by air or water-cooling depending on the horsepower of transmission.

### Regulation by varying speed – variable speed drives

These are also known as inverters, AC drives, and adjustable frequency drives (AFDs), VFDs operate by varying the frequency and voltage to AC motors.

The frequency of the applied power to an AC motor determines the motor speed. The variable speed can be used in certain applications for improved efficiency, energy savings, and to optimize the motor-driven process. The energy savings is generally proportional to the motor speed.

VFDs can also extend the speed range of a motor by applying adjustable frequencies up to 120 Hz. At speeds below 60 Hz, torque is constant and HP diminishes proportionately to speed. At speeds above 60 Hz, HP is constant and torque gradually diminishes to roughly 66% of maximum at 120 Hz.

Motor cooling is usually not a concern but mechanical balance, rotational stresses, and bearing life are important considerations.

Most drives are programmable and have internal adjustments, which allow the user to select operating conditions most suitable for his application. Some common applications for VFDs are pumps, fans, mixers, blenders, air handlers, and conveyor systems.

A comparison of the various methods of regulation is depicted in Figure 5.26 and it is obvious that regulation using variable frequency drives is one of the most economical methods of pump discharge regulation.

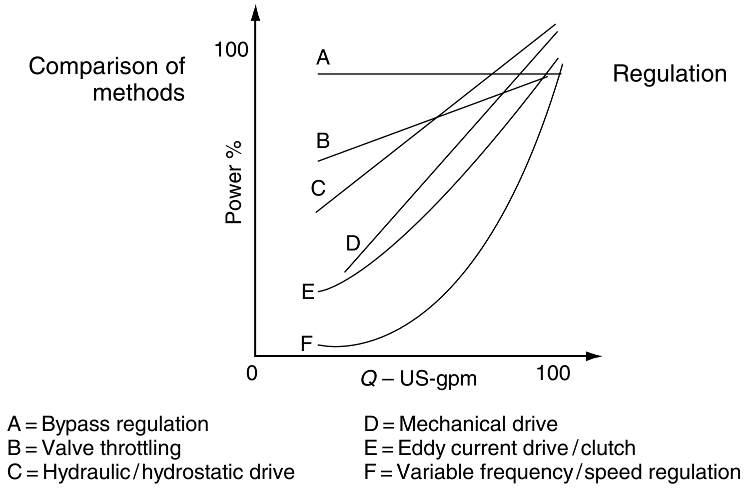



**Figure 5.26**
*Comparison of methods of regulation*

## 5.13 Range of pump operation

When sizing and selecting centrifugal pumps for a given application, the pump efficiency at design should be taken into consideration. The pump characteristics indicate the point of BEP and its variation with respect to the flow rate $Q$.

This is an important factor; however, it is almost improbable that the pump operation in actual service would be at the BEP. Thus, if the above is not possible in most of the cases, the selection of the pump has to be made in a manner so that the range of operation is near the BEP.

This range of operation is important to define to avoid excessive hydraulic thrust, temperature rise, and erosion and separation cavitation.

These phenomena occur when the operation of a centrifugal pump is to the furthest left or right of the $Q$–$H$ curves. Performance in these areas induces premature bearing and mechanical seal failures due to shaft deflection, and an increase in temperature of the process fluid in the pump casing causing seizure of close tolerance parts and cavitation.

### 5.13.1 Pump operation to the left of BEP

In a centrifugal pump when operating toward shut-off or the far left on the $Q$–$H$ curve, a percentage of the process fluid recirculates in the eye of the impeller and between the impeller shroud and back-plate. Evidence of minimum flow problems is more dramatic on applications where NPSH-a exceeds the NPSH-r by a given pump two-, three- and fourfold.

This liquid that churns in the casing keeps absorbing the 'inefficiency' of the pump. Efficiency is a factor of useful conversion of mechanical energy into liquid head and flow. However, the inefficient part of the power goes into heating of the liquid.

Heating up of the liquid can lead to potential problems such as vapor formation, expansion of internals leading to seizure, crossing the operating temperature limits of the material of construction, or any other.

To avoid thermal problems during low flow operation and prevent a potentially hazardous temperature rise within the pump, the temperature rise at shut-off and the minimum flow required for thermal protection must be calculated and the required volume of the fluid to be bypassed to dissipate this heat established.

Prior to calculating the minimum flow required for a given application, the maximum allowable temperature rise must be established. This defines the temperature that exceeds the corresponding saturation temperature of the impeller eye.

While most maximum allowable temperature increases are based on the temperature where flashing; vaporizing, of the process fluid occurs, it is important to realize other pump components may dictate a lower temperature to insure long trouble-free service.

For example, while the maximum allowable temperature to avoid cavitation maybe 210 °F, the upper temperature limitations for a polypropylene pump maybe 180 °F.

Other pump components that may require consideration will include mechanical seals, packing and bearings, and wear ring tolerances.

When handling non-Newtonian, viscous, and shear sensitive fluids, where temperature may affect product integrity, the maximum allowable temperature introduced by the pump at the design capacity should be considered. The temperature rise at various parts on a centrifugal pump head capacity curve should be identified and the required bypass volume established.

During the centrifugal pump selection process, consideration must also be given to minimum flow for mechanical protection. With cool clean liquids, the required minimum flow for thermal protection maybe minimal.

However, excessive shaft deflection due to unbalanced radial loads, vibration and rotating element instability will result, should the mechanical minimum flow requirements not be met. These scenarios become more evident as suction pressures increase further beyond the NPSH-r by the pump.

The rate of temperature rise when a pump operate with fully closed discharge valve can be calculated from:

$$T_{\mathrm{R}} = \left[ \frac{(5.1 \times \mathrm{HP})}{(Q \times S_{\mathrm{H}} \times S_{\mathrm{G}})} \right]$$

Where

$T_{\mathrm{R}}$ = rate of temperature rise, °F/min
HP = BHP at shut-off
$Q$ = volume of liquid in pump case, gallons
$S_{\mathrm{H}}$ = specific heat of liquid
$S_{\mathrm{G}}$ = specific gravity of the liquid.

This equation neglects the heat loss through the pump case so its result is conservative. The volume of the casing can be geometrically estimated.

Based on the above rate of temperature rise, the duration of safe operation at shut-off can then be calculated by dividing the allowable temperature rise with the rate of temperature rise.

For safe operation the allowable temperature rise should be limited to 50 °F maximum. A lower value of allowable temperature rise should be considered if the 50° limit would result in exceeding the case design temperature, or to an increase in vapor pressure that could result in critical NPSH or to potentially damaging vaporization at mechanical seal face. As in the case of boiler feed water applications, this rise in temperature is limited to 15 °F.

At any other point of operation other than the shut-off conditions, the following equation can be used to estimate the temperature rise of a liquid being

$$T_R = \left[ \frac{H}{778 \times S_H \times E} \right]$$

Where

$T_R$ = temperature rise, °F
$H$ = total pump head, FT
$S_H$ = specific heat of liquid
$E$ = pump efficiency, in decimal point.

In case the allowable temperature rise of a pump is available, the following formula can be used to estimate the minimum flow through the pump.

$$Q = \frac{5 \times \mathrm{HP}}{(S_H \times T_R)}$$

Where

HP = Horse Power of the pump at shut-off conditions
$T_R$ = temperature rise of liquid in °F
$S_H$ = specific heat of liquid.

Once the minimum flow of a pump is established, different techniques are available to insure that the minimum amount flow is passing through the pump casing.

These techniques are:

- The minimum flow bypass will remain open at all times with a fixed orifice installed in the line.
- The bypass is opened during start-up and then closed when the pump is operating at design capacity.
- One of the most practical, cost efficient methods of maintaining the bypass capacity would be an automatically controlled device, which senses changes in the process pressure.
- As this pressure increases to a predetermined setting the device will open diverting the bypass flow, as the pressure decreases the device will modulate limiting the bypass flow.

### 5.13.2 Pump operation to the right of BEP

The pump operation to the right of the BEP has different set of problems.

The characteristic curves of the lower specific speed pumps indicate that such pumps have overloading features. This implies that the power requirement of the pumps keeps on increasing which ultimately leads to tripping of the motor.

Another problem associated with operation to the far right of the BEP is that of the breakaway NPSH-r. At higher flow, the internal fluid velocities are higher, and according to Bernoulli (sum of velocity head, static head, and gradient is constant), the static pressure (or static head) part becomes less and thus comes closer to vapor pressure.

The static pressure, therefore, must be increased externally, that is a higher value of NPSH-r is needed for higher flows.

Even though the pump may have adequate NPSH margin, at higher flow rates this margin can be reduced very rapidly as shown in Figure 5.27.

At higher flows the NPSH margin can become negative leading to the phenomenon of cavitation.

Thus considering both the problems associated with the minimum and higher flows from a centrifugal pump, the range of operation of pumps is usually limited within the 70–120% of the flow rate BEP.

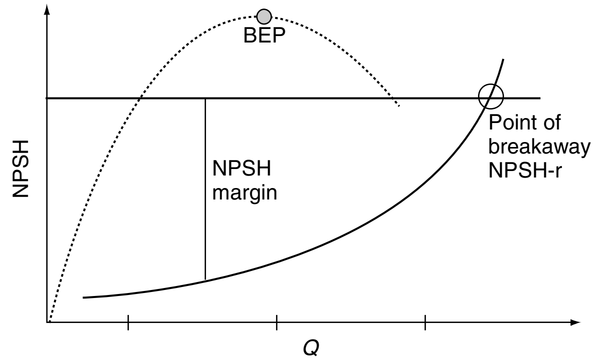



**Figure 5.27**
*NPSH* vs Q

Flow rate at 120% of the BEP in some texts is called as the 'end-of-the-curve'. Some of the motor selection and sizing for lower specific speed pumps is done considering this point as the last operating point on the curve.

# 6

# Pump specification and selection

Centrifugal pumps are used for a wide variety of applications. These applications may or may not be critical in nature.

In some instances, it suffices to buy a pump off-the-shelf merely on the horsepower rating of its motor or by its overall dimensions.

The pumps procured in the above manner may not provide the best fit of the pump to the application, but the penalties associated with such a mismatch may not be significant.

However, in process industries, the pumps have to perform vital functions on a continual basis. In difficult environments, there are significant penalties associated with downtime and maintenance costs.

It then becomes important to specify the process and its requirements. This aids in a selection of a pump, which is designed and manufactured with features such that it can operate under specified conditions in a reliable, efficient, and cost-effective manner.

A good pump specification is often considered as the foundation or the basis of pump reliability over its years of operation. It is the first document prepared for a pump by the user and is often the most vital.

A pump specification is a document that is preceded by:

- System analysis (covering the hydraulic aspects)
- Mechanical requirements.

Preparation of a pump specification document is a multidisciplinary team effort, which involves the process engineers, mechanical engineers, contractors, and the pump vendors.

Once a pump specification has been laid out, it is followed by a bid request, a bid analysis and this process finally culminates with the selection of the right pump.

Failure to define or specify the pump completely for the anticipated process conditions often results in poor operating experience and high maintenance costs.

In any pump specification, the following process requirements are of prime importance:

- Maximum differential head at the specified flow rates
- Available Net Positive Suction Head, NPSH-a
- Anticipated flow range or the required flow flexibility
- Properties of the liquid that include its hazard potential, the specific gravity, vapor pressure, viscosity within the range of pumping temperatures. This also includes information of its composition and possibilities of solids in the liquid and its properties
- Anticipated transient conditions.

The above form the basis of a detailed 'system analysis' and these often result in specifying certain mechanical requirements.

For example, handling of a liquid that generates carcinogenic vapors may demand selection of a sealless pump instead of a pump with mechanical seals.

It is essential not to be over conservative in the specification process as this can lead to selection of pumps that not only mismatch with the process but also are expensive in the due to over sizing.

## 6.1 System analysis

System analysis is the first step leading to the preparation of the pump specification.

This includes the following evaluations:

- Pump boundary conditions
- Flow requirements
- Fluid specifications
- Criticality of service.

### 6.1.1 Pump boundary conditions

To evaluate the pump boundary conditions, it is essential to have a comprehensive knowledge about the pump hydraulics and the various issues related to pump operation and associated problems.

#### NPSH-available and NPSH margin

The first boundary condition is related to the suction conditions of the pump and the factor under discussion is the Available Net Positive Suction Head.

This parameter has been discussed with in detail in Section 3.12. In the Section 3.12, different cases are solved to arrive at the available NPSH for the different suction conditions of any pump.

It is observed NPSH-a is dependent on the following factors:

- Vapor pressure of liquid
- Pipe losses
- Pressure in the suction vessel
- Gradient height
- Absolute pressure.

Among the listed factors, the last factor is constant for a location. However, this makes it necessary to know the location where the pump would be installed. This is especially true in many cases when the process/pump specifications are made for plants to be erected in some other part of the globe.

The geographical conditions of the location play an important role in pump specification. In the above case, the height of the location from the mean sea level determines the absolute pressure. Other factors like the ambient temperature ranges and rainfall through out the year also determine some of the pump features.

In pumps with a long suction pipe, the added heat due to both heat tracing lines and ambient conditions can raise the pumping temperature and simultaneously affect the vapor pressure of the liquid.

An increase in vapor pressure due to increase in temperature reduces the NPSH-a; therefore, it maybe prudent to specify insulation for suction piping.

Another factor associated with long suction pipelines are the pipe losses. These are easy to compute and add to the system resistance. However, there exist certain applications where fouling of pipes, fittings, strainer, and others can also be expected.

In such cases, the extent of fouling before a cleanout needs to be anticipated and taken into consideration and take into account the drop in the NPSH-a along with the rise in system resistance.

Figure 6.1 indicates the effect of fouling in the suction header on NPSH-a. Fouling results in a drop in NPSH-a and consequently, in the margin between the NPSH-a and NPSH-r.

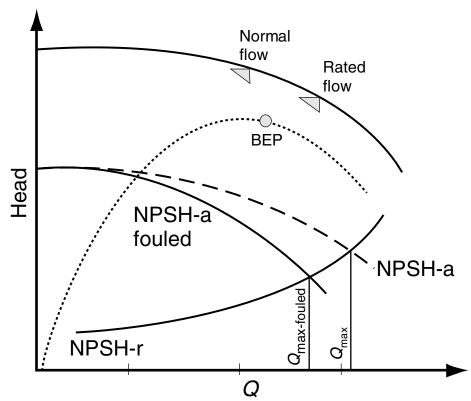



**Figure 6.1**
*Effect of fouling on NPSH-a and flow rate*

The point where this margin reduces to a minimum positive value is specified as the maximum flow rate, $Q_{max}$. The fouling of the suction header causes a steeper droop in the NPSH-a curve and this reduces the $Q_{max}$ flow rate.

It is important to consider the $Q_{max}$ for NPSH-a calculations, as the frictional pipe losses are a maximum for this flow rate.

The other two factors that can affect the NPSH-a are the pressure in the suction vessel and the height of the suction column.

While computing the NPSH-a, the process has to be evaluated considering the minimum operating level of the liquid so as to obtain the minimum height of the suction column. This could be the level at which an alarm is placed or it could be the height at which the suction pipe draws off from the vessel. This is shown in Figure 6.2.

Thus, the structural design and the process operational limits need to be considered, the minimum level at which the maximum flow rate is expected should provide the limiting guidelines.

Once the NPSH-a over the range of operation is determined, the next step is to compute the margin. The acceptable NPSH margin or ratio is also covered in the Section 3.12.

As mentioned in Section 3.12, the definition of NPSH-r provided by the Hydraulic Institute does not adequately cover the high suction energy pump. The damage free value of NPSH-r could be 2 to 20 times the NPSH-r obtained by the method of 3% head drop (Figure 6.3).

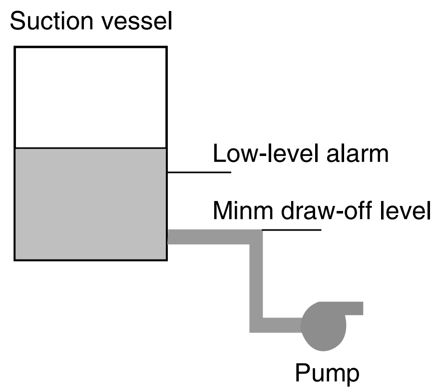



**Figure 6.2**
*Minimum height of suction column*

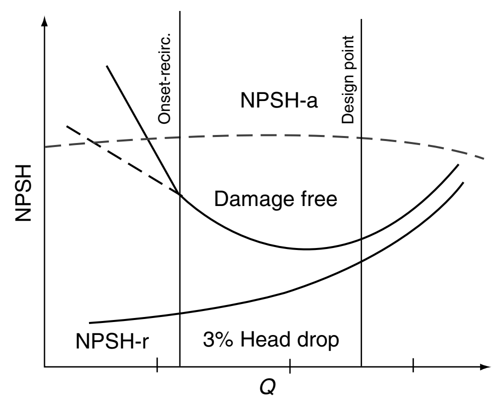



**Figure 6.3**
*Comparison of damage free NPSH-r to NPSH-r$_3$*

## Suction-specific speed

When minimal values of NPSH-a are obtained, it is essential to make a note of it so that the vendors select a pump, which typically has a lower NPSH-r. However, when vendors offer low NPSH-r pumps, one has to evaluate the suction-specific speed (covered under Section 3.13) number of the pump.

Under Section 3.13, it was stated that when suction-specific speed, $N_{ss}$ is greater than 11 000, the pump can experience hydraulic and mechanical problems, especially when the flow rates are further away from the BEP (Figure 6.4).

The above basis comes from a statistical study that was conducted in a refinery for 480 pumps over a period of 5 years. Jerry L. Hallam presented the results in a paper in 1982.

At the 13th International Pump Users Symposium conducted by the Texas A&M University in 1996, Bernd Stoffel and Ralf Jaeger presented a paper called, 'Experimental Investigations in Respect to the Relevance of Suction Specific Speed for the Performance and Reliability of Centrifugal Pumps'.

The paper presented the results of a study whose aim was to investigate the possible effects of high $N_{ss}$ on the operational behavior of centrifugal pumps, especially on the measurable dynamic quantities that can serve as indicators for the risk of failures.

In this study, three standard pumps of different specific speeds and sizes were designed and manufactured by three different German manufacturers. All these pumps were alternatively equipped with two impellers. One of them was designed for a high $N_{ss}$ value and one impeller with an $N_{ss}$ value lower than the recommended limit of 11 000.

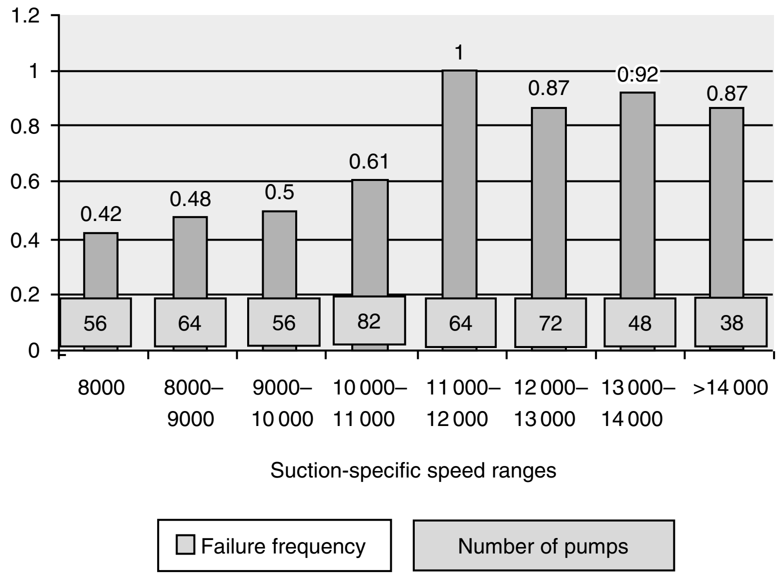



**Figure 6.4**
*Suction-specific speed*

The results of the experiment indicated that the measured quantities that could be considered as indicators for the dynamic loading of bearings, seals, and others did not show trends or cause effects in pumps with $N_{ss}$ > 11 000 which would have indicated a higher failure probability.

Therefore, there exists a debate over the issue of selection of pumps with an $N_{ss}$ value more than 11 000. Until there is a clear consensus in the Pumps community over the issue, it is advisable to be conservative and stay specified within the recommended limit.

## System resistance – differential head

The next step in the process of evaluating the complete system analysis is the accurate determination of the System Resistance Curve.

The defining of the NPSH-a covered earlier more or less evaluates the suction side of the centrifugal pump. In a similar way, the discharge side should be worked out.

To determine the system resistance on the discharge side, the following factors have to be considered:

- Static head built in the pump discharge in terms of downstream pressure in a discharge vessel
- The height which must be overcome to reach the discharge vessel
- Rate of increase of system resistance with respect to the flow rate.

A quickly rising system resistance curve may preclude a maximum flow rate they maybe required occasionally. This is especially the case when the maximum flow rate is far in excess of the normal flow rate.

A regulating valve (control valve) sizing is based on the rate of rise of the system resistance curve as well as the size of the pump (Figure 6.5).

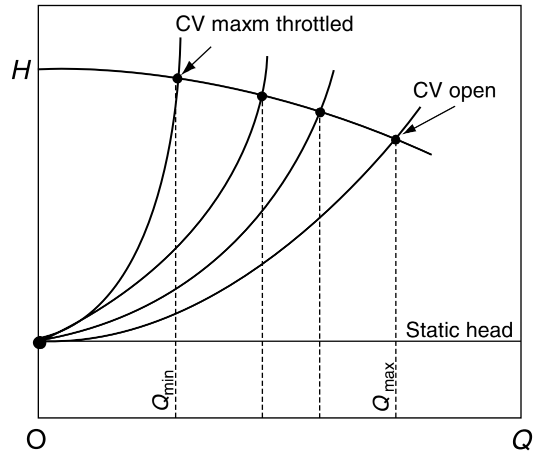



**Figure 6.5**
*System resistance with regulating valve*

It has to be sized to provide the artificial head loss at the rated, minimum flow rates, and the minimum loss at the maximum flow rates.

One way to flatten the system resistance is to install a higher size of pipe.

The evaluation of the system resistance on the suction and the discharge sides of the centrifugal provides for the differential head as required from the pump.

## 6.1.2 Flow requirements

The flow requirements are often determined on the basis of meeting process demands. The process decides the flow rate.

In a pump specification usually, two flow rates are stated:

1. *Normal flow rate*: This is the flow rate at which the pump will usually operate.
2. *Rated flow rate*: This is the flow rate guaranteed by the pump vendor for the specified operating conditions.

Usually the Rated Flow Rate is 10% in excess of the Normal Flow rate for low to medium flow rates and 5% in excess of the normal flow rate for pump delivering higher flow rates.

The rated flow rate should reflect the maximum flow the system can envisage under current consideration. In addition, it should be selected keeping in mind any future increases in process throughout.

The minimum flow rate requirements of the pump may conflict with the rated flow rate of the pump. In such cases, provisions should be made for recycle of the process liquid.

If it is possible, one should indicate the periods for which the pump shall be operating at minimum, maximum, and rated flows.

A longer period of operation at low flows could imply higher radial loads. This can greatly affect the life of the bearings.

In addition to the minimum and maximum flow rates, it is possible that under certain operating conditions the pump could be physically or hydraulically shut-off at the discharge.

When such is the case, it is recommended to consider the various options stated under Section 5.13 to insure minimum flow rate through the pump.

If the specified pump is required to operate in parallel with another pump, a lot of care has to be taken to insure the minimum stable flows. This is especially true in case of pumps with dissimilar *Q–H* curves (Parallel Operation with different *Q–H* curves, see Section 5.8). In this case, below a certain flow rate, one of the pumps with a lower shut-off head will begin to operate under shut-off conditions.

A similar event occurs when the *Q–H* curves are very flat and one the pumps have a shut-off head slightly lower than the other.

It is for this reason that API 610 (7th Ed.) specifies that pumps with one or two stages and operating parallel should have rising *Q–H* curves. The percentage rise of the head at rated capacity to shut-off conditions should be 10–20%. For 3 or more stages, a slightly lesser percentage rise is allowed as this can lead to excessive high shut-off heads.

Probably flow requirement of a pump is one factor that demands maximum team effort to arrive at a definitive value.

Flow requirement determines the sizing and reliability of pumps. Pumps specified with flow rates that match closely to actual operations generally have lower life cycle costs.

### 6.1.3 Fluid specification

Pump specifications should provide as much information as possible about the liquid properties.

The liquid properties determine the pump's:

- Material of construction
- Pump design like its support, cooling water jackets, etc.
- Impeller design
- Mechanical seal, sealant and piping plan
- Construction features like wear plates, hard coatings, etc.
- Driver horsepower.

The liquid should be checked for its hazard and toxicity potential, which may include higher flammability, acidic or alkaline nature, health hazard, and carcinogenic potential.

Corrosive liquids chemically attack or oxidize the pump material. For example, handling sulfuric acid of 65–70% concentration at temperatures above 70 °C may require special materials like High Silicon Iron (13–15% Silicon).

Materials selection should consider possibilities of electrolytic reaction, particularly in seawater applications.

Liquids that contain abrasive particles have a potential to cause considerable erosion of the wet parts of a pump and may lead to performance deterioration. It may become necessary to specify a semi-open or open impeller if the particles are larger.

The abrasive nature of the particles may necessitate specification of hard coatings or wear plates to prevent wear of pump parts.

As mentioned in the earlier Section on NPSH, cold water has the maximum damage potential due to cavitation and the NPSH margin/ratio has to be evaluated carefully.

Liquids that contain dissolved gases have to be treated carefully, as potassium carbonate solution could evolve carbondioxide gases under certain pumping temperatures. Evolution of gases can cause cavitation. It can also affect pump's capability to build pressure. In such cases, suction vent joining an upstream vessel can help the situation.

The pumping temperature as mentioned earlier has an impact on the viscosity and the vapor pressure of the liquid and can affect the pump performance and the available NPSH.

At higher pumping temperature, horizontal pumps with a centerline support are selected. API 610 recommends this feature when the pumping temperature is above 177 ºC (350 ºF). The seal housing and in some cases even the bearing housing may have cooling water jackets based on this factor.

The corrosiveness of any liquid is a function of its temperature so it is essential to confirm the adequacy of the material of construction at the pumping temperature.

Dangerous liquids that could be toxic, inflammable, or carcinogenic may demand stringent pump designs. For example, an application in which no leak maybe acceptable under any condition may necessitate sealless pumps (Section 1.4).

### 6.1.4 Criticality of service

The criticality of a pump is based on the following factors:

- Failure can affect plant safety and it does not have a standby.
- Pumps are vital for plant operation and its shutdown will curtail the process.
- It is a part of a large horsepower train, where better operation can save energy or improve yields.
- The capital cost is high, and very expensive to repair or may take a long time to repair.
- Perennial 'bad actors' or machines that wreck on the slightest provocation of an off-duty operation.

In most of the cases, a standby pump is specified for a critical process and a continuous operation.

At times, the pump and its standby are specified with different types of drivers. For example, one pump maybe driven by a steam turbine and the other pump maybe coupled to an electric motor.

In this arrangement, process steam can be utilized and power can be conserved. This scheme may also ensure operation even in the failure of power supply.

## 6.2 Data sheet – the pump specification document

The pump specification document or a data sheet is an organized format in which the information obtained from the above studies is made available to the contractor or the pump vendor.

It also includes the notes providing information about various aspects and includes the compulsory or optional features desired in any pump.

A blank data sheet or a format for centrifugal pumps is available in the API 610 standard. This is attached at the end of this section.

Another typical data sheet is attached to depict as to how these can be modified to suit a particular user.

A data sheet format is organized to for providing or demanding the following information:

- Project information
- Operating (liquid) data
- Pumping (system) data
- Site conditions
- Pump driver information
- Design operating conditions
- Pump design

- Mechanical seal information
- Bearings and lubrication
- Material of construction
- Weight
- Cooling requirements
- Piping connections
- Accessories
- Inspection and test requirements
- Pump drawings, design and data documents
- Additional information (notes/comments).

The last point is covered in the blank pages of the data sheet and may seek the following information form the pump vendor.

- Demand for deviations from the specified standard
- Requirement of start-up and minimum spares
- Quote the pump minimum flow and its basis
- Specify impeller to the volute cutwater clearance for pumps developing a head greater than 200 mlc.
- Requirement of any special type of seals and bearings and their manufacturers
- Specification of noise limits
- Requirement of rotordynamic studies that could include lateral analysis or a torsion vibration analysis of the full train
- Mill reports of certain materials
- Any special welding and attachment procedures
- Wear plate or hard coat requirements
- Any particular painting/packaging requirements
- Shipment details
- Requirement of pump information in soft and hard copies.

## 6.3 Bid request

The bid request is the process in which the pump data sheet is sent to the pump vendors for a quote of a suitable pump.

Usually a data sheet should be sent to three to five qualified vendors. When it is known that the specified pump falls among the standard offering, three bids are sufficient.

In the case of specialized pumps, the number of vendors could vary from three to six in number.

It is recommended to enclose, along with the pump datasheet, a covering form that enlists the mandatory and optional documents and their order.

This acts like a checklist for the pump vendor and assists in the next step, which is the bid evaluation process.

While selecting the pump vendors, keep in mind the existing inventory of pump spares with a user. Many times similar pumps can be quoted which might not demand any additional spares.

Alternatively, a pump vendor could be excluded merely because of poor follow-up and delivery of spares.

A preparation of a pump bid consumes time, effort, and money for the vendor and the user who reviews it.

Therefore, it is necessary to forward the bid request only to those vendors whose bids will be seriously considered if their pumps meet all the requirements.

## 6.4 Bid review/analysis

A clear and comprehensive specification enables a purchaser to compare the bids on an equal basis. The exceptions made by the vendors need to be weighed and factored against the desired features and prices offered.

An analytical approach demands a tabulation of bids to ease the comparison of the offers. This is typically classified into the following.

### Hydraulic performance

- Percentage of rated flow to the flow rate at BEP
- Pump numbers – specific speed and suction-specific speed
- NPSH-r, NPSH-a margin (at rated and maximum flows)
- Percentage of head rise from rated flow to shut-off
- Pump efficiency at rated and normal flow rates
- Minimum continuous/stable flow
- Maximum hydraulic power
- Noise levels.

### Construction

- Pump types
- Orientation of suction flange and rating
- Cooling water jackets for seal housing, bearing housing, or pedestals
- Impeller size; minimum and maximum sizes possible in the volute casing
- Single or double volutes
- Material compliance
- Mechanical Seal type and materials offered
- Bearing types and lubrication
- Coupling type
- Maximum thrust load
- Baseplate grouting facilities.

### Driver

#### Steam turbine

- Steam rate (kW h/kg)
- Direction of rotation
- Maximum horsepower at worst steam conditions
- Governor type
- Bearing type and lubrication
- Sealing arrangement
- Trip and throttle valve
- Materials of construction.

### Electrical motor

- Horsepower rating
- Type of enclosure
- Service factor
- Voltage, speed
- Efficiency vs motor load
- Frame and its size.

### Price

- Price of pump
- Price of turbine or motor
- Price of spare parts offered
- Price of inspection and testing
- Installation and commissioning charges.

Bid analysis almost rarely brings about a clear winner. There are some pumps, which may have some advantage over the other in regard to certain features.

Thus, it is essential to give weightage to all the factors to arrive at a pump that gets the maximum marks.

Factorial weightage is dependent on many factors. A correct system analysis provides a sound basis for weighting factors.

Standard computer-bid analysis spreadsheets assist in making the analysis convenient and accurate.

The results of the bid analysis end the selection process of the pump.

## 6.5 Conclusion

Thus a clear understanding of the pump operating and system requirements leads to selection of a pump that would be efficient and reliable.

Omissions and ambiguity at this early stage can prove to be very an expensive mistake.

To lower pump life cycle costs it is essential to specify and select the pumps correctly.

# 7

# Pump testing and inspection

The previous chapter covered the aspects of pump specification, bid request, and bid analysis. This process culminates with the raising of a purchase order for the selected pump.

Once the order is placed, the design effort shifts to the pump manufacturer and the requisition officer's priorities shift to drawing and delivery schedules and conformance to the quality standards established during the selection process and laid down in the data sheet or the purchase order.

It is important to make precise specifications and insure that the pump manufacturer supplies a product in line with these specifications.

However, the actual process of design and fabrication involves many organizational sub-units of the supplier and its sub-vendors. Lack of communication or commitment can result in nonconformance.

This can result in the selection of a pump that may not be in line with the requirements.

However, though rectifications can be made subsequently, these can prove to be very expensive especially in a project driven by tight schedules.

Thus, an approach based on the concept of partnership is essential to work toward a common goal to obtain the right pump for an application. This approach demands regular meets between the purchaser and the pump vendor to insure proper communication in regard to design engineering, fabrication, and quality control.

These are still predominantly in the domain of the pump manufacturer. Once the product is ready, it is essential to insure that the pump is really made to the specifications and will stand guarantee to the specified performance when it is installed in the field.

This brings about the need for an effective testing and inspection plan.

Even in this process, there is a need for effective communication and coordination with the purchaser, vendor, and sub-vendors on the extent of participation and the requirements of the testing and inspection plans.

A general overview of the inspection and testing requirements and guidelines are as follows:

- Inspection and testing requirements are based on
  - ANSI B73.1M: Horizontal Centrifugal Pumps
  - ANSI B73.2M: Vertical Centrifugal Pumps
  - API 610 – 9th Ed.: Centrifugal Pumps
- Specifying inspection and testing requirements
- Shop test acceptance criteria

- Preparation of inspection and testing checklist
- Review of the shop performance and procedures
- Reporting of test.

Usually, the above requirements are communicated through the pump specifications.

## 7.1 Material inspection requirements

The inspection requirements related to the material of construction of the pumps include:

- Material checks (chemical composition and physical properties)
- Casting defects and their classification
- Non-destructive testing (NDT)
- Repairs procedures of castings and welding.

### 7.1.1 Material checks

The material specifications of pump components are usually based on the API 610. The 8th Edition of API 610 – Table H-1 tabulates the material class for various pump components and Table H-2 provides with the ASTM specifications.

Even the purchaser can specify the materials based on its liquid properties and refer the applicable ASTM, ANSI, AISI, BS, DIN, or equivalent standards.

The standards usually provide with the chemical composition limits and the desired physical properties of the specified material grade. For example, the ASTM standard specifies the tests for the above from the heat from which the material is supplied.

Such tests are usually recommended in special applications like:

- Pump components are exposed to traces of hydrogen sulfide (in this case, materials of specific components have to conform to NACE standard MR-01-75)
- Pumps in highly corrosive or hazardous services
- Pumps in low temperature applications (less than –50 °C, some specifications consider –29 °C as the limit).

### 7.1.2 Casting defects and classification

Special attention is paid to the pressure containing parts of the pump, which include the pump casing and seal housings. Usually, the casting process is adopted to manufacture these components and hence special requirements are laid down to insure integrity of the components.

It is considered mandatory that the castings are free from any defects such as porosity, cracks, blowholes, shrink holes, scales, and any other serious defects.

In case any major defect is observed in the casting of a pump for cryogenic application or any special material, it is recommend carrying out the inspection in the following order:

- Check the Mill test reports for chemical composition and physical properties for compliance
- Check if any post repair heat treatment charts
- Check or witness non-destructive examination as required and specified
- Check and review welding procedures.

As per ASTM, the following repairs on a casting classify as 'major' repairs:

- Casting fails in the hydrostatic pressure test
- Repairs for which the depth, of any cavity prepared for welding, exceeds 20% of the wall thickness or 1 in. (25.4 mm) or whichever is lesser
- The cavity prepared for welding is greater than 10 square inch (65 $cm^2$)
- Any repair to a Cast Iron casting is treated as a major repair.

### 7.1.3 Non-destructive testing (NDT)

Material Inspection for Castings and Welding are carried out by the following NDTs:

- Visual Inspection (VI) – This has to conform to MSS SP-55
- Magnetic Particle Inspection (MPI)
  - Applicable Code – ASME E 709
  - Acceptance Codes
    (a) *Castings*: ASTM E 125
    (b) *Welding*: ASME Section VIII – Div. 1, Appendix 6.
- Dye Penetrant Checks (DP)
  - Applicable Code – ASTM E 165, Sec V, Article 6
  - Acceptance Codes
    (a) *Castings*: ASME Section VIII, Div.1, Appendix 7
    (b) *Welding*: ASME Section VIII – Div.1, Appendix 8.
- Ultrasonic Examination (UT)
  - Applicable Code – ASTM A 609, Sec V, Article 5
  - Acceptance Codes
    (a) *Castings*: ASME Section VIII, Div.1, Appendix 7
    (b) *Welding*: ASME Section VIII – Div.1, Appendix 12.
- Radiography (RT)
  - Applicable Code – ASTM E 94, ASTM E 142, Casting – E 446, E 186, E 280
  - Acceptance Codes
    (a) *Castings*: ASME Section VIII, Div.1, Appendix 7
    (b) *Welding*: ASME Section VIII – Div.1, W 52.
- *Impact Test*: Carbon steel materials in low temperature service below the ductile-brittle transition require careful selection to avoid brittle failure. In applications where the pumping temperature is –29 °C and below, the selected carbon steel materials shall meet the minimum Charpy 'V' notch impact energy requirements at the lowest specified temperature in accordance with paragraph UG-84 of ASME Section VIII, Div. 1. Some purchasers broadly classify the pumps in to three categories based on the pump pressure and liquid temperature.
- *Category A*: Casing pressure upto 40 bar-a with a pumping temperature range from 0 to 300 °C.
- *Category B*: Casing pressure upto 60 bar-a and pumping temperature range from – 29 to 400 °C and excluding the range covered in Category A.
- *Category C*: Range not covered by the above categories.

For Casing castings and welds falling under Category A, only visual inspection is carried out. For Category B components, it is either the dye-penetrant checks or magnetic particle tests that are carried out.

However, for Category C, it is DP or MPI and followed by UT or RT. Geometric and other considerations may make one or the other test unfeasible to conduct, however, these exceptions have to be agreed upon by both the inspector and the pump vendor.

These tests are carried out after a final heat treatment or final machining as maybe the case.

### 7.1.4 Repairs procedures of castings and welding

When any noticed defect qualifies for major repairs, a written procedure has to be accepted by the purchaser prior to carrying out the repairs.

The procedure should include:

- How the defect was detected
- Sketch/drawing indicating the location and depth of the defect
- The method of repair that includes heat treatment and inspection procedure after repair.

Usually peening, plugging, or impregnation of casting is not allowed as a repair procedure for ferrous pressure containing casings. Welding grades of steel maybe repaired in accordance with ASME Section VIII, Division 1 and ASME Section IX.

The above ASME codes are also applicable to other critical welds of pump components. However, structural welding that may include base plates or any other not covered by the ASME codes should be welded considering AWS D1.1 standard as a minimum.

## 7.2 Shop tests

The shop tests include the Hydrostatic Test and Shop Running test.

The shop acceptance tests as per the API are classified as:

- Witnessed
- Observed.

Witnessed Test as stated by API 610 is an agreement in which a hold is applied to the production schedule and the inspection or test is carried out in the presence of the purchaser or its representative. In case of performance or mechanical running test, the manufacturer has to notify the purchaser of a successful preliminary test.

Observed Test as stated by API 610 is an agreement in which a manufacturer informs the purchaser of the forthcoming inspection and testing of the pump. The inspection and testing is done as per the schedule. If, however, the purchaser or a nominee is not present for the test, the vendor may not proceed to the next step.

### 7.2.1 Hydrostatic test

All pressure containing parts that include the auxiliary components/piping shall be hydrostatically tested at 1.5 times the maximum allowable pressure that can be anticipated from the pump or system.

All cooling water jackets or passages should be tested at 7.9 bar-g or as specified by the purchaser.

When hydrotesting stainless steel pumps, it is advisable to insure that the chloride content in the testing water is less than 50 ppm.

The hydrotest is considered satisfactory when no seepage or leak is observed for atleast 30 min as per API 610. ANSI standard specifies 10 min and the Hydraulic Institute Standard specifies 3 min for pumps less than 100 HP and 10 min for pumps more than 100 HP.

A failure of a hydrotest due to leaks from other than bolted or threaded joints is considered a major failure and the purchasers' written approval is required prior to any repairs to rectify the defect.

### 7.2.2 Shop running test

These tests are carried out to verify the pump performance and the mechanical integrity of the unit, which includes the vibration and noise levels.

The requirements of the pump hydraulic and mechanical performance are clearly defined in all applicable specification/standards and in data sheets prior to issue of the bid inquiry.

Not all pump standards specify a mandatory performance test.

ANSI B73.1M and B73.2M specifications for horizontal and vertical centrifugal pumps do not mandate hydraulic or mechanical performance test. When test is specified, the Hydraulics Institute Standard or more stringent limits could be imposed.

As the ANSI pumps are manufactured and sold as standard products and sold as of the shelf equipment, the standardization allows for skipping of the tests.

However, Industry standards for ANSI pumps recommend a performance test in the following conditions:

- Pumps are operating in parallel
- Suction-specific speed is more than 11 000
- Normal flow is less than 10% above minimum continuous flow rate.
- NPSH-r test is required when the NPSH-a to NPSH-r difference is less than 1.8 m.

The API 610 standard makes the hydraulic and mechanical performance test mandatory for all pumps.

### 7.2.3 Shop acceptance criteria

Following are the acceptance criteria for pumps as per the two main pump standards.

| | **Pump Specification Requirements** | | | | | |
|---|---|---|---|---|---|---|
| | **API 610 (7th Ed.)** | | | **Hydraulic Institute Standard, ANSI and Industry Practice** | | |
| | | | | | A | B |
| Rated differential head | Condition | Rated | Shut-off | Under 200 ft, 2999 gpm | +5/−0% | +5/−3% |
| | Hd: 0–150 m | –2/+5% | +10/−10% | Under 200 ft, 3000 gpm and above | +5/0% | +5/−3% |
| | 150–300 m | −2/+3% | +8/−8% | 200–500 ft, any flow rate | +5/0% | +5/0% |
| | Over 300 m | −2/+2% | +5/−5% | 500 ft and over | +3/0% | +3/0% |

*(Continued)*

| | **Pump Specification Requirements** | | | |
|---|---|---|---|---|
| | **API 610 (7th Ed.)** | **Hydraulic Institute Standard, ANSI and Industry Practice** | | |
| Capacity at rated head | – | | +10/−0% | +5/−5% |
| Efficiency | −0.5% | | 0% | Formula |
| Rated Power | +4% | | – | – |
| NPSH-r−3% head drop | +0 m | | | |
| Retest | When reduction in impeller diameter demands more than 5% of original diameter | When reduction in impeller diameter demands more than 5% of original diameter | | |
| Mechanical seal leakage test | Mutual agreement | Not specified | | |

## 7.3 Performance test procedure

The performance test of horizontal or vertical pumps needs different configurations of the pump testing facility

Typical layout of a test stand for a horizontal pump is shown in Figure 7.1. This is a closed loop configuration, which implies that the same water is recirculated for the entire test duration, however, in some cases there could be an open loop configuration.

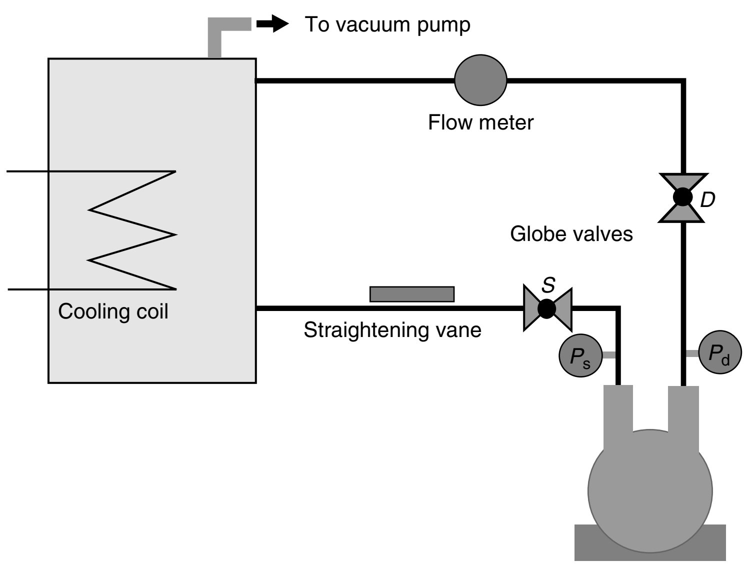



**Figure 7.1**
*Typical layout for a horizontal pump shop running test*

A test stand for a pump comprises of various equipment instrumentation. These include:

- Storage tank or pit for test fluid
- Slotted base for supporting the pump, drivers, and auxiliary equipment with different sizes
- Piping on the suction and discharge of the pump, equipped with globe valves for a proper flow control and a flow meter
- A negative and positive pressure source for conducting NPSH tests
- Pressure Indicators on the suction and discharge installed as close as possible to the pump casing.
- Prime mover to drive the pump
- Instrumentation to accurately measure the power delivered by the prime mover
- Monitor horsepower with a torque meter or calibrated motor
- Measure speed of the pump to check for variations in electrical supply frequency
- Data collector/analyzer/proximity probes to measure and record data
- Decibel meter to measure noise levels
- Thermometers to measure bearing housing temperatures.

All the above-mentioned instrumentation should be calibrated semiannually or annually based on the instrumentation used and the experience of the shop testing facility.

A calibration certificate should be available on-demand and the instruments should carry calibration stickers.

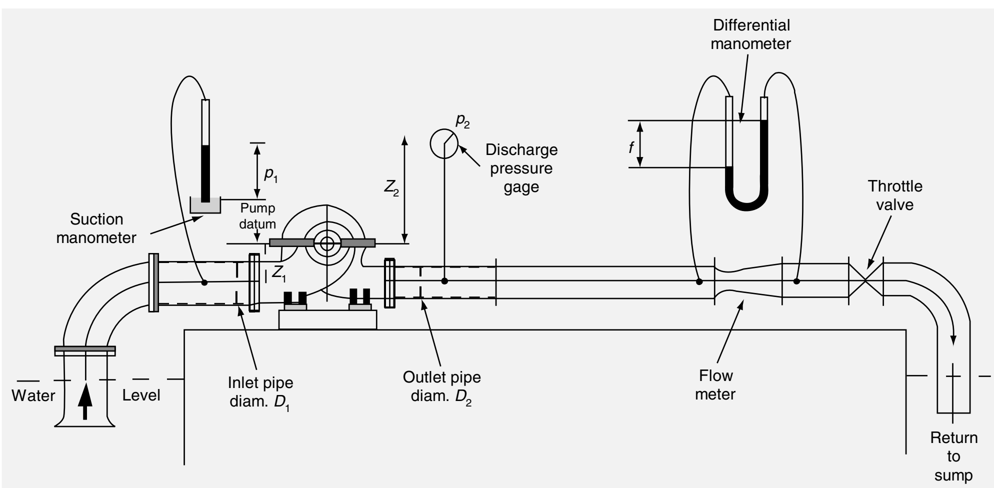



**Figure 7.2**
*Typical test rig for split casing pump*

The pump is mounted on the test base and clamped. Depending on its size, rating, and test specifications, either the vendor's motor or the motor procured for the pump is used for the test. This motor is then aligned to the pump and electrical connections are made. The suction and discharge piping are connected (Figure 7.2).

The required instrumentation is connected and activated. The pump is then started and allowed to stabilize. Once the pump and motor temperatures flatten out, the following readings are taken.

A recorded test data comprises of:

- Flow rates
- Discharge pressure
- Suction pressure
- Elevation corrections
- Test fluid temperature
- Test fluid specific gravity
- Power reading
- Voltage at driver
- Current to the driver
- Power factor of the supply
- Frequency of the electrical supply
- Vibration levels
- Bearing temperatures
- Noise levels
- Speed.

Usually, five to seven test points' data are recorded from a low range of shut-off to 120% of the rated flow. The test points must include the normal and the rated flow rates.

The above is achieved starting the pump with a closed discharge valve, which is the shut-off condition. This valve is then opened in the five to seven steps as mentioned to get the different flow rates.

When the data collection is completed, it is tabulated. Computations for speed, density, viscosity, and elevation corrections are made. Corrected flow rates and differential head in meters of liquid column are filled in the table. Power and efficiency of each point is calculated and entered in the table. A standard pump performance test log is shown in Table 7.1.

Once the computations are done, the deviations are checked to be within the tolerance of the acceptance criteria.

The computation procedure is covered in Chapter 3 on Pump Hydraulics.

The next step is to plot the tested performance curve based on the data collected during the running test.

In case, the shop running test fails to meet the acceptance criteria, corrective actions have to be taken and a possible retest.

The common causes of pump test failure are:

- Incorrect Impeller diameter
- Impeller with high residual imbalance
- Unexpected seal failure
- Rubbing at wearing rings
- Poor Impeller surface quality
- Uncalibrated instruments
- Incorrect data collection and computational errors
- Misinterpretation of test results and acceptance criteria.

TEST LOG NO.

CLIENT DETAILS: JOB NO: TEST:

Barometric Pressure 979 mo; Water Temperature 16 ºC; Pump Number ZT 5500; Chart Number T45500; Date 31 Jan 86
Pump Size/Type: 200 × 250 – 500 2T; Impellet Dia 449 mm; Inlet Dia 250 mm; Oulet Dia 200 mm
Motor Make: TECO Frame 280MC; Serial No 5C10 961; kW 150; V 415; A 256; r/min 1460
Motor Efficiency @ 1.00/0.75/0.50 Load = 0.939/0.945/0.945; kW: w 100; Drive: Direct Coupled
kq 8.671; kp1 = 013595; kp2 1019; z1 – 0.3 m; z2 – 0.815 m; kd 3.0528E-5; Start Time

| No | F | p1 | p2 | W | EkW | ee | kW | L/s | H | ep | eo | k | r/min | NPSH |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | 615 | 198 | 494 | 0.925 | 148.5 | 0.941 | 139.8 | 215.0 | 55.56 | 0.838 | 0.789 | 0.1918 | 1483 | 7.79 |
| 2 | 526 | 178 | 534 | 0.899 | 143.8 | 0.942 | 135.5 | 198.9 | 59.16 | 0.851 | 0.602 | 0.2009 | 1484 | 7.92 |
| 3 | 408 | 152 | 589 | 0.846 | 135.4 | 0.943 | 127.7 | 175.1 | 64.14 | 0.863 | 0.814 | 0.2147 | 1485 | 8.08 |
| 4 | 328 | 135 | 631 | 0.806 | 129.0 | 0.944 | 121.8 | 157.0 | 68.00 | 0.860 | 0.612 | 0.2201 | 1486 | 8.16 |
| 5 | 249 | 117 | 663 | 0.741 | 118.6 | 0.945 | 112.0 | 138.8 | 70.84 | 0.848 | 0.802 | 0.2407 | 1487 | 8.90 |
| 6 | 191 | 103 | 888 | 0.684 | 109.4 | 0.946 | 100.5 | 119.8 | 73.06 | 0.830 | 0.765 | 0.2537 | 1458 | 8.40 |
| 7 | 132 | 91 | 712 | 0.612 | 97.9 | 0.946 | 92.6 | 99.6 | 75.21 | 0.793 | 0.750 | 0.2730 | 1459 | 8.47 |
| 8 | 72 | 78 | 731 | 0.515 | 82.4 | 0.945 | 77.9 | 13.6 | 76.83 | 0.712 | 0.673 | 0.3111 | 1491 | 8.55 |
| 9 | 21 | 66 | 740 | 0.373 | 59.7 | 0.943 | 56.3 | 35.7 | 77.47 | 0.537 | 0.506 | 0.4172 | 1493 | 8.53 |
| 10 | 0 | 60 | 747 | 0.311 | 49.8 | 0.941 | 46.8 | 0.0 | 78.05 | 0.000 | 0.000 | 0.000 | 1495 | 8.88 |

Finish Time: 1130 Guaranteed Duty—175 L/s. 64 m. 86% Efficiency.

The Pump was tested in accordance with Australian Standard No. 2417 Part 3–1980 [Class B Tests].
The Results obtained comply with the guarantee and the recommendation is that the pump be accepted.

Tested By: Witnessed By:

**Table 7.1**
*Standard pump performance test log (Courtesy of the Australian Pump Handbook)*

**Notation**

| | |
|---|---|
| I | – Venturl or Orifice Differential Pressure |
| A | – Rated Full Load Motor Amps |
| Inlet Dia | – Inlet Diameter in mm (Pipe) |
| kd | – Velocity Head Conversion Coefficient<br>velocity head = $\frac{v_2^2 - v_1^2}{2g} = kd \times (l/s)^2$ |
| kq | – Venturl of orifice Flow Coefficient |
| kp1&kp2 | – Factors to convert inlet and outlet readings to m |
| KW: w | – Instrument Transformer Ratio (Current and Potential Transformers) |
| Outlet Dia | – Outlet Diameter in mm (Pipe) |
| p1 | – Inlet Pressure (mm Hg) |
| p2 | – Outlet Pressure (kPa) |
| V | – Rated Motor Voltage (Volts) |
| w | – Wattmeter reading |
| z | – Corrections for gauge height = $z2 - z1$ |
| z1 | – Distance from pump datum to p1 (m) |
| z2 | – Distance from pump datum to p2 (m) |
| ee | – Drive Efficiency |
| EkW | – Electric Kilowatts to drive = $w \times kW{:}\ w$ |
| eo | – Overall Efficiency (pump and Drive) |
| ep | – Pump Efficiency |
| H | – Total Head.<br>Metres = $(p2 \times kp2) - (p1 \times kp1) + z + \text{Vel Hd.1}$ |
| K | – KWh/1000 Litres = $\frac{EkW}{3.6 \times L/s}$ |
| kW | – Pump Input Power (Kilowatts) = $EkW \times ee$ |
| L/s | – Litres per second = $kq\sqrt{1}$ |
| NPSH | – Nett Positive Suction Head at Pump Datum (m) = Barom.<br>Pressure + $(p1 \times kp1) - \text{Vap. Press} + \frac{v^2}{2g} + z1$ |
| r/min | – Pump Revolutions per Minute |
| v1 | – Fluid Velocity at tapping point p1(m/s.) |
| v2 | – Fluid Velocity at tapping point p2(m/s.) |
| Vp | – Vapour Pressure (PSI) of Water |
| ºC | – Water Temperature (celsius) |
| g | – Acceleration due to Gravity (9.7982 m/s$^2$) |

**Evaluating pump performance from data logged at the Pump Performance Test**

Calculation of Total Head – H

$$H = (p2 \times kp2) - (p1 \times kp1) + z + \text{velocity head}$$

Now;
Z = gauge height correction = 0.815 − (−0.3) = 1.12 m

$$\text{Velocity head} = (v_2^2 - v_2^1)/2g = kd \times (L/s)^2 = (3.0526 \times 10^{-5}) \times (215)^2 = 1.41 \text{ m}$$

Therefore, evaluating pump performance based on Test point No 1 in the table, we get
Total head 'H' = (494 × 0.1019) − (198 × (−0.013595)) + 1.12 + 1.41 = 55.56 m

Input power at the pump = (Electrical power to drive × drive efficiency)

$$= w \times kW{:}w \times ee = 0.928 \times 160 \times 0.941 = 139.8 \text{ kW}$$

$$\text{Pump efficiency} - ep = (L/s \times H)/kW \times 101.972 = (215 \times 55.6)/(139.8 \times 1010.972)$$
$$= 0.838 \text{ or } (83.3\% \text{ efficient})$$

$$\text{Over all efficiency 'eo' of the pump set} = ep \times ee = 0.838 \times 0.941 = 0.789 \text{ or } (78.9\%)$$

The net positive suction head available (NPSHa) = Barometric Pressure + (p1 × kp1) − Vapour pressure (at 17 °C = 0.19 m) + velocity head + z1 (gauge height correction)

$$\text{Note} - \text{Flow velocity 'v'} = \text{'f'/cross-sectional area of suction pipe}$$
$$=(215 \times 1000)/(3.14 \times (2502/4)) = 4.38 \text{ m/s}$$

and $v^2/2g = (4.38)^2/2 \times 9.8 = 0.98$ m
Therefore, in this typical pump test rig set up,

$$\text{NPSHa} = (979/10 \times g) + (198 \times (-0.013595)) - 0.19 + 0.98 - 0.30$$
$$= 9.99 - 2.69 - 0.19 + 0.98 - 0.30 = 7.79 \text{ m}$$

## 7.3.1 NPSH testing

NPSH-r testing of pumps is usually done by two methods:

1. Reducing the suction pressure by throttling the suction valve
2. Reducing the suction pressure by pulling a vacuum in the suction vessel.

The pump is run at a constant flow rate and speed while the suction pressure of the pump is reduced by any one of the methods mentioned above.

The reduction in the suction head leads to a reduction in the discharge head. For a particular flow rate, 5 to 12 readings maybe taken and plotted on a graph.

Such a set of readings maybe repeated for a number of flow rates ranging from 1 to 4.

As mentioned in the earlier chapters, the Hydraulics institute defines the NPSH-r as the value of the suction head at which the discharge head drops 3% from the rated head at that flow rate (Figure 7.3).

In case of multistage pumps, the 3% drop in discharge head is not the total head. The discharge head considered is of the first stage. This is obtained by dividing the total head of the pump by the number of stages to get an approximate head of the first stage.

In case the pump is dismantled to correct the NPSH-r of the pump, it calls for a retest.

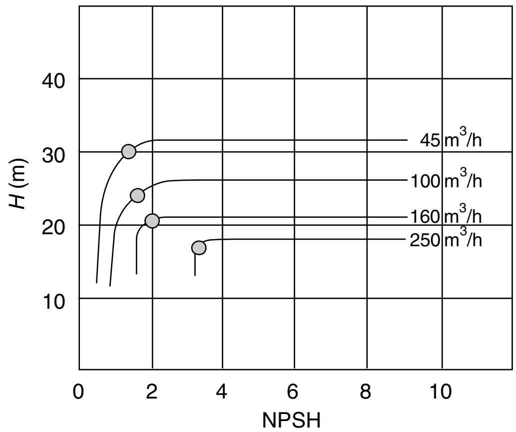



**Figure 7.3**
*NPSH test curves – the dots indicate the 3% head drop*

## Dismantling of pump

Some standards specify that the pump be dismantled after the shop running test. This is generally done to inspect the bearings and mechanical seal.

However, if the pump has performed satisfactorily during the shop test, the dismantling of the pump can be skipped out.

It is only in special circumstances like inconsistent test results, abnormal noise, higher vibrations, or a prototype design that the dismantling of the pump after the test is required.

The review of the pump hydraulic and mechanical performance by shop tests insures that there are no unexpected deviations or ambiguity as regards to its performance after installation and commissioning.

Correct and accurate pump specification; inspection and testing are the building blocks for a reliable operation. The next block is the process of installation of the pump and this is covered in the next chapter.

# 8

# Pump installation and commissioning

After proper specification, selection, sizing, inspection, and testing, installation is the next key factor in the reliable operation of any centrifugal pump. When properly installed, operated, and maintained, a pump can offer many years of trouble-free service.

However, when pumps are incorrectly installed, maintenance and operational problems will impede its performance.

Careful preparation and planning is needed to insure proper installation of all pumps. It is a coordinated effort between the supervising engineer, the mechanical and electrical contractors, and suppliers.

The installation process is a series of many steps.

## 8.1 Site location

This is one of the pre-installation activities. Prior to the receipt of the equipment, drawings specifying layout dimensions are available and this enables one to select and mark out the site location for the equipment.

Ergonomic considerations are a prime factor in the selection of a proper site. When equipment is accessible for maintenance, technicians perform better and operators activate and control it more efficiently.

A pump or its motor that is difficult to access and maintain becomes a cause for longer downtime and lower availability.

Poor workmanship due to difficult access leads to lower reliability.

Safety is the main aspect. A difficult site has a higher probability of accidents.

Thus, this is a factor that requires considerable attention and has to be done by a person with good practical experience in operation and maintenance of pumps.

## 8.2 Receipts and physical inspection

All pumps and auxiliary equipment or components should be examined upon receipt for any signs of apparent damage. If any damage is indicated, it should be notified.

In case the installation is not planned immediately, it is best to store it in a clean, dry location where it will be protected from possible damage. When storing equipment, it is best to follow the manufacturer's recommendations and protect it from environmental extremes.

It is also advised to check the equipment prior to storage to anticipate any possible problems at the time of installation. It is a good idea to check sizes, design features called for on the plans and specifications, and all interface components.

With these simple early checks, installation problems can be minimized to a great extent.

Following are some good recommended practices:

- All nozzles, openings should be kept covered or plugged until the piping is attached.
- The bearing housings should be filled with oil of the recommended viscosity. If greased bearings are installed, new grease should be pumped in and old one should be displaced.
- All exposed surfaces should be coated with a rust preventive. If the pump is anticipated for preservation for more than 6 months then the internals too should be coated with suitable rust preventive or an oil mist.
- Pump packings with sleeves should be removed.
- Careful handling has to be taken for pumps installed with mechanical seals. They should not be subjected to impact or excessive vibration.

## 8.3 Pre-alignment checks

In the event of an immediate installation after the receipt of the pump, an alignment check of the pump with its motor on the base-plate should be carried out. This is to insure that it is possible to achieve the final alignment tolerances as per the specification. This check is recommended using a reverse dial indicator method or laser alignment method.

If the specified alignment is not achieved, there is still time to rectify the faults. This saves time and money and prevents quick fixes if the problem was detected at the final stages of installation. A correct alignment is necessary for pump operation.

If correct alignment is achieved, the pump and motor can be removed from the base to ease the installation and help prevent damage to critical components.

## 8.4 Location of pump foundation

Once the site location has been fixed, the location of pump foundation is often a job of fine-tuning to bring it in line with the existing equipment like piping, vessels, and any other.

The factors for site location are not lost here and one has to keep in mind when choosing the exact location. It should allow room for walkways, existing piping, new piping, operator, technician accessibility, and aesthetic considerations.

The aspects that can usually be altered are the orientation of the pumps face, their closeness to walls, and the height and depth of the pump foundation.

If the pump height or location is altered, one has to insure that the operating parameters of the pump can still be met. The prime consideration is the NPSH margin or ratio. If installation standards or guidelines already exist, it is essential to obtain the concerned engineer's consent.

## 8.5 Design and dimensions of pump foundation

The pump foundation has two specific purposes:

1. It serves as a support for the pumps to operate in a safe manner.
2. The foundation mass will damp the pump vibrations.

The pump foundation must provide enough rigidity to absorb axial, transverse, and torsion loads that the rotating pump imposes. Proper structural design of a foundation requires an evaluation of soil conditions so that both dynamic and static forces are considered.

A foundation design has to take into consideration the following aspects:

- Functional support to the pump; it should have a mass – that is, at least thrice the total weight of supporting equipment.
- The foundation rests on solid or stabilized earth that is completely independent of other foundations, pads, walls, or operating platforms.
- A minimum of 3000 psi steel reinforced concrete should be used.
- The foundation's resonant frequency cannot be excited by pump operating speed or multiples of operating speed.
- All units, including the pump, gearbox, and motor rest on a common foundation.
- The foundation is designed for uniform temperatures to minimize distortion and misalignment.
- The foundation is designed taking into account the seismic activity in the region.

Some rules of thumb are followed for general pump foundation design and dimensions (Figure 8.1).

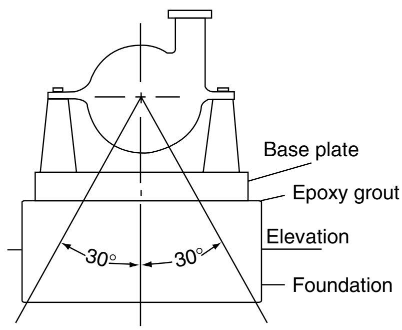



**Figure 8.1**
*A pump on its foundation*

- Drop two lines from the pump center that are 30° to the vertical. The width of the foundation should be more than its spread.
- Weight of foundation should be a minimum 3 times more than the mass of supported equipments.
- For pumps less than 500 HP, the distance between base plate edge and foundation edge, all the way around should be at least 3 in. For pumps with higher horsepower, it should be 6 in.

## 8.6 Excavation and forms for pump foundation

Once the site location, orientation, and dimensions of the pump foundation are fixed, the data can be used to mark the location, and excavate a correctly sized cavity to the correct depth. The usual requirements are a cement saw, jackhammer, air supply, wheelbarrow, shovels, a small front-end loader, a disposal method for the excavated material, and permits for digging.

It usually takes a maximum of two days to excavate for an average-sized centrifugal pump.

Once the right size excavation has been carried out, the next step is placing the forms. These can be reusable forms or one-time use forms. It is important that the forms are true, solid, well-braced, and liquid-tight. The leak tightness is required as epoxies have a tendency to seep through small holes, a lot more readily than the cements.

Once the forms are installed, it is essential to insure that these are level, square, and securely fastened to the floor or the ground.

This is essential as the weight of the cement and epoxy would bear on the forms and could easily cause it to shift. If the form shifts during pouring, the job has to be terminated and restarted.

When epoxy pours are considered, it is essential to the insides of the forms with anti-stick materials. This helps in removal of the forms without having to destroy them.

Wax is a good anti-stick material, it coats well and the forms can be coated before they are assembled and installed.

## 8.7 Rebar and anchor bolts

The next step in the installation process is to install the rebar and pump hold-down bolts.

Depending on the dimensions of the foundation, the numbers of rebar rings and posts are installed. A foundation of 60"(L) × 60"(W) × 36"(D) would typically have three rings of rebar tied to eight posts of rebar (Figure 8.2).

The rebars are placed several inches from the sides, top, and bottom of the foundation. These are equally spaced from the top and bottom.

**Figure 8.2**
*Rebar bolts*

Once the rebar is installed, anchor bolts are provided. The length of the anchor bolts is typically 10 to 15 times the bolt diameter. This is required for proper stretch to develop the design holding force (Figure 8.3). If epoxy grout is allowed to grip the anchor bolt, the bolt will break at the grout surface even when tightened to the design torque. This requirement is met at the foundation design stage and this recommends the use of bolt sleeves in the concrete. When sleeves are used, they should be filled with non-binding material like sand, flexible foam, or wax to prevent epoxy from bonding to the anchor bolt.

The exposed length of the anchor bolt from top of the concrete to the bottom of the base plate could also be wrapped with one layer of weather stripping and one layer of duct tape.

An acceptable hold-down or an anchor bolt for most ANSI pumps is 5/8 in. J-bolt mounted in a 1-1/2 in. pipe. The pipe is at least 6 in. long.

The J-bolt extends past the top of the pipe by a minimum of the pump base height plus1-1/2 in.

The J-bolt extends past the bottom of the pipe an inch or two before the 'J' bends. A washer is welded to the bottom of the pipe. The J-bolt passes through it. Subsequently, the J-bolt is welded to the washer.

**Figure 8.3**
*Straight and J-type anchor bolts*

It is important that the bolts are true to themselves and with respect to the other bolts. This can be achieved by having a single bracket that secures all the bolts.

The stage is now set for pouring of concrete and epoxy.

## 8.8 Pouring

The pouring of the concrete mix is carried out slowly while stirring the cement. A cement truck does this; however, if the site is not accessible by a truck, the cement can be pumped or loaded onto a wheelbarrow and transferred to the site.

### 8.8.1 Concrete mix pour

Curing of freshly poured concrete must occur before epoxy grout is applied. The epoxy-concrete bond is sensitive to the presence of moisture and this needs to be prevented at all times. In either instance, trowel-finish the top of the pad when the cement is ready for finishing.

It is recommended to carry out ASTM 157-80 concrete shrinkage test to determine when the shrinkage drops to the minimum. This is an indicator of the end of the chemical reaction between cement and water, which causes the concrete to cure.

In the absence of the above test the following thumb rule is adopted:

- Standard concrete (5 bags mix) – 28 days
- Quick setting concrete (6–7 bags mix) – 7 days.

One method to check on moisture is to tape one square feet of plastic sheet over the concrete block and leave it overnight. If there is moisture on the underside of plastic, the concrete is still not ready for epoxy grout. This should be repeated until no moisture is seen under the plastic sheet.

Surface preparation of the new concrete begins two to three days after the pour. It maybe necessary to chip the top 1/2" to 1" from the surface of the foundation to remove the cement-rich surface called laitance. The laitance is a weak surface created when concrete is cast and would not provide for proper adhesion or support for the grout that is added under the base plate.

Usually sand blasting is the technique used to remove the laitance and expose the aggregate. The most common method is to wait for the concrete to cure and then chip the laitance with light-duty pneumatic hammers. Jackhammers and sharp pointed chisels should not be used for chipping.

All foundation edges should be chamfered at least 2 to 4 in. at 45° to remove stress concentration. All dust, dirt, chips, oil water, and any other contaminants should be removed and the foundation should be covered.

### 8.8.2 Epoxy pour

Sometimes it maybe necessary or advantageous to have the entire foundation made out of epoxy.

The initial costs are higher but these foundations offer:

- Superior vibration dampening
- Better chemical resistance
- Faster curing time. Even the quick setting cement does not cure before 7 days to bear the forces needed to attach the equipment. The epoxy gets cured in 24 h, and allows the equipment installation to begin almost immediately.

The procedure involves mixing the epoxy parts A (resin) and B (hardener) as per the instructions. It is better to mix both the parts completely and not leave behind unmixed parts. Unmixed parts are a hazard and need special disposal. However, the mixture of the two parts is not hazardous.

The next step involves adding the mixed resin into an empty mortar mixer, and then adding the aggregate. The aggregate consists of pure silica, some with the texture of sand, and some with the texture of pea gravel mixed at a specific ratio.

The ratio is based on the ambient conditions. Pure silica is used instead of sand and pea gravel because of its superior heat-sink capabilities. This also adds to the overall strength of the pour. Once the silica aggregates are added to the mixed resin, they should be mixed until a uniform consistency is achieved. The mixing is done slowly to ensure that no air is entrained in the batch, as air is detrimental to the overall strength of the pour. A spiral-blade mortar mixer is best suited for this application.

The mixed epoxy should then be poured into the hole, and the process repeated until the pour is complete.

An average-size foundation maybe poured in less than 4 h when properly administered. Any finishing touches need to be completed before the epoxy cures.

The next step involves placing the base plate and grouting it.

## 8.9 Base plate and sole plate preparation

It is recommended to remove all the equipment from the base plate or sole plate prior to grouting.

This helps to:

- Level the plate
- Reduce unwanted distortion.

The pump and motor/turbine/engine can be mounted after the base plate has been properly grouted.

The base plate surface to be in contact with the grout should be coated with an inorganic zinc silicate or any compatible primer. The base plate should have bare or rusted surface and should be free of blisters. The surface should also not be smooth, as this may not allow for proper bonding with the epoxy grout.

It should be checked that the base plate is provided with at least one grouting opening in each bulkhead section and/or each 12 sq. feet of base area as a minimum. Vent holes should be provided at the corners of each bulkhead compartment. These insure that no voids are created by trapped air.

The corners of all base plates should be rounded to 20 in. radius. When epoxy cures, it shrinks and rounding prevents stress corrosion in the grout. If sharp corners were left, it would eventually cause cracking of the grout.

Before placing the base plate on the prepared foundation, it should be free from oil, grease, and rust.

After the base plate is rested on the foundation it should be supported on leveling screws, rectangular leveling shims or taper wedges placed close to the foundation bolt to prevent distortion. Leveling screws should be adequately coated with grease/wax to prevent adhesion of epoxy to the screws.

The base plate should then be leveled side-to-side, end-to-end, and diagonally to within 0.002 in. per foot. The machined surfaces have to be flat and parallel. The mounting surface tolerance should remain the same even after the anchor bolts have been tightened.

Once leveling has been achieved, it should be confirmed that all the wedges/shims are in contact with the base plate and foundation. The foundation bolts are then evenly tightened and the levels are rechecked.

Before the grout is poured, the elevation of the machined surfaces should be checked to insure that it would allow for a minimum of 1/8" shim thickness under the driving equipment.

Insure that eight alignment positioning screws are provided for positioning the driver.

The machined mounting surfaces should extend 0.1 in. beyond the pump and driver feet on all sides.

Two holes should be drilled and tapped on the base plate flanges on each side of the anchor-bolt holes to make provision for 1-1/2 in. jack/leveling screws.

The coupling guard bolts should be greased and inserted in the base plate. It is difficult to drill and tap holes in case they are filled up with epoxy grout.

## 8.10 Grouting

The term ‘grout’ refers to a hardenable material such as a mortar, concrete, or epoxy, which is placed under and around the base plate to assure intimate contact with the foundation.

The main reasons for grouting are:

- To provide uniformly distributed load bearing surface
- To provide effective damping to machinery vibration
- To fill cavities and cover projections thereby eliminating unsafe conditions and improving performance.

There are basically two types of grouts in use:

1. Epoxy grout, consisting of three parts, resin, hardener, and an aggregate
2. Cement plus a natural or metallic aggregate.

### 8.10.1 Epoxy grout

The first step in the process to carry out an epoxy grout is to layout the forms. These forms should be of heavy-duty design as the weight of epoxy is nearly 2.5 times than that of the concrete. It is recommended to use 3/4 in. plywood with adequate bracing.

The surface of the forms that will come in contact with the epoxy needs to be wax coated to insure their easy separation after the epoxy has hardened. Usually, three coats of wax are applied with sufficient intervals between the coats to allow for penetration of wax in the wood and drying.

The forms should have 1 in., 45° chamfer in the vertical and horizontal edges so that the mating of the forms allows for minimum seepage of epoxy from the forms. If required a leak-tight joint can be formed with the use of a plastic type of sealant at all joints and at the interface with the foundation.

The base plates that have been designed as per API requirement require two-pour grouts:

- To fill the void between the concrete and the base plate flanges
- To fill the void between the base plate flange and the top of the base plate.

If the free surface of the grout at the base-plate flanges is confined the 6–7 in. higher, grout level at the top of the base plate can be filled in one pour.

The vent holes usually 1/2 in. in diameter that are drilled allow the air to escape as the grout is poured from the center of the base plate to the edges. When the grout overflows from the vent holes, duct tape is used to cover the holes and the filling operation is continued.

One-pour grout can be accomplished in 45 min and after the curing is complete in 24 h, the forms can be removed. In case the ambient temperature is above 25 ºC, the pump and its driver can be installed after the forms are removed.

A two-pour job needs more time and cost.

Epoxy grouts have a narrow range for mixing and placement. This range is from 10 ºC to 35 ºC for best life, flow ability, and curing. When the temperatures are lower, low-temperature accelerators can be added with the foundation and base plate kept heated. When temperatures are high, temporary shades can be placed over the base plate 24 h before the pouring and 48 h after pouring of the grout.

With a temperature-conditioned grouting, insure that all tools tackles are in place and that all items in the checklist have been ticked.

The next step is to mix the grout. This can be done in a wheelbarrow with a mortar-mixing hoe or in a motorized concrete or mortar mixer

The latter maybe used in case of when the grout area is larger (10 units or more). Care should be taken to keep the blade speed limited to 15 rpm.

When hand mixing is used, two wheelbarrows are used to insure that there is a continuous supply of grout to men pouring it in the forms.

In either of the cases, mixing has to be done slowly with due care taken to prevent formation of froth or air entrapment in the pour mix. Mixing is done for 3–5 min after adding the hardener to the resin.

It is better to record the timing of mixing and pouring of the grout and insuring that it is done as per the specifications.

The pouring of the grout is done slowly using a large funnel placed about a meter above the base plate to provide the necessary force to push the grout out of the vent holes. Alternately, a positive displacement pump maybe used to perform the same function.

Random grout samples maybe taken along with the ambient temperature and location of grout for analysis and records.

When the epoxy begins to harden, it is better to form domes to insure that no water accumulates in the low areas. These can be removed after 24 h. The jackscrews can be relieved after 3 days.

After full-cure, sealing material (duct tape) and the wedges or shims should be removed. Silicone caulking is used to fill the shim holes. The anchor bolts are then tightened to the recommended torque.

The grout job should be checked for voids by ringing the base plate with a hammer. A good job will sound like hitting a lead plate and the one with voids will ring like a bell.

### 8.10.2 Concrete grout

The procedure for the cement-based grout is quite similar to the one adopted for the epoxy grout (Figure 8.4).

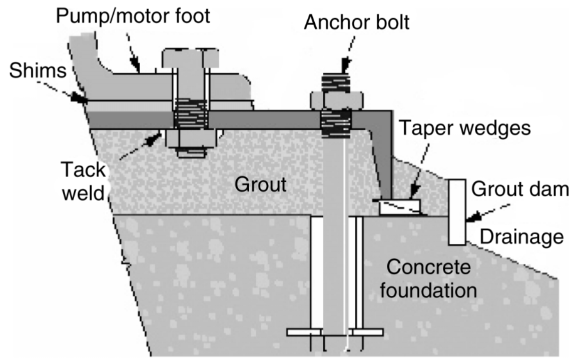



**Figure 8.4**
*Concrete grout (Image source – Berkeley pumps – USA)*

As in the epoxy grout procedure, the first step is to layout the forms and similar precautions need to be taken.

When the foundation is of concrete, the top surface should be kept saturated with water for a specified period of time as per the recommendations of the grout manufacturer. This water from the top of the foundation and boltholes should be removed just prior to placing the grout.

Precautions should be taken to insure that grout does not enter the anchor bolt sleeves and hence the sleeves are filled with non-bonding pliable material such as asphalt or silicone rubber molding compound to prevent a water pocket around the bolt.

A duct tape maybe used to wrap around the exposed threads of anchor bolts to prevent direct contact between the grout and the anchor bolts.

The temperature conditions required are again quite similar to the one stated for the epoxy grout. The temperature range has to be maintained in this case for a minimum of 24 h after pouring of the grout.

The preparation of grout has to be done as per the instructions with the right amount and quality of oil-free water.

The placement of the grout has to be done rapidly and continuously. It is recommended to start placing the grout from one end of the base plate and work toward the other end to insure that all air is positively vented and no air pockets are trapped.

Grout should be cut back to the bottom outer edge of the base plate or sole plate and tapered to the existing concrete. The top of the grout on base plates with flange-type support should be at the top of the flange. The top of the grout on base plates with solid sides and soleplates should be 1 in. above the bottom of the base plate or underside of the sole plate. The outside top edges of the grout should be chamfered at 45º.

After the initial set, it should be trimmed to the levels indicated in the drawings. After the complete curing of the grout has been achieved, it should be checked for air voids with a hammer test. Any voids detected should be pressure grouted.

The forms can be taken out after 24 h. The leveling shims and wedges may also be removed after the grout has cured. The voids from these should be filled with grout without the aggregate.

When leveling screws are used these should be removed after the grout has cured to allow the full equipment weight to be distributed evenly over the grouted area. The holes should be caulked with putty.

Subsequently, the anchor bolts should be tightened to the right torque.

The pump and its driver are then ready for installation.

## 8.11 Installation of pump and driver

Once the grout is cured and the pump base is clean, the pump and its driver can be installed. The bolts that were used to seal the needed boltholes can be removed. The pump should be in operation-ready condition when it is installed on the base.

Whether the driver is an electrical motor or a turbine, they should be placed at the locations indicated in the drawings. It is essential to confirm the distance between the shaft ends (DBSE).

The DBSE should be set with the pump and driver shafts pulled toward each other. For motor drives with sleeve bearings, the DBSE should be set with the motor shaft at its magnetic center.

In case of an electric motor, the motor should be wired correctly to insure the correct direction of rotation. This check has to be carried out before the equipment is coupled up. Pumps, seals, or magnetic drive bearings can be ruined if operated dry or in reverse.

After the rotation check, the motor should be de-energized and breaker should be locked out.

In the subsequent step, the pump and the motor are aligned to the final tolerances using a reverse dial gage or a laser alignment tool.

This is also the stage during which a soft foot condition of the pump could manifest itself. The hold-down bolts are loosened one at a time and a dial gage is used to record the movement between the machine foot and the base plate or the sole plate. Any movement in excess of 1 mil (0.025 mm) is an indicator of soft foot and should be corrected by adding the required amount of shims under the feet.

When the pump is being aligned with a steam turbine, it is usually carried out at ambient conditions. When steam is introduced, the centerline of the turbine is raised leading to misalignment. To account for this phenomenon, the vertical growth is computed. The easy rule to compute the growth is:

The rise of equipment centerline from base is 1.2 mils for every inch of height from base to centerline for every 100 ºC rise in temperature; or 1.2 mm for every meter height for every 100 ºC rises in temperature.

Thus, if an impulse steam turbine has an exhaust temperature of 130 ºC with an ambient temperature of 30 ºC, the rise in temperature is 100 ºC. If the height from base of foot to shaft centerline is 12 in., then the rise due to thermal growth is 14.4 mils.

After the alignment is completed, the piping associated with the pump and steam turbine should be bolted.

Once this is completed, the alignment should be checked and similar readings should be obtained. If this is not the case, then the piping should be investigated and suitable corrections should be made. If this is left unattended, this can cause stress on the pump casing and nozzles.

After the alignment has been approved, the support pads for the pumps and drivers should be drilled at two locations for providing taper dowels. These dowels should be preferably located at the end, which has the thrust bearings.

## 8.12 Associated piping and fittings

The following are some of the recommended practices in regard to piping associated with the centrifugal pumps (Figure 8.5).

Piping associated with the pump must be anchored and supported independently of the pump. In absence of adequate anchorage, the expansion and contraction of line can cause the transfer of forces to the pump casing. When the pipes are not supported, their weight is borne by the pump casing and nozzles causing them to deflect and crack. The seal life of the pump also gets affected due to this strain.

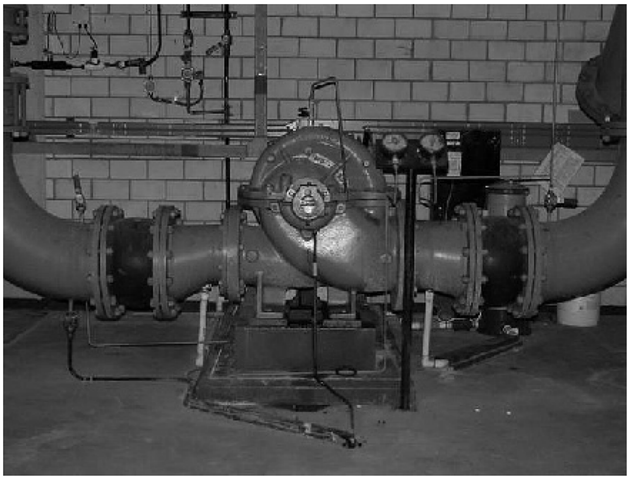


**Figure 8.5**
*Typical piping layout associated with a horizontal centrifugal pump*

It is important that the connections be carefully aligned axially, angularly, and in length. The flange boltholes too have to be in phase with the pump nozzle holes.

One good check to perform is to disconnect all the suction and discharge flanges on the pumps. If levers are required to force the pipe flange on to the pump nozzles (to facilitate bolting of the flanges), one can be certain that the pumps will sooner or later start giving bearing and other problems.

## 8.12.1 Inlet piping (Figure 8.6)

- The piping run and the connection fittings should be properly aligned and supported separately to reduce strain on the pump casing.
- The straight run of the piping leading to pump suction nozzle should be at least 3 to 6 times the diameter of the pipe from the upstream elbow.
- The elbow should be of a standard type or of the long radius type.

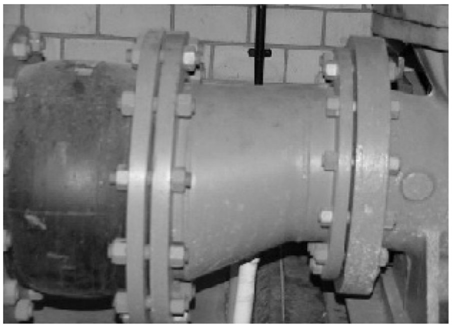


**Figure 8.6**
*Inlet piping*

- If the pump has a negative suction, all suction piping must be airtight.
- Suction pipe size should be at least one commercial size larger than the opening of the pump inlet.
- The reducer joining the straight length of the pipe in the pump line should be an eccentric reducer with the flat side of the reducer as the topside.
- The straight length of the pipe after the eccentric reducer should be 2 times the pipe diameter.
- The suction pipe should be sized to insure a liquid velocity of not more than 2 to 3 m/s.
- All suction pipes in negative suction should have a continuous rise to the pump suction inlet. A 6 mm per 100 mm minimum slope is recommended. This may not be required in a flooded suction.
- In a negative suction, no isolation valves are recommended but can be provided in the flooded suction. Isolation valves even in open condition contribute to pressure losses due to friction and result in lowering of the available NPSH. In pumps with higher negative suction lift, NPSH-a is on the lower side and addition of a valve does not help the cause in any way.
- In a negative suction, the minimum depth of submergence of the strainer should be at least 3 times the pipe diameter, measured from the upper row of holes of the strainer. The distance between the bottom of the strainer and the floor of the sump should be considered as 2 times the pipe diameter.
- In case of a bellmouth or funnel with $D = 2d$, the optimum distance between the rim of the bellmouth and the bottom of the sum should be approximated $0.5d$. If this is larger it leads to the formation of eddies and vortices as shown in Figure 8.7. Swirling vortices can cause the air to be drawn in the suction pipe interfering with the pump performance.
- The minimum submergence should at least $2d$.
- The suction strainer must be at least 4 times the suction pipe area and the mesh size should screen out solid particles that could clog the impeller (Figure 8.8).

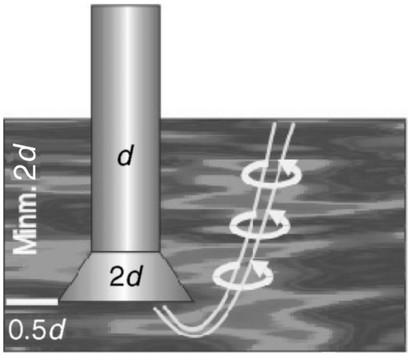



**Figure 8.7**
*Formation of eddies and vortices*

- There should be a provision to drain the contents between the isolation valves in the suction and pump casing.
- There should be a tapping provided for installing a pressure gage in the suction.

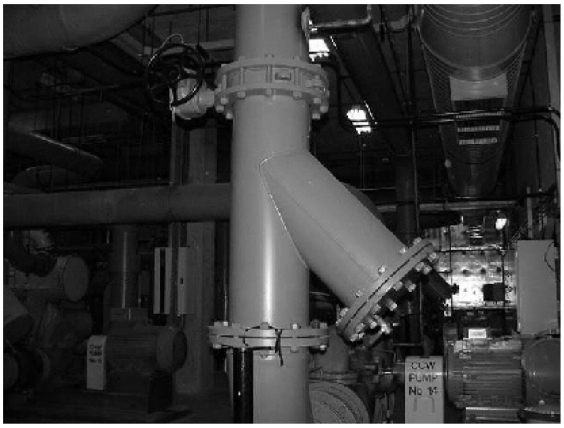


**Figure 8.8**
*A Y-type suction strain in the pump suction*

### 8.12.2 Discharge piping (Figure 8.9)

- The piping run and the connection fittings should be properly aligned and supported separately to reduce strain on the pump casing.
- Discharge pipe size should be at least one commercial size larger than the opening of the pump outlet.
- The number of fittings and size changes should be minimum to prevent fluid friction losses.

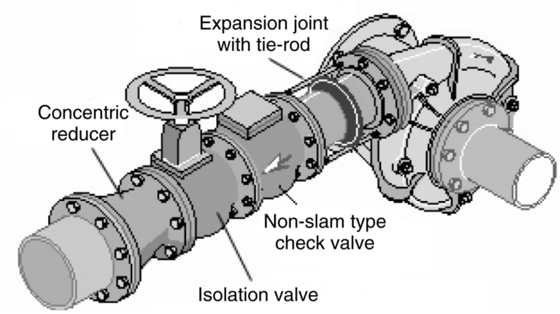



**Figure 8.9**
*Discharge piping (Image source – Berkeley pumps – USA)*

- The check valve used in the discharge should be of the non-slam type to prevent hydraulic shocks.
- The isolation valve is provided downstream of the check valve so that these can be taken up for servicing whenever required.
- Concentric reducers are installed in the discharge pipe to minimize friction losses.
- There should be a pressure tapping as close as possible to the pump outlet and before the isolation valve to measure the pump shut-off head (Figure 8.10).
- Another pressure tapping downstream of the reducer is a good indicator of the pump operating pressure.
- Expansion joints maybe used only after a careful piping analysis, especially when the discharge pressures are on the higher side.

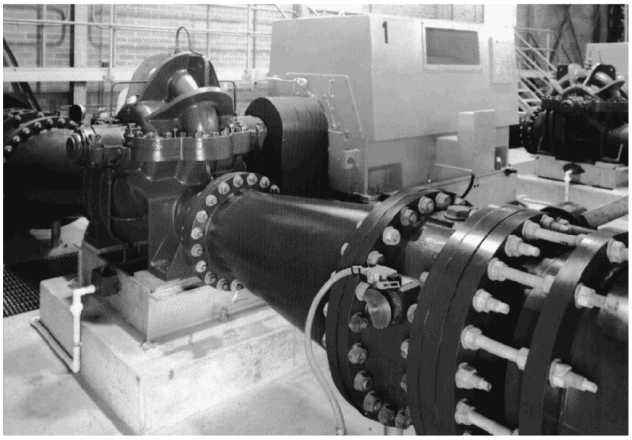



**Figure 8.10**
*Notice the pressure tapping on the pump discharge*

## 8.13 On-site installation and commissioning of the pump set

Once the shop tests have been witnessed and verified by the owner or the owner's appointed representative, the pump is then disconnected and transported to the site for installation and commissioning.

It is important to note that once the pump has been installed, checked, pump and system properly primed, and finally tested and set up, the pump may or may not operate at the design point (flow – $Q$ and head – $H$) as calculated. There are a number of reasons why a difference in $Q$ and $H$ values could arise. These are:

- There could be a difference between the 'as designed' and the 'as constructed' system configurations, i.e. differences in the run of the system pipe work, the number of bends and/or quality of fittings could have altered, etc. This would impact on frictional resistance that the pump must overcome in operation, which could either increase or decrease, thus causing the operating point to differ from the design point.
- If at the design stage too much allowance is made when evaluating frictional resistance – $H_f$, then it is possible that the pump selected could be oversized, i.e. operate off the performance curve or too far to the right of the BEP.
    - If during commissioning of the pump the operating point is measured to be within a margin of ±2.5% of the design point, then the evaluation of the total system resistance $H_{ts}$ is considered to be good.
    - If the operating point is measured to be within a margin of ±5% of the design point, then the evaluation of the total system resistance $H_{ts}$ is considered to be satisfactory.
    - If, however, operating point is measured to be greater than 5% of the design point, then the system would need to be checked out. It is always good practice to have the pump manufacturer's representative involved at the time of installing, testing, and commissioning of the pump set(s) to insure satisfactory operation of the pumps right through their designed life cycle.
- Change in the properties of the fluid being pumped would affect the operating point.

- There could also be other anomalies that could be introduced from the time of the shop test to the time of installation and testing – such as the wrong impeller size, pump speed, or incorrect direction of rotation. All these could adversely affect the operating point.

It is worth noting though, that once the pump performance curve has been established and drawn up at the shop test, the performance curve tend to remain consistent so long as there is no mechanical damage or wear and tear to the pump. So if anomalies arise and the pump has not been dismantled or mechanically altered in any way, the reason for the discrepancy could be with the system layout or system components, which would need to be checked.

Commissioning test logs, similar to those drawn up at the pump shop tests must be filled and held on record as part of the offer and acceptance protocol. Further, based on the operating point at commissioning, the system resistance curve needs to be developed and drawn up for the pump and system. These documents are to be referred to at a later date at the time of verifying the pump and system performance.

## 8.14 Pre-operational checks

Pre-operational checks are mainly focused toward the auxiliary systems:

- For pumps fitted with double mechanical seals (back-to-back) with an external pressurized sealant supply, it is necessary to flush and clean the lines prior to their connection with the pump.
- If the pump casing has been pressurized, it is essential to check if the sealant supply pressure is 1–1.5 $kg/cm^2$ above the pump casing pressure. Pressure lower than this can cause the inner seal to open up, which contaminates the process fluid.
- When the pumps are equipped with tandem mechanical seals, the overhead reservoir that facilitates the thermosyphon circulation for the outboard seal needs to be thoroughly cleaned by oil flush prior to its connection with the pump.
- All cooling water lines connected to the pumps and turbines need to be flushed.
- It is necessary to confirm that the supply and return cooling water lines are connected to their correct headers.
- Bearing housings should be drained of their oil and refilled with fresh oil of correct viscosity.
- If the pump lubrication is with the oil mist system, it should be up and running for almost 12 h before the start-up.
- The oil mist piping should be sloping toward the equipment without any sags or low spots.
- Verify that oil reaches all bearings.
- It is necessary to reconfirm
  - Direction of rotation of the motor
  - Over speed trip setting of a steam turbine
  - Greasing of coupling
  - Earthing of equipment.
- If the pump and turbine are provided with a separate lube oil system then all alarms and trips have to be checked and tested.
- Rotate the pump by hand and look for any rubs.
- Place the coupling guard and tighten the bolts/screws.
- Inform the electrical department to energize.

## 8.15 Preparation for start-up

After the above-mentioned checks have been completed, the stage is set for the start-up.

The sequence from now is:

- Pump suction valve is opened slowly and all the joints are checked for any leakage.
- The pump casing is opened to vent any vapor. This can be tricky in case of flashing hydrocarbons so it has to be done a numerous times.
- If the pumping temperature is high, the pump should be allowed to warm up. For multistage pumps with long rotors, it would be a good idea to keep rotating the rotor 180° after every 30 min.
- The sealant and cooling water lines are opened and circulation of the liquids is insured.
- The opening of the discharge valve is dependent on the type of centrifugal pump. For low specific speed pumps, it is kept closed and opened for higher specific speed pumps. This prevents overloading of the motor drive.
- Once these checks are made, it is time to confirm if the electrical supply has been energized.
- The pump is started!

## 8.16 Pump in operation

- Once the pump has started, check the discharge pressure and insure it is along expected lines. If the pressure does not come up, the most probable reason is that the pump has not been primed properly. The pump should be stopped and re-primed.
- In case of a low specific speed pump, the discharge pressure falls when the discharge valve is opened.
- The flow rate should be confirmed.
- Vibration measurements of the entire train should be taken with a data collector. The overall and filtered readings should be recorded. The frequency plots be recorded and stored. These should be studied for possible defects.
- The overall vibration reading can vary with the point of operation on the pump curve. Therefore, it is recommended to record vibration reading and frequency plots at the 4 or 5 operating points that include the normal and rated points.
- The mechanical seal leakage should be confirmed. It is possible that there could be leakage in the initial stages, which may settle down after wear-in.
- The bearing temperatures of both the pump and motor/turbine should be not more than 10–20° above ambient. A temperature higher than this is an indicator of bearing in distress unless they are of the greased type. In that case, the most probable cause is over greasing. A vibration or a shock pulse analysis can confirm this fact. In such a case, it is better to wait for 24 h and allow the flow out of excess grease.
- Once steady state has been achieved, it is recommended to carry out a performance check of the pump and know its efficiency. It is recommended that this should be plotted on the pump characteristic curves.

When the trial is completed, shut the discharge valve partially and switch off the motor. An eye should be kept on any reverse rotation of the pump. This allows a check of the non-return valve.

# 9

# Centrifugal pump maintenance

## 9.1 Introduction

The basis of centrifugal pump maintenance is a direct function of its criticality to its application. For example, an ordinary garden water pump would not require the same kind of attention as a boiler feed water pump in a major power plant or a firewater pump in a refinery.

The criticality of any pump equipment is based on the following criteria:

- Failure can affect plant safety
- Essential for plant operation and where a shutdown will curtail the process throughput
- No standby or installed spares
- Large horsepower pumps
- High capital cost and expensive to repair or longer repair lead time
- Perennial 'bad actors' or pumps that wreck on the slightest provocation of an off-duty operation
- Finally, pump trains, where better operation could save energy or improve yields are also likely candidates.

Once the criticality of a pump can be ascertained based on the factors mentioned, the pumps can be classified as:

- Critical
- Essential
- General purpose.

After this categorization, the type of maintenance philosophy can be assigned. The pumps, which fall in the category of critical machines, are usually maintained with the predictive and proactive techniques.

The essential category pumps are assigned with preventive maintenance whereas maintenance for the general-purpose pumps maybe less stringent.

In actual operations, a mix and match of techniques is applied with a prime intention of maximizing runtime lengths and reducing downtime and costs.

The present day focus on continuous process plant pumps is to adopt a mix of Predictive and Preventative Maintenance (PPM).

There are four areas that should be incorporated in a PPM program. Individually, each one will provide information that gives an indication of the condition of the pump; collectively, they will provide a complete picture as to the actual condition of the pump.

These include:

- Performance monitoring
- Vibration monitoring
- Oil and particle analysis
- System analysis.

### 9.1.1 Performance monitoring

The following six parameters should be monitored to understand how a pump is performing:

1. Suction pressure ($P_s$)
2. Discharge pressure ($P_d$)
3. Flow ($Q$)
4. Pump speed ($N$)
5. Pump efficiency ($\eta$)
6. Power.

When the driver is an electrical motor, the power is easiest to measure using portable instruments. These can be clipped on the electrical phase cables and used to measure the ampere, voltage, power factor, and eventually the power.

In conjunction with the process parameters such as suction and discharge pressures, flow rate and fluid characteristics, the efficiency of the pump can be computed and benchmarked against expected values provided by the OEM.

The suction and discharge pressures are measured using calibrated pressure gages screwed on to the provided fittings on pipelines with isolation valves.

The flow measurement is usually done with obtrusive instruments such as venturi tubes, orifice plates, and pitot tubes. When such devices are not installed, the non-obtrusive types like the doppler or transit time devices can be used. These can be portable but caution must be exercised because each device must be calibrated, and independent testing has shown these devices are sensitive to the liquid being pumped and are not precisely accurate.

A detailed method to compute the pump performance is given in Section 3.14.

Analysis of efficiency or inefficiency can help one to determine whether the losses are on account of:

- Hydraulic losses
- Internal recirculation
- Mechanical losses.

Direct reading thermodynamic pump efficiency monitors (such as the Yates meter) are now available and capable of interpreting the pump's operating efficiency in a dynamic manner by measuring and computing the rise in temperature (albeit in mK) of the fluid as it moves from the suction to the discharge side of the pump.

Highly sensitive and accurate temperature probes and pressure transducers are positioned on the suction and discharge pipe work to measure temperature and pressure, whilst a power meter provides information on the power absorbed by the electric motor.

Motor efficiency obtained from the manufacturer of the electric motor and drive losses are factored in to the software program, to then calculate the operating efficiency of the pump. This reading could be compared with the pump's commissioning data and drop in performance or efficiency could be determined.

The advantages of this set up is that:

- No dismantling of the pump is necessary.
- Offers cost savings and energy savings by increasing the pump availability and reliability factors.
- The time to maintain the pump set maybe predicted and planned more accurately and in a qualified manner in line with predictive and planned maintenance strategies.
- If a flow meter is installed to measure process liquid flow, then the pump monitor is able to verify the accuracy of the meter readings by calculating '*Q*' from the empirical formula for power '*P*'.

Though the capital cost for such meters maybe high, the return on investment may soon be realized for large capacity pumping systems involving numerous pumps. Alternatively, the service could be purchased at an affordable cost, or services provides could be contracted to periodically measure and report on the performance of the pumps as part of the PPM program.

### 9.1.2 Vibration monitoring

Vibration monitoring on pumps can be online or offline based on the criticality of its pump. Online vibration measurements would use proximity probes to measure shaft vibrations. The most common type of vibration monitoring is offline using portable data collects and measuring casing vibrations.

The vibrations are measured for both the pump and its driver. Vibration measurements are collected at bearing locations of these machines. While reporting, it is necessary to follow a convention for labeling the various bearings of a machine train from where measurements are made. The general convention followed is to start labeling from where the power comes in.

So, for a simple machine train consisting of a motor and pump will be labeled in the following way (Figure 9.1):

- Motor non-drive end bearing – A
- Motor drive end bearing – B
- Pump outboard bearing (next to the coupling) – C
- Pump inboard bearing (away from coupling) – D.

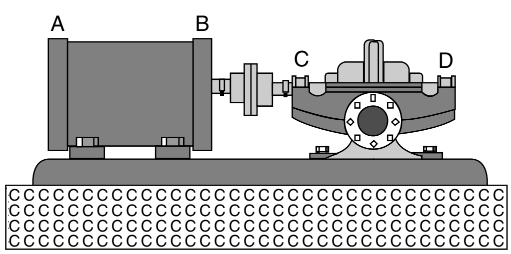



**Figure 9.1**
*Bearing naming nomenclature*

Once having labeled the bearings, it is important that vibrations are taken in three Cartesian directions. In vibration nomenclature, these are vertical, horizontal and axial directions. This is because of the construction of machines; their defects can show up in any of the three planes and hence should be measured.

Vibration reporting is generally done in the manner shown below.

| **Velocity (mm/s-pk) 20/8/02** | **Vertical** | **Horizontal** | **Axial** |
|---|---|---|---|
| A | 2.4 | 1.7 | 1.0 |
| B | 2.1 | 1.9 | 1.2 |
| C | 4.3 | 5.6 | 2.7 |
| D | 3.7 | 4.1 | 2.1 |

Hand held measurements are subject to a number of errors. It is thus important that personnel carrying out this task is aware of the possible errors that can occur while taking measurements.

The errors can occur due to:

- Position on machinery
- Probe angle
- Probe type
- Pressure.

As mentioned in Section 8.15, a base reading of vibrations is taken at various operating points. The vibration levels are minimum at BEP and rise when the operating point is away from the BEP. A comparison with the original base readings can indicate the deterioration in the mechanical health of the pump.

In absence of the base reading, the limits provided in API 610 can be used as a reference.

Frequency analysis of vibrations helps to determine the nature of the pump fault.

The three most common types of problems associated with pumps are:

1. Unbalance
2. Misalignment
3. Bearing defects.

These can be easily detected using vibration analysis. Some of the infrequently occurring problems of the hydraulic nature like cavitation, improper impeller to volute tongue gaps, can be detected using vibration analysis.

Infrared thermography measurements may also be utilized in conjunction with vibration analysis to pinpoint the source of a problem due to the pick up of an elevated temperature at the trouble spot.

### 9.1.3 Oil and particle monitoring

The testing of the oil and possible particles in it can be a good indicator of the condition of the pump bearings and the oil program of the company.

There are several tests that can be performed on the lubricant to determine the condition of the bearing or determine why a bearing failed so appropriate corrective action can be taken.

These tests include:

- Spectrographic analysis
- Viscosity analysis
- Infrared analysis

- Total acid number
- Wear particle analysis and wear particle count
- Moisture content.

Most of these tests have to be performed under laboratory conditions. Portable instruments are now available that enable the user to perform the test on site.

### 9.1.4 System analysis

A system analysis is a method of determining the system resistance curve of the pump as depicted in Section 3.6. This can be further used to evaluate the NPSH-a for the pump as in the installed condition.

A typical system analysis will include the following information, NPSH-a, NPSH-r, static head, friction loss through the system, and a complete review of the piping configuration and valves and fittings.

System analysis is often overlooked because it is assumed the system was constructed and operation of the pumps is in accordance with the design specifications.

This is often not the case. As has been stated previously, it is imperative to know where the pumps are being operated to perform a correct analysis and this is dependent on the system.

In addition, the process must be understood because it ultimately dictates how the pumps are being operated. All indicators may show the pump in distress when the real problem could be that it is run at low or high flows due to excessive hydraulic forces inside the pump.

By now, we see that pump maintenance is not all about dismantling and assembling pumps. Pump maintenance is an activity to prolong the efficient runtimes of the equipment to improve its overall effectiveness.

The techniques mentioned above are used to monitor the health of the equipment and on indication of pump distress, the cause and severity of this distress is also indicated.

If a pump is observed to be in distress, it should be taken up for corrective actions immediately for to prolong maintenance can lead from primary failures to secondary failures. This increases the downtime and the maintenance costs.

In case of non-essential pumps, it is possible that a pump is left unmonitored or unanalyzed until failure is encountered.

It is important to define the term ‘failure’ now. The word ‘failure’ does not necessarily imply a pump seizure or a heavy seal blowout forcing a shutdown of the pump.

The pump is installed to perform a function under specified conditions. When it is no longer capable of performing a function under the specified conditions it is defined as a failure.

For example, if the pump has to deliver a certain flow rate, consuming a certain horsepower and if this horsepower consumption exceeds a specified limit, the pump is stated to have failed even though the operation of pump may appear to be very smooth.

## 9.2 Pump breakdown and removal

In real applications, breakdown of pumps is a common event. The typical failure causes are:

- Mechanical seal failure
- Excessive vibrations
- Pump rubbing or seizure
- Inadequate performance (flow rate, head developed, power consumption)
- Leaking casing.

Before a pump is removed for a repair or an overhaul, it is essential to know the failure type before its dismantling in the workshop.

Thus prior to the removal of the pump the following exercise must be conducted:

- Check with the operating cause on the perceived cause of failure
- If it is safe to operate the pump, rerun it to diagnose failure by
  - Sight, smell, noise, and touch
  - Perform a vibration analysis
  - Measure bearing temperatures
  - Carry out a performance check.

The above steps will positively indicate the probable cause of failure and ascertain that it is specific to the pump and not to the process or pumping system, which includes the suction and discharge system associated with the pump.

If the above analysis indicates the problem is due to the pump, the following sequence of field checks are recommended:

- Check flush lines and quench lines for leak, corrosion, or plugging
- Check the balance line if included in the pump design
- Check the suction strainer for blockage and insure all valves are open
- Visually check the oil condition and the oil level
- Check coupling for wear or lack of grease
- Check the condition of the gages
- Observe the condition of the base plate and pump supports
- Isolate and danger tag the pump valves and motor (pad lock motor switch)
- While dismantling notice the pipe strain on the pump
- Once the pump is decoupled, measure the radial and end clearance of both the pump and the motor
- Run the motor and confirm satisfactory operation. If unsatisfactory, the motor should also be removed for repairs.

## 9.3 Single-stage pump dismantling and repair

Once the pump is taken out for an overhaul, the observation process does not end but becomes even more focused.

A recommended procedure is to match and mark all parts prior to the dismantling of the pump and make the following checks.

- Open the pump casing bolts and separate the pump casing or the volute.
- Inspect the pump casing for corrosion and erosion after it is removed. The gasket face should be particularly inspected to insure that there is sufficient land area and the surface is smooth. In case damage is observed, these should be repaired in appropriate manner.
- Similarly, inspect the impeller and the nut for wear, corrosion, and erosion. Impeller vanes should be inspected for pitting or cavities in the areas mentioned in Section 3.12.2 to look for signs of cavitation, suction, or discharge recirculation.
- Inspect the impeller and casing wearing rings for any signs of rub and/or detachment. This can be an indication of excessive shaft deflection, improper assembly, or piping strain.
- Open the seal flange nuts and check for the seal tension.

- Remove the impeller from the shaft after opening the nut. In case it has balancing holes, these should be unplugged. Wearing rings on the backside should be inspected carefully.
- The pump shaft sleeve is then pulled out along with the rotary head of the seal. All these should be carefully dismantled and checked for weak signs on the primary seal face, as well as, the secondary seals like the O-rings and gaskets if any.
- The bearings are checked for roughness. The endplay is measured as shown in Figure 9.2. Shaft is also checked for any signs of corrosion of erosion.

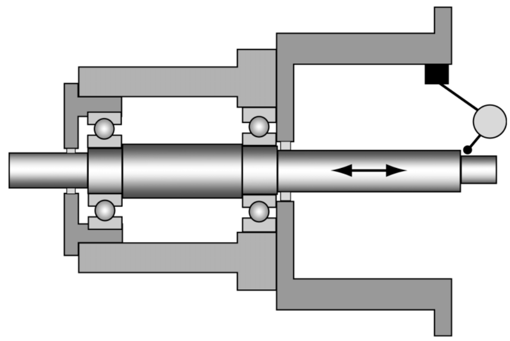


**Figure 9.2**
*Method to check endplay or axial float of a pump shaft*

The endplay or axial float of a pump shaft is measured by placing the pointer of a dial gage against any step on the shaft. The magnetic base maybe fixed to seal housing or any other convenient location.

The shaft end is then lightly pumped over to one end and the dial reading is set to zero. The shaft is then moved over to the other end and a reading is recorded again. The endplay should be in the region of 0.001–0.003 in. (0.02–0.07 mm).

An axial float greater than this leads to pitting and fretting in the points of contact in the shaft packing or mechanical seal areas.

On finding a higher float, one should ascertain whether this is due to an improper assembly of bearing or bearings in the housing or due to defective bearings.

- Shaft runout is a measure of shaft roundness combined with a permanent bow or bend. The effect of shaft runout is greatest on a mechanical seal due to the orbital motion of the shaft. A shaft with a 3 mils runout (0.075 mm) will move the seal faces 3 mil per side or a total of 6 mils. In addition, a bent shaft tends to cause vibrations that have a large impact on the life of mechanical seals and bearings. This check can be carried out after the shaft is bare and the bearings have been removed. The shaft is placed on V-blocks that are resting on a flat machinist's table.

Dial gage pointer is placed at different locations and readings are taken while the shaft is rotated by hand.

Alternately, the shaft can be clamped at lathe centers and reading. The bearings journals are set at zero and runout at other locations is measured by keeping the dial gage on the tool post.

The next method, as shown in Figure 9.3, is recommended only when the pump bearings are in good condition.

In this method, the dial gage is fixed on any part of the housing. The pointer is placed on the OD of the shaft at various locations. The shaft is rotated by hand.

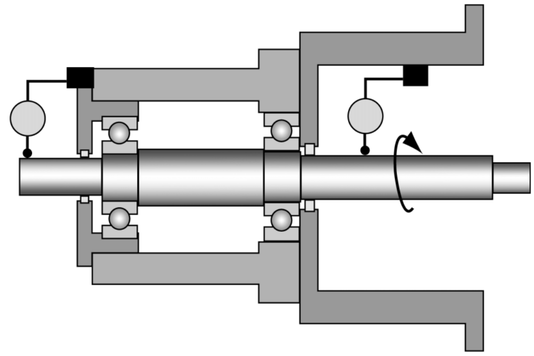


**Figure 9.3**
*Method to check bent and runout of a pump shaft*

The runout should not be greater than 0.05 mm. When a pump sleeve is mounted on the shaft, then the runout should be measured on the sleeve OD. The combined runouts of the shaft and the sleeve should be 0.05 mm. It is recommended to contain the maximum runout of the shaft and the sleeve to within 0.025 mm. Thus even in a worst case, the runout will not be more than 0.05 mm. In case the reading is higher, the shaft should be straightened.

- Shaft radial movement check is next on the list and this can cause shaft whip, deflection, and ultimately, vibrations. This usually occurs when the bearing fits are loose due to corrosion, wear, or improper machining. Even defective bearings can cause this to happen.

  The method to check this condition is shown in Figure 9.4. A dial gage is fixed to the housing as close as possible to the inboard bearing housing and the pointer is placed on the shaft outside diameter (OD). The shaft is then lightly lifted and pressed downward and the dial reading is recorded.

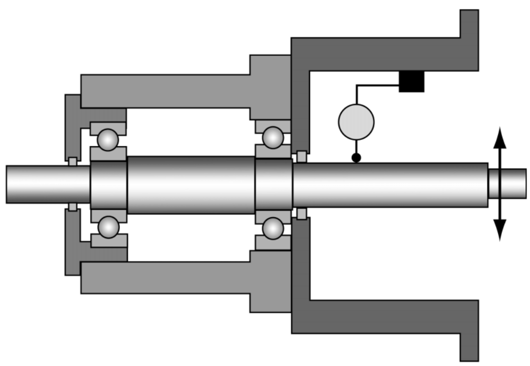


**Figure 9.4**
*Method for checking for shaft whip or deflection*

  Any reading above the value of 0.07 mm is unacceptable and calls for corrective actions.

- Seal housing squareness is an important check to insure that the seal faces are parallel to each other when they are installed. A seal housing that does not sit square will cause an angularity between the seal faces and cause an unequal wear of the seal faces leading to shortened seal life (Figure 9.5).

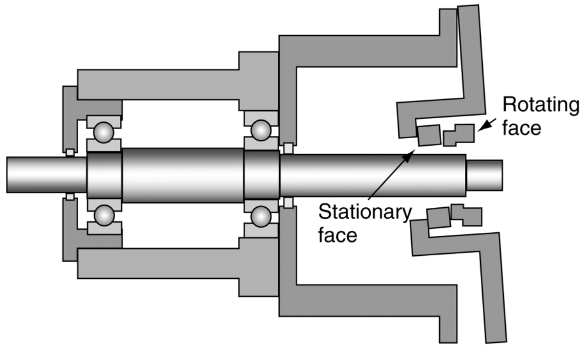



**Figure 9.5**
*Notice the effect of angularity of seal housing on seal faces in this exaggerated view*

This check is slightly different from the checks mentioned above. In this case, the magnetic base of the dial gage is placed on the shaft and the pointer is placed on the face of the seal housing as shown in Figure 9.6.

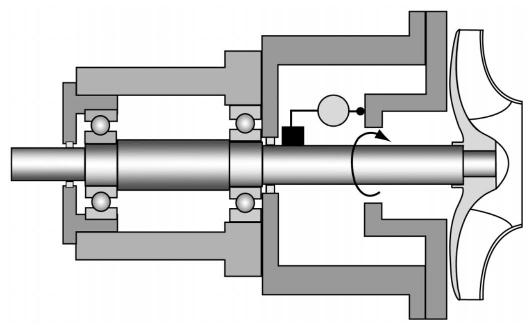


**Figure 9.6**
*Method for checking for stuffing box squareness*

The shaft is then rotated and facial reading is recorded. The allowable value is 0.05 mm.

In case a higher reading is obtained, the seal housing should be put on a lathe and face should be squared with the mating face of the seal housing with the bearing housing.

The seal housing face should be free of any nicks, burrs, or any other surface defect.

- One type of seal assembly problem occurs due to the angularity of the seal housing as mentioned above. The next problem is the offset of the seal faces. This occurs when the seal housing is not concentric with the shaft axis (Figure 9.7).

The way to record the seal housing concentricity is shown in Figure 9.8:

A dial gage is fixed to the OD of the shaft and the pointer sweeps the inside bore of the seal housing. The inside of the bore may have a rough surface because of corrosion or wear. In such a case, it should be cleaned with a sandpaper, washed, and dried with a solvent.

The allowable limit for concentricity is 0.05 mm. In case the reading is in excess of the allowable limit it should positioned and doweled until concentricity is achieved.

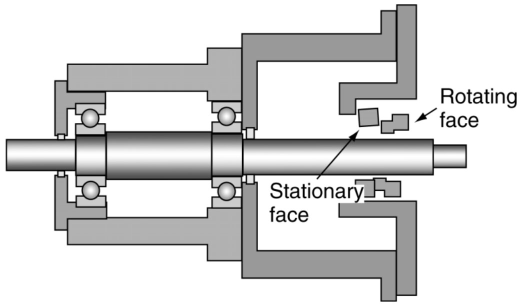



**Figure 9.7**
*Notice the effect of non-concentricity of seal housing on seal faces – exaggerated view*

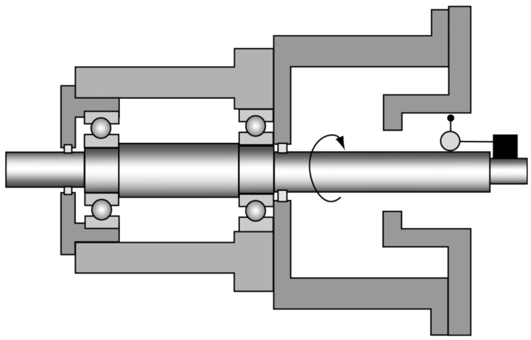


**Figure 9.8**
*Method for checking for stuffing box concentricity*

In in-between bearing pumps which have two seal housing located at the two ends, this check is even more critical.

- In case the bearings are found to feel rough and/or there is excessive radial and axial float, the bearing housing should be dismantled to replace the bearings. Usually, the outer race of the bearing has a transition fit with the housing and the inner races of the bearings have an interference fit on the shaft OD.

Thus, the pump shaft with the bearings can be drawn out of the bearing housing with slight tap of the hammer.

The bearings can be removed from the shaft usually by using a hydraulic press. Once the bearings have been removed, the dimensions of the bore and the shaft OD need to taken and the bearing fits should be worked out.

The shaft at this stage can be checked for straightness as mentioned earlier. After this is checked, it is polished and kept ready for reassembly.

If the bearing fits are found to be excessive, suitable repair procedures have to be worked out depending on the material of construction and size of the shaft.

The housings can be re-sleeved with rings of the same material of construction. The shaft diameter too can be subjected to weld build-up and re-machining. However, this has to be done considering the material of construction and thickness of the section.

Some materials may need a heat treatment and a smaller diameter shaft could warp if the weld procedure results in excessive heat build up.

# 9.4 Preparation for reassembly

## 9.4.1 Impellers

The pump impeller is a rotating part, subjected to high velocity; and the adverse effects of poor hydraulic design can be a cause of wear and erosion. As mentioned earlier the vanes need to be examined for cavitation and recirculation phenomenon. This information needs to be communicated to the operations.

Any corrosion of impeller should be viewed seriously and a review of the material of construction should be made if the corrosion rates are excessive.

A badly worn/corroded impeller should be replaced.

When the impeller is to be reused, the fit of its bore with the shaft should be checked and that of its keyway.

The diameter of the wearing rings should be measured and compared with its pump casing, its mating ring. The clearances should be in line with the API 610 (7th Edition) recommendations. These are given in Section 2.3.

This clearance is probably the one factor that has a major impact on the efficiency of the pump. A higher clearance leads to higher recirculation losses.

Figure 9.9 is a graph for double suction pumps showing the relationship between the percentage power loss and the specific speed. (Adapted from – When To Maintain Centrifugal Pumps – I.J.Karassik, HP-April 1993.)

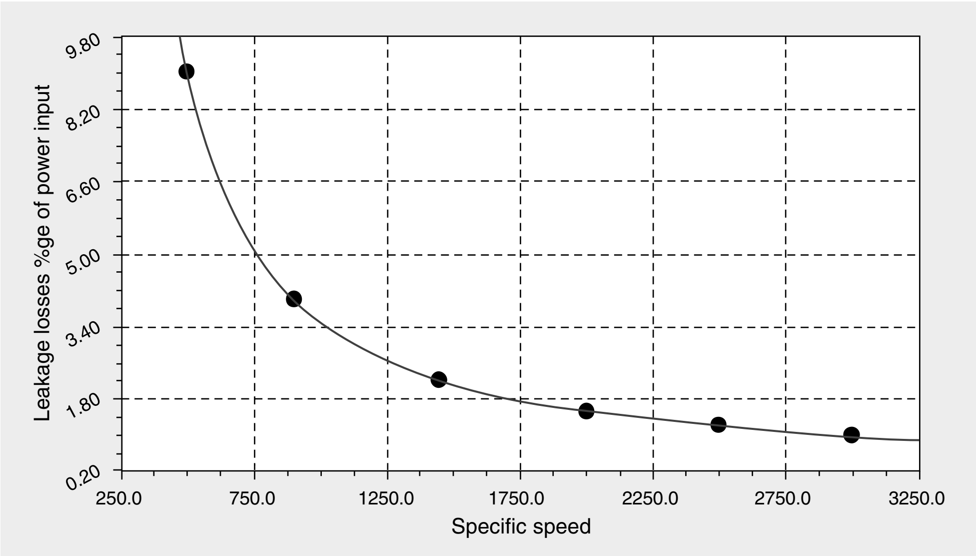



**Figure 9.9**
*Results from an experiment of wearing ring losses in double suction pumps*

From the graph, it can be seen that the percentage power losses due to wearing clearances are much higher in lower specific speed pumps.

Restoring back the clearances for such pumps gives higher returns in terms of leakage loss reduction. Thus, such pumps need more attention with respect to wearing ring clearances.

The wearing rings are press fitted to the impeller, casing, and held in place by the following ways:

- Tack welding and then grinding off the excess bead
- Threaded wearing rings (against the direction of rotation)

- Drill, tap, and fasten with grub screws in the radial direction. The diameter of the hole is not more than one-third the width of the ring
- Drill and tap at the ring and impeller/casing interface and fasten with grub screws in the axial direction.

API 610 generally does not recommend the first two methods.

After the wearing rings have been renewed and machined to obtain the requisite clearance, it is recommended to balance the impellers with allowable limits as specified by the API. The ISO 1940 standard that recommends a grade of G 6.3 for centrifugal pumps is far too liberal.

However, in case of pumps with long shafts and big impellers and heavier coupling halves, it is recommend to carry out the balancing after assembling the complete rotor. Two-plane rotor balancing is recommended over single-plane balancing of the pump impeller.

## 9.4.2 Pump casing

All fits of pump casing and mating parts should be measured and recorded. All plugs and fittings must be removed, cleaned, inspected, and fitted back. The mounting pads should be flat and parallel to the pump centerline.

The casings should be cleaned. The gasket area should be polished with emery paper to ensure that no pieces of the old gaskets adhere to the surface.

In case there are signs of corrosion or wear, the possibility of coating the internals with resistant epoxy or its equivalent should be considered. A repair work is shown in Figure 9.10 for the internals of casing affected corrosion and pitting in salt-water application.

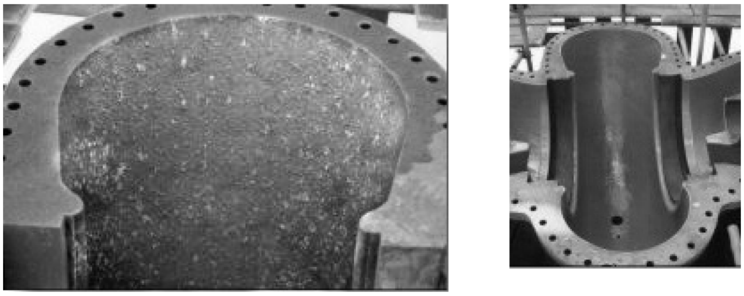


**Figure 9.10**
*Surface of a pump casing before and after a finished coat*

The procedure involves descaling of the internals by ash, sand, or grit blasting. In the case of salt-water applications, steam cleaning is applied to insure all salt contamination is removed. This is again followed by grit blasting.

The final coating is done using epoxy compounds to provide the finish as shown in Figure 9.10.

Such coatings smoothen the surface roughness of the liquid path and thus reduce the hydraulic losses within the pump. This has positive impact on the efficiency of the pump; in some cases, this can be as high as 10–15%!

The casing bolts are cleaned and the right number are arranged and coated with anti-seize compounds.

### 9.4.3 Seal and bearing housing

The cleanliness of the seal and bearing housings is of utmost importance. The cleanliness required in the housing has a large impact on the life of the bearings and seals.

If the seal housing has cooling water jackets, it is recommended to get them chemically cleaned periodically.

The bearing housing mating faces with the seal housing should be inspected and checked to insure a proper fit.

The faces of the bearing housing covers are checked for smooth surface finish. The oil seals or any other shaft seal is replaced.

### 9.4.4 Shaft

The shaft is checked for straightness, condition of threads at the impeller nut location, keyway fit at impeller and coupling locations and fits for the anti-friction bearings.

In case the shoulders or steps are rounded, these are dressed up/repaired.

The location of lip seal area is inspected. In case these are damaged it has to be repaired or compensated by a sleeve of appropriate size.

### 9.4.5 Mechanical seals

Mechanical seals are the prime causes of pump failure and many efforts to improve the reliability have led to a building of standard for seals called as API 682. This standard recommends the use of cartridge seals. Cartridge seals eliminate the skilled workmanship required to assemble a seal at the users end.

### 9.4.6 Pump shaft sleeve

The preparation for seal assembly begins with the check of the pump sleeve condition. The sleeve area under secondary seals such as O-rings/PTFE wedges are mostly prone to fretting as shown in Figure 9.11.

To eliminate this problem, this area of the pump sleeve can be plasma sprayed to provide a hard ceramic coating like chromium oxide, tungsten carbide, or aluminum oxide. The hardness of nearly 70 Rc is obtained. After spraying, this area is ground to the exact dimension.


**Figure 9.11**
*Pump shaft sleeve*

### 9.4.7 Seal faces

The seal faces should be renewed. It is advisable to check the faces for the flatness using an interferometer and an optical flat. Here a monochromatic light source (helium or sodium) is used and an optical flat is placed between the seal face and the light source. The patters observed on the optical flat indicate the flatness of the seal faces (Figure 9.12).

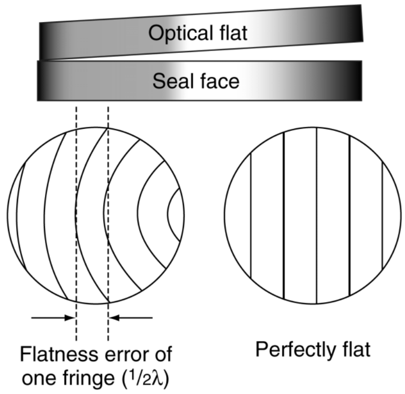



**Figure 9.12**
*Principle of confirming flatness of seal faces*

The optical flat is placed on the piece to be measured. The monochromatic light is aimed at the piece and this light reflects off the piece back through the optical flat causing interference light bands (Figure 9.13).

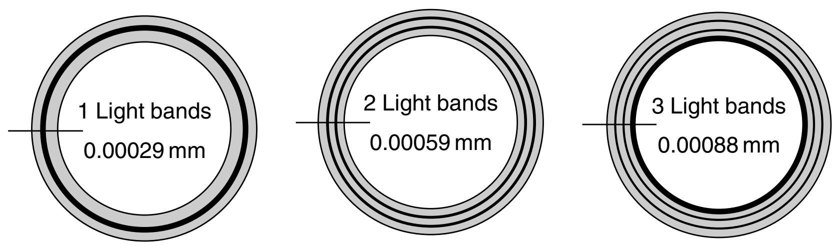



**Figure 9.13**
*Reading the flatness of the seal faces under the monochromatic light source*

If the distance between the optical flat and the piece we measure is one-half the wavelength of helium, or an even multiple of the number, the band will show black. This is referred to as a helium light band and because it is one-half the wavelength of helium it measures 0.3 microns or 0.0000116 in.

The concentric rings indicate that the surface is not concave or convex. When these are present, the concentric rings give way to arcs. However, the number arcs or rings cut by a straight line across the face is the number of light bands.

The allowable light bands for a helium source check is 0.9 microns or 3 light bands. This degree of flatness will allow a mechanical seal to seal vacuum down to a measurement of one Torr (one millimeter of mercury). With this limit, the mechanical seal can easily pass fugitive emission specifications of less than one hundred parts per million.

The carbon graphite faces are known to relax after lapping. Although lapped to less than one light band by the seal manufacturer, we may see the readings as high as three light bands during the checks. These faces should return to flat once they are placed against a hard face that is flat.

It is preferred that the carbon/graphite seal faces should not be relapped because the relapping procedure drives the trapped solids further into these faces. Lapping powder or paste should not be used to lap carbon/graphite faces. They should be lapped dry on ceramic stones of varying grit or finish.

In addition to seal faces, the rotary-head that houses one of the seal faces comprises of a stainless steel retainer, which may house a single or multiple springs. The retainer may have anti-rotation provision.

All these should be checked to insure that they are capable of performing their function. It is advisable to replace the springs.

The grub screws used to fix the retainer on the pump sleeve should be unscrewed, cleaned, and reused.

When bellow seals are used, these should be inspected carefully to look for any cracks at the weld joints.

### 9.4.8 Secondary seals

Once the faces have been checked, they are wrapped and stored carefully in an air-conditioned and dust-free room until it is time for reassembly.

The secondary seals include all O-rings or gaskets to be used with the stationary seat, rotating head, and the pump sleeve.

During inspection if any of the O-rings were found to have swelled, shrunk, corroded, nicked, cut, extruded, or blistered the cause should be investigated.

For no reason the secondary seals should be reused. The savings will not justify even a pump leaking on the test bench.

Therefore, a new set with confirmed dimensions should be procured and kept along with the seal faces.

### 9.4.9 Seal plate

The seal plate that houses the stationary seat should be free of corrosion. Sometimes these are fitted with a stop pin to prevent the rotation of the seat. The condition of the pin should be checked and if necessary, it should be replaced.

The gasket/O-ring surface should be cleaned and polished.

### 9.4.10 Seal housing

The seal housing needs to be clean and dry. The internals maybe coated with an anti-rust compound.

The throat bush is fitted to the seal housing. Its clearance with the pump sleeve OD should be confirmed. If it is higher, it should be replaced.

In case it has integral cooling water jackets, these should flushed and periodically chemically cleaned.

### 9.4.11 Bearings and bearing housing

New bearings should left packed until they are required for assembly. The housing fits with the bearing should be confirmed when there are repeated bearing failures or when the pump has high bearing temperatures.

Another important check is the concentricity of the bearing housing bores. To carry out this check the bearing housing is placed on a horizontal boring machine. The dial gage is fixed to the boring bar and the readings are measured in the vertical and the horizontal plane.

Alternately, dummy shaft having zero clearance with the housing bore can be used to confirm non-concentricity of the bores.

Bearing housing is typically made of cast iron and in areas where humidity is high and heavy rainfall is common, the inner surface is prone to corrosion. The corrosion dust or particles can be washed by the splashing oil and act as abrasives for the bearings.

Thus, it is necessary to paint the internals of the housing with an oil resistant paint. Prior to assemble, the paint condition should be inspected.

The bearing housings have to be sealed to prevent moisture ingress. The bearing covers that are bolted to the bearing hosing should be installed with proper bearing isolators.

The oil level gage opening should be cleaned and new one should be fitted to it.

The bearing housing vent should also be removed, cleaned and screwed back.

### 9.4.12 Coupling

The coupling should be inspected for wear and its fit with the shaft. If these are normal, they should be greased if applicable.

### 9.4.13 Pump clearance/overhaul chart

Preparation for assembly is a detailed exercise and lays the basis for a quick assembly and reliable repair. The preparations require acceptable clearance and accurate data to insure that assembly will not be hampered because of incorrect dimensions and unclear expectations of any sub-assembly procedure.

A standard clearance chart is given at the end of this topic.

## 9.5 Pump assembly

The pump assembly described below (Figure 9.14) is for a single-stage overhung impeller centrifugal pump.

The reassembly process begins at the pump shaft, which has undergone checks for runouts, condition of steps/shoulders, keyways, and fits at the journals.

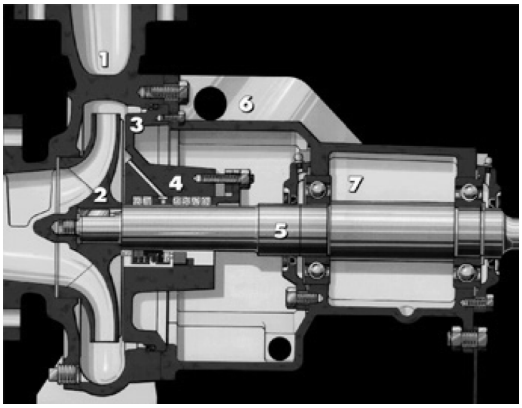



**Figure 9.14**
*Cross-section of a single-stage overhung centrifugal pump*

### 9.5.1 Bearing assembly

- Apply oil on the bearing seat of the shaft lightly.
- Shielding, if any, must face in proper direction. Angular contact bearings, on pumps where they are used, must also face in the proper direction. Duplex bearings must be mounted with the proper faces together. Mounting arrangements vary from model to model (refer to the OFM pump manual).

- The bearing should be pressed on squarely. Do not cock it on to the shaft. Be sure that the sleeve used to press the bearing on is clean, cut square, and contacts the inner race only.
- The bearing should be pressed firmly against shaft shoulder. The shoulder helps to support and square the bearing.
- In case the fits are tight, the bearings maybe heated using an induction heater with a de-magnetizing cycle. The temperature should not increase beyond 110 ºC.
- The snap rings are then properly installed with the flat side against the bearing, and that lock nuts are tightened.
- The bearings should be lubricated.
- This assembly should then be wrapped in a plastic cover while the bearing housing is made ready.
- Prior to placing the rotor in the bearing housing, it is insured that it is spotlessly clean.
- In pumps, the inboard bearing (closer to the impeller) is near the locating bearing (fixed bearing). The bearing cover that provides the step to the outer race of the bearing should be fixed. This is done after a sealing compound has been applied to the mating faces.
- Oil is smeared on the bearing housing bores.
- Shaft with the mounted bearings is tapped in the bearing housing till the outer race of the inboard bearing rests against the step.
- Apply sealing compound on the outboard mating face for the bearing cover and tighten the bolts.
- Another type of bearing assembly is depicted in Figure 9.15. The outboard bearing is the locating bearing and is placed in a housing that can be screw jacked axially after loosen the clamping bolts. The advantage of this arrangement is the rotor can be moved axially without dismantling the pump. This is particularly useful in setting the clearance between the back vane impeller and the casing wall.

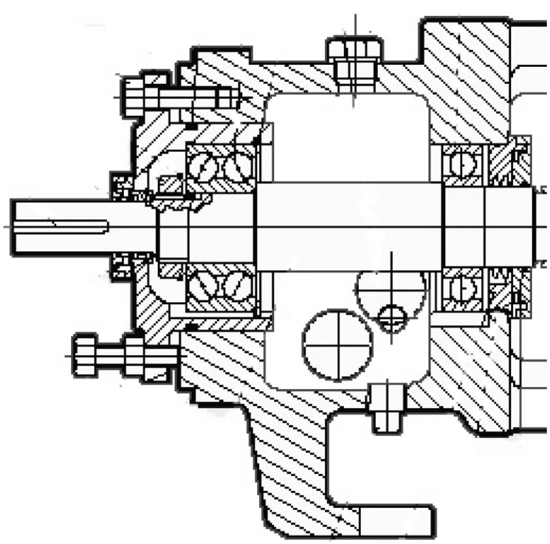


**Figure 9.15**
*A bearing assembly – outboard bearing is the locating bearing*

- In this assembly, the bearings are fitted on the shaft and the lock nut is tightened against the outboard bearing. The housing is then slid over the outer race and snap ring is slipped in the slot.
- This rotor assembly is then placed in the bearing housing and the bolts are tightened.

### 9.5.2 Seal assembly

- The stationary seat is firmly clamped with the seals in the seal end plate.
- This is then bolted to the seal housing.
- The seal housing is then carefully slid over the shaft so that there is no impact between the shaft surface and the stationary seat.
- It is then bolted to the bearing housing taking care of match marks made prior to dismantling.
- The above is necessary for if even a single bolthole is out of phase it may not be possible to make the sealant connections in the field.
- Once this is complete, the next step is to adjust the seal tension.
- The springs of a seal have a certain effect on the face pressure and a certain seal tension has to be maintained. In a seal design comprising of multisprings, the seal tension is generally of the order of 3–4 mm depending on the size of the seal. Single spring seals have a much higher seal tension.
- To arrive at the correct seal tension, the following measurements need to be taken. The figures below help to explain this procedure.

  The rotary head of the seal is assembled and total length is measured. This is the free length from the edge of the sleeve to the seal face without compression. Let us call this length as *F* (Figure 9.16).

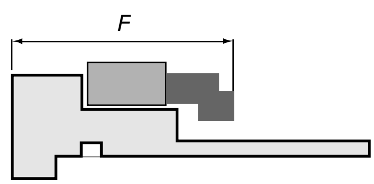



**Figure 9.16**
*Sleeve installation*

Before we install the sleeve with the rotary head on the shaft, we need to measure the distance between the shaft step and the face of the stationary seat. This can be measured directly or by using a frame of reference that is based on the construction of the pumps. This is shown in Figure 9.17. Let us call this distance as *A*.

The next step is to know the length *S* of the sleeve. This is a differential between easily measurable lengths '*L*' and '*M*' (Figure 9.18).

Once *F*, *S*, and *A* are known, then $F - (S + A) = 3$ mm. This seal tension will be achieved once the seal is assembled.

- Some sleeve designs have an O-ring to prevent leakage through the gap between shaft OD and the sleeve ID while some designs may require a gasket as shown in Figure 9.19.

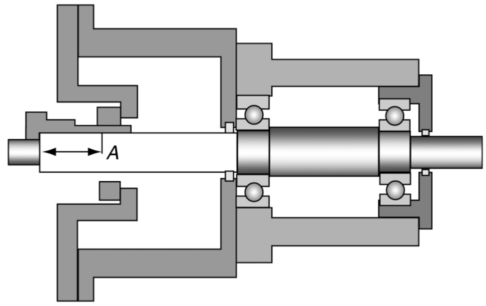



**Figure 9.17**
*Sleeve installation step 1*

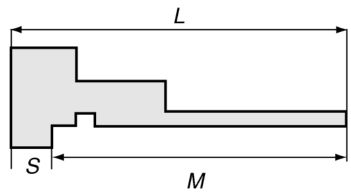



**Figure 9.18**
*Sleeve installation step 2*

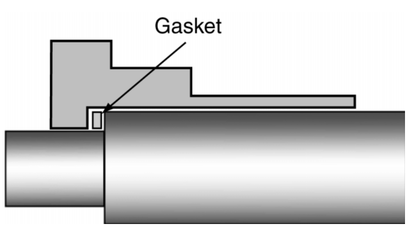



**Figure 9.19**
*Typical sleeve design*

In this case, the thickness of the gasket has to be taken in account for computing the seal tension. If $G$ is the gasket thickness, the new formula is

$$\text{Seal Tension} = F - (S + A + G)$$

In case the seal tension is improper, cuts on the sleeve steps can be taken.

*Caution*: Care has to be taken that the step of balance carbon and sleeve step does not foul up once the seal is under compression.

- Squirt oil from an can to lubricate the faces of the seals
- Insert the sleeve with the rotary head
- The seal tension adjustment can be eliminated by the use of cartridge seals (Figure 9.20). These come with pre-adjusted seal tension.

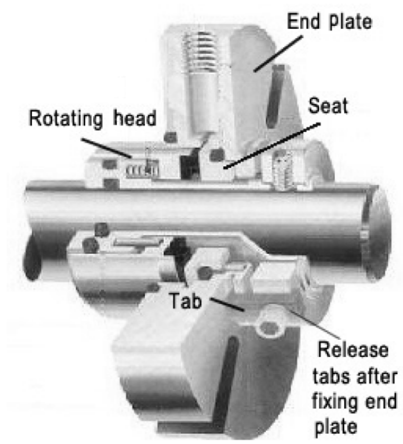



**Figure 9.20**
*Cartridge seal – seal tension is achieved after releasing tabs*

## 9.5.3 Impeller and casing assembly

- Once the sleeve with rotary head is placed, the compression is achieved after installing the impeller and locking it to the shaft with the help of the nut.
- After the impeller is fixed, it is a good idea to measure the runout of the impeller wearing ring in this installed condition. This should be within 0.05 mm.
- The casing gasket is placed in the pump casing.
- The casing is then bolted to the seal-housing flange, once again taking care of the match marks.
- The bolts are tightened to the specified torque value.

*Note*: The correct torque for bolting is based on the size, thread, and material of the bolt. Another very important factor is whether the bolt has been lubricated as say by application of any anti-seize compound.

If such a compound has been applied, then the tightening torque values should be set to approximately 2/3rd of the torque as specified for dry bolts.

- This completes the main assembly.

## 9.5.4 Seal hydrotest

- If the workshop has a provision to test the seals, it should be done so.
- To carry out the above test, blind flanges are bolted to the inlet and outlet nozzles of the pump.
- The blind flanges should be drilled and tapped to screw in fittings for filling the pump with water and venting air from the top most point.
- Water is then filled in the pump and all air is vented out.
- Pump is then pressurized to a minimum of 1–2 kg/cm$^2$ above the shut-off discharge pressure of the pump.
- Once pressurized, compressed air should be used to dry the pump externally, especially near the seal and shaft interface.
- The shaft is then rotated slowly by hand and pump should be observed carefully for any leaks from the seal.

- The casing joint should also be examined.
- Any drop in the hydrotest pressure should be investigated.
- If the hydrotest is satisfactory, the pressure should be released and the water should be drained.

### 9.5.5 Installation of coupling, lines, and fittings

- The pump coupling half can now be mounted onto the shaft.
- Shaft end should preferably be flush with the coupling half. However, if it was not so originally, this should have been recorded and kept along similar lines.
- The sealant lines, oil level gages, cooling water lines, or any other should be fitted.
- After the blind flanges are removed, the pump nozzles should be completely covered by a tape.
- The pump is ready for installation at site.

## 9.6 Vertical pump repair

The construction of the vertical pumps basically comprise of:

- Head/driver assembly
- An electrical motor and a cast or fabricated base from which the column and the bowl assembly is suspended
- Column and shaft assembly
- The column pipe is the link between the head assembly and the pump bowl assembly as shown in Figure 9.21. Its main function is to conduct the liquid from the bowl to the discharge. Within the column pipe is the pump shaft that transmit the power from the driver to the pump impellers. The pumped liquid generally lubricates the line shaft bearings in the column
- Pump bowl assembly
- Each bowl comprises of impeller, suction, and discharge casing.

There are three main reasons to overhaul or repair a vertical turbine pump.

1. One of them is the complete pump breakdown leading to unexpected disruption of a critical system or plant operations.
2. Reduced pumping efficiency due to increased clearances between the bowls and enclosed impeller skirts, increased clearances between the bowl shaft and bearings, and breakdown of the fluid passages. Over time, running a badly worn pump can be just as costly as some system shutdowns.
3. When operational conditions demand altered requirements in pressure and capacity.

The vertical pump comprises of a lot many more components than a simple horizontal pump and the maintenance problems are also different and can be substantially higher.

The rotor of the vertical pumps not stabilized by gravity is prone to gyroscopic effects that can cause damage to the rotor and the casing. Thus, the repair of such pumps demands care and attention.

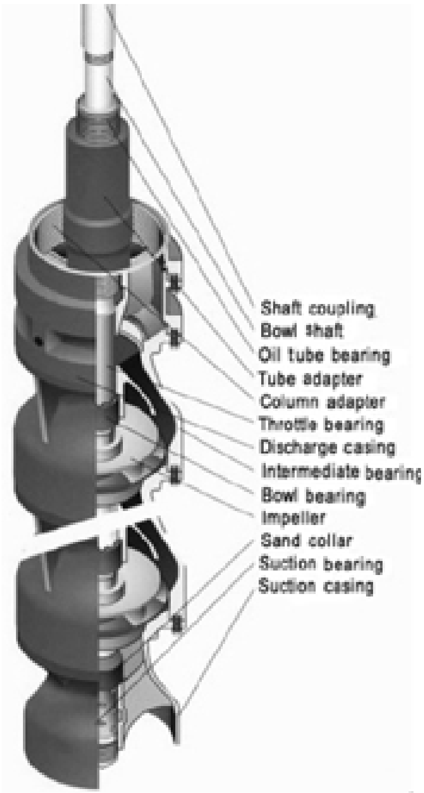



**Figure 9.21**
*Details of vertical turbine pump (American turbine pumps)*

## 9.6.1 Pump dismantling and repair

The repair and dismantling procedure of a vertical turbine pump is influenced by its construction. Pump repair is a lot more complex than a single-stage pump and it is advisable to prepare a datasheet for the pump.

This datasheet should comprise of all pump details, clearance and fits data, and any special remarks and procedures.

- The first step in the disassembly procedure should be to make the pump lie horizontally on the floor with the discharge nozzle facing the ground.
- Take a paintbrush and match mark all the joints. As the bowl joints can look similar, it is advisable to punch consecutive numbers on the joints taking care that these are not in addition to some other numbers of a previous overhaul. This would lead to a lot of confusion during assembly.
- Push the pump upper shaft toward the suction and measure the distance to the motor adapter face and adapter face configuration.
- Now pull the rotor outward and re-measure this distance to get the total axial float of the rotor.
- The pumps' bowls maybe individually screwed together, bolted together with a series of bolts around the bowl flange (most common design) or bolted together with tie bolts that extend from the bottom of the bottom to the top of the pumping section through the bowls. These maybe opened one section at a time.

- In the pump design, where a split collet bushing is used to lock the impeller on the shaft there the distance between the shaft end and the impeller must be measured and recorded.
- The float is measured and recorded at each stage (Figure 9.22).
- As the disassembly progresses, it is essential to match mark the impeller, collets, keys, locking pins, split keeps, and keeper plates. These should be arranged and kept in an organized manner.
- Similarly, the line shaft couplings should be match marked along the mating shaft on each end.

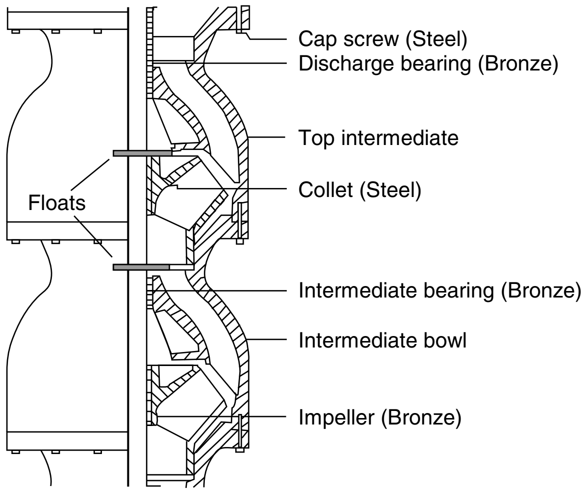



**Figure 9.22**
*Sketch indicating floats in assembly*

- The line shaft couplings should be removed using a pipe or chain tongs on the mating shafts. Wrenches applied to couplings can cause the coupling to collapse and seize on the shaft. If normal methods fail, the coupling can be cut using thin grinding wheel. The main aim is to prevent damage to the shaft, as it is the most expensive item.
- The rigid coupling halves from the motor and pump should be checked for trueness and squareness.

## 9.6.2 Preparation for assembly

Many checks and repairs have to be carried out in this stage:

- The fluid passages of the bowl and impeller need to be carefully examined for wear. A significant impact on efficiency can be observed if the above are damaged, pitted, have a particle build-up or have eroded coatings.

  Coatings are normally applied to the interior of the bowl assembly, mainly to enhance efficiency or for the protection of internals. However, the velocity of the fluids and suspended solids being pumped often damage the coating. Cleaning, recoating, or replacing the affected parts are options dependent on the severity of the problem.

- The next step is to check the condition of the shaft. The bearing areas are susceptible for wear. If the condition is found to be good, the next step is to measure the shaft runout. A typical layout for this procedure is shown in Figure 9.23.

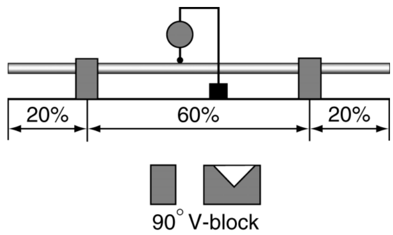



**Figure 9.23**
*Measuring shaft straightness*

The shaft is placed on V-blocks or knife-edge rollers, if available at the recommended distances. The dial gage is placed at various locations and the shaft is rotated a full circle by hand. The dial readings are measured and recorded.

The shaft runout should be within 3–4 mils (0.075–0.1 mm). If it is higher, efforts should be made to straighten it within the tolerances using a press.

Whenever possible, a single-piece shaft is recommended. If more than one piece is used and couplings are required, a radial weep hole should be provided in the center of the coupling. The shaft should be screwed together and runout is checked.

The bowl shaft is typically made of some grade of stainless steel (typically 12% Cr steel, ASTM – A-581, A-582, hot rolled bar, turned, ground and polished) rather than carbon steel because of the resistance of stainless to abrasion/corrosion.

If abrasive solids are pumped, shrunk-fitted sleeves can be fitted in the wear areas. Another solution can be the application of chrome or other hard coatings at these locations on the shaft.

The shaft could be in lengths of 20–24 in. with a straightness tolerance of 1.5 mils (0.4 mm) per foot (300 mm). The recommended surface finish is 16 μin. RMS.

- Impeller to bowl wearing ring clearances are measured and recorded. The wearing ring clearances should be in accordance with the API 610 (7th Ed.) as tabulated in Section 2.3.

  If the bowl does not have a wearing ring, then it should be checked to see if there is sufficient material available to bore the bowl and/or turn the impellers. Nevertheless, wear rings can replace the worn material and bring the parts back into tolerance.

  The fluid contains some particularly abrasive solids and the wear rings are made of materials to minimize the abrasive erosion. If wear rings are ordered with the original unit for easy replacement later, both bowl and impeller pieces are necessary to return the bowl assembly to its original tolerances.
- All impellers should be balanced individually as per API specifications.
- All bearing bores should be checked for parallelism with the shaft column.

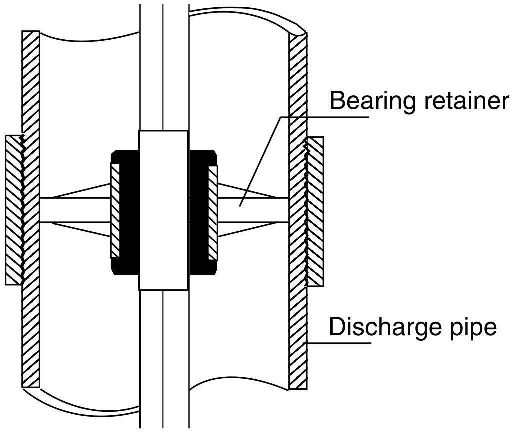



**Figure 9.24**
*Sketch showing line shaft bearings*

- All bushings in the bearing retainers should be measured for their clearance with the shaft. If they are replaced because of excessive clearance, they should be properly secured to the retainer as per their design and material of construction. These bushings generally have a spiral groove for lubrication as the clearances are generally on the tighter side (refer Figure 9.24).

  Most often, the bushings are bronze, rubber, or a combination of the two. Other materials used include bronze-backed rubber bearings, Teflon, and carbon composites.

  Bronze bushings are not used in corrosive, sandy, dirty water service. The allowable temperature range is from –30 to 90 °C.

  Above a pumping temperature of 90 ºC, cast iron bushings maybe used but these can withstand mild corrosive duty.

  When corrosive or lube lubricity products like hydrocarbons are to be handled, bushings made from carbon are used. These are not recommended for temperatures below –30 ºC.

  The rubber bearings are typically made from Neoprene. These are mostly used in sandy or dirt sump application. These may not be used at higher temperatures.

  When left with little choice coke-filled or glass-filled bushing could be used after proper consideration.

  The typical shaft/bearing clearance is 0.010", except when rubber bearings are used. In that case, the clearance can run up to 0.030", which makes it a much more forgiving material in applications where abrasive solids are present.
- The pipe maybe pitted or encrusted with scale, or the coating maybe damaged. Considering the degree of roughness it can be decided whether the pipe should be reused, recoated, cleaned, or replaced.

  The faces of the tubes should be inspected for leakage or damage, and the tubes themselves should be replaced when necessary.

  Enclosing tube bearings should have approximately the same tolerances as the bowl bearings (10 mils or 0.25 mm). Figure 9.25 show another drive shaft configuration with a separate oil lubricated column which offers extended bearing performance, but at a higher first cost, with pay back through the service life of the pump. Open-line shaft bearings should be replaced when they are visibly damaged. Usually, they are rubber and the condition of these bearings does not necessarily affect efficiency, but it is vital for supporting the shaft.

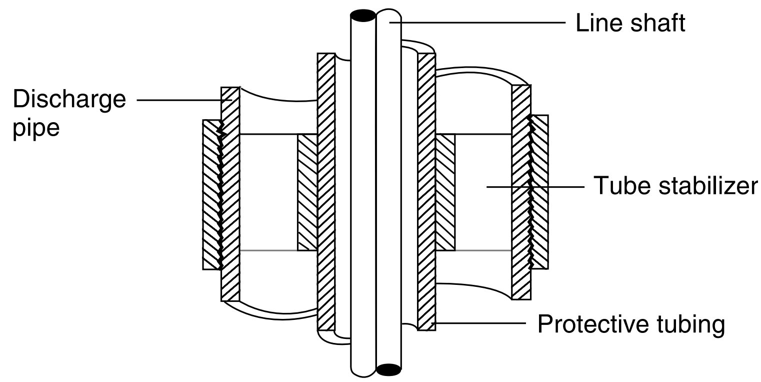



**Figure 9.25**
*Sketch of a screw-coupled discharge pipe with an oil-lubricated column*

- The discharge head of the pump generally requires the least maintenance. Most of the repairs are performed on the shaft seal, whether it is a stuffing box or a mechanical seal. Packing in the stuffing box should be replaced to control leakage. Stuffing box or seal-housing bearings need to be checked to make sure the shaft is properly supported.

  The repair of the mechanical seals is not very different from what is covered for the horizontal pumps.

  In vertical pumps, it makes more sense to have cartridge seals. In horizontal pumps of the overhung impeller types, the pump rotor has to be dismantled to access the seal, even though it maybe of the cartridge type.

  However, for in-between bearing pumps and vertical pumps, the impeller assembly need not be touched. Thus, seal replacement jobs can be done at site without removing the pump to a shop.

  Changing seals onsite can be tricky and prone to errors on account of site conditions, thus it make a strong case for cartridge seals in these type of pumps.
- The last check is the condition of the driver. When a vertical pump is started, it has a momentary up-thrust hydraulic action. When the pump is operated at high flows on a continuous basis it is likely to have this continuous upward acting axial force. This can stress the motor thrust bearing besides causing the probability to buckle line shafts, rub impellers, and leak of mechanical seals.

  The thrust bearings should be checked to ensure that they are not excessively worn.

  Motors upgraded to premium efficiency or at least replaced or rewound can significantly increase overall pump performance.

### 9.6.3 Pump assembly

- The assembly of the pump is carried in the vertical position. This prevents the risk of shaft or bowl assembly to develop sag. The bowl bushing clearance is typically 9 mils and in a horizontal position it is bound to touch after 3–4 bowl assemblies as the bowl have a spigot fits of 2–3 mils. In a vertical position, these get distributed.
- The jack-bolt in the suction piece must be used to position the end of the shaft to allow for accurate spacing of the impellers. If a load applied on a previously installed impellers can knock them off their collet while installing the next

impeller. If the impellers become loose, there is this grave danger of a rub on starting.

- Impellers fitted with collets in comparison to those held by splittings; keys, snap rings, and others need a lot more attention. Excessive tightening of the collet can lead to cracking of the impeller in the hub area.
- After the pump bowls have been assembled, the lift should be checked and the rotor should to rotated by hand to check for any rubbing of internals.
- If the length of the pump is large, column sections have to be assembled in a horizontal position.
- The assembled positions should be rotated 180° with installation of every additional component. This aids in staggering the alignment clearances, fits through the length of the columns, and helps to keep the assembly along the shaft centerline.
- After the columns have been fitted, the discharge head is installed. At this stage, the shaft extension length can be compared with the ones taken before dismantling of the pump. Deviations of 1/16th to 1/8th are considered as normal and can be compensated with the use of gaskets.
- The shaft extension maybe supported and the shaft can be locked before dispatching it to the site. If the jack bolt at the suction bell is left in place, it should accompany a warning tag to remove it before installation.

## 9.7 Multistage pump repair

The multistage pump considered here are the multistage barrel type double case pumps. It comprises of two casings; an inner assembly that comprises of the complex shape of the stationary hydraulic passages. This casing sees the external differential pressure so any bolting required to hold it is minimal. The casing can be of the type that is split axially into similar halves and cast from the same pattern as shown in Figure 9.26. Else, this could be of the multiple diffuser casing type.

The outer casing acts as a pressure boundary for the pumped liquid and is designed as a pressure vessel.

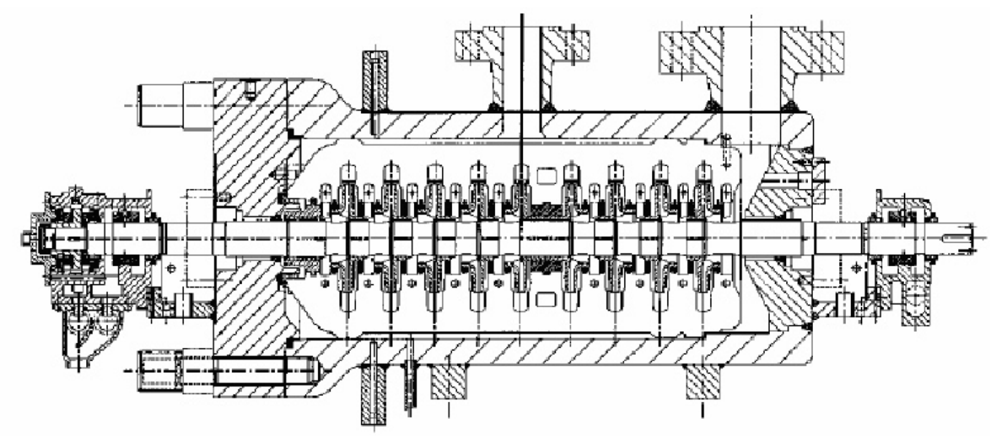


**Figure 9.26**
*Double case pump – cross section*

The rotor can be of opposed impeller configuration or of the inline type. The latter has balance disk to take care of axial thrust, which is much higher in this type of design.

These pumps are generally used as boiler feed water pumps, charge pumps handling hydrocarbons at high temperatures and pressures, water-flood pumps and pipeline pumps. Each of these applications falls under the critical category and their downtimes can be quite expensive.

There are designs that have a fully separable inner case subassembly that includes the rotor. This is done by removing the outboard cover and without breaking the suction line joint.

As downtimes are expensive, a spare inner subassembly is usually made ready and kept as a spare. During a repair requirement, the inner assemblies are just replaced. For some designs that include the journal and thrust bearing, the downtime is further reduced as even these adjustments are made beforehand.

Once the inner subassembly has been dismantled, the shaft runout is obviously the prime check. The runout should be with 1 mil. In the bearing areas, it should be less than half a mil.

In some designs, the impellers are shrunk fitted on the shaft to prevent mechanical looseness and positioned axially by split rings on the suction side of each impeller hub.

The grooves for the split rings cause the shaft to develop the run outs and because of the shrink fits, the shaft fits should be measured and recorded.

The next step should be to balance the shaft with the half keys taped. The shaft is mounted on a balancing machine and spun to a speed of 300 rpm for about 10 min. The shaft runout is then measured at mid-span and the angular position is marked on it.

The balance should be achieved by machining the face of the steps of the shaft. Once a balance is achieved, the runout should be rechecked.

The next step is assembling the rotor for balancing. To do this the shaft is placed vertically. All the rotating components that include the impellers, thrust collar and balance, drum (if installed) are stacked sequentially.

Since most of the components are shrunk fit, these are heated to about 110–120 °C depending on the bore and the shaft diameters.

The match marked split ring and the key for the first-stage impeller are first held in place. The impeller keyway is marked with a pen in case it is blind at one end.

The impeller is then heated to 120 °C and carefully slipped over the key to its position until it makes contact with the split rings. A constant downward pressure is then applied until the impeller has firmly gripped the shaft. To aid this process, cooling air may then be passed from the OD of the impeller to the suction eye until the temperature of the impeller is ambient.

Install the other impellers in a similar manner.

Once all the impellers are firmly fixed, the runout of the rotor is measured. If the runout is found to be within limits, the impeller nuts are tightened and the runout is rechecked.

If this is also found acceptable, the rotor should be fitted with all the rotating components and sent for balancing.

In case excessive runout is detected, the shaft nuts are the probable causes and their faces should be rectified.

The next step is to balance the rotor. The reader is recommended to read a detailed procedure covered in the paper, 'Monitoring Repairs to Your Pumps' by W. Edward Nelson presented at the 12th International Pump Users Symposium in 1995.

Once the rotor has been balanced, the rotor should be unstacked.

Once again, the shaft is placed vertically and this time the impellers are stacked along with the diffuser elements in a manner similar to the one explained above.

After installation of every stage, the float should be checked. For this purpose, a jack maybe placed under the coupling end of the shaft. The required lift should be maintained throughout the assembly.

The diffuser covers have a small interference fit of 0.01 mm and may require some heat for installation. In case the cover does not fit, the preceding cover should be heated with a small torch around entire circumference for a minute. This cover can then be tapped lightly using a hammer.

After all the impellers and diffusers have been installed, the balance piston is installed. Once it is cooled, the nut is tightened to the match marks made earlier.

The impeller diffuser lineup or location of the rotor axially to have a gap of '*Z*' (shown in Figure 9.27) is adjusted by the thickness of the locator ring placed between the thrust collar and the shaft step (see Figure 9.27).

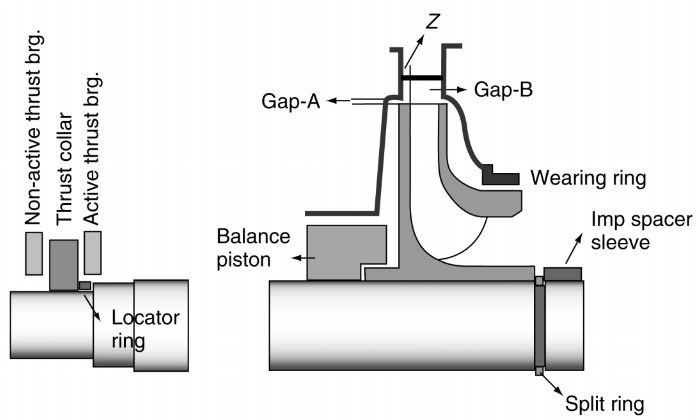



**Figure 9.27**
*Gap-A and Gap-B*

The relation of shroud and hub positions to the diffuser/volute and the casing must be verified and determined during assembly. Axially, the impeller discharge should be entirely within the axial width of the diffuser. If this is not the case, the flow from the impeller will discharge against the diffuser or casing walls. The rotor sensitivity to impeller and diffuser/volute alignment is dependent on the ratio ($B_2/B_3$ – see Figure 9.28) of the impeller exit vane width and diffuser vane width and must be judged on a case-by-case basis. However, a typical overlap of not less than 1/2 of the impeller side-plate (shroud/cover) thickness is preferred.

During dynamic balancing, if grinding of the impeller sidewall extends to the impeller OD such that it affects the overlap, the pump performance can get affected.

In addition to this axial location, there are two more famous dimensions, which should measured, recorded/corrected during an overhaul. These have a pronounced impact on the hydraulic stability of any high-energy pump. These are called as Gap-A and Gap-B as shown in Figure 9.28.

Gap-A is the radial clearance between the outer diameters of the impeller side plates; hub/cover, shroud, and the inner diameter of the diffuser channel side plates.

Gap-B is the radial clearance between the impeller vanes at its outer diameter and the diffuser or volute vane in the inner diameter. This is expressed as a percentage with respect to the impeller vane diameter.

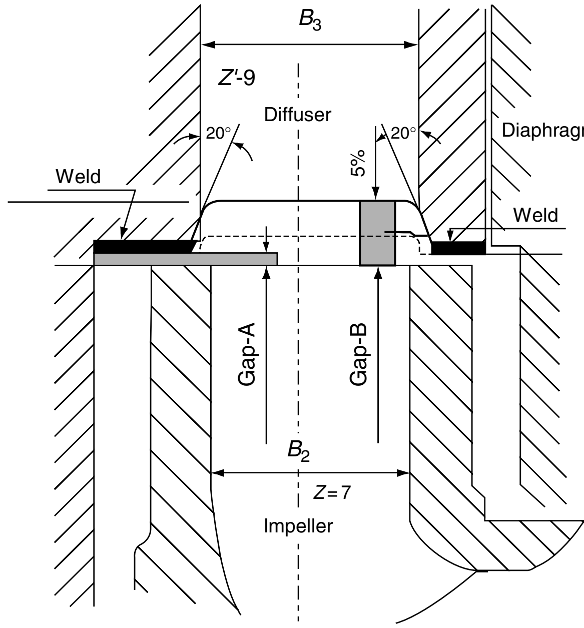



**Figure 9.28**
*Expansion gaskets and spacers*

The above gaps have to be maintained in accordance with the specific speed and construction of the pump. Correct gaps eliminate vane pass vibration frequency. Machining of the diffuser/volute vane to achieve the right Gap-B without affecting Gap-A causes no loss of efficiency. In fact, the high noise level, shock, and vibration caused by vane pass frequency are eliminated improving efficiency.

Gap-A has a typical value within the range of 35–85 mils. A preferred gap is of 50 mils, anything beyond 125 mils it losses its effectiveness.

In case of Gap-B, those pumps with a diffuser type construction the range is from 4 to 12% with a preferred gap of 6%. In a volute type of construction, this range varies from 6 to 12% with a preferred gap of 10%.

In case the number of impeller and diffuser vanes is even in number, the radial gap must be larger by 4%.

This assembles the inner subassembly comprising of the inner casing and the rotor.

These are then inserted in the outer casing using a guide fixture.

The expansion gaskets and spacers between the inner casing and outer head are installed as shown in Figure 9.29. Improper inner head gasket spacers can result in improper makeup of the intermediate diffuser covers, the suction spacer and the discharge diffuser spacer could lead to liquid recirculation resulting in drop in efficiency and erosion. The thickness of the spacer along with the gasket should achieve approximately 0.4–0.5 mm in compression on each gasket.

Next, the head is installed and tightened. In most designs, the head fit is all-metal and the gap at the head should be approximately 1–1.5 mm.

The bearing housings are installed with the lower half journal bearing in place. The coupling end housing is moved until suction impeller is centered in wearing rings within 0.025 mm. This measurement can be made using feeler gages.

The dial gage readings of the seal housing bores and faces are recorded.

The locating spacer and the thrust collar are inserted.

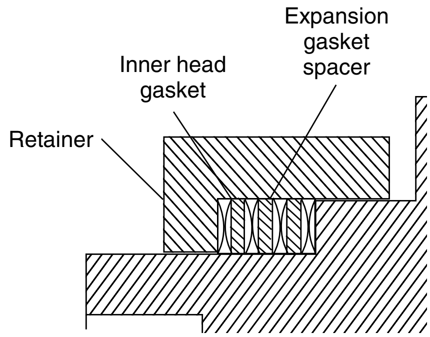



**Figure 9.29**
*Expansion gasket spacer*

Before the bearings are placed in the housings, the seals are installed.

The bearing housing dowels are inserted and these are bolted.

Finally, the journal and thrust bearings are installed. The location of the thrust collar lines up the rotor axially to get the right gap between the impeller and diffuser.

Coupling halves are installed and made ready for alignment.

## 9.8 Optimum time to maintain pumps

Over a period, the performance of the pumps deteriorates and it is always a debate whether to live with the problem or take the pump for an overhaul. In the first case, there is a continuous loss of money due to inefficient operation and for the latter; one has to incur the maintenance cost.

One has to work out an equation to determine the optimum time to take up the pump for an overhaul.

The algorithm presented here is the work of Ray Beebe presented in his paper 'Condition Monitoring of Pumps can save Money' at the APMA Seminar.

As per the theory put forward, the economic time to restore lost performance by overhaul will vary with the circumstances.

The basic assumption that is made is whether the:

- Deterioration is constant over a period of time
- Deterioration rate increases with time.

In the first case, when the deterioration is constant over time, then a cash flow analysis can be done to insure that the investment in overhaul will give the required rate of return. The same process is used in deciding on any investment in plant improvement.

When the deterioration rate increases with time, then the optimum time for overhaul will be when the accumulated cost of the increased power consumption equals the cost of the overhaul.

Deterioration in the pump performance can cause:

- Limitation in plant throughput: In such cases, the loss of production is quite high to justify an immediate cost of the pump overhaul and a matter of time to find an opportune window to carry out the overhaul.
- The pump operation is intermittent and does not affect the plant throughput. Any deterioration in pump performance only results in longer duration of

operation; typically, in case of pumps used for loading products in tankers. The additional time required for operation resulting in excess power consumption can be compared with the cost of the overhaul.

- In the initial stage, the pump performance deterioration does not affect the plant throughput and it is offset by opening a control valve or increasing the speed but over a period, a threshold is reached and the plant throughput begins to get affected. In some cases, the power requirement may exceed the motor rating leading to overload trips.

The performance calculation procedures described in the previous sections are used to determine the power that a pump consumes.

This can be benchmarked from the pump performance curves.

Using the two, one can determine excess power consumption of the pump and this can be used to trace lost expenses due to poor performance.

The pumps that use up excess power are primary candidates and should take a higher priority. The next thing factored is the cost of the overhaul. This now becomes a case of costs and benefits.

The following example is taken from Ray Beebe's paper for the second case where the pump runs intermittently and loss of performance does not affect the plant throughput.

Duration of pump operation: 24 months – operation is 27% of this time
Cost of the overhaul: \$50 000
Cost of power: 10 c/kWh
Excess power consumption: 167 kW

If the excess power consumption rate increase is assumed to be linear then,

Rate of increasing cost of overhaul per month is computed as follows:

- Number of hours in one month: 30 × 24 = 720 h
- Number of hours of operation (27%): 720 × 0.27 = 194.4h
- Excess power units consumed in 1 month: 167 × 194.4 = 32464.8 kWh
- Cost of this excess power @10 c/kWh: 32464.8 × 0.1 = \$3246.5

This is the cost of deterioration at the 24th month.

However, this is not the same for all of the 24 months. In fact, it must have been quite less in the initial stages. So, an assumption is made that the rate of deterioration is linear. Therefore, the rate of cost of deterioration is spread over 24 months.

The cost of deterioration increases by \$3246.6 per month/24 months

This average cost of deterioration is \$135.3/month/month.

From here on the process is iterative but can be easily solved using a spreadsheet calculation.

The procedure to do it in an MS-Excel spreadsheet is shown in Figure 9.30.

The optimum time for an overhaul is when the average cost of overhaul per month is equal to average cost of excess power.

For the iterative process an optimum time is assumed, in this example, say we consider 22 months.

The average cost of overhaul is: \$50 000 ÷ 22 = \$2273/month.

The average cost of extra energy used is: \$135 × 1/2 × 22 = \$1488/month.

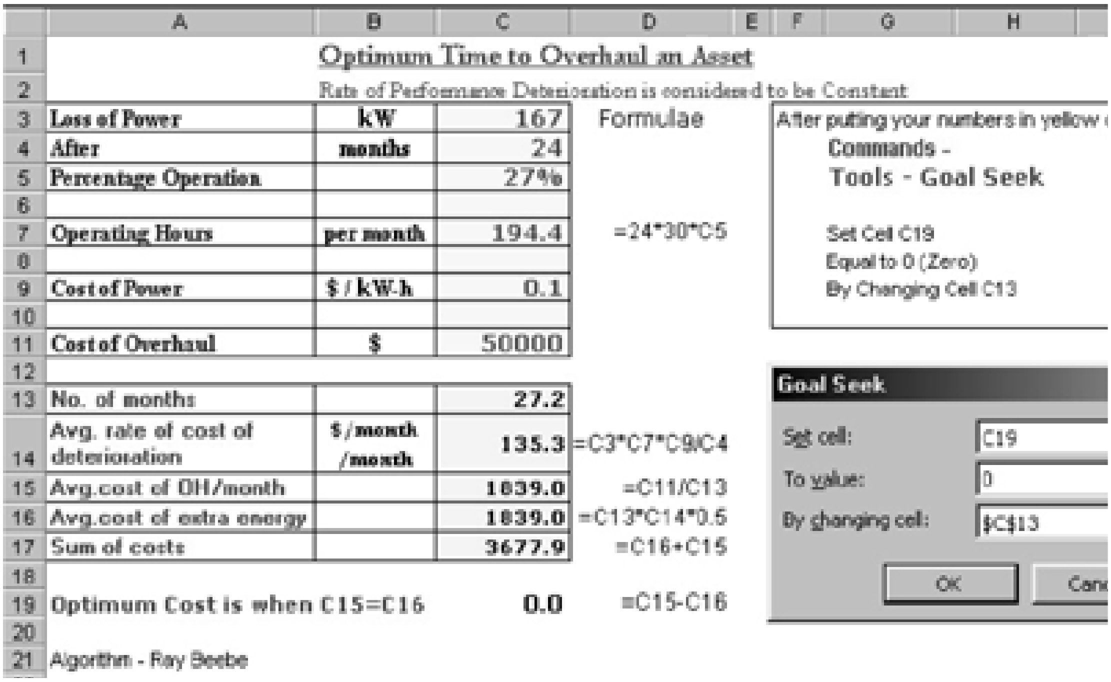
The total average cost/month is therefore the sum of these two figures, or \$3760.7.

Here we see the two costs are not equal and hence the total cost is not optimum.

This process is repeated till both the costs become equal.

This process is repeated and it is found that for this example the optimum duration is computed to be 27.2 months.

The total cost works out to $3677.9.



| | A | B | C | D |
|---|---|---|---|---|
| 1 | Optimum Time to Overhaul an Asset | | | |
| 2 | Rate of Performance Deterioration is considered to be Constant | | | |
| 3 | Loss of Power | kW | 167 | Formulae |
| 4 | After | months | 24 | |
| 5 | Percentage Operation | | 27% | |
| 6 | | | | |
| 7 | Operating Hours | per month | 194.4 | =24*30*C5 |
| 8 | | | | |
| 9 | Cost of Power | $ / kW-h | 0.1 | |
| 10 | | | | |
| 11 | Cost of Overhaul | $ | 50000 | |
| 12 | | | | |
| 13 | No. of months | | 27.2 | |
| 14 | Avg. rate of cost of deterioration | $/month /month | 135.3 | =C3*C7*C9/C4 |
| 15 | Avg.cost of OH/month | | 1839.0 | =C11/C13 |
| 16 | Avg.cost of extra energy | | 1839.0 | =C13*C14*0.5 |
| 17 | Sum of costs | | 3677.9 | =C16+C15 |
| 18 | | | | |
| 19 | Optimum Cost is when C15=C16 | | 0.0 | =C15-C16 |
| 20 | | | | |
| 21 | Algorithm - Ray Beebe | | | |



**Figure 9.30**
*An MS-Excel worksheet calculation to work out the algorithm*

Note that this calculation is only correct if the wear progresses at a uniformly increasing rate with time. Information may not be available to make any other assumption, but decision makers have to start somewhere.

Note that some relatively small pumps may never justify overhaul on savings in energy use alone, but maybe justified on reduced plant production rate when the pump deterioration does not affect production, at least initially.

For a pump where the speed is varied to meet its desired duty, the effect of wear on power required is much more dramatic than for the case of a constant speed throttle controlled pump. This is because the power increases in proportion to the speed ratio cubed.

Unless the pump output is limited by the pump reaching its maximum speed, or by its driver reaching its highest allowable power output, then no production will be lost.

However, power consumed will increase more dramatically for a given wear state than for a constant speed pump. The performance monitoring tools can be used or portable instruments can accurately indicate the excess power being consumed.

The same method and calculations can be used as before to find the time for overhaul for minimum total cost.

This algorithm is quite general in nature and can be used not just for pumps alone. Where there is deterioration in performance leading to loss of energy, it is a handy tool for maintenance engineers and managers in their role of managing assets to provide capacity for production.

**Clearances Chart for Single-Stage Centrifugal Pumps**

**All Values are in mm (Else Stated)**

| | | Minm. | Maxm. | Remarks |
|---|---|---|---|---|
| *Bearing assembly* | | | | |
| 1. Shaft runout | | | 0.025 | |
| 2. Bearing ID fit | | –0.003 | 0.02 | Positive here is interference |
| 3. Bearing OD fit | Less than 2" | –0.0017 | 0.0016 | Positive here is interference |
| | More than 2" | –0.02 | 0.03 | Positive here is interference |
| 4. Brg. housing runout | Radial | | 0.05 | |
| | Facial | | 0.05 | |
| 5. Shaft axial float | | 0.01 | 0.03 | |
| 6. Sleeve runout | | | 0.05 | Shaft to sleeve clr. – 0.07 to 0.1 mm |
| 7. Imp. wear ring runout | Front | | 0.07 | Shaft to imp. clr. – 0.02 to 0.03 mm |
| 8. Imp. wear ring runout | Back | | 0.07 | |
| *Seal housing assembly* | | | | |
| 9. Brg hsg. spigot & seal hsg. clearance | | | 0.1 | |
| 10. Case wear ring runout | | | 0.07 | |
| 11. Seal cavity concentricity with shaft | | | 0.1 | |
| 12. Throat bush clearance | | 0.15 | 0.2 | |
| *Seal assembly* | | | | |
| 13. Rotating Face flatness – light bands | | 1.0 | 3.0 | Seal retainer with sleeve – 0.07 to 0.1 clearance fit |
| 14. Stationary Face flatness – light bands | | 1.0 | 3.0 | |
| *Casing assembly* | | | | |
| 15. Impeller wear ring clearance | Front | | | Imp wearing ring clearance (WRC) – |
| 16. Impeller wear ring clearance | Back | | | Under 4" – [(Inch dia./1000)+10] mils |
| 17. Back vane clearance | | 0.8 | 1.0 | 4" & above – [(Inch dia./1000)+12] mils |
| 18. Impeller diameter | | | | Add 5 mils for galling materials & |
| 19. Imp. residual unbalance – gm-mm/Kg | | | | product temp. above 260 °C |
| *Coupling assembly* | | | | |
| 20. Coupling (keyed) to shaft fit | | –0.01 | 0.01 | |

| **Additional Fits for Multistage Pumps** | | | |
|---|---|---|---|
| *Rotor assembly* | | | |
| 1. Shaft runout | | 0.07 | |
| 2. Shaft and bottom sleeve | 0 | –0.02 | Interference |
| 3. Shaft and spacer sleeve | 0.02 | 0.03 | |
| 4. Bottom and intermediate bush with body | 0 | –0.03 | |
| 5. Bottom bush and sleeve | 0.1 | 0.15 | |
| 6. Intermediate bush and sleeve | 0.7 * WRC | 0.8 * WRC | Rule of thumb |
| 7. Throat bush and sleeve | 0.25 | 0.3 | |
| 8. Shaft and bearing sleeve | 0.02 | 0.04 | |
| 9. Diffuser face and drum spigot | 0.2 | 0.5 | |
| 10. Drum spigot diameter clearance | 0.07 | 0.1 | |
| 11. Assembly total float | | | Pump specific |
| 12. Wear ring clearance – 1st stage | | | Refer single stage form – 16 |
| 13. Wear ring clearance – balance stages | | | Refer single stage form – 16 |
| 14. Impeller wear ring runouts | | 0.07 | |
| 15. Case wear ring runouts | | 0.07 | |
| 16. Intermediate sleeve runouts | | 0.07 | |

# Appendix A

# Pump types

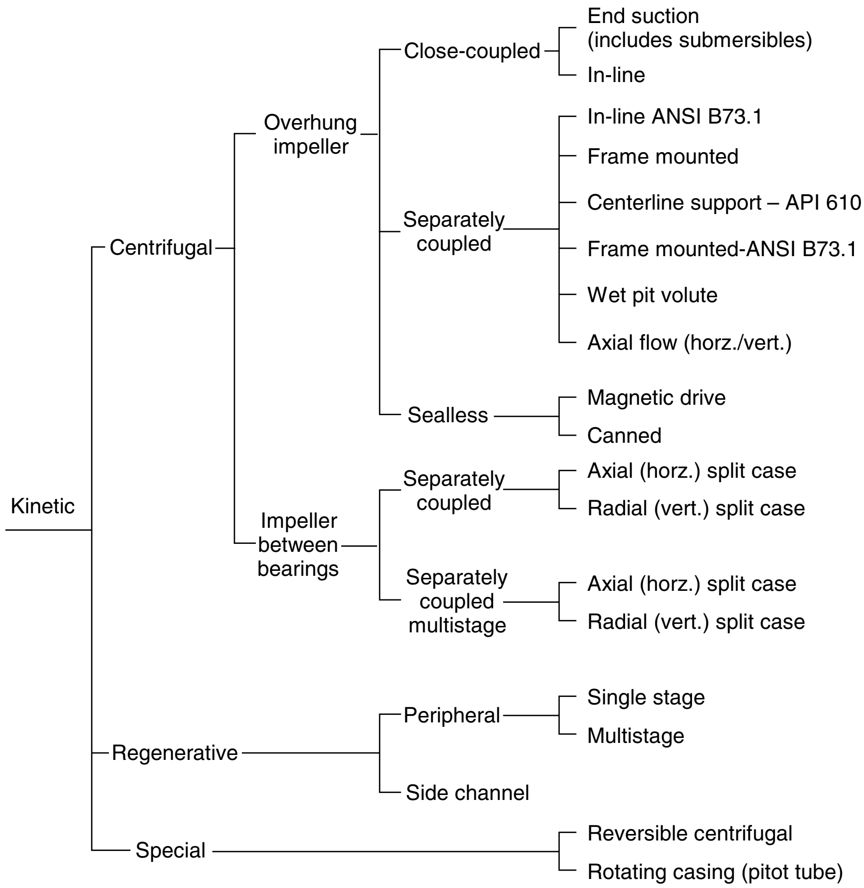



**Figure A.1**
*Classification of pumps*

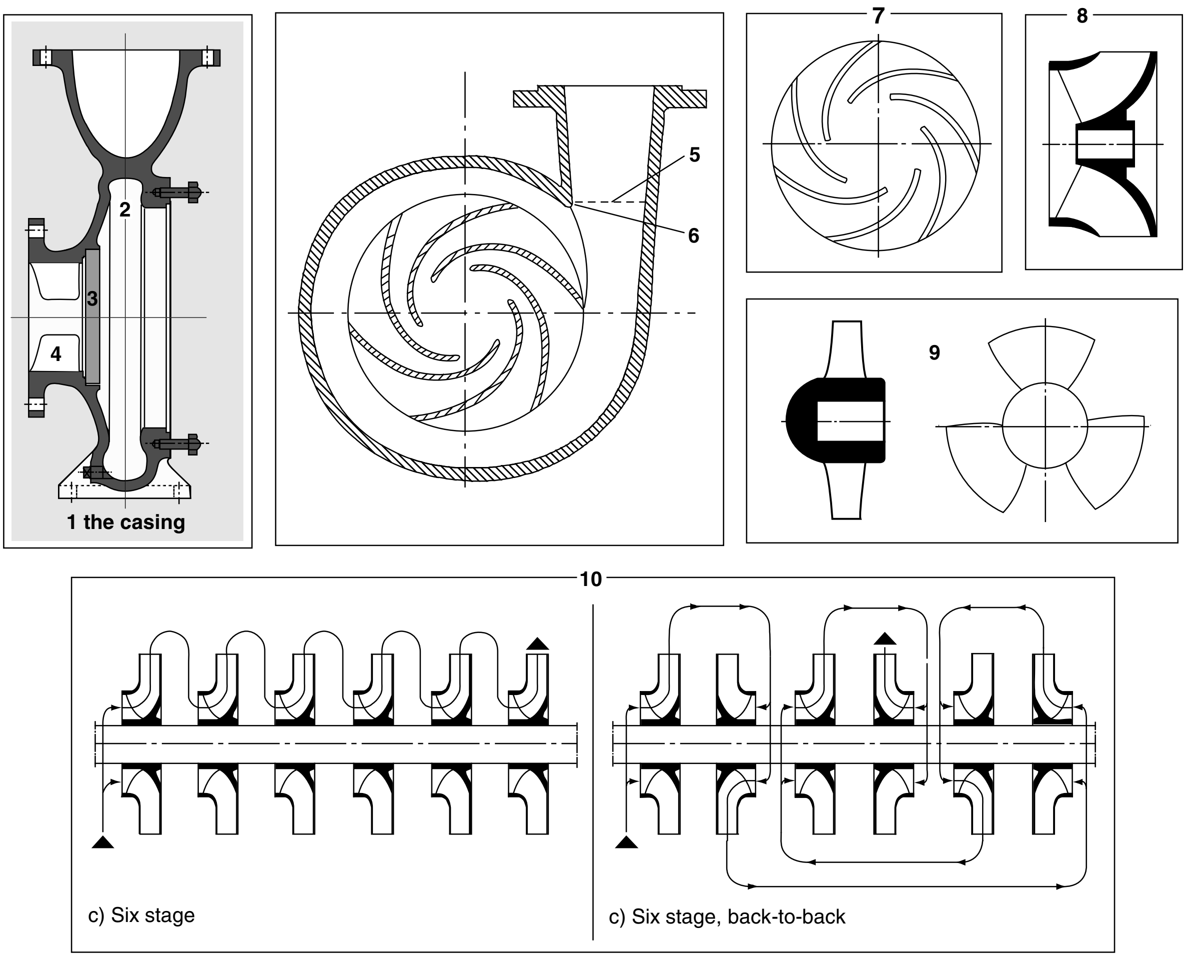
# A.1 Tutorials

## A.1.1 Tutorial 1

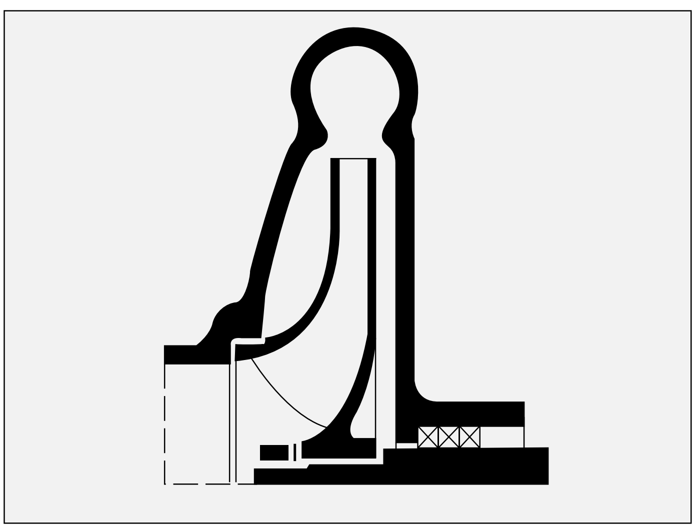
1. What three classifications can be given to the properties of a good pump?
2. List at least eight issues to be considered when evaluating risks of a new engineering project.
3. Why is risk assessment an essential part of any task?
4. List the main components of life cycle cost of any engineering project.
5. What would be appropriate measures for comparative assessment of life cycle cost of alternative pumping plants?
6. List some areas where statutory requirements apply to pump installations.

## A.1.2 Tutorial 2

| Question Numbers Refer to Numbers on the illustrations on the Next Page | Enter Your Answer |
|---|---|
| 1 | List the functions of the pump casing. |
| 2 | Name the element and describe its purpose. |
| 3 | Name the element and describe its purpose. |
| 4 | Name the element and describe its purpose. |
| 5 | Name the element. |
| 6 | Name the element. |
| 7 | Name the impeller type. |
| 8 | Name the impeller type. |
| 9 | Name the impeller type. |
| 10A | List the relative advantages/disadvantages of the two impeller layouts illustrated. |
| 10B | Why are multistage pumps required? |



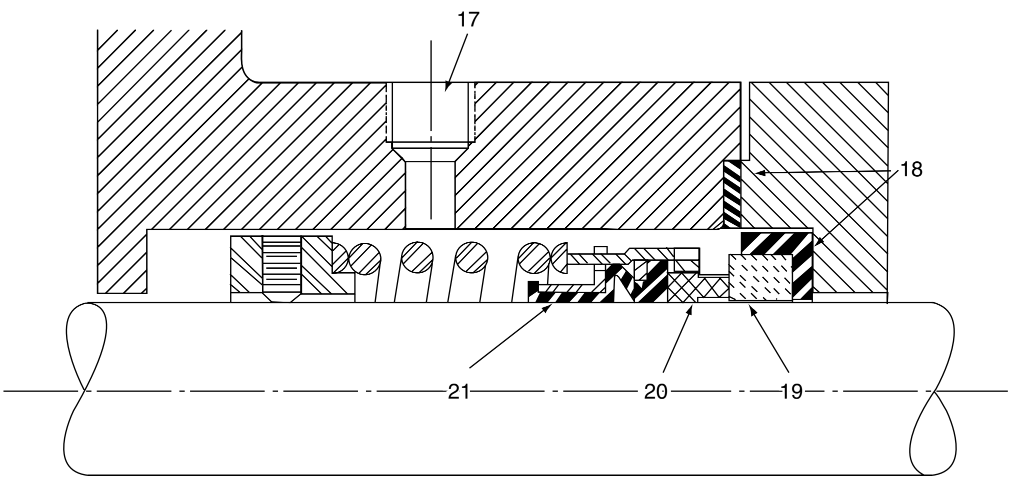

11. Sketch the distribution and direction of hydraulic axial forces applied to the illustrated pump impeller during pump operation.



12. How would the force distribution be changed if the impeller was of the open type? (Sketch the pressure distribution)

13. List possible sources of axial forces acting on the pump?
14. At which operating point are radial forces in a centrifugal pump at a minimum and a maximum?

| **Question Numbers Refer to Numbers on the Illustrations on the Next Page** | **Enter Your Answer** |
|---|---|
| 15 | List the functions of the shaft seals. |
| 16 | Name the main types of shaft seals in use. |
| 17 | Name the element and describe its purpose. |
| 18 | Name the element and describe its purpose. |
| 19 | Name the element and describe its purpose. |
| 20 | Name the element and describe its purpose. |
| 21 | Name the element and describe its purpose. |

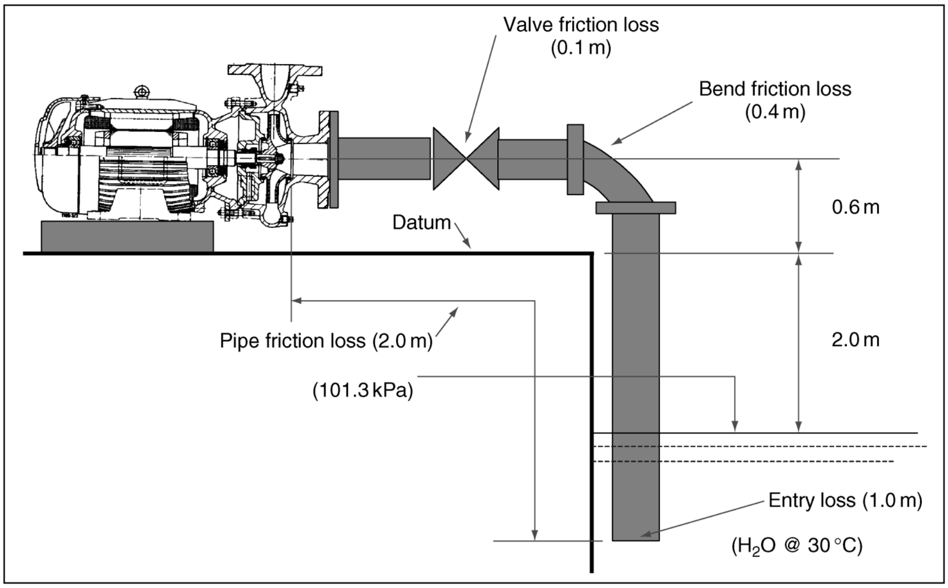



Section: Mechanical seal

## A.1.3 Tutorial 3

1. Correct the following incorrect pump laws:
   For similar conditions of flow (i.e. the same efficiency):
   - *Capacity* is directly proportional to speed and or impeller diameter.
   - *Power* is directly proportional to the square of speed and or impeller diameter.
   - *Head* is directly proportional to the cube of speed and or impeller diameter.
2. Express the corrected version of each law as a formula.

3. Define Net Positive Suction Head.
4. What is the difference between NPSH-a and NPSH-r?
5. Calculate NPSH-a for the following example:

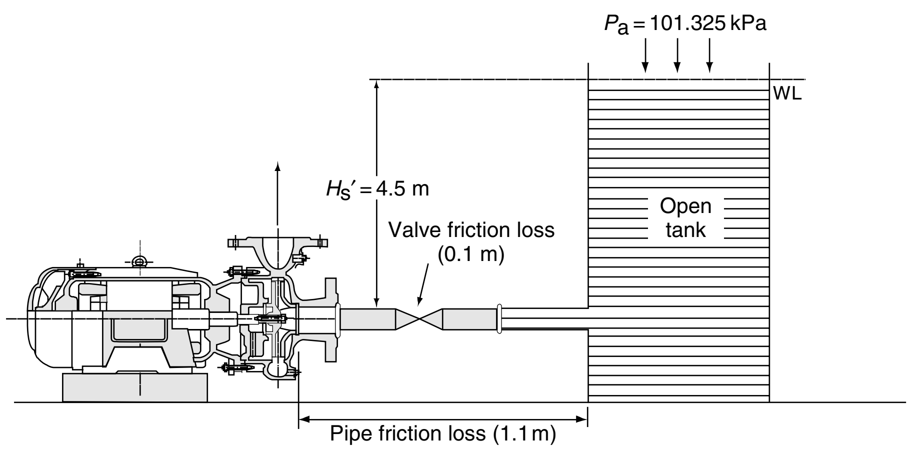



6. List ways of avoiding cavitation.
7. Sketch a drooping pump curve.
8. What are the relative merits of steep an flat pump curves.
9. Additional NPSH-a calculations.

*9.1 Suction head/open tank – given*

Valve friction loss = 0.1 m, Pipe friction loss = 1.1 m, Static head ($H_s$) = 4.5 m, Water temperature = 20 °C, Density ($\rho$) @ 20 °C = 998.3 kg/m$^3$, Atmospheric pr ($H_a$) = 101.325 kPa (absolute), Vapor pr = 2.337 kPa (absolute), Gravitational acceleration = 9.81 m/s$^2$.

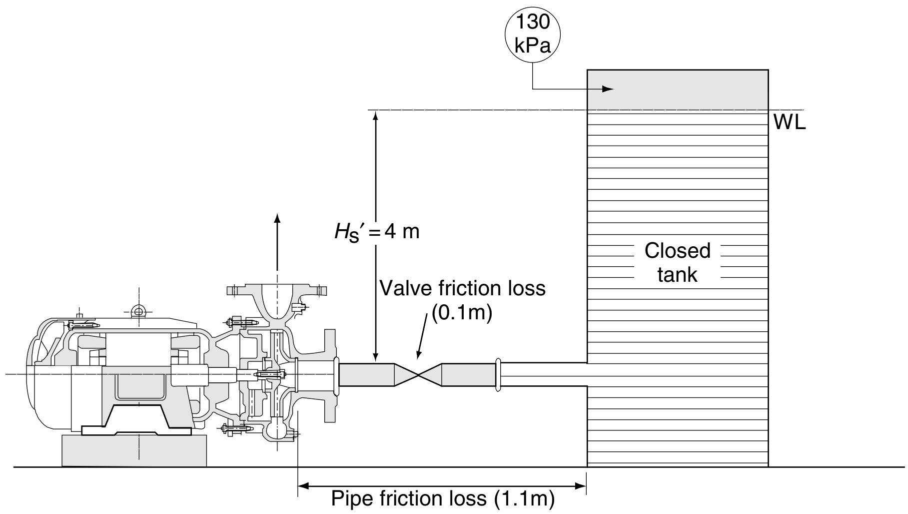
$$P = \rho \times g \times H \quad \text{and} \quad H = P / \rho \times g$$



*9.2 Suction head/closed tank*

Water temperature = 130 °C, Density @ 130 °C = 934.6 kg/m$^3$, Vapor pr @ 130 °C = 270.13 kPa (absolute), Gage pr = 130 kPa$_g$, Gravitational acceleration = $g$ = 9.81 m/s$^2$, Suction head ($H_s$) = 4.0 m, Pipe friction loss = 1.1 m, Valve friction loss = 0.1 m.

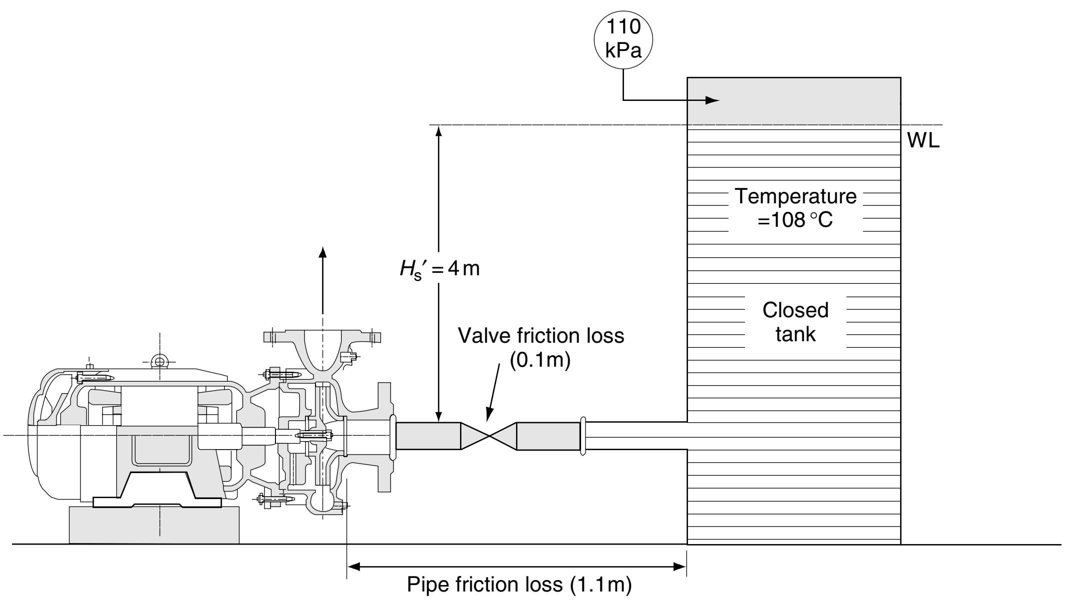
$$\text{NPSH-a} = H_{st} - H_{vap}$$




*9.3 Suction head/closed tank – given*

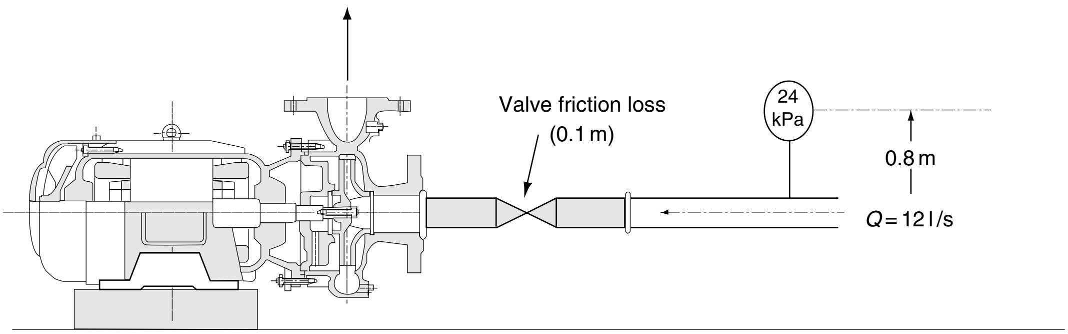
Water temperature = 108 °C, Density @ 108 °C = 952.2 kg/m$^3$, Vapor pr @ 108 °C = 133.90 kPa (absolute), Gage pr = 110 kPa$_g$, Gravitational acceleration = $g$ = 9.81 m/s$^2$, Suction head ($H_s$) = 4.0 m, Pipe friction loss = 1.1 m, Valve friction loss = 0.1 m.

$$\text{NPSH-a} = H_{st} - H_{vap}$$



*9.4 Suction head / suction gage reading – given*

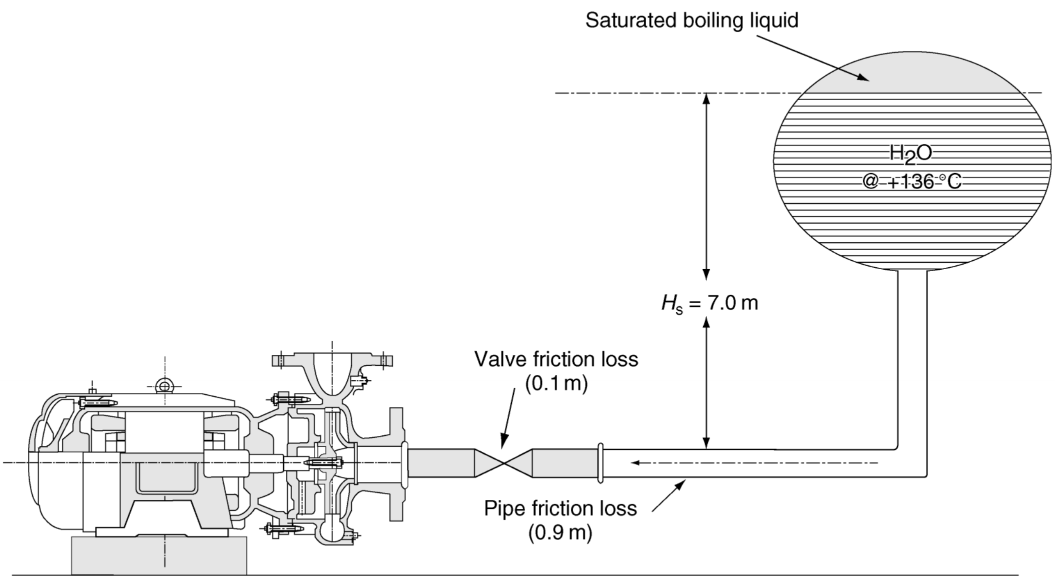
Liquid flow = 12 l/s = $12 \times 10^{-3}$ $m^3/s$, Gage pr = 24 $kPa_g$, Suction pipe diameter = 58.2 mm = 0.0582 m, Water temperature = 50 °C, Density @ 50 °C = 988 $kg/m^3$, Vapor Pr @ 50 °C = 12.335 kPa (absolute), Gravitational acceleration = 9.81 $m/s^2$, Atmospheric pressure = 101.325 kPa.



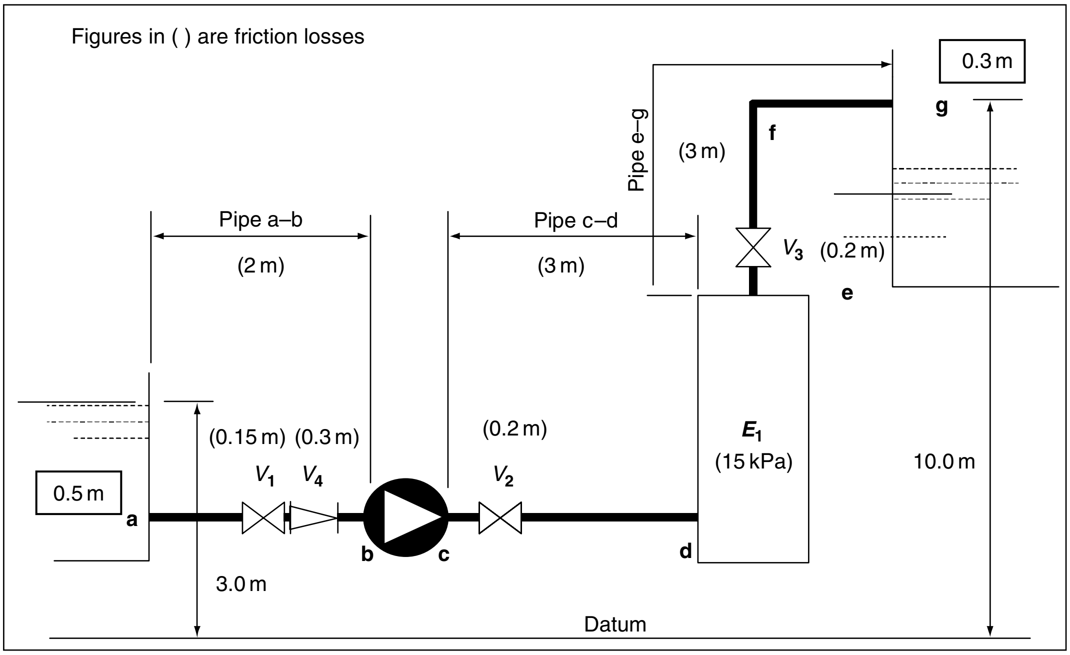

*9.5 Suction head/saturated (boiling) liquid – given*

Water temperature = 136 ºC, Density of water = $\rho = 929.4$ $kg/m^3$, Pressure in tank = vapor pr = 322.3 kPa (absolute), Gravitational acceleration = 9.81 $m/s^2$, $H_s$ = 7.0 m, Valve friction (head loss) = 0.1 m, Pipe friction (head loss) = 0.9 m.

*As the liquid is in equilibrium – (saturated state), the vessel pressure* $H_{vap}$ *and gage pressure (if fitted) would cancel out.*

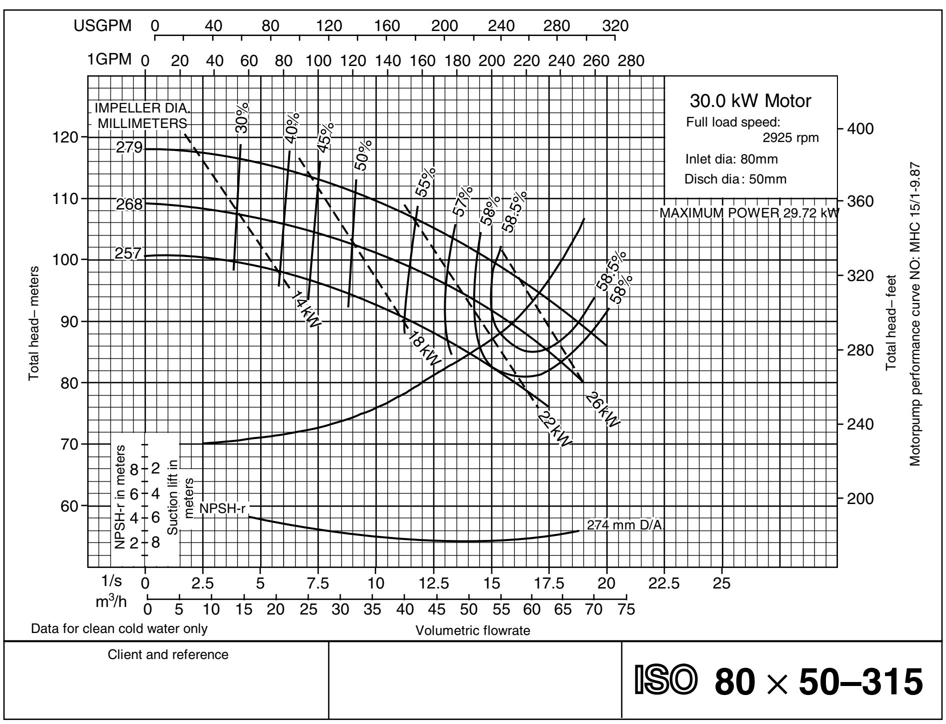



10. Calculate the total system head for the hydraulic system illustrated. Assume density of water is 1000 $kg/m^3$ and entry loss at '*a*' = 0.5 m and sudden expansion loss at '*g*' = 0.3 m and 0.4 for bend '*f*'.




11. Plot a system resistance curve, on the attached pump performance chart, for a system with the following parameters:

 Impeller Diameter = 268 mm, Operating point; $Q = 15.75$ l/s, $H_t = 90$ m
 Static head = 70 m, Fluid pumped is water of density = 1000 kgs/m$^3$.
 The plot should show the resistance curve from zero to 17.5 l/s flow.
 (Supply Grundfos 80 × 50–315 pump curves.)



## A.1.4 Tutorial 4

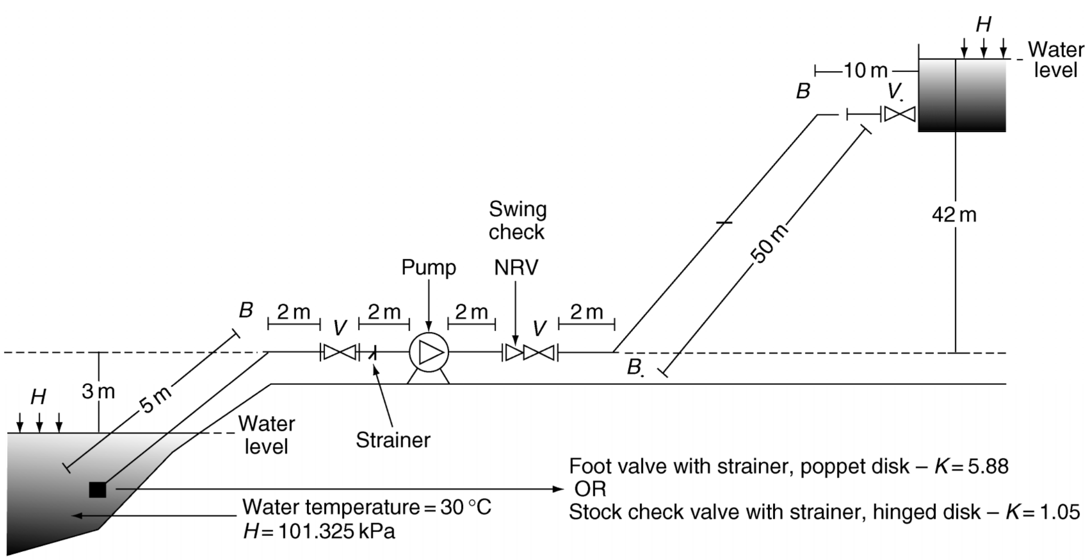
1. Given that the pump with an **impeller diameter of 268 mm** is putting out 15.75 l/s @ 90 m head, calculate the pump impeller diameter to put out 14.5 l/s.
2. What would be the output if a second identical pump with impeller diameter 268 mm was run in parallel with the system pipe work remaining the unchanged? How could flow in parallel operations be improved?
3. How would the head, of the pump in 1 above, vary if the kinematic viscosity of the liquid pumped was $10^{-4}$ $m^2/s$ and the specific gravity was 0.85? (Use viscosity charts provided)
4. List the issues that elevated fluid temperatures raise for pump selection and operation.

## A.1.5 Tutorial 5

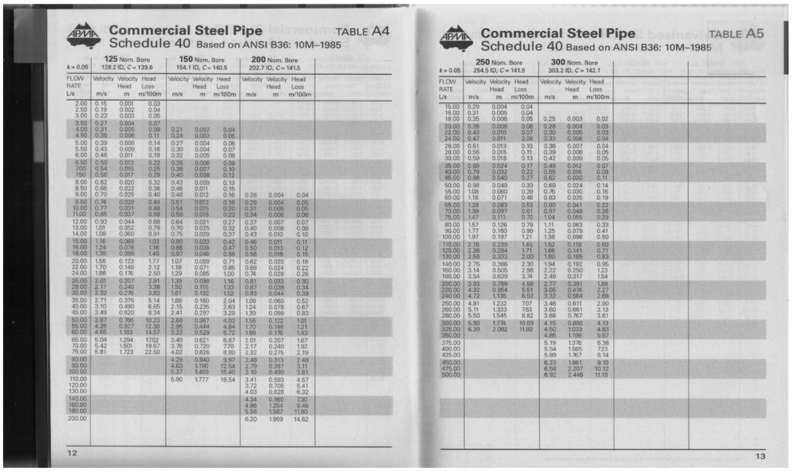
### Pump selection exercise:

In the example provided below, calculate the following:

- NPSH-a for the system.
- NPSH-r from the pump performance chart at the operating point.
- The total system head utilising the equivalent length – valves and fittings chart and tables provided.
- Flow, head and power absorbed by the pump when operating unrestricted.
- Power absorbed by the pump when operating with the discharge throttled to achieve 140 l/s flow.
- If the drive efficiency is 96.5% and the motor efficiency is 91%, calculate the motor running amps with flow at 156 l/s.
- What size (high efficiency) motor must be selected for the pump to operate safely and reliably?



| GIVEN Data | |
|---|---|
| **Process requirements '$Q$'** | **140.00 l/s** |
| **Design Point, head '$H$'** | **51.50 m** |
| Water temperature | 30.00 °C |
| Water density $\rho =$ | 995.70 kg/m$^3$ |
| Gravitational acceleration – $g$ | 9.81 m/s$^2$ |
| Pump rpm (4 × pole motor) | 1490.00 rpm ((Hz × 60)/pairs of poles) |
| Pump impeller diameter = | 406.00 mm |
| Pump Suct/Disch diameter | 250.00 mm |
| Atmospheric Pr ($H_a$) | 101.33 kPa |
| 9.789 kPa = 1 m or 9.8 kPa = | 1.00 m |
| From chart, friction loss in m/100 m length for 250 mm nominal bore Schedule 40 commercial steel pipe = | 2.30 m/100 m and velocity = 2.75 m/s$^2$ |
| *Note*: resistance coefficient '$K$' for foot valve with strainer poppet disk (refer tables for '$K$' value) = | 5.88 |
| Head loss for strainer = $K \times v^2 / 2g$ | 2.27 (m – equivalent length of pipe) |
| Entry loss (refer tables for value '$K$' = 0.5) then loss = | 0.19 (m – equivalent length of pipe) |
| Exit loss (refer tables for value '$K$' = 1.0) then loss = | 0.39 (m – equivalent length of pipe) |



APMA Commercial Steel Pipe TABLE A4
Schedule 40 Based on ANSI B36: 10M–1985

| $k$ = 0.05 | 125 Nom. Bore 128.2 ID, $C$ = 139.6 | | | 150 Nom. Bore 154.1 ID, $C$ = 140.5 | | | 200 Nom. Bore 202.7 ID, $C$ = 141.5 | | |
|---|---|---|---|---|---|---|---|---|---|
| FLOW RATE L/s | Velocity m/s | Velocity Head m | Head Loss m/100m | Velocity m/s | Velocity Head m | Head Loss m/100m | Velocity m/s | Velocity Head m | Head Loss m/100m |
| 2.00 | 0.15 | 0.001 | 0.03 | | | | | | |
| 2.50 | 0.19 | 0.002 | 0.04 | | | | | | |
| 3.00 | 0.23 | 0.003 | 0.05 | | | | | | |
| 3.50 | 0.27 | 0.004 | 0.07 | | | | | | |
| 4.00 | 0.31 | 0.005 | 0.09 | 0.21 | 0.002 | 0.04 | | | |
| 4.50 | 0.35 | 0.006 | 0.11 | 0.24 | 0.003 | 0.06 | | | |
| 5.00 | 0.39 | 0.008 | 0.14 | 0.27 | 0.004 | 0.06 | | | |
| 5.50 | 0.43 | 0.009 | 0.16 | 0.30 | 0.004 | 0.07 | | | |
| 6.00 | 0.46 | 0.011 | 0.19 | 0.32 | 0.005 | 0.08 | | | |
| 6.50 | 0.50 | 0.013 | 0.22 | 0.36 | 0.006 | 0.09 | | | |
| 7.00 | 0.54 | 0.015 | 0.25 | 0.38 | 0.007 | 0.10 | | | |
| 7.50 | 0.58 | 0.017 | 0.29 | 0.40 | 0.008 | 0.12 | | | |
| 8.00 | 0.62 | 0.020 | 0.32 | 0.43 | 0.009 | 0.13 | | | |
| 8.50 | 0.66 | 0.022 | 0.36 | 0.46 | 0.011 | 0.15 | | | |
| 9.00 | 0.70 | 0.025 | 0.40 | 0.48 | 0.012 | 0.16 | 0.28 | 0.004 | 0.04 |
| 9.50 | 0.74 | 0.028 | 0.44 | 0.51 | 0.013 | 0.18 | 0.29 | 0.004 | 0.05 |
| 10.00 | 0.77 | 0.031 | 0.48 | 0.54 | 0.015 | 0.20 | 0.31 | 0.005 | 0.05 |
| 11.00 | 0.85 | 0.037 | 0.58 | 0.59 | 0.018 | 0.23 | 0.34 | 0.006 | 0.06 |
| 12.00 | 0.93 | 0.044 | 0.68 | 0.64 | 0.021 | 0.27 | 0.37 | 0.007 | 0.07 |
| 13.00 | 1.01 | 0.052 | 0.79 | 0.70 | 0.025 | 0.32 | 0.40 | 0.008 | 0.08 |
| 14.00 | 1.08 | 0.060 | 0.91 | 0.75 | 0.029 | 0.37 | 0.43 | 0.010 | 0.10 |
| 15.00 | 1.16 | 0.069 | 1.03 | 0.80 | 0.033 | 0.42 | 0.46 | 0.011 | 0.11 |
| 16.00 | 1.24 | 0.078 | 1.16 | 0.86 | 0.038 | 0.47 | 0.50 | 0.013 | 0.12 |
| 18.00 | 1.39 | 0.099 | 1.45 | 0.97 | 0.048 | 0.58 | 0.56 | 0.016 | 0.15 |
| 20.00 | 1.55 | 0.123 | 1.77 | 1.07 | 0.059 | 0.71 | 0.62 | 0.020 | 0.18 |
| 22.00 | 1.70 | 0.148 | 2.12 | 1.18 | 0.071 | 0.85 | 0.68 | 0.024 | 0.22 |
| 24.00 | 1.86 | 0.176 | 2.50 | 1.29 | 0.085 | 1.00 | 0.74 | 0.028 | 0.26 |
| 26.00 | 2.01 | 0.207 | 2.91 | 1.39 | 0.099 | 1.16 | 0.81 | 0.033 | 0.30 |
| 28.00 | 2.17 | 0.240 | 3.36 | 1.50 | 0.115 | 1.33 | 0.87 | 0.038 | 0.34 |
| 30.00 | 2.32 | 0.276 | 3.83 | 1.61 | 0.132 | 1.52 | 0.93 | 0.044 | 0.39 |
| 35.00 | 2.71 | 0.375 | 5.14 | 1.88 | 0.180 | 2.04 | 1.08 | 0.060 | 0.52 |
| 40.00 | 3.10 | 0.490 | 6.65 | 2.15 | 0.235 | 2.63 | 1.24 | 0.078 | 0.67 |
| 45.00 | 3.49 | 0.620 | 8.34 | 2.41 | 0.297 | 3.29 | 1.39 | 0.099 | 0.83 |
| 50.00 | 3.87 | 0.766 | 10.23 | 2.68 | 0.367 | 4.03 | 1.55 | 0.122 | 1.01 |
| 55.00 | 4.26 | 0.927 | 12.30 | 2.95 | 0.444 | 4.84 | 1.70 | 0.148 | 1.21 |
| 60.00 | 4.65 | 1.103 | 14.57 | 3.22 | 0.529 | 5.72 | 1.86 | 0.176 | 1.43 |
| 65.00 | 5.04 | 1.294 | 17.02 | 3.49 | 0.621 | 6.67 | 2.01 | 0.207 | 1.67 |
| 70.00 | 5.42 | 1.501 | 19.67 | 3.76 | 0.720 | 7.70 | 2.17 | 0.240 | 1.92 |
| 75.00 | 5.81 | 1.723 | 22.50 | 4.02 | 0.826 | 8.80 | 2.32 | 0.275 | 2.19 |
| 80.00 | | | | 4.29 | 0.940 | 9.97 | 2.48 | 0.313 | 2.48 |
| 90.00 | | | | 4.83 | 1.190 | 12.54 | 2.79 | 0.397 | 3.11 |
| 100.00 | | | | 5.37 | 1.469 | 15.40 | 3.10 | 0.490 | 3.81 |
| 110.00 | | | | 5.90 | 1.777 | 18.54 | 3.41 | 0.593 | 4.57 |
| 120.00 | | | | | | | 3.72 | 0.705 | 5.41 |
| 130.00 | | | | | | | 4.03 | 0.828 | 6.32 |
| 140.00 | | | | | | | 4.34 | 0.960 | 7.30 |
| 160.00 | | | | | | | 4.96 | 1.254 | 9.46 |
| 180.00 | | | | | | | 5.58 | 1.587 | 11.90 |
| 200.00 | | | | | | | 6.20 | 1.959 | 14.62 |

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APMA Commercial Steel Pipe TABLE A5
Schedule 40 Based on ANSI B36: 10M–1985

| $k$ = 0.05 | 250 Nom. Bore 254.5 ID, $C$ = 141.9 | | | 300 Nom. Bore 303.2 ID, $C$ = 142.1 | | |
|---|---|---|---|---|---|---|
| FLOW RATE L/s | Velocity m/s | Velocity Head m | Head Loss m/100m | Velocity m/s | Velocity Head m | Head Loss m/100m |
| 15.00 | 0.29 | 0.004 | 0.04 | | | |
| 16.00 | 0.31 | 0.005 | 0.04 | | | |
| 18.00 | 0.35 | 0.006 | 0.05 | 0.25 | 0.003 | 0.02 |
| 20.00 | 0.39 | 0.008 | 0.06 | 0.28 | 0.004 | 0.03 |
| 22.00 | 0.43 | 0.010 | 0.07 | 0.30 | 0.005 | 0.03 |
| 24.00 | 0.47 | 0.011 | 0.08 | 0.33 | 0.006 | 0.04 |
| 26.00 | 0.51 | 0.013 | 0.10 | 0.36 | 0.007 | 0.04 |
| 28.00 | 0.55 | 0.015 | 0.11 | 0.39 | 0.008 | 0.05 |
| 30.00 | 0.59 | 0.018 | 0.13 | 0.42 | 0.009 | 0.05 |
| 35.00 | 0.69 | 0.024 | 0.17 | 0.48 | 0.012 | 0.07 |
| 40.00 | 0.79 | 0.032 | 0.22 | 0.55 | 0.016 | 0.09 |
| 45.00 | 0.88 | 0.040 | 0.27 | 0.62 | 0.020 | 0.11 |
| 50.00 | 0.98 | 0.049 | 0.33 | 0.69 | 0.024 | 0.14 |
| 55.00 | 1.08 | 0.060 | 0.39 | 0.76 | 0.030 | 0.16 |
| 60.00 | 1.18 | 0.071 | 0.46 | 0.83 | 0.035 | 0.19 |
| 65.00 | 1.28 | 0.083 | 0.53 | 0.90 | 0.041 | 0.22 |
| 70.00 | 1.38 | 0.097 | 0.61 | 0.97 | 0.048 | 0.26 |
| 75.00 | 1.47 | 0.111 | 0.70 | 1.04 | 0.055 | 0.29 |
| 80.00 | 1.57 | 0.126 | 0.79 | 1.11 | 0.063 | 0.33 |
| 90.00 | 1.77 | 0.160 | 0.99 | 1.25 | 0.079 | 0.41 |
| 100.00 | 1.97 | 0.197 | 1.21 | 1.38 | 0.098 | 0.50 |
| 110.00 | 2.16 | 0.239 | 1.45 | 1.52 | 0.118 | 0.60 |
| 120.00 | 2.36 | 0.284 | 1.71 | 1.66 | 0.141 | 0.71 |
| 130.00 | 2.56 | 0.333 | 2.00 | 1.80 | 0.165 | 0.83 |
| 140.00 | 2.75 | 0.386 | 2.30 | 1.94 | 0.192 | 0.95 |
| 160.00 | 3.14 | 0.505 | 2.98 | 2.22 | 0.250 | 1.23 |
| 180.00 | 3.54 | 0.639 | 3.74 | 2.49 | 0.317 | 1.54 |
| 200.00 | 3.93 | 0.789 | 4.58 | 2.77 | 0.391 | 1.89 |
| 220.00 | 4.32 | 0.954 | 5.51 | 3.05 | 0.474 | 2.27 |
| 240.00 | 4.72 | 1.135 | 6.53 | 3.32 | 0.564 | 2.68 |
| 250.00 | 4.91 | 1.232 | 7.07 | 3.46 | 0.611 | 2.90 |
| 260.00 | 5.11 | 1.333 | 7.63 | 3.60 | 0.661 | 3.13 |
| 280.00 | 5.50 | 1.545 | 8.82 | 3.88 | 0.767 | 3.61 |
| 300.00 | 5.90 | 1.774 | 10.09 | 4.15 | 0.880 | 4.13 |
| 325.00 | 6.39 | 2.082 | 11.80 | 4.50 | 1.033 | 4.83 |
| 350.00 | | | | 4.85 | 1.198 | 5.57 |
| 375.00 | | | | 5.19 | 1.376 | 6.38 |
| 400.00 | | | | 5.54 | 1.565 | 7.23 |
| 425.00 | | | | 5.89 | 1.767 | 8.14 |
| 450.00 | | | | 6.23 | 1.981 | 9.10 |
| 475.00 | | | | 6.58 | 2.207 | 10.12 |
| 500.00 | | | | 6.92 | 2.446 | 11.19 |

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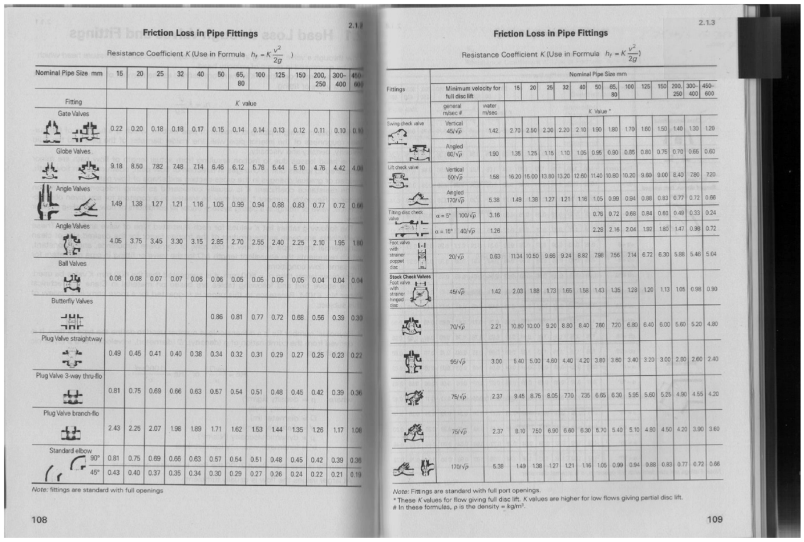

2.1.2

**Friction Loss in Pipe Fittings**

Resistance Coefficient $K$ (Use in Formula $h_f = K\frac{v^2}{2g}$ )

| Nominal Pipe Size mm | 15 | 20 | 25 | 32 | 40 | 50 | 65, 80 | 100 | 125 | 150 | 200, 250 | 300–400 | 450–600 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Fitting | $K$ value | | | | | | | | | | | | |
| Gate Valves | 0.22 | 0.20 | 0.18 | 0.18 | 0.17 | 0.15 | 0.14 | 0.14 | 0.13 | 0.12 | 0.11 | 0.10 | 0.10 |
| Globe Valves | 9.18 | 8.50 | 7.82 | 7.48 | 7.14 | 6.46 | 6.12 | 5.78 | 5.44 | 5.10 | 4.76 | 4.42 | 4.08 |
| Angle Valves | 1.49 | 1.38 | 1.27 | 1.21 | 1.16 | 1.05 | 0.99 | 0.94 | 0.88 | 0.83 | 0.77 | 0.72 | 0.66 |
| Angle Valves | 4.05 | 3.75 | 3.45 | 3.30 | 3.15 | 2.85 | 2.70 | 2.55 | 2.40 | 2.25 | 2.10 | 1.95 | 1.80 |
| Ball Valves | 0.08 | 0.08 | 0.07 | 0.07 | 0.06 | 0.06 | 0.05 | 0.05 | 0.05 | 0.05 | 0.04 | 0.04 | 0.04 |
| Butterfly Valves | | | | | | 0.86 | 0.81 | 0.77 | 0.72 | 0.68 | 0.56 | 0.39 | 0.30 |
| Plug Valve straightway | 0.49 | 0.45 | 0.41 | 0.40 | 0.38 | 0.34 | 0.32 | 0.31 | 0.29 | 0.27 | 0.25 | 0.23 | 0.22 |
| Plug Valve 3-way thru-flo | 0.81 | 0.75 | 0.69 | 0.66 | 0.63 | 0.57 | 0.54 | 0.51 | 0.48 | 0.45 | 0.42 | 0.39 | 0.36 |
| Plug Valve branch-flo | 2.43 | 2.25 | 2.07 | 1.98 | 1.89 | 1.71 | 1.62 | 1.53 | 1.44 | 1.35 | 1.26 | 1.17 | 1.08 |
| Standard elbow 90° | 0.81 | 0.75 | 0.69 | 0.66 | 0.63 | 0.57 | 0.54 | 0.51 | 0.48 | 0.45 | 0.42 | 0.39 | 0.36 |
| Standard elbow 45° | 0.43 | 0.40 | 0.37 | 0.35 | 0.34 | 0.30 | 0.29 | 0.27 | 0.26 | 0.24 | 0.22 | 0.21 | 0.19 |

*Note:* fittings are standard with full openings

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2.1.3

**Friction Loss in Pipe Fittings**

Resistance Coefficient $K$ (Use in Formula $h_f = K\frac{v^2}{2g}$)

| Fittings | Minimum velocity for full disc lift: general m/sec # | water m/sec | 15 | 20 | 25 | 32 | 40 | 50 | 65, 80 | 100 | 125 | 150 | 200, 250 | 300–400 | 450–600 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | $K$ Value * | | | | | | | | | | | | |
| Swing check valve, Vertical | $45/\sqrt{\rho}$ | 1.42 | 2.70 | 2.50 | 2.30 | 2.20 | 2.10 | 1.90 | 1.80 | 1.70 | 1.60 | 1.50 | 1.40 | 1.30 | 1.20 |
| Swing check valve, Angled | $60/\sqrt{\rho}$ | 1.90 | 1.35 | 1.25 | 1.15 | 1.10 | 1.05 | 0.95 | 0.90 | 0.85 | 0.80 | 0.75 | 0.70 | 0.65 | 0.60 |
| Lift check valve, Vertical | $50/\sqrt{\rho}$ | 1.58 | 16.20 | 15.00 | 13.80 | 13.20 | 12.60 | 11.40 | 10.80 | 10.20 | 9.60 | 9.00 | 8.40 | 7.80 | 7.20 |
| Lift check valve, Angled | $170/\sqrt{\rho}$ | 5.38 | 1.49 | 1.38 | 1.27 | 1.21 | 1.16 | 1.05 | 0.99 | 0.94 | 0.88 | 0.83 | 0.77 | 0.72 | 0.66 |
| Tilting disc check valve, $\alpha = 5°$ | $100/\sqrt{\rho}$ | 3.16 | | | | | | 0.76 | 0.72 | 0.68 | 0.84 | 0.60 | 0.49 | 0.33 | 0.24 |
| Tilting disc check valve, $\alpha = 15°$ | $40/\sqrt{\rho}$ | 1.26 | | | | | | 2.28 | 2.16 | 2.04 | 1.92 | 1.80 | 1.47 | 0.98 | 0.72 |
| Foot valve with strainer poppet disc | $20/\sqrt{\rho}$ | 0.63 | 11.34 | 10.50 | 9.66 | 9.24 | 8.82 | 7.98 | 7.56 | 7.14 | 6.72 | 6.30 | 5.88 | 5.46 | 5.04 |
| Stock Check Valves: Foot valve with strainer hinged disc | $45/\sqrt{\rho}$ | 1.42 | 2.03 | 1.88 | 1.73 | 1.65 | 1.58 | 1.43 | 1.35 | 1.28 | 1.20 | 1.13 | 1.05 | 0.98 | 0.90 |
| | $70/\sqrt{\rho}$ | 2.21 | 10.80 | 10.00 | 9.20 | 8.80 | 8.40 | 7.60 | 7.20 | 6.80 | 6.40 | 6.00 | 5.60 | 5.20 | 4.80 |
| | $95/\sqrt{\rho}$ | 3.00 | 5.40 | 5.00 | 4.60 | 4.40 | 4.20 | 3.80 | 3.60 | 3.40 | 3.20 | 3.00 | 2.80 | 2.60 | 2.40 |
| | $75/\sqrt{\rho}$ | 2.37 | 9.45 | 8.75 | 8.05 | 7.70 | 7.35 | 6.65 | 6.30 | 5.95 | 5.60 | 5.25 | 4.90 | 4.55 | 4.20 |
| | $75/\sqrt{\rho}$ | 2.37 | 8.10 | 7.50 | 6.90 | 6.60 | 6.30 | 5.70 | 5.40 | 5.10 | 4.80 | 4.50 | 4.20 | 3.90 | 3.60 |
| | $170/\sqrt{\rho}$ | 5.38 | 1.49 | 1.38 | 1.27 | 1.21 | 1.16 | 1.05 | 0.99 | 0.94 | 0.88 | 0.83 | 0.77 | 0.72 | 0.66 |

*Note:* Fittings are standard with full port openings.
* These $K$ values for flow giving full disc lift. $K$ values are higher for low flows giving partial disc lift.
# In these formulas, $\rho$ is the density = kg/m³.

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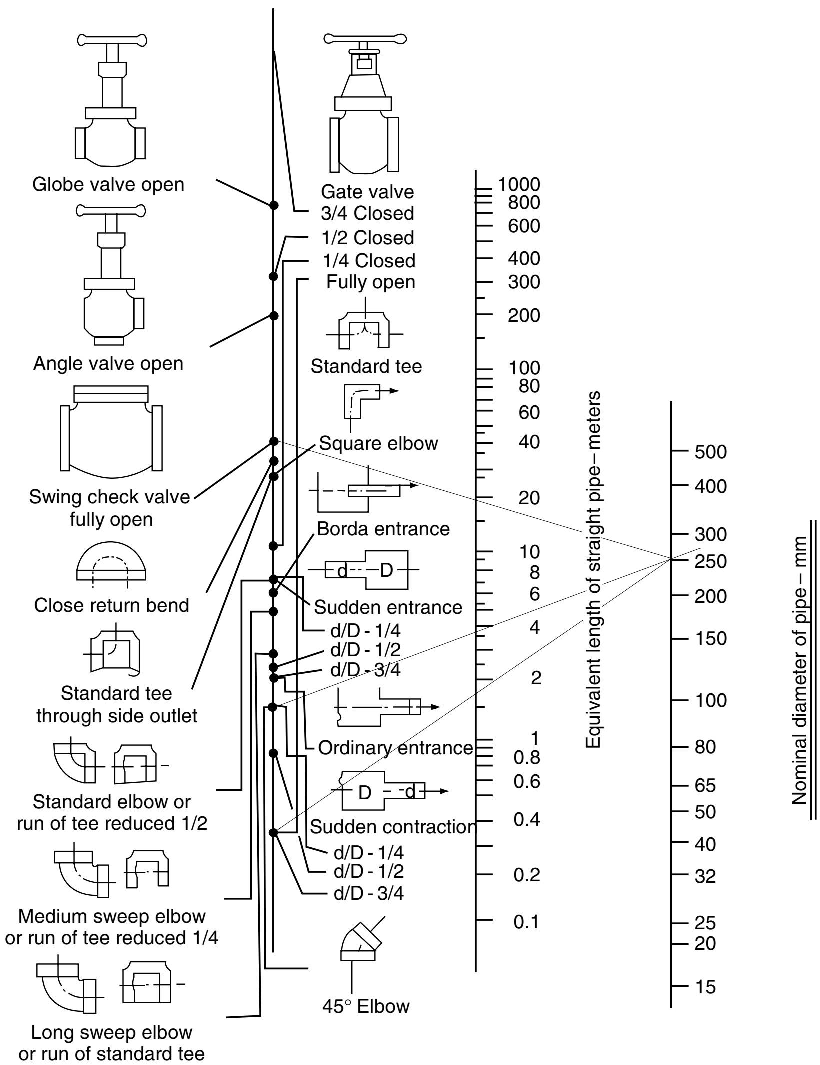
## Equivalent length – valves & fittings



### A.1.6 Tutorial 6

1. Calculate the reduced impeller diameter for the pump that will put out 140 l/s. Also
   (a) Calculate the energy cost saving per annum in trimming the impeller instead of throttling the discharge valve to achieve the same result. Cost of 1 unit of electricity (1 kW h) = $0.11 with the pump running 24 h/day and
   (b) How else may the pump output be controlled?
   (c) Which option is most efficient and/or offers more flexibility?

### A.1.7 Tutorial 7

Refer to the values for Pump test readings set out in the table below. Utilising the Equation provided in AS 2417.3, verify mathematically at which points the pump has met it's guaranteed performance in line with the ellipse analysis or value being = or »1 rule. Accordingly, state in each case if the Pump guarantee parameters have been met.

*Given*: Manufacturers Guaranteed duty point for the Pump is 140 l/s @48 head. AS 2417.3 stipulates acceptable deviation in terms of $H_G$ as (±2.0%) and (±4.0%) for $Q_G$

**Ellipse analysis – calculations**

| Value $\Delta H$ (m) | Value $\Delta Q$ (l/s) | $[H_G \times X_H / \Delta H]^2 +$ $[Q_G \times X_Q / \Delta Q]^2$ >>= 1 | Pump $H_G$ Guarantee (m) | Pump $Q_G$ Guarantee (l/s) | Deviation $X_H = 2.0\%$ | Deviation $X_Q = 4.0\%$ |
|---|---|---|---|---|---|---|
| | | **Design point** | **51.5** | **140** | 0.02 | 0.04 |
| 3.9 | 16.0 | | 51.5 | 140 | 0.02 | 0.04 |
| 3.0 | 14.0 | | 51.5 | 140 | 0.02 | 0.04 |
| 2.0 | 11.0 | | 51.5 | 140 | 0.02 | 0.04 |
| 1.0 | 9.0 | | 51.5 | 140 | 0.02 | 0.04 |
| 0.5 | 7.0 | | 51.5 | 140 | 0.02 | 0.04 |
| –1.0 | 5.0 | | 51.5 | 140 | 0.02 | 0.04 |
| –2.0 | –5.0 | | 51.5 | 140 | 0.02 | 0.04 |
| –3.0 | –8.0 | | 51.5 | 140 | 0.02 | 0.04 |
| –4.0 | –10.0 | | 51.5 | 140 | 0.02 | 0.04 |
| 1.0 | –12.0 | | 51.5 | 140 | 0.02 | 0.04 |
| –1.0 | –15.0 | | 51.5 | 140 | 0.02 | 0.04 |
| 5.0 | 5.5 | | 51.5 | 140 | 0.02 | 0.04 |
| 10.0 | 5.6 | | 51.5 | 140 | 0.02 | 0.04 |
| 1.03 | 5.6 | | 51.5 | 140 | 0.02 | 0.04 |

Testing for various values of $\Delta H$ (m) and $\Delta Q$ (m) eg.
Line 1: 3.9 = 55.4 – 51.5 and 16 = 156 – 140, etc.

### A.1.8 Tutorial 8

If a pump's speed is 2450 rpm and duty is 200 l/s @ 60 m head whilst drawing 120 kW, what would be the pump duty and duty power if the pump speed is increased to 2950?

*Given*: $Q_1 = 200$ l/s, $N_1 = 2450$ rpm, $kW_1 = 120$ kW, $H_1 = 60$ m and $N_2 = 2950$ rpm.

## A.2 Miscellaneous tutorials

### A.2.1 Centrifugal pumps

#### Work group 'problem-solving' exercise

This tutorial session has been designed to go over a range of fault situations that could be encountered with centrifugal pump operations.

The class will be divided into groups and the presenter will ask each group to work through a couple of fault scenarios. The group is required to list all possible causes that could lead to the fault occurring and appoint a spokes person to present the list prepared when asked to do so by the presenter.

The list of topics are:

1. Pump does not develop any head pressure nor does it deliver any liquid.
2. Pump delivers no liquid but develops some pressure.
3. Pump output is below the manufacturer's provided performance data as in the associated performance curve provided with the pump.
4. Pump draws higher amps than specified.
5. Pump performance is compromised, although nothing appears to be wrong with pumping system.
6. Pump operates satisfactorily during start up, however, performance deteriorates in a relatively short time.
7. Pump operates with noise or vibrations, or both.
8. Stuffing box leaks abnormally.
9. Gland packing has short working life.
10. Mechanical seal fails prematurely.
11. Bearings fail prematurely.
12. Bearings run noisily, i.e. steady high-pitched or intermittent/continuous low-pitched noise or intermittent rumble/rattle and/or clicks, Intermittent squeal or high-pitch tone.
13. Pump overheats or seizes, or both.
14. Pump cavitates when the NPSH-a is increased.
15. List all the 'dos' that come to mind in regard to *safe* operation of pumps.
16. List all the 'don'ts' that come to mind in regard to *safe* operation of pumps.

| Q1. | Pump does not develop any head pressure nor does it deliver liquid |
|---|---|
| A1. | • |
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| Q2. | Pump delivers no liquid but develops some pressure |
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| A2. | • |
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| **Q3.** | **Pump output is below the manufacturer's provided performance data as in the associated performance curve provided with the pump** |
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| A3. | • |
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| **Q4.** | **Pump draws higher amps than specified** |
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| A4. | • |
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| **Q5.** | **Pump performance compromised, although nothing appears to be wrong with pumping system** |
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| A5. | |
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| **Q6.** | **Pump operates satisfactorily during start up, but performance deteriorates in a relatively short time** |
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| A6. | • |
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| **Q7**. | **Pump operates with noise or vibrations, or both** |
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| A7. | • |
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| **Q8.** | **Stuffing box leaks abnormally** |
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| A8. | • |
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| **Q9**. | **Gland packing has short working life** |
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| A9. | • |
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| **Q10.** | **Mechanical seal fails prematurely** |
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| A10. | • |
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| **Q11.** | **Bearings fail prematurely** |
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| A11. | • |
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| **Q12.** | **Bearings run noisily i.e. steady high-pitched or intermittent/continuous low-pitched noise or intermittent rumble/rattle and or clicks, Intermittent squeal or high-pitch tone** |
|---|---|
| A12. | *A. Steady high-pitched tone* |
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| | *B. Continuous or intermittent low-pitch tone* |
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| | *C. Intermittent rattles, rumbles, and/or clicks* |
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| | *D. Intermittent squeal or high-pitch tone* |
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| **Q13.** | **Pump overheats or seizes, or both** |
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| A13. | • |
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| **Q14.** | **Pump cavitates when the NPSH-a is increased** |
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| A14. | • |
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| **Q15.** | **List all the 'dos' that come to mind in regard to *safe* operation of pumps**<br>**---------- Dos ---------** |
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| A15. | ☐ |

| **Q16.** | **List all the 'don'ts' that come to mind in regard to *safe* operation of pumps**<br>**-----Don'ts -----** |
|---|---|
| A16. | ☐ |

# A.3 Tutorials (Answers)

## A.3.1 Tutorial 1: Answers

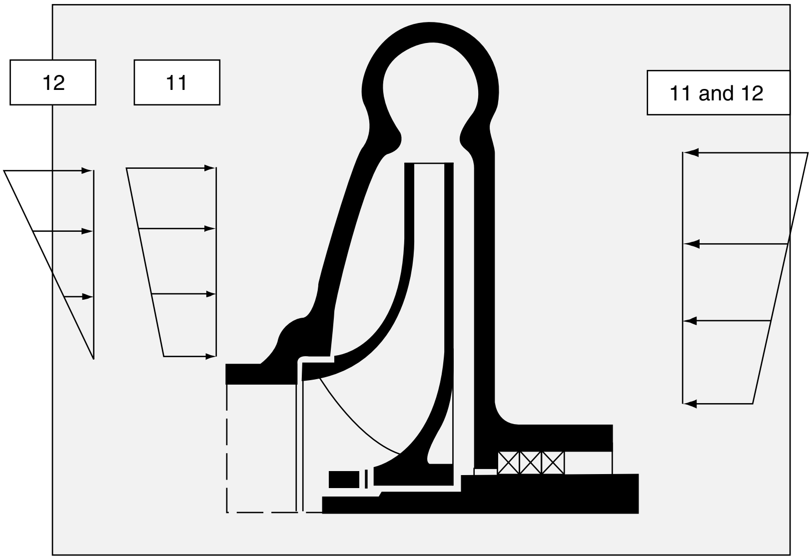
1. What three classifications can be given to the properties of a good pump.
   (a) Safety
   (b) Reliability
   (c) Efficiency.
2. List at least eight issues to be considered when evaluating risks of a new engineering project.
   (a) Continuity of supplies
   (b) Economic
   (c) Environmental
   (d) Financial
   (e) Human error or behavior
   (f) Natural hazards
   (g) Occupational safety and health
   (h) Product liability
   (i) Professional liability
   (j) Property damage
   (k) Public liability
   (l) Security.
3. Why is risk assessment an essential part of any task?
   (a) If being a good corporate citizen isn't enough, *duty of care* imposed by
   (b) legislation and the penalties, personal and corporate, for breaches of the
   (c) legislation make careful risk management an essential part of any activity.
4. List the main components of life cycle cost of any engineering project.
   (a) Design costs
   (b) Design development costs
   (c) Capital costs
   (d) Running/operational costs
   (e) Maintenance costs
   (f) Replacement costs
   (g) Process improvement costs
   (h) Demolition costs
   (i) Environmental costs.
5. What would be the appropriate measures for comparative assessment of life cycle cost of alternative pumping plants?

   *Total costs per unit of productive output.*
6. List some areas where statutory requirements apply to pump installations.
   (a) Occupational safety and health regulations – *Duty of care perspective*
   (b) Fire protection systems
   (c) Pumping cryogenic liquids
   (d) Pumping flammable liquids
   (e) Pumping liquids at high temperature
   (f) Pumping in hazardous atmospheres.

## A.3.2 Tutorial 2: Answers

| Question Numbers Refer to Numbers on the Illustrations on the Next Page | Enter Your Answer |
|---|---|
| 1 | *List the functions of the pump casing.*<br>Pressure containment<br>Fluid containment<br>Attachment to the fluid transmission system<br>Structural support<br>Conversion of kinetic energy to pressure |
| 2 | *Name the element and describe its purpose.*<br>Volute<br>Channels fluid flow to pump outlet at constant velocity and converts kinetic energy to pressure |
| 3 | *Name the element and describe its purpose.*<br>Wear ring<br>Reduces flow recirculation and hence improves efficiency. Provides a cheap maintenance alternative to casing repair or replacement |
| 4 | *Name the element and describe its purpose.*<br>Guide vanes<br>Controls fluid pre-rotation |
| 5 | *Name the element.*<br>Throat |
| 6 | *Name the element.*<br>Tongue |
| 7 | *Name the impeller type.*<br>Radial |
| 8 | *Name the impeller type.*<br>Mixed flow |
| 9 | *Name the impeller type.*<br>Axial |
| 10A | *List the relative advantages/disadvantages of the two impeller layouts illustrated.*<br>The back to back layout is self-balancing for axial hydraulic forces.<br>The back to back design requires more complex and therefore more expensive casing castings. |
| 10B | *Why are multistage pumps required?*<br>To produce higher discharge pressures than can be economically achieved with single-stage designs. |

11. Sketch the distribution and direction of hydraulic axial forces applied to the illustrated pump impeller during pump operation.




12. How would the force distribution be changed if the impeller was of the open type? (Sketch the pressure distribution)

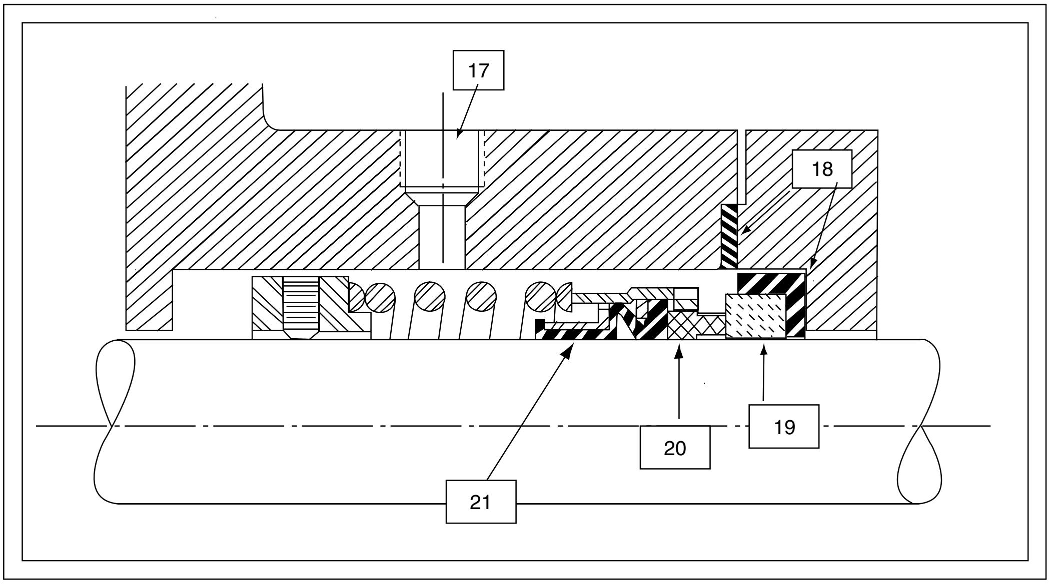
13. List possible sources of axial forces acting on the pump?

    Pressure forces acting on the impeller shroud(s).
    Momentum forces due to differences in axial components of fluid velocity at impeller inlet and outlet.
    Differences in static pressure in front of and behind shaft seals.
    Mechanical forces from the driver.
    Forces due to the weight of pump components (on non-horizontal pump designs).
    Transitionary forces during startup conditions.

14. At which operating point are radial forces in a centrifugal pump a minimum and a maximum?

    Minimum at best efficiency point.
    Maximum at flow shut-off.

| Question Numbers Refer to Numbers on the Illustrations Below | Enter Your Answer |
|---|---|
| 15 | *List the functions of the shaft seals.*<br>• To prevent leakage of the pumped fluids between the pump shaft and casing<br>• To contain the lubricant (grease or oil) with in the bearing housing<br>To prevent air ingress into the pump casing in certain application |
| 16 | *Name the main types of shaft seals in use.*<br>• Packed stuffing box – commonly encountered – low cost<br>• Lip seals – widely used – for oil / grease sealing – low pressure<br>Mechanical seal – widely used – high performance – improved reliability – trouble-free operation |
| 17 | *Name the element and describe its purpose.*<br>Sealing liquid connection – offers cooling, lubrication, and flushing action in certain applications. Liquid source could be external or a bleed off from the volute casing |
| 18 | *Name the element and describe its purpose.*<br>Elastometric-moulded rubber seal to prevent fluid leaking past the stationary element |
| 19 | *Name the element and describe its purpose.*<br>Stationary seal element provides an effective seal with the rotating element. Usually of a softer material compared with the rotating element |
| 20 | *Name the element and describe its purpose.*<br>Rotating element provides an effective seal with the stationary element. Usually of a harder material compared with the stationary element |
| 21 | *Name the element and describe its purpose.*<br>Elastometric-moulded nitrile rubber bellows seals in the axial and radial planes |



## A.3.3 Tutorial 3: Answers

1. Correct the following incorrect pump laws:

   For similar conditions of flow (i.e. the same efficiency):

   – *Capacity* is directly proportional to speed and or impeller diameter.
   – *Head* is directly proportional to the *square* of speed and or impeller diameter.
   – *Power* is directly proportional to the *cube* of speed and or impeller diameter.

2. Express the corrected version of each law as a formula.

$$Q_2 = Q_1 \times N_2 \div N_1$$
$$H_2 = H_1 \times (N_2 \div N_1)^2$$
$$P_2 = P_1 \times (N_2 \div N_1)^3$$

3. Define Net Positive Suction Head.

   The difference between absolute head available at the pump inlet and the head corresponding to the vapor pressure of the pumped fluid.

4. What is the difference between NPSH-a and NPSH-r?

   NPSH-r is the minimum value of NPSH that a pump requires for satisfactory operation at a given speed.

   NPSH-a is the value of NPSH after deducting the sum in meters head of the – suction lift + the frictional resistance of pipe and fittings (due to flow) + the vapor pressure from 10.35 m which is the 'free lift' provided by the atmosphere. For satisfactory operation of the pump, the value of NPSH-a must be equal to and preferably greater than NPSH-r stated by the pump manufacturer.

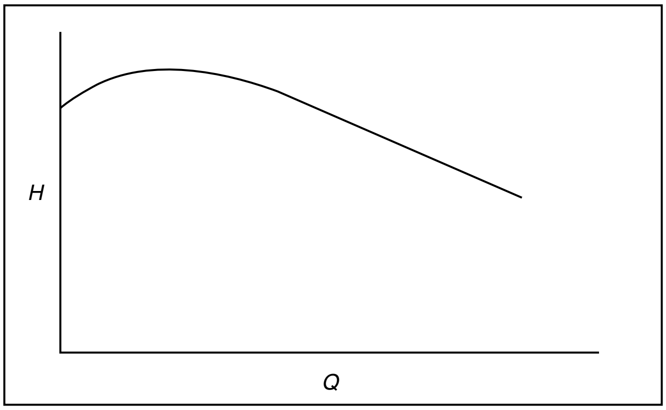
5. Calculate NPSH-a for the following example:

$$\begin{aligned}\text{NPSH-a} &= 1 \text{ atm } (H_a) - H_f - H_{vap}\\ &= (101.325 \times 1000 / 995.7 \times 9.81) - (0.1 + 0.4 + 2.6 + 1.0 + 2.0) - H_{vap}\\ &= 10.373 - 6.1 - (4.241 \times 1000 / 9.81 \times 995.7)\\ &= 10.37 - 6.1 - 0.434 = 3.84 \text{ m}\end{aligned}$$

6. List ways of avoiding cavitation.

   - Establish accurate range of flow for pump.
   - If running speed higher than pump curve data multiply NPSH-r at curve speed by speed ratio ($N_2 = N_1 \times Q_2 / Q_1$).
   - Avoid suction-specific speeds of over 6750 (metric) 11 000 (USA – units).
   - Where able to, select a pump with rated flow within 10% of best efficiency point (BEP).
   - Select for NPSH-a 10% or 0.5–1.0 m above NPSH-r.
   - Require witnessed pump tests over flow range for critical service applications.
   - Don't accept model tests as a basis for larger pumps.

7. Sketch a drooping pump curve.




8. What are the relative merits of steep and flat pump curves.

*Steep*: Relatively constant flow at small variations in head;
Usually has an non-overloading power characteristic.
*Flat*: Relatively constant head with variations of flow.

9. Additional NPSH-a calculations

*Ans 9.1 Suction head/open tank – Answer*

$$\text{Total Suction head } H_{\text{st}} = H_{\text{a}} + H_{\text{s}} - \text{friction losses } H_{\text{f}}$$

$$H_{\text{st}} = \frac{101.325 \times 1000}{998.3 \times 9.81} + 4.5 - (1.1 + 0.1)$$

$$= 10.35 + 4.5 - 1.2 = 13.65 \text{ m}$$

$$H_{\text{vap}} = \frac{2.337 \times 1000}{998.3 \times 9.81} = 0.239 \text{ m}$$

$$\text{NPSH-a} = H_{\text{st}} - H_{\text{vap}}$$

$$\text{NPSH-a} = 13.65 - 0.24 = 13.4 \text{ m}$$

*Ans 9.2 Suction head/closed tank*

$$H_{\text{st}} = H_{\text{a}} + H_{\text{s}} - H_{\text{f}}$$

$$= \frac{(101.325 + 130) \times 1000}{9.81 \times 934.6} + 4 - (1.1 + 0.1)$$

$$= 25.24 + 4 - 1.2 = 28.04 \text{ m}$$

$$H_{\text{vap}} = \frac{270.13 \times 1000}{9.81 \times 934.6} = 29.47 \text{ m}$$

$$\text{NPSH-a} = 28.04 - 29.47 = -1.43 \text{ m}$$

*(as* NPSH-*a is negative, the pump would cavitate under these operating conditions)*

*Ans 9.3 Suction head/closed tank*

$$H_{st} = H_a + H_{gage} + H_s - H_f$$

$$= \frac{(101.325 + 110) \times 1000}{9.81 \times 952.2} + 4 - (1.1 + 0.1)$$

$$= 22.63 + 4 - 1.2 = 25.43\,\text{m}$$

$$H_{vap} = \frac{133.90 \times 1000}{9.81 \times 952.2} = 14.34$$

$$\text{NPSH-a} = H_{st} - H_{vap} = 25.43 - 14.34 = 11.10\,\text{m}$$

*Ans 9.4 Suction head/suction gage reading*

$$\text{Velocity } V = \frac{\text{Flow}}{\text{Area}} = \frac{12 \times 4 \times 10^{-3}}{3.14 \times 0.0582^2} = 4.513\,\text{m/s}$$

$$\text{Velocity head} = \frac{v^2}{2g} = \frac{(4.513)^2}{2 \times 9.81} = 1.038\,\text{m}$$

$$H_{st} = \text{Gage reading + gage ht + atmospheric pr + velocity head}$$

$$= \frac{24 \times 1000}{9.81 \times 988} + 0.8 + \frac{101.325 \times 1000}{9.81 \times 988} + 1.038$$

$$= 2.48 + 0.8 + 10.4593 + 1.038 = 14.77\text{ m}$$

$$H_{vap} = \frac{12.34 \times 1000}{9.81 \times 988} = 1.27\text{ m}$$

$$\text{NPSH-a} = 14.77 - 1.27 = 13.50\text{ m}$$

*Ans 9.5 Suction head/saturated (boiling) liquid*

$$\text{NPSH-a} = H_s - H_f$$

$$= 7 - (0.1 + 0.9)$$

$$= 7 - 1$$

$$= 6.0\,\text{m}$$

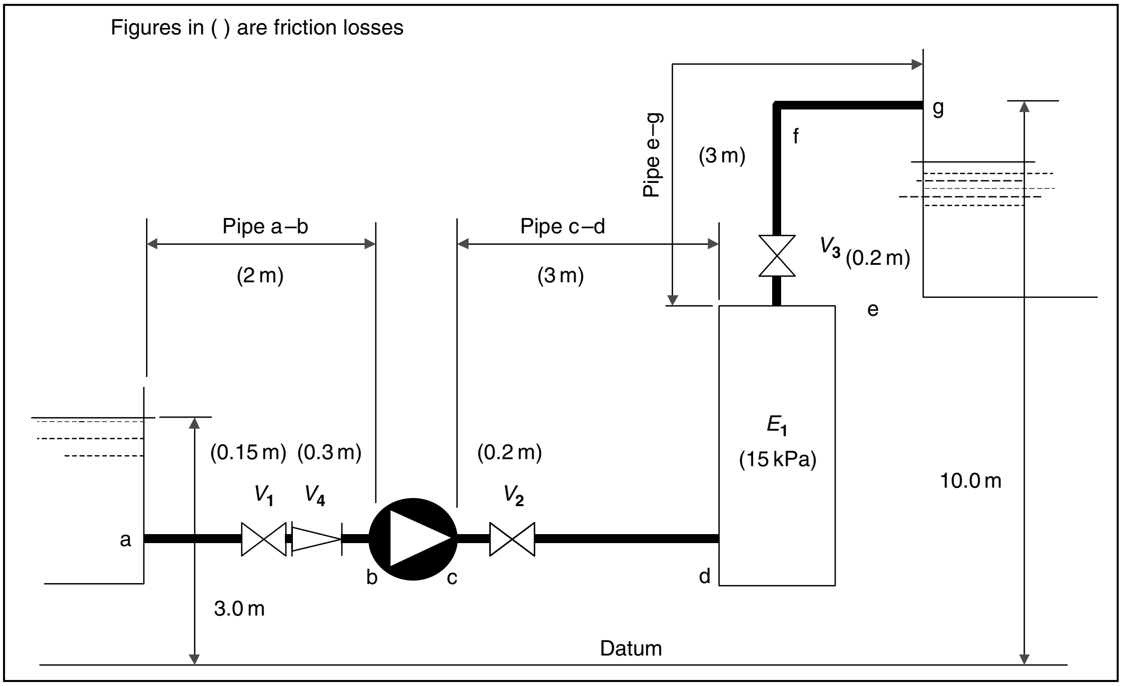
10. Calculate the total system head for the hydraulic system illustrated. Assume density of water is 1000 kg/m$^3$ and entry loss at '*a*' = 0.5 m and sudden expansion loss at '*g*' = 0.3 m and 0.4 for bend '*f*'.

    The total system head ($H_t$) = $H_{ts}$ + $H_f$

    The static head on the system $H_{ts}$ = 10 m – 3 m = 7 m head and

    The total frictional head for the system $H_f$ = the sum of all the frictional losses indicated = (0.5 + 0.15 + 0.3 + 2.0 + 0.2 + 3.0 + (15 / 9.81) + 0.2 + 0.4 + 3.0 + 0.3) = 11.58 m

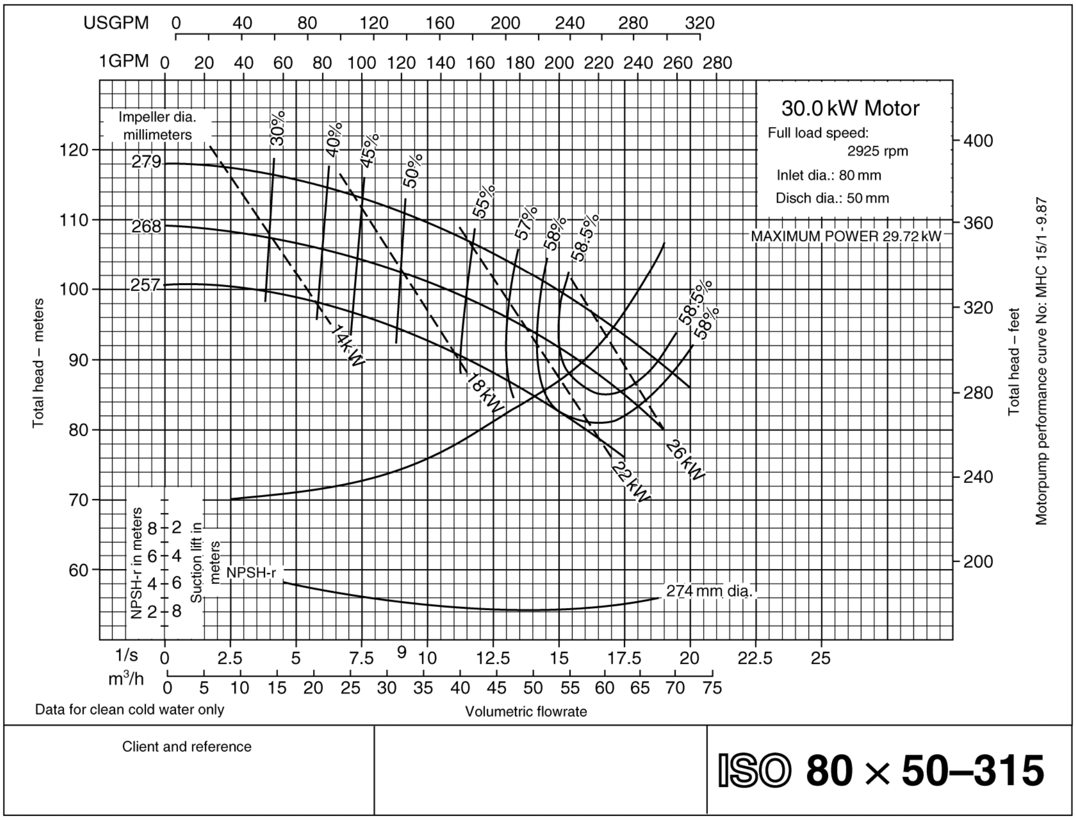
    Therefore, the Total system head ($H_t$) = 7 + 11.58 = 18.58 m.




11. Plot a system resistance curve, on the attached pump performance chart, for a system with the following parameters:

    Impeller diameter = 268 mm, Operating point; $Q$ = 15.75 l/s $H_t$ = 90 m Static head = 70 m, Fluid pumped is water of density = 1000 kgs/m$^3$. The plot should show the resistance curve from zero to 17.5 l/s flow.
    (Supply Grundfos 80 × 50–315 pump curves.)

| **Plotting System Resistance Curve – Impeller Diameter 268 mm** | | | | |
|---|---|---|---|---|
| **Liquid = Water** | | | | |
| **Flow (l/s)** | **$H_t$ (m [Total Head])** | **$H_{st}$ (m [Static Head])** | **$H_f = k \times Q^2$ (m [Friction Head])** | **k (Loss Coeff.) = $H_f / Q^2$** |
| 2.5 | 70.47 | 70 | 0.47 | 0.07574 |
| 5.0 | 71.89 | 70 | 1.89 | 0.07574 |
| 7.0 | 73.71 | 70 | 3.71 | 0.07574 |
| 9.0 | 76.13 | 70 | 6.13 | 0.07574 |
| 11.0 | 79.16 | 70 | 9.16 | 0.07574 |
| 13.0 | 82.80 | 70 | 12.80 | 0.07574 |
| 15.0 | 87.04 | 70 | 17.04 | 0.07574 |
| 16.25 | 90.00 | 70 | 20.00 | 0.07574 |
| 17.0 | 91.89 | 70 | 21.89 | 0.07574 |




## A.3.4 Tutorial 4: Answers

(Following on from Tutorial 3)

1. Given that the pump with an impeller diameter of 268 mm is putting out 15.75 l/s @ 90 m head, calculate the pump impeller diameter to put out 14.5 l/s

   Using the KSB – 'Centrifugal Pump Design' recommended approximate formulae;

$$\frac{(D_2)^2}{(D_1)^2} \text{ approx} = \frac{Q_2}{Q_1} \text{ approx} = \frac{H_2}{H_1}$$

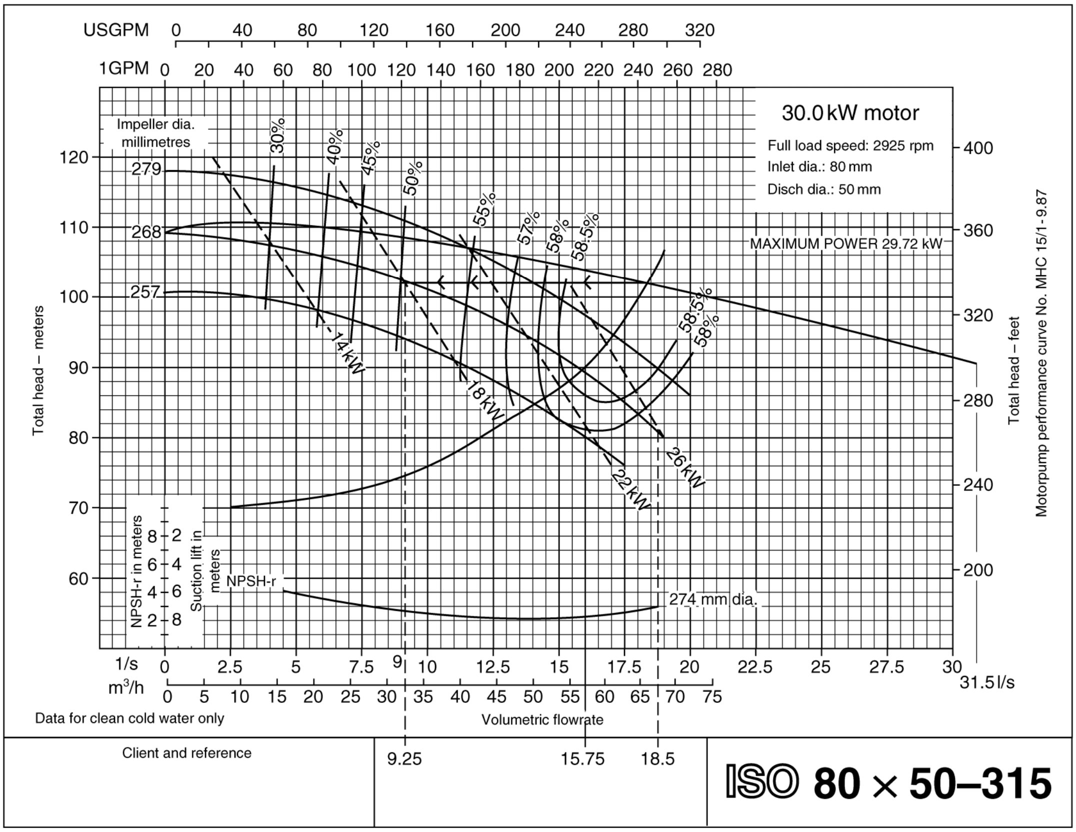
   We get:

$$D_2^{\,2} = Q_2 \div Q_1 \times D_1^{\,2} = 14.5 \div 15.75 \times 268^2 \quad \text{or}$$
$$D_2 = (0.9206)^{-2} \times 268 = 0.9595 \times 268 = 257.15\,\text{mm}$$

   *Note*: Trimming of the Impeller follows the Affinity laws within limits and with more tendency for inaccuracy with some pumps compared to others. Hence proceed with caution!

2. What would be the output if a second identical pump with impeller diameter 268 mm was run in parallel with the system pipe work remaining the unchanged? How could flow in parallel operations be improved?

   Refer to the pump performance chart below. In parallel operation, the pumps would try and put out twice the output of a single pump. However, with the increase in flow resistance, the actual output is restricted to 18.5 l/s instead of 31.5 l/s. To optimize flow conditions in parallel operation, it is important to size the system pipe work correctly to keep flow resistance to a minimum.




3. How would the head, of the pump at question 1 (impeller diameter of 268 mm delivering 15.75 l/s (or 56.7 m$^3$/h) @ 90 m head) vary if the kinematic viscosity of the liquid pumped was $1.14 \times 10^{-4}$ m$^2$/s and the specific gravity was 0.85? (Use viscosity charts provided, note the kinematic viscosity of water is $1 \times 10^{-6}$ m$^2$/s)

   Static head = 59.5 m (70 × 0.85)
   Total head = 85.5 m (read from the chart provided, i.e. $H_z = K_H \times H_t = 0.95 \times 90$)
   Total Flow = 15.59 l/s (read from chart provided, i.e. $Q_z = K_Q \times Q = 0.99 \times 15.75$)

   Pump efficiency would drop to = 46.02% (0.78 × 59% at operating point read off the chart)

4. List the issues that elevated fluid temperatures raise for pump selection and operation.

   - Changes to liquid vapor pressure
   - Changes to viscosity
   - Changes to mechanical strength and chemical resistance of pump materials
   - Increase in safety hazards of the pumping process
   - Introduction of expansion and contraction stresses to pump components.

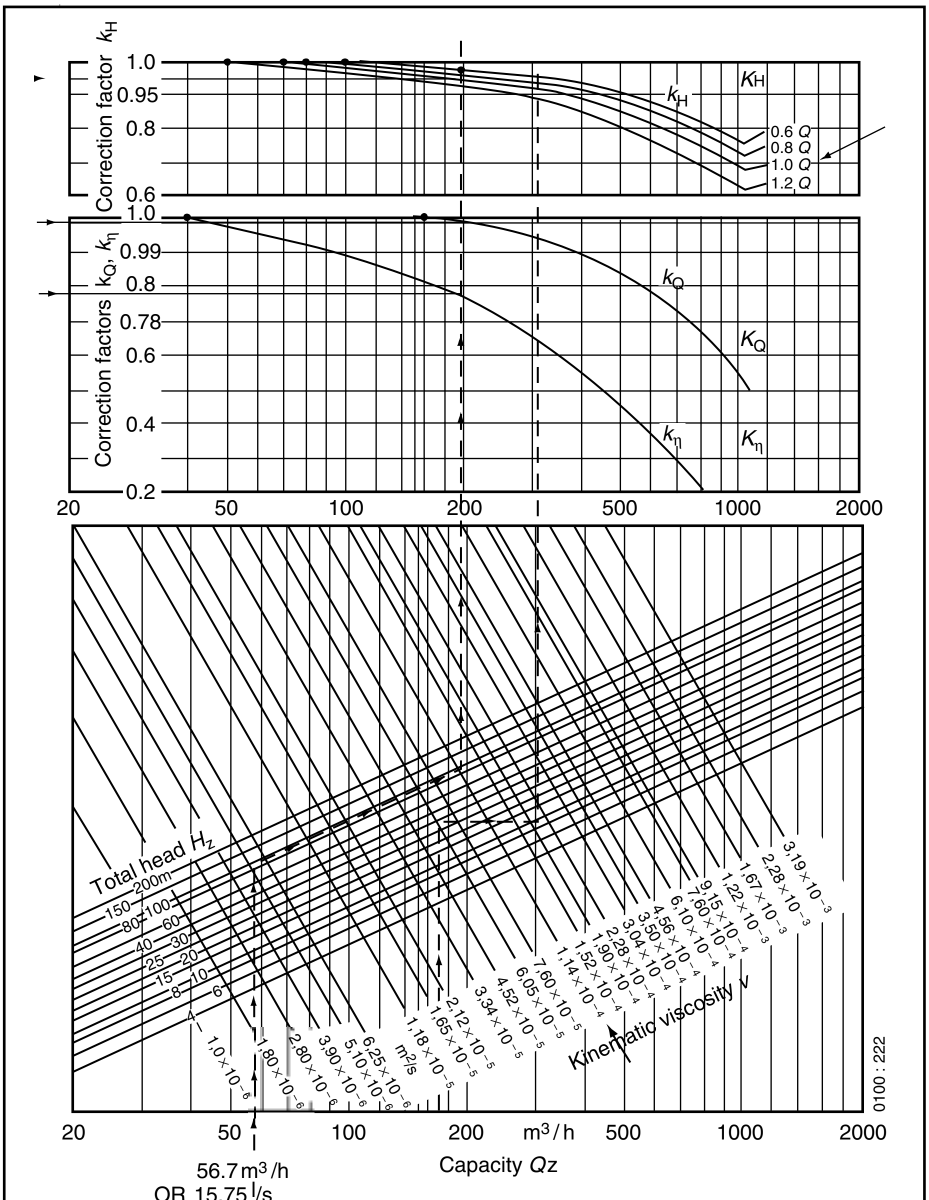


Fig. 2 relating to 'Viscosity'

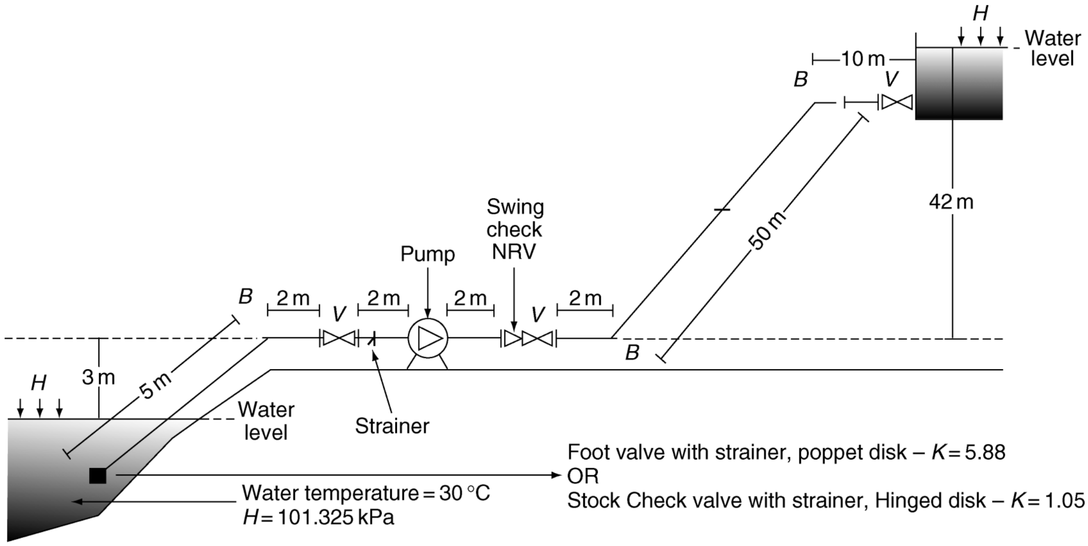
## A.3.5 Tutorial 5

### Pump selection exercise

In the example provided below, calculate the following:

- NPSH-a for the system
- NPSH-r from the pump performance chart at the operating point
- The total system head utilising the equivalent length – valves and fittings chart and tables provided
- Flow, head, and power absorbed by the pump when operating unrestricted
- Power absorbed by the pump when operating with the discharge throttled to achieve 140 l/s flow

- If the drive efficiency is 96.5% and the motor efficiency is 91%, calculate the motor running amps with flow at 156 l/s
- What size (high efficiency) motor must be selected for the pump to operate safely and reliably?




| **NPSH-a** | $H_a$ **(atmos)** | $H_{ts}$ **(Suct lift)** | $H_f$ **(Frict Hd)** | $H_{vap}$ |
|---|---|---|---|---|
| NPSH-a = | $\frac{101.325 \times 1000}{995.7 \times 9.8}$ | 3 m | [2.27 + 0.19 + 5 + 3.6($B_1$) + 2 + 1.7 ($V_1$) + 2 ] equivalent length of pipe | $\frac{4.241 \times 1000}{995.7 \times 9.8}$ |
| NPSH-a = 10.38 | | − 3 | 16.76 Equivalent length of pipe | − 0.43 |
| NPSH-a = 10.38 | | − 3 | − 0.33+ 2.46 | − 0.43 |

NPSH-a = 10.38 − (3 + 2.18 + 0.43) =10.38 − 3.82 = 6.56 m

**Calculating total system head** (i.e. $H_t = H_{ts} + H_f$, using equivalent pipe Length – $V/V_s$ & Fittings method)

$H_{ts}$ = 3 m + 42 m = 45.00 m
$H_f$ (equi. pipe meters) = 0.19 + 2.27 + 5 + 3.6($B_1$)+2 + 1.7 ($V_1$)+2 + 2 + 20 + 1.7 ($V_2$) + 2 + 3.6($B_2$) + 50 + 3.6($B_3$) + 10 + 1.7($V_3$) + 0.39 m, but 100 m of 250 mm pipe is equivalent of 2.3 m head therefore,
$H_f$ (equi. pipe meters) = 111.75
$H_f$ (equi. pipe meters) = 2.57 m
Therefore, Total system head = 45 m + 2.57 m = 47.57 m
From Pump performance chart, we note that at 47.57 m total hd, the NPSH-r = 3.8 m
We calculated NPSH-a as 6.56 m. Hence NPSH is adequate for satisfactory operations as NPSH-a >> NPSH-r by

6.56 − 3.8 = 2.76 m (refer to the pump performance chart)

Further, the pump will operate at the point where the total system head intersects the pump curve, i.e. at 47.6 m and 156 l/s and a power consumption of approx. 84 kW.

The Flow put out by the pump (with 406 mm dia impeller and 47.6 m total head) = 156 l/s

Operating efficiency of the pump – read from the pump curve = 86.50%

Power absorbed by the pump = $\dfrac{\rho \times g \times Q \times H}{10^6 \times \eta}$ $P$(kW) = 83.85 kW (calculating Power for flow 156 l/s & t/head 47.6 m)

*Note*: The actual power drawn at the operating point will depend up on 2 other factors –

1. The efficiency of the Drive – let us assume it is = 96.50% and
2. The efficiency of the motor (at full load – usually provided by the Manufacturer of the Motor) let us assume it is a high eff. Motor! = 91%.

Therefore the total power drawn by the motor will be (83.85 / 0.965) / 0.91 = 95.49 kW.
Per annum Energy costs to run the pump 24 h × 365 days a year, @ 0.11A$ / kW h = $95.49 \times 24 \times 365 \times 0.11$ \$92,014.16.

*Note*: When sizing the motor, a motor of approx. 10% higher capacity is usually ordered to provide a safety margin for Motors above 75 kW capacity.

Therefore, the right size motor for this pump would be approximately – 96.5 + 9.65 = approx. 103 kW.

(*Note*: It is prudent to order a high-efficiency motor as the energy savings over the life cycle of the pump unit would be significant, i.e. usually 2–3%. This amounts to a total of \$ p.a. of 0.025 × 96.5 × 0.11 × 24 × 365 = \$2324/annum or 2324 × 20 = \$46480 over 20 years!!)

If process requirements are 140 l/s, this flow output if achieved by throttling would reduce power consumption from approx. 84–81 kW. However, if the impeller was trimmed down from 406 to 384.6 mm diameter, the power absorbed by the pump would reduce to 68 kW, i.e. a savings of 13 kW!!

Per annum if the pump was run 24 h a day and 365 days a year, the savings would amount to = $13 \times 24 \times 365 \times 0.11$ \$12,526.80

Life cycle cost savings over 20 years = $12526.80 \times 20$ \$250,536.00

## A.3.6 Tutorial 6: Answers

1. Calculate the reduced impeller diameter for the pump that will deliver 140 l/s. Also
   (a) Calculate the energy cost saving per annum in trimming the impeller instead of throttling the discharge valve to achieve the same result. Cost of 1 unit of electricity (1 kW h) = \$0.11 with the pump running 24 h/day, 365 days a year and
   (b) How else may the pump out put be controlled?
   (c) Which option is most efficient and/or offers more flexibility?

*Given*:

$$\text{Power } P \text{ (kW)} = \rho \times g \times Q \times H / 1000 \times \eta$$

Where

$\rho$ = kg/m$^3$, $g = m/s^2$, $Q$ = l/s, $H$ = m and

$\eta$ = pump efficiency is read off the pump performance curve provided by the manufacturer.

1. The reduced impeller diameter, for the pump that operates unrestricted delivering 156 l/s @ 47.6 m head, is calculated using the formula

$$\left(\frac{D_2}{D_1}\right)^2 = \frac{Q_2}{Q_1} \quad \text{or} \quad D_2 = \left(\frac{Q_2}{Q_1}\right)^{-2} \times D_1$$

Where $D_2$ = unknown?, $D_1 = 406$ mm, $Q_1 = 156$ l/s and $Q_2 = 140$ l/s .

Therefore, $D_2 = 384.6$ mm.

To determine the head at which the pump will deliver 140 l/s with impeller size 384.6 mm, join the operating point (140, 51.5) to the origin (0, 0). Now read off the '*Y*' axis (*H*) at the point where this line through the origin intersects the vertical line drawn through 140 l/s (*Q*) on the '*X*' axis – i.e. 42.5 m.

1. (a) Calculating power absorbed by the pump with reduced impeller of 384.6 mm, delivering 140 l/s @ 42.5 m head and pump efficiency (read from the performance chart as 86.2%)

We get

$$P = \frac{(995.7 \times 9.81 \times 140 \times 42.5)}{(10^6 \times 0.862)} = 67.4 \text{ kW} = \text{approx. } 67\,\text{kW}$$

Similarly, power absorbed with impeller size 406 mm (pump discharge unrestricted (with $Q = 156$ l/s, $H = 47.6$ m and $\eta = 86.5\%$) = 83.85 kW = approx. 84 kW and

Power absorbed with impeller size 406 mm (pump discharge throttled (with $Q = 140$ l/s $H = 51.5$ m and $\eta = 87.1\%$) = 80.85 kW = approx. 81 kW.

Hence, if the process requirements is 140 l/s, the power absorbed by the pump with flow achieved through throttling would be 81 kW. However, if the impeller is trimmed down from 406 to 384.6 mm diameter, the power absorbed by the pump will further reduce to 67 kW, i.e. a savings of 14 kW!!

Refer to the table below for running cost per annum and cost savings comparison.

| **For Pump Operating 24 h a Day, 365 Days a Year and Electricity Costs = \$0.11/ kW h** | | | |
|---|---|---|---|
| **Power Absorbed by the Pump** | **Annual Cost @ 84 kW** | **Annual Cost @ 81 kW** | **Annual Cost @ 67 kW** |
| Unchecked | \$80,942 | | |
| Throttled | | \$78,052 | |
| Impeller trimmed | | | \$64,561 |
| **Annual cost savings** | | **\$2,891** | **\$16,381** |

1. (b) How else may the pump out put be controlled?

Other options available to control centrifugal pump output are;
Speed control of the Motor and
Bye pass control

1. (c) Which option is most efficient and / or offers more flexibility?

Trimming of the impeller offers best efficiency, but no flexibility once the impeller is trimmed down – i.e. cannot increase impeller size if required in the future.

Speed control is the best choice as it offers pump capacity control efficiently, along with flexibility (i.e. by varying speed).

Throttling provides a small reduction in power absorbed along with flexibility.

Bye pass control is most inefficient for **Radial flow pumps** but provides flexibility. However, it provides pump capacity control efficiently, along with flexibility (i.e. by varying the amount of flow bypassed) for **Axial flow pumps!!**

## A.3.7 Tutorial 7: Answer

Refer to the values for Pump test conditions set out in the table below. Utilising the Equation provided in AS 2417.3 to verify if a pump has met its guarantee performance. Accordingly, determine mathematically under which conditions the Pump guarantee parameters have been met.

*Given*: Manufacturer's guaranteed duty point for the pump is 140 l/s @51.5 m head. AS 2417.3 stipulates acceptable deviation in terms of $H_G$ as (±2.0%) and (±4.0%) for $Q_G$. In this case the center point for the ellipse will be (140, 51.5) and the major and minor axis for the ellipse will be (140 × 0.04 × 2 = 11.2 and 51.5 × 0.02 × 2 = 2.06).

### Ellipse analysis – calculations

| Value $\Delta H$ (m) | Value $\Delta Q$ (l/s) | $[H_G \times X_H/\Delta H]^2 + [Q_G \times X_Q/\Delta Q]^2$ >>= 1 | Pump $H_G$ Guarantee (m) | Pump $Q_G$ Guarantee (l/s) | Deviation $X_H = 2.0\%$ | Deviation $X_Q = 4.0\%$ |
|---|---|---|---|---|---|---|
| | | Design point | **51.5** | **140** | 0.02 | 0.04 |
| 3.9 | 16 | 0.19 | 51.5 | 140 | 0.02 | 0.04 |
| 3 | 14 | 0.28 | 51.5 | 140 | 0.02 | 0.04 |
| 2 | 11 | 0.52 | 51.5 | 140 | 0.02 | 0.04 |
| 1 | 9 | **1.45** | 51.5 | 140 | 0.02 | 0.04 |
| 0.5 | 7 | **4.88** | 51.5 | 140 | 0.02 | 0.04 |
| − 1 | 5 | **2.32** | 51.5 | 140 | 0.02 | 0.04 |
| − 2 | − 5 | **1.52** | 51.5 | 140 | 0.02 | 0.04 |
| − 3 | − 8 | 0.61 | 51.5 | 140 | 0.02 | 0.04 |
| − 4 | − 10 | 0.38 | 51.5 | 140 | 0.02 | 0.04 |
| 1 | − 12 | **1.28** | 51.5 | 140 | 0.02 | 0.04 |
| − 1 | − 15 | **1.20** | 51.5 | 140 | 0.02 | 0.04 |
| 5 | 5.5 | **1.08** | 51.5 | 140 | 0.02 | 0.04 |
| 10 | 5.6 | **1.01** | 51.5 | 140 | 0.02 | 0.04 |
| 1.03 | 5.6 | **2.00** | 51.5 | 140 | 0.02 | 0.04 |

## A.3.8 Tutorial 8: Answer

If a Pump's speed is 2450 rpm and duty is 200 l/s @ 60 m head whilst absorbing 120 kW, what would be the pump duty and duty power if the pump speed is increased to 2950?
*Given*:

$$Q_1 = 200 \text{ l/s},\ N_1 = 2450 \text{ rpm},\ P_1 = 120 \text{ kW},\ H_1 = 60 \text{ m and } N_2 = 2950 \text{ rpm}.$$

From the Centrifugal pump Affinity Laws, we know that for variation in speed with constant impeller diameter the following rule applies

$$\frac{Q_1}{Q_2} = \frac{N_1}{N_2} \text{ or}$$

$$Q_2 = \left(\frac{N_2}{N_1}\right) \times Q_1 = 200 \times \frac{2950}{2450}$$

$$Q_2 = 240.82 \text{ l/s}$$

Further, we know that the pump head varies with the square of the speed – i.e.

$$\frac{H_1}{H_2} = \left(\frac{N_1}{N_2}\right)^2 \text{ or}$$

$$H_2 = \left(\frac{N_2}{N_1}\right)^2 \times H_1$$

$$H_2 = \left(\frac{2950}{2450}\right)^2 \times 60$$

$$H_2 = 86.98 \text{ m or } 87 \text{ m}$$

We also know that power absorbed varies with the cube of the speed – i.e.

$$P_1 / P_2 = \left(\frac{N_1}{N_2}\right)^3 \text{ or}$$

$$P_2 = P_1\left(\frac{N_2}{N_1}\right)^3 \text{ or}$$

$$P_2 = 120\left(\frac{2950}{2450}\right)^3$$

$$P_2 = 209.48 \text{ kW or } 209.5 \text{ kW}$$

*Note*: The steep increase in power absorbed which could overload the motor. Hence proceed with caution!

# A.4 Miscellaneous practicals

## A.4.1 Centrifugal pumps

### Work group 'problem-solving' exercise

This practical session has been designed to go over a range of fault situations that could be encountered with centrifugal pump operations.

The class will be divided into groups and the presenter will ask each group to work through a couple of fault scenarios. The group is required to list all possible causes that could lead to the fault occurring and appoint a spokes person to present the list prepared when asked to do so by the presenter.

The list of topics are :

1. Pump does not develop any head pressure nor does it deliver any liquid.
2. Pump delivers no liquid but develops some pressure.

3. Pump output is below the manufacturer's provided performance data as in the associated performance curve provided with the pump.
4. Pump draws higher amps than specified.
5. Pump performance is compromised, although nothing appears to be wrong with pumping system.
6. Pump operates satisfactorily during start up, however, performance deteriorates in a relatively short time.
7. Pump operates with noise or vibrations, or both.
8. Stuffing box leaks abnormally.
9. Gland packing has short working life.
10. Mechanical seal fails prematurely.
11. Bearings fail prematurely.
12. Bearings run noisily, i.e. steady high-pitched or intermittent/continuous low-pitched noise or intermittent rumble/rattle and or clicks, Intermittent squeal or high-pitch tone.
13. Pump overheats or seizes, or both.
14. Pump cavitates when the NPSH-a is increased.
15. List all the 'dos' that come to mind in regards to safe operation of pumps.
16. List all the 'don'ts' that come to mind in regards *to safe* operation of pumps.

| **Q1.** | **Pump does not develop any head pressure nor does it deliver liquid** |
|---|---|
| A1. | • Pump not primed (not full of liquid – air locked) |
| | • Shaft could have sheared |
| | • Check pump drive coupling |
| | • Impeller key could have sheared or missed out |
| | • Pump impeller not installed |
| | |
| | |
| | |

| **Q2.** | **Pump delivers no liquid but develops some pressure** |
|---|---|
| A2. | • Air lock in pump casing or in pipelines |
| | • Foot valve malfunctioning – frozen or possibly clogged |
| | • Strainer restricted with a build up of usually stringy matter |
| | • Suction line clogged |
| | • Strainer element restricted with solid deposits – such as sand |
| | • Discharge pressure required by system is higher than maximum pressure developed by pump |
| | • Operating speed too low |
| | • Wrong direction of operation |
| | • Available NPSH inadequate |

*(Continued)*

| **Q2.** | **Pump delivers no liquid but develops some pressure** |
| --- | --- |
| | • Excessive amounts of gas or air entrained in pumped liquid |
| | • Under size impeller or machined down to a smaller diameter that required |
| | |
| | |
| | |
| | |

| **Q3.** | **Pump output is below the manufacturer's provided performance data as in the associated performance curve provided with the pump** |
| --- | --- |
| A3. | • Air enters pump during operation, or pumping system not de-aerated (primed properly) before starting |
| | • Verify pump rpm – *n* |
| | • Incorrect direction of rotation |
| | • System resistance higher than that developed by pump |
| | • Measuring instruments inaccurate, i.e. not properly calibrated or incorrectly installed |
| | • NPSH-a (actual) too low |
| | • Excessive amount of air or gas entrained in pumped liquid |
| | • Excessive leakage through wearing rings or other sealing faces |
| | • Viscosity of liquid higher than that for which pump has been designated |
| | • Fouled Impeller or casing, i.e. partially clogged with solid matter |
| | • Fins, burrs, or sharp edges in path of liquid |
| | • Incorrectly installed impeller |
| | • Pump operating too far out of the head-capacity curve (incorrect selection) |
| | • Obstruction to flow in suction or discharge piping |
| | • Foot valve clogged or jammed |
| | • Suction strainer fouled thus restricting flow and lowering NPSH |
| | • Incorrect layout of suction or discharge piping |
| | • Incorrect layout of sump suction |
| | • Excessive leakage through stuffing box or seal |
| | • Excessive amount of liquid recirculated internally to stuffing box lantern or seal |
| | • Excessive leakage through hydraulic balancing device |
| | • Liquid level in suction tank or sump lower than originally specified |
| | • In a system with more than one pump, operation of one pump may affect operation of others |

| Q4. | Pump draws higher amps than specified |
|---|---|
| A4. | • Speed too high |
| | • Process liquid of higher viscosity/specific gravity than originally quoted being pumped |
| | • Oversized impeller installed |
| | • Total head of system either higher or lower than anticipated |
| | • Misalignment between pump and driver |
| | • Rotating parts in contact with stationary parts |
| | • Worn or damaged bearings |
| | • Packing improperly installed |
| | • Incorrect type of packing |
| | • Mechanical seal exerts excessive pressure on seat |
| | • Gland too tight |
| | • Improper lubrication of bearings |
| | • Too much lubricant in bearings |
| | • Bent shaft |
| | • Uneven thermal expansion of different parts of pumping unit |
| | • Faulty power-measuring instruments |
| | • Power-measuring instruments incorrectly mounted or connected |
| | • Wrong direction of rotation |
| | • Liquid not preheated to keep viscosity below specified limits |
| | • Impeller or casing partially clogged with solid matter |
| | • Wetted surfaces of impeller or casing very rough |
| | • Damaged impeller |
| | • Faulty casting of impeller or casing |
| | • Impeller incorrectly located in casing |
| | • Impeller inversely mounted on shaft |
| | • Pump operating too far out on head-capacity curve |
| | • Breakdown of discharge line |
| | • Faulty motor bearings |

| Q5. | **Pump performance compromised, although nothing appears to be wrong with pumping system** |
|---|---|
| A5. | This is usually due to incorrect testing. The reasons for this maybe as follows: |
| | • Incorrect measuring instruments |
| | • Measuring instruments damaged during installation |
| | • Measuring instruments mounted in wrong locations |
| | • Tubing that leads from pipelines to measuring instruments clogged |
| | • Instrument-connecting tubing that should be full of liquid not de-aerated completely |
| | • Instrument-connecting tubing that should be full of air contains some liquid (manometers) |
| | • Leakage in instrument-connecting tubing or in its fittings |
| | • Burrs or fins at mouth of connections between tubing and piping |
| | • Incorrect connections of wiring to electrical instruments |
| | • Loose connections at wires terminal |
| | • Fouled/oxidized electrical terminals or contacts |
| | • Dust or dirt in torque bar |
| | • Torque bar incorrectly mounted |
| | • Misalignment or fouled bearings in a dynamometer produces false readings |
| | • Excessive friction in pivots or pulleys that guides the levers and cables in a dynamometer produces false readings |
| | • Weight and stiffness of the electrical cables in a dynamometer affect torque readings |
| | • Turbulence in pipelines where instruments are hooked up |
| | • Discrepancy with inner diameter of piping not matching design specification |

| Q6. | **Pump operates satisfactorily during start, but performance deteriorates in a relatively short time** |
|---|---|
| A6. | • Air leaks into pump |
| | • Pumped liquid contains high percentage of entrained air or gas |
| | • Rate of make up liquid into suction sump in adequate – results in air being drawn into pump |
| | • Air pocket in suction line has moved into pump |
| | • Air funnels in sump suction (vortex formation) |
| | • Drop in NPSH |

| Q7. | **Pump operates with noise or vibrations, or both** |
|---|---|
| A7. | • Misalignment between pump and driver |
| | • Rotating parts rubbing against stationary parts |
| | • Worn bearings |
| | • Incorrect direction of rotation |
| | • Available NPSH too low |
| | • Impeller or casing partially filled with solid matter – out of balance |
| | • Fins, burrs, or sharp edges in waterways causing cavitation |
| | • Damaged impeller |
| | • Impeller incorrectly mounted |
| | • System requirements too far out on head-capacity curve |
| | • Suction strainer filled with solid matter |
| | • Strainer covered with fibrous matter |
| | • Incorrect layout of suction sump |
| | • Air enters pump during operation |
| | • Mutual interaction of several pumps within one common system |
| | • Incorrect layout of suction or discharge piping |
| | • Piping imposes strain on pump |
| | • Pump operating at critical speed |
| | • Rotating elements not balanced |
| | • Excessive radial forces on rotating parts |
| | • Too small distance between impeller outer diameter and volute tongue |
| | • Faulty shape of volute tongue |
| | • Undersized suction or discharge piping and fittings causing cavitation somewhere in system |
| | • Loose valve disk in system |
| | • Bent shaft |
| | • Impeller bore not concentric with its outer diameter or not square with its face |
| | • Misalignment of pump parts |
| | • Pump operates at very low flow rates |
| | • Improperly designed base plate or foundations |
| | • Resonance between pump speed and natural frequency of base plate or foundations |
| | • Resonance between pump speed and natural frequency of base plate or foundations |
| | • Resonance between operating speed and natural frequency of piping |
| | • Resonance between operating speed and valve discs |
| | • Loose bolts |
| | • Uneven thermal expansion |
| | • Improper installation of bearings |
| | • Damaged bearings |
| | • Improper lubrication of bearings |
| | • Obstruction to flow in suction or discharge piping |

*(Continued)*

| Q7. | **Pump operates with noise or vibrations, or both** |
|---|---|
| | • Total head of system either higher or lower than expected |
| | • Excessive amount of air or gas entrained in liquid |
| | • Waterways of impeller or casing badly eroded or rough |
| | • Cavitation in pipelines |

| Q8. | **Stuffing box leaks abnormally** |
|---|---|
| A8. | • Worn bearings |
| | • Improperly installed packing |
| | • Incorrect type of packing |
| | • Rotating element not balanced |
| | • Excessive radial forces on rotating parts |
| | • Bent shaft |
| | • Bore of impeller not concentric with outer diameter, or not square with face |
| | • Misalignment of pump parts |
| | • Rotating parts running off-center |
| | • Water-seal pipe clogged |
| | • Seal cage improperly located |
| | • Shaft sleeve worn or scorched at packing |
| | • Failure to provide cooling liquid to water-cooled stuffing boxes |
| | • Excessive clearance at bottom of stuffing box (between shaft and box bottom) |
| | • Dirt or grit in sealing liquid |

| Q9. | Gland packing has short working life |
|---|---|
| A9. | • Worn bearings |
| | • Improperly installed packing |
| | • Incorrect type of packing |
| | • Gland too tight |
| | • Rotating element not balanced |
| | • Excessive radial forces on rotating parts |
| | • Bent shaft |
| | • Bore of impeller not concentric with its outer diameter or not square with its face |
| | • Misalignment of pump parts |
| | • Rotating parts running off-center from damaged bearings or other parts |
| | • Water-seal pipe clogged |
| | • Seal cage improperly located in stuffing box, preventing sealing fluid from entering |
| | • Shaft scorched where it contacts packing |
| | • Failure to provide cooling liquid to water-cooled stuffing box |
| | • Excessive clearance at bottom of stuffing box, between shaft and stuffing box's bottom |
| | • Dirt or grit in sealing liquid |
| | • Improper lubrication of packing |
| | • Space in stuffing box where packing is located is eccentric to the shaft |

| Q10. | Mechanical seal fails prematurely |
|---|---|
| A10. | • Worn bearings |
| | • Rotating elements not balanced |
| | • Excessive radial forces on rotating parts |
| | • Bent shaft |
| | • Misalignment of pump parts |
| | • Rotating elements running off-center from damage to bearings or other parts |
| | • Dirt or grit in seal-flushing liquid |
| | • Sealing face not perpendicular to pump axis |
| | • Mechanical seal has been run dry |
| | • Abrasive particles in liquid coming in contact with seal |
| | • Mechanical seal improperly installed |
| | • Incorrect type of mechanical seal |
| | • Misalignment of internal seal parts preventing proper mating between seal and seat |

| **Q11.** | **Bearings fail prematurely** |
|---|---|
| A11. | • Damaged impeller |
| | • Impeller partially clogged |
| | • Rotating elements not balanced |
| | • Excessive radial loads on rotating parts |
| | • Excessive axial loads |
| | • Bent shaft |
| | • Bore of impeller not concentric with outer diameter or not square with hub face |
| | • Misalignment of pump parts |
| | • Misalignment between pump and driver |
| | • Pump operates for prolonged time at low flow rate |
| | • Improper base plate or foundations |
| | • Rotating parts running off-center from damaged or misaligned parts |
| | • Improper installation of bearings |
| | • Bores of bearing housing not concentric with bores in water-end |
| | • Cracked or damaged bearing housing |
| | • Excessive grease in bearings |
| | • Faulty lubrication system |
| | • Improper workmanship during installation of bearings |
| | • Bearings improperly lubricated |
| | • Dirt ingress in to bearing housing |
| | • Water has entered bearing housing |
| | • Excessive wear of impeller wear rings which adversely affects support of the rotating element |
| | • Excessive suction pressure |
| | • Too tight fit between line bearing and seat (may prevent it from sliding under axial load, transferring this load to the line bearing) |
| | • Inadequate cooling of bearings where applicable |
| | • Inadequate cooling of lubricant where applicable |
| | • Source of cooling media restricted or shut-off from bearing housing |

| Q12. | Bearings run noisily |
|---|---|
| A12. | *A. Steady high-pitch tone* |
| | • Excessive radial load |
| | • Excessive axial load |
| | • Misalignment |
| | • Too much clearance between bearing and shaft, and/or housing |
| | |
| | *B. Continuous or intermittent low-pitch tone* |
| | • Bearing brinelled |
| | • Pitted raceway, from dirt |
| | • Resonance with other structural pump parts |
| | |
| | *C. Intermittent rattles, rumbles, and/or clicks* |
| | • Loose machine parts |
| | • Dirt in bearings |
| | • Clearance between balls and races too large for given application |
| | • Bearings that require preloading not adequately preloaded |
| | |
| | *D. Intermittent squeal or high-pitch tone* |
| | • Balls skidding from excessive clearance between balls and races |
| | • Balls skidding from insufficient preloading (whenever required) |
| | • Shaft rubbing against housing from improper mounting of housing |
| | • Shaft rubbing against housing from bent shaft |
| | • Shaft rubbing against housing from having been machined eccentrically |

| Q13. | Pump overheats or seizes, or both |
|---|---|
| A13. | • Pump allowed to run dry |
| | • Vapor or air pockets inside pump |
| | • Pump operates near shut-off |
| | • Simultaneous operation of poorly matched pumps |
| | • Internal misalignment from too much pipe strain, poor foundations, or faulty repair work |
| | • Internal rubbing of rotating parts against stationary parts |
| | • Worn or damaged bearings |
| | • Poor lubrication |
| | • Rotating and stationary wearing rings made of identical, galling-prone materials |

| **Q14.** | **Pump cavitates when the NPSH-a is increased** |
|---|---|
| A14. | This may happen when the increase in the available NPSH has reduced the system resistance so far that the pump operates far out on the *Q–H* curve. This happens when |
| | • Oversized impeller installed in pump |
| | • Pump operates at excessive speed |
| | • Breakdown or serious leak in discharge line |
| | • Open bypass in discharge line |
| | • Extremely large clearances between impeller and casing |
| | • Hole in casing allowing liquid from pressure side of casing to return to its suction inlet. |

| **Q15.** | **List all the 'dos' that come to mind in regard to *SAFE* operation of pumps**<br>**-----------------Dos -----------------** |
|---|---|
| A15. | • Read manufacturers instructions prior to operation<br>• Check sumps and piping are clean<br>• Prime the pump before starting<br>• Regulate flow if necessary with the discharge valve<br>• Pump maybe run up with the discharge valve closed only for a Limited time<br>• Never over tighten packing glands<br>• Check alignment prior to placing a pump in service<br>• Operate with in manufacturers speed recommendations<br>• Support piping adequately to avoid stresses on pump casings<br>• Allow pump foundation to set firmly prior to bolting down the pump<br>• Ensure that NPSH-a exceeds NPSH-r by a margin of 0.5–1.0 m<br>• Ensure proper pipe gradients / lay out to avoid trapping air in suction lines (air lock)<br>• Guard pump drives and all moving or high temperature parts |

| **Q16.** | **List all the 'don'ts' that come to mind in regard to *SAFE o*peration of pumps**<br>**-----------------Don'ts -----------------** |
|---|---|
| A16. | • Fail to read manufacturers instructions<br>• Fail to check sumps and piping is clean<br>• Run a pump without liquid<br>• Regulate flow with a suction valve<br>• Run a pump with a closed discharge<br>• Over tighten packing glands<br>• Fail to check alignment<br>• Exceed manufacturers speed recommendations<br>• Impose piping stresses on pump casings<br>• Bolt a pump to a foundation until it is firmly set<br>• Attempt suction lift greater than a safety margin under NPSH-a<br>• Trap air in suction lines<br>• Allow pumps to run with unguarded moving or high temperature parts |

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