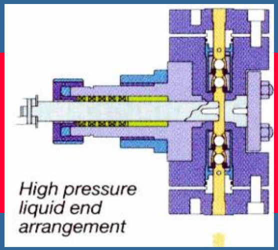
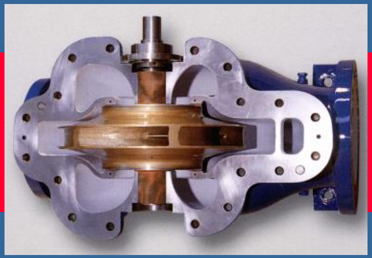
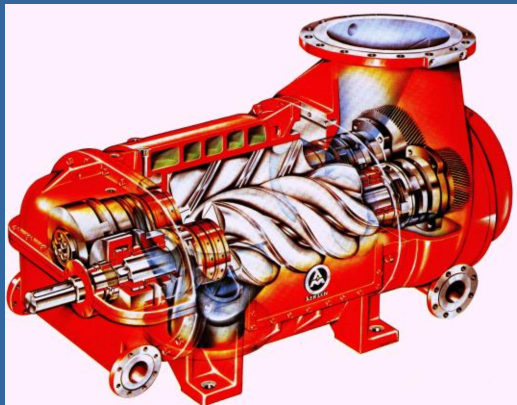


# Industrial Compressors & Pump: Operation & Maintenance



# Rotating Equipment

- Pumps
- Mixers
- Compressors
- Internal Combustion Engines
- Gas Turbines
- Steam Turbines

**A very wide horizon with a vast number of topics**

# Equipment in Service

It is very desirable to achieve

- **Maximum Reliability of the Equipment**
- **Increased Effectiveness of the Maintenance and Operating Personnel**
- **Reduced Maintenance Cost with Optimum Results**
- **Increased Plant Availability and Maximum Profitability**

# Diagnostics

- Operating Characteristics
  - Comparisons with Original Performance Tests
    - General Operating Features

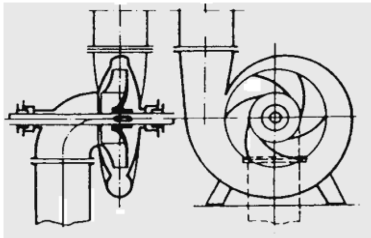
## Assist in Fault Finding and Diagnostics

# Pumps

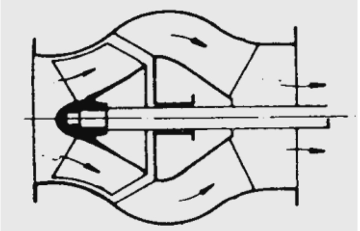
- Centrifugal
  - Partial Emission
  - Regenerative
  - Radial
  - Mixed Flow
  - Axial
- **Positive Displacement**
  - Rotary
    - Gear, Lobe, Vane, Screw
  - Reciprocating
    - Piston, Diaphragm
  - Miscellaneous

# Pump Types (i)

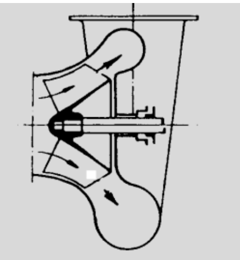
In the Radial Pump the flow is substantially perpendicular to the shaft i.e. in the radial direction.



In the Axial Pump the flow is substantially parallel to the shaft i.e in the axial direction.

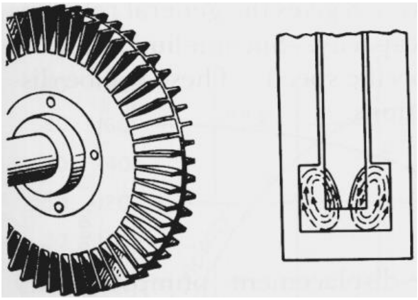


In the Mixed Flow Pump the flow is a combination of the previous two i.e at an acute angle to the shaft.

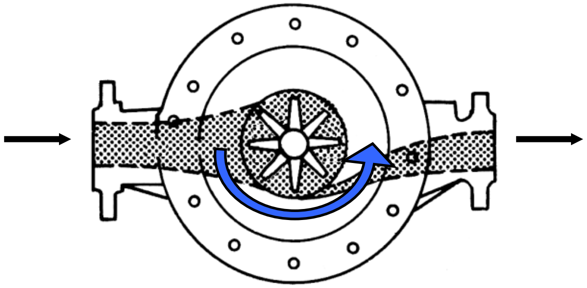


# Pump Types (ii)

In the Regenerative Pump the liquid enters at the periphery of the impeller in the appropriate channels of the impeller and of the casing where the liquid is centrifuged. The flow pattern gives the pump another name "periflow"



**Regenerative Pump**



**Partial Emission Pump**

In the Partial Emission Pump the flow enters at the central part of the impeller where it is taken up by the vanes and is centrifuged in a circular forced vortex path. Part of this liquid is discharged whilst an equal amount enters the impeller.

# Pump Types (iii)

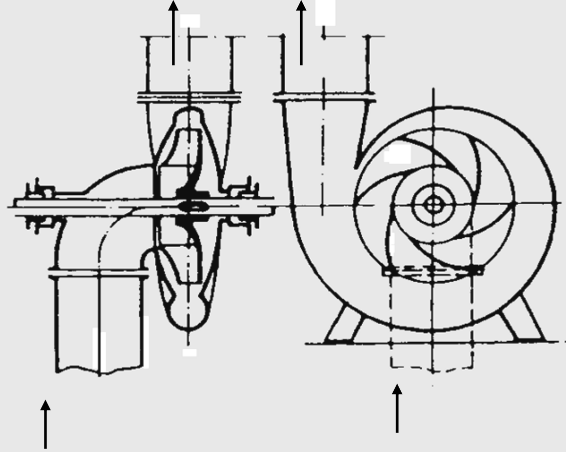
In the Rotary Positive Displacement Pumps  a specific quantity of liquid is taken by the rotor and it is rotated to the discharge where by the positive action of the rotor the liquid is delivered against the discharge pressure.

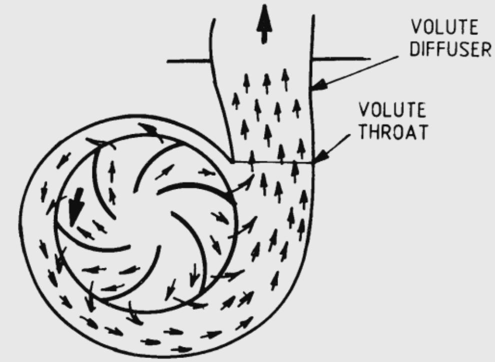
In the Reciprocating Pump the pumping element (piston or plunger) reciprocates between the rest position at suction to accelerate and then discelerate to the rest position at discharge and then again back to suction.

The design of the Miscellaneous Pumps combines the Rotary and the Reciprocating action of the previous two types in several design variations.

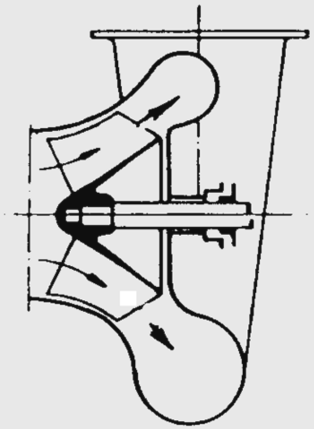
# Typical Flow Paths of Centrifugal Pumps

Radial Pump

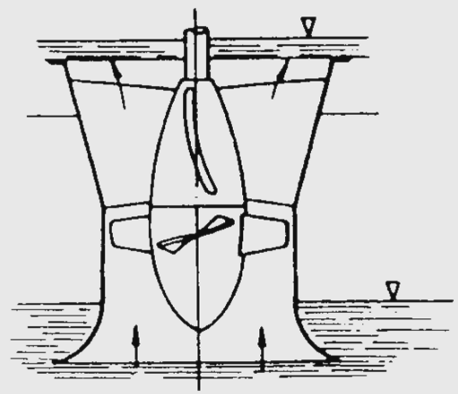




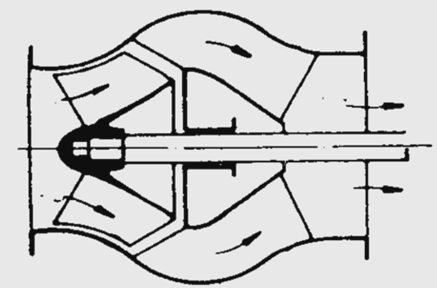

Mixed Flow (Francis Type) Pump



Axial Pump



**Vertical Axial Pump**

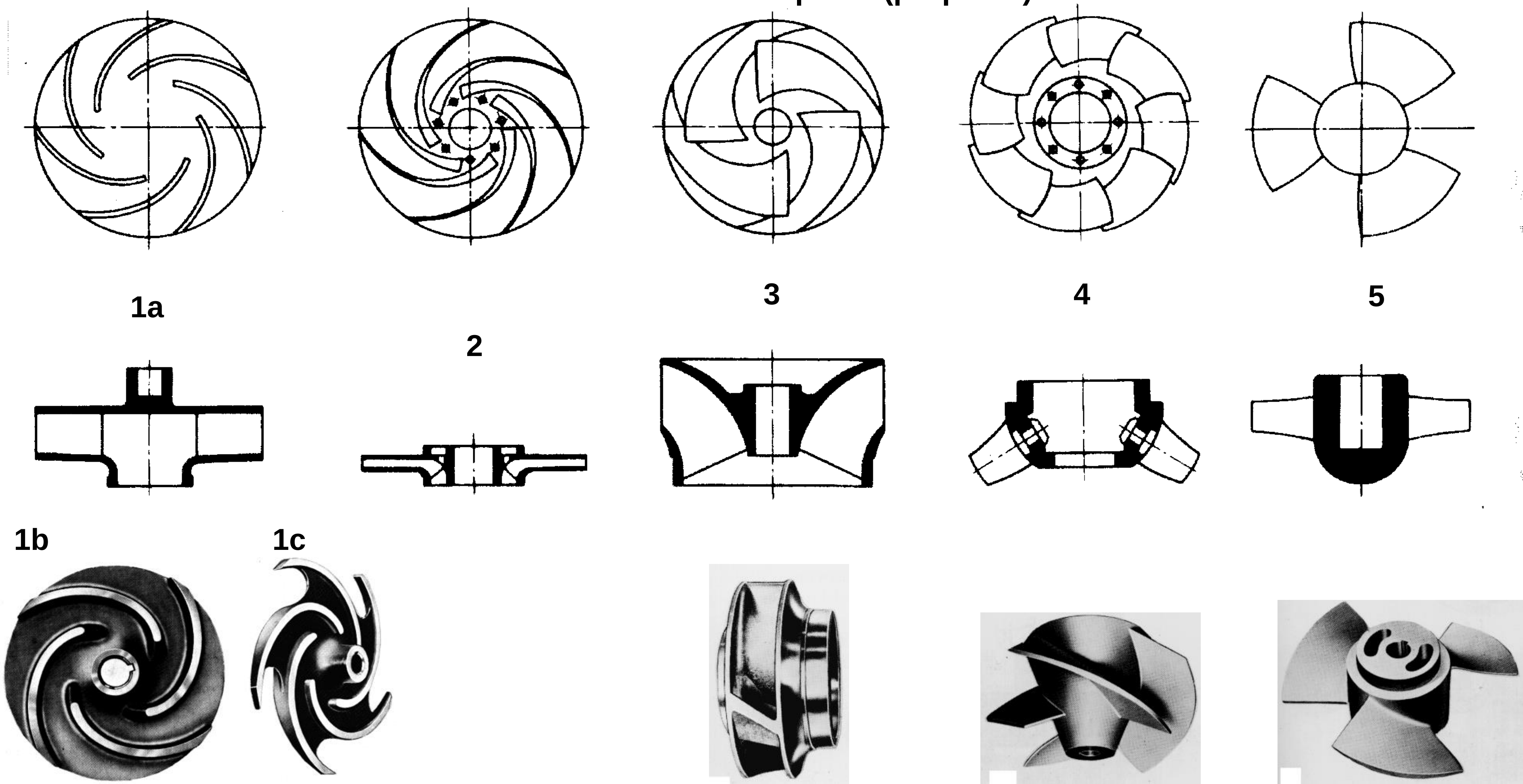


**Mixed Flow Pump**

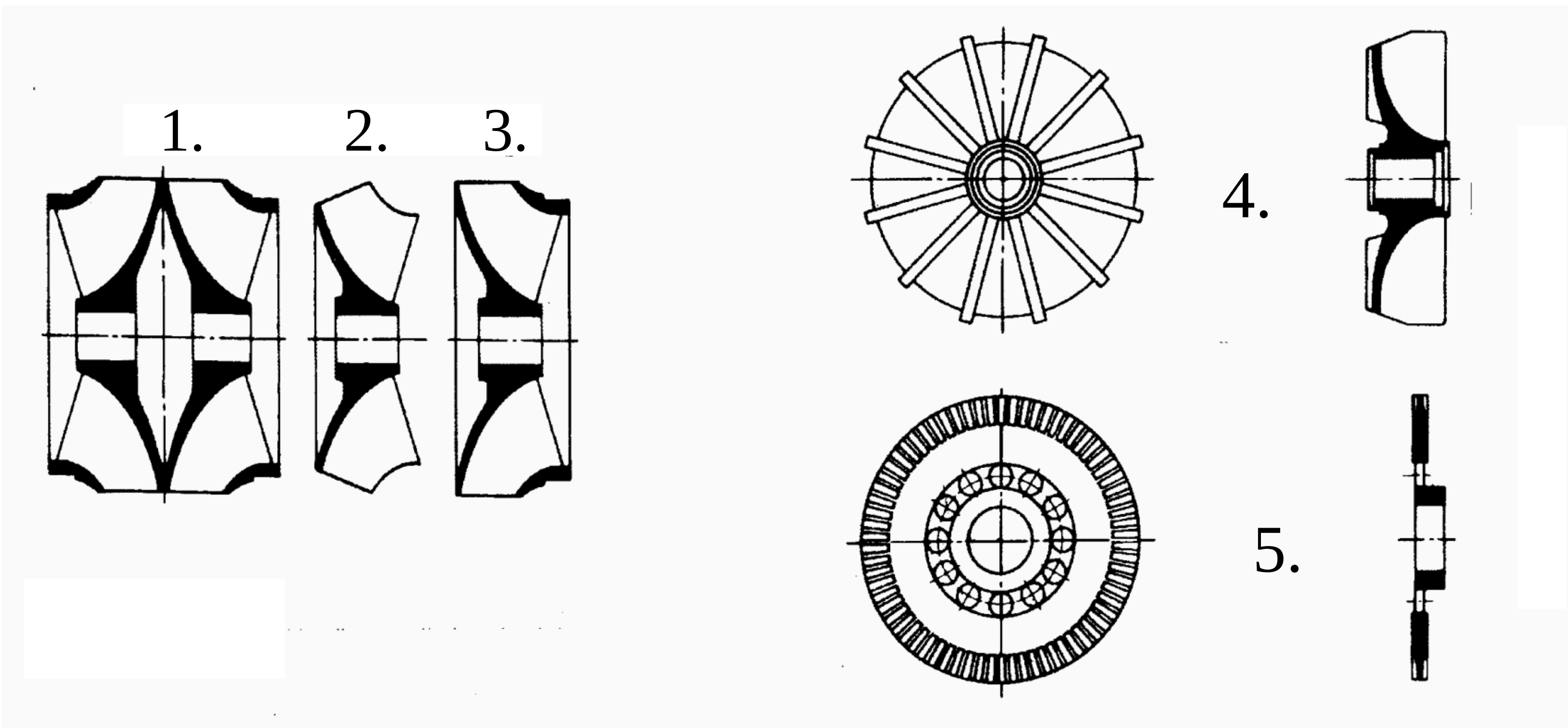
# Centrifugal Pump Impellers

1a Closed Radial Impeller
1b Semi-open Radial Impeller
1c Open Radial Impeller

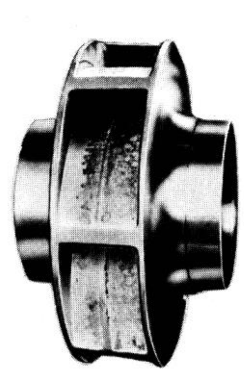
2 Radial Impeller (with vanes extending into the eye)
3 Mixed Flow (Francis) Impeller
4 Mixed Flow (semi axial) Impeller
5 Axial Impeller (propeller)



# Some Rotodynamic Pump Impellers

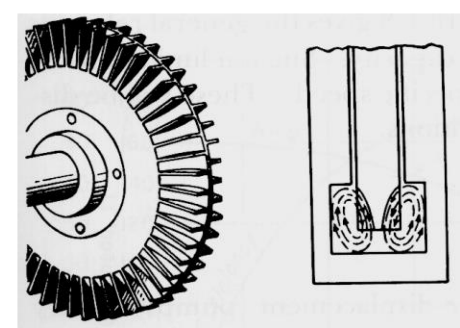



1.

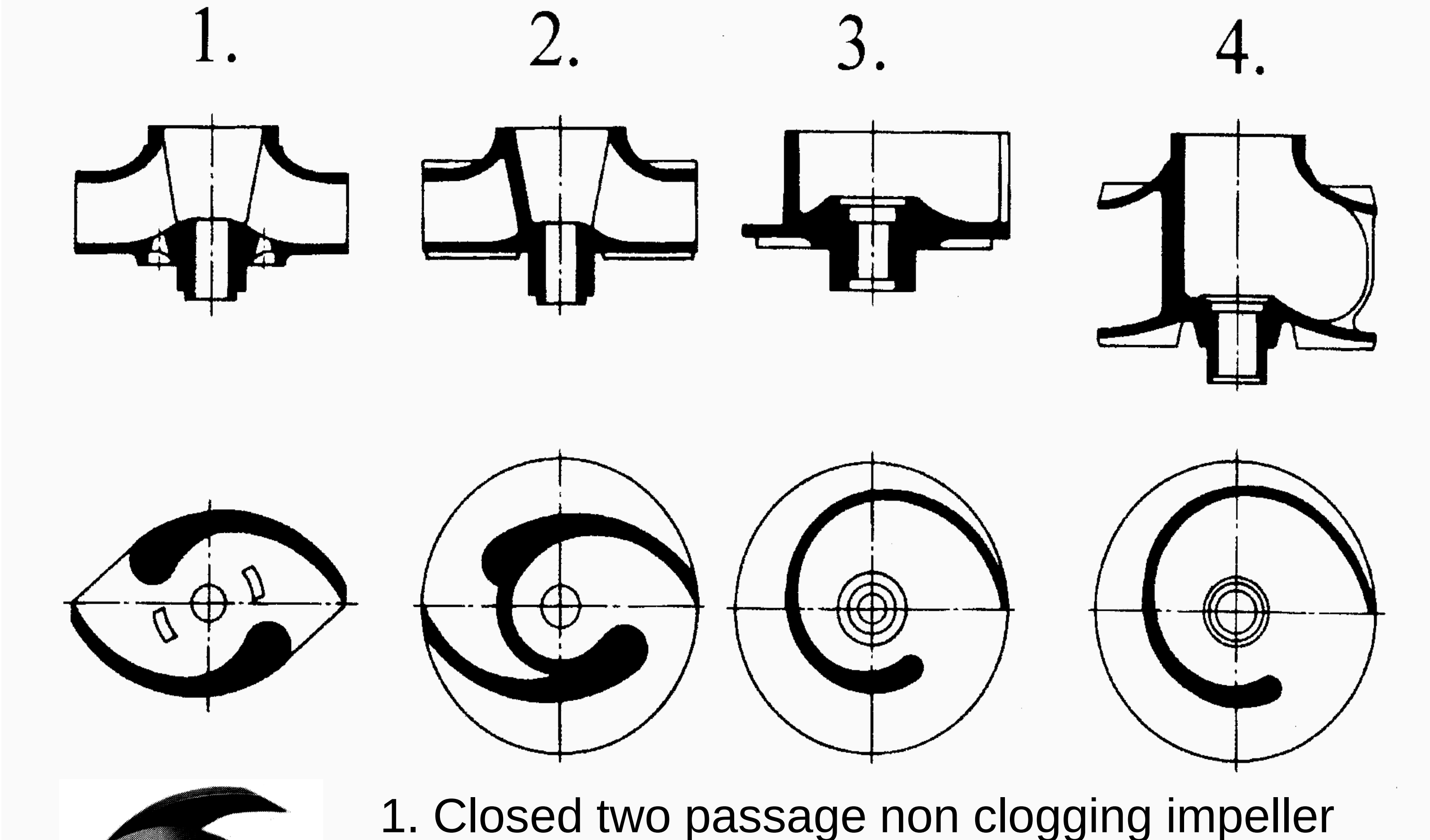


1. Double Suction Closed Impeller
2. Semi open Single Flow Impeller
3. Closed Single Flow Impeller
4. Partial Emission (Barske) Impeller
5. Regenerative (Periflow) Impeller

5.



# Some Solids Handling Centrifugal Pump Impellers




5.

1. Closed two passage non clogging impeller
2. Closed single passage impeller
3. Open single vane impeller
4. Closed single vane impeller
5. Open two vane non clogging impeller

# Pump Impeller & Casing Function

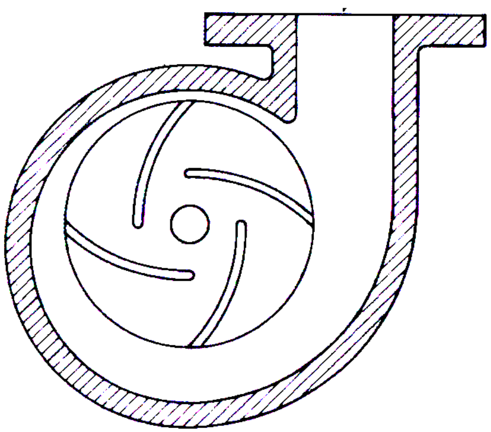
The **impeller impels** the liquid in a vortex motion to a higher velocity and higher pressure**.**

In the **volute casing** the pumped liquid is gradually decelerated as the flow area of the volute gradually increases and so the liquid pressure rises as the velocity decreases.. There is a further reduction in velocity and corresponding increase in pressure in the discharge nozzle diffuser.

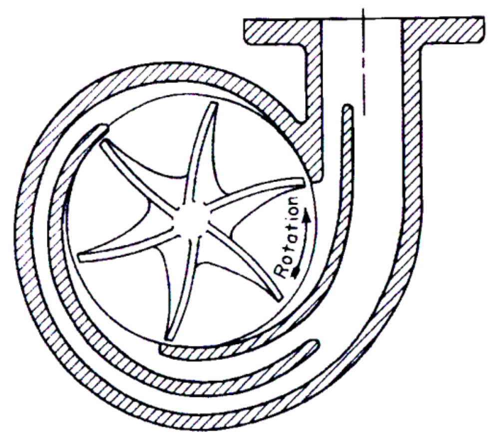
In certain cases of large pumps instead of the simple volute casing stationary vanes are installed into the volute. In this case the casing is called the **diffusion casing** where there is a more efficient conversion of velocity head into pressure head.

# Some Pump Casings

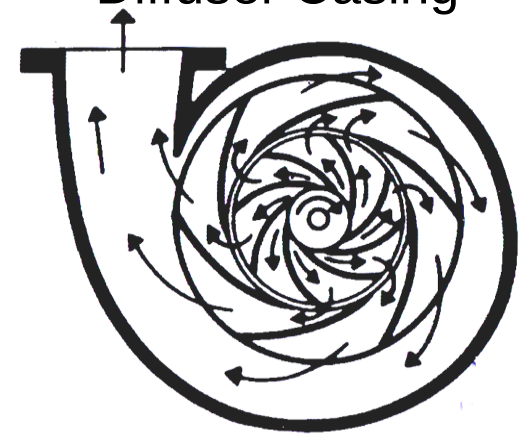
Volute Casing

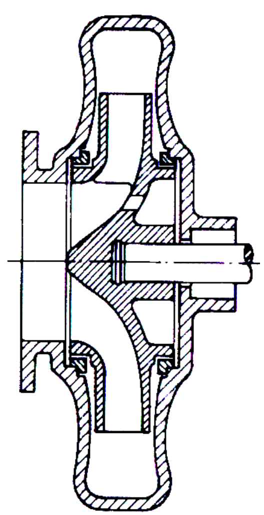


Double Volute Casing

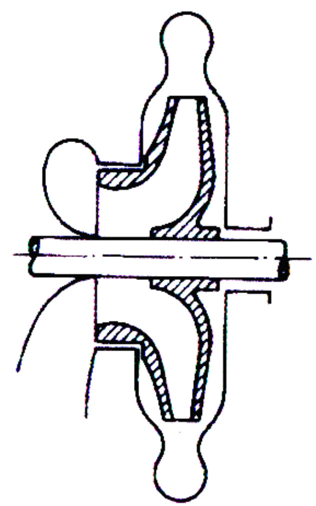



Diffuser Casing

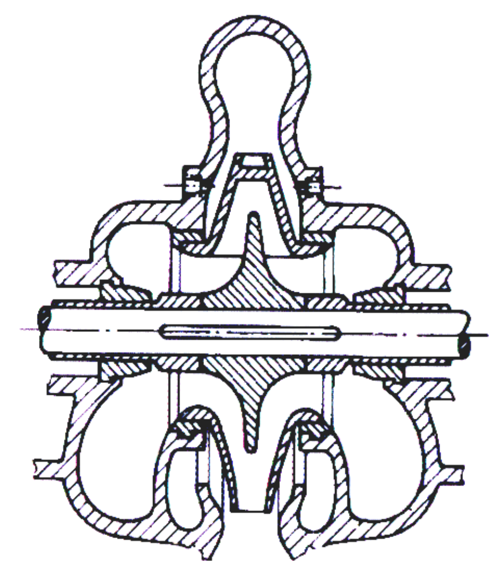




End Suction Volute



Side Suction Volute



Double Suction Volute

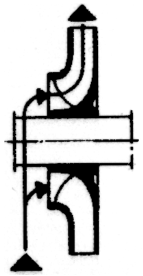
# Multistage Pumps

When the required pressure for a given capacity cannot be achieved efficiently by one impeller a number of impellers, as required, can be arranged in series on the same shaft. The result is a **multistage pump**.

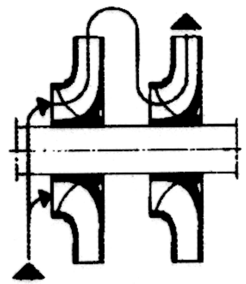
It is normal practice to install identical impellers, so that each stage gives the same pressure rise to the pumped liquid.
The **multistage pump discharge pressure is** slightly lower than **the sum of the pressures developed in each stage** due to the interstage frictional losses.

When the required capacity for a given pressure rise cannot be efficiently achieved it is possible to install **double suction or common suction impellers.**

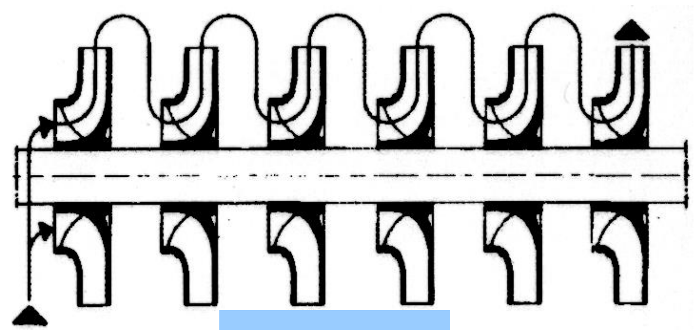
# Multistage Pumps - Casing Arrangements



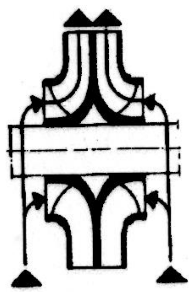
Single Stage



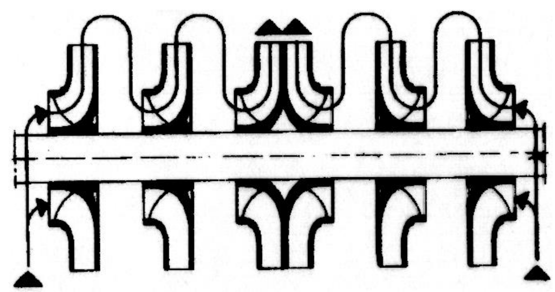
Two Stage



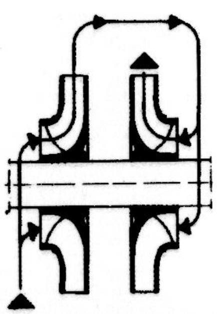
Six Stage



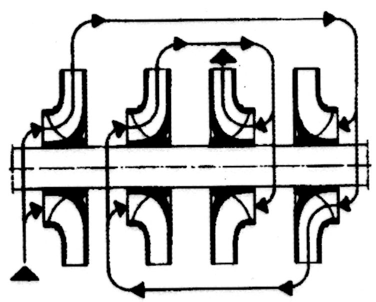
Single Stage, Double Flow



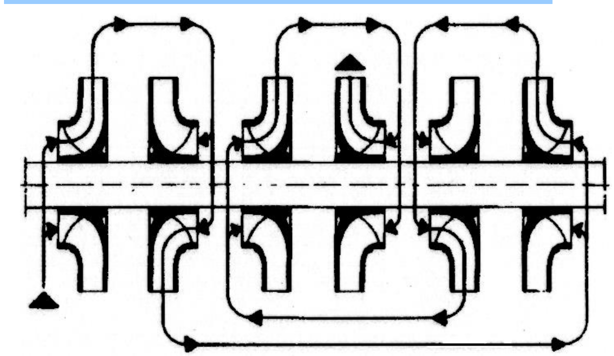
Three Stage Double Flow



Two Stage Back to Back

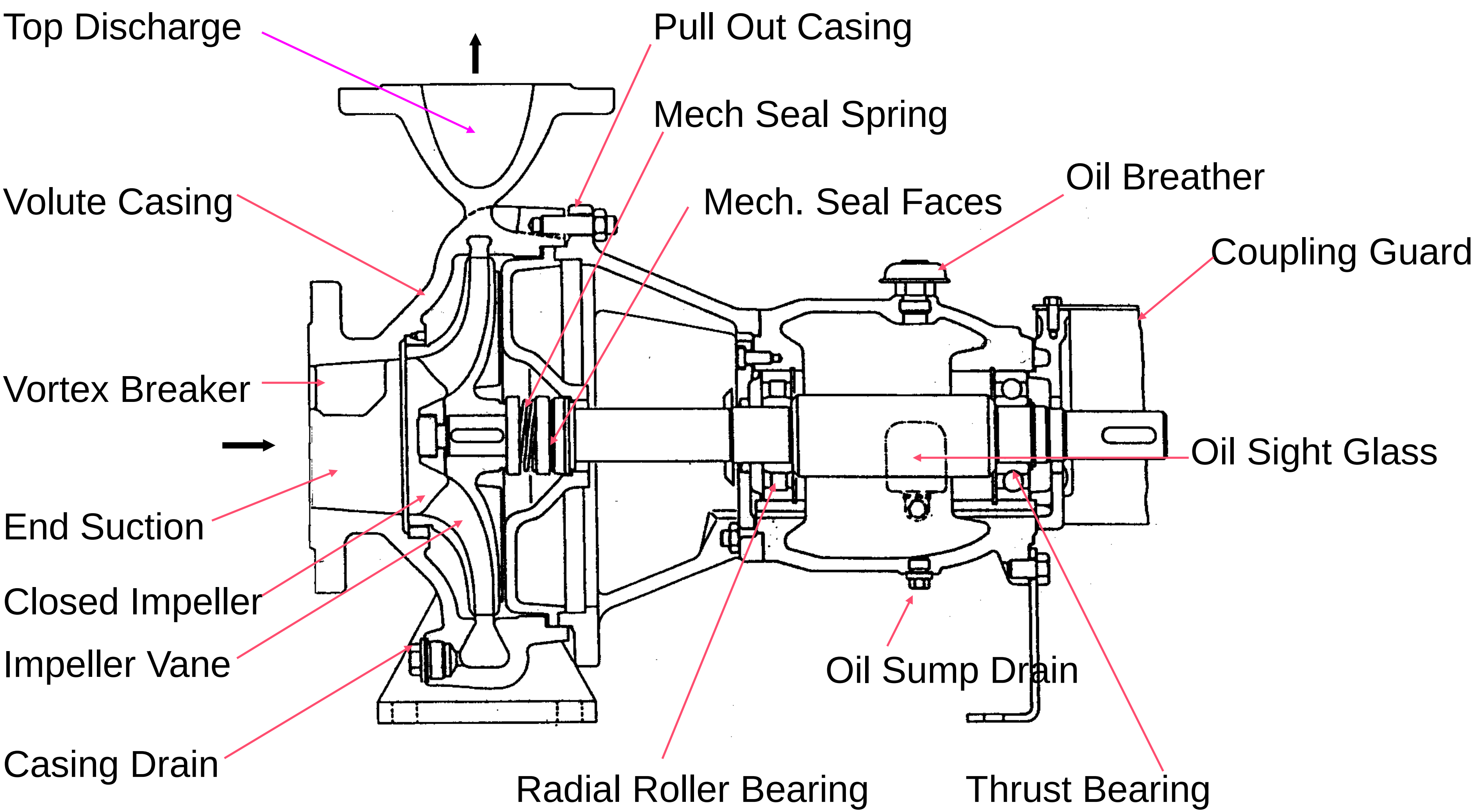


Four Stage Cross Over

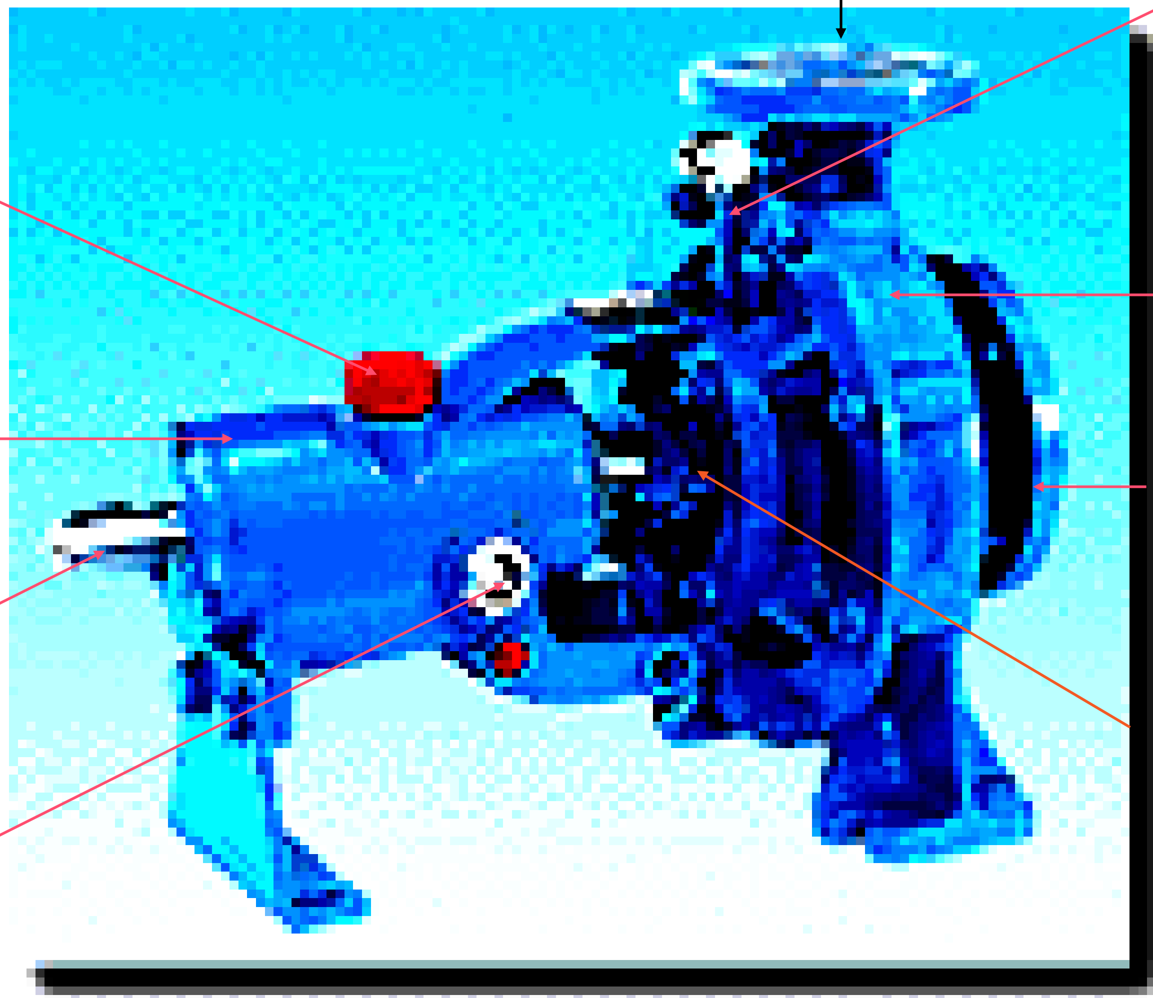


Six Stage Back to Back

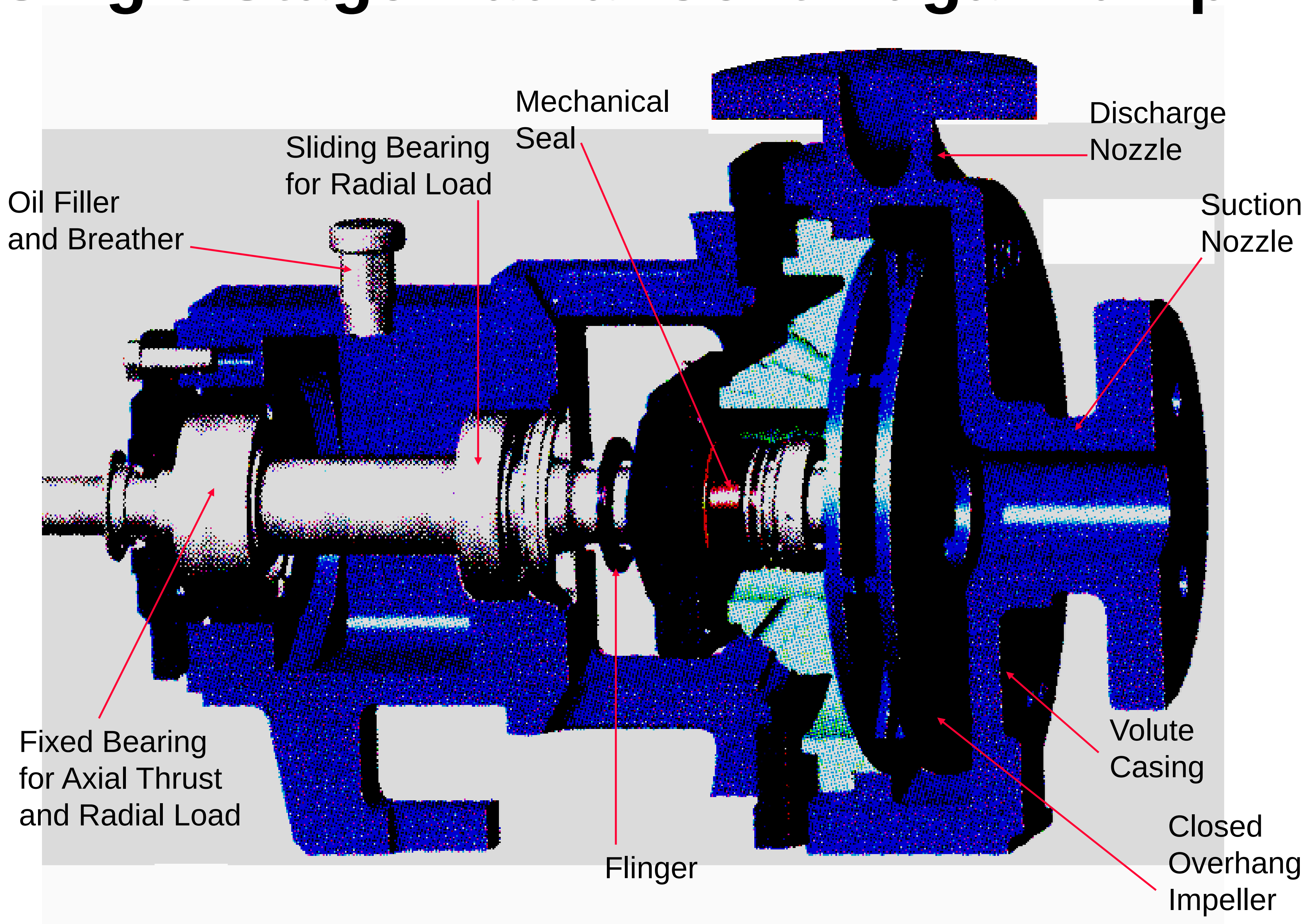
# End Suction Overhang Impeller Centrifugal Pump



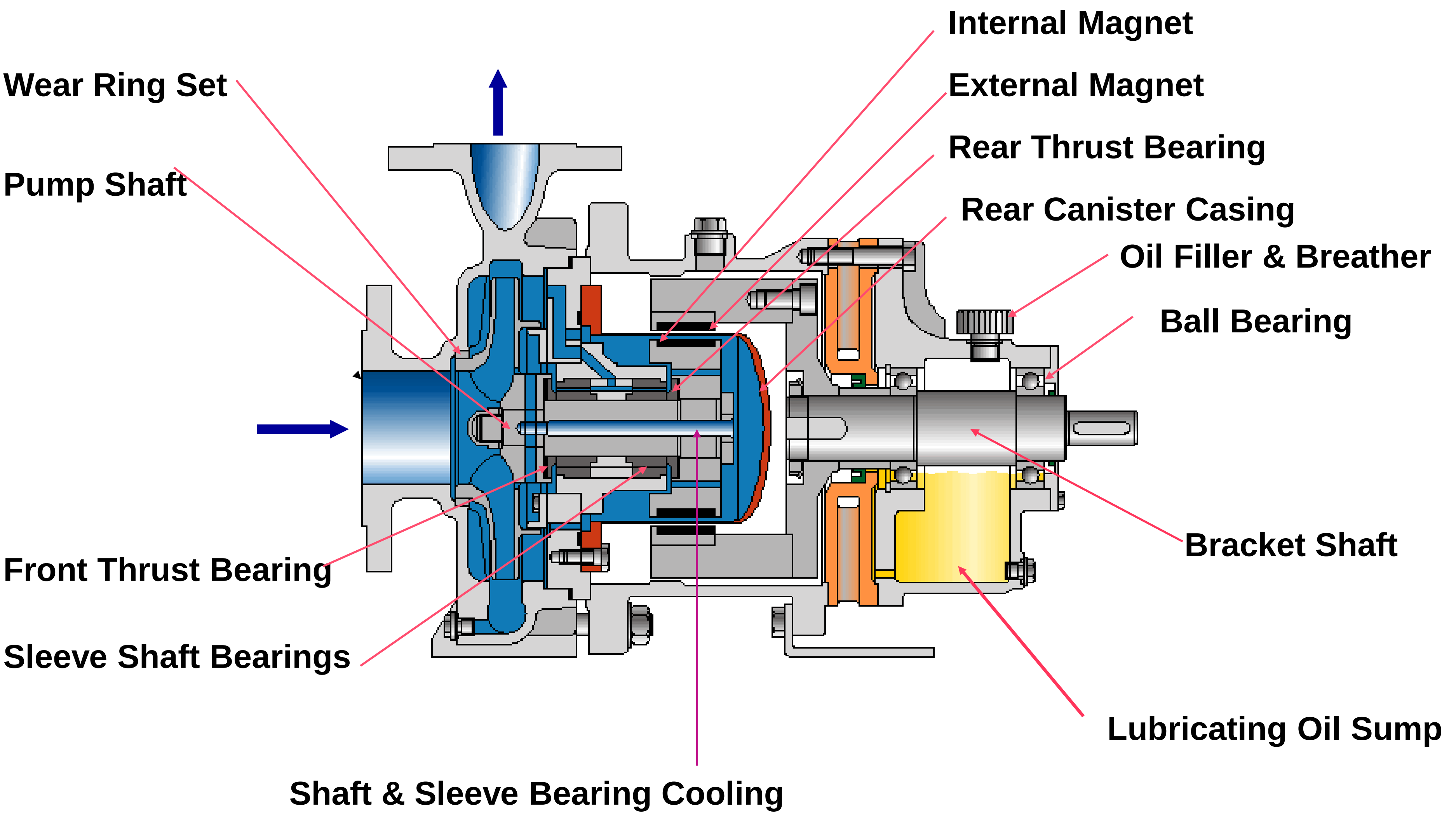
# Single Stage Overhang Impeller Pump



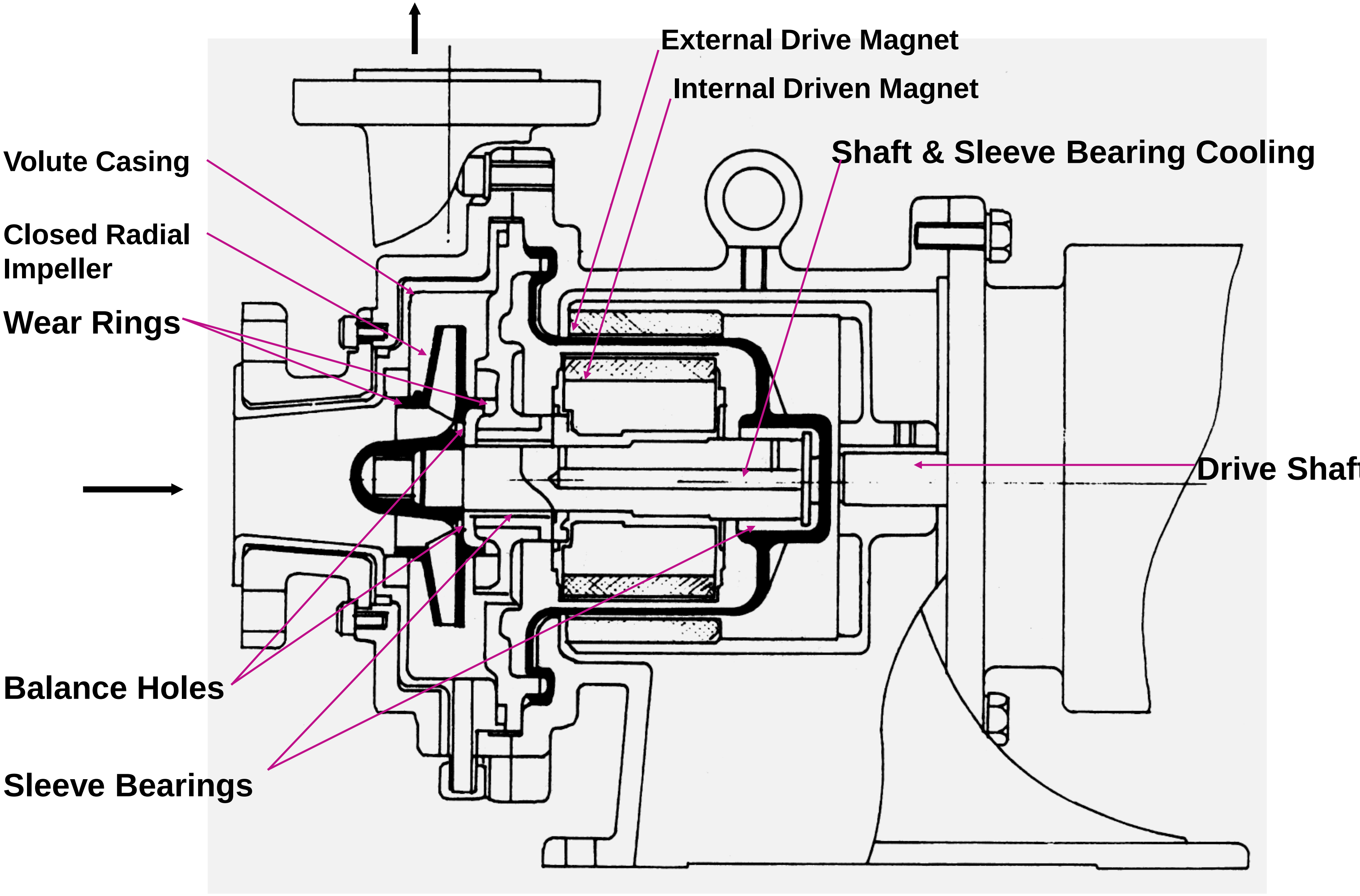
# Single Stage Radial Centrifugal Pump



# Magnetic Drive Centrifugal Pump

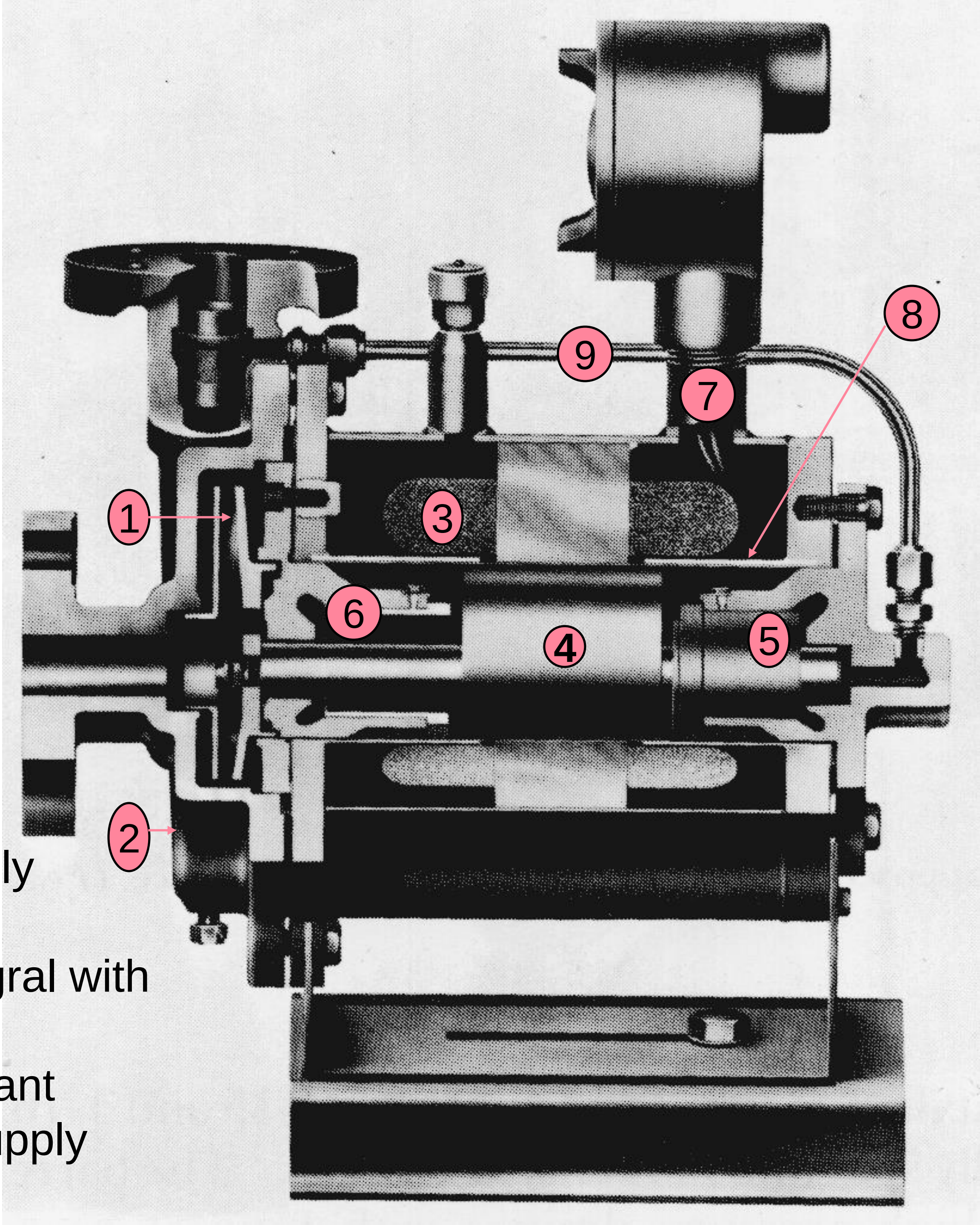


# Magnetic Drive Centrifugal Pump



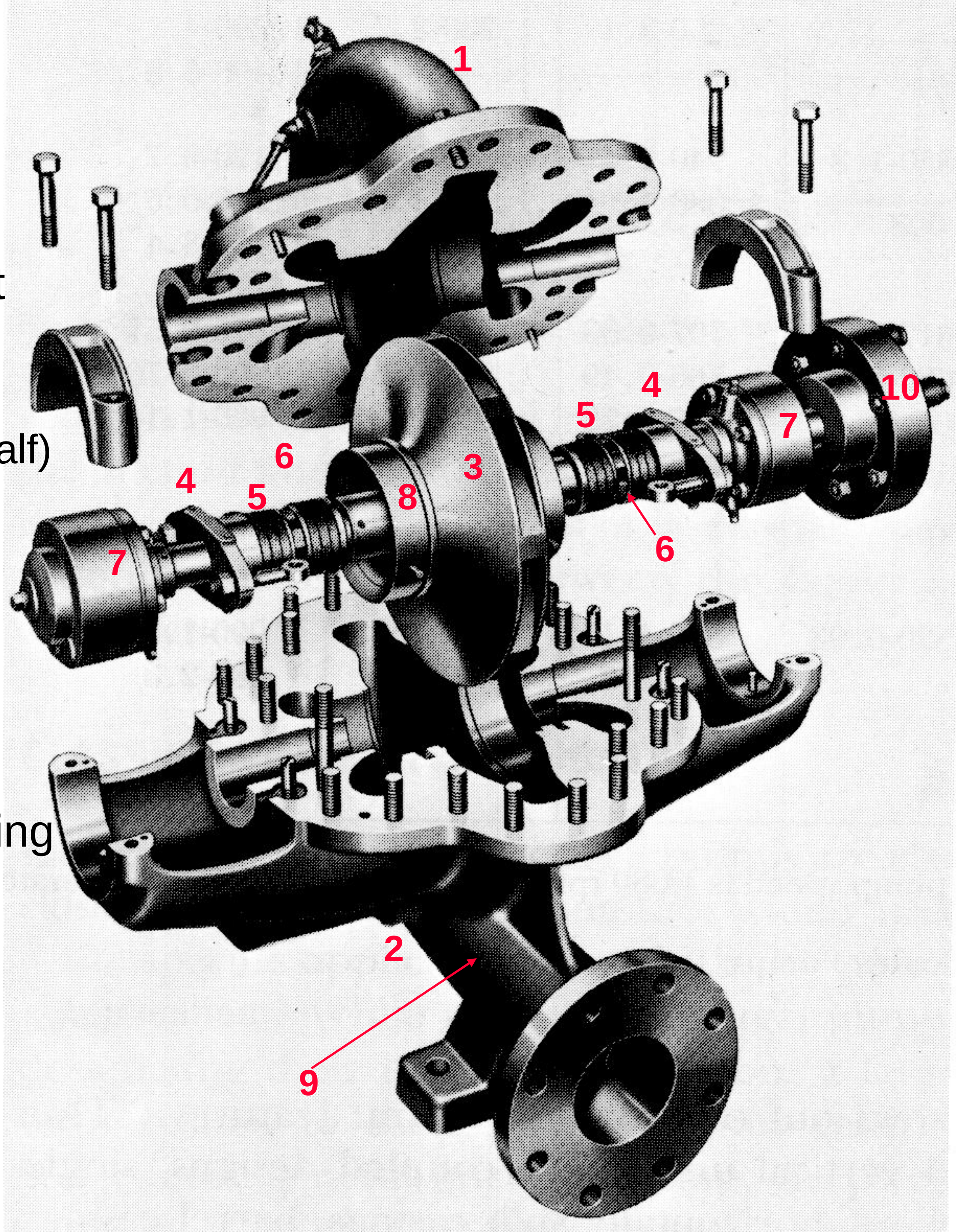
# Closed Coupled Seal-less Canned Pump

1. Radial Closed Impeller
2. Volute Casing
3. Motor Stator
4. Motor Rotor
5. Thrust Bearing (lubricated by pumped fluid )
6. Radial Bearing (lubricated by pumped fluid )
7. Electrical Supply to Stator
8. Canister - integral with pump casing
9. Coolant/Lubricant (pumped fluid) Supply

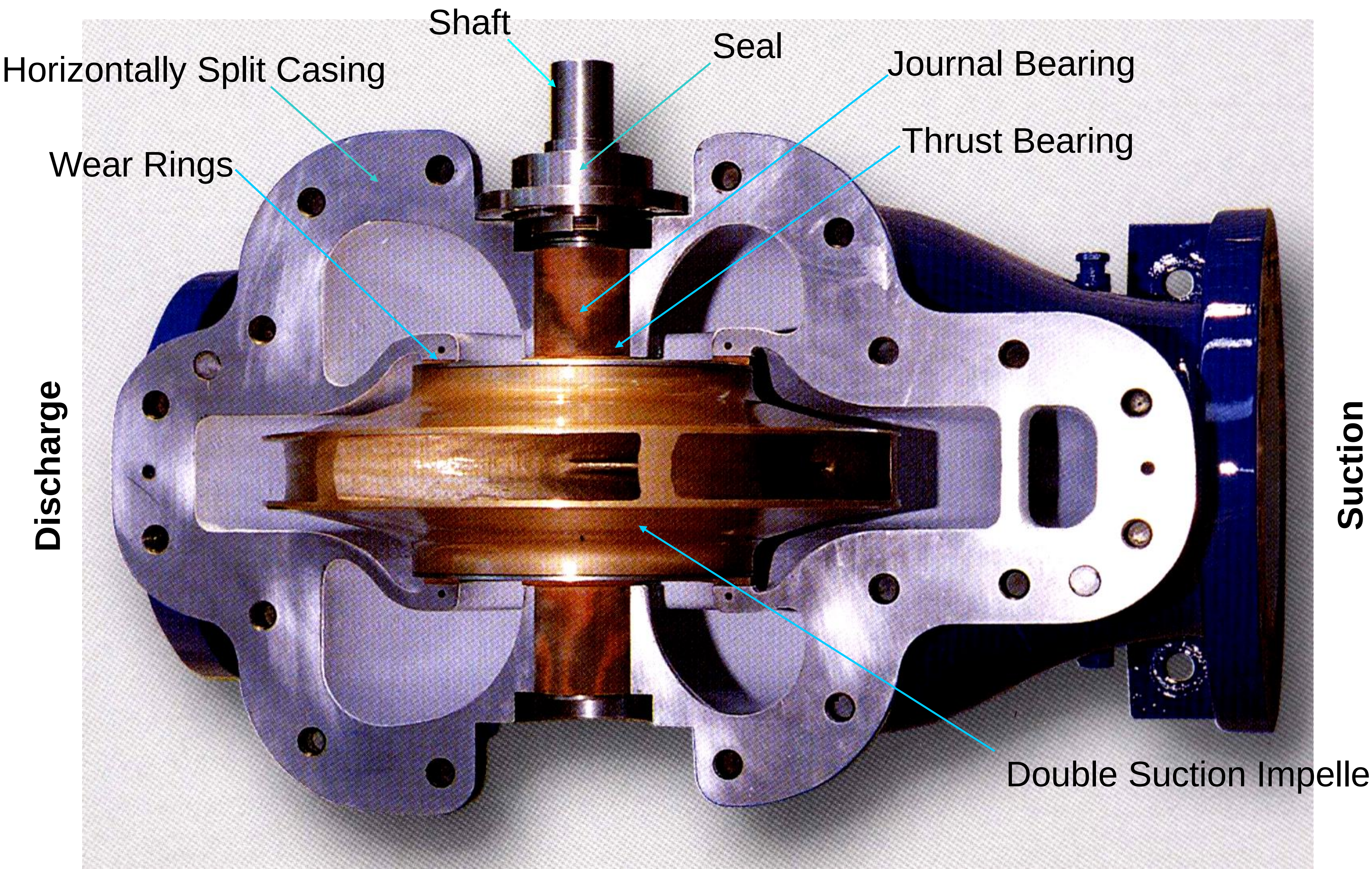


# Double Suction Single Stage Pump

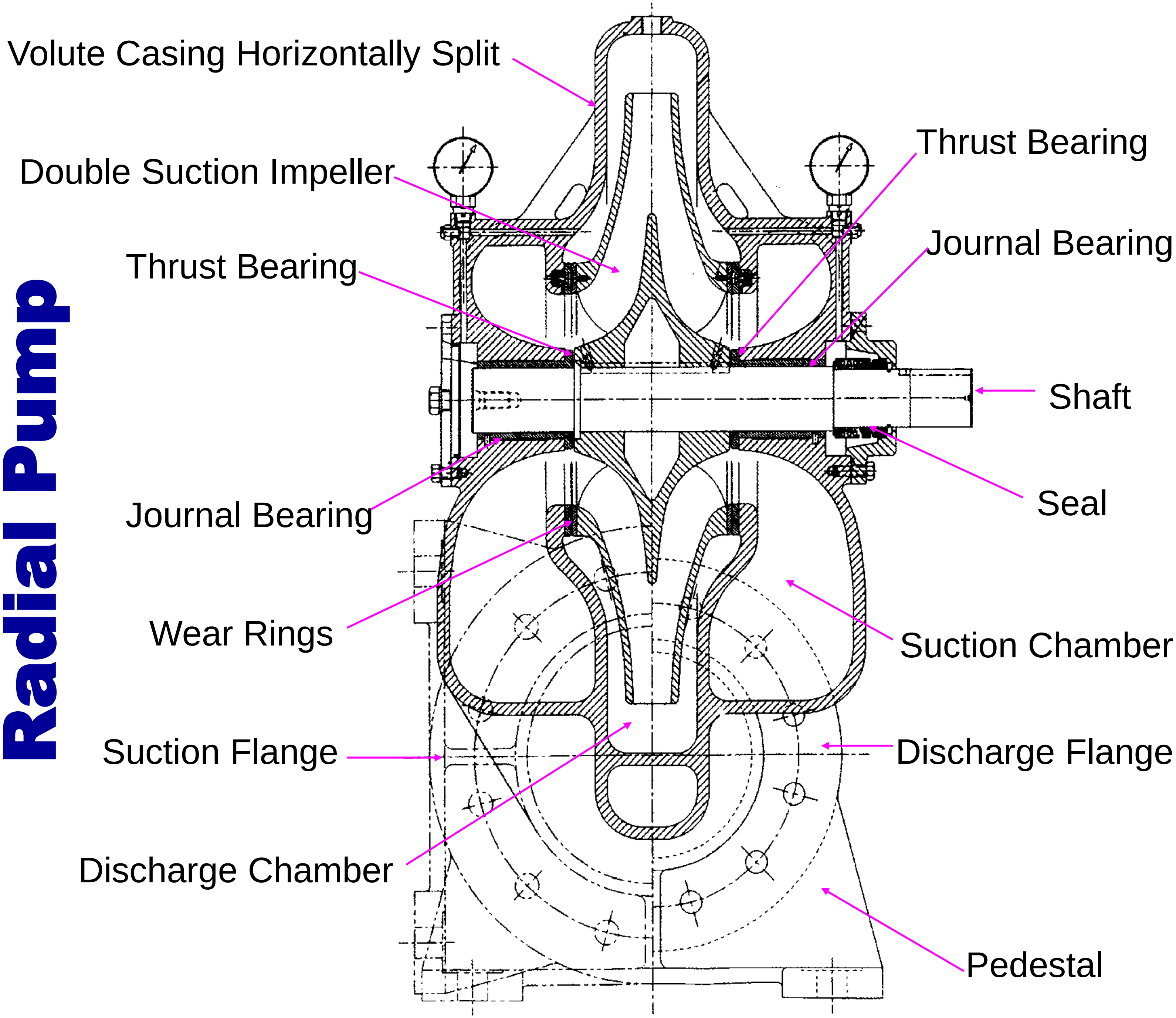
1. Horizontally Split Casing (top half)
2. Horizontally Split Casing (bottom half)
3. Double Suction Impeller
4. Gland
5. Packing
6. Lantern Ring
7. Bearing
8. Impeller Wear Ring
9. Suction Nozzle
10. Coupling



# Double Suction Single Stage Radial Pump

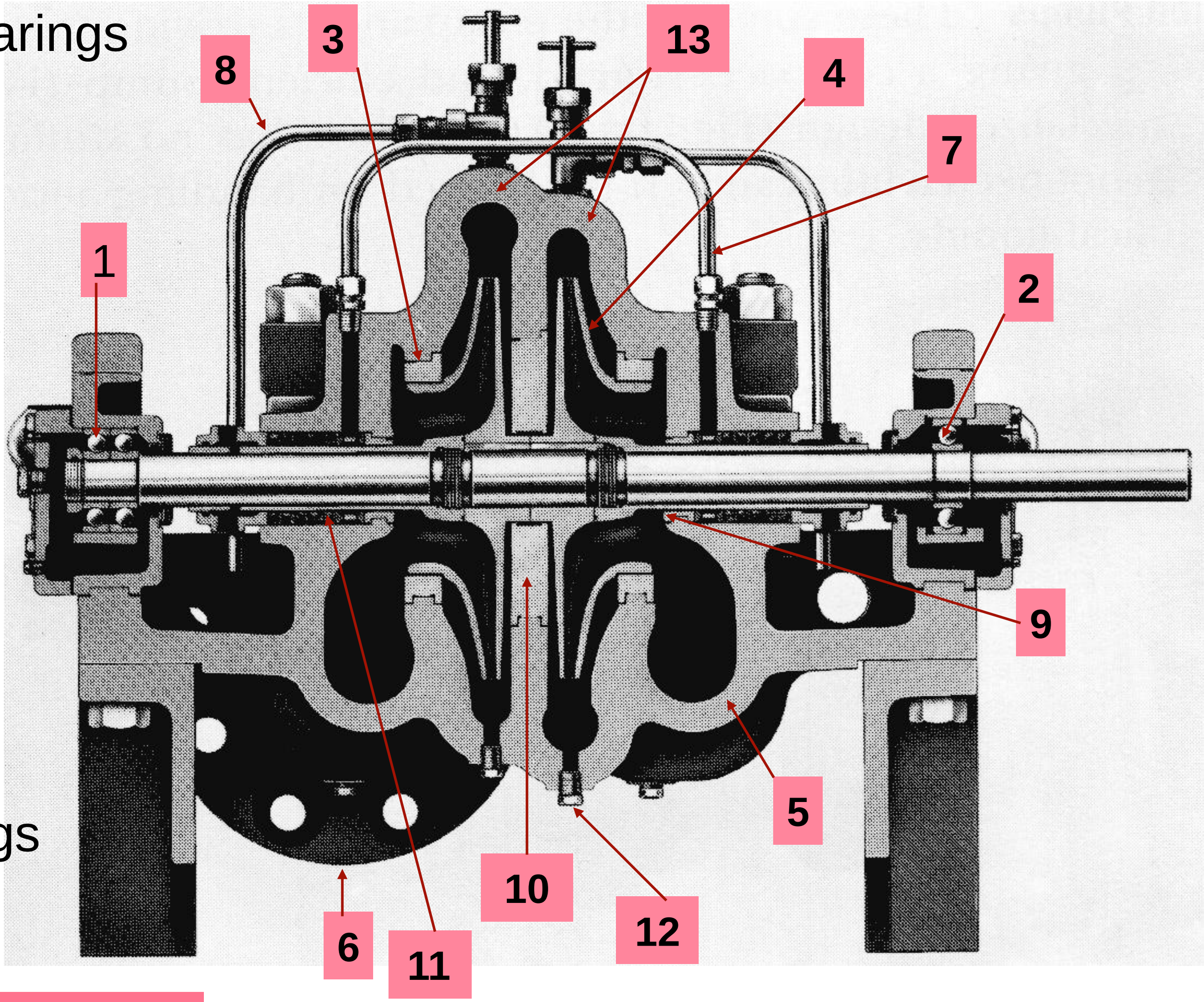


# Double Suction Single Stage Radial Pump



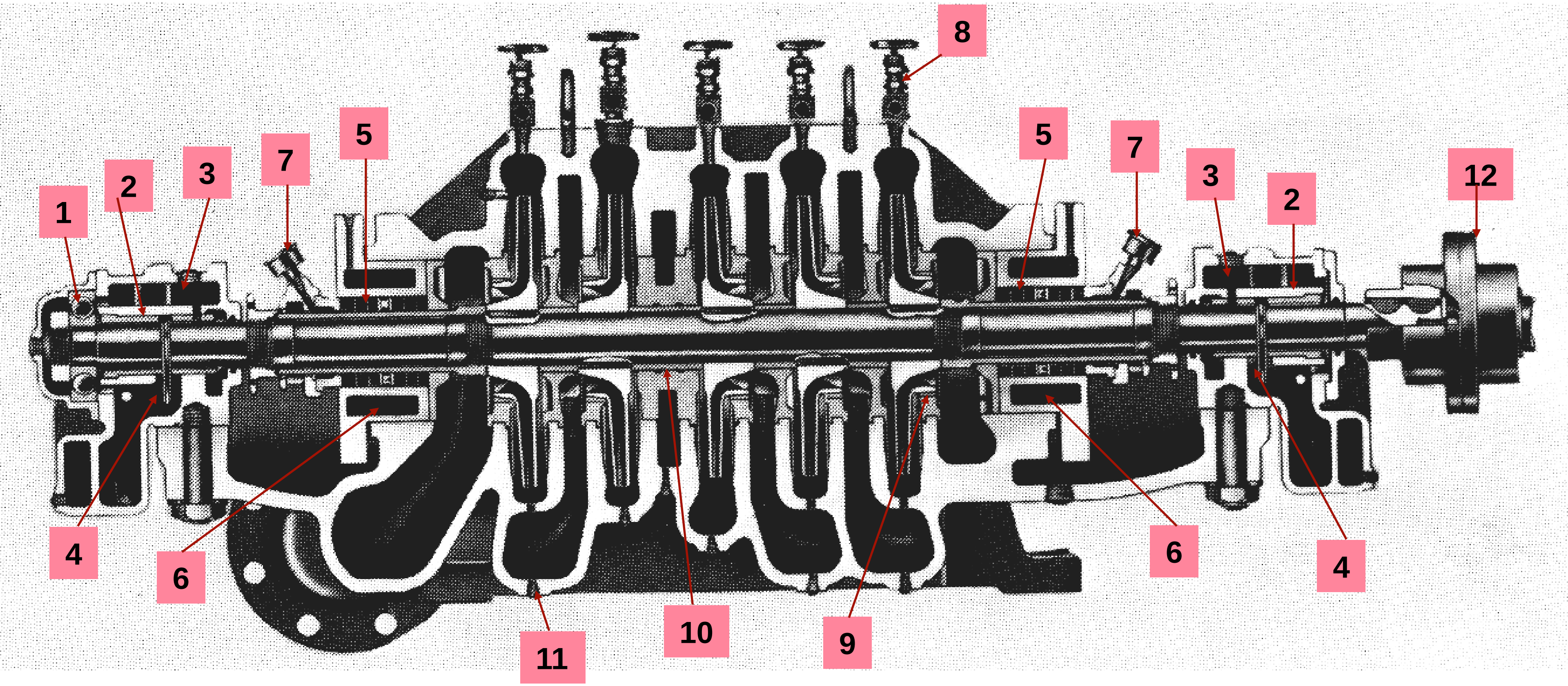
# Two Stage Radial Pump

1. Angular Contact Ball Bearings
   (thrust & radial load)
2. Radial Ball Bearing
3. Wear Ring
4. Closed Radial Impeller
5. Horizontally Split Casing
6. Suction Nozzle Flange
7. Balance Line
8. Vent Line
9. Neck Bush
10. Interstage Bush
11. Shaft Sleeves
12. Drain Plug
13. Staggered Volute Casings
    (radial thrust balance)



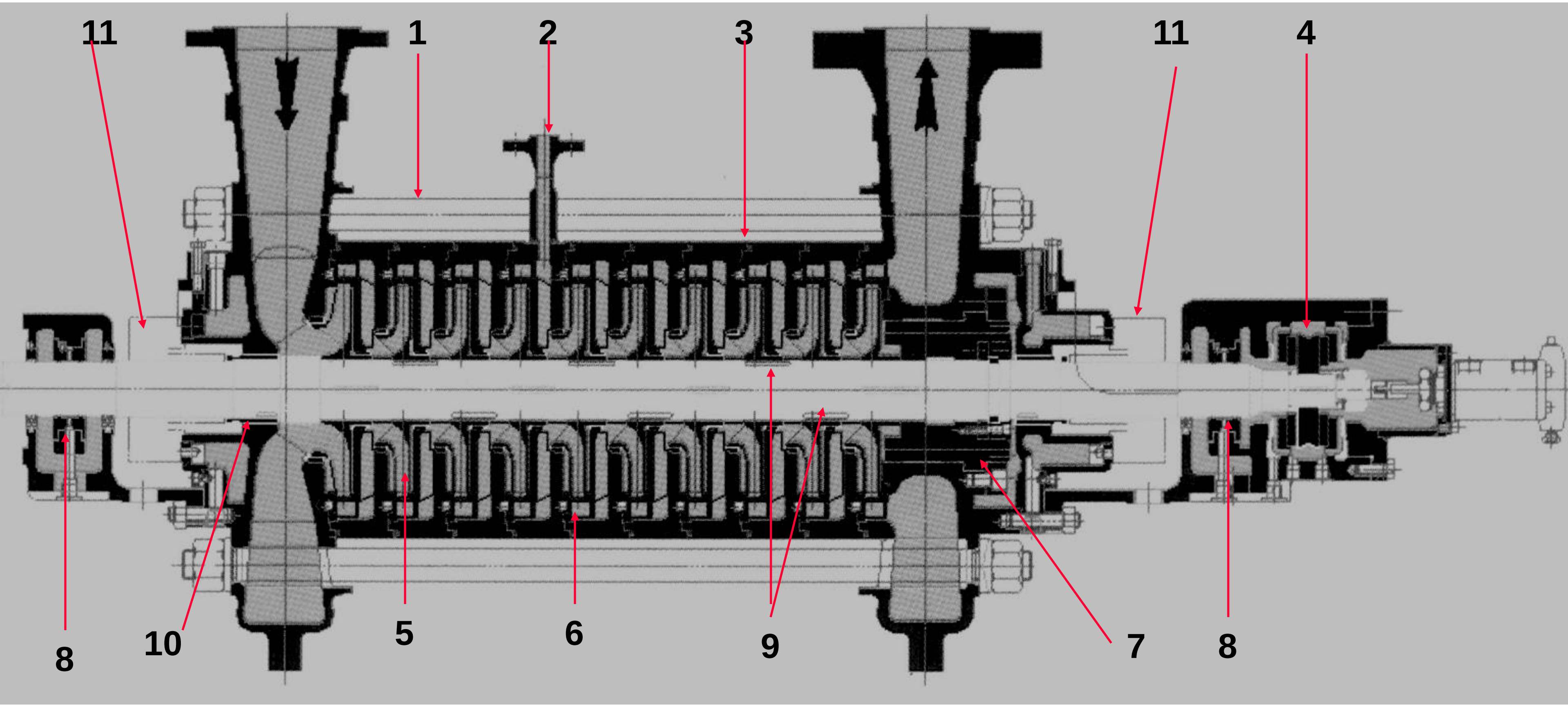
# Multistage Radial Pump

## (with opposed impellers)




1. Axial Thrust Bearing
2. Radial Load Bearing
3. Bearing Cooling Chamber
4. Oil Ring
5. Stuffing Box
6. Stuffing Box Cooling
7. Gland Grease Applicator
8. Stage Venting
9. Impeller & Casing Wear Rings
10. Interstage Bush & Bearing
11. Drain
12. Shaft Coupling

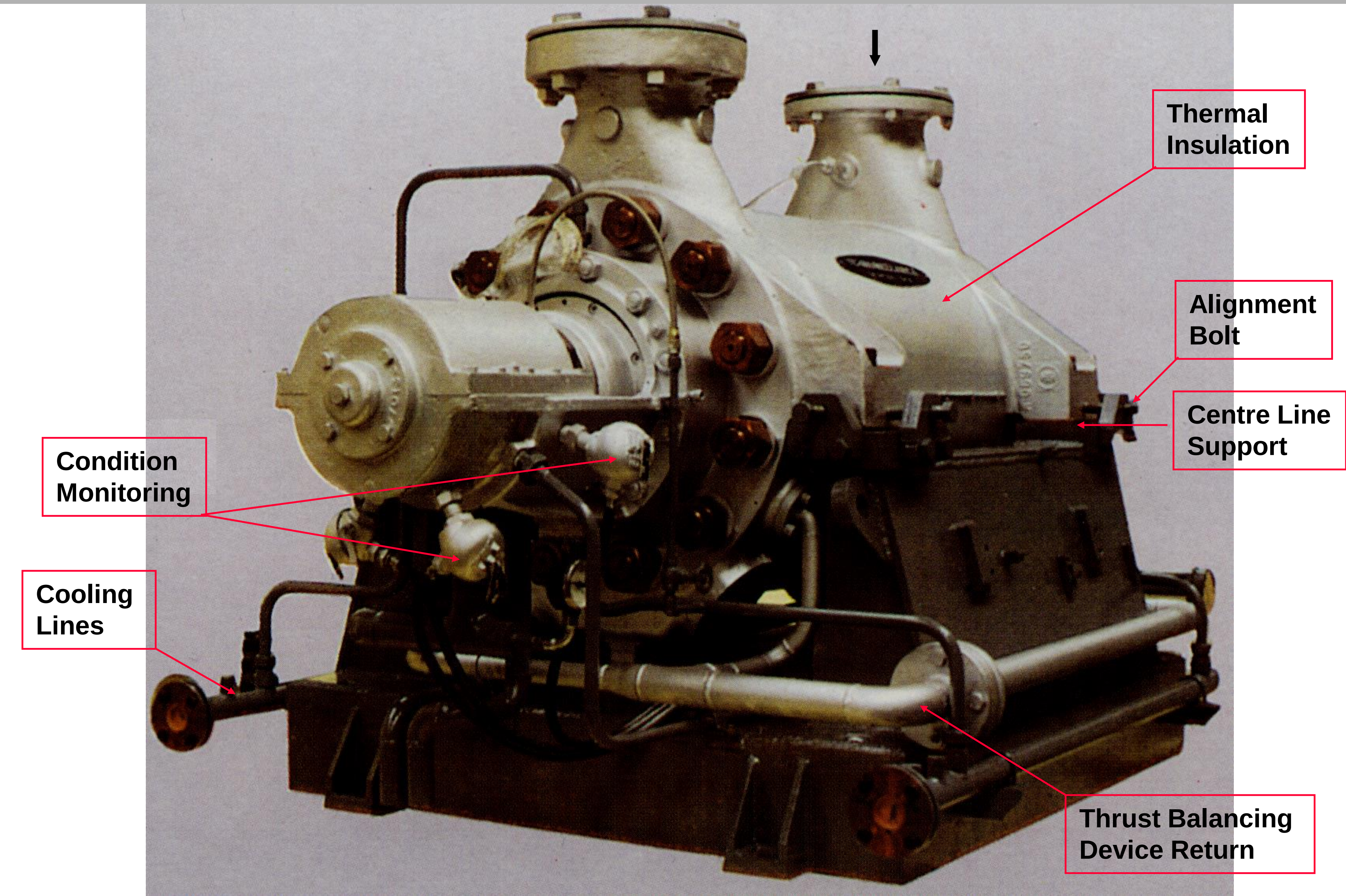
# Multistage Ring Section Pump




1. Tie Rod
2. Piloting Connection
3. Ring Section Casing
4. Thrust Bearing
5. Impeller
6. Diffuser
7. Axial Thrust Balancing
8. Radial Bearing
9. Impeller Keys
10. Shaft Sleeves
11. Mechanical Seal Position

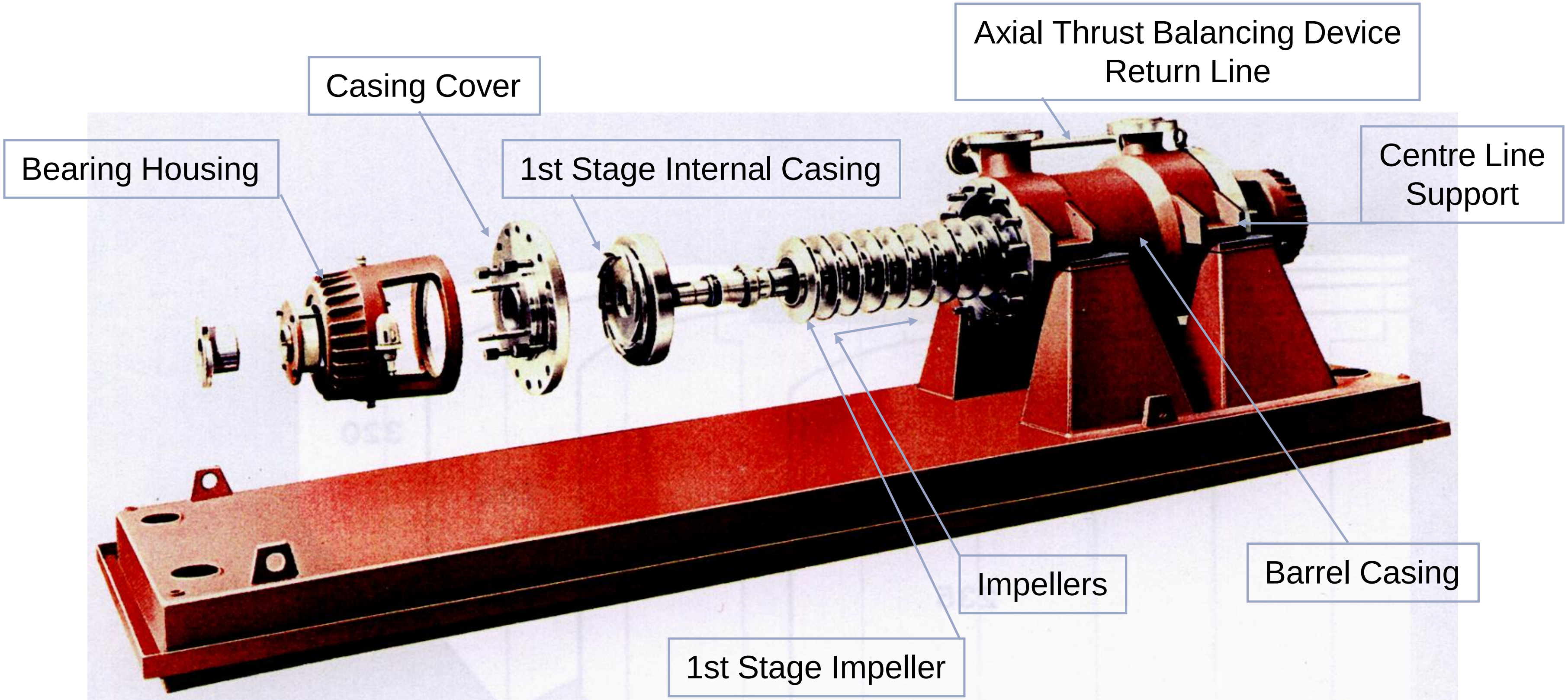
# Multistage Ring Section Pump

(ref previous slide for section of the pump)



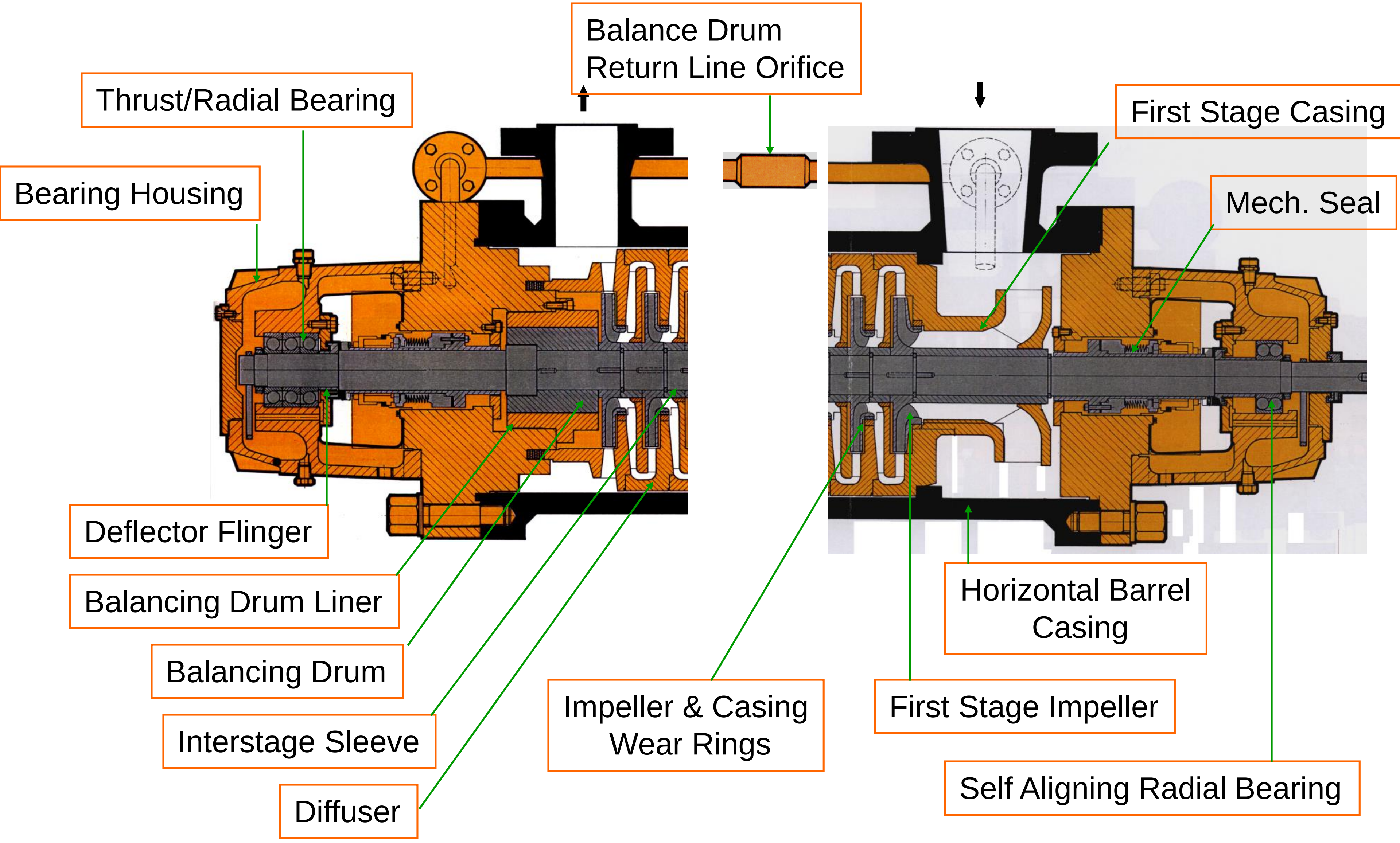
# Heavy Duty Horizontal Barrel Pump

(ref next slide)



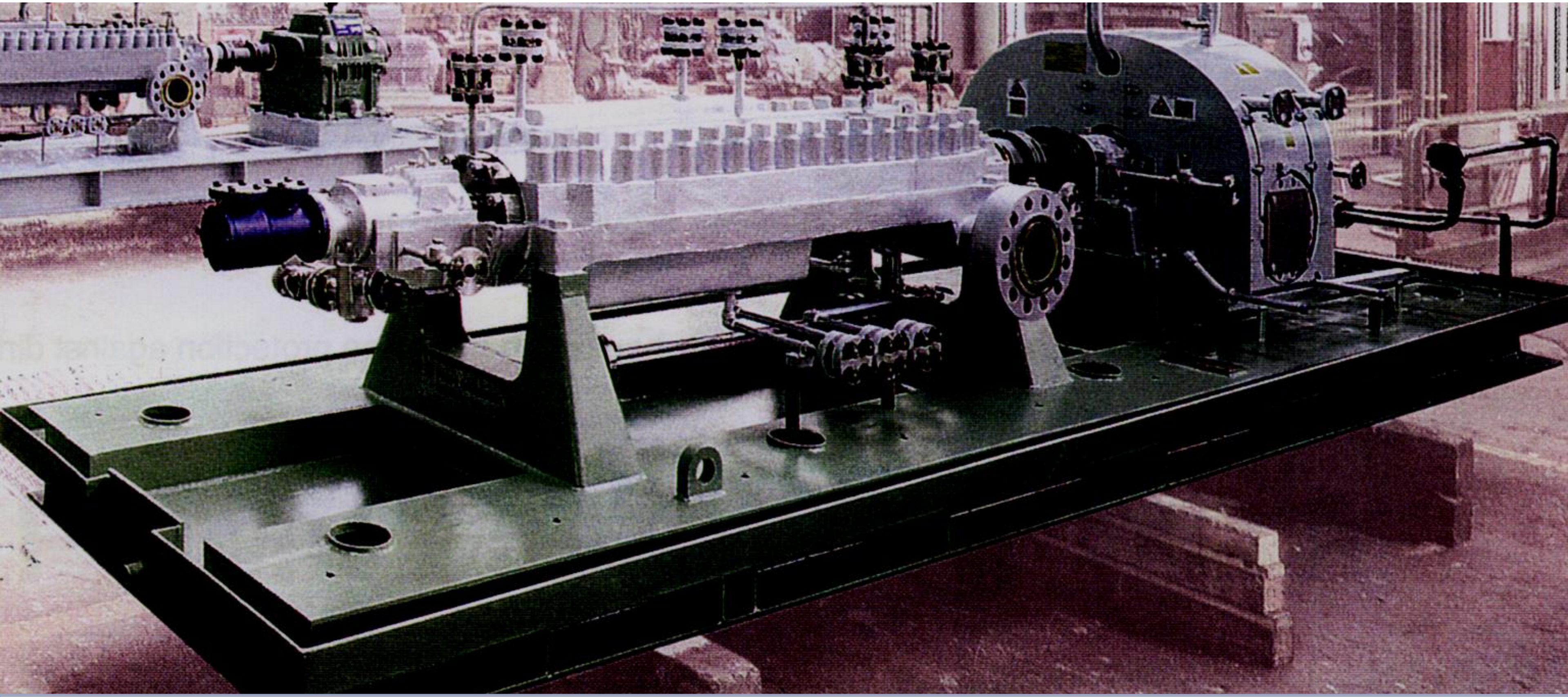

**Capacity 650m³/h, Head 1500m, Temperature 455°C, max speed 4000rpm**

# Heavy Duty Horizontal Barrel Pump
## (ref previous slide)



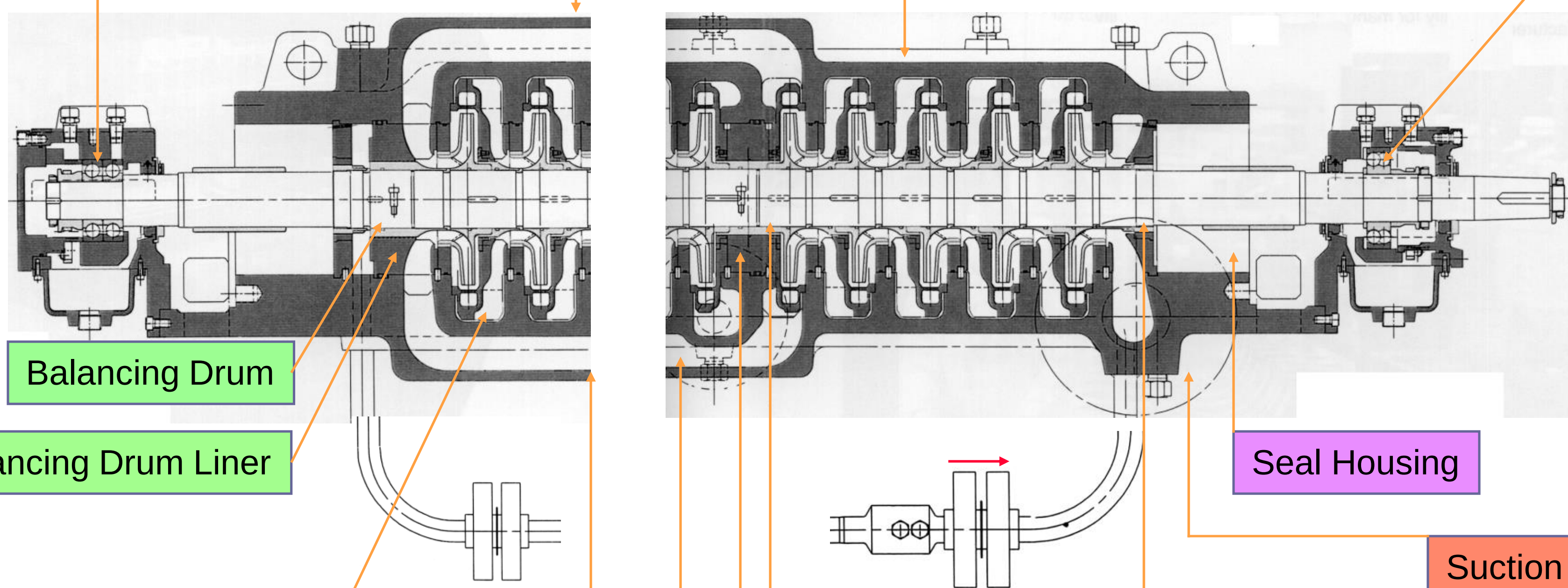
# Horizontal Multistage Axially Split Pump

**Suction & Discharge nozzles are integral part of the lower casing half, so that the pump can be dismantled without disturbing the piping connections.**
**The casing is semi-centreline supported to minimize the effects of high operating temperatures on shaft alignment - hot alignment at operating temperature has to be checked.**
**This particular model may be fitted with a shaft driven lub-oil pump**
**Pumps in hot service are fitted with a warm up system.**



**Pompe Cabbioneta, Milano**

# Section of the Horizontal Multistage Axially Split Pump (ref. Previous slide)



# Typical Petroleum Refinery Pumphouse

# Vertical Centrifugal Pumps (i)

**Vertical Centrifugal Pumps are preferred in cases of:**

- **Low Net Positive Suction Head -** lowering the pump suction increases the Net Positive Suction Head Available {NPSH(A)}.
- **Limited Floor Space** e.g. marine applications
- **Requiring the motor to be at a higher elevation when the pump has to be low closer to the pumped liquid.**

# Vertical Centrifugal Pumps (ii)

**Vertical Centrifugal Pumps are those which have their shaft arranged in the vertical.**

**There are two types of vertical centrifugal pumps**

- **Dry Pit Pumps**
- **Wet Pit Pumps**

**The Dry Pit Pumps have a vertical shaft and their suction and discharge are connected to the piping system in the same way as the normal horizontal pumps.**

**The Wet Pit Pumps have have a vertical shaft and their suction is designed for submerged operation in a pit with the operating liquid and their discharge is connected to the piping system.**

# Vertical Dry Pit Pumps (i)

Many vertical dry pit pumps **are basically horizontal pump designs** with minor modifications to adapt them for the vertical installation.

The vertical design is preferred in cases of limited floor space as in marine applications (ships, oil platforms etc). In cases when the pump has to mounted low a vertical pump allows higher more convenient mounting of the motor.

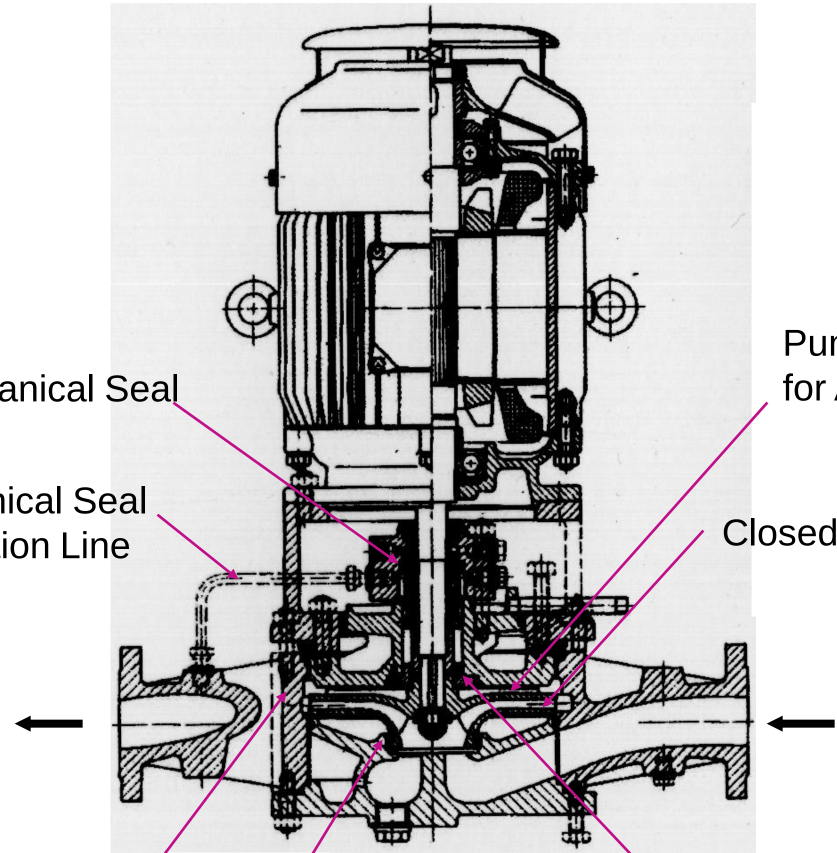
# Vertical Dry Pit Pumps (ii)

For process vertical dry pit pumps a flexible coupling connects the pump and motor shafts leaving a space between the pump seal and motor to safeguard the motor form seal leakage effects.The pump and motor have their own thrust bearings.

The major parts of the vertical pumps are usually made to allow the pump and motor to be assembled **without the need of pump/motor shaft alignment.**

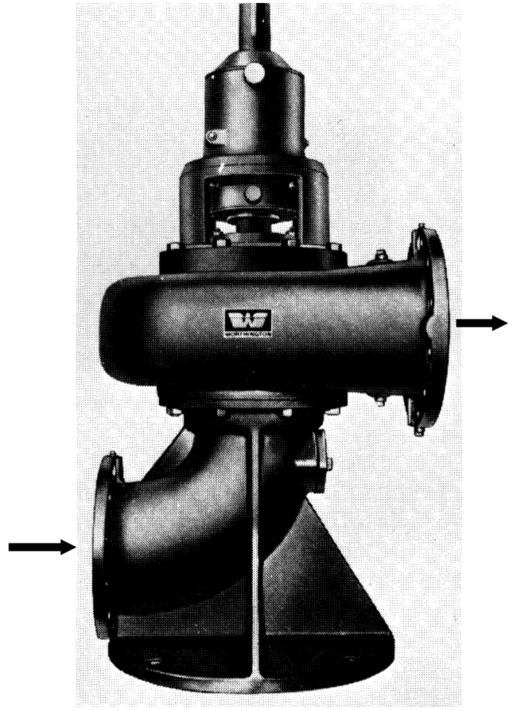
The bearings of smaller pumps are usually grease lubricated while those of larger pumps are usually oil lubricated. However the grease lubricated bearings in vertical pumps simplify the problem of having to retain the liquid lubricant in a housing with the shaft passing vertically through it.

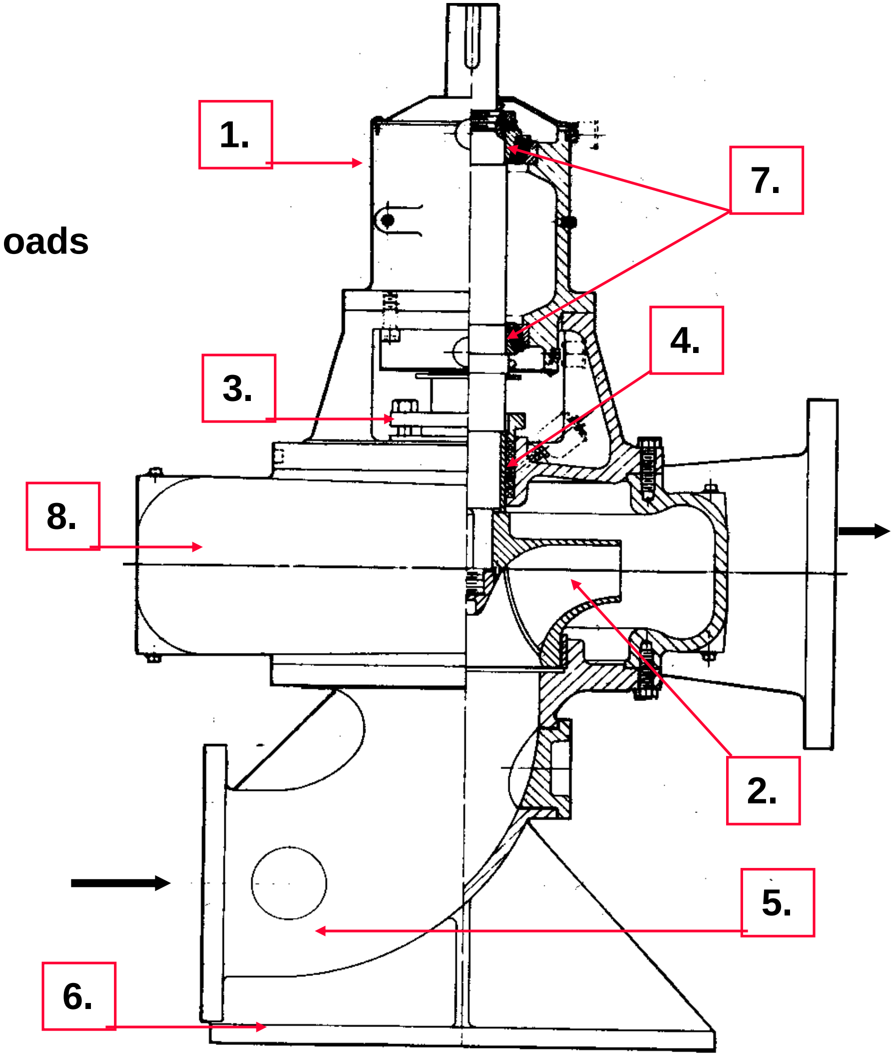
# Centrifugal In Line Pump



# Vertical Dry Pit Pump

1. Bearing Housing
2. Closed Impeller
3. Gland
4. Packing
5. Suction Elbow
6. Soleplate
7. Taper Roller Bearings for radial & axial loads
8. Volute Casing





# Vertical Wet Pit Pumps (i)

The vertical wet pit pumps have the major advantage of having low NPSH(R) requirements due to the low suction. They also have a very small foot print, and especially by comparison the multistage vertical type.

The vertical wet pit pumps can be grouped as follows:-

**Vertical Turbine Pumps**
**Vertical Volute Pumps**
**Vertical Propeller Pumps**

The **Turbine Pumps** are characterized by a multistage design and the use of a diffuser in the casing.Semi-open or closed impellers are used. The semi-open impeller type require care in ensuring the correct location of the impellers with respect to the casing to secure the correct casing/impeller clearances. For this purpose the vertical position of the shaft can be adjusted with the pump in situ at base plate level.

# Vertical Wet Pit Pumps (ii)

The vertical turbine pumps can be provided either with a shaft that is exposed to the pumped fluid or with an enclosed shaft that is protected from the pumped fluid . In the first case the bearings are lubricated and cooled by the pumped liquid and in the second case the bearings are greased or lubricated by a suitable arrangement from the pump base plate level.

The **Vertical Volute Pumps** are provided with a volute casings and impellers according to the operating conditions and liquids handled e.g. closed, semi-open, non-clogging etc.

The radial shaft bearings are either greased for life or are greased by means of grease injection with the injection points at the baseplate. These injection points are connected by tubing to the bearings. In other cases journal sleeve bearings can be used for the long shaft. These bearings can either be lubricated by the pumped liquid or they can be of the non lubricated type such as p.t.f.e. (Teflon) or graphite.

# Vertical Wet Pit Pumps (iii)

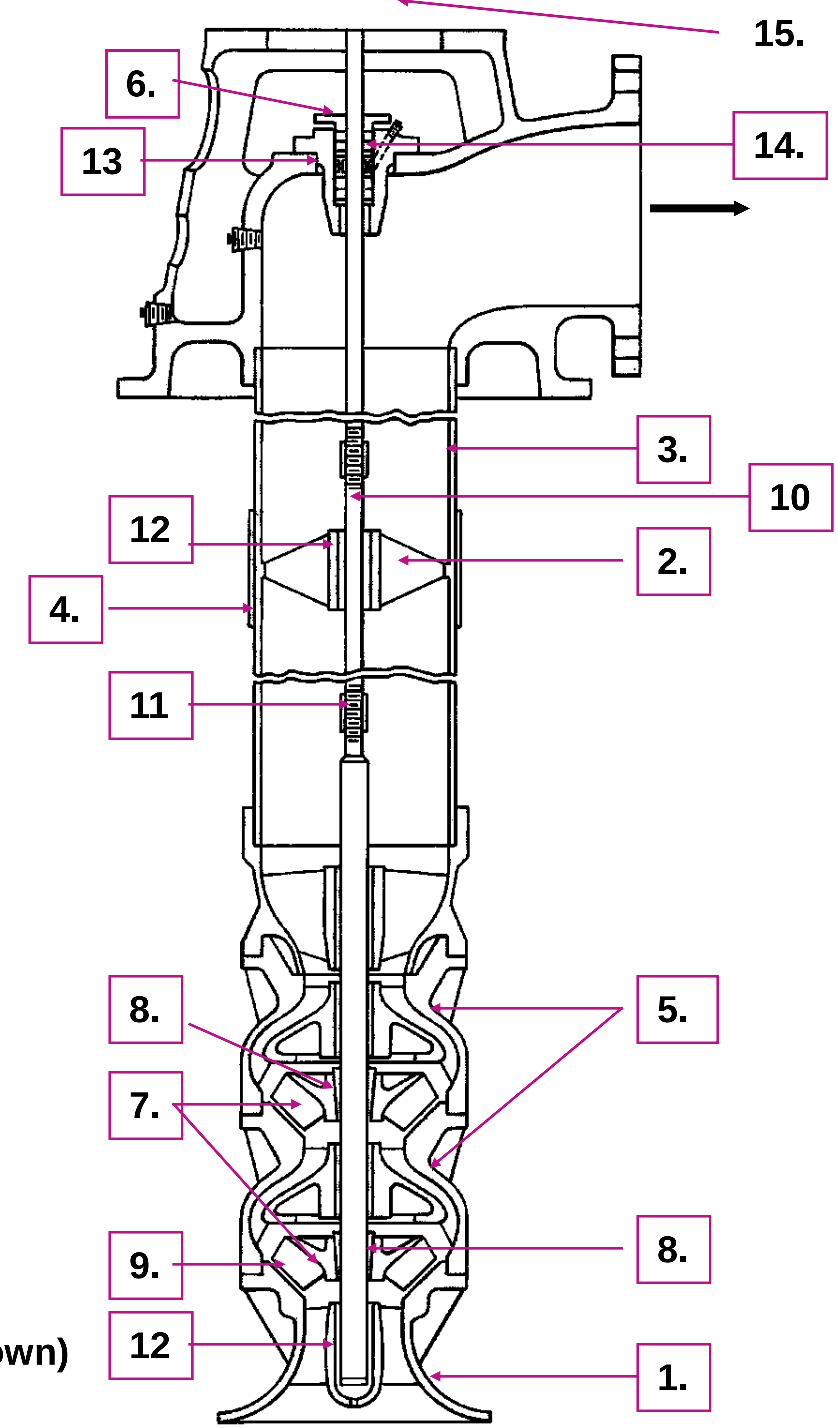
The **Vertical Propeller Pumps** are pumps with axial impellers (propellers) when large flows at low heads are required. Mostly they are installed in open sumps. Their construction is similar to the other vertical pump types. In some large pumps the impeller, diffuser and shaft assembly can be removed without disturbing the column and pipe assembly, for ease of maintenance.

Vertical wet pit pumps due to their relatively large axial dimension and heavy weight need sufficient free headroom for withdrawal and for accommodating the overhead lifting facility where this is fixed or mobile.

To avoid the necessity of lifting the large mass of the entire pump assembly some designs allow withdrawal of the shaft assembly from the top without disturbing the the column assembly.

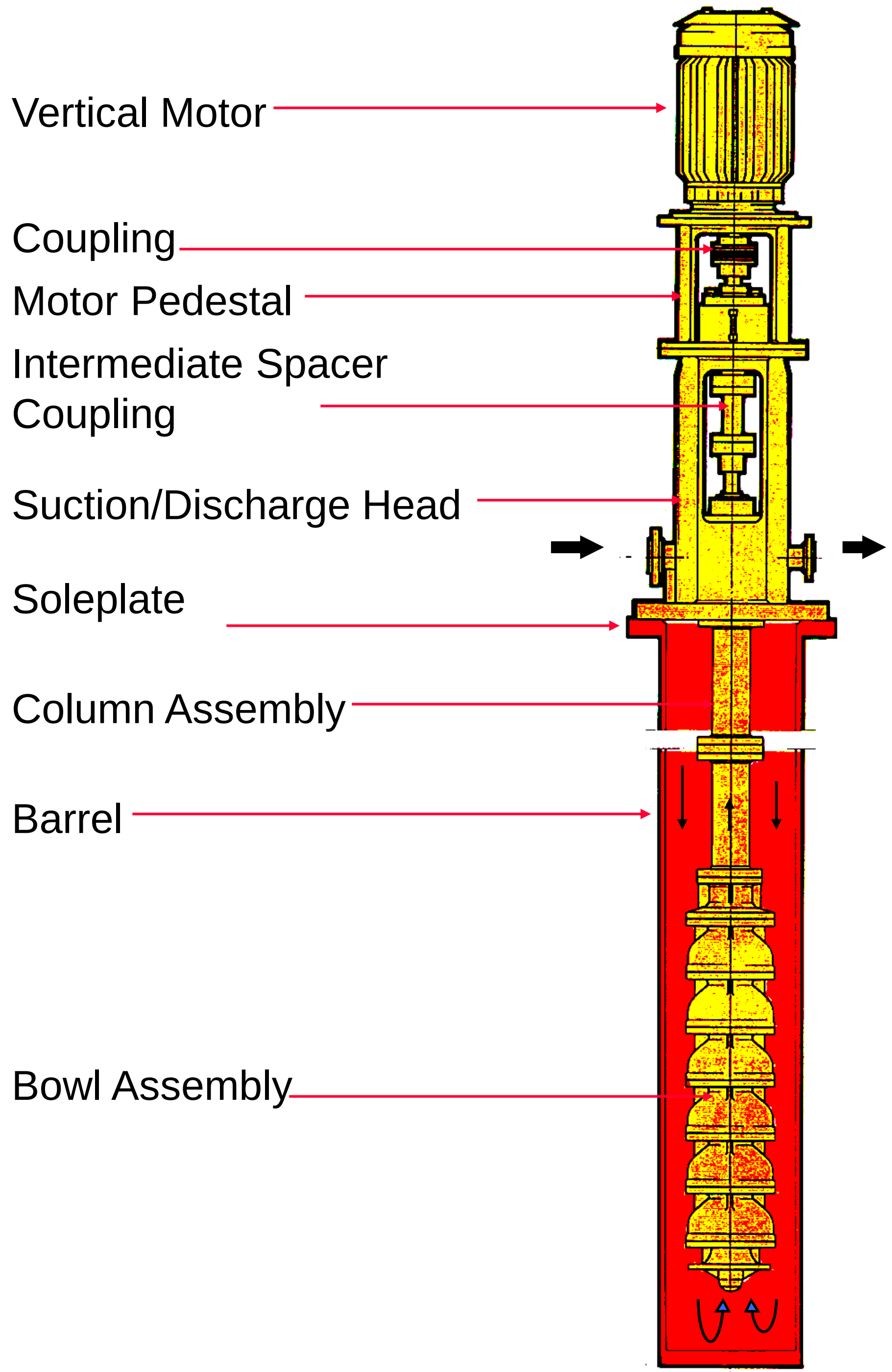
# Vertical Wet Pit Turbine Pump

1. Bell Mouth (Suction)
2. Bearing Retainer
3. Column Pipe
4. Column Pipe Coupling
5. Diffuser
6. Gland
7. Mixed Flow Impeller
8. Impeller Lock Collet
9. Impeller Vane
10. Line Shaft
11. Line Shaft Coupling
12. Sleeve Bearing
13. Stuffing Box
14. Packing
15. Thrust Bearing (not shown)



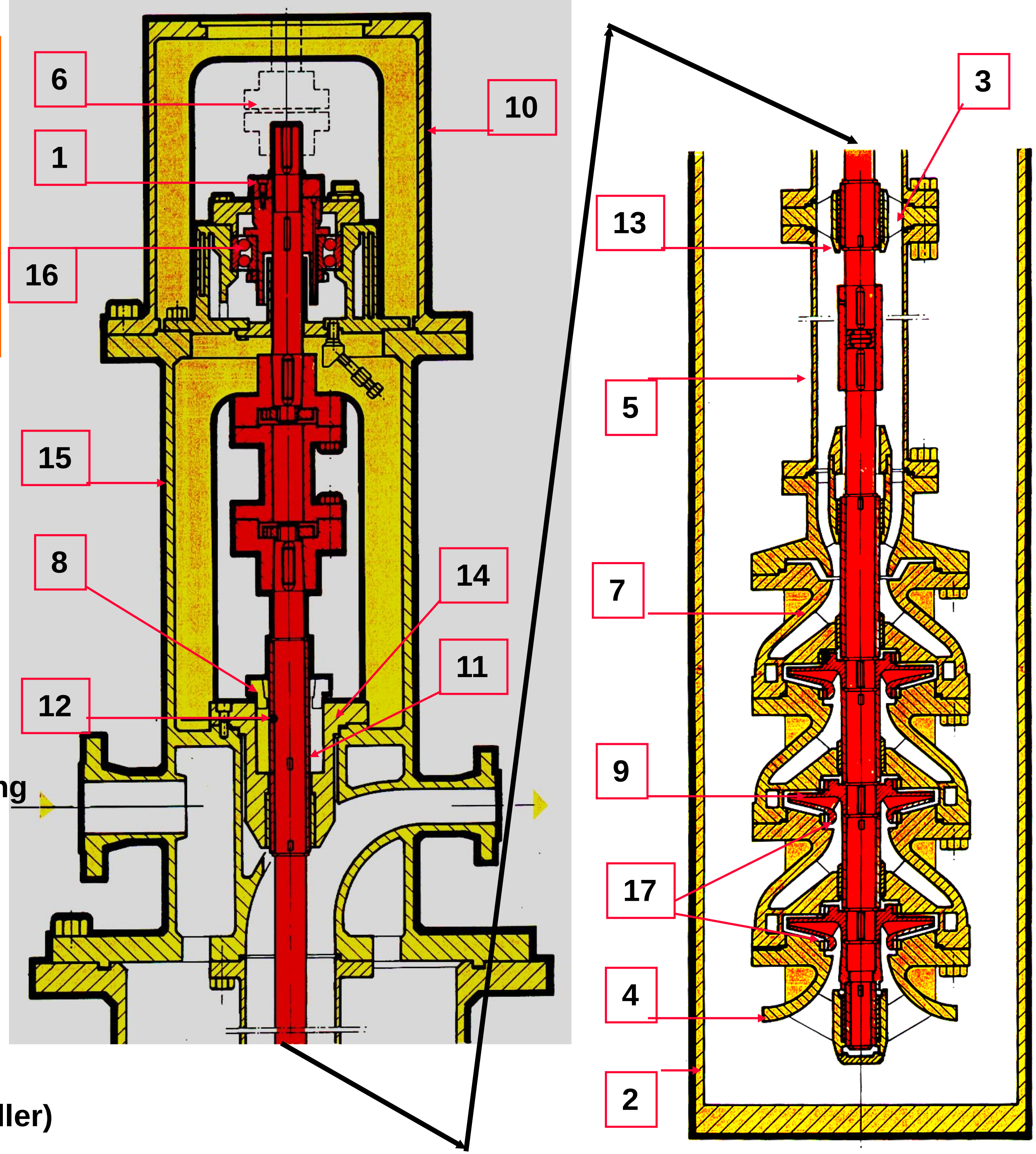
# Vertical Turbine Barrel Pump

Ref Section of the Pump on the following Slide



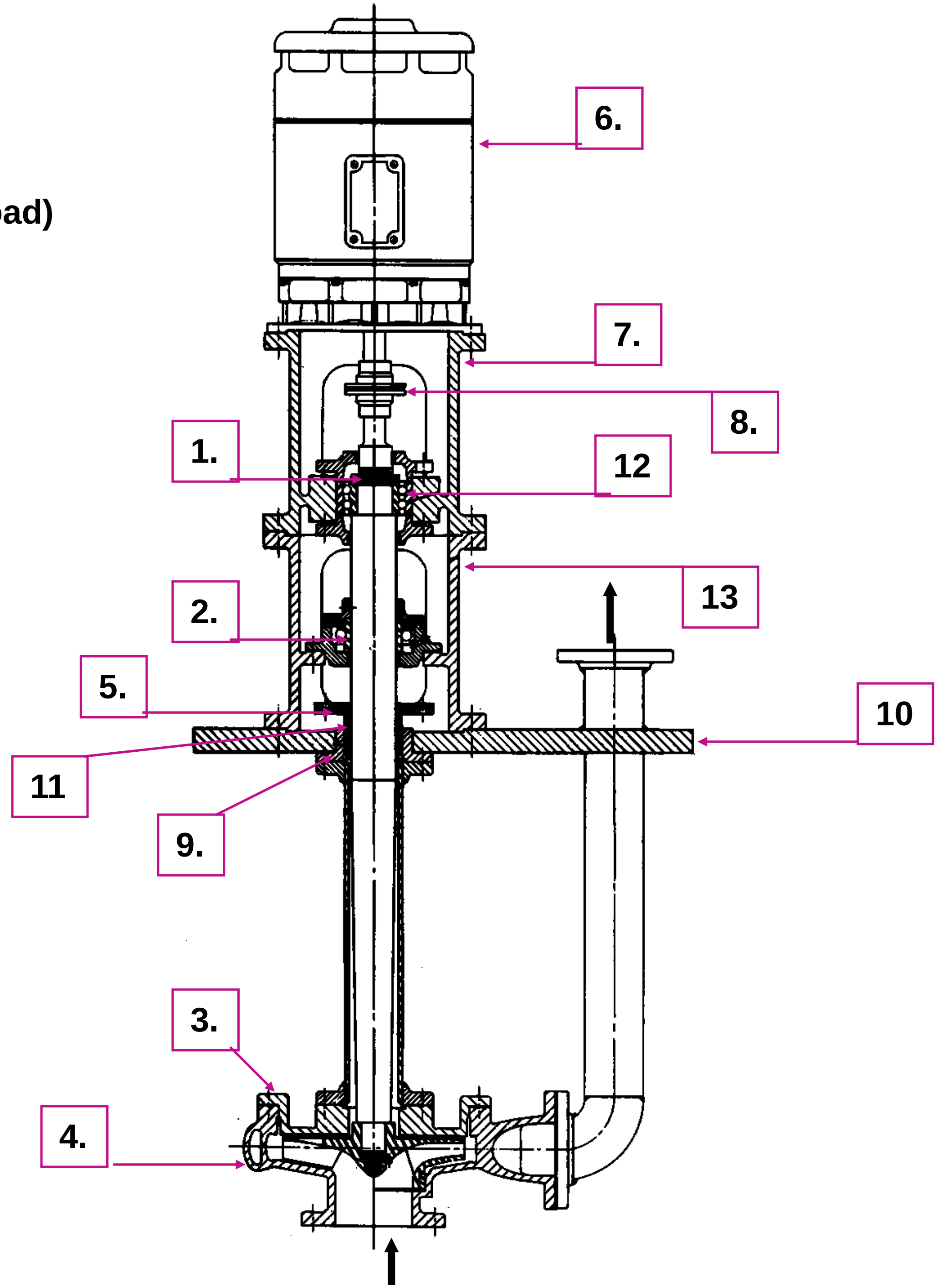
# Section of the Barrel Pump on the Previous Slide

1. Adjuster Nut
2. Barrel
3. Bearing Sleeve Carrier
4. Bellmouth Suction
5. Column
6. Coupling
7. Diffusion Casing
8. Gland
9. Impeller
8. Intermediate Spacer Coupling
9. Line Shaft Coupling
10. Motor Pedestal
11. Packing
12. Shaft Sleeve at Packing
13. Sleeve Bearing
14. Stuffing Box
15. Suction/Discharge Head
16. Thrust Bearing
17. Wear Rings (casing & impeller)



# Vertical Wet Pit Volute Pump

1. **Adjuster Locknut**
2. **Ball Bearing (radial load)**
3. **Casing Back Cover**
4. **Double Volute Casing**
5. **Gland**
6. **Motor**
7. **Motor Pedestal**
8. **Shaft Coupling**
9. **Stuffing Box**
10. **Sump Cover**
11. **Packing**
12. **Thrust Bearing**
13. **Thrust Pedestal**



# Pump Main Operating Parameters

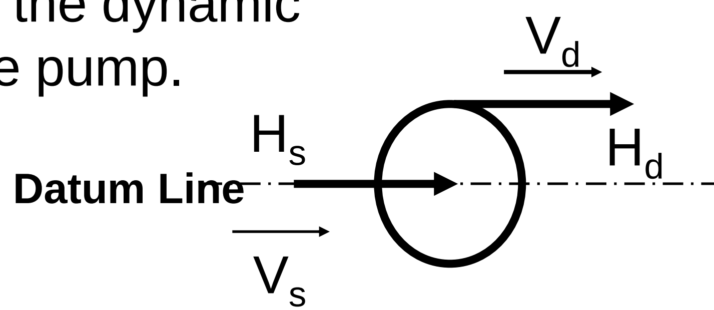
**Capacity** The liquid volume flow rate is referred to as the capacity. The most common units units used are US gallons per minute (g.p.m.), cubic feet per minute ($ft^3$**/**min), or per second (cusec) or cubic meters per hour ($m^3$/hr) or per minute ($m^3$/min).

**Head** The head is the corresponding height to which the pumped liquid can be raised by the pressure difference raised by the action of the pump. Datum line for determining the head in the case of horizontal pumps is taken the horizontal line through the centre line of the pump impellers and for vertical pumps the centre line of the pump outlet nozzle. **The head developed is independent of the specific gravity of the liquid** but it is affected by high values of liquid viscosity at pumping conditions. However the pressure difference developed by the pump is proportional to the liquid density.

$h=H_d \pm Hs$ i.e. static head = discharge head  suction head

**Total Dynamic Head** the sum of the head and of the dynamic head (velocity or dynamic head) developed by the pump.

$$H= H_d \pm H_s +(V_d^2/2g) - (V_s^2/2g)$$



# Pump Efficiency

The efficiency is a measure of the mechanical and hydraulic effectiveness of the pump. It is a comparison of the energy imparted to the liquid by the pump to the energy into the pump i.e energy input at the pump shaft and not the energy into the pump driver.

Pump Efficiency=(Liquid Energy imparted by the pump)/(Pump Energy Input)

$$\eta = Q.\rho.H/P$$

where $\eta$ = Pump Efficiency

$Q$ = pump flowrate

$H$ = Total Head Developed at Q

$\rho$ = liquid density at pump conditions

$P$ = power input to the pump for an electric motor $P=3VI\cos\varphi * \eta_{motor}$

$V$ = voltage inot motor

$I$ = motor current

$\cos\varphi$ = motor power factor

# The Affinity Laws (i)

If the rotational speed of a pump is changed from $N_1$ to $N_2$ it can be shown that the following relations are applicable:-

"The flowrate (Q) varies directly as the rotational speed"
"The total head (H) varies directly as the square of the rotational speed"
"The power input (P) varies directly as the cube of the rotational speed"

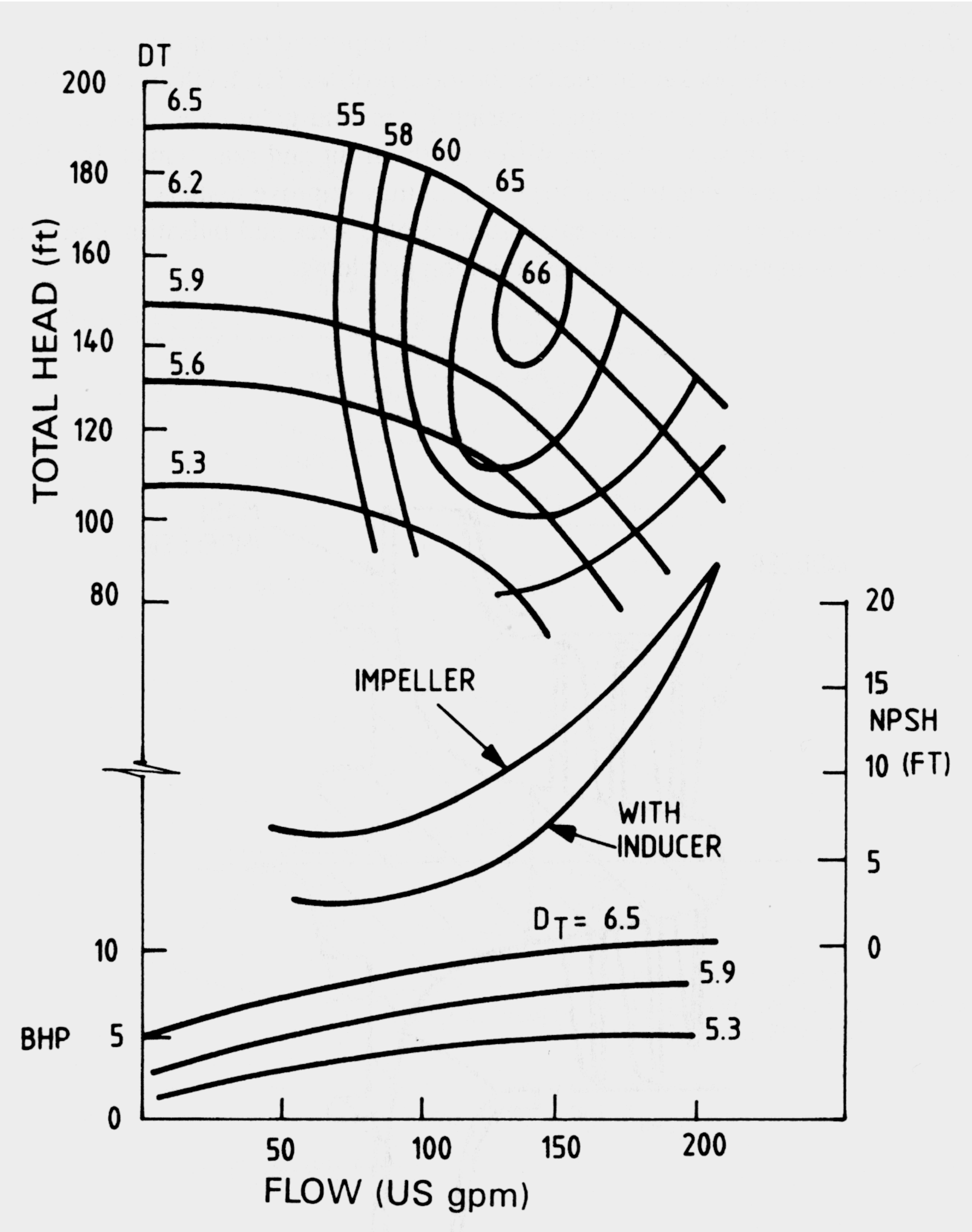
i.e. $Q_1/ Q_2 = (N_1/ N_2)$ $H_1/ H_2 = (N_1/ N_2)^2$ $P_1/ P_2 = (N_1/ N_2)^3$ at constant D

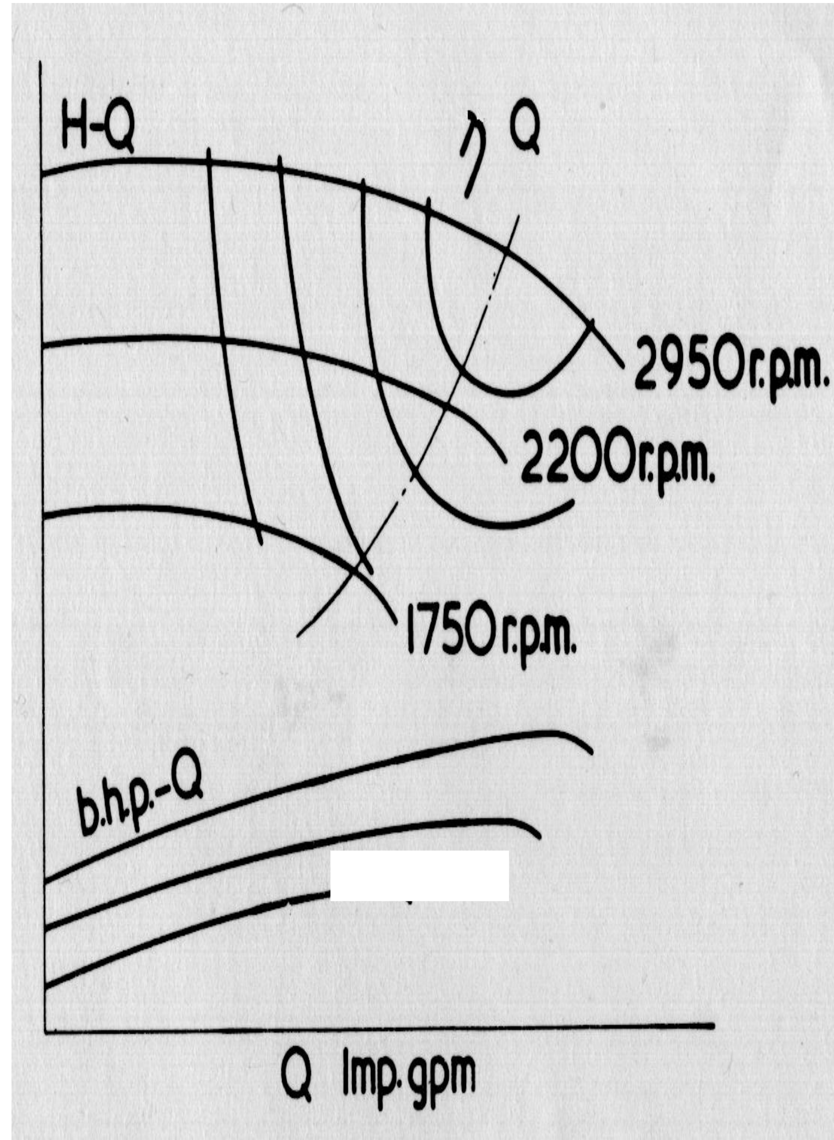
Similarly if the diameter D of the impeller is changed and the pump is run at the same rotational speed the following relations are applicable provided that the change in impeller diameter is 10% to 15% below the maximum that the casing can accommodate. If the change is greater the lack of flow guidance lowers the pump efficiency. Correction factors are given by the chart that follows.
"The flowrate (Q) varies directly as the impeller diameter"
"The total head (H) varies directly as the square of the impeller diameter"
"The power input (P) varies directly as the cube of the impeller diameter"

i.e. $Q_1/ Q_2 = (D_1/ D_2)$ $H_1/ H_2 = (D_1/ D_2)^2$ $P_1/ P_2 = (D_1/ D_2)^3$ at constant N

# Variation of Head & Power Vs Capacity

## with various impeller diameters or speeds



# Affinity Laws - Corresponding Points

According to the Affinity Laws

$Q_1/ Q_2 = (N_1/ N_2)$ $\quad$ $H_1/ H_2 = (N_1/ N_2)^2$

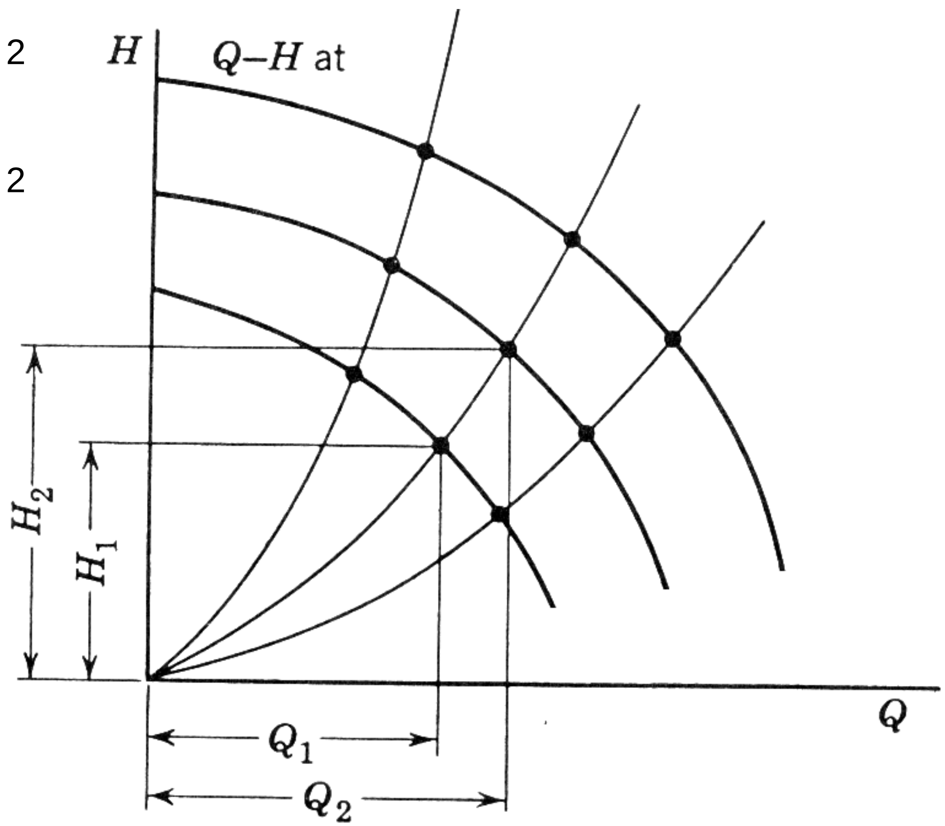
or

$Q_1/ Q_2 = (D_1/ D_2)$ $\quad$ $H_1/ H_2 = (D_1/ D_2)^2$

Points such as 1 & 2 connected by the Affinity Laws are called corresponding points and are connected by curved lines through the origin.

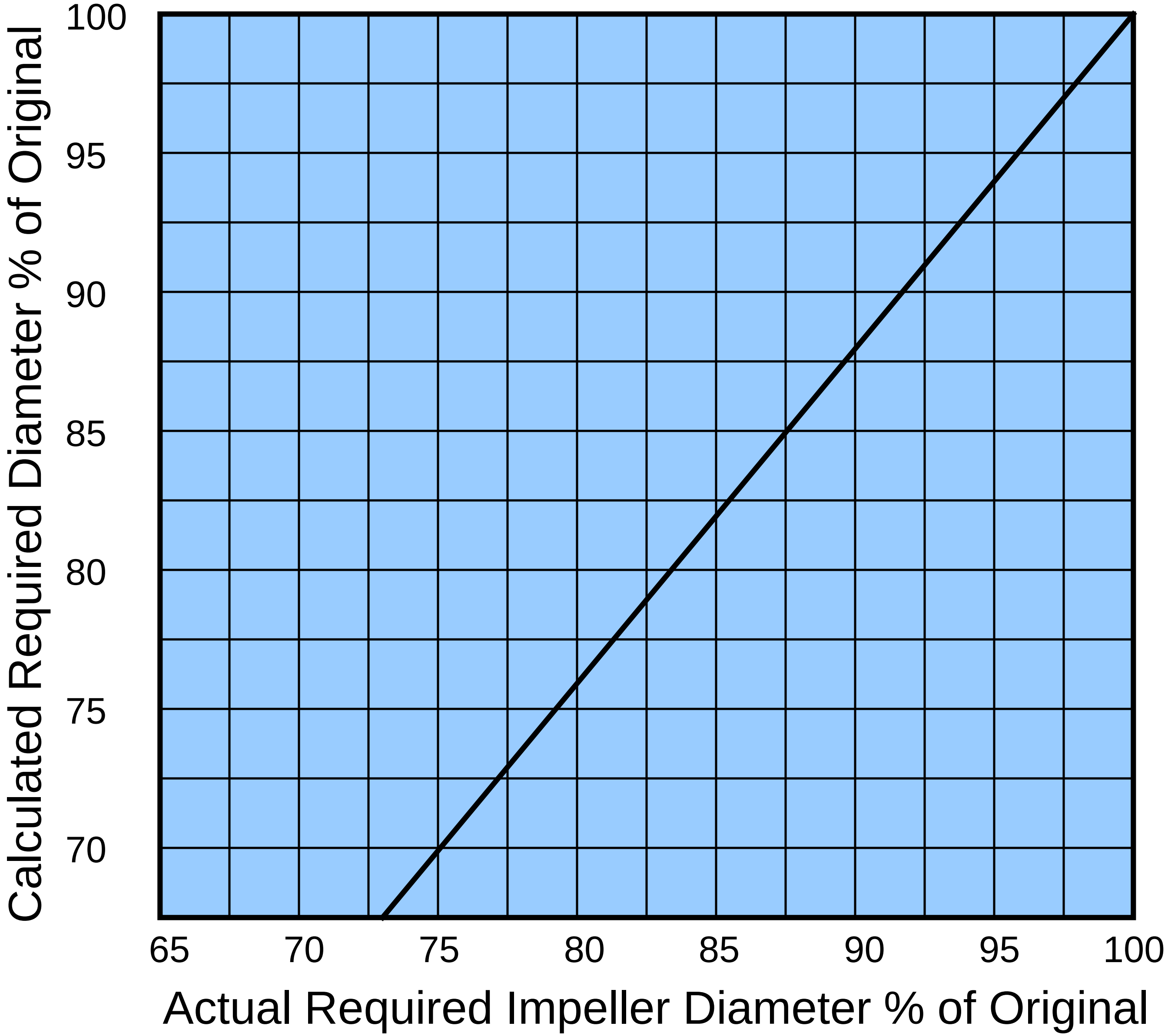
The curved lines through the origin are :-

1. Parabolas in case of the Head/ Capacity (H/Q) relationship
2. Cubic in the case of the Power/ Capacity (P/Q) relationship



# Impeller Trim Correction

## Re Affinity Laws



# The Affinity Laws (ii)

When the speed of the pump takes different values whilst the diameter is unaltered a family of head capacity and power capacity curves are obtained. Similarly if the diameter of the pump impeller takes different values whilst the speed of the pump remains unaltered a different family of head capacity and power capacity curves are obtained. If **corresponding points** in each family are joined together **a number of parabolic (for the head capacity group of curves) and a number of cubic curves (for the power capacity group of curves) through the origin are obtained.**

When the affinity relationships at constant speed and with constant diameter are combined the result is as follows:-

$$Q_1 / Q_2 = (N_1 / N_2)\,(D_1 / D_2)$$

$$H_1 / H_2 = (N_1 / N_2)^2\,(D_1 / D_2)^2$$

$$P_1 / P_2 = (N_1 / N_2)^3\,(D_1 / D_2)^3$$

# Specific Speed (i)

A characteristic which describes the hydraulic type of a pump is the "**Specific Speed Number**"

This number is an index of the hydraulic type (radial, axial, positive displacement etc) of a pump using **head, capacity and speed values** at the point of **maximum pump efficiency**.

The Specific Speed $N_s = NQ^{1/2} / H^{3/4}$

a consistent system of units has to be used for calculating its value. The calculated value depends on the system of units used although the number is dimensionless, if used in the form shown below.

$N_s = NQ^{1/2} / (gH)^{3/4} = k_s$, the Characteristic Number in the SI system.

However the omission of the gravitational constant g makes the value of the calculated number dependent on the system of units used for the calculation. N is in rpm, Q in US gallons/min, H in ft or Q is in $m^3/s$ and H in metres.

In the SI system of units N is in rad/s, Q in $m^3/s$ and gH in J/kg

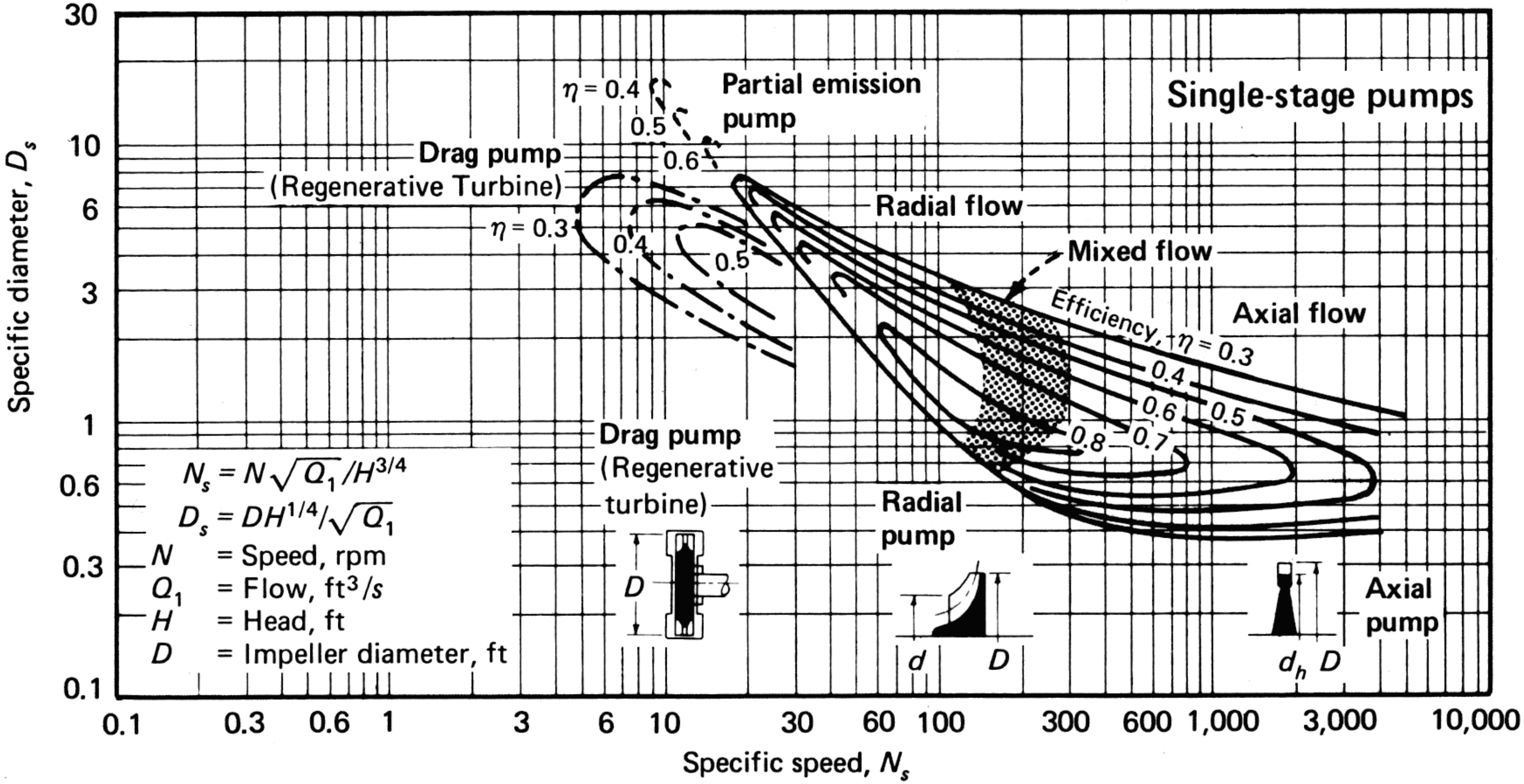
# Specific Speed (ii)

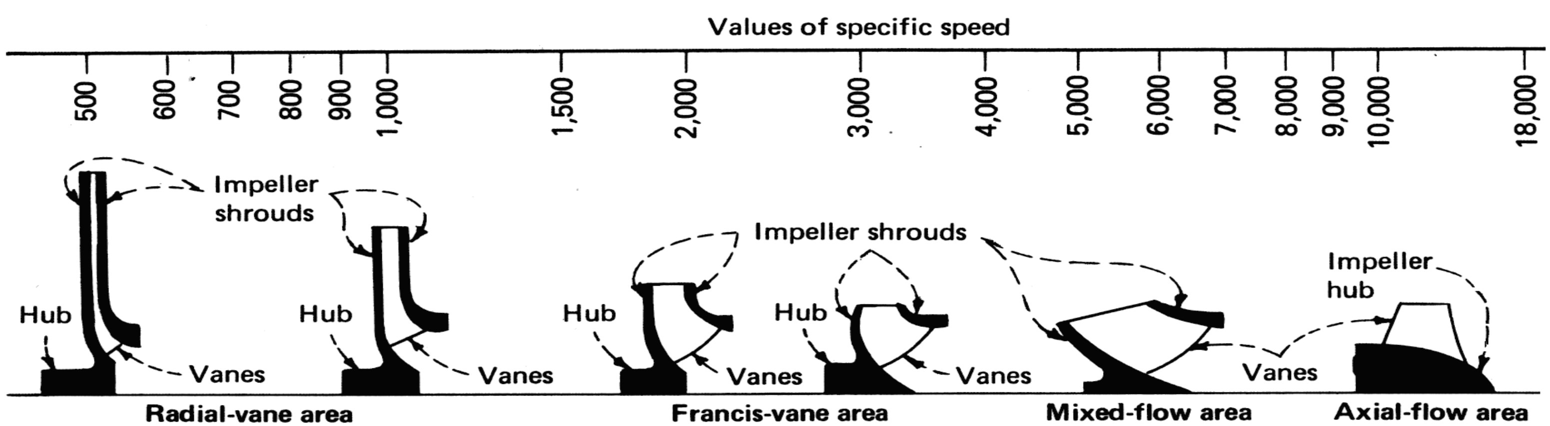
The Specific Speed Number (or Characteristic Number in the SI system) is calculated on the basis of head per stage (i.e if the total head developed at maximum efficiency by a pump with n stages H/n is used for the calculation) for multistage pumps and for double suction impellers the capacity at maximum efficiency is divided by 2 so that the numbers used refer to a single stage single suction pump.

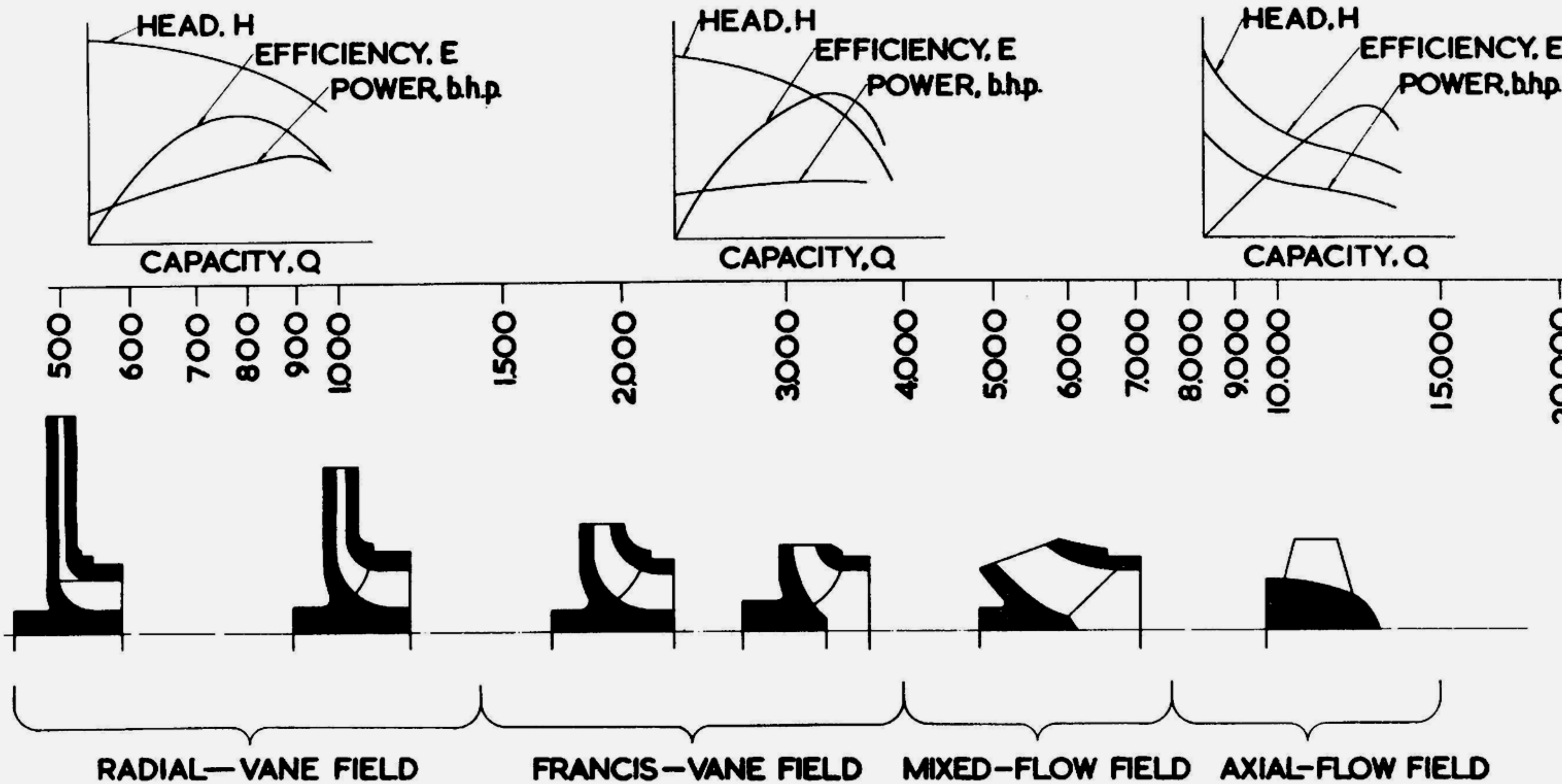
In the study of pump performance the Specific Speed Number is a criterion of hydraulic similarity as Reynolds Number is for pipe flow.

The specific speed equation shows that higher specific speed pumps run at higher speeds. At the same speed and capacity higher specific speed pumps run at lower head or for the same speed and head the higher specific speed pump will deliver higher capacities.

# Preliminary Pump Selection Using Specific Speed Number

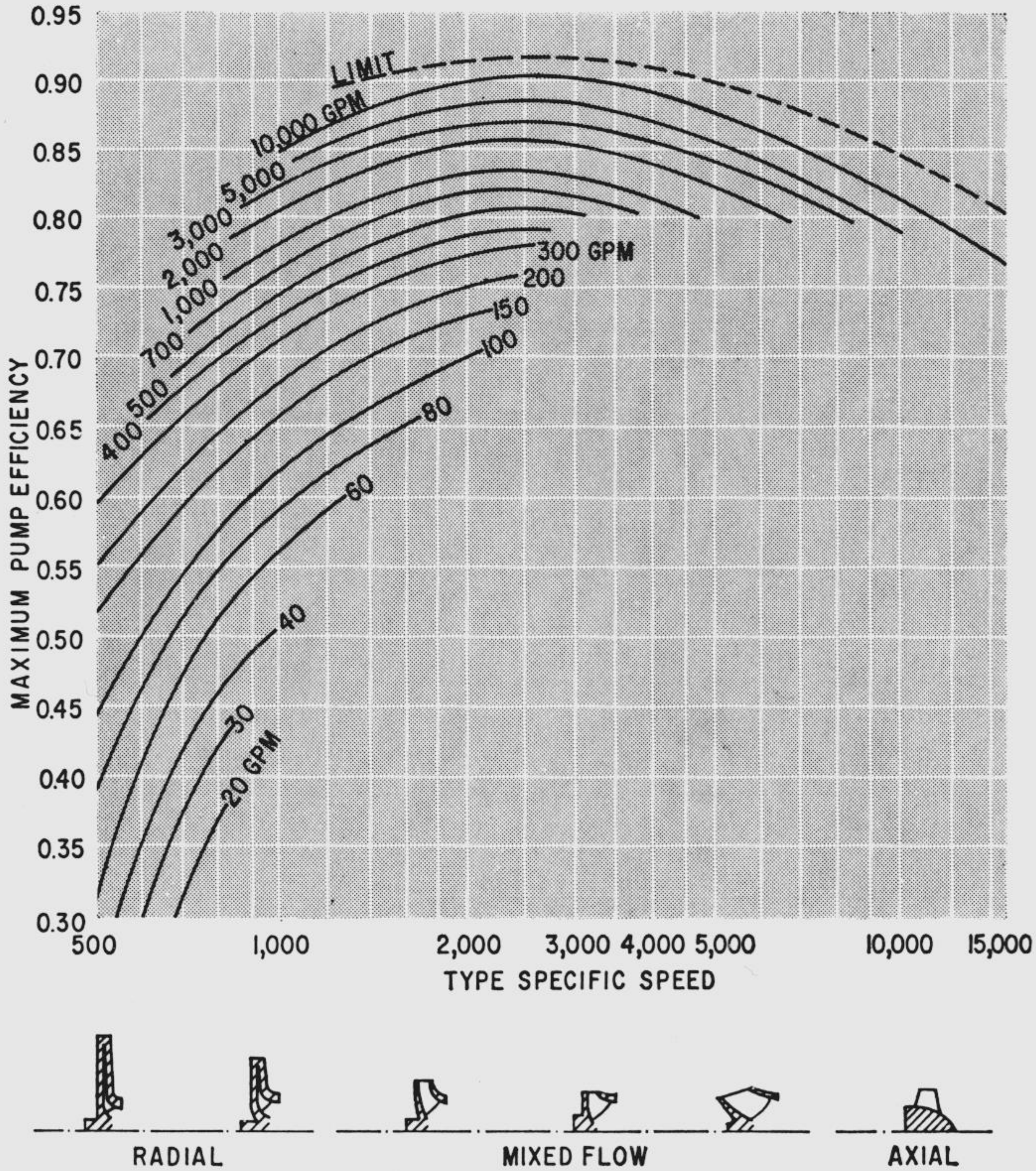




HEAD, H
EFFICIENCY, E
POWER, b.h.p.
CAPACITY, Q
HEAD, H
EFFICIENCY, E
POWER, b.h.p.
CAPACITY, Q
HEAD, H
EFFICIENCY, E
POWER, b.h.p.
CAPACITY, Q
500
600
700
800
900
1,000
1,500
2,000
3,000
4,000
5,000
6,000
7,000
8,000
9,000
10,000
15,000
20,000
RADIAL—VANE FIELD
FRANCIS—VANE FIELD
MIXED—FLOW FIELD
AXIAL—FLOW FIELD

# Variation of Maximum Pump Efficiency with Specific Speed and Pump Type



# Suction Specific Speed

Impeller Suction Conditions are very important in the performance and behaviour of a pump. Suction conditions of pumps with respect to cavitation can be expressed by another “dimensionless” number, the Suction Specific Speed. It is a term that relates capacity, speed and NPSH(R) and gives criteria necessary to evaluate the cavitation characteristics of pumps.

The Suction Specific Speed $S_s$ is based on **NPSH(R), capacity and speed values** at the point of **maximum pump efficiency**.

The Specific Speed $S_s = NQ^{1/2} / NPSH(R)^{3/4}$

a consistent system of units has to be used for calculating its value. The calculated value depends on the system of units used although the number is dimensionless, if used in the form $S_s = NQ^{1/2} / (gNPSH(R))^{3/4}$ .
However the omission of the gravitational constant g makes the value of the calculated number dependent on the system of units used for the calculation. N is in rpm, Q in US gallons/min, H in ft or Q is in $m^3/s$ and H in metres.
In the SI system of units N is in rad/s, Q in $m^3/s$ and gNPSH(R) in J/kg

# Head Capacity Characteristic (i)

## Centrifugal Pumps

The actual head capacity characteristic of high specific speed pumps rises fairly sharply with decreasing rate of pump flow. As the specific speed number value decreases the increase of pressure rise with decreasing flow rate becomes flatter and flatter. For some radial impellers it is possible to observe a maximum pressure at some low value of capacity and then reduction in the total head developed as the capacity decreases further. This latter type of head capacity curve is called an **Unstable Characteristic**.

This will give rise to instability in the operation of the pump over the region in which there are two values of flow rate for the same value of total head. Under these conditions the pumped volume may fluctuate between the two values corresponding to the same total head.The instability can be eliminated by introducing a suitably sized restriction orifice at the pump discharge or by changing the impeller to another with vanes further away from the radial configuration. A decrease in the pump efficiency will be observed, especially in the first case.

# Head Capacity Characteristic (ii)

## Centrifugal Pumps

The capacity fluctuations due to an unstable characteristic will give rise to power fluctuations and in the case of a turbine or an engine drive to speed fluctuations as well. If the frequency of such fluctuations happens to coincide with that of the natural frequency of the associated piping system severe vibrations -with the possibility of failure-can result.

To develop capacity and power fluctuations certain conditions must prevail in conjunction with an unstable head capacity characteristic.

- The mass of the liquid must be free to oscillate e.g. pumping from a vessel with a free liquid surface to another with free liquid surface
- There must be a member in the system which can absorb and give back energy e.g. a long piping system
- There must be a member in the system to provide impulses to start the fluctuations e.g. an automatic control valve.
- When there are two similar pumps in the system running together
- When running values of rate of flow are within the instability region.

# Head Capacity Characteristic (iii)
## Centrifugal Pumps

Changing the width only of an impeller alters the gradients of a head capacity characteristic. A wider impeller is capable of handling a greater volume than an equivalent narrower one and therefore the characteristic will be flatter. Furthermore the narrower the impeller the higher the frictional losses per unit of flowrate and therefore the somewhat steeper the characteristic.

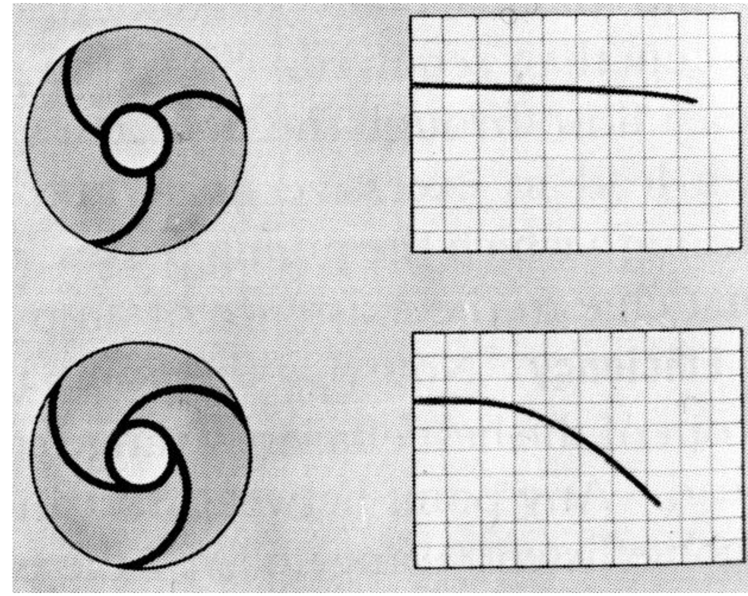
Increasing the number of vanes produces a flatter characteristic as the liquid guidance is more effective and the corresponding losses are lower.

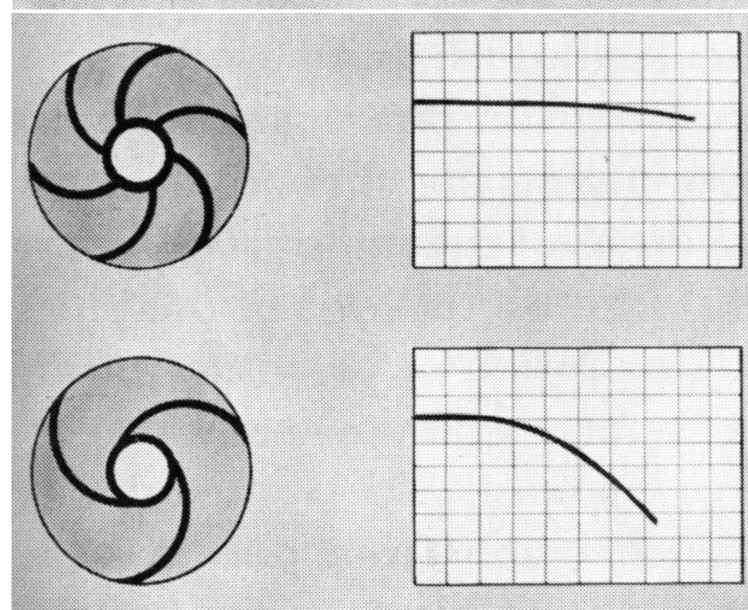
Increasing the radial curvature of the vanes (i.e. the swept back effect) gives a steeper characteristic and a somewhat lower pump efficiency.

Decreasing the radial curvature of the vanes may introduce pump instability as discussed previously.

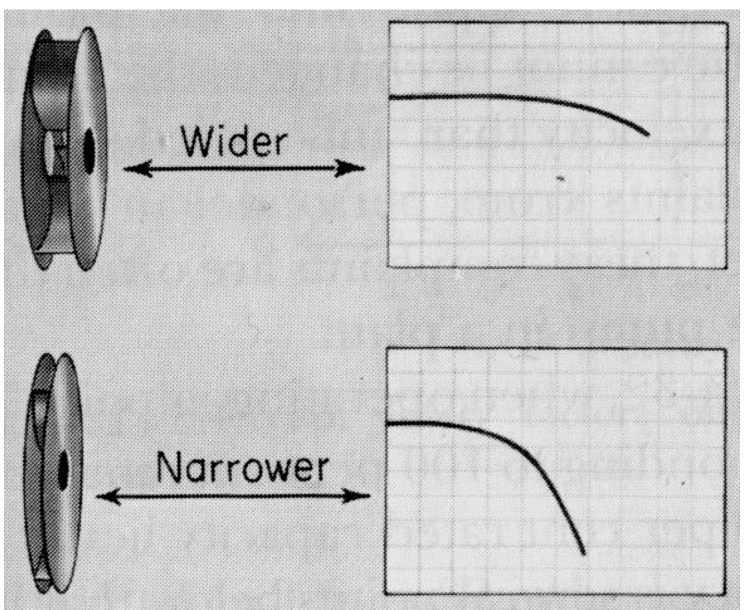
# Head Capacity Characteristic (iv)

## Centrifugal Pumps





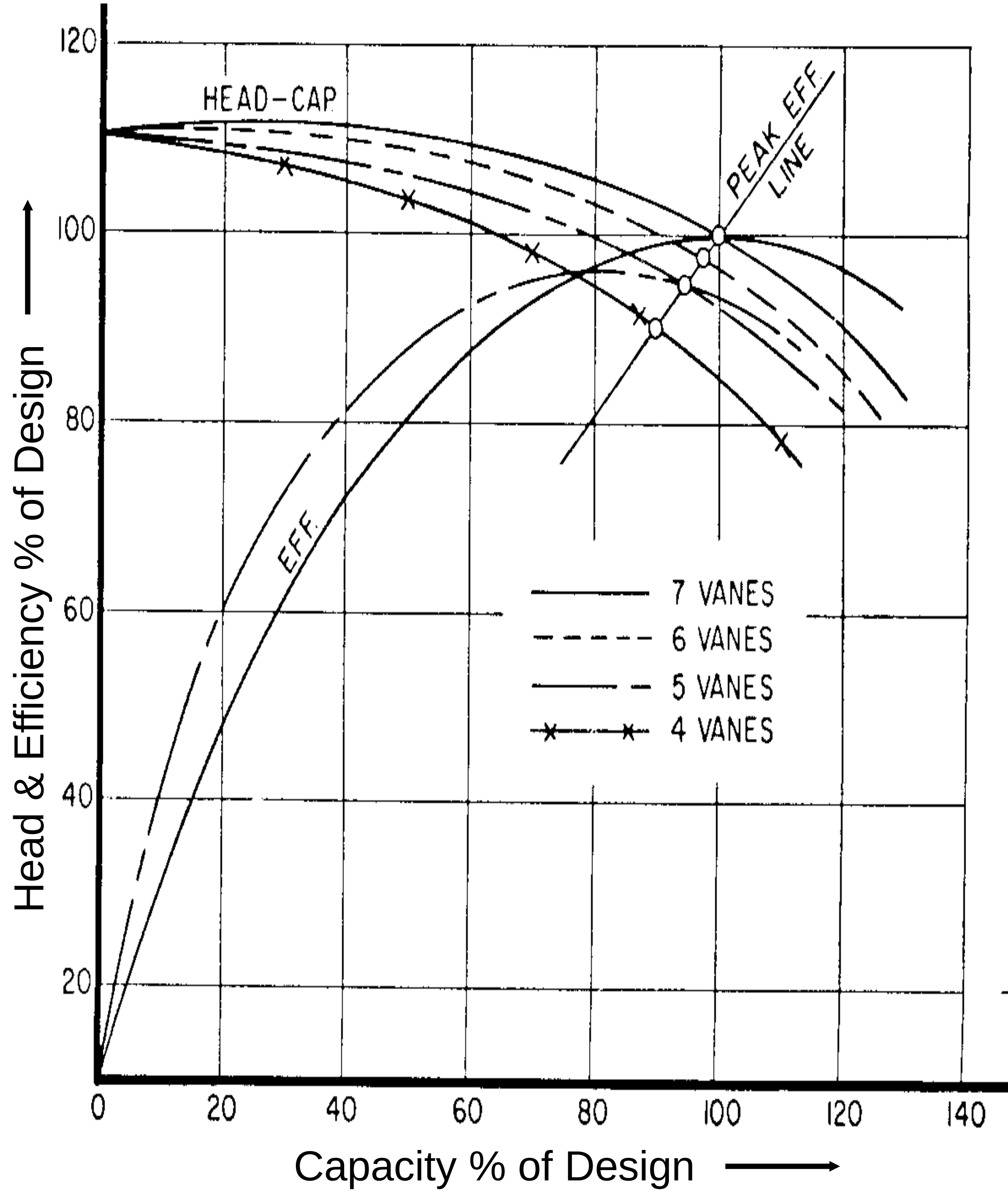
Varying the impeller gives the pump a different head/capacity, efficiency and power/capacity characteristics.
This property can be used initially in choosing the correct impeller for the required performance or at a later stage when the pump may need upgrading

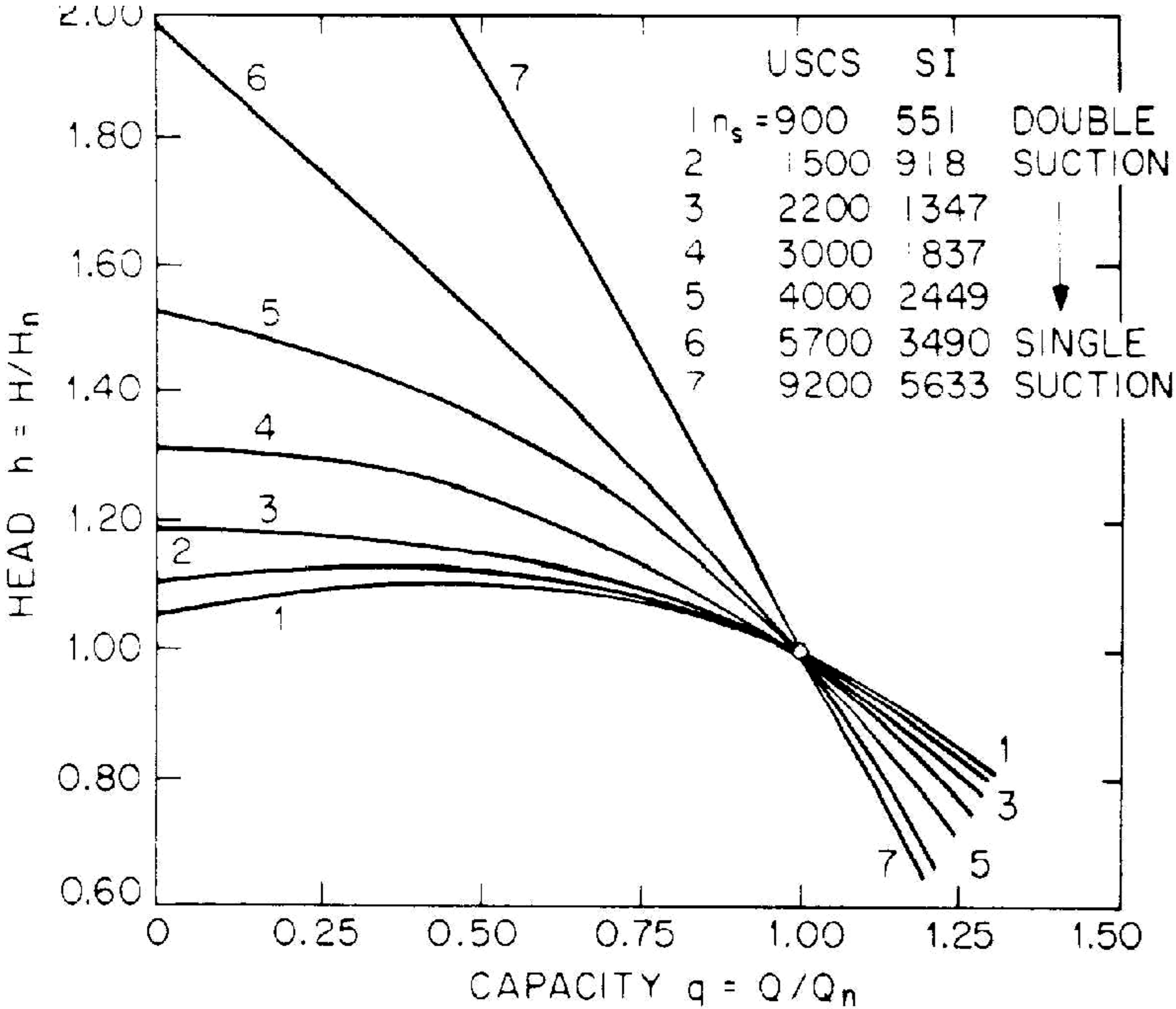


# Effect of Low $N_s$ Impeller Vane Number on Pump Performance

The lower the number of impeller vanes the lower the total head away from no flow conditions. Similarly the Best Efficiency Point has a lower efficiency.

As the number of impeller vanes decreases there is a corresponding decrease in the power input requirement.



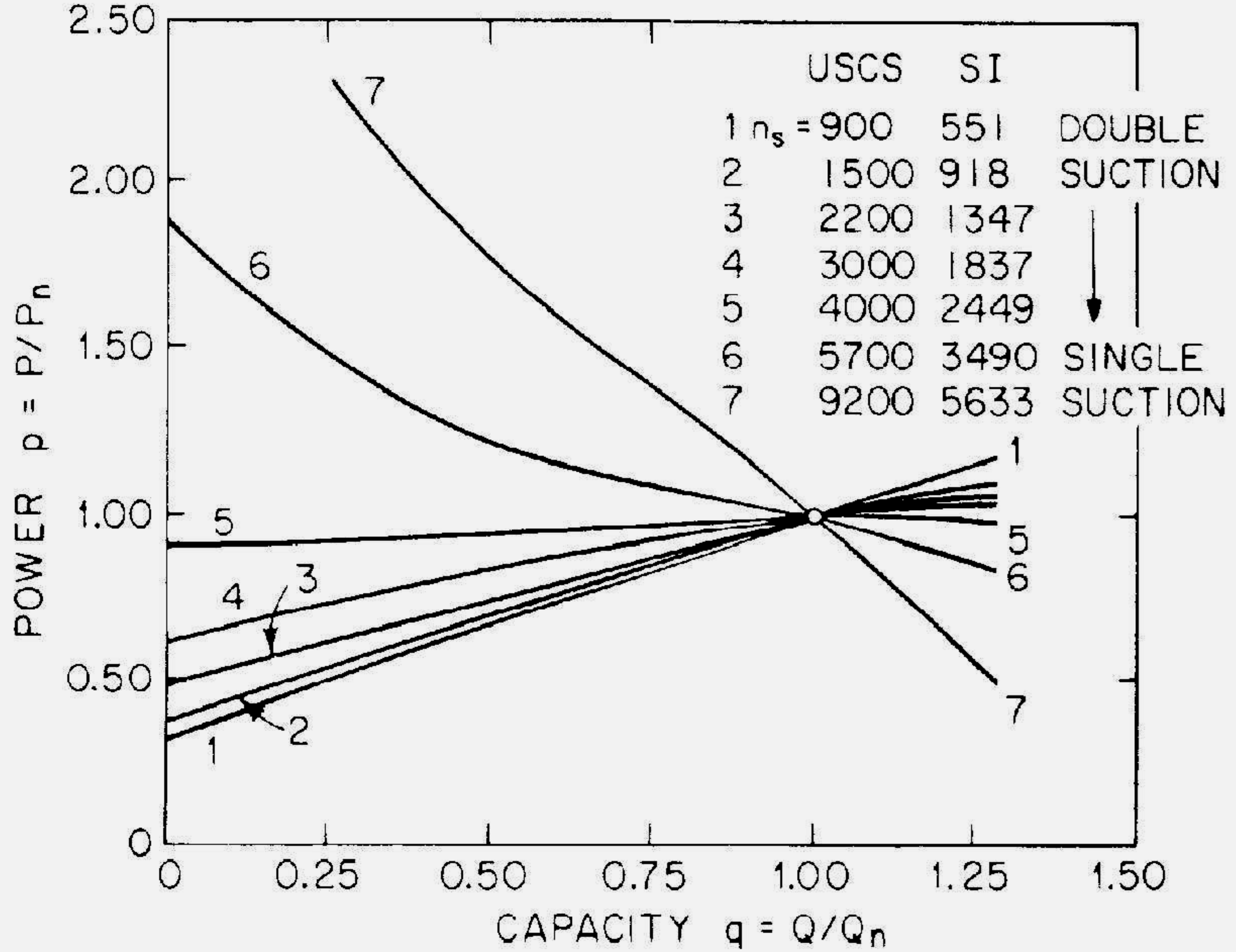



Head-capacity curves for several specific speeds.

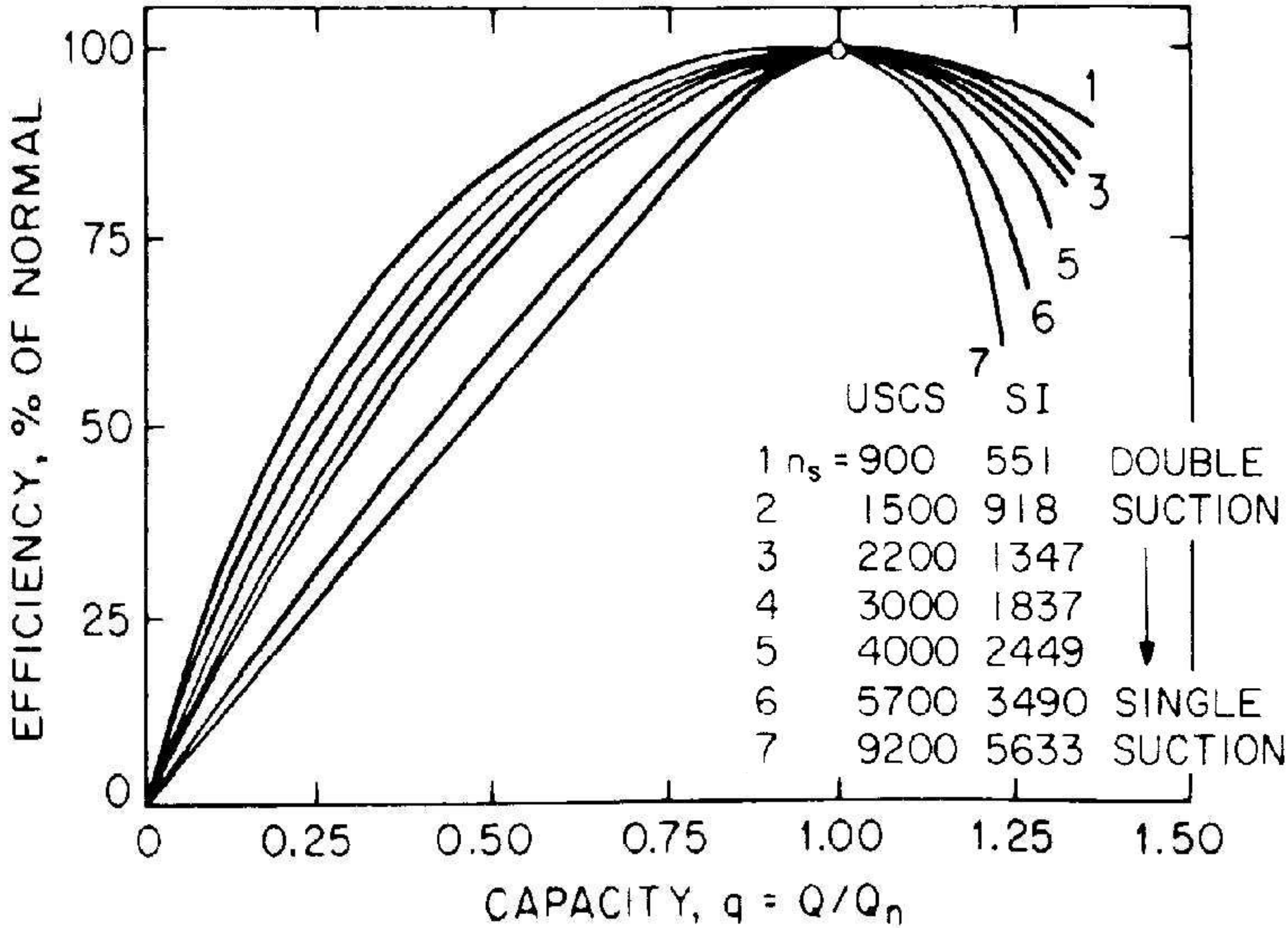
# Power Capacity Characteristic
## Centrifugal Pumps

The power capacity characteristics of high specific speed pumps rise fairly sharply with decreasing rate of pump flow. As the specific speed number value decreases the increase of power rise with decreasing flow rate becomes flatter and flatter and eventually in the case of radial pumps the power requirement decreases with decreasing flow rate. I.e. axial and mixed flow centrifugal pumps have maximum power demand at very low flow rates and radial centrifugal pumps have minimum power demand at very low rates.

This nature of the Power Capacity characteristics indicates that a radial centrifugal pump should preferably be started with a closed discharge valve and a mixed flow or an axial flow pump with an open discharge valve to minimize driver system starting load. However as soon as the pump reaches running speed the discharge valve must be opened to prevent pump temperature rise and at a steady gradual rate to prevent system hammering. In the case of large radial pumps a small pump bypass line with a globe valve is usually installed to direct a small flow from the pump discharge into the suction system for delaying the temperature rise. Clearly the bypass valve should be open and the discharge valve shut when starting.




Power-capacity curves for several specific speeds.




Efficiency Capacity Curves of Centrifugal Pumps for Several Specific Speeds

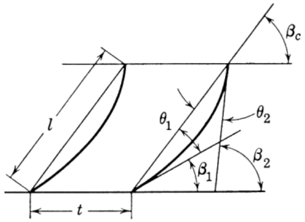
# Axial Pump Design Characteristics (i)

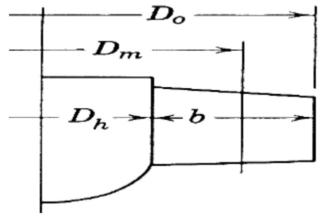
The Axial Flow Pump Field has the range of Specific Speed Numbers from about 6000 to over 18000.

The following design elements have an important practical effect on the axial pump performance.

1. Hub Ratio - The ratio of the hub diameter to the outside diameter
2. Number of Vanes and their Spacing
3. Vane Thickness
4. Angle of Vanes on the Hub
5. Pump Casing

# Axial Pump Design Characteristics (ii)



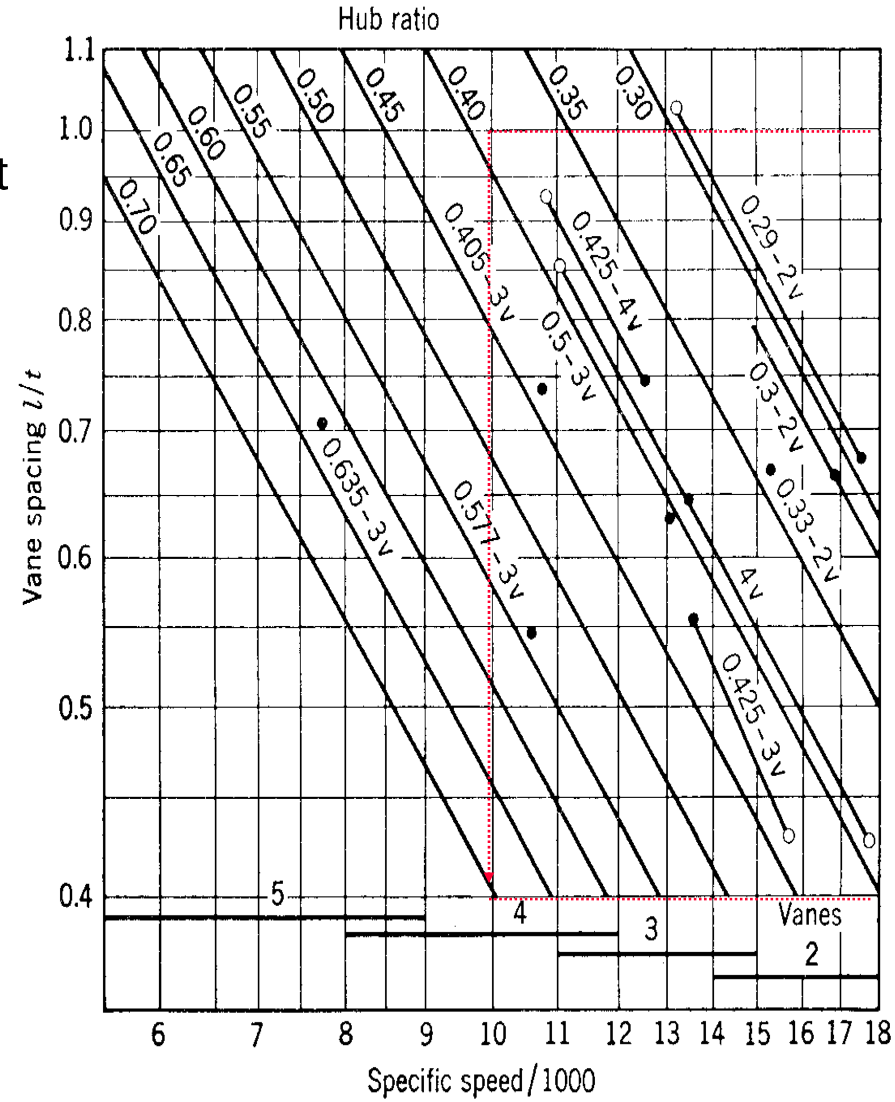

The diagram gives the Hub Ratios $D_h/D_o$. for various modern axial pumps for different specific speeds and their relationship to the Vane Spacing $l/t$.

The relationships are established experimentally.

In general higher specific speed pumps have smaller hubs which give a greater free area for the larger flows.

Furthermore the number of vanes decreases with increasing specific speed number as the head developed decreases and the capacity increases.

The hub ratio is less than unity for specific speeds over 10000.

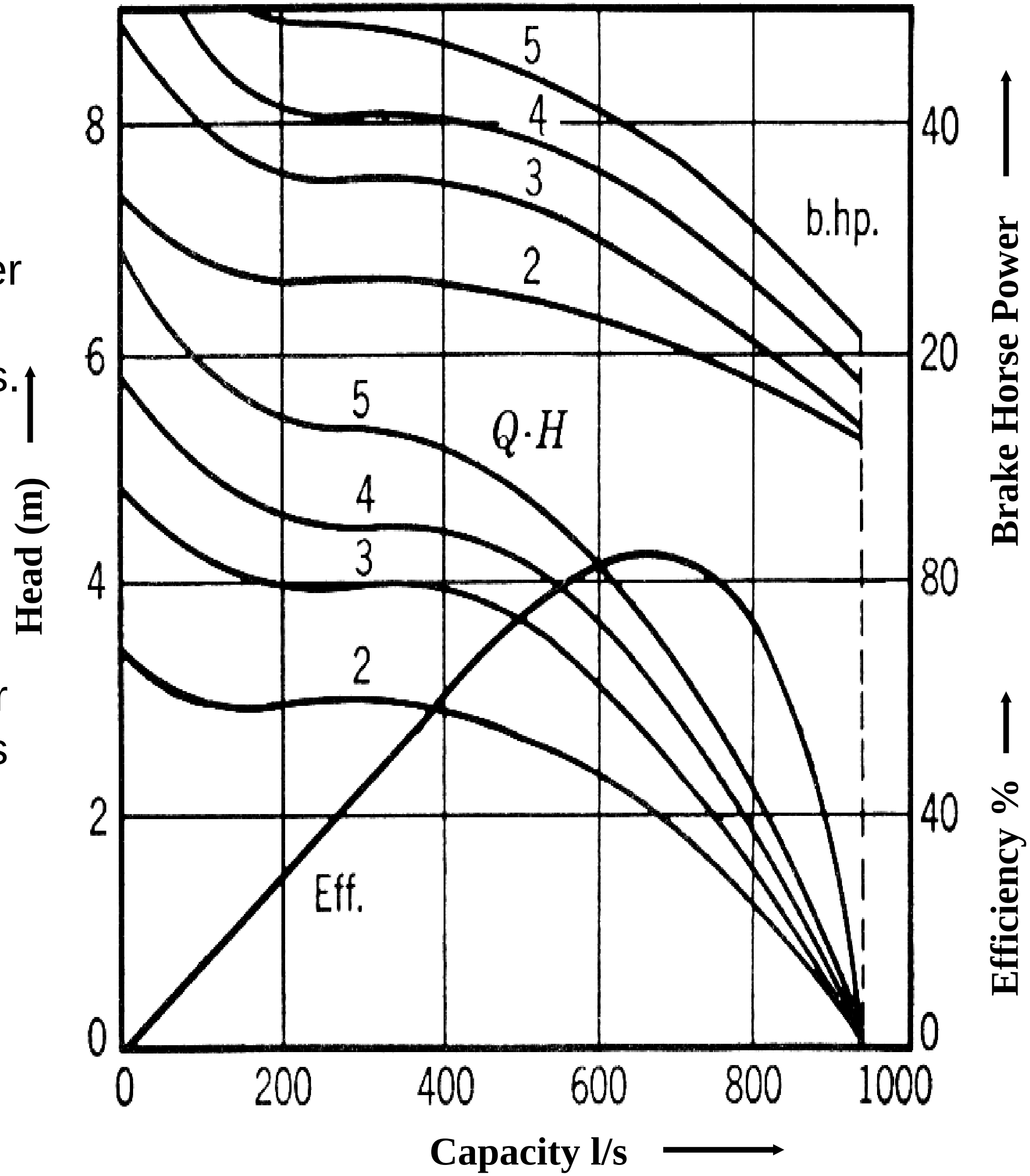


# Axial Pump Design Characteristics (iii)

The Head and Power Input increase with the number of vanes.

The capacities at zero head are the same

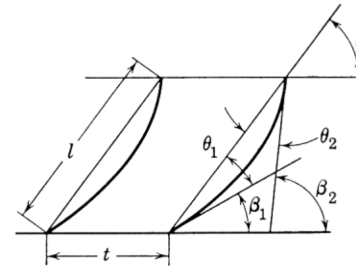
However the smaller the number of vanes the higher the efficiency.

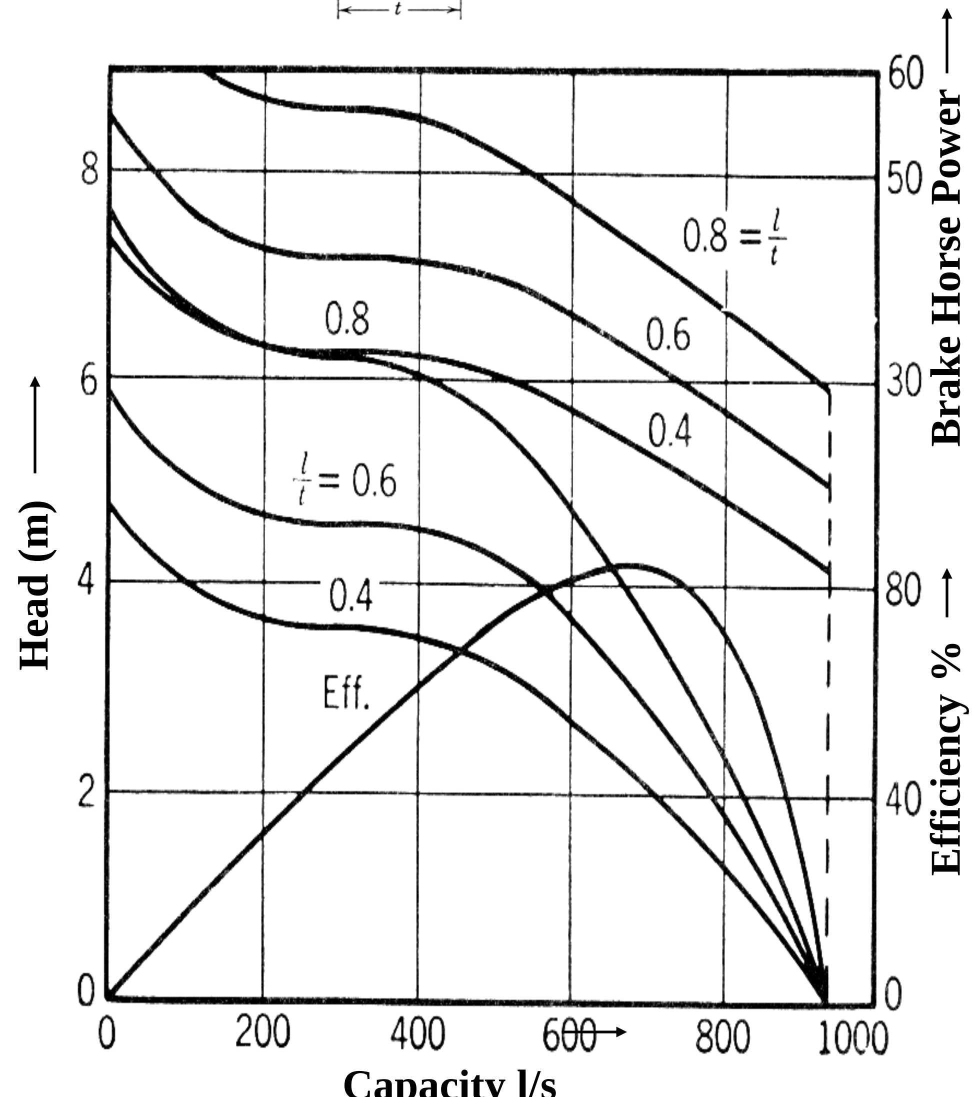


# Axial Pump Design Characteristics (iv)

The increase of the vane chord length relative to the spacing of the vanes increases the head developed for a given capacity.

This is similar to the effect of increasing the number of blades.

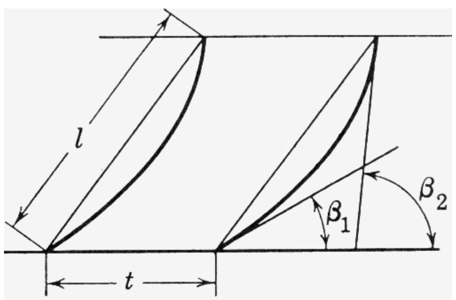


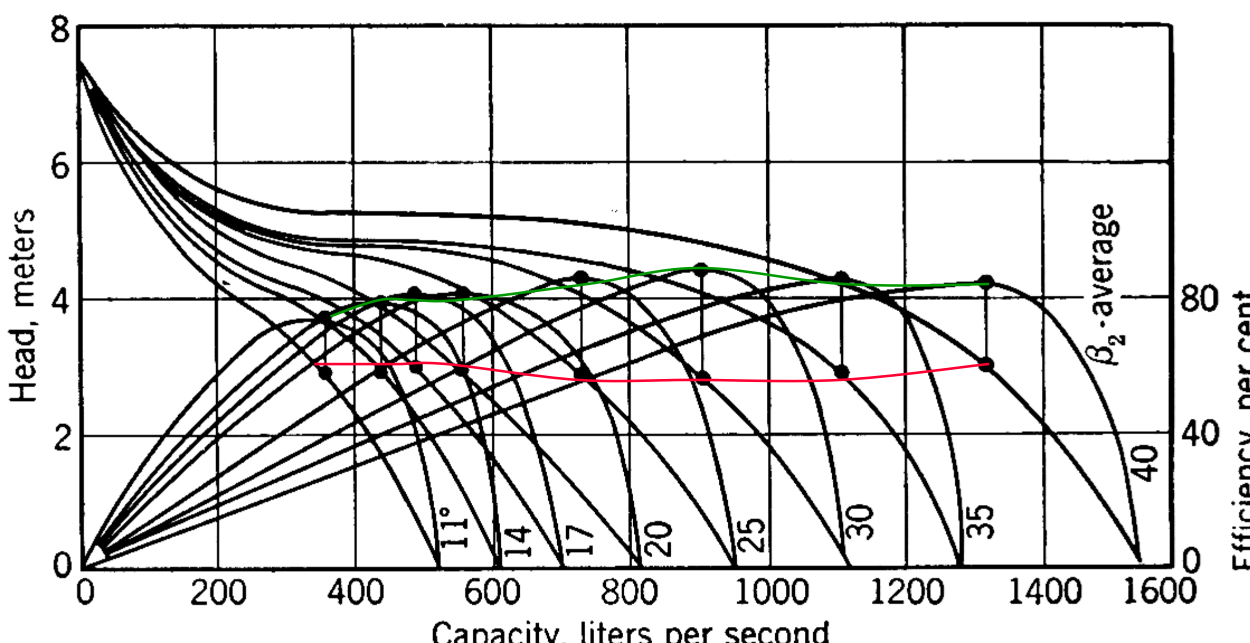



# Axial Pump Design Characteristics (v)

Experiments show that when the vane curvature ($\beta_2$ -$\beta_1$) **remains constant** and the angle $\beta_2$ is varied the head developed is essentially unchanged and similarly the efficiency.

In effect by varying the impeller vane angle the pump can operate over a much wider capacity range developing a substantially constant head at high efficiency.



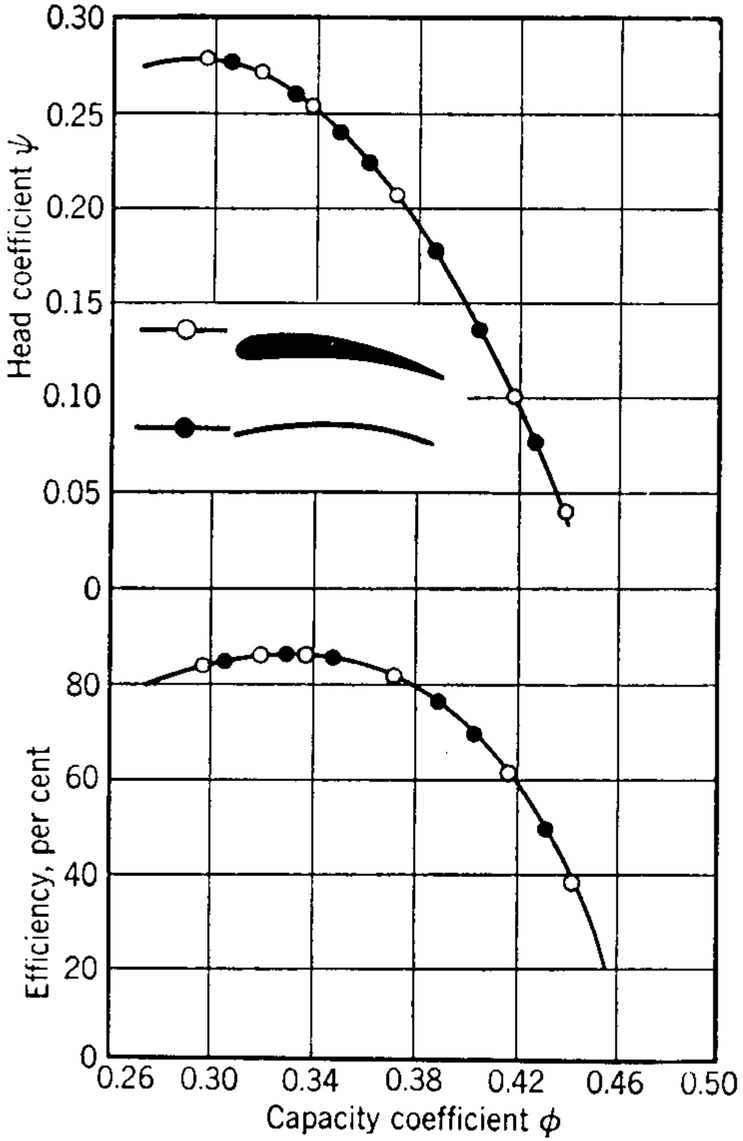



# Axial Pump Design Characteristics (vi)

It has been shown experimentally that the performance of two impellers, one with thin hydrofoil vanes and the other with stamped steel vanes, can be identical.

It was also shown that excessive vane thickness results in flow separation and turbulence with high speed impellers.

The advantage of the hydrofoil sections is that they give the desired mechanical strength with the minimum sacrifice in efficiency.



# Performance with Viscous Liquids

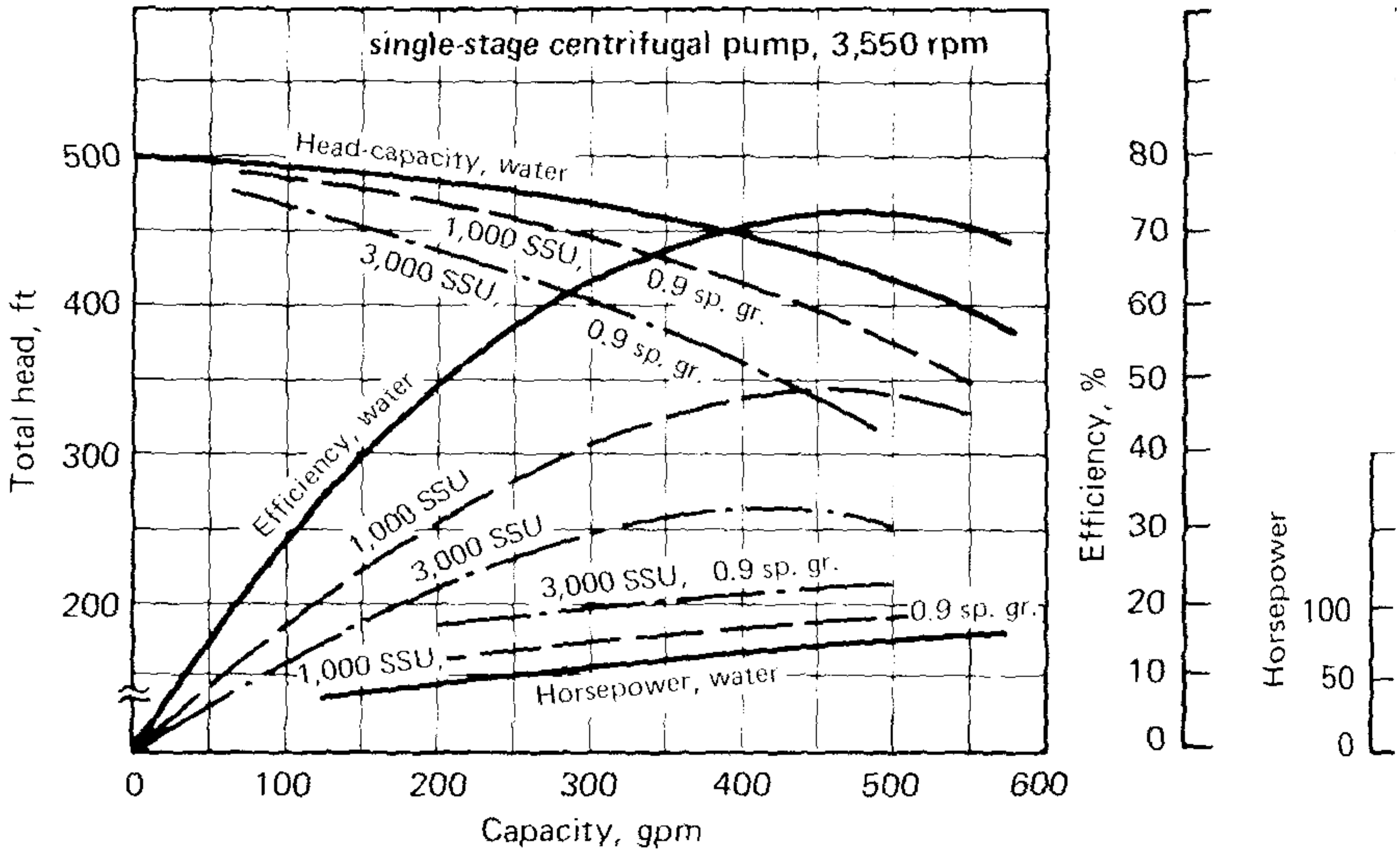
The hydrodynamic losses in a pump are influenced by the viscosity, the higher the viscosity the higher the losses, resulting in a respective reduction in the head developed, a corresponding increase in power input requirements and so a consequential reduction in pump efficiency.

The geometry of the pump also influences significantly the effect of viscosity on performance. The smaller the internal flow passages the greater the effect of viscosity.

The performance of a pump handling a viscous liquid is generally estimated by applying correction factors to the water performance of the pump. Experience suggests that correction is needed when the kinematic viscosity exceeds 20cSt.

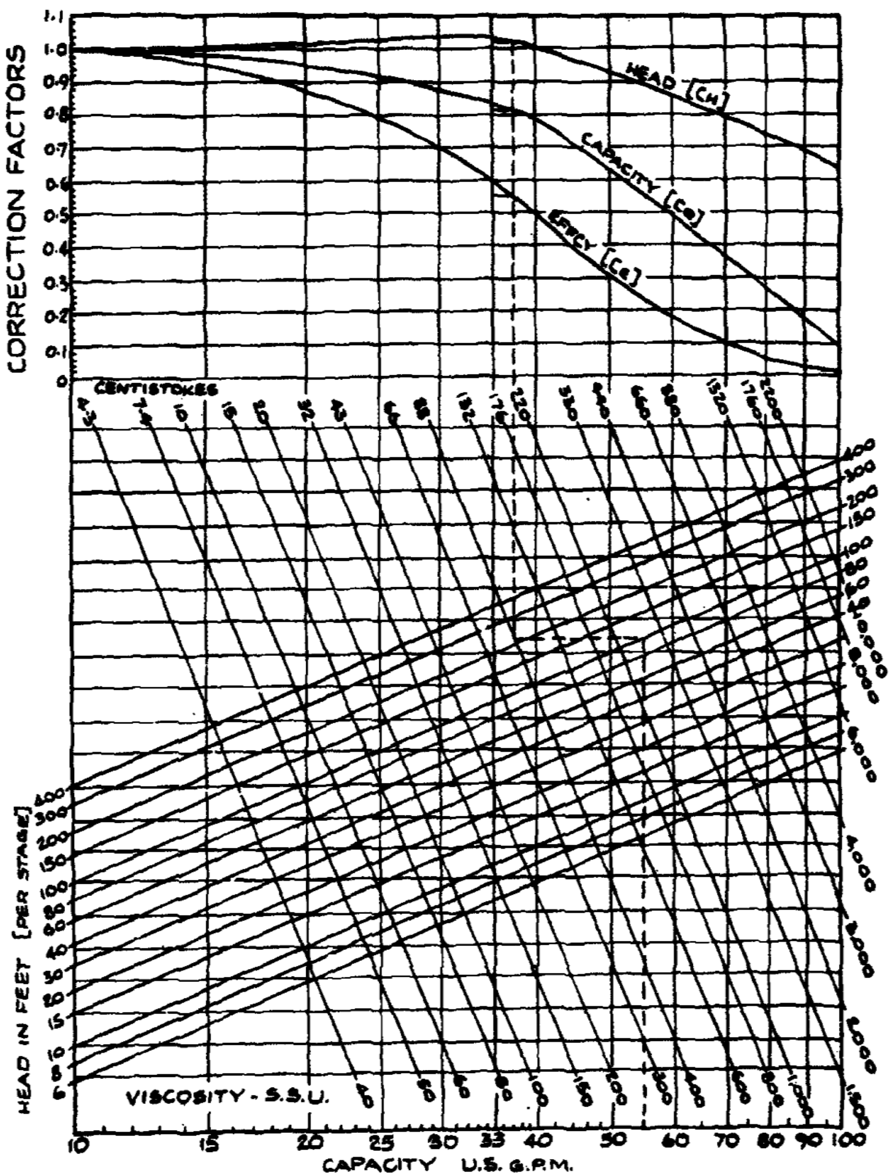
As the viscosity increases the head developed is lower, especially if the impeller is smaller than the maximum that can be fitted in the casing. However if the impeller is large and the viscosity is not very high a somewhat higher head than when pumping water can develop at low capacities due to the viscous drag.

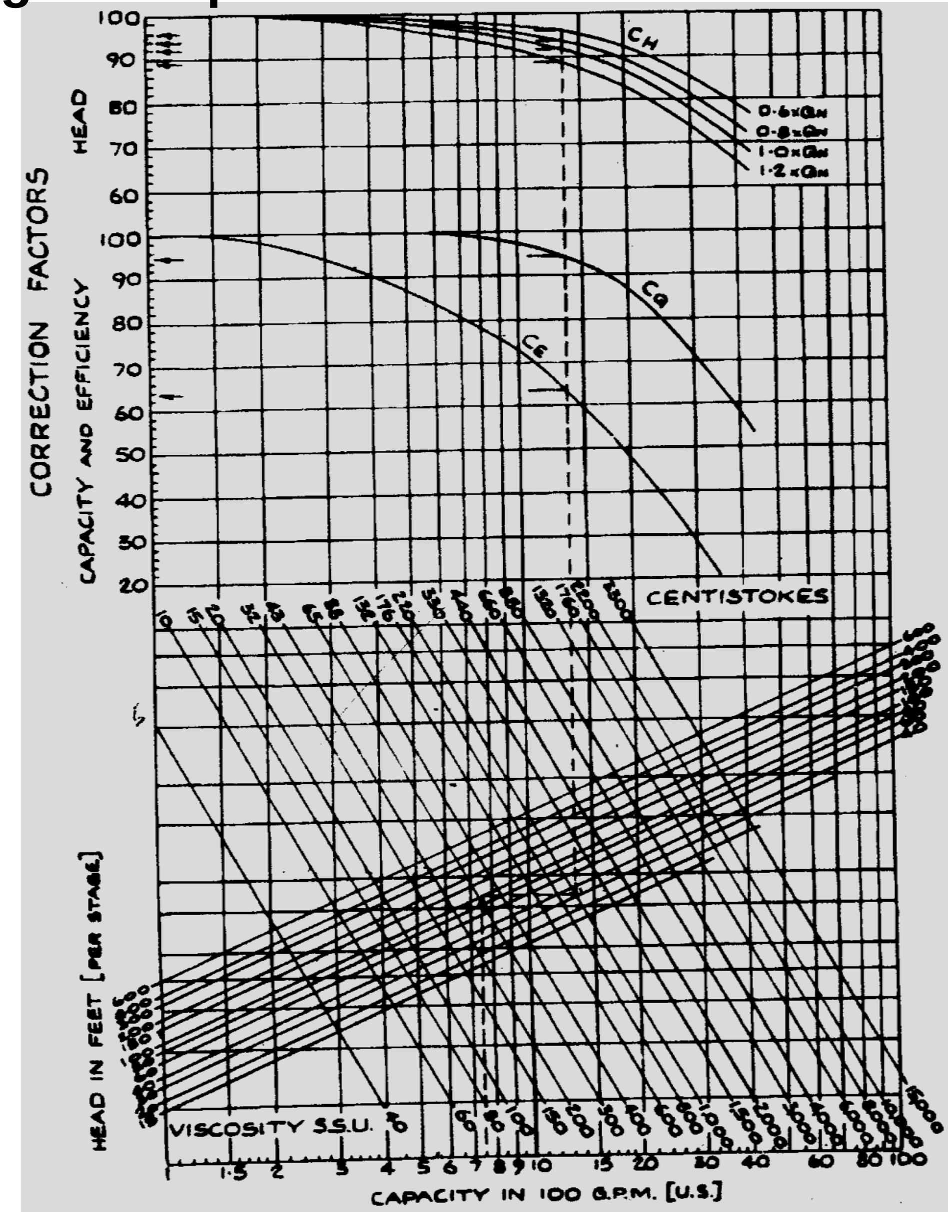
# Performance Reduction with Increased Viscosity



# Performance Correction Charts

## for Centrifugal Pumps



# Controlling Pump Flowrate (i)

In general the control of the flow rate of centrifugal pumps is achieved by varying

1. the resistance against which the pump has to deliver I.e. the pump head
2. the pump speed
3. the pump speed and the system resistance together.

The capacity and power of pumps with specific speeds below 4000 (USCS) increase with decreasing head. It is thus possible for the drivers of such pumps to be overloaded if the system resistance is lowered below a specific limit, depending on the specific driver.

The head and power of pumps with specific speeds over 4000 (USCS) increase with decreasing capacity. The drivers of such pumps and especially as the specific speed increases should be either sized to cope with the high power demand of no flow conditions or they should be provided with overload protection.

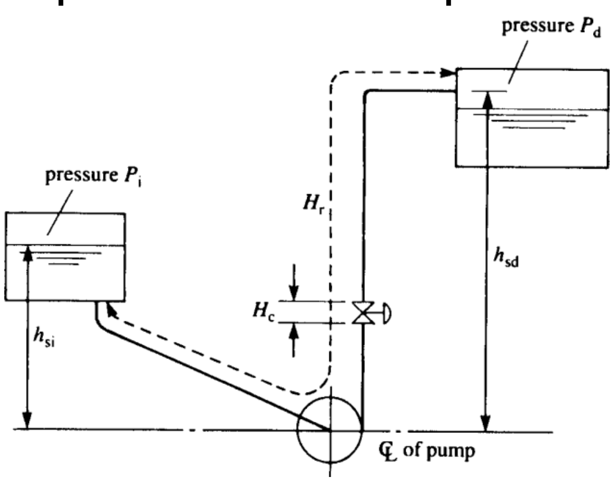
It is possible that systems connected to such pumps should also have overpressure protection for the eventuality of the pumps being started with closed discharge valve or if they are operated at too low a flowrate.

# Controlling Pump Flowrate (ii)

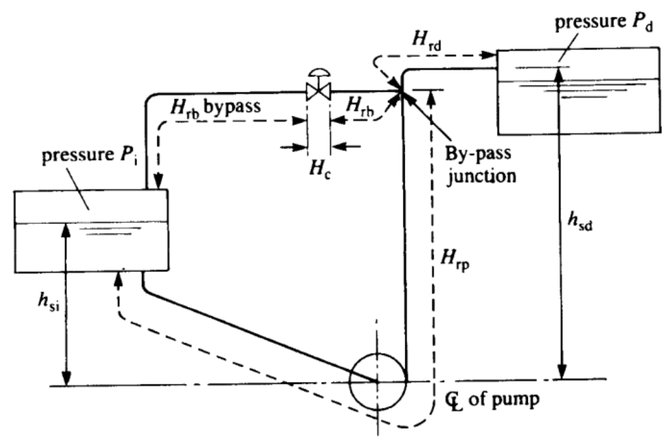
Capacity regulation may be manual or automatic by:-

1. Discharge Throttling. It is the cheapest and most common method for low and medium specific speed pumps. However power is lost at the throttle valve This power loss may be important in large installations.

2. By-pass Regulation. The pump is running at high capacity and the excess is diverted to a suitable point in the suction system through the by-pass line. For low and medium specific speed pumps this may entail considerable energy degradation with possible overheating but for high specific speed pumps considerable power saving is possible.




**Discharge Throttling**




**By-pass Regulation**

# Controlling Pump Flowrate (iii)

3.Speed Regulation. This is the most energy efficient method as it minimizes power requirements and overheating. Steam turbines, internal combustion engines, where practically or economically possible can give attractive solutions. Variable speed hydraulic drives or electric motors could also be attractive.

Variable speed drives are generally costly and more complex. They are justified for

a) large pumps when a continuous flow regulation is required

b) pumps operating for significant periods at low flows, flows in the re-circulation cavitation zone, or in systems with high control valve pressure drop.

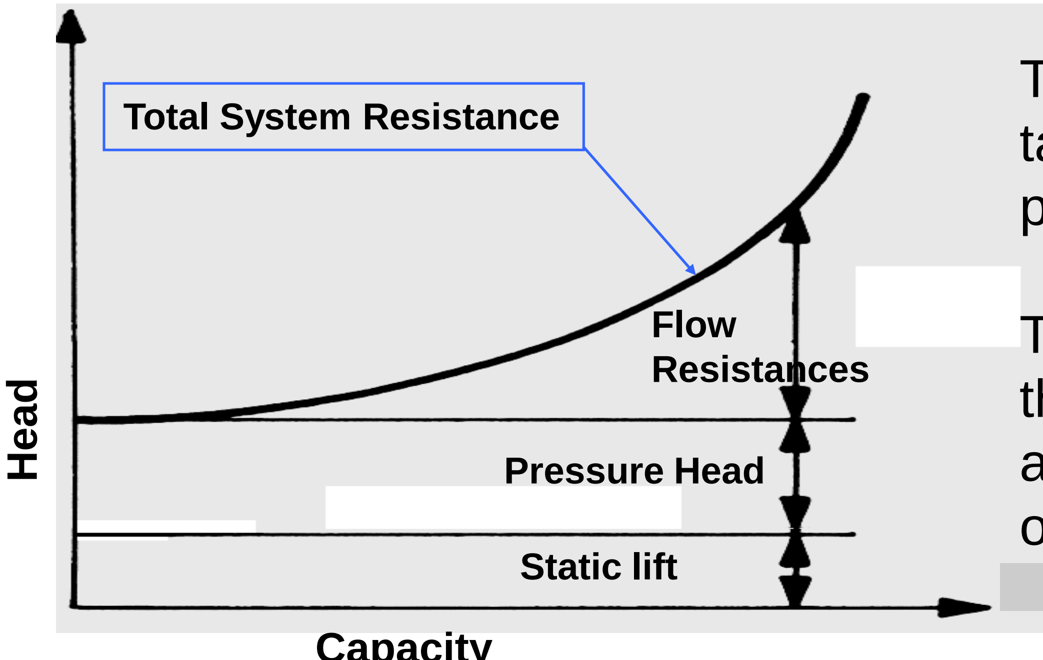
4. Adjustable Vane Regulation. Propeller pumps with adjustable pitch blades give good results as a wide capacity variation at practically constant head and substantially constant efficiency. It has limited practical application due to the complexity of the system. Ref slide on “Axial Pump Design Characteristics (v).

# Controlling Pump Flowrate (iv)

The resistance to the flow in a piping system have a

- static component → static lift from the suction to the discharge levels; pressure head increase from suction to discharge

- dynamic component   the line and equipment flow resistances in the system from the suction to the discharge points

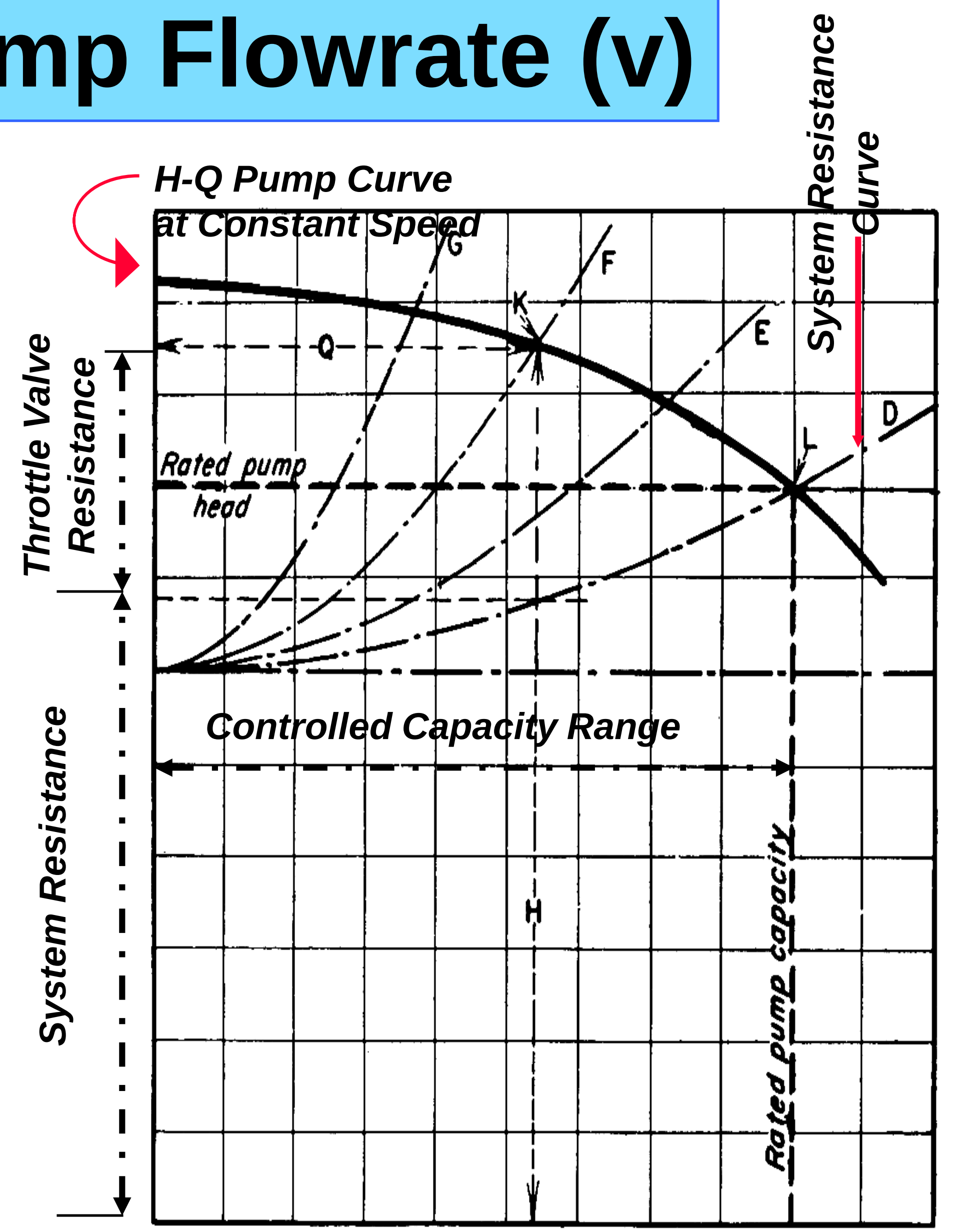



The pressure head increase is taken as constant since the process is assumed steady.

The flow resistances vary with the square of the flow velocity and therefore with the square of the capacity.

# Controlling Pump Flowrate (v)

If the pump in the system is coupled to a constant speed driver it develops a total head along the constant Head Capacity characteristic. This constant speed curve meets the system resistance line at only one point, (L) the “match point”. When the system resistance is lower than the pump head the pump delivers but it can only do so until the match point. The system flowrate can increase no further than the match point. However for capacities lower than the match point there must be a means of taking up the difference in the head developed by the pump and the system resistance head. In throttling systems this is done by a control or a manual throttling valve. System curve L does not include any valve throttling action. Curves E,F,G correspond to different positions of the throttling valve

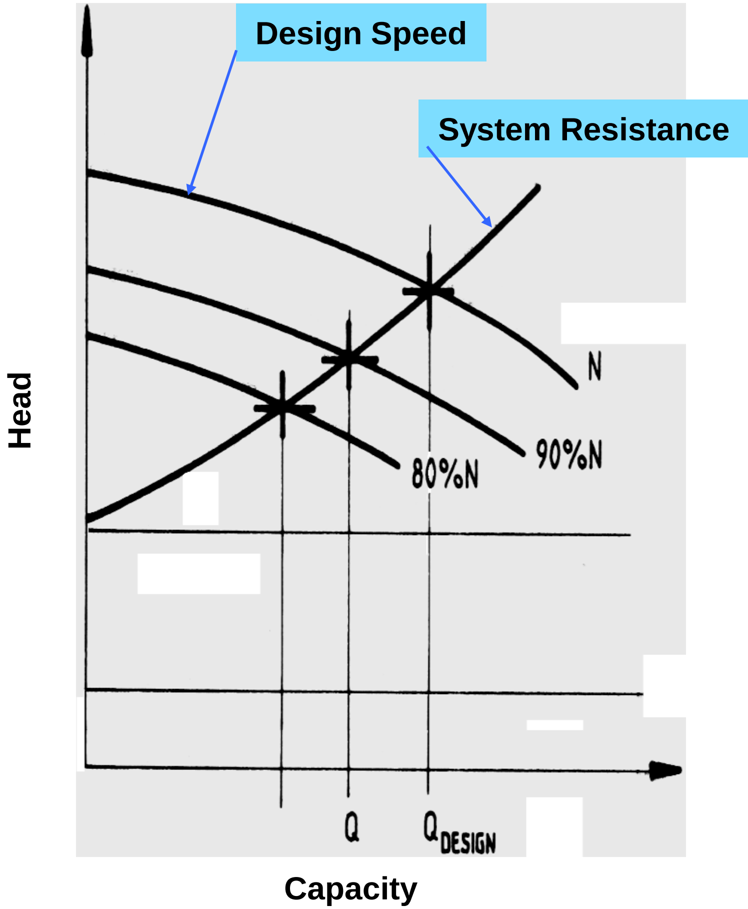



Total Head Vs Capacity

# Controlling Pump Flowrate (vi)

With Speed control a wide range of flow rates can be achieved. In fact a range of capacities are possible at substantially the same discharge pressure by varying the pump speed. The advantage here is that the pump with the speed control develops as much head as the system requires for the given flow. This results in energy conservation reducing pump driver operating energy costs.

The extra cost of the variable speed drive and its control must be justified against the economics of reduced driver energy consumption.

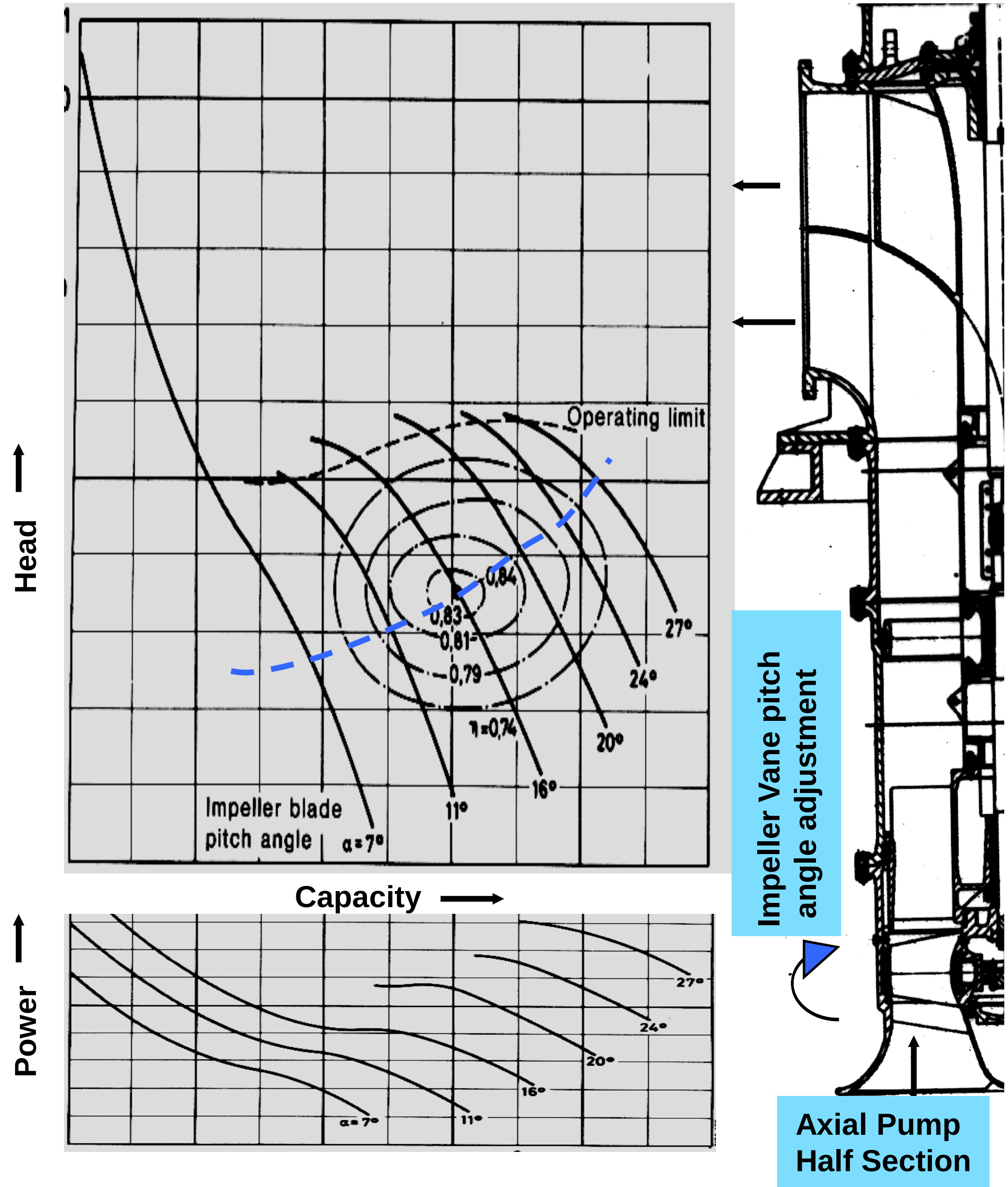


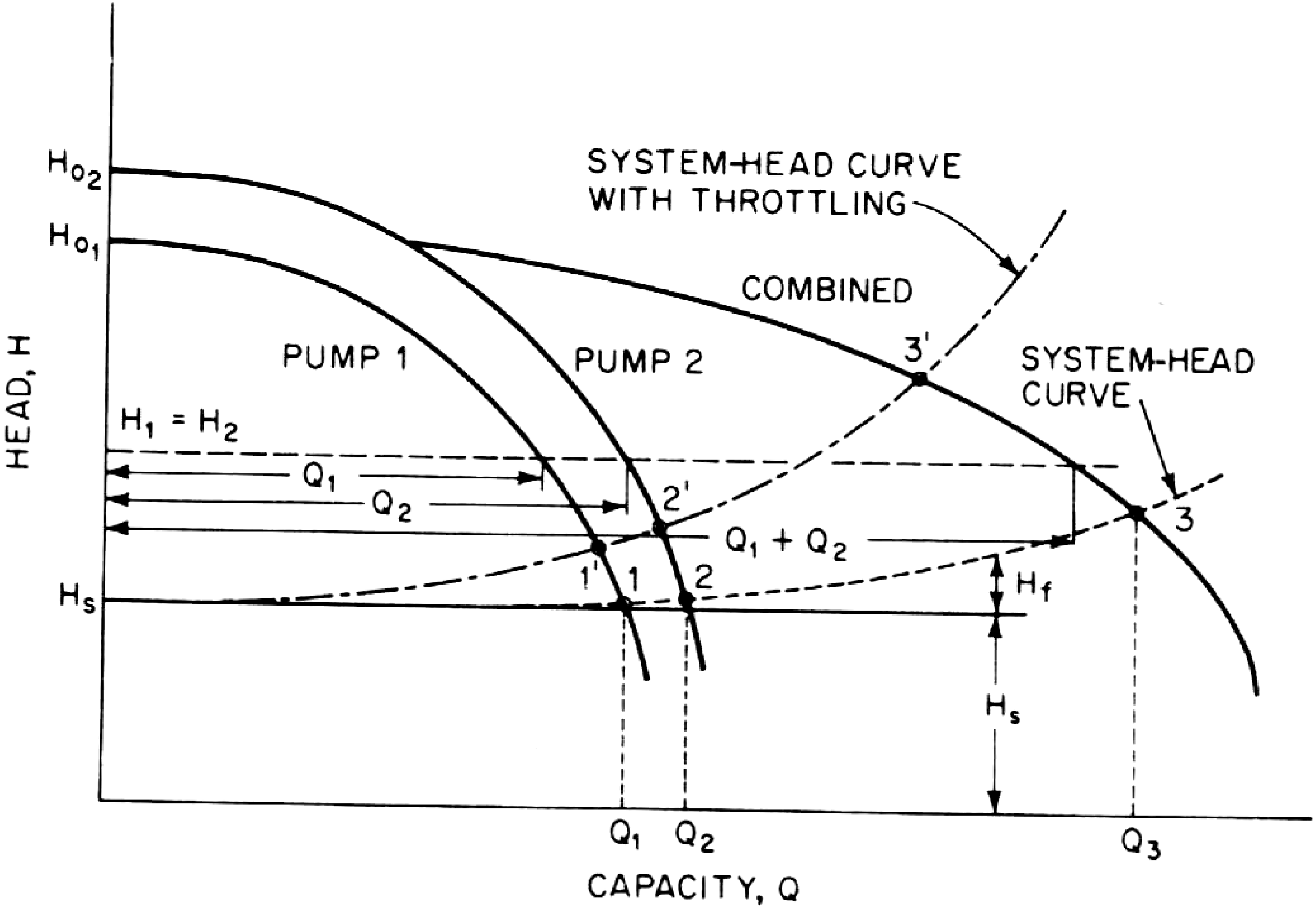
# Controlling Pump Flowrate (vii)

Varying the impeller vane pitch angle a stepless control of the Head/Capacity characteristic is achieved.

In effect by varying the impeller vane angle the pump can operate over a much wider capacity range developing a substantially constant head at high efficiency.

The mechanical complication of the adjuster mechanism increases the initial and maintenance costs. It is only possible on the larger pumps.






**Combined Head Capacity Characteristics of Pumps Operating in Parallel**

# Pumps in Parallel

The combined Head Capacity characteristic of pumps in parallel is obtained by adding the capacities at a given head as the pumps take suction and discharge from/to common suction and discharge headers.

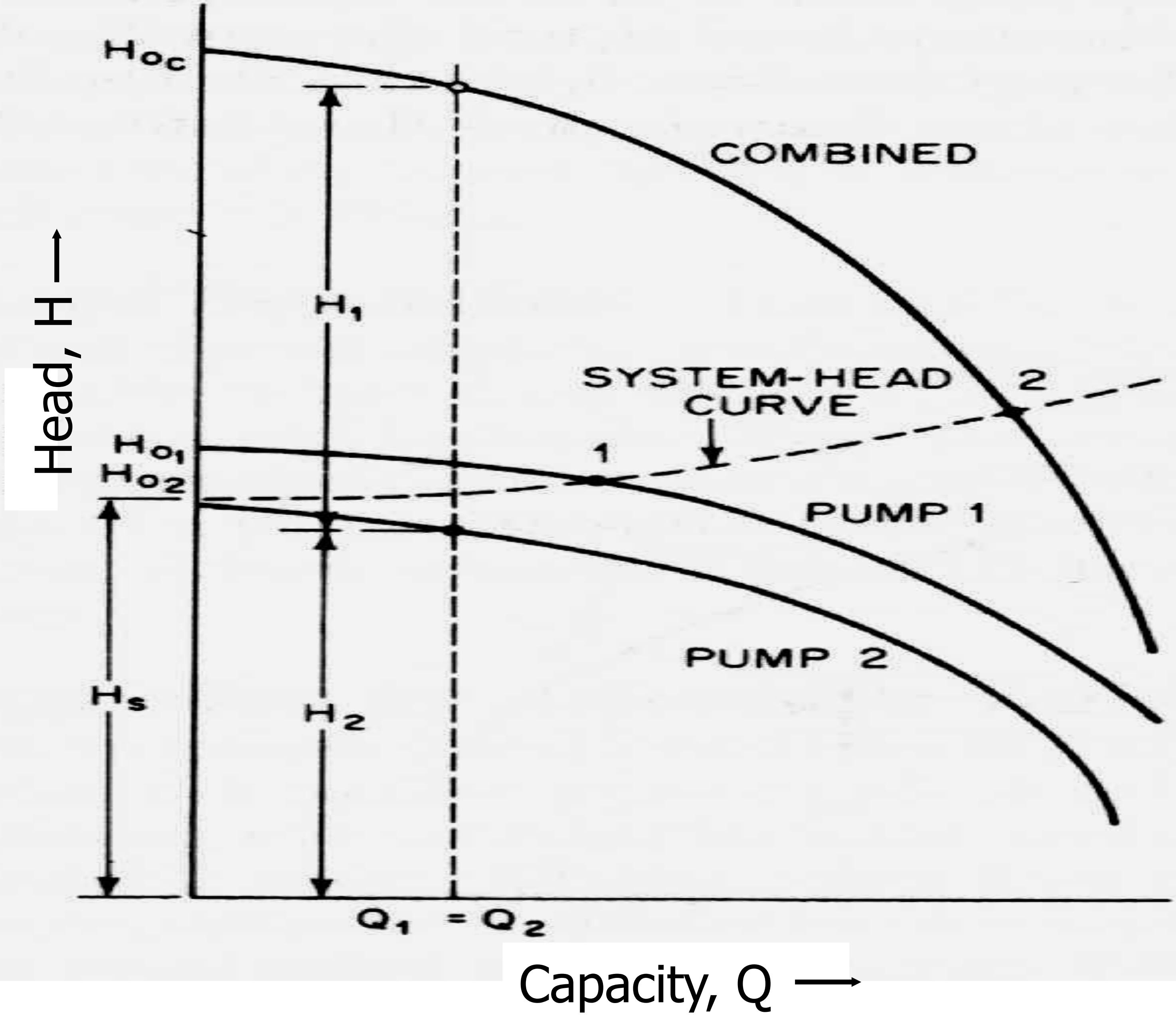
If one of the pumps develops a higher head than the other it is possible that at the lower capacities the higher head pump runs at a head higher than the maximum of the other pump. In such a case the lower head pump will not be delivering while it will be consuming energy and thus it will overheat.

It is possible that if the suction piping system is not adequately designed for cavitation to set in when the pumps are running in parallel.

In case of unstable pump characteristics system hunting will occur when the pumps are running in the instability region.

Pumps in parallel are used for stepwise capacity control requirements or for emergency capacity make up situations.

# Combined Head Capacity Curve of Pumps Operating in Series



# Pumps in Series

The combined Head Capacity characteristic of pumps in series is obtained by adding the head developed by each pump at a given capacity as one pump takes suction from the other. This assumes that the head loss between the discharge of the first and and the suction of the second is negligibly small. This is the case when the pumps are close together, otherwise the piping system loss between the two pumps must be deducted fro the combined head.

When the first pump cannot deliver the flow rate at which the second pump is running then instability and cavitation problems will disrupt the series operation. When the second pump is smaller then the first pump can only be utilized in its lower capacity range with the possibility of low capacity associated problems, unless part of its flowrate is taken away from the second pump suction as is the case of smaller booster pumps. The multistage pump is essentially a number of identical pumps in series closely connected together.

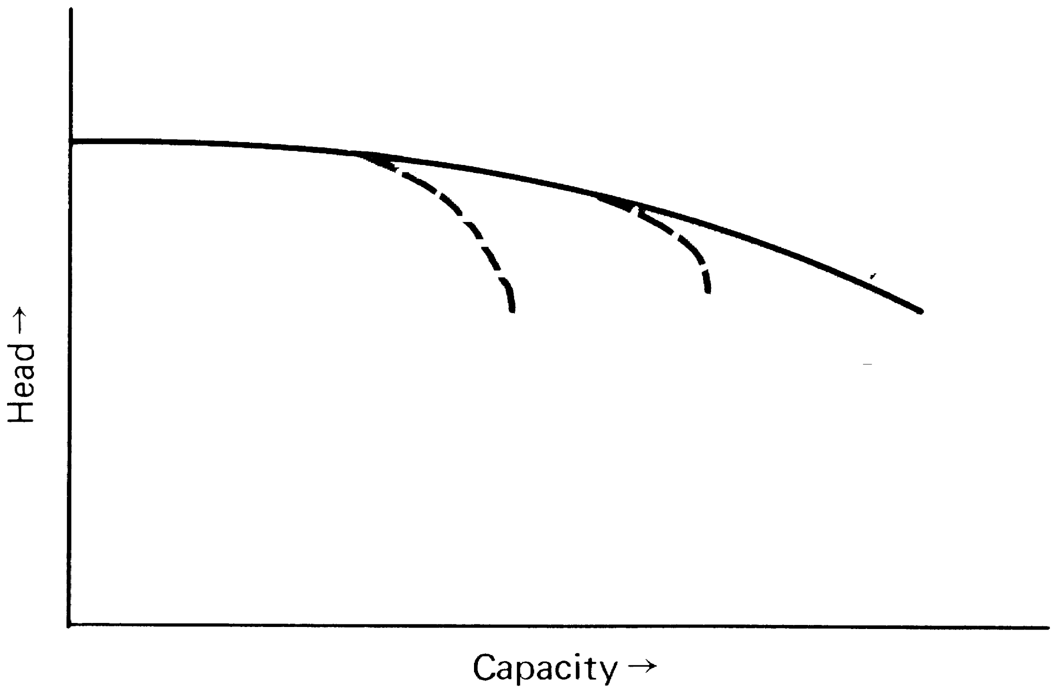
# Cavitation

When the absolute pressure at some point in a liquid reaches the vapour pressure corresponding to the liquid temperature there, bubbles, cavities, form within the liquid at this point. These cavities contain vapour and possibly some gases coming out of solution. When the liquid is in motion and a pressure increase occurs without a substantial change in temperature, as in the case of flow through a running impeller, the vapour in the bubbles cannot be sustained, it condenses very rapidly and therefore the bubbles collapse quite suddenly, giving rise to shocks in the nature of implosions. This phenomenon is called cavitation.

The formation of bubbles gives rise to hydraulic performance reduction possibly to an unacceptably large scale,to rough running and possibly to noise. As the vapour bubbles collapse in an extremely short period of time, in microseconds, pressure shocks traveling at the local speed of sound,  meet the metallic boundaries at very high local surface pressure that can be higher than the ultimate strength or fatigue limit of the material, causing material break up, pitting and erosion followed by corrosion. The damage is more extensive if the bubbles implode on the metallic boundaries.

# Reduction of Radial Pump Performance due to Cavitation

Normally, the NPSH(R) plotted on the traditional pump curve is based on a 3% head loss due to cavitation, a convention established many years ago in the Hydraulic Institute Standards. Permitting this head loss means that cavitation would already have been occurring, at some higher flow condition, before performance loss was noticed.



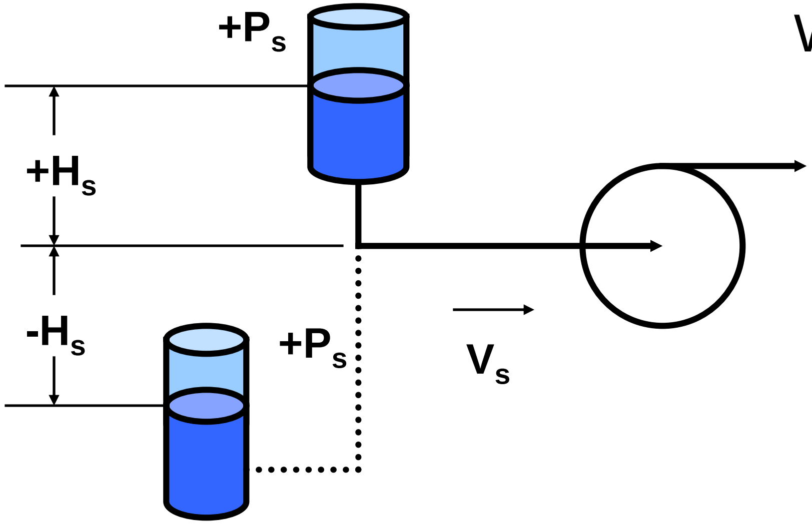
# Net Positive Suction Head (i)

It is possible that the pressure at some point in the liquid flowing into a pump may reach the vapour pressure value of the liquid at the prevailing temperature at this point. Cavitation will then be initiated. It is therefore useful to know the suction head (or energy) at the conditions when cavitation sets in. For the liquid to flow into the pump there must be a positive head causing (energizing) the flow. However the liquid vapour pressure is opposing this flow. The difference between the two is termed the Net Positive Suction Head (NPSH), or Net Positive Suction Energy (NPSE) if energy instead of head is considered.

The cavitating conditions depend not only on the head or energy available but also on the pump geometry and surface finish. Therefore the pump requirement is distinguished from the system head availability by the Net Positive Suction Head Required NPSH(R) and the Net Positive Suction Head Available NPSH(A).

NPSH(A) depends on the pump suction system (piping etc) design and conditions and NPAH(R) depends on the pump design, construction and condition. It is advisable to have the NPSH(A) by at least 10% higher than NPSH(R), or by at least 1.5m whichever is higher.

# Net Positive Suction Head (ii)




Where $V_s$ = flow velocity in pump suction
$P_s$ = pressure in suction vessel at liquid interface
$H_s$= liquid interface level elevation relative to pump datum line
$H_f$ = frictional head loss in suction system
$P_v$= liquid vapour pressure at pumping temperature
ρ = liquid density at pumping conditions

$$NPSH(A) = P_s/\rho + V_s^2/2g - P_v/\rho \pm H_s - H_f$$

It follows that NPSH(A) can be increased by increasing $P_s$ and reducing $P_v$ but in cases when the liquid in the suction vessel is condensing from the vapour increasing $P_s$ will increase $P_v$ by the same amount and therefore there will be no net change in the value of NPSH(A). Increasing $V_s$ will have a very small effect and it will also increase $H_f$.Simplifying the suction line can reduce $H_f$. Raising the suction vessel increases $H_s$ and so NPSH(A). It can be increased by reducing $P_v$ but cooling the liquid going into the pump presents difficulties.

# Net Positive Suction Head (iii)

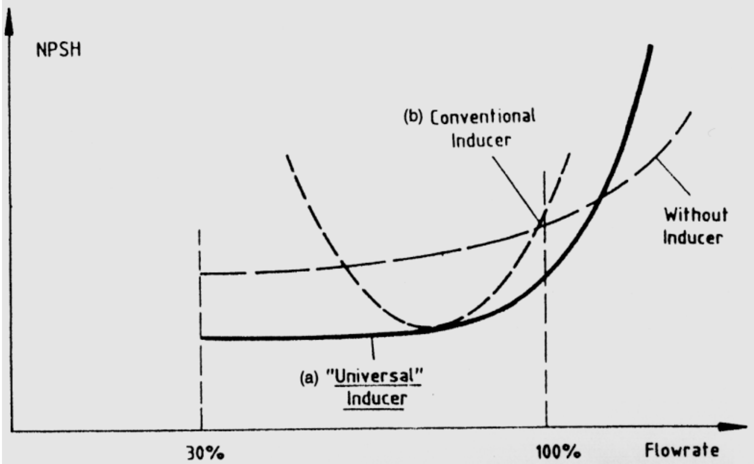
Other practical means of improving suction conditions are:

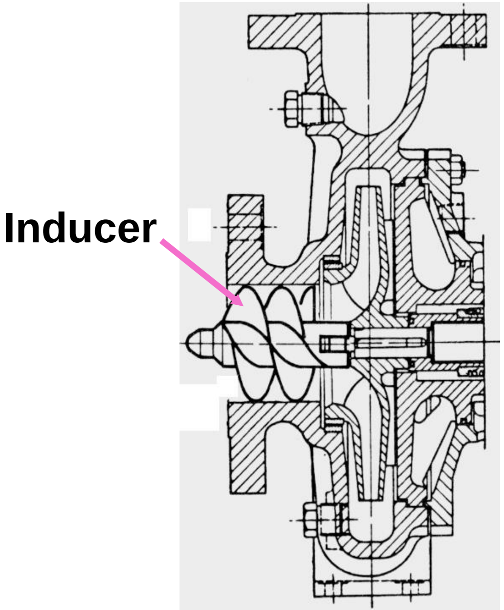
- Keep suction strainers clean
- Prevent liquid pre-rotation at entry by introducing a vortex breaker at the outlet of the suction vessel, the inlet of the pump, relocating pipe elbows well away from pump suction. Flow streamlining baffles may also help, especially in the case of large diameter suction pipes.
- Reduce shock losses away from the design point, particularly if the pump runs over prolonged periods away from design. (e.g. bull ring installation in the impeller eye, sharpening vanes at entry - for low flowrate range - or enlarging impeller inlet area by boring and removing small part at the inlet of the vanes - for large flow rates. Interfering with the impeller is not recommended unless in cases of emergency needs).

# Net Positive Suction Head (iii)

If the pump NPSH(R) is too high for the prevailing conditions and increasing the suction diameter is impractical, an inducer may be fitted in the case of overhang end suction impellers. The inducer becomes effectively an axial first stage. The hydraulic penalty is increased power input requirement. Furthermore the inducer and impeller can only be matched at one flow rate causing additional losses away from the design point and further efficiency reduction.

Well designed inducers are capable of lowering NPSH(R) over a large part of the operating range.



# Normal Cavitation

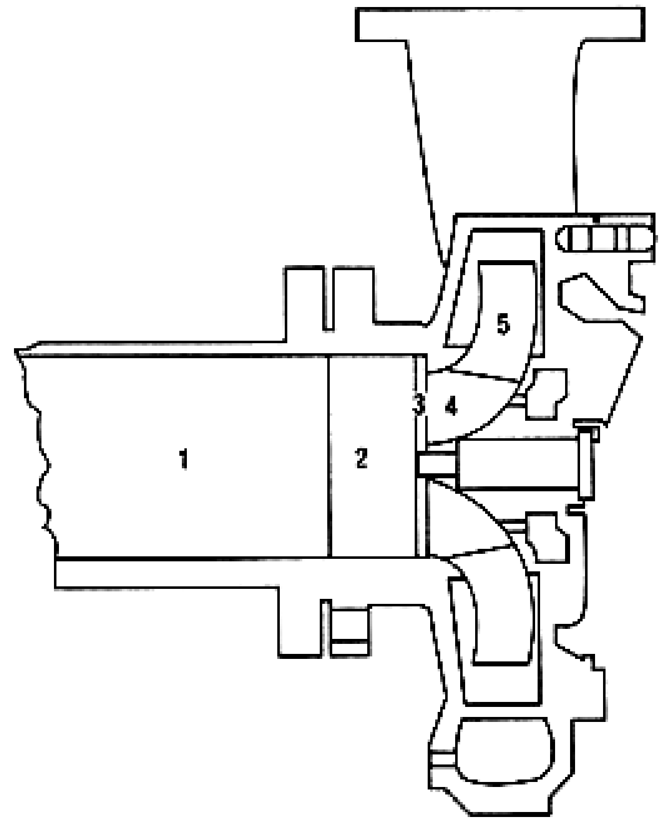
When cavitation occurs due to insufficient head at the pump suction the type of cavitation taking place is termed **Normal Cavitation.**

**Normal Cavitation** results in:-

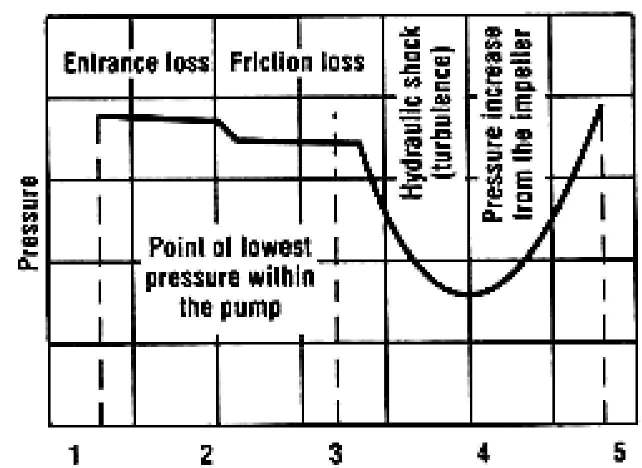
- Pump performance deterioration - reduction in total head developed, flowrate reduction, pump efficiency reduction.
- Increase in the level of pump vibrations and of noise level, flowrate pressure and power input fluctuations if the cavitation is severe enough.
- Pump impeller and possibly casing pitting and erosion.

In carrying out cavitation tests (constant speed and flowrate but decreasing suction head) the point at which the **total head** developed falls by a predetermined amount (usually 3%) is regarded as the point at which the pump becomes critical.

# Normal Cavitation




The pressure of the liquid going into a centrifugal pump drops as it flows from the suction pipe through the suction nozzle and into the impeller. The pressure drop is a function of many factors, including pump geometry, rotational speed, frictional and hydraulic shock losses, and rate of flow.




If the pressure at any point within the pump falls below the vapor pressure of the liquid being pumped, vaporization and so bubble formation will occur. As the pressure increases the local pressure is above the vapour pressure and so the bubbles collapse spontaneously and cavitation occurs

# Evolved Gas Cavitation

Pumped liquids may contain gas or gases in solution. As the pumped liquid pressure is minimum at the impeller eye or at other points due to re-circulating flow gas may come out of solution in the form of bubbles. This form of cavitation is called “**Evolved Gas Cavitation”**

In the case of a moderate amount of gas release a stream of bubbles evolve causing a reduction of the pump flowrate. The bubbles are drawn by the liquid stream and may cause erosion of the metal boundaries. In addition the impeller may hold gas bubbles near the periphery of the eye causing an unsteady blockage causing fluctuations in the flow. Eventually if the release of as out of solution is taking place at a high rate the pump performance could be severely hindered.

# Re-circulation Cavitation

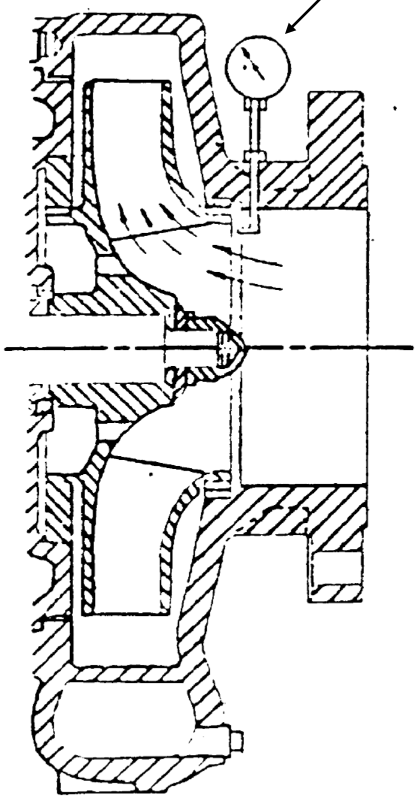
As the flowrate decreases below the design value the pressure at the periphery of the impeller tends to decrease in value just after each vane as the passage between the vanes is larger than necessary. At the lower pump flow-rates there actually can be a flow reversal when part of the liquid re-enters the impeller in these lower pressure regions. The re-entering liquid follows the back of the vane until it meets the liquid entering from the eye of the impeller. At this point there are abrupt flow reversals that may cause cavitation. This form of cavitation is termed **"discharge re-circulation cavitation".**

On the other hand when with the reduction of flow eddies and flow reversals take place at the eye of the impeller incoming flow velocity increase and pressure reduces. When the due to pressure reduction the vapour pressure is reached cavitation occurs. This is termed **"suction re-circulation cavitation"**

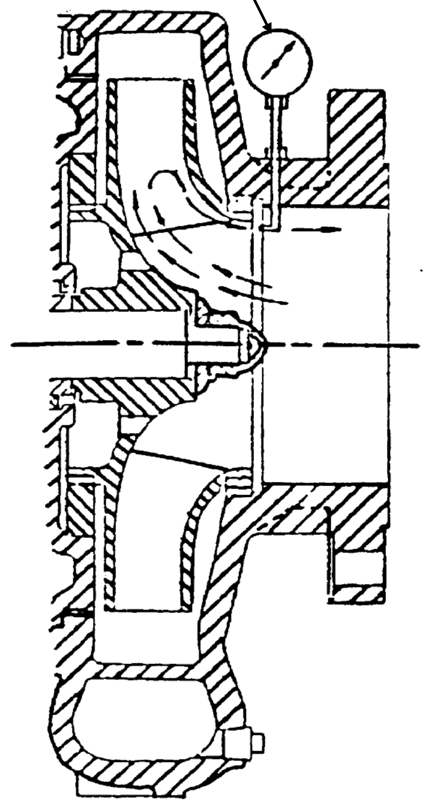
Re-circulation cavitation  is accompanied by crackling noise and high intensity knocks and fluctuations. It causes pitting and erosion at the point of the abrupt flow reversals - **at the visible part of the vane at impeller outlet and at the non-visible part of the vane at the impeller inlet.**

# Recirculation Cavitation Flow Regime

Directional Head Probes



Normal Flow



Suction Recirculation Flow

During Normal Flow the probe is essentially reading the suction pressure less the velocity head pressure.

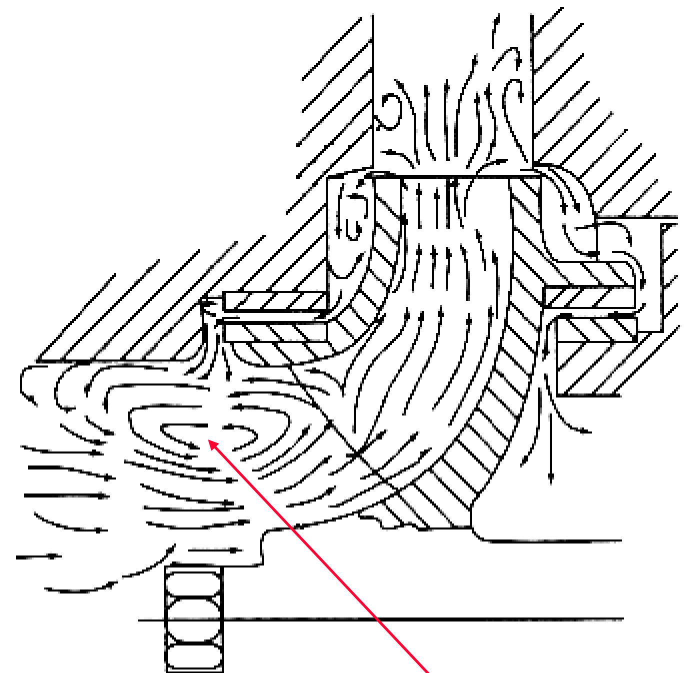
When suction re-circulation flow occurs the probe shows a sudden rise in pressure due to the flow reversal there

When discharge recirculation occurs a substantial increase in pressure pulsations occurs .

# Suction Recirculation Caviation (i)

As the pump flow is reduced below the Best Efficiency Value, flow reversals and eddy currents begin to form in the eye of the impeller. That is, the re-circulating flow eddy currents at the eye have effectively reduced the flow channel size, thereby increasing liquid velocity for the fixed flowrate.

When the velocity increases, the pressure reduction increases and more so due to increased friction. If the drop is large enough to cause the pressure to fall below the vapor pressure of the liquid, the pump will develop cavitation because of this action of recirculating flow. The liquid enters the impeller at a central core directing the cavities at the low pressure side of the vanes at inlet.



**Vortices in suction pipe and impeller eye.**

# Suction Recirculation Caviation (ii)

Symptoms associated with suction recirculation:-

1. Cavitation damage to the pressure side of the vanes at impeller inlet

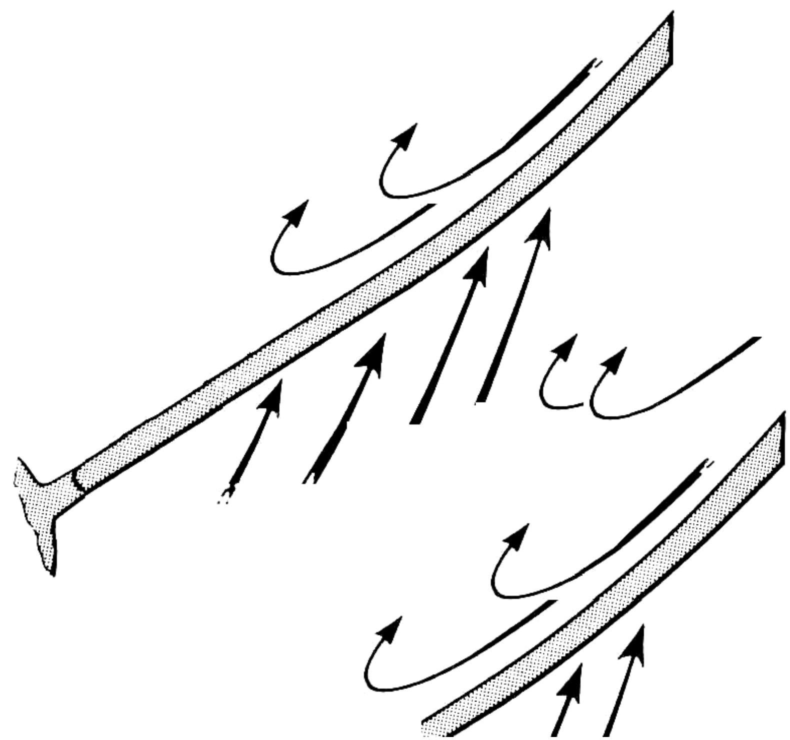
2. Cavitation damage to the stationary vanes (if fitted) in the pump suction

3. Random crackling noise in the suction

4. Surging and pulsations

# Discharge Recirculation Cavitation (i)

The discharge recirculation phenomena are due to collapsing cavities (bubbles of liquid vapour). However it is difficult to imagine how localized pressure at the discharge of an impeller developing high head is reduced to vapour pressure value. When the recirculating flow entering the impeller discharge passage under low capacity conditions meets the oncoming flow from the impeller eye a high velocity vortex is formed. Cavitation will follow as soon as the pressure at the vortex core reaches vapour pressure.



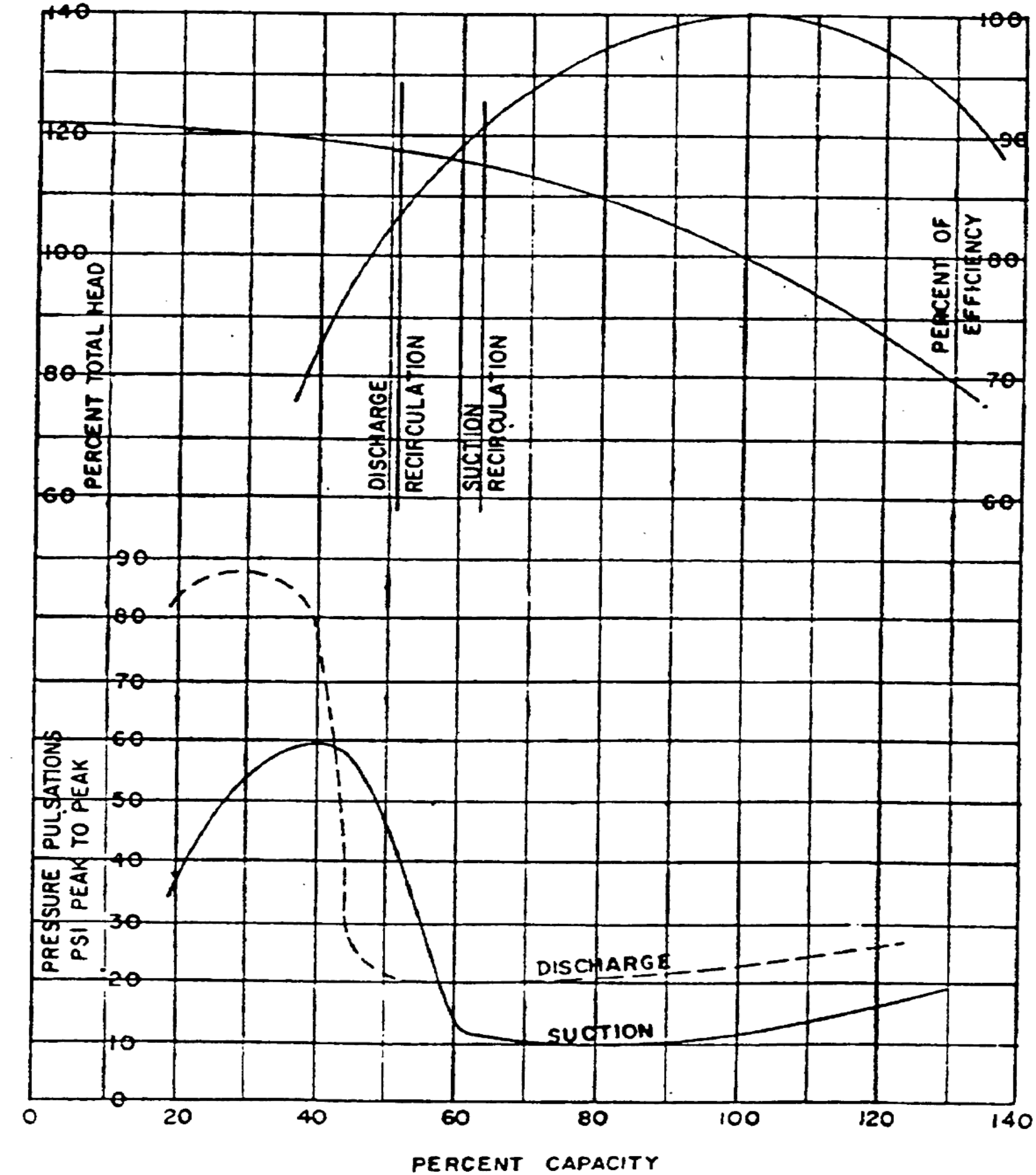
The bubbles formed in this way travel outwards and collapse in the high pressure regions causing damage to the high pressure side of the outward part of the vane and possibly the impeller shroud.

# Discharge Recirculation Cavitation (ii)

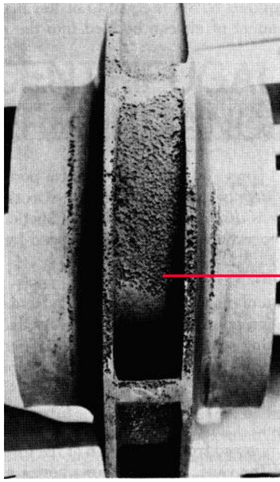
Symptoms associated with discharge recirculation:-

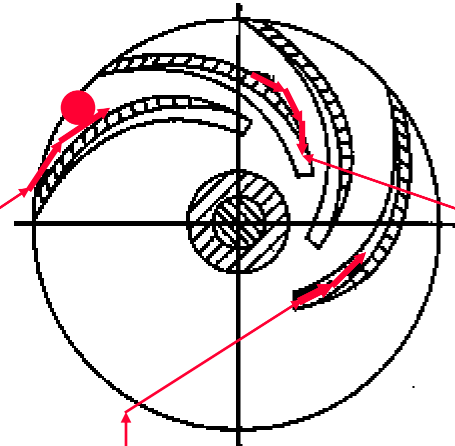
1. Cavitation Damage to the pressure side of the vane at the discharge
2. Axial movement of the shaft with or without damage to the thrust bearing
3. Failure of the impeller shrouds at the discharge of the impeller
4. Shaft failures on the outboard end due to fatigue
5. Cavitation damage to the casing throat or the diffuser vanes of a diffusion casing

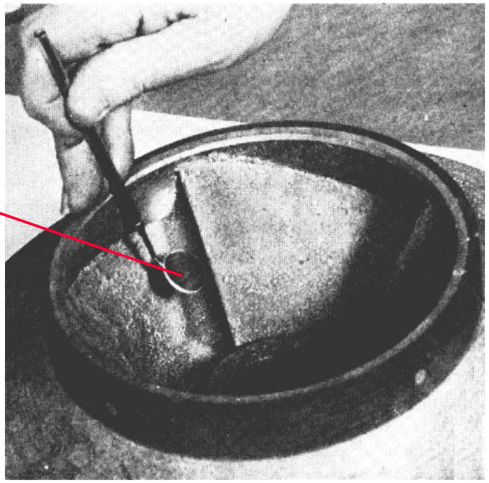
# Tendency for Recirculation Cavitation



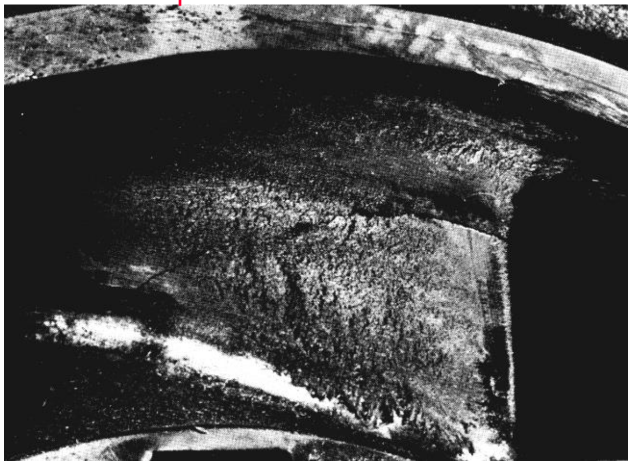
# Cavitation Damage on Pump Impellers







**Discharge Recirculation** damage on the pressure surface of the discharge part of the impeller vane. The damage can extend to the impeller shrouds at the discharge part of the impeller.



**Examining the Non Visible Side of the Vanes at Inlet for Suction Recirculation Cavitation Damage**

**Normal Cavitation Damage on the Visible Part of the Vanes at Inlet**

# Avoiding Cavitation (i)

To correct pump cavitation of a pump already installed implies modifications and additional cost. Some suggestions are given below:-

A. Normal Cavitation

Increase Pump Suction Pressure. However in cases when the liquid in the suction vessel is condensing from the vapour increasing the suction pressure will increase the vapour pressure by the same amount and there will therefore be no net effect {in increasing the value of NPSH(A)}.

Reduce Liquid Vapour Pressure. - Cooling the liquid going into the pump on many occasions is not a practical possibility.

Increase Static Head at Suction. Increasing the elevation of the suction vessel or lowering the pump on certain occasions may be possible. This could be one very effective solution.

Simplifying the Suction Line Configuration with the object of reducing frictional line losses.

# Avoiding Cavitation (ii)

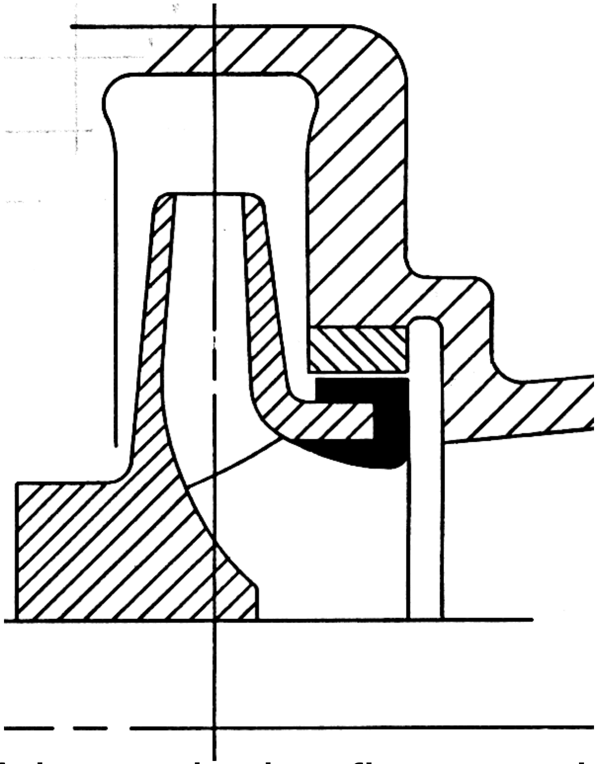
Keep Suction Strainers Clean.

Prevent Liquid Pre-rotation at Entry by introducing a vortex breaker at the outlet of
the suction vessel, the inlet of the pump. Also it may be necessary to relocate pipe
elbows well away from pump suction.

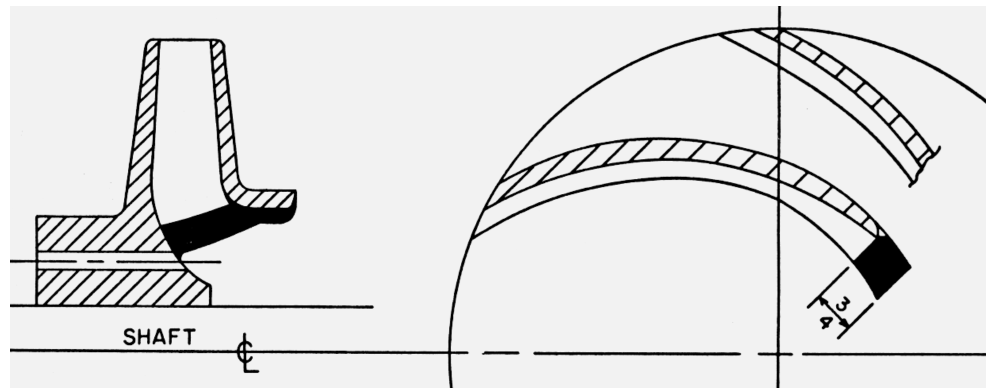
Control Pump Flow Away from the Regions of Cavitation. Controlling process flow is not always possible. A pump by pass may be opened to increase the pump flow away  from cavitation at low flows or a pumping parallel to reduce high cavitating pump flows.

Install an Inducer. This could be possible for an overhang impeller type pump. However the inducer has to be matched with the impeller.

# Avoiding Cavitation (iii)



If the cavitating flows are low in comparison to the design capacity the installation of a bull ring streamlined to match the impeller is beneficial.




If the cavitating flows are high in comparison to the design capacity removing metal from the vanes at inlet increases the flow area and reduces inlet velocity, reducing cavitating conditions. In many cases it is not a very practical solution but in emergencies could give relief.

# Avoiding Cavitation (iv)

B. Recirculation Cavitation.

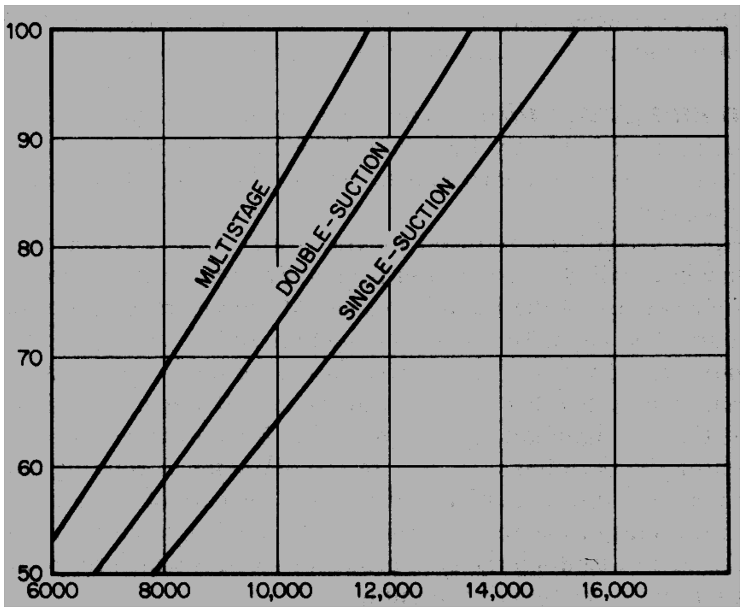
1. Control the flow rate away from the recirculation cavitation range.
   (a) By process flow control. It is not always appropriate from the process requirements to do so.
   (b) By bypassing a sufficient quantity back to the suction system so that the pump deals with a quantity larger than the process requirement although the process flow rate remains unaltered.
2. By installing a second smaller pump to deal with the lower capacity requirements whilst the bigger pump remains stand-by for the higher process flow requirements.
3. The range of the expected capacity range must be specified prior to the purchase of the pump so that the necessary considerations can be made prior to the purchase.

NB It is inescapable that suction and discharge recirculation will occur at some critical capacity range below the Best Efficiency Point and measures must be taken at the design stage to take into account the additional forces and shocks associated with recirculation cavitation.

# Avoiding Cavitation (v)

For Pumps with Suction Specific Speeds 500 to 2500 (USCS)

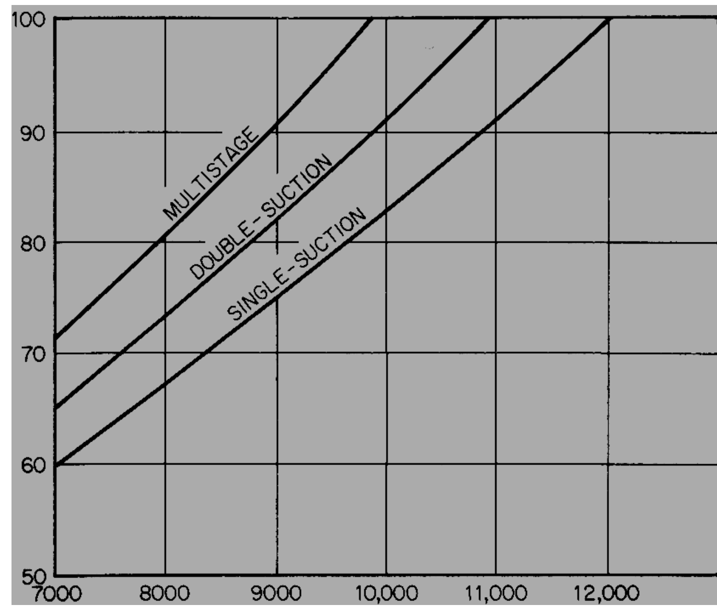
Suction Recirculation Incidence % of Best Efficiency Point Flow




Suction Specific Speed $S_s$ in US Consistent System

For Pumps with Suction Specific Speeds 500 to 2500 (USCS)

Suction Recirculation Incidence % of Best Efficiency Point Flow




Suction Specific Speed $S_s$ in US Consistent System

These graphs are of value to the user as they establish limits and guide lines in the selection of centrifugal pumps. It is evident that recirculation cavitation can be delayed by seeking lower suction specific speed values and limiting the range of operation to capacities above the point of recirculation.

# Avoiding Cavitation (vi)

Discharge re-circulation cavitation has been reported to be put into advantage in increasing head at low flow and at the same time avoiding the re-circulation damage. Two methods are under conideration.

1. Open up slots on the closed impeller shrouds at the convex part of the vanes as in Fig 1. These slots allow liquid  from the discharge to return and fill up the passages and at the same time pick up additional energy thus increasing the pressure and reducing cavitation

2. Remove parts of the shrouds near the impeller outlet as in Fig 2. This has a similar effect as the slots in the previous case.

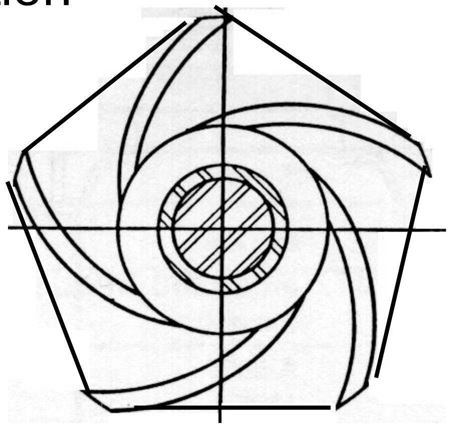


Fig 2

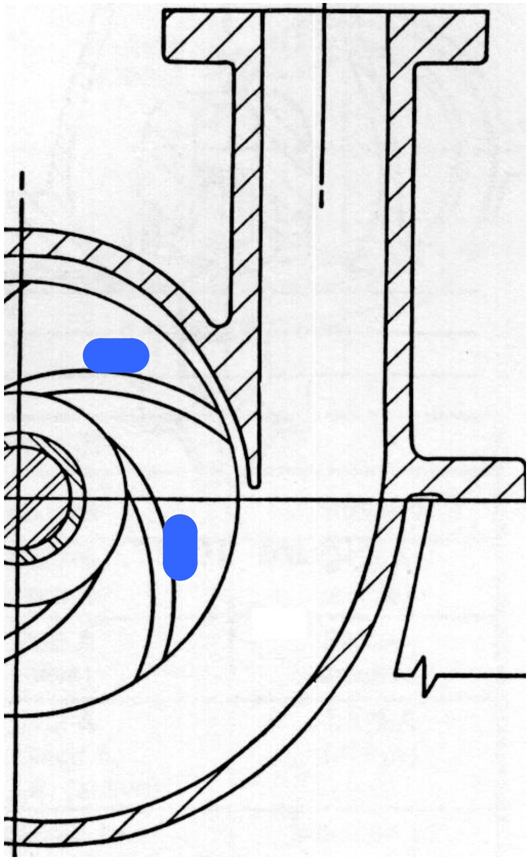


Fig 1

# Axial Thrust

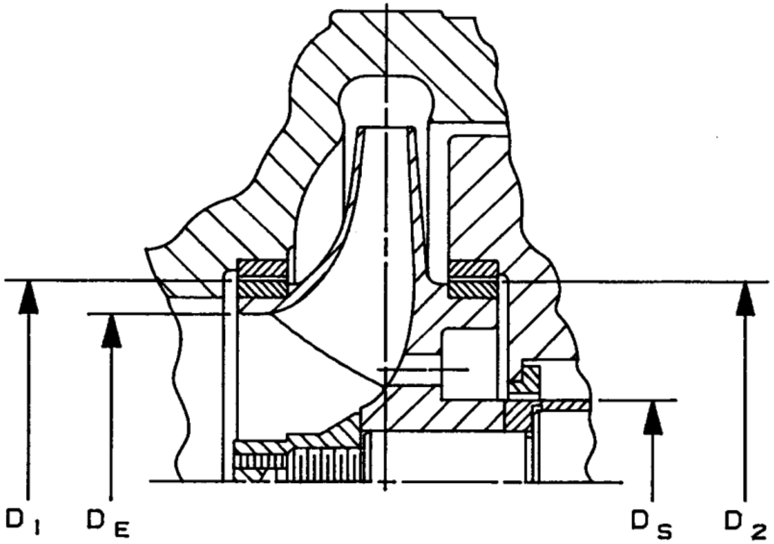
Axial Thrust loads result from internal pressures acting axially on the exposed areas of the rotating element. If these axial loads are not balanced there is a net axial thrust on the rotating element. Since the radial stage differential pressure increase is high axial thrusts are high whilst as the pump specific speed number increases the stage differential pressures decrease so does the net axial thrust - being comparatively very low in the case of axial pumps.

Axial thrust acts on single suction impellers because the area of the eye of the impeller is under suction pressure and the back shroud of the impeller is acted upon by a pressure higher than suction pressure and closer in value to discharge pressure. Calculating the axial thrust on an impeller requires consideration of the different design variations used in these pumps.

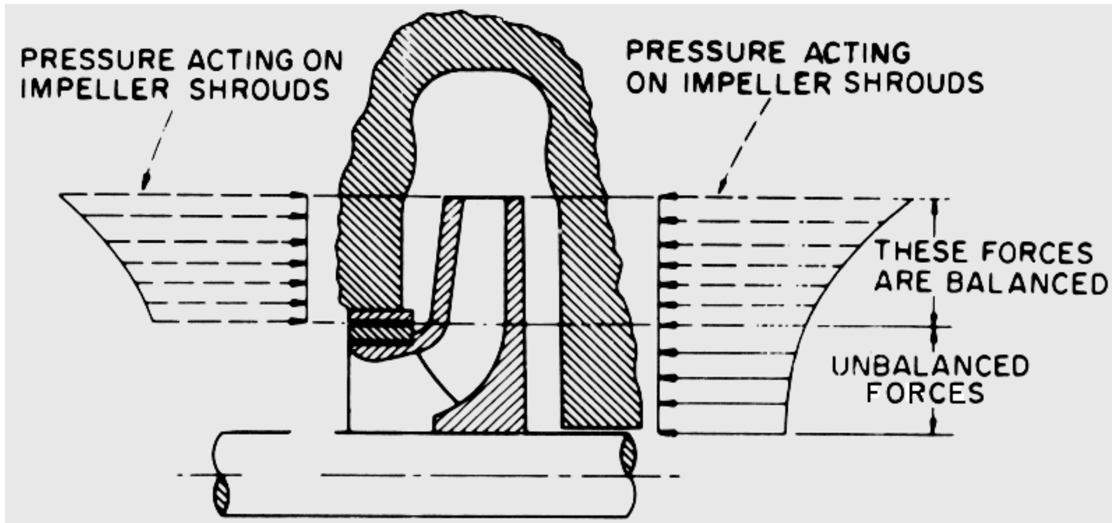
# Balancing of Axial Thrust (i)

The axial thrust can be reduced or counterbalanced by several methods

By introducing a set of wear rings at the back of the impeller of approximately the same diameter as the wear rings at the eye of the impeller and drilling balance holes in the back shroud connecting the area enclosed by the wear rings at the back of the impeller with the impeller suction area.
This will increase by a small amount the internal leakage loss from discharge to suction and so somewhat the power input to the pump, simultaneously reducing slightly the total head developed and the pump efficiency. In addition the jetting action of the flow through these balance holes causes some hydraulic disturbance to the impeller inlet flowpattern. To avoid this disturbance instead of the balance holes a channel can be used to connect the balancing space to the pump suction.



# Magnitude of Axial Thrust (i)




The liquid pressure on the closed impeller shrouds is not uniform but parabolic as shown by the diagram. For simplification of the calculations the parabolic distribution can be ignored and instead an average uniform pressure of 75% of the discharge pressure (3/4$P_D$**) may** be assumed as acting on the shrouds.

# Magnitude of Axial Thrust (ii)

The Axial Thrust for the example of the horizontal pump under steady conditions in the diagram is given approximately by:-

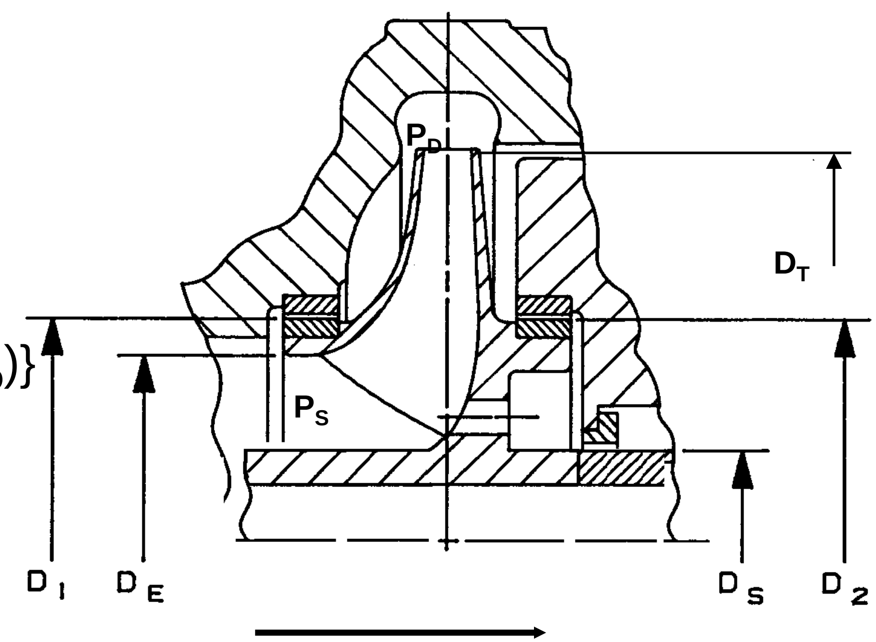
In Direction X = $(T_X)$

$= \pi/4\{(D_T^2 - D_1^2)*(3/4P_D) + (D_1^2 - D_S^2)*(P_S)\}$

Opposite to Direction X = $(T_{OX})$ =

$= \pi/4\{(D_T^2 - D_2^2)*(3/4P_D) + (D_2^2 - D_S^2)*(3\%P_D)\}$
$+\rho Q(\Delta V_{ax})$

**Total Axial Thrust = T**
**$T = (T_{OX}) - (T_X)$**

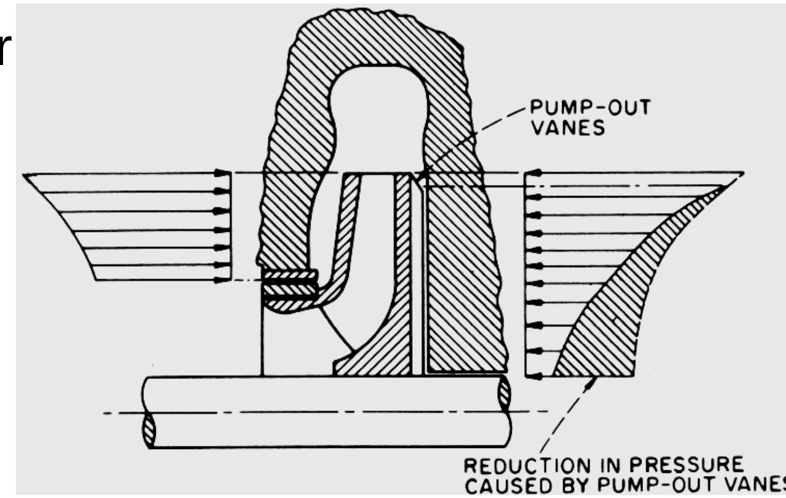



Direction X

where $\rho Q(\Delta V_{ax})$ is the change in momentum in the axial direction,
$\rho$ is the density @ pumping conditions, Q the quantity and $(\Delta V_{ax})$ change in the value of the axial velocity

# Magnitude of Axial Thrust (iii)

The Axial Thrust for the example of the horizontal overhang impeller pump under steady conditions in the diagram is given approximately by:-




In Direction X = ($T_X$)

$= \pi/4\{(D_T^2 - D_1^2)*(3/4P_D) + (D_1^2)*(P_S)\}$

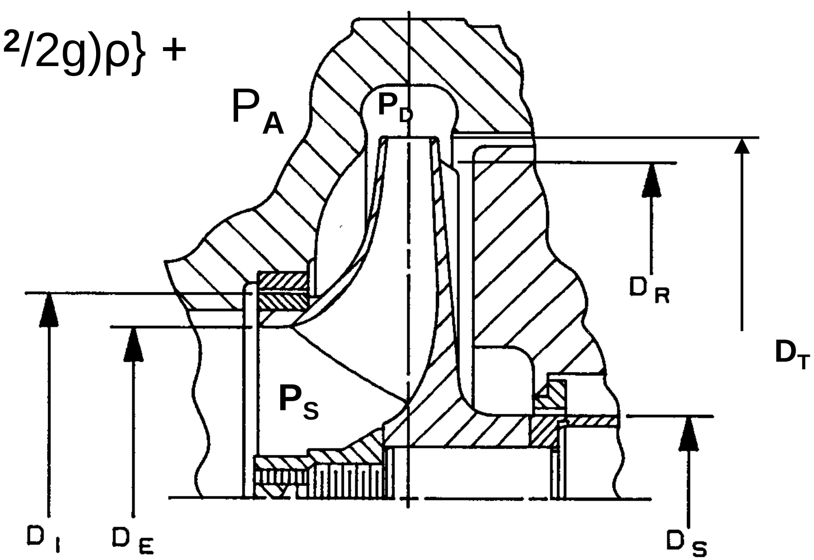
$- (\pi/4)\, D_S^2 P_A$

Opposite to Direction X = ($T_{OX}$) =

$=\pi/4\{(D_T^2 - D_s^2)*(3/4*P_D) + (D_R^2 - D_S^2)*(3/8*\Delta u^2/2g)\rho\} +$

$\rho Q(\Delta V_{ax})$

**Total Axial Thrust = T = ($T_{OX}$) - ($T_X$)**



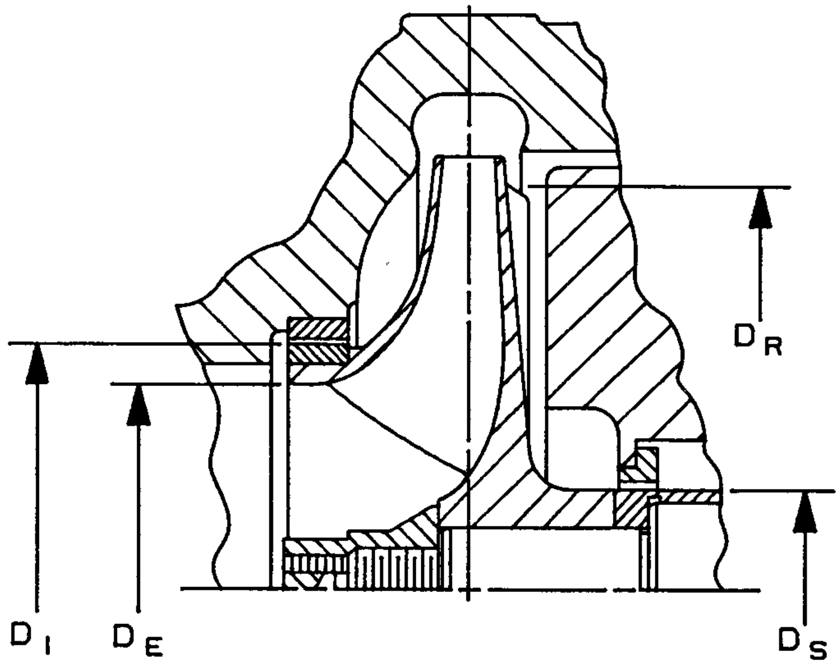

where pressures P are in absolute values and $P_A$ is atmospheric pressure

$\Delta u^2$ is the difference of the square of the peripheral velocities at the tip and root diameters of the pump out vanes.

$\rho Q(\Delta V_{ax})$ as on the previous slide

# Balancing of Axial Thrust (ii)

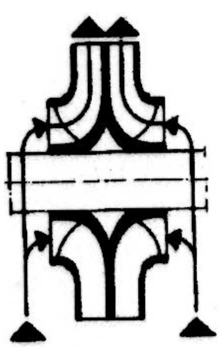
Another method is to provide pump out vanes on the reverse of the impeller back shroud. The pump out vanes require a small amount of additional power to drive the impeller for a certain capacity. In case the pump handles liquids with a certain amount of solids, the pump out vanes keep the space at the back of the impeller clean, an advantage in case mechanical seals and especially of packed glands It has been shown that this method requires no more power than that which is lost by the wear ring increased leakage in the previous method of back shroud wear rings.

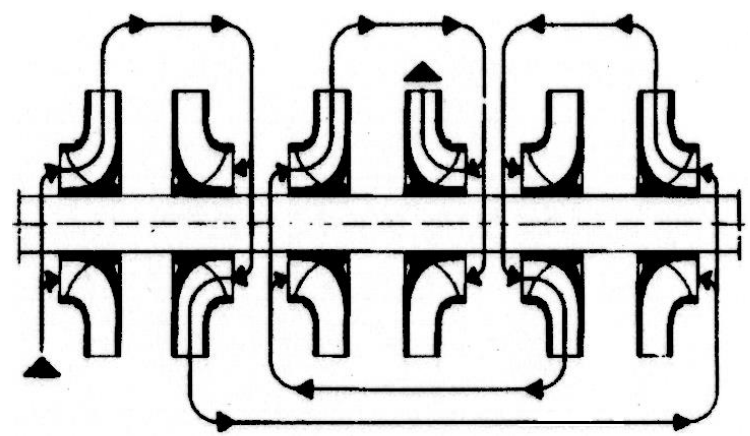


# Balancing of Axial Thrust (iii)

Double suction impellers are theoretically balanced as regards axial thrust due to symmetry. To achieve axial thrust balance it is important to achieve hydraulic symmetry in the approach to the eye of the double suction impeller. Symmetrical double suction pumps have a small thrust bearing to locate the rotating element and to take up any possible axial thrust due to casing and impeller machining and installation variations, shaft end play, fluid flow anomalies and turbulence.

A similar situation prevails when a multistage pump with an even number of stages has the impellers arranged in two opposing groups. In this case in addition to the drawbacks mentioned above there are increased internal losses due to friction in the interstage liquid passages.



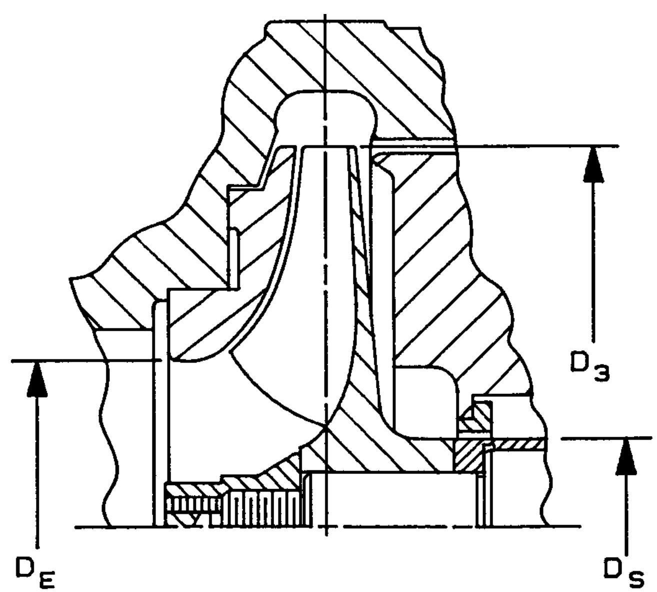


# Balancing of Axial Thrust (iv)

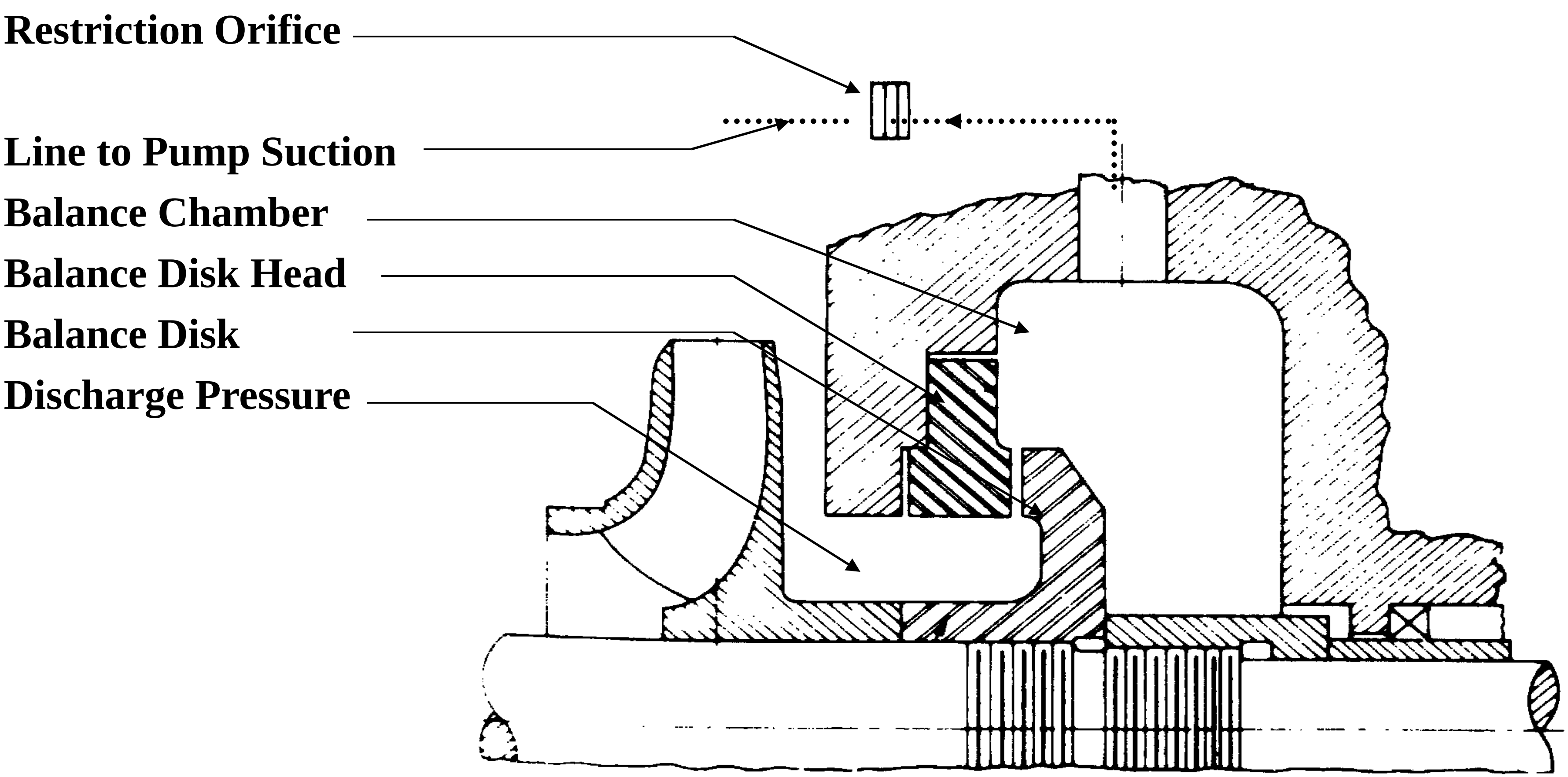
Semi-open impellers produce higher axial thrust than closed impellers. In the case of semi-open impellers the gap between the vanes and the wear liner must be carefully controlled at assembly as larger gaps will increase interstage leakage loss and power input and lower total head and overall efficiency.

Pump out vanes will reduce axial thrust but additional assembly difficulties arise as the back impeller clearance together with the front clearance has to be adjusted.

To reduce the axial thrust on semi-open impellers part of the back shroud can be removed.

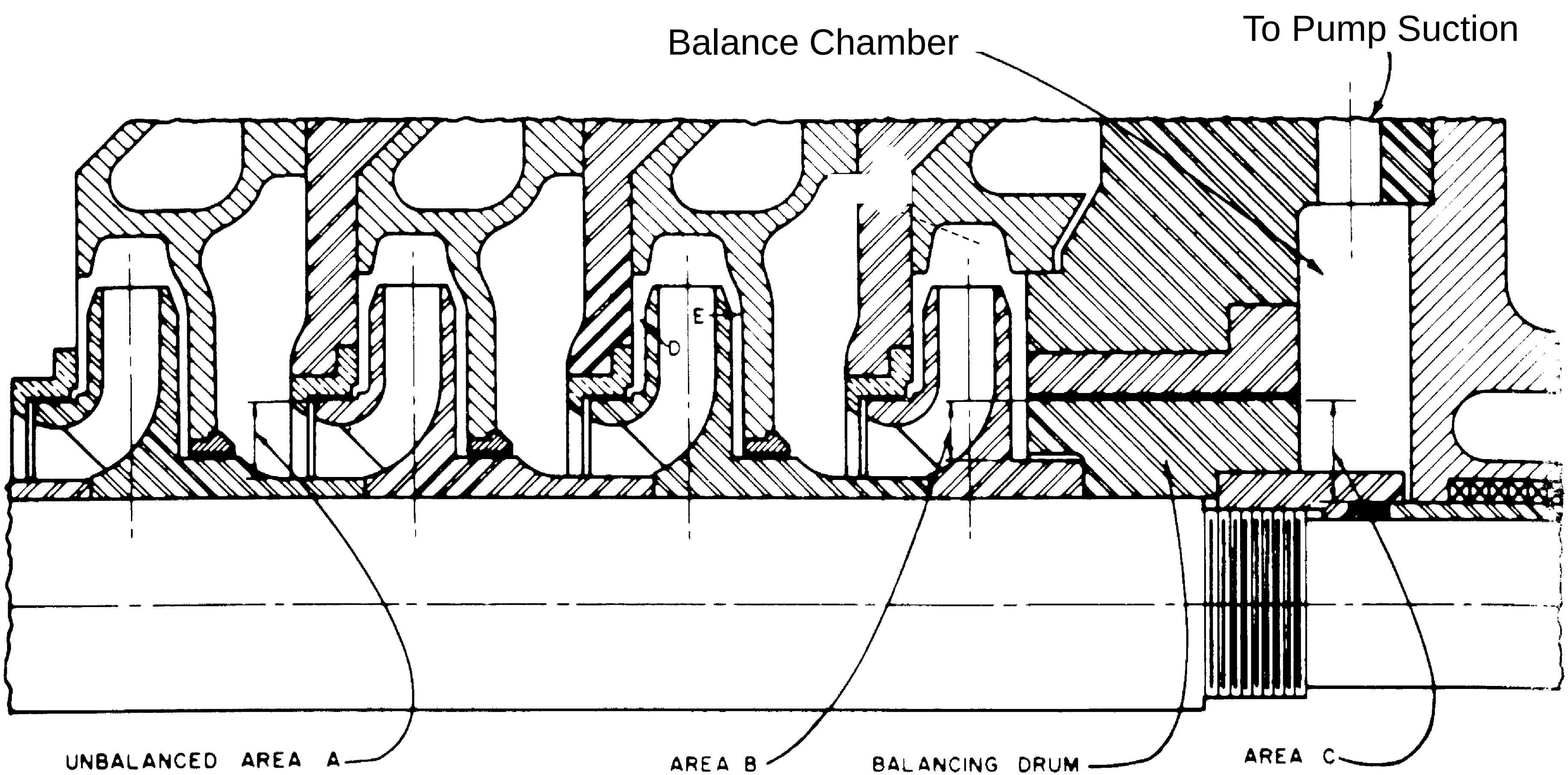


# Simple Axial Thrust Balancing Disk
## for a Multistage Pump



# Axial Thrust Balancing Drum

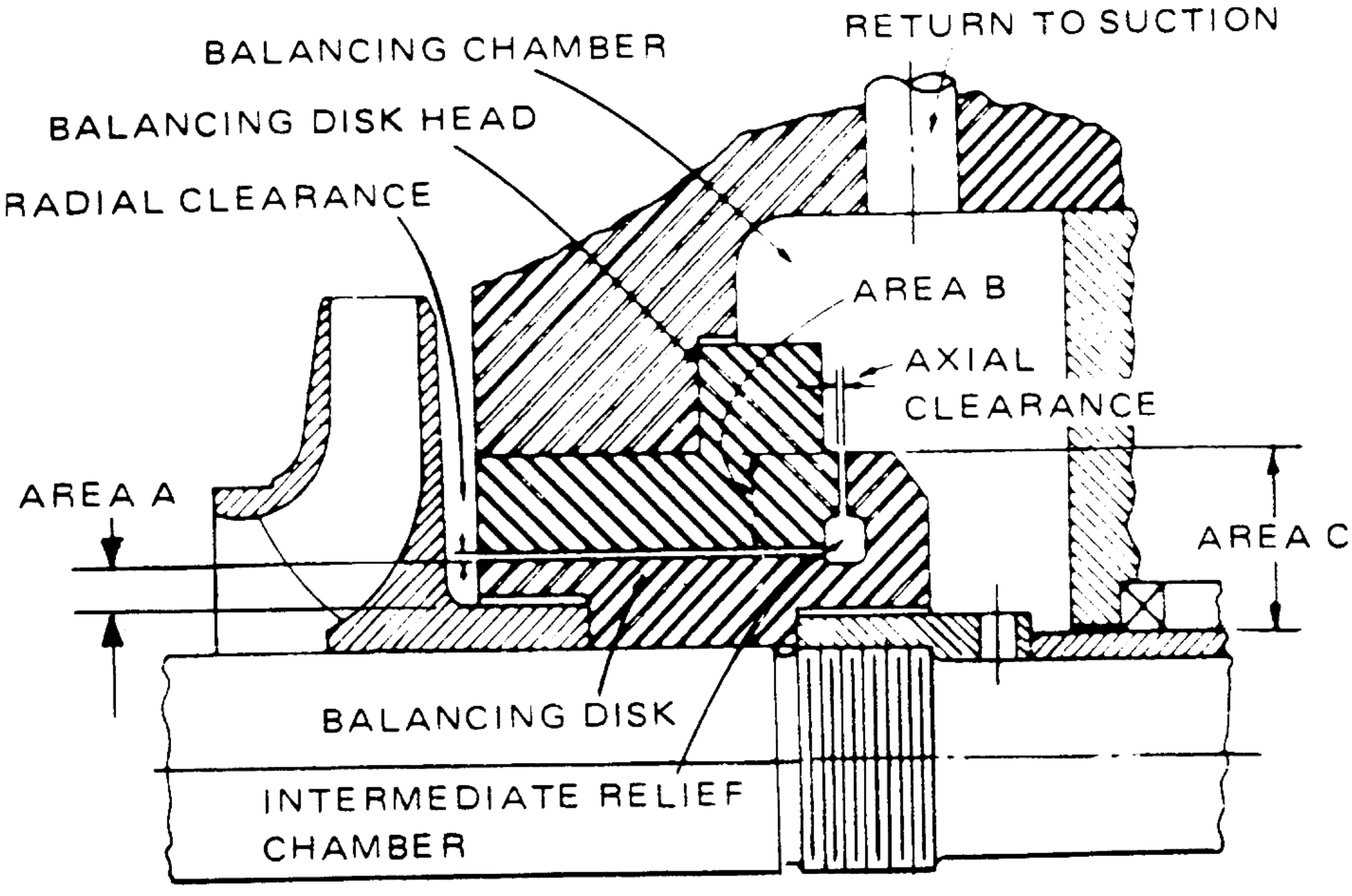
## for a Multistage Pump




At pump design conditions $\{(P_d - P_s)/n\}*A - P_d *B + P_c *C = 0$
where n = number of pump stages, $P_d$ =pump discharge pressure,
$P_c$ =balance chamber pressure = pump suction pressure ($P_s$) approximately

# Combined Axial Thrust Balancing Disk & Drum

## for a Multistage Pump



# Axial Thrust in Practice

In practice Axial Hydraulic Thrust on impellers are affected by:-

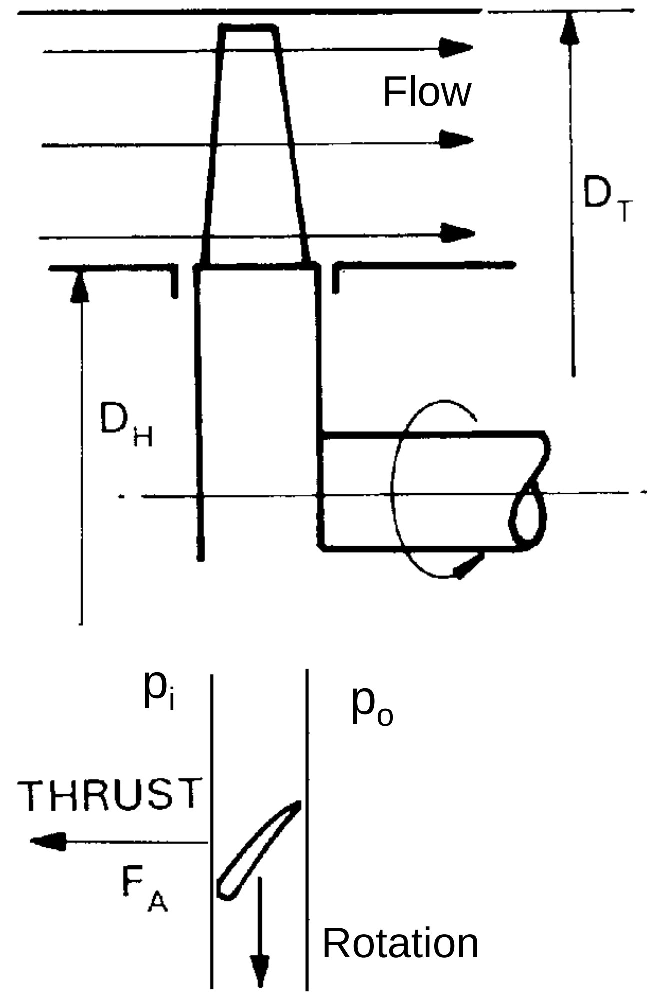
1. Suction Passages.
2. Suction Piping Conditions
3. Location of the Impeller in relation to the Casing
4. Internal Leakage through leakage passages such as wear rings, balance lines etc.
5. Cavitation
6. Pump Load
7. Condition of Impellers

# Axial Thrust in an Axial Flow Pump

There is an axial thrust due to the change in the direction of flow and the pressure difference across the pump stage.

$$F_A=(p_o-p_i)*(\pi/4)*(D_T^2-D_H^2)$$

The pressure all round the circumference is uniform so the radial thrust is fully balanced.



# Radial Thrust (i)

In a centrifugal pump if the flow rate is close to the design value the pressure distribution around the impeller does not vary substantially and so the net radial load on the impeller is small. When however the the flow rate reduces the pressure distribution variation around the impeller shows marked variation and this results in a large net radial load on the impeller.

Double volute casing design are introduced to significantly reduce the net radial thrust. The greater the symmetry of the double volute design the greater the reduction of the radial thrust. It is worth noting that tests have shown that double volute casing designs result in one (1%) to one and a half (1 1/2%) percentage points reduction in pump Best Efficiency Point (BEP) but up to (2%) two percentage points improvement in pump efficiency on either side of BEP. For large pumps double volutes can be fairly easily constructed but difficulties increase as the pump becomes smaller.

# Radial Thrust (ii)

The closer a casing is to a circular configuration the lower the net radial thrust but a reduction in efficiency values is observed. Partial emission pumps which have circular casings show a far grater uniformity of velocity and pressure distribution but only at no flow conditions where the radial thrust is a minimum. The value increases steadily as the flow rate increases.

The radial thrust $F_R$ is a function of the width (b) and diameter (D) of the impeller and of total head (H) developed i.e.

$$F_R = k.K.w.H.D.b$$

where k is a dimensional constant K is an experimental constant depending on the pump specific speed ($N_s$) and $Q/Q_{BEP}$, the smaller $Q/Q_{BEP}$ the higher the value of K.  w is the specific weight of the pumped liquid (at pumping conditions).

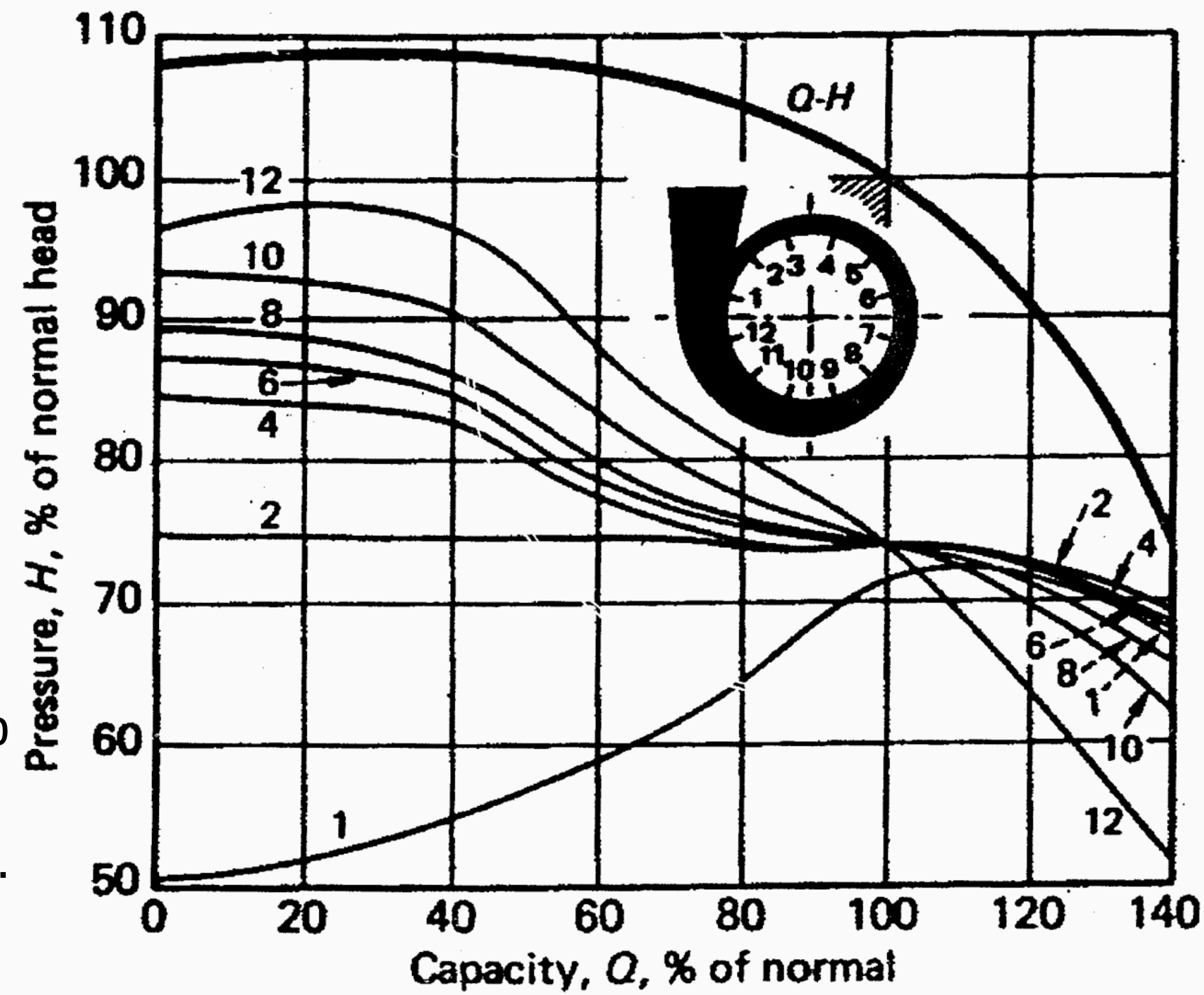
Many multistage casing designs have staggered volutes so that the stage radial thrusts tend to balance out.

Shaft diameter and bearing load capacity have to be carefully designed to take radial thrusts at low and maximum flow rates. Fatigue failure of shafts occurred when high radial thrusts at low flow rates were not taken into account.

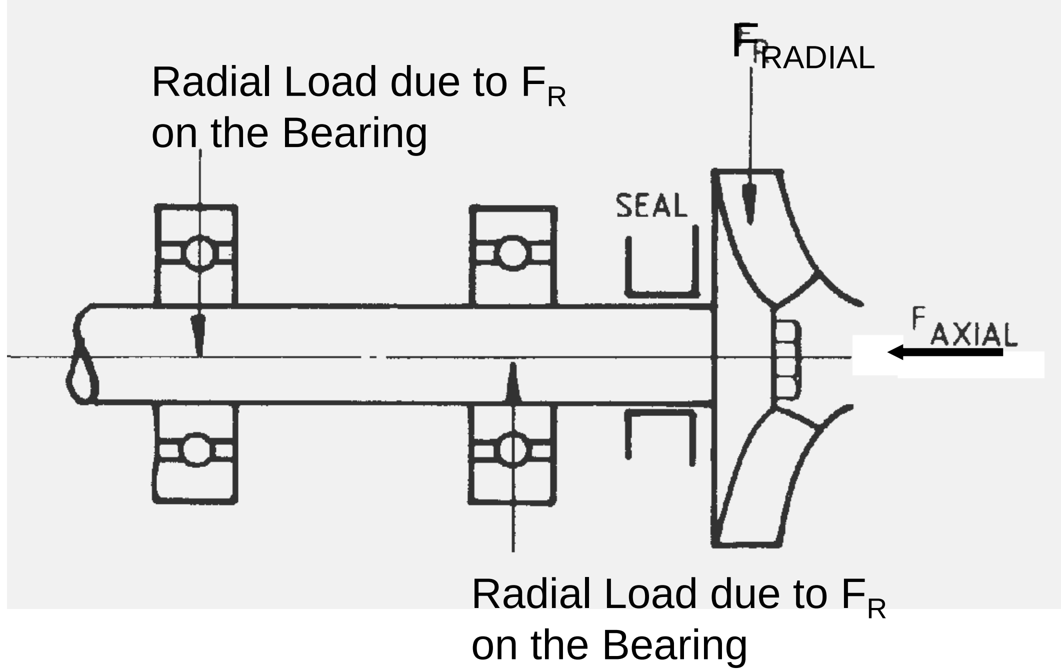
# Pressure Variation in a Pump Volute.

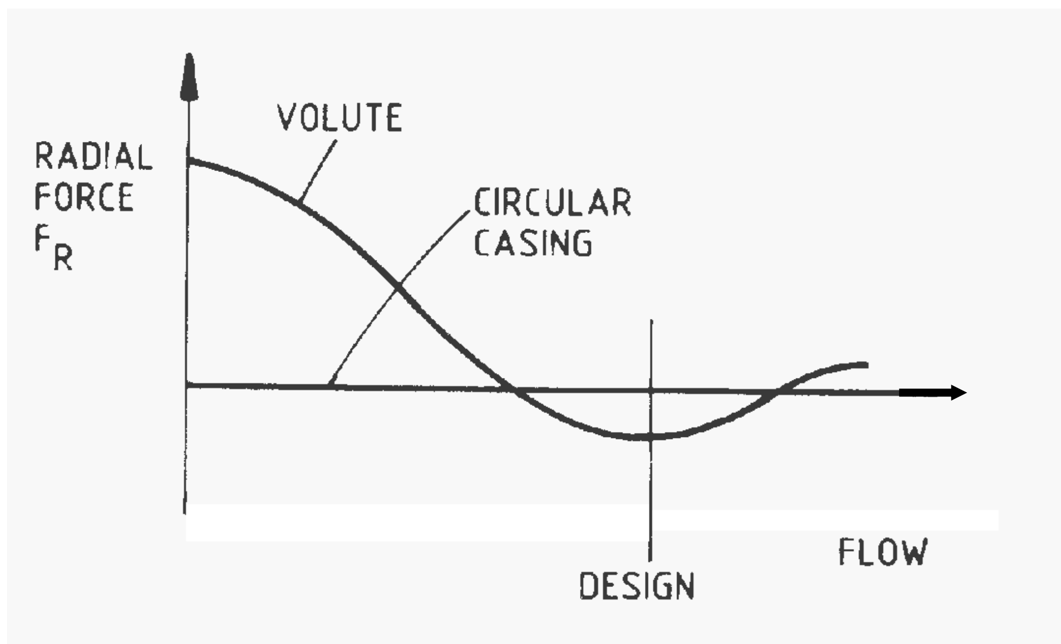
At the design point the pressure in the volute is practically uniform. The maximum pressure variation occurs in the region below about 50% flow when the radial thrust on the shaft takes high values.

The pressure in the pump volute varies with the location and the flow rate.

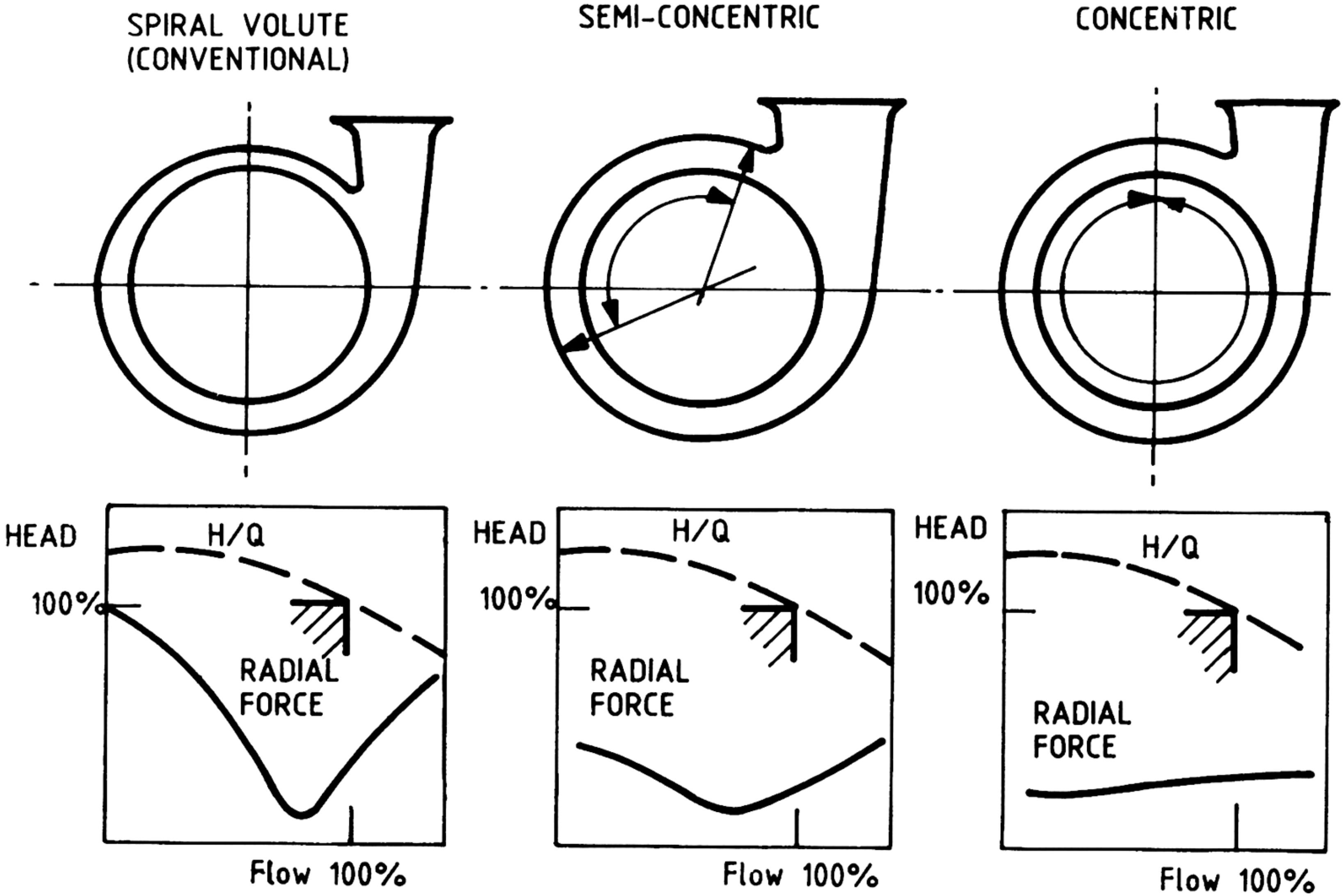


# Radial Thrust Variation with Pump Flow Rate for a Volute Pump

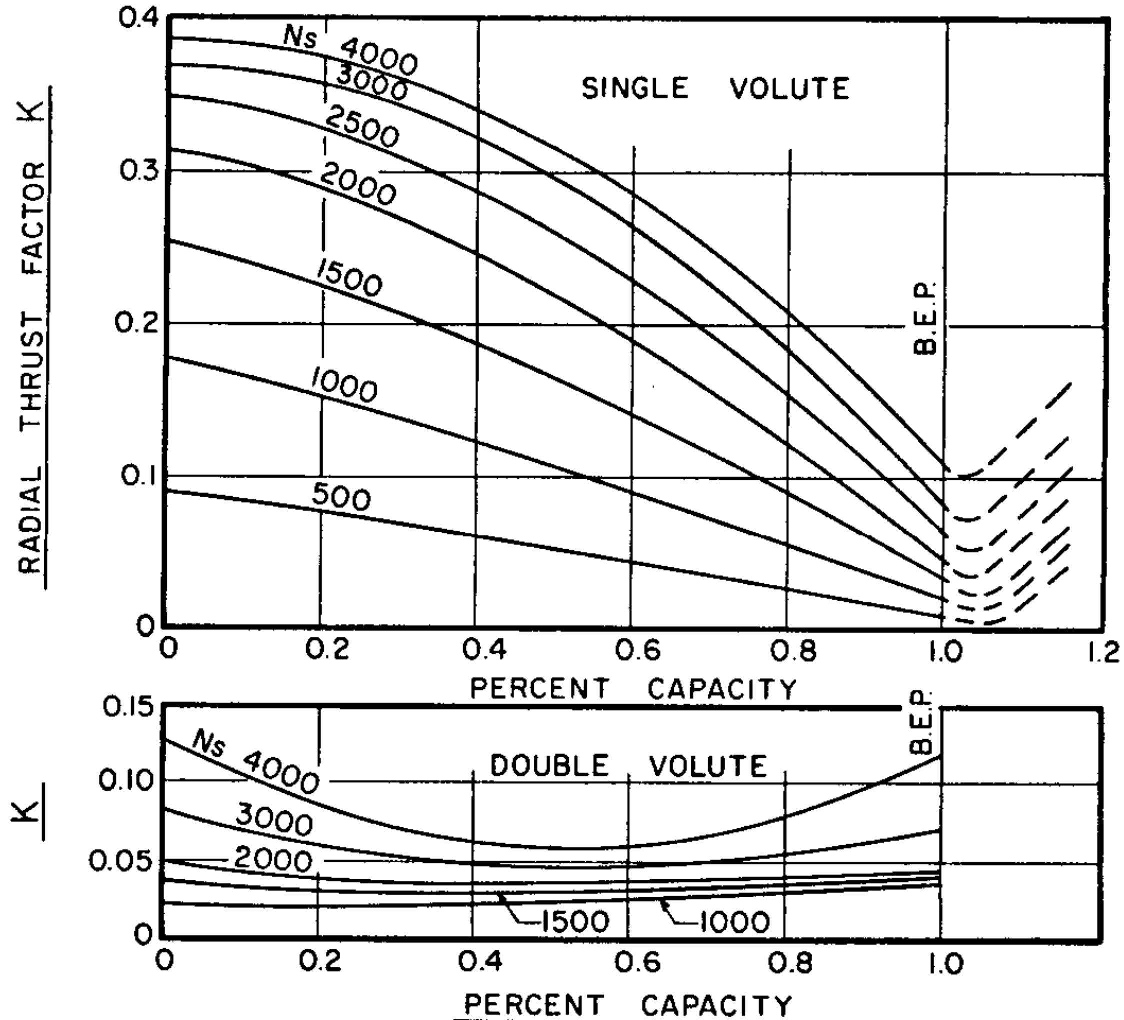


# Centrifugal Pump Radial Thrust Change with Volute Shape

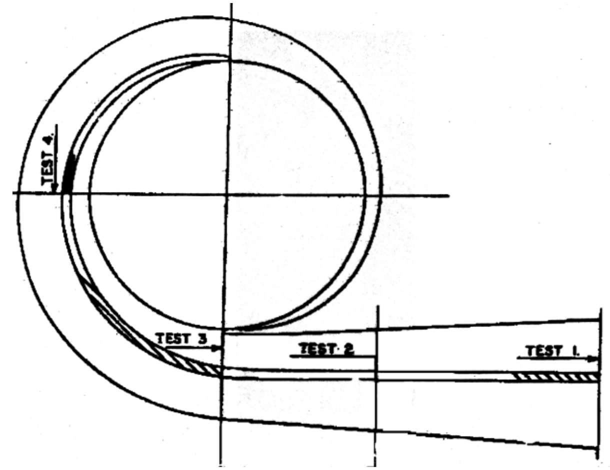


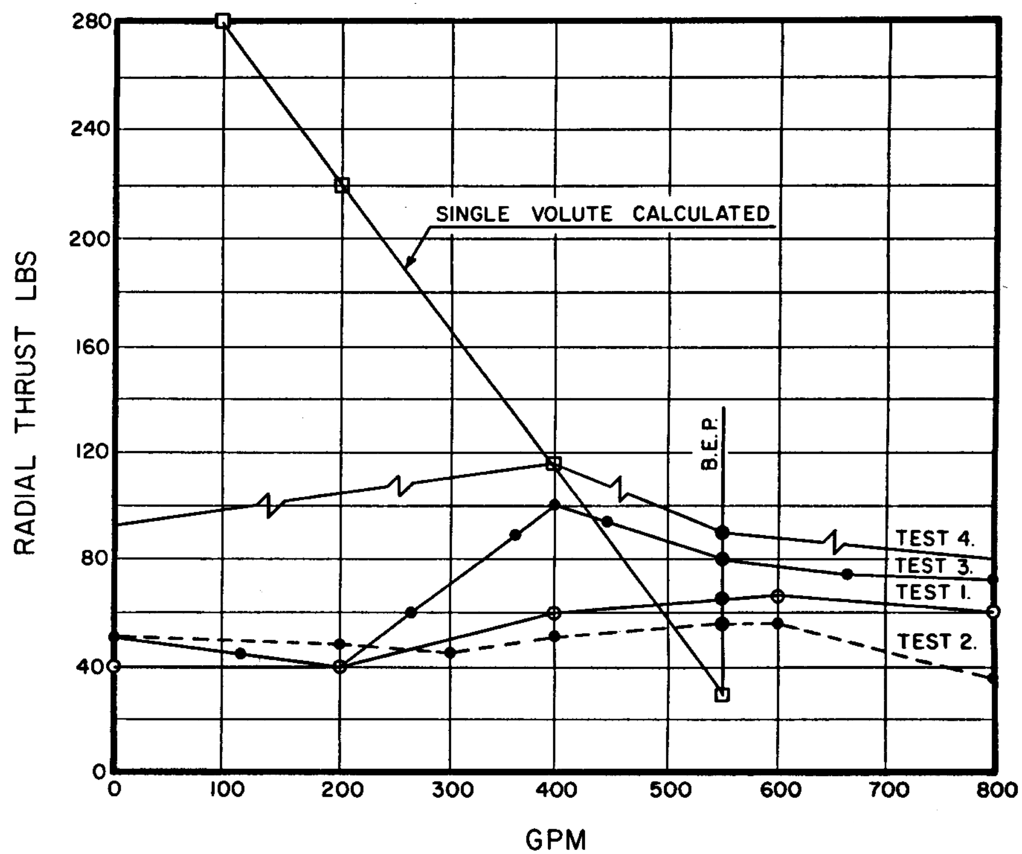
# Radial Thrust Factors for Single and Double Volute Configurations




**NOTE :- for circular casings of pumps with Specific Speed up to 750 K can be taken as K=0.05**

# Effect on Radial Thrust of the Volute Rib Length



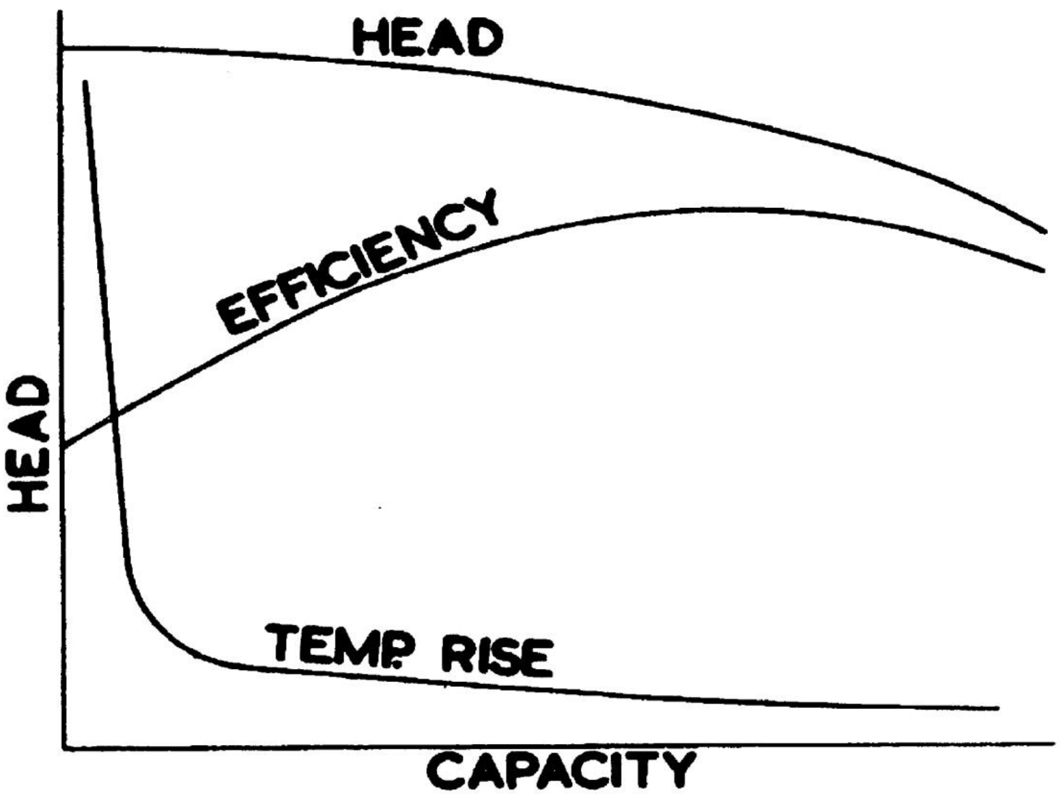
# Temperature Rise at Low Pump Flow

Assuming that the mechanical and radiation losses are small then the temperature rise is due to power input into the pump minus the liquid power i.e. power transmitted into the pumped liquid :-

$$\Delta T = (1-\eta)*H/(\eta*S_{sp.h})$$

Where H = total pump head
$\eta$ = pump efficiency at actual flow rate
$\Delta T$= temperature rise
$S_{sp.h}$= Specific heat

To prevent a high temperature rise at low flow rates a pump valved by pass can be installed.



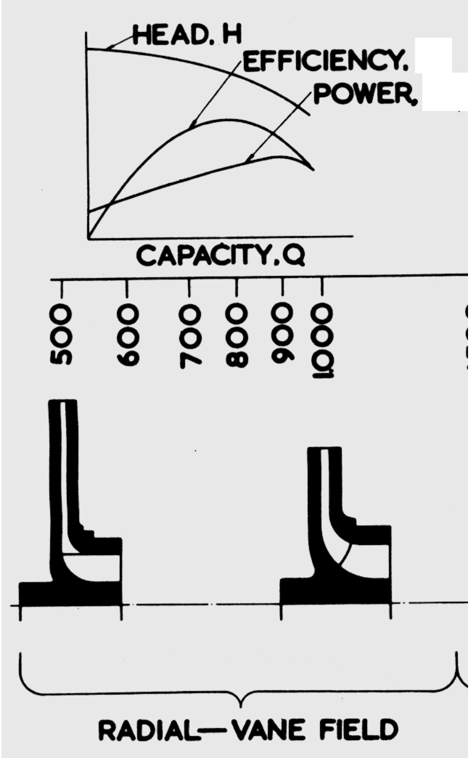
# Operation Away from the Best Efficiency Point

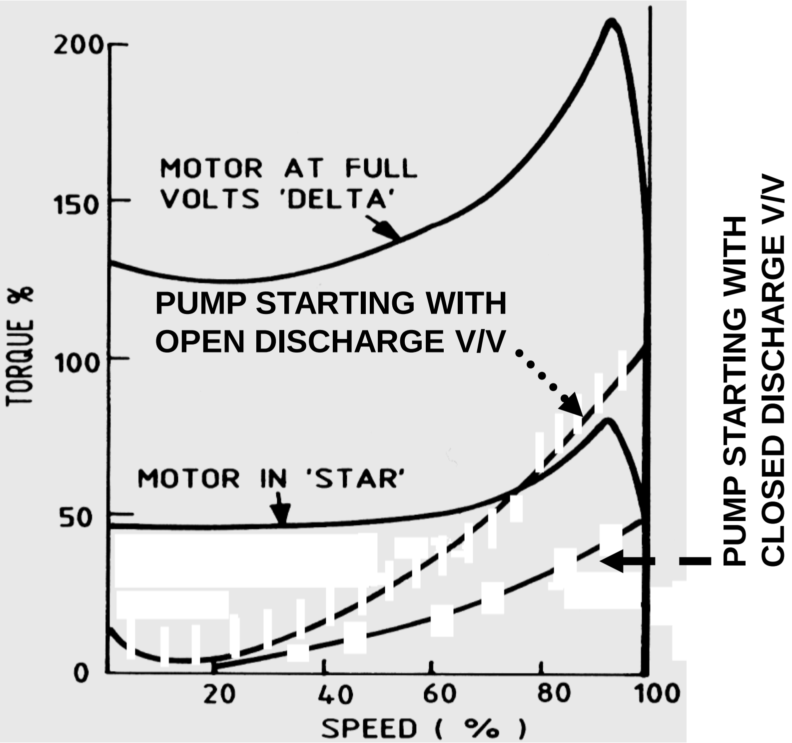
- As the flow rate increases from the Best Efficiency Point turbulence increases, suction losses too and so NPSH(R). Normal cavitation tendency increases until it sets in heavily at the break off point.
- Away from the Best Efficiency Point there is an increase in the radial thrust, especially with lower flow rates, below 50% BEP capacity.
- As flow rate decreases turbulence increases and re-circulation cavitation as well as normal cavitation become prevalent.
- Further decrease to very small flow rates causes the temperature in the impeller and therefore the casing to rise rapidly.

# Some Other Operational Aspects (i)

## Radial Pumps

The head capacity curve shows that the pressure developed at no flow conditions is not far higher than the design value. On the other hand the power capacity curve shows that the power requirement at no flow conditions has its minimum value. Therefore radial pumps firstly have to have the suction valve open and after priming to ensure that the discharge valve is closed. Therefore the motor should be energized after opening the suction valve, ensuring that the pump is primed and that the discharge valve is closed.

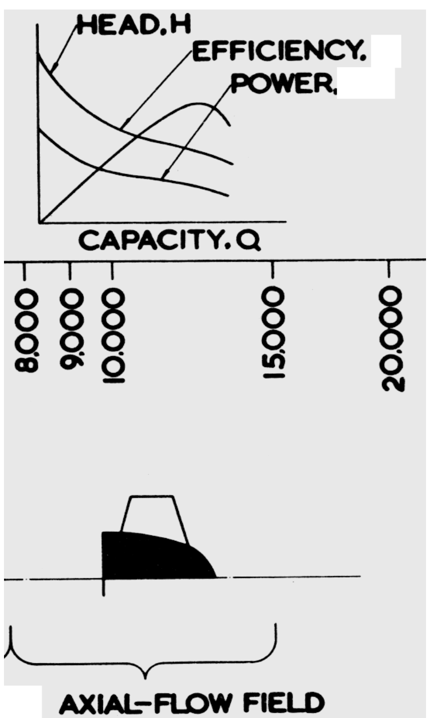


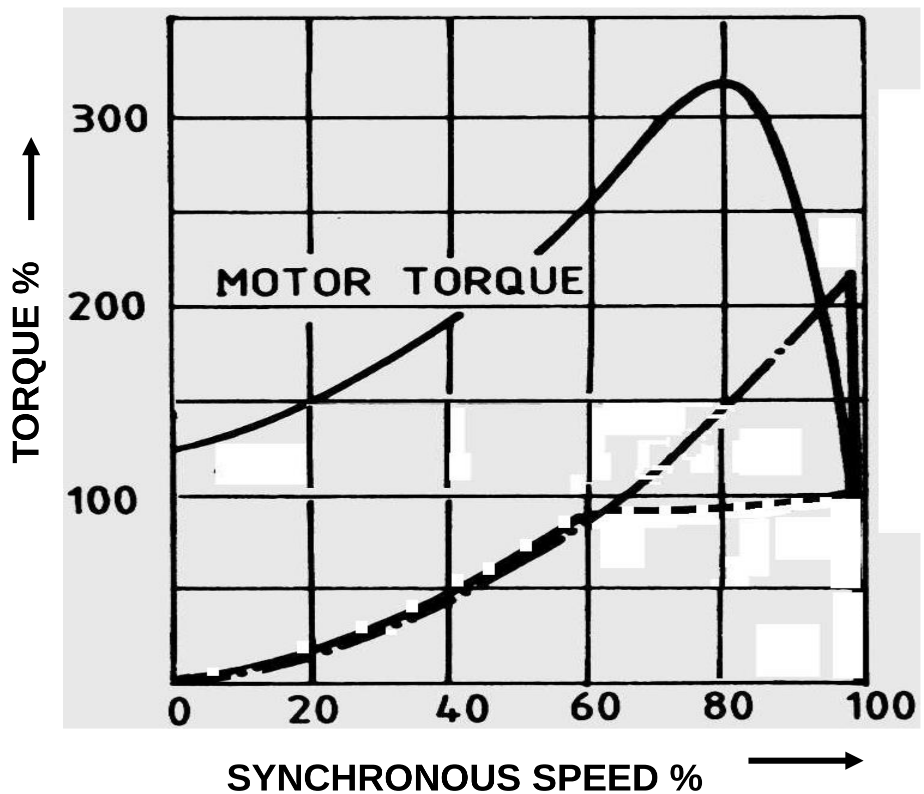



# Some Other Operational Aspects (ii)

## Axial Pumps

The head capacity and power capacity curves show that it is possible for the pressure developed by an axial pump at very low and no flow conditions to exceed the system design pressure and the power requirements to exceed the motor power rating. Therefore axial pumps firstly have to have the suction valve open and after priming to open the discharge valve. Therefore the motor should be energized after opening the discharge valve.



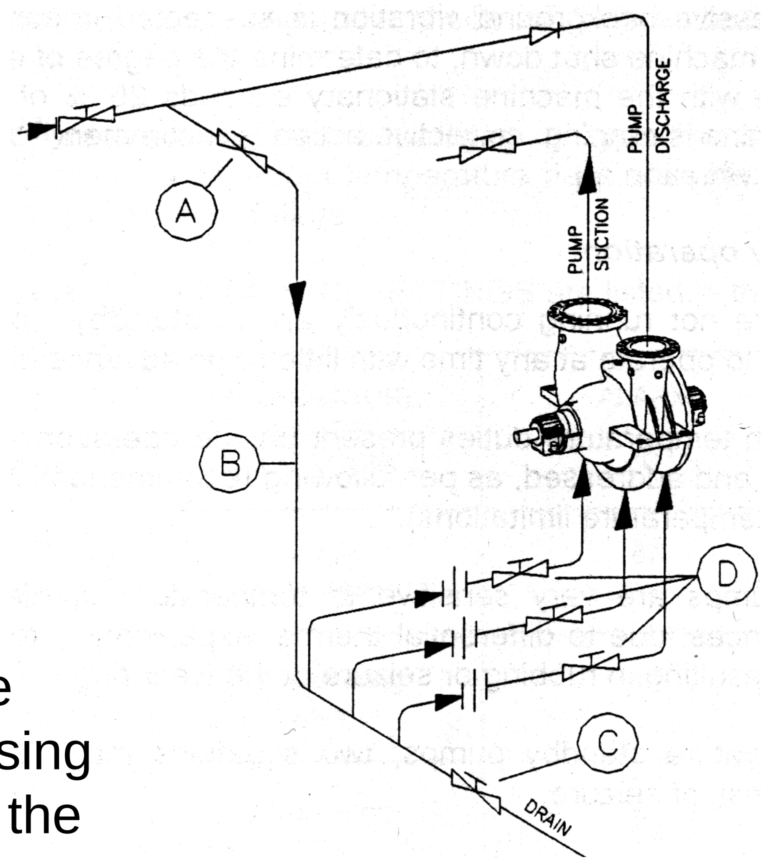



# Some Other Operational Aspects (iii)

Pumps in hot service must be warmed up gradually prior to start up.

Rapid temperature increases or thermal shocks can cause:-

- bending of the shaft
- rubbing and possibly binding of parts having narrow clearances such as wear rings, neck bushes etc.
- leakage at joints
- mechanical seal leakage
- vibrations

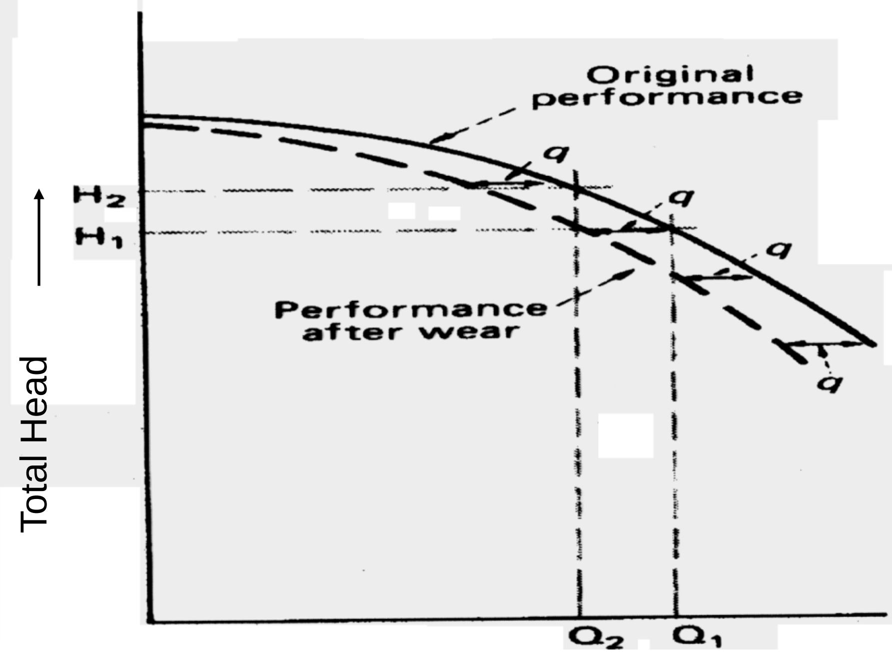


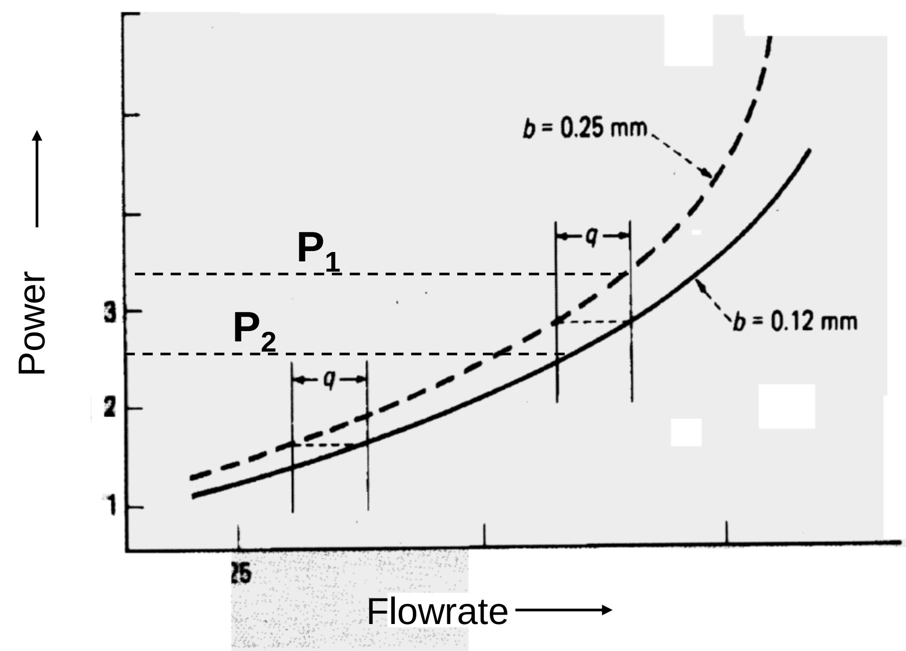

A warming up connection to the discharge header supplies hot liquid to the pump casing from where it flows to the suction through the open suction valve either through a discharge valve by-pass or as shown in the diagram.

# Pump Performance Deterioration

## Increase of Wear Ring Clearance

As the wear ring clearances increase with pump service time, the amount of liquid flowing internally from impeller discharge to the suction of the impeller increases due to the increased clearance area between the wear rings. Thus if the process flow required is $Q_2$ and the increased recirculation flow is q the impeller must pump $Q_2$+q i.e. $Q_1$. At $Q_1$ the head is lower $H_1$ and the power input is higher $P_1$. Similarly when the pump does not deliver any flow, internally the impeller pumps the additional recirculation flow and therefore absorbs more power and develops a lower head corresponding to the additional recirculation flow

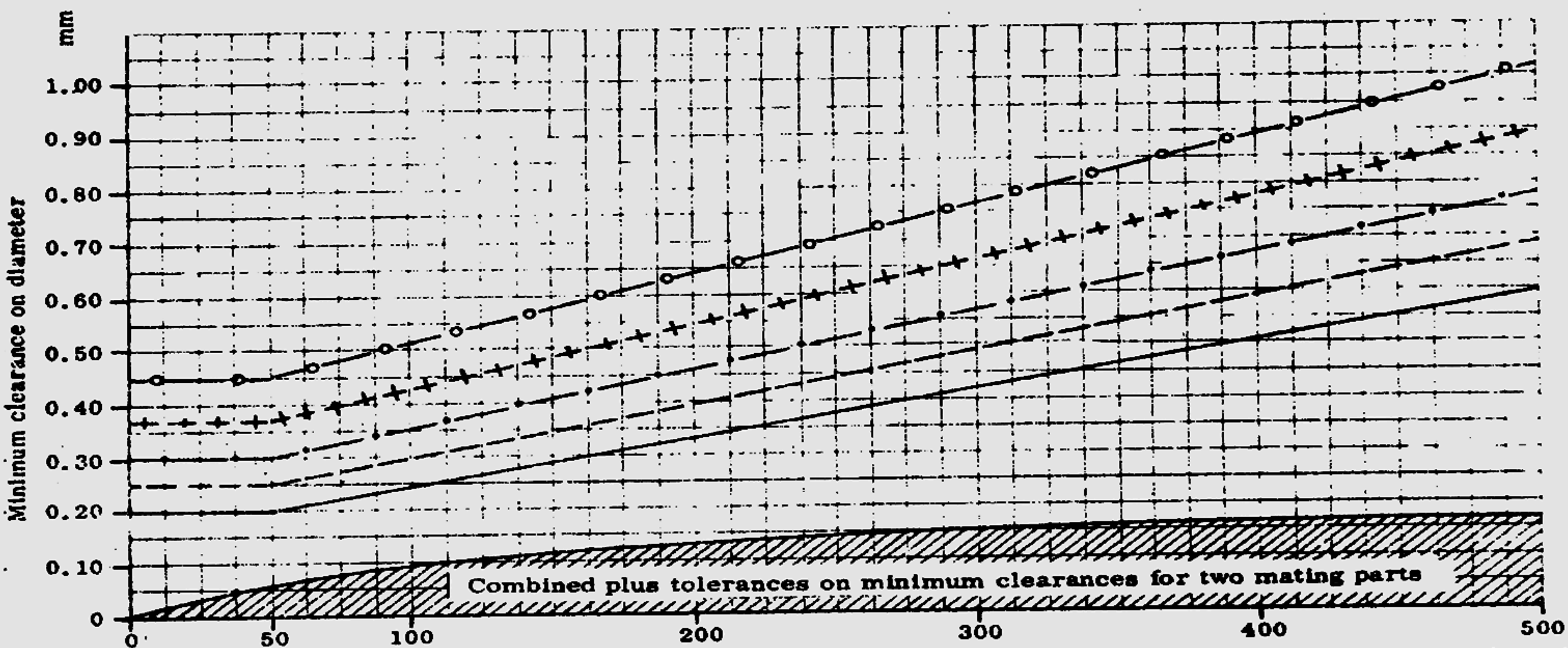


# Recommended Clearances for Centrifugal Process Pumps (I)

____ 100°C single/two stage
- - - - 100 to 250°C single/two stage
. . . . . 250°C & over single/two stage

_ . _ . _ 100°C multistage
+ + + + 100 to 250°C multistage
° ° ° ° ° 250°C & over multistage




Diameter of Impeller Rings or sleeves mm

# Internal Leakage Loss (i)

Internal leakage loss is the quantity of liquid flowing through the running clearances between the wetted rotating parts and the corresponding stationary parts.

**Internal leakage** can take place:-

- Between the casing and the impeller, with liquid flowing through clearances from the periphery to the eye of the impeller.
- Past vanes in open impeller pumps
- Between two adjacent stages in multistage pumps
- Through axial thrust balancing devices
- Through circulation connections of mechanical seals.

# Internal Leakage Loss (ii)

Internal leakage can be controlled by setting clearances at generally accepted values. In general leakage loss is greater in relation to pump capacity in small pumps than in larger pumps because clearances must not be reduced below the practical minimum value.

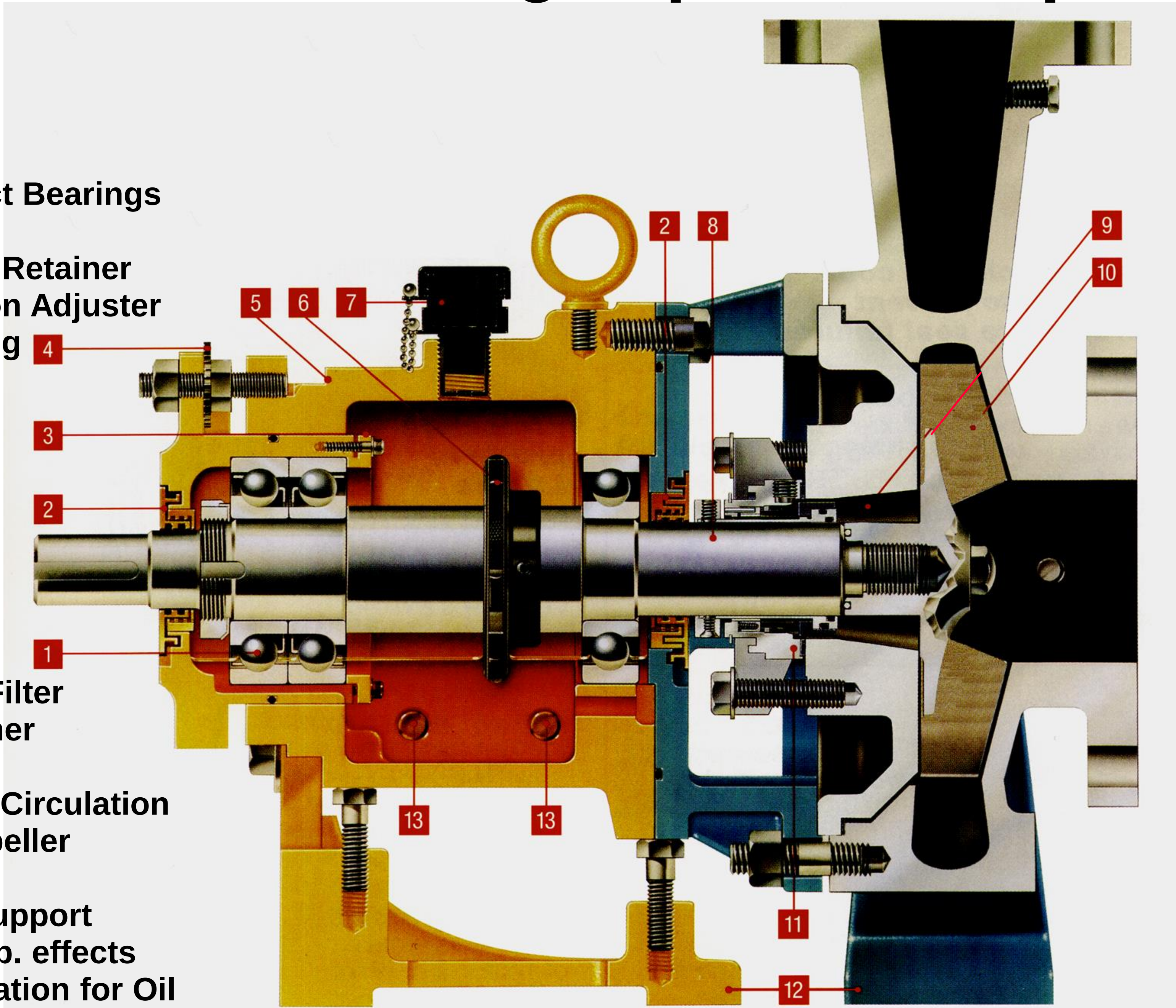
The internal leakage loss decreases rapidly with increasing specific speed since the differential pressure across the impeller becomes smaller.

The circulation connection of mechanical seals are controlled by orifices which are designed to allow the required amount to pass. Worn orifices must be replaced. Too high a recirculation quantity can cause erosion damage to seal components especially carbon faces, in addition to pump head and efficiency reduction.

Too high or too low a re-circulation quantity in the case of axial thrust balancing devices is detrimental on the effectiveness of the balancing device resulting in early thrust bearing or balancing device failure.

# Chesterton Overhang Impeller Pump

1. Angular Contact Bearings
2. Labyrinth Seal
3. Bolted Bearing Retainer
4. Impeller position Adjuster
5. Bearing Housing




6. Oil Thrower & Filter
7. Oil Filler Breather
8. Heavy Shaft
9. Seal Cooling & Circulation
10. Semi-open Impeller
11. Mech. Seal
12. Center Line Support
    to reduce temp. effects
13. Coolant Circulation for Oil

# External Leakage Loss (i)

External leakage loss can take place through :-

- Stuffing Boxes. A small amount of leakage through the pump stuffing box is necessary to keep the stuffing box lubricated and cool. When the stuffing box is malfunctioning larger amounts of leakage emerge. Valve stuffing boxes allow leakage either as emissions or as liquid leakage when glands are not properly adjusted or when the packing allows leakage.
- Mechanical Seals allow a very small amount of leakage - emission - of vapour to pass through the seal faces for lubrication purposes, provided the seal is in good condition.
- Bleed off e.g. stuffing box pressure reduction, pump by pass etc.
- Defective stationary seals such a pump or valve body joints.
- Defective casing castings.

# External Leakage Loss (ii)

External leakage loss may have safety, health and environmental hazards, especially if the pumped liquid is flammable, toxic, odorous or polluting.

External pump leakage can be controlled by the correct choice of sealing materials or devices such as gaskets, pump and valve packing, mechanical seals and of valve sealing devices other than normal packing such as Expandable Valve Stem Packing (EVSP) or bellows valve stem sealing.

Good inspection and workmanship in installing these sealing devices is essential. It is also extremely important to avoid operating practices which tend to damage sealing devices such as hydraulic or thermal shocks, mechanical vibration, cavitation (recirculating or normal), running the pump for prolonged periods in the low flowrate region.

Defects in casings can be avoided by careful selection of materials, inspection and testing. It is good practice to have pump casings and valve bodies carefully inspected and pressure tested according to the applicable code.

# Centrifugal Pump Malfunction

Pump malfunction can be attributed to various causes but these can be grouped under three main categories.

1. Hydraulic

2. Mechanical

3. External. (resulting from faulty piping,
external influences such as imposed vibration,
hydraulic or thermal shocks,
faulty testing and operating procedures,
incorrect maintenance practices,
driver problems,
expectations based on incorrect or incomplete information etc.

# Hydraulic Malfunction

Many of the hydraulic faults can be diagnosed by carefully considering the actual Total Head/Capacity characteristic and comparing it to the manufacturer's original Total Head/Capacity curve.

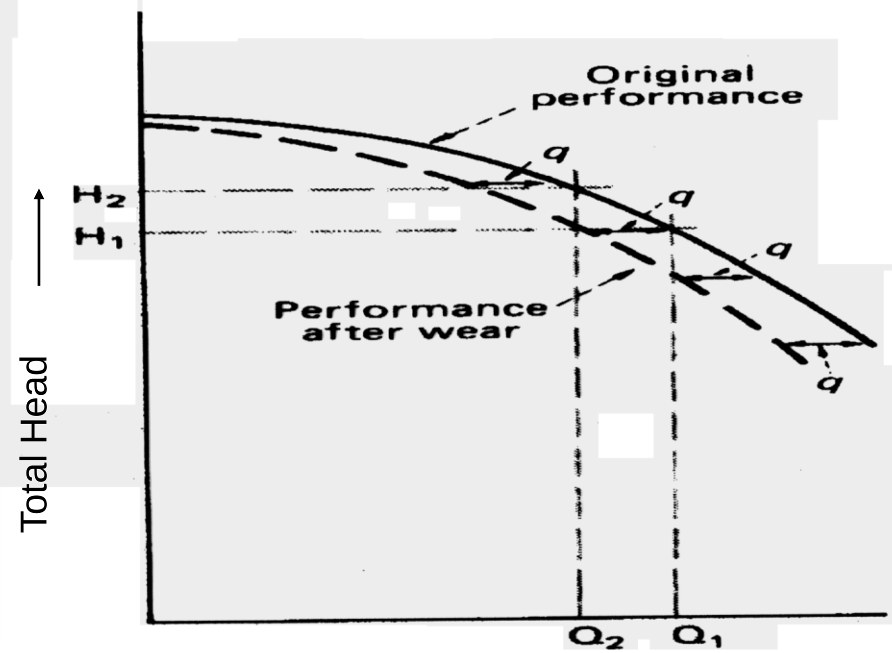
The method is fairly simple and very effective. It can contribute significantly in improving the reliability of the equipment as well as help in more effective and quick repairs.

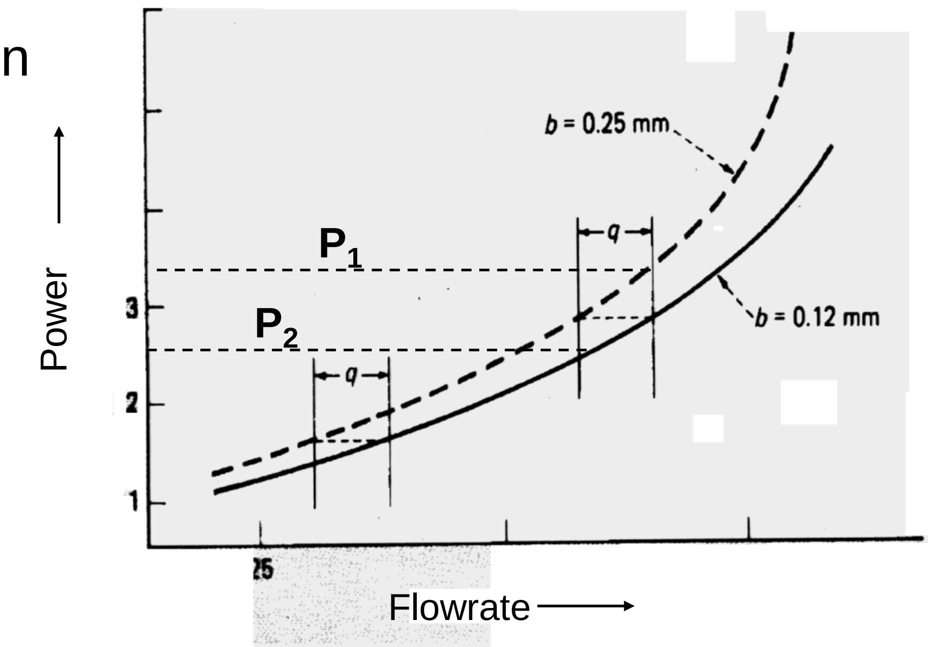
Prior to taking a pump out of service due to a suspected malfunction a simple test must be carried out to establish the value of the total head developed and the power absorbed at the measured flowrate at the pump outlet. It may be of great help if a no pump flowrate total head developed and power absorbed values can also be obtained. These are compared to the manufacturer's original equivalent figures and conclusions drawn, taking into consideration viscosity and density values of the pumped liquid.

# Pump Performance Deterioration
## Increase of Wear Ring Clearance

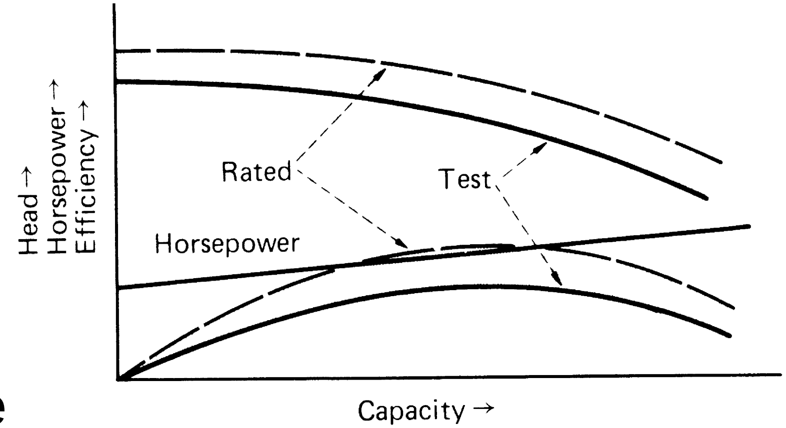
Lower head and higher power than the original performance test at the pump outlet flowrate as well as at zero flow out of the pump.

This is a clear indication that the internal re-circulation quantity has increased. The wear ring clearances must be checked and re-established.

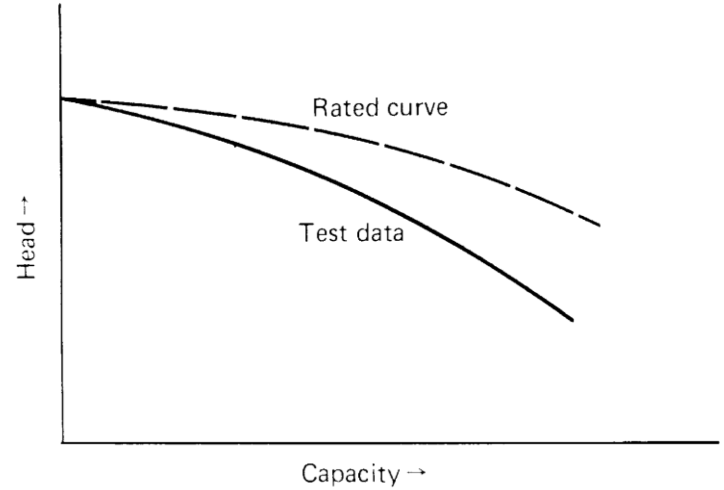


# Pump Performance Deterioration




**Roughness.** When surface roughness develops without impeller shape damage, the frictional losses internally increase but the power input remains substantially the same as due to the undamaged shape it still imparts the same amount of energy to the liquid part of which is absorbed in overcoming friction instead of increasing the head developed. Therefore the head is lower at substantially the same (possibly somewhat higher) power input

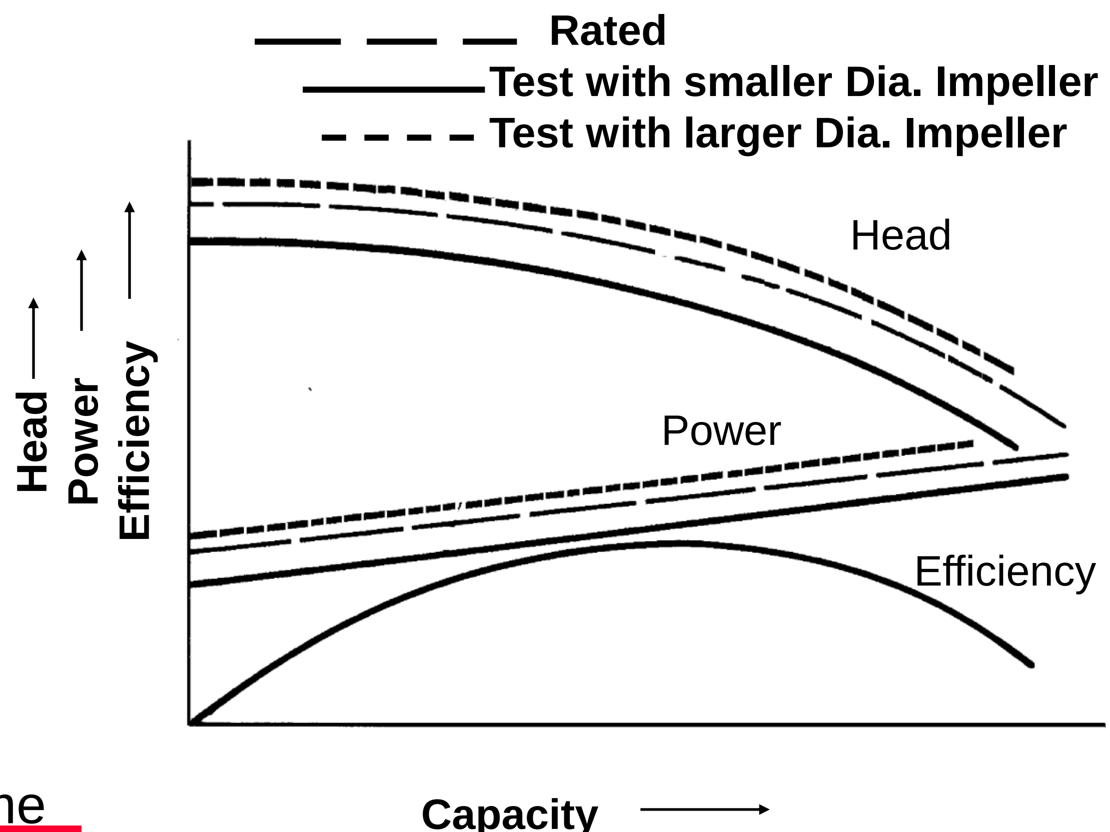



**Obstruction in the Discharge Passages.** The power input is unaltered as the obstruction in the discharge passages does not affect the action of the impeller. However the obstruction causes additional head loss as it restricts the flow and the pump head developed is lower except at no pump flow as at this point there is no flow in the discharge passages.

# Alteration of Pump Performance

If after a pump has been for overhaul and on the first test it is observed that the total head is lower (higher) than the pump rating test as well as the power input is lower (higher) than the rating tests at any capacity and at no flow conditions it can be concluded that the newly installed impeller is of a smaller (larger) diameter than the impeller used for the rating tests (if the pump is tested at the same speed.

Similarly if thespeed is lower or higher and the dia. Is the same

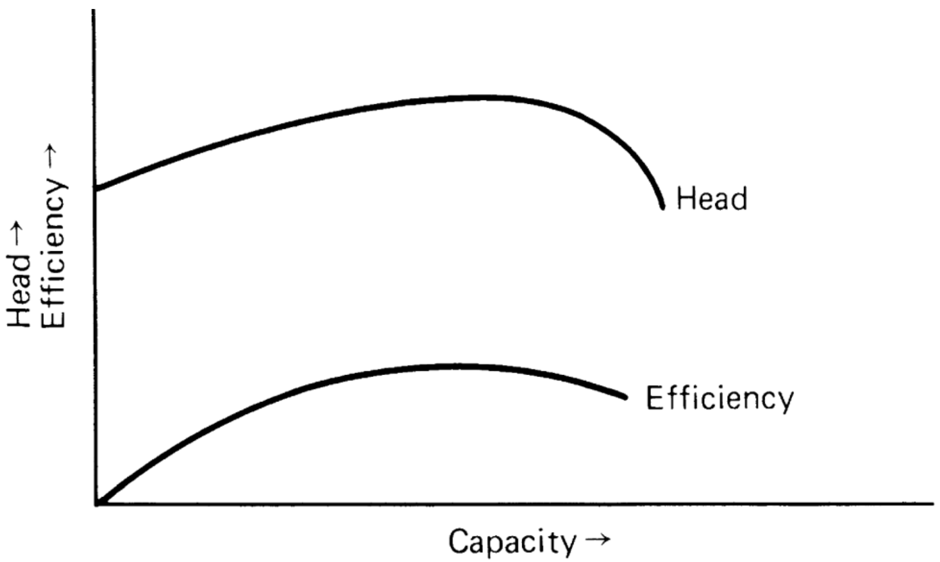


# Alteration of Pump Performance

It is possible that in the case of a motor driven pump after electrical disconnection and re-connection of the motor the power supply phases are reversed. This causes reverse rotation. When the pump is started it will develop a very low head at a very low efficiency.

Reversing the electrical supply phases will restore the pump performance to the levels prior to the electrical disconnection.

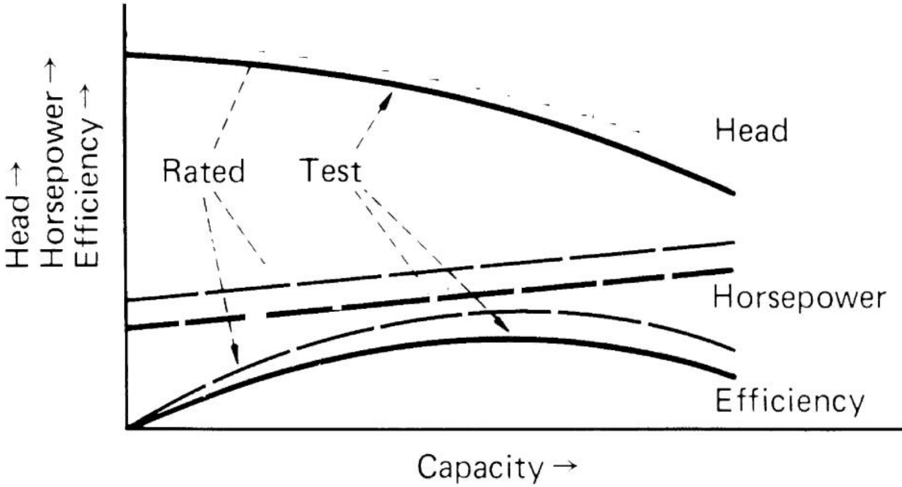
It is always advisable to check the pump rotation after power supply cable re-connection.



# Alteration of Pump Performance

Extremely low wear ring clearances and neck bushes, very tight gland packing, damaged bearings, rubbing components due to undue casing stresses caused by maladjusted piping loads or other reasons causing mechanical tightness and internal rubbing increase the pump driver power demand. These malfunctions may possibly be accompanied by characteristic noises, vibrations and overheating of the respective parts. If the additional heat generated by rubbing is released to the pumped liquid it may not be noticed.

The head developed will be substantially following the original test curve values but the power input will be higher and so the pump efficiency will be lower.



# Malfunction due to External Influences

**Air or Gas in the Pump.**

Air or gas entering or present in the pump causes pump performance deterioration or even complete stoppage of the pumping action.

The amount of air or gas that can be handled by a centrifugal pump without substantial reduction in performance is about 0.5% by volume at suction conditions. If the amount of air or gas increases the performance of the pump suffers - reduction of head and capacity - until the gaseous content reaches about 6% by volume at suction conditions when the pump cannot pump any more. When the gas content is low the pumped liquid sweeps the gas through the pump to discharge, but when the gas volume increases the liquid volume decreases and there is a point when the liquid can no longer sweep the air or gas to discharge and the pump suffers vapour lock.

# Malfunction due to External Influences

**Air or Gas in the Suction Piping**

Air or gas pockets can form in the suction piping if pipe leveling is not properly attended to. Pumps can function with small stationary pockets in the suction piping. Problems arise when the gas moves and enters the pump. However even stationary pockets disturb flow patterns and flow velocities. It is advisable to eliminate pockets.

Horizontal eccentric reducers with the straight part at the top must be used instead of concentric ones. In addition the suction line must either be laid horizontal or with a steady rise towards the pump to eliminate gas pockets. Vertical loops in suction lines should be avoided or if this is not possible suitable vents must be installed.

# Malfunction due to External Influences

**Piping System**

If the associated piping system is not properly supported either due to incorrect location of supports, errors in construction or due to incorrect adjustment of supports (initial or gradual deterioration) unduly heavy piping loads can be transmitted to the pump casing. Distortion of the casing can follow or even in the worst cases casing failure. Casing distortions can cause rubbing or binding of parts with narrow clearances.

Piping must not transmit vibrations as these can cause mechanical seal and bearing problems whether the pump is idle or running.

In the case of double suction impeller pumps a bend in the suction line just before the pump may cause uneven distribution of flow to the impeller, causing axial thrust unbalance and lower pump head and efficiency.

# Increasing Pump Performance

The performance of a particular pump can be increased in several ways.
It is assumed that the pump is in good condition mechanically and structurally and that it is running in the as new condition.

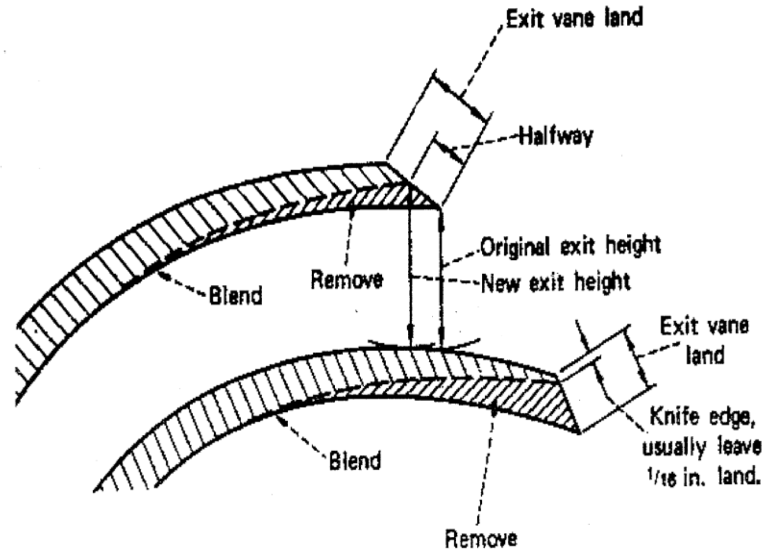
**1. Increase of Impeller Diameter.** A range of impeller diameters can be installed in a particular casing. It is usual practice when a new pump is chosen to avoid choosing one with the largest diameter impeller that can be installed in the particular casing. On the other hand choosing too small an impeller may result in a noticeably lower efficiency. The affinity laws can be used to predict the performance with the impeller with the larger diameter. It may be necessary that an upgraded driving system may be required with the larger impeller.

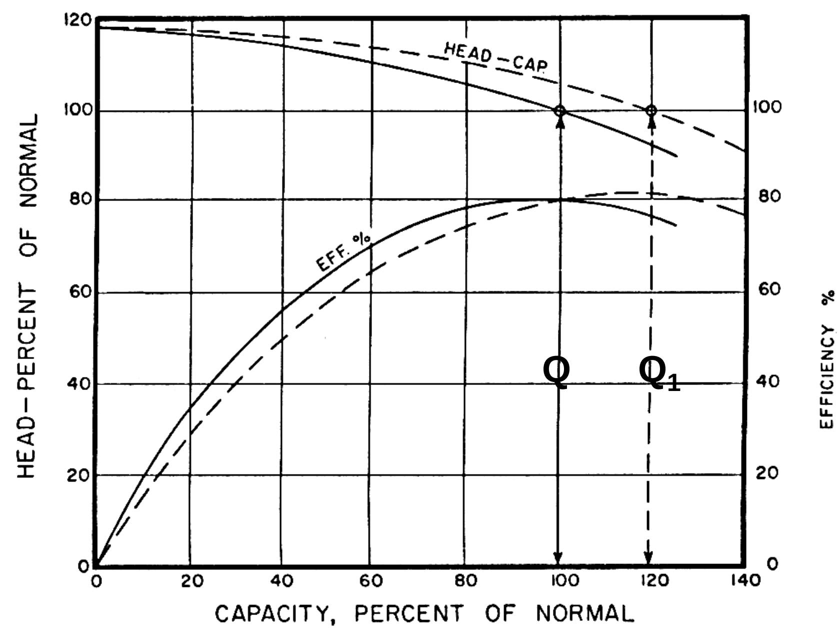
**2. Increase in Impeller Speed.** It may be possible to increase the pump driver speed (if it is turbine or engine driven - synchronous motors are running at a substantially constant speed). The affinity laws can be used to predict the performance with the impeller with the larger diameter. It may be necessary that an upgraded driving system may be required with the larger impeller.

# Increasing Pump Performance

## 3. Vane Underfiling

It is possible to increase pump performance by a small amount by underfiling the exit part of the vanes with the aim of increasing the outlet area, as shown. There will be no increase at shut off. The efficiency suffers over a wide  operating range but will increase over the high flowrate range. The extend of underfiling is either “halfway” when about half of the exit vane thickness is removed as shown or “knife edge” when the land of the vane is reduced to about 1.5mm as shown. This method is only recommended under high priority or emergency requirements. The impeller should be dynamically balanced after the filing operation.



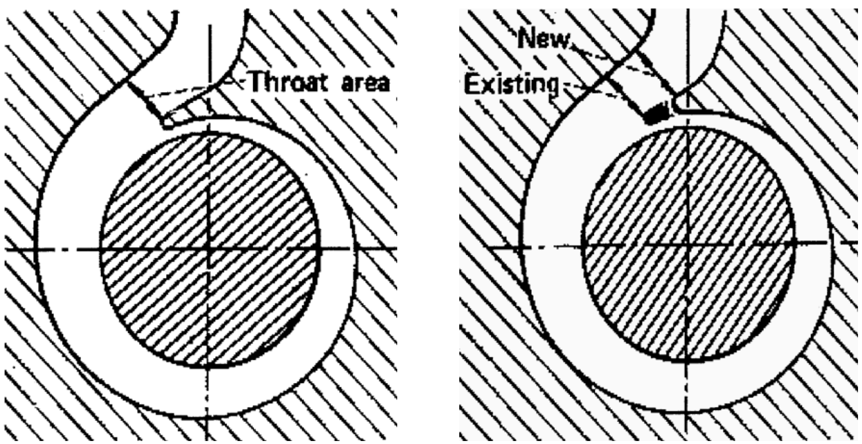

The new capacity $Q_1 = Q_{Original}$ x(new exit height/original exit height

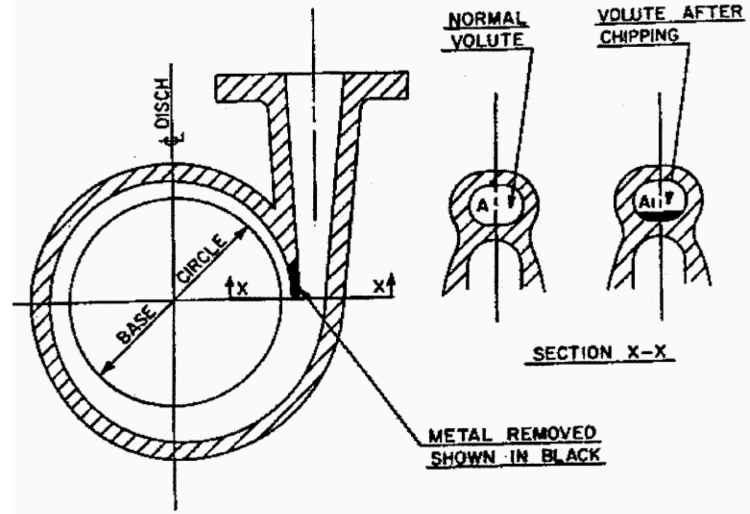
# Increasing Pump Performance

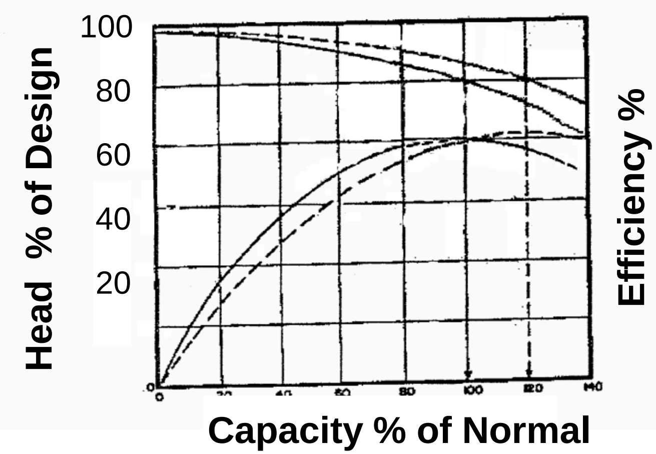
## 4. Volute Throat Chipping

$$Q_1 = Q^*(A_1/A)^{1/2}$$

$Q_1$ = Capacity after Throat Chipping
$Q$ = Normal Capacity
$A_1$ = Throat Area after Chipping
$A$ = Normal Throat Area.



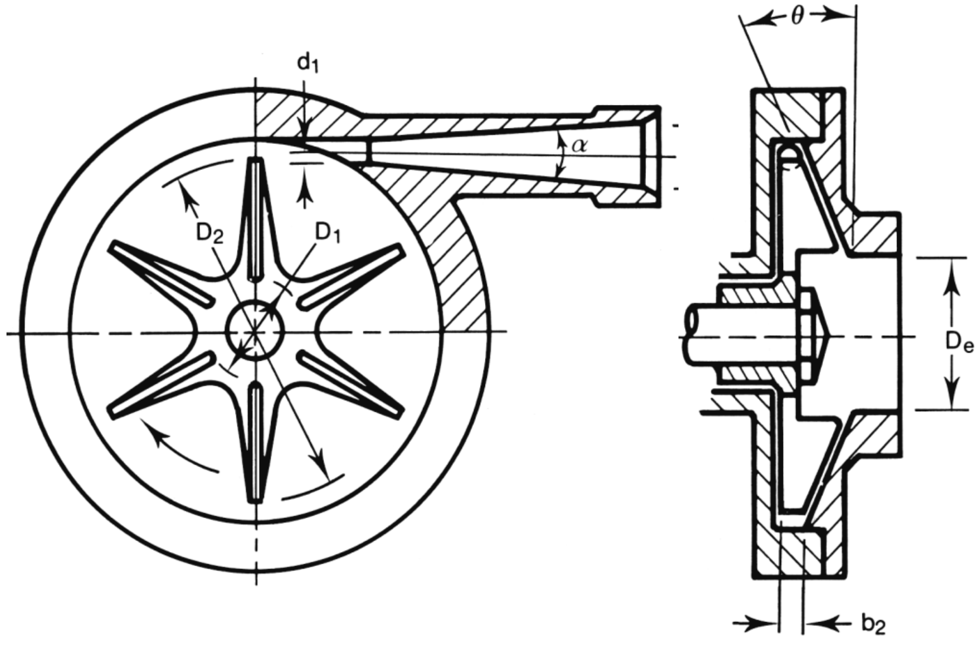




Low specific speed centrifugals are very sensitive to the area of the casing throat. A slight change in this area has a pronounced effect on performance. Overfiling could easily disturb the flow pattern resulting in performance loss.

# Partial Emission Pumps (i)

The typical partial emission pump has a simple open impeller with straight radial vanes, rotating concentrically in a circular casing. A simple conical diffuser tangential to the casing forms the outlet whilst the inlet is opposite to eye of the impeller. The conical outlet diffuser is one with a cone angle of 8º to 10º for maximum efficiency converting velocity to pressure head.
The impeller tip height is equal or somewhat greater than the outlet diffuser throat. The height of the vanes at inlet must provide an area at least equal to the area of the suction passage.



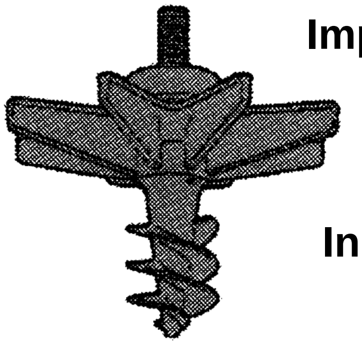
# Partial Emission Pumps (ii)

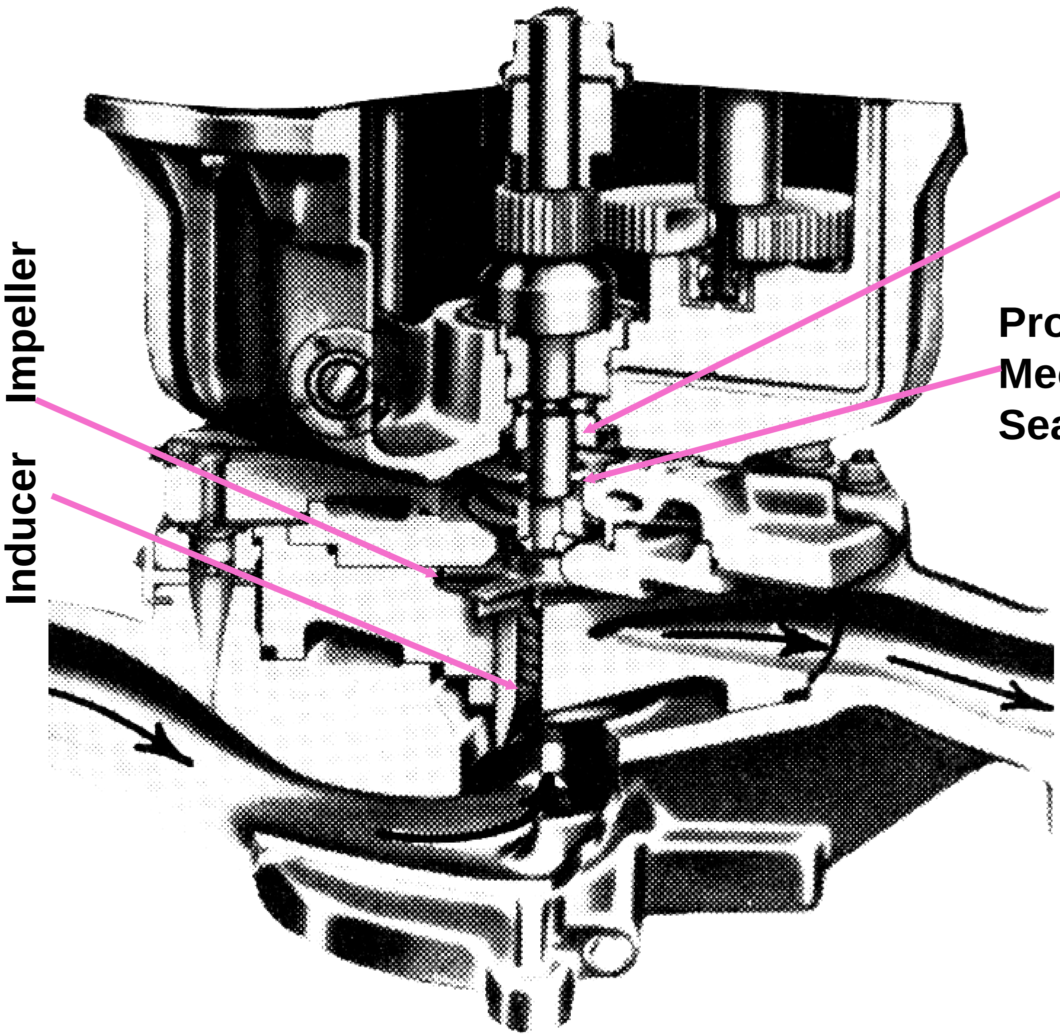
The impeller creates a forced vortex with high circumferential liquid velocities An element of liquid makes a number of full circuit turns within the forced vortex moving successively to a larger diameter and to a higher velocity from impeller eye to the outlet. Only part of the liquid moving in the vortex leaves to the discharge diffuser. For this reason this pump is called a partial emission pump in contrast to the usual centrifugal pump in which the amount of liquid that enters the eye moves once through the impeller and then to the volute casing and to the outlet. The latter is for this reason sometimes referred to as a full emission pump.

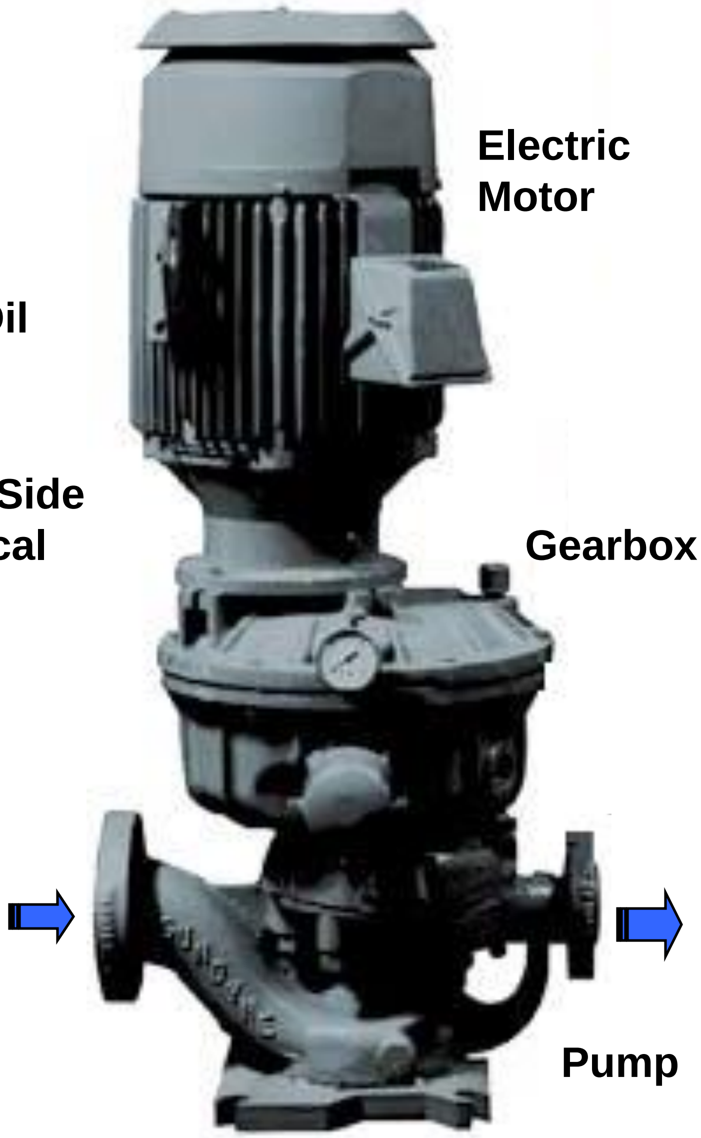
The result is a simple lightweight pump producing high heads at low flow rates.

The high total head developed in the partial emission impeller makes it suitable to replace several full emission pump stages. The higher the speed the higher the total head developed for a given capacity. Commercial partial emission pumps running up to about 30000rpm have been used successfully to replace multistage centrifugal full emission pumps and even positive displacement ones.

# Partial Emission Pump

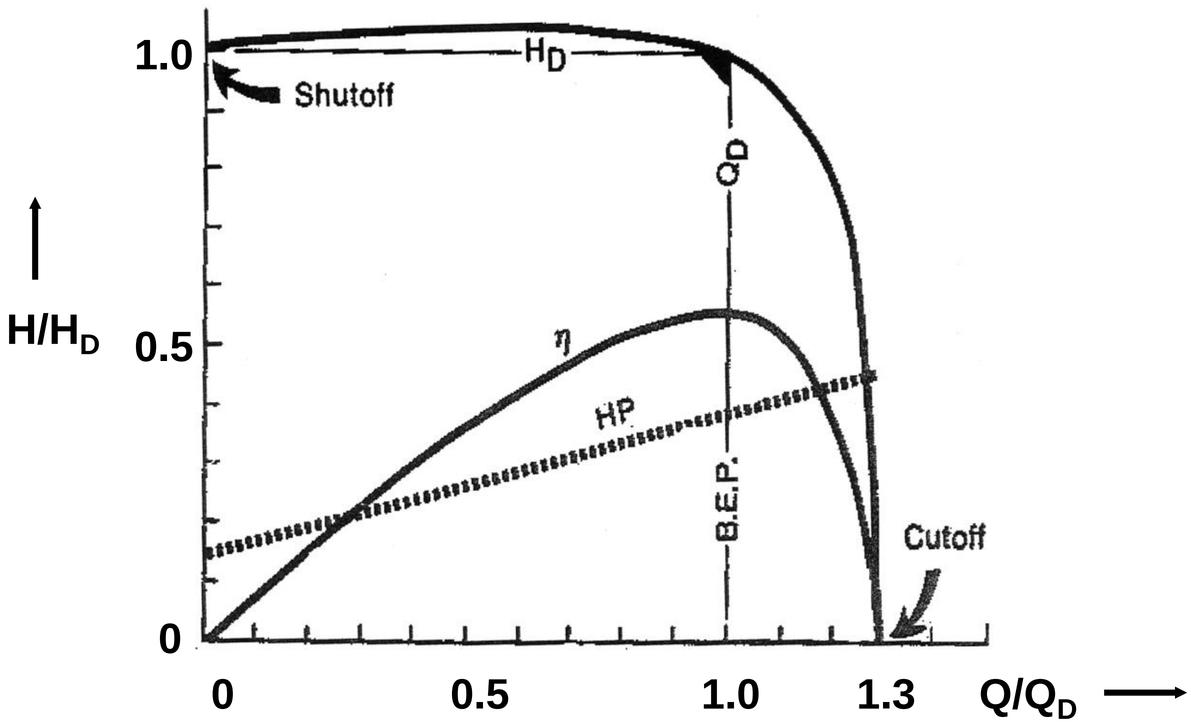


# Partial Emission Pumps (iii)

Head capacity performance curves are similar to those of radial centrifugal but they are flatter and on many occasions the head at shut off is roughly equal to the head at BEP with a maximum in between. This implies that it is possible for the pump at two different values of fowrate for a given head. This is termed an unstable characteristic which can give rise to system pulsations and hunting. Stability can be achieved by installing a suitable orifice at pump discharge but at the expense of head and efficiency.



# Partial Emission Pumps (iv)

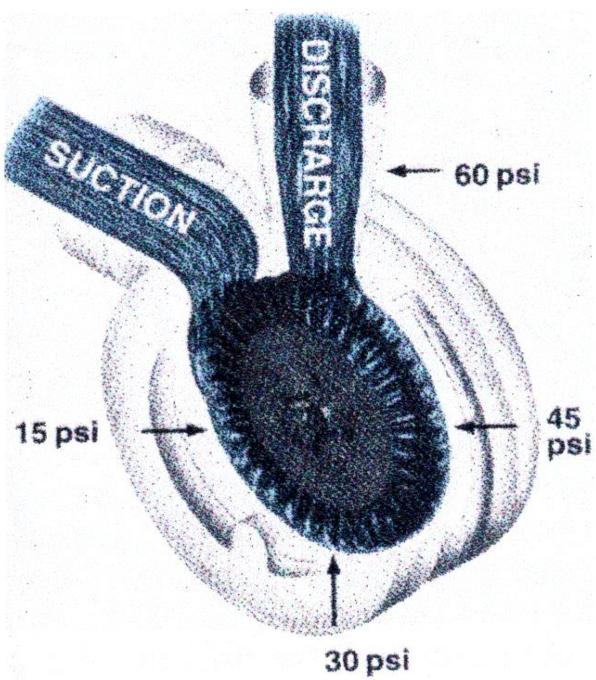
The high impeller speed imposes and increased requirement for NPSH. An inducer can be used for the required improvement but for a pump efficiency reduction away from BEP.

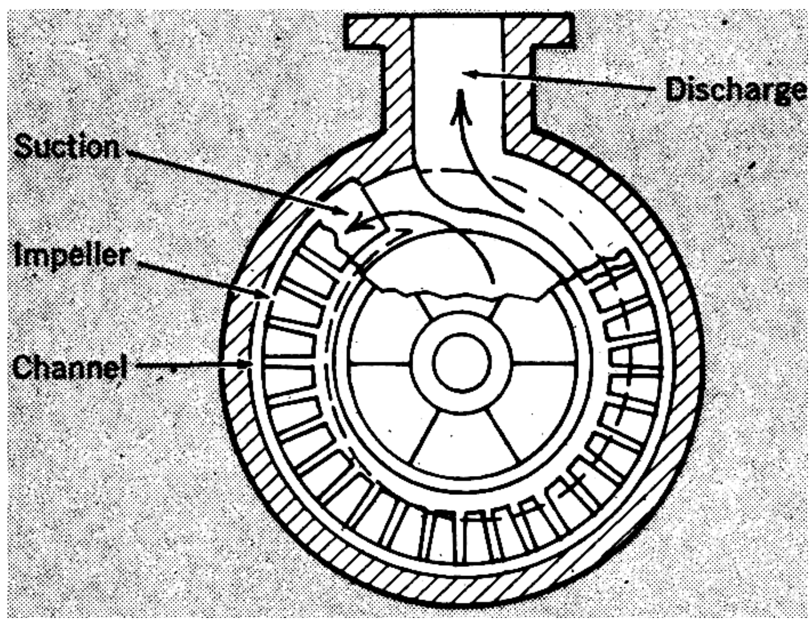
The partial emission pump does not require close operating clearances to achieve good performance. Due to the high velocity vortex motion, the leakage from the impeller outlet back to the impeller eye is not significant, especially in the case of the higher speed pumps. Clearances of the order of 0.8mm to 1.3mm are common.

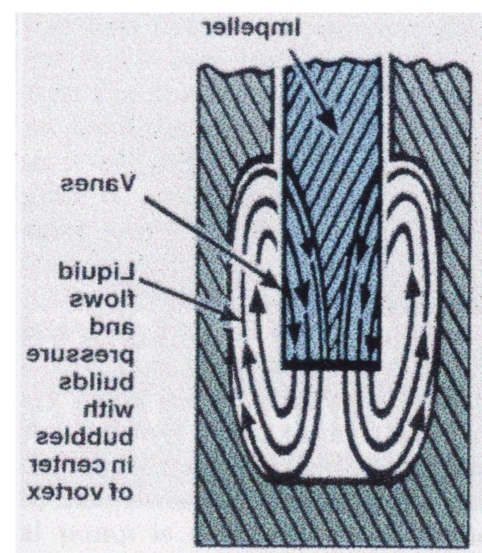
The partial emission pump mainly due to its circular casing has a very low radial thrust at shutoff conditions. The radial thrust increases with increasing flowrate.

# Regenerative Pumps (i)

The impeller of a regenerative pump has radial cuts along the periphery that are opposite corresponding similar cuts in the pump casing or stator. The arrangement imparts to the liquid motion along successive forced vortices that correspond to each impeller cut and in planes perpendicular to the rotor or impeller disc. The liquid circulates around the impeller periphery almost over one complete revolution before it discharges. In reality the regenerative pump stage corresponds to a multistage pump with a large number of simple stages.



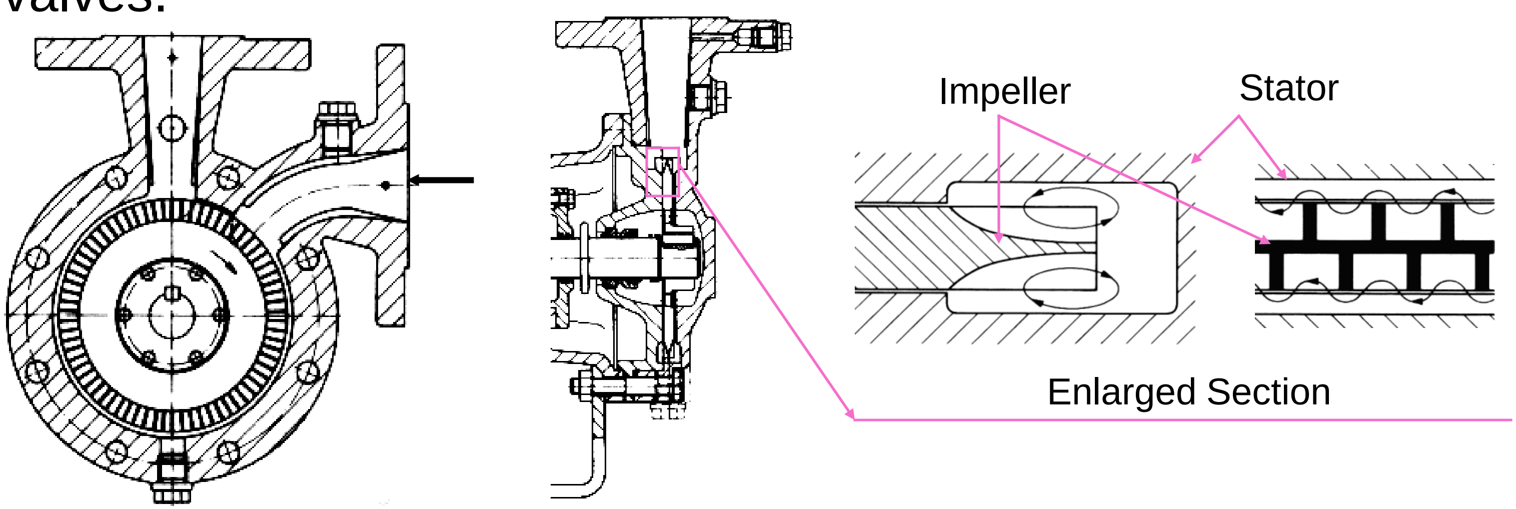



# Regenerative Pumps (ii)

The fluid flow path gives this type of pump several other names such as vortex, periphery, periflow or drag pump. Due to the high pressures developed across the pump impeller very high sensitivity to internal leakage exists. Thus the clearances between impeller and stator have to be very carefully adjusted to low values. These narrow clearances make the pump not suitable for liquids carrying solids even in small concentrations.

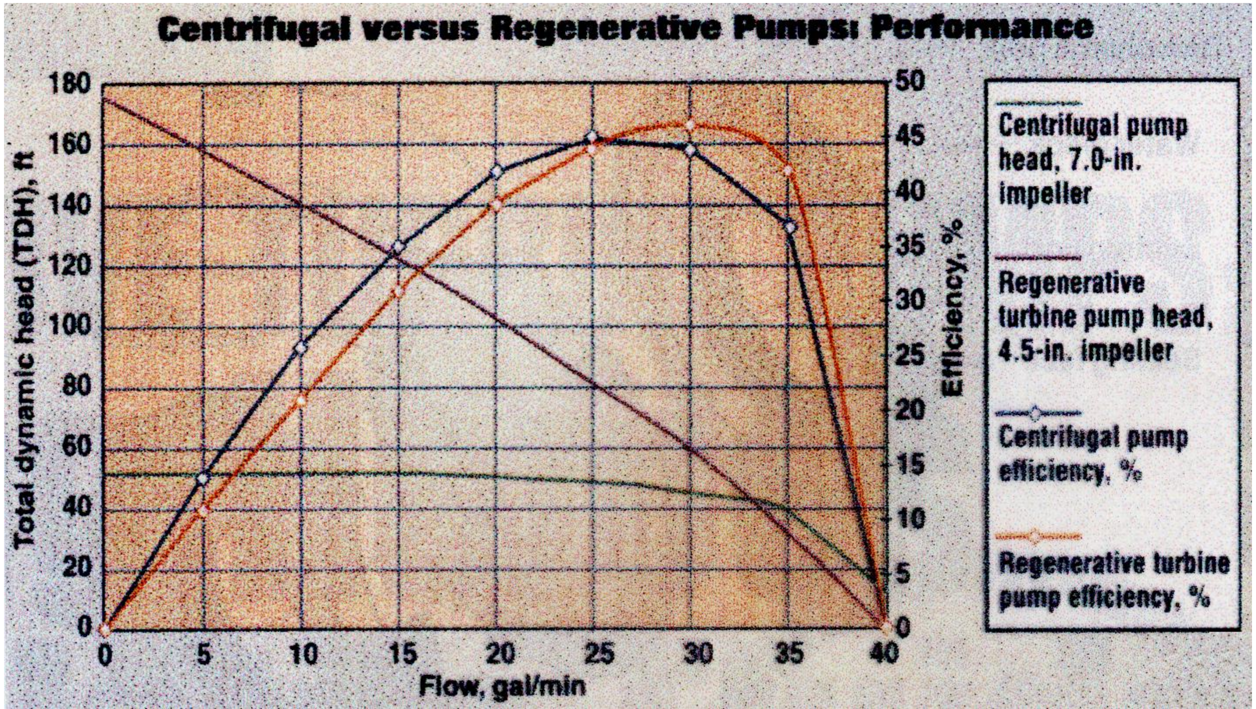
The increase in pressure around the periphery of the impeller from suction to discharge gives rise to a net unbalanced radial force, increasing rapidly with decreasing flow rate. The regenerative pumps should therefore have heavy duty radial bearings and stiff heavy shafts. Operation at very low loads and at no flow should be avoided. The pump should be started with open suction and discharge valves.



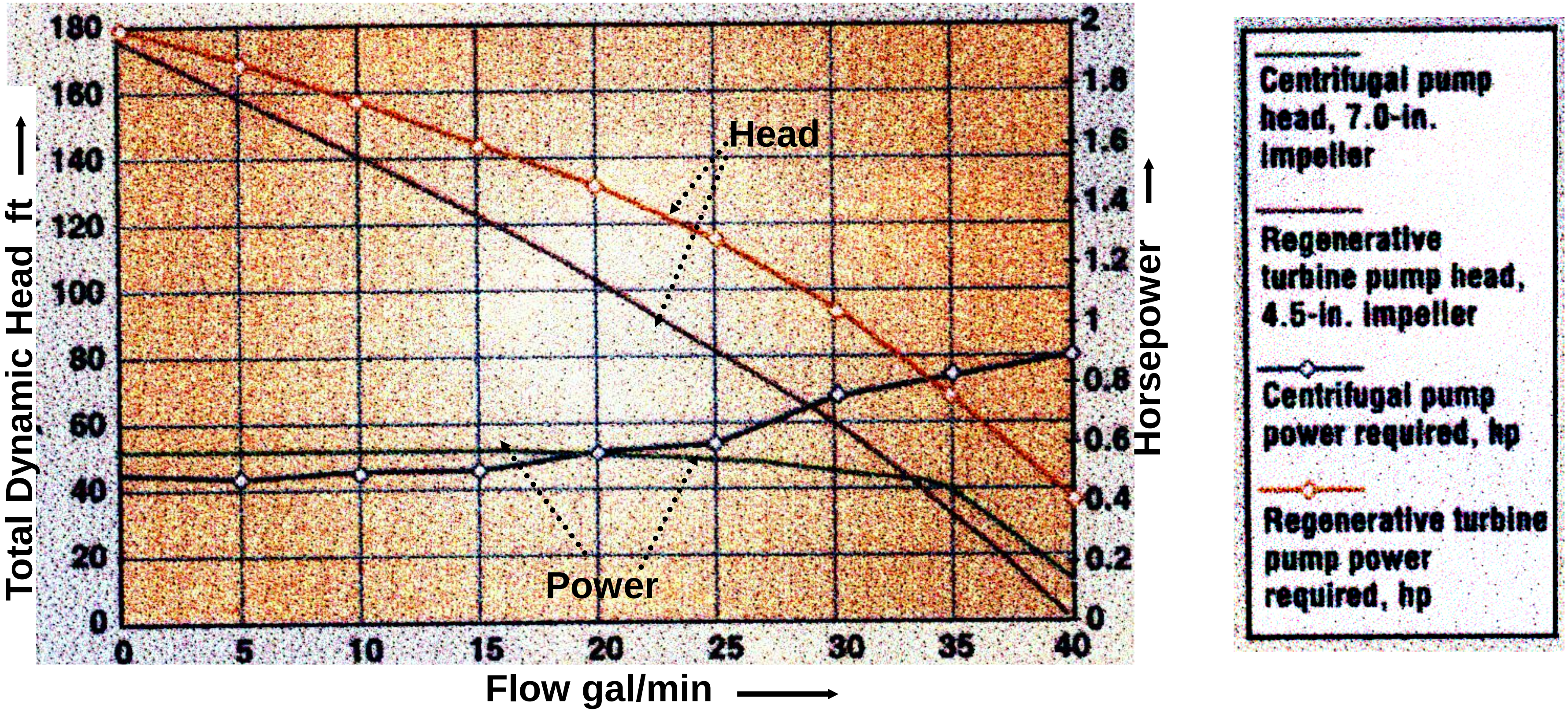
# Regenerative Pumps (iii)

Regenerative pumps have steep head/capacity curves rising to the maximum at shut off conditions. Considerations for inclusion of a relief valve at pump discharge, relieving to the suction system should be made to protect the pump from the high very low flow pressures.

The efficiency falls off very rapidly with increasing viscosity and with a corresponding high increase in power requirements. These pumps are used for clean, low viscosity liquids when small quantities have to be pumped to high pressures.



# Regenerative Pumps




Several impellers may be assembled on the same shaft giving a very high pressure low capacity pump. Multistage regenerative pumps need particular attention in their assembly to ensure setting the impellers and stator components to maintain the correct narrow internal clearances, otherwise the performance and efficiency will suffer noticeably.