Cranfield
UNIVERSITY

PUMP BASICS II

Mr Tony Salisbury

ROTODYNAMIC PUMP DESIGN 1

1. INTRODUCTION

This lecture deals with the main design principles applied rotodynamic pumps where centrifugal pumps are the main subject. The order of discussion will follow the flow path from inlet to outlet, and will cover moving and static parts in contact with the liquid, then move to discuss sealing and bearing arrangements. The correct selection of materials will be outlined in the lecture on Selection, as will the influence of such industry codes as API 610 and ISO and US standards.

2. CENTRIFUGAL PUMP DESIGN

2.1 Selection of Driving Speed

A look at the Euler Equation indicates that for a given pressure rise from an impeller driving at higher speed reduces the impeller diameter. Construction transport and installation costs are related to size so that a smaller pump for a given duty will in the first analysis cost less. This idea was followed through by the Power generation Utility, the CEGB some years ago Figure 1 shows how size of boiler feed pumps has reduced with increasing rotational speed. The well known plot of overall efficiency against characteristic number k_s, Figure 2, demonstrates the relation between hydraulic performance, speed and the efficiency, hence the power.

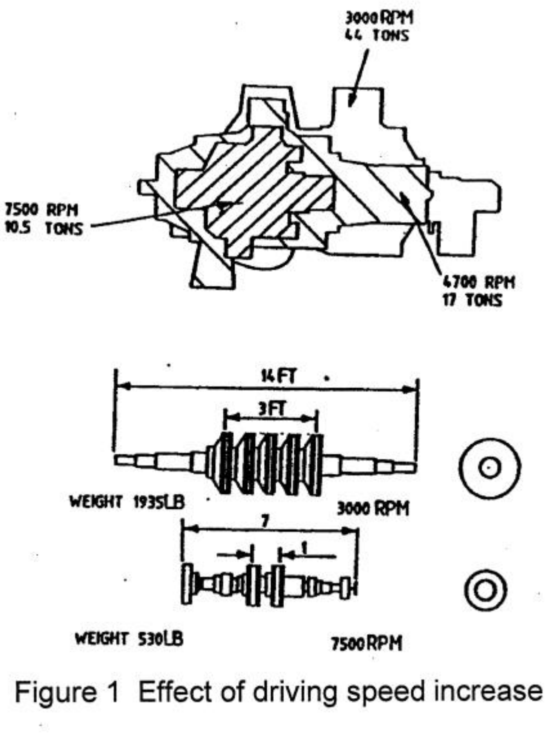


Figure 1 Effect of driving speed increase

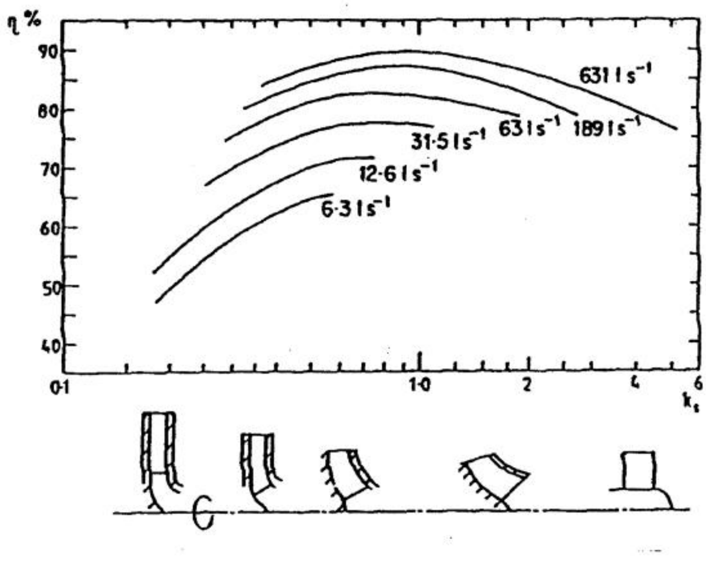


Figure 2 Plot of efficiency

The practical problem is that as the size reduces the local velocities must increase, so that a limit to performance is imposed by cavitation inception. Figures 3 and 4 present rotational speed limits against flow rate for a range of $NPSH_A$ values for typical single and double suction pump designs based on American experience. When a machine has a high added value as in boiler feed duties, development work can be afforded in improving cavitation behaviour, but the plots provide a useful guide for general pump designs.

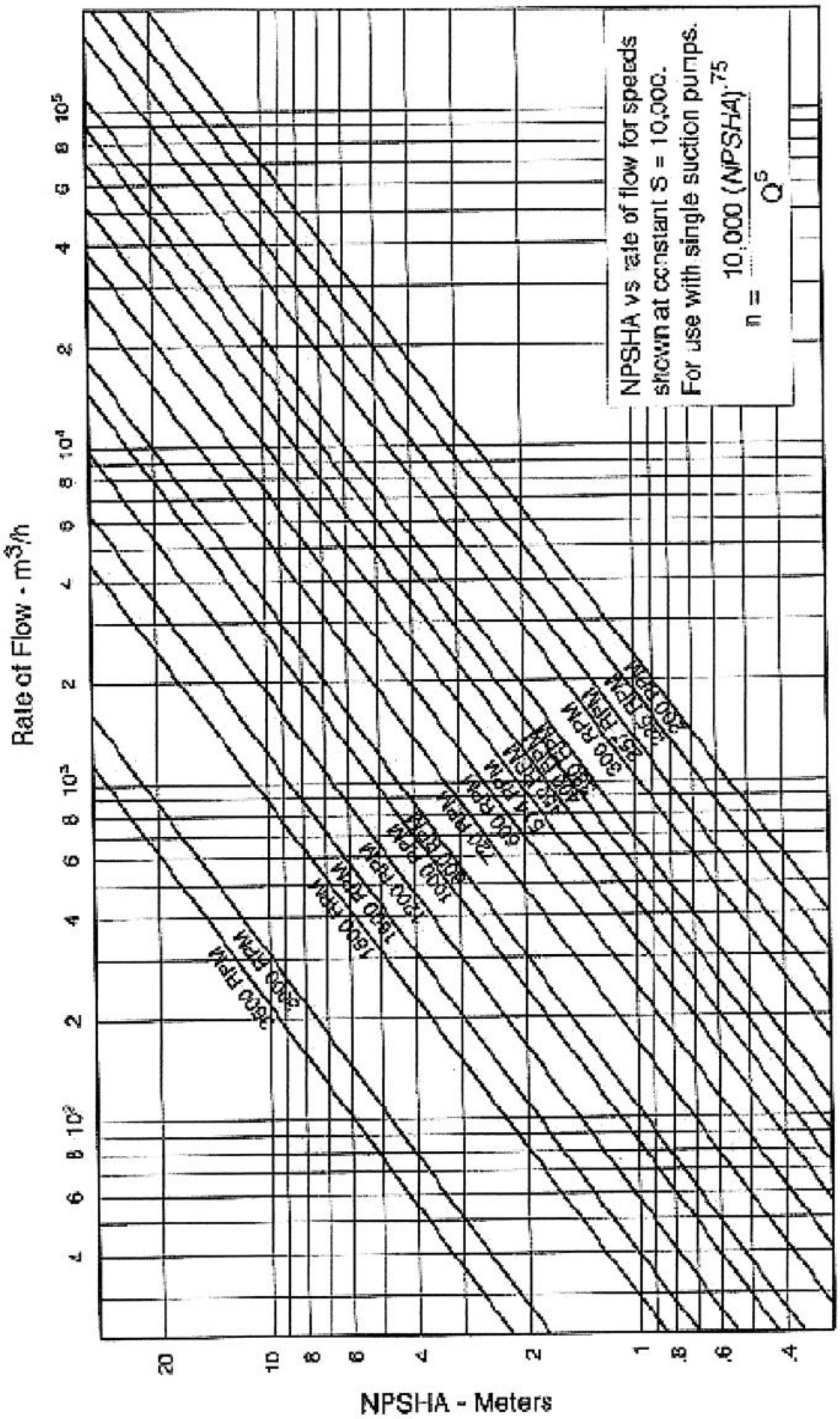


Fig 3 Recommended typical operating speed limits for single suction pumps

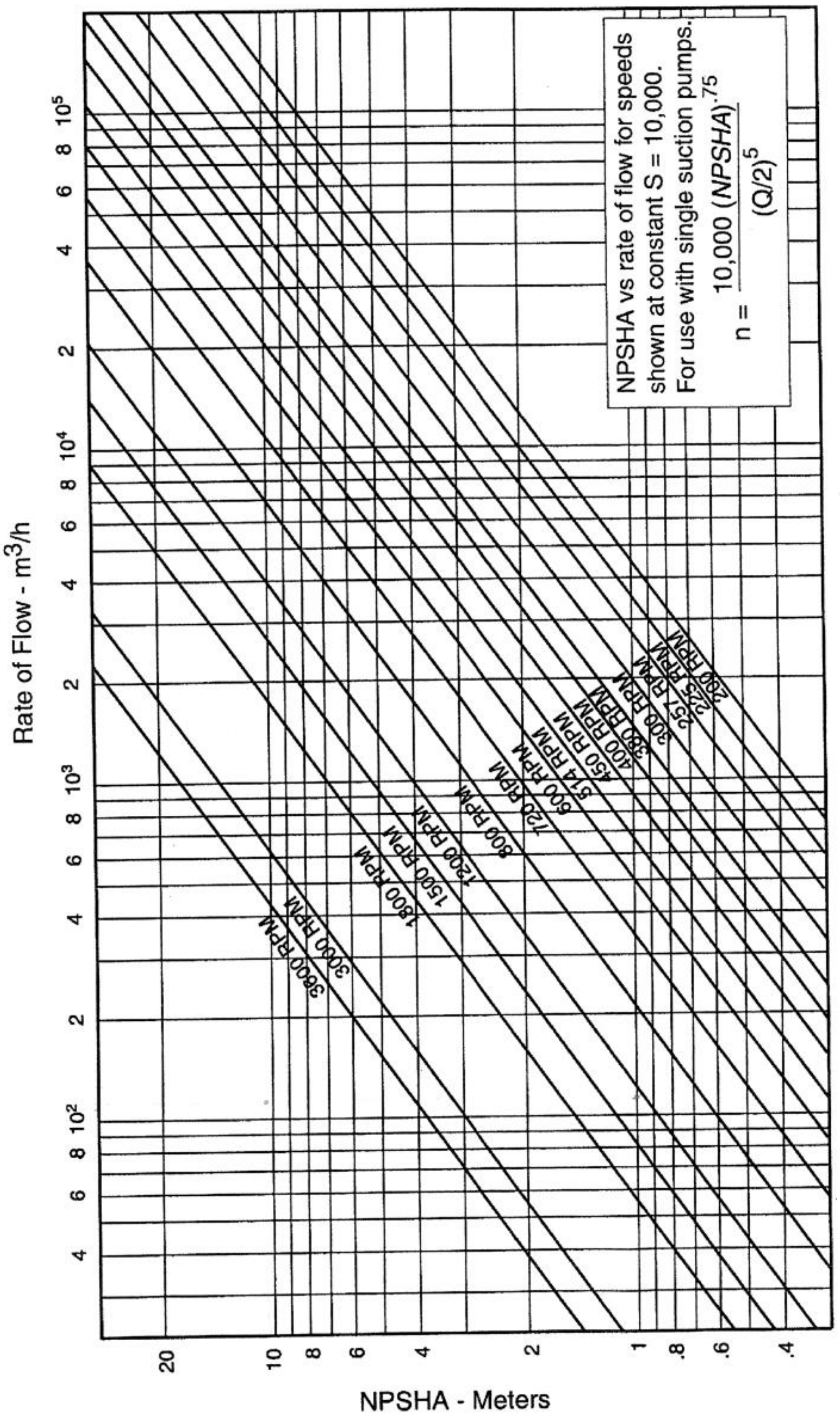


Fig 4 Recommended typical operating speed limits for double suction pumps (metric)

2.2 Hydraulic Passage Design

The passage shapes will be discussed in the order that the liquid passes through the machine: inlet ducts, impeller and collector casings (diffusers and volutes).

2.2.1 Inlet Ducts

The simplest duct is the straight inlet (figure 5), it is also the most desirable as the liquid enters the impeller without added swirl and this should lead to better efficiency. In some applications, particularly multi-stage designs, side entry ducts have to be provided. A typical design being illustrated in figure 6. The dimensions shown have been developed to give an acceptable flow into the impeller at the design flow and to minimise flow problems at flow rates away from the design value.

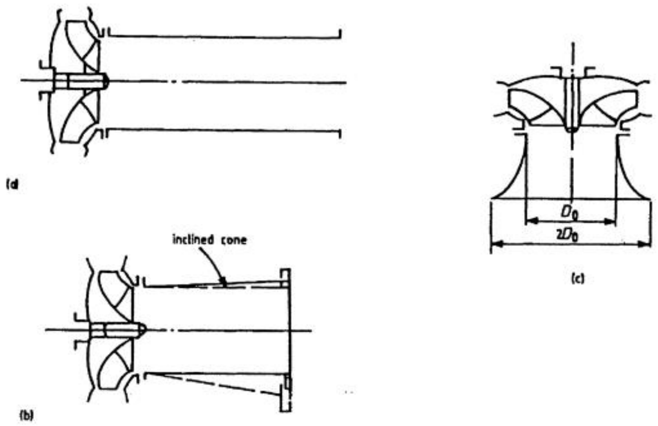


Figure 5 Alternative suction system for an end suction pump

a. Co-axial cylindrical straight suction line
b. Inline cone type suction
c. Flared inlet that may be used for vertical design

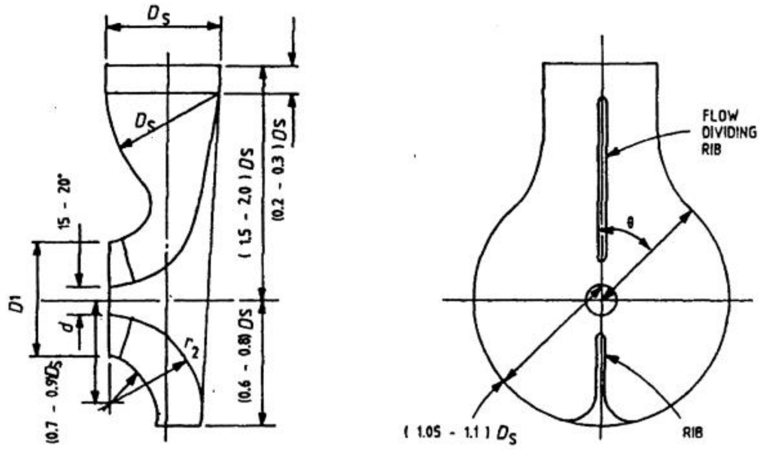


Figure 6 A typical side suction duct for a single suction pump showing a flow dividing rib

2.2.2 The Impeller

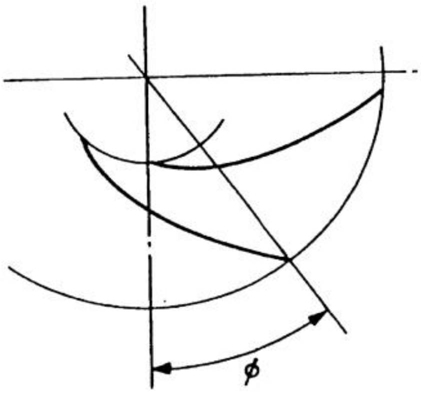
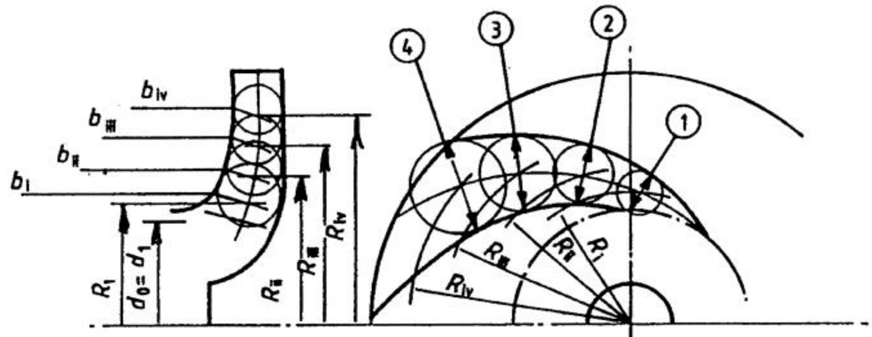


Figure 7 Definition of blade overlap

The Euler equation shows that at the design flow condition the pressure rise is related to the outer diameter, the outlet angle β_2, and the driving speed at any given flow rate. Typically β_2 lies in the range of 15 to about 35^0, and the number of blade passages will be between 5 and 8 as experience has shown that providing more than this affects the efficiency. The blade profiles follow an approximation to the Archimedian spiral as being the best shape. A good modern practice uses an overlap of about 45^0 (figure 7.)



DETERMINING THE CROSS-SECTIONAL DIMENSIONS OF AN IMPELLER PASSAGE

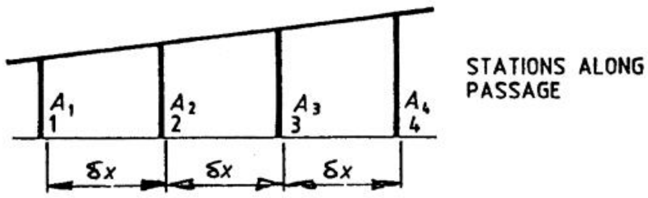


Figure 8: Cross sectional area of impeller passage plotted against passage length

The relative velocity decreases through the impeller passages so that the passage shape has to be designed to give a smooth change of velocity and to avoid sudden changes of area. Figure 8 shows a drawing board method of checking the area change. Modern CAD software will do this and check the best shape of blade profile is used. A correction to the outlet angle is used to allow for the velocity distortions, as shown in figure 9. The technique is known as slip factor correction. This effectively modifies the Euler triangle at outlet as sketched in figure 10. Several approaches are available for correcting the velocity diagram and are described in contributions by Phleiderer (1), Stanitz (2), Karassik (3) and Weisner (4). All reduce the value of the theoretical slip, V_{U2}, and a summary of the common factors that have been used can be found in Turton (5).

The outside diameter is found using well established empirical equations, due to Pfleiderer and others, which have been incorporated into manufacturers computing software. The inlet or suction zone geometry is established by using the principles of cavitation avoidance already discussed in the lecture on cavitation with empirical approaches based on suction specific speed limits and experience being covered in the listed references.

The impeller material is related to the fluid being pumped as discussed later in the lecture on Selection. Stress calculations are a specialised subject and will not be discussed here.

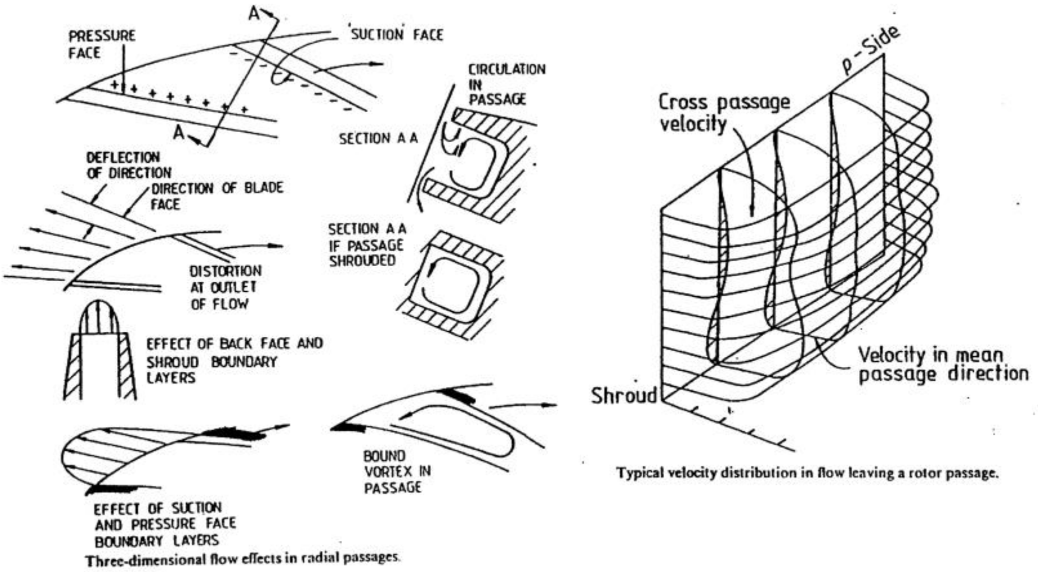


Figure 9 3-dimensional flow patterns in radial impeller passages

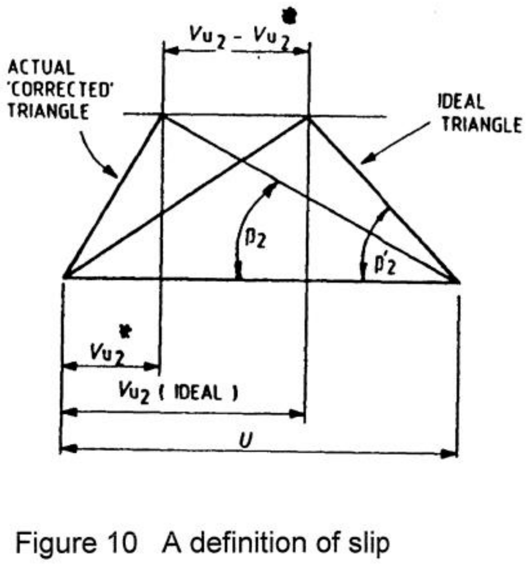


Figure 10 A definition of slip

2.2.3 The Collector Casing

Two different collector systems are used; in the single stage machine the usual casing used is the volute, figure 11, in the multi-stage designs a diffuser is fitted.

The volute casing collects flow from the impeller then reduces the velocity in a diffuser before the liquid is discharged into the delivery pipe. The volute cross section area increases, as figure 11 shows, to the throat area. This area increase is related to the quantity of flow that has left the impeller; so looking at figure 11 after 90^0 from the cutwater 25% of the total flow rate has emerged into the volute, after 180° 50% and so on. The areas may be found assuming a velocity equal to the throat velocity applies round the impeller or by assuming a free vortex distribution across each volute section. As can be seen from figure 13, the velocity distribution is not affected to any significant degree. The throat area may be found using empirical equations or by using the area ratio principle proposed by Anderson and filled out by Worster (6).

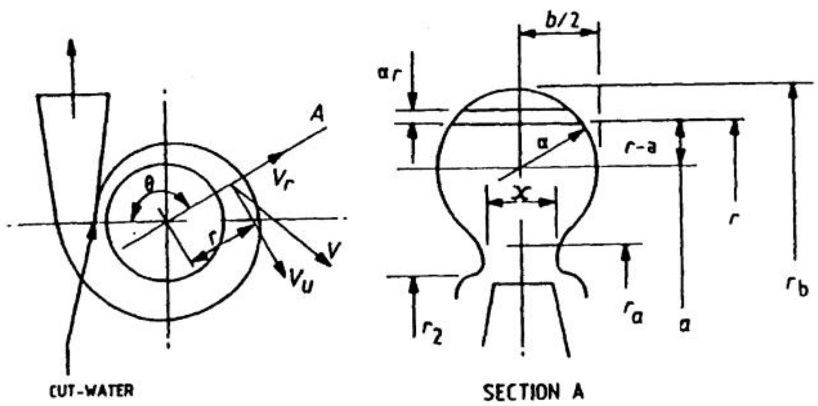


Figure 11 A typical pump volute

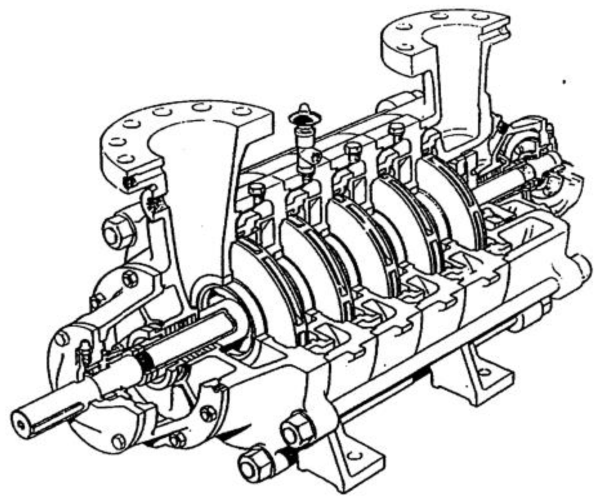


Figure 12 A multistage pump

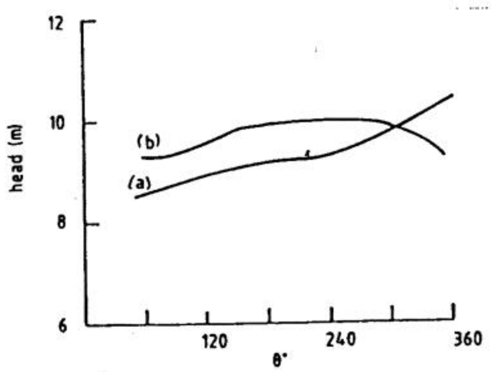


Figure 13
A comparison of the distribution of pressure around the
periphery of a centrifugal pump due to a volute. (a) Designed on the assumption of constant angular momentum and (b) on the assumption of constant velocity distribution

In multi-stage machines the flow path is more complex as shown in Figure 12. A typical diffuser is sketched in Figure 14, and as can be seen the diffuser passage expands in area to give a pressure increase. After passing through the diffuser the liquid is then conveyed round a bend and down into the inlet of the next impeller. Several different guidance ducts are sketched in Figure 15. The passage guiding flow into the impeller is designed to give a slight pre-whirl into the impeller to promote stable flow conditions.

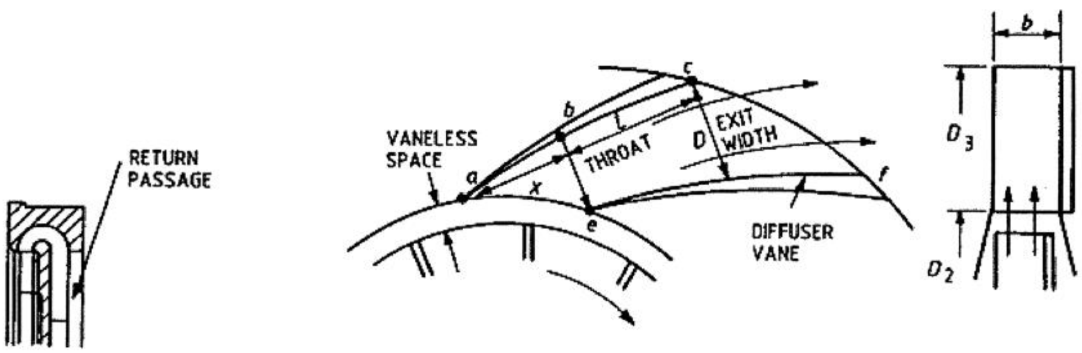
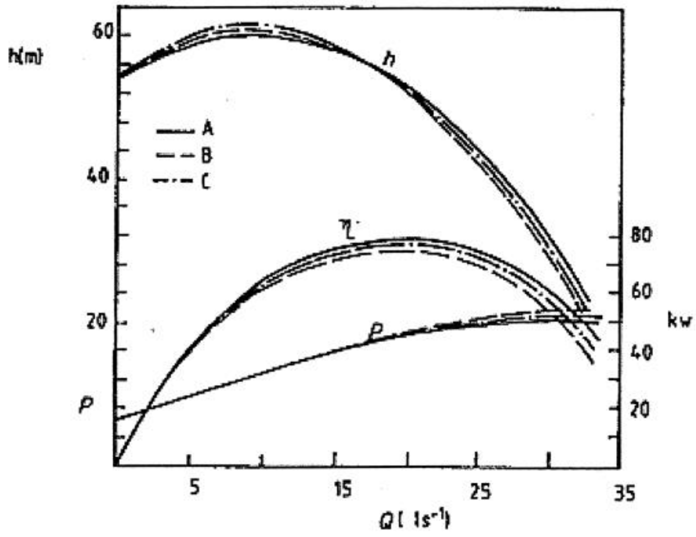


Figure 14 A typical vaned diffuser

The casing that surrounds the flow system just described has to be pressure tight and tested according to API 610 if used in the petrochemical field or to ISO or other standards for other uses. Two types of casing are in use split casing with the joint on the horizontal centre-line, or barrel type casings with the internal parts mounted as a cartridge. The choice is dictated to some extent by the pressure rise provided. In either case the mounting brackets are placed as near to the horizontal axis as possible to minimise lateral stress problems due to expansion, and the brackets at the thrust bearing end act as location on the bed plate and those on the other end allow for some sliding to allow for expansion of the casing under the normal temperatures reached. Usually, the bearing housings are mounted separately for ease of inspection. Typical machines are shown in Figure 12 and 16.



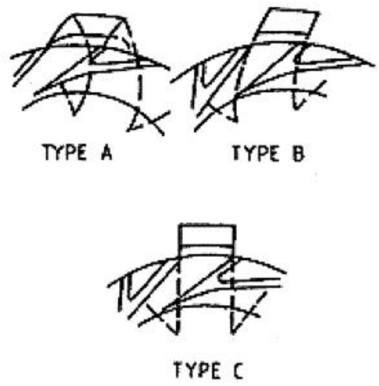


Figure 15 A comparison of the cross over duct designs that may be used to connect stages in a multi-stage pump, and their effect on the stage performance

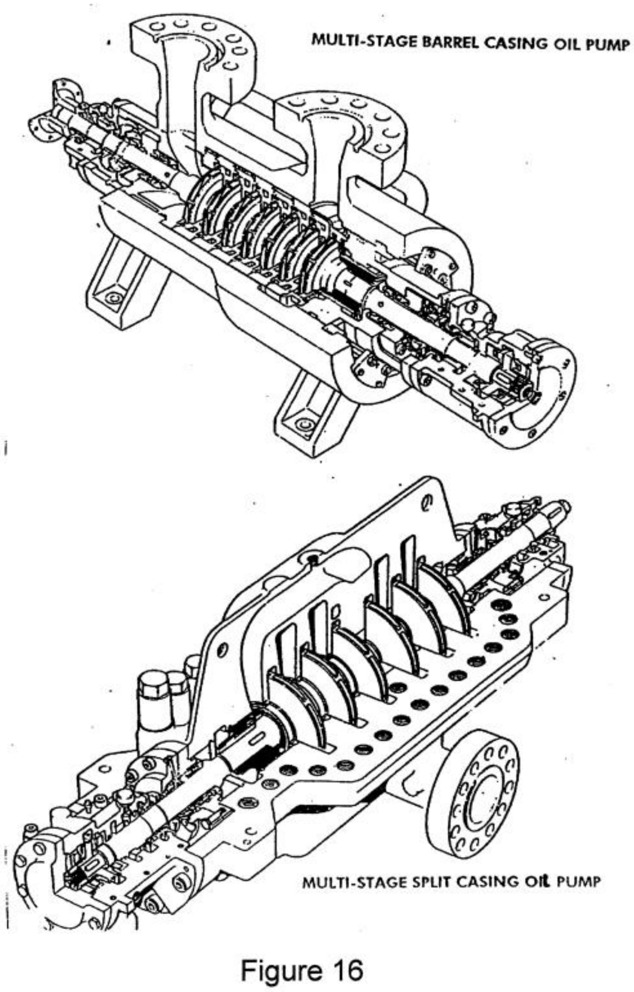


Figure 16

2.2.4 Sealing System

Two types of seal are used in rotodynamic pumps: the mechanical seal and the packed gland. The Mechanical seal is used singly for normal duties or in series for applications where leakage is not desirable for safety and environmental reasons. The packed gland is the older system and is still preferred for some applications.

2.2.4.1 The Single Mechanical Seal

A typical single seal is shown in Figure 17. The active part of the seal consists of a hard ring in contact with a softer ring. The two mating rings are lapped together to give a very high degree of surface finish. A very thin film of liquid separates the two surfaces, and for this to exist great care over cooling is needed. Effort is also needed to ensure that no grit of contamination can enter the seal. If the product is suitable a flush from the discharge flange of the pump is used with the flow either being taken back into the suction line or through the balance chamber, if that is the means of axial thrust balance. If the product is at the wrong temperature and is dirty or at the wrong pressure, action has to be taken to ensure the health of the seal and a more complicated system installed. Examples of these will be discussed. The materials used for the contact rings vary with the product being sealed and the pressure differential that is applied. One ring will, in many cases, be a carbon graphite compound impregnated with resin metal or electrographite, the latter being good at temperatures of 250°C but not resistant to wear. A wide range of materials are available for the other ring and the IMechE guide on mechanical seals edited by Summers-smith (7) should be consulted for these.

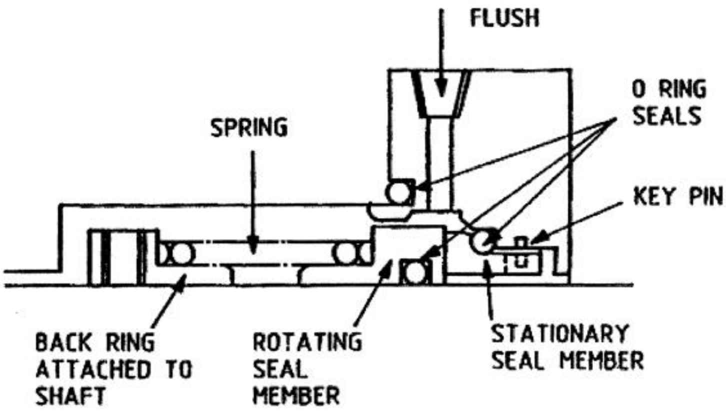


Figure 17 A simple mechanical seal

The simple seal shown in Figure 17 is limited to fairly low sealing pressures, and for higher differential pressures force balance has to be provided as shown in Figure 18.

The balance ratio of a mechanical seal is defined as:

B = hydraulic loading area/sealing interface area

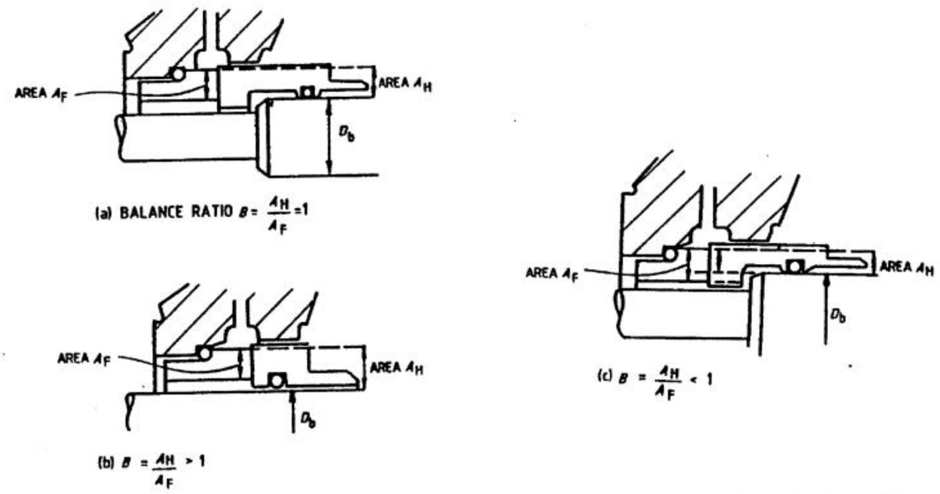


Figure 18 Force balance in a mechanical seal

If B $\geq$ 1 the seal is unbalanced and used for pressures up to and 10 bar gauge, if B < 1 the seal is balanced and will cope with the pressures of up to about 70 bar gauge.

Typically B will be in the range 0.65 to 0.85 for these devices, with the higher value rendering them less likely to "blow open" under the hydrostatic pressure between the sealing surfaces but at the expense of higher surface loading. The guide mentioned above details how to calculate spring loads and surface pressures.

2.2.4.2 Double Mechanical Seals

Where leakage is not acceptable due to the properties of the product sealed double seals are required. Typical arrangements are shown in Figure 19.

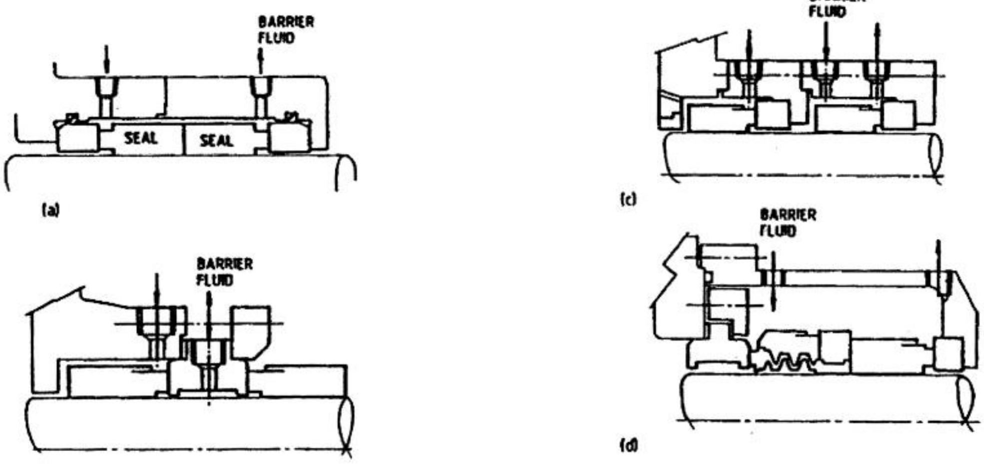


Figure 19 Double mechanical seals

In Figure 19, several arrangements are shown; in (a) a back to back design, the two seals are flushed by a barrier fluid that will not affect the product pumped if leakage occurs into the pump, and does not cause unacceptable pollution if the outer seal leaks: in (b) the inner seal is flushed by the product sealed, and the outer is cooled by a barrier fluid and by being exposed to the environment: in (c) the seals are in series with the inner flushed by product and the outer by a barrier fluid: in (d) an alternative to springs is shown fitted to the inner seal – a bellows type spring. These designs are used for high temperature duties (up to 4000°C) with appropriate provision of cooling, for food and pharmaceutical duties where contamination cannot be allowed, and other difficult applications. The implications of using these arrangements is of course that a secondary chemical system has to be provided and maintained. Modern designs also fit a neck ring at the outer end of the assembly to limit leakage if a catastrophic failure occurs. The selection and application of such seals depends on the advice of the seal manufacturers and the IMechE guide already mentioned may also be consulted.

2.2.4.3 Packed Glands

These consist of soft rings compressed into a housing as sketched in Figure 20.

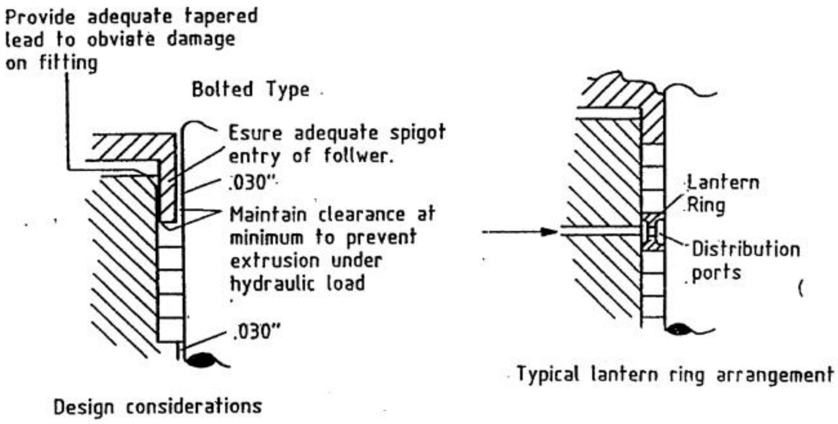


Figure 20 Packed Gland Details

Materials used for the rings vary from impregnated yarns made from natural materials and synthetics, and for high temperature applications reinforcing wires are introduced with limits of temperature ranging from copper (80°C) to stainless steel (875°C). The material has to cope with the product sealed and the rubbing temperature generated.

Table 2.1 presents some data on materials commonly used for normal temperatures.

Four or five rings is regarded as the usual number at present with the conventional compression used, usually of 5mm cross section, and a surface finish of 0.2 to 0.4 μm. Good fitting is the secret of successful glands (1-2 drops per minute being the usual controlled leakage to provide some lubrication). The run out of the shaft sealed should be less than 0.05mm total indicator reading.

Table 2.1. A Selection of Soft Packaging Material

Packaging Type	Applications
Cotton – grease impregnate	Water and Aqueous based liquids Max temperature 80°c Max sliding speed 7.5 m/s
Hemp-grease impregnated with	Hydrocarbons max temperature and speed as cotton
Crimped metal...aluminium, lead or white metal foil on mineral fibre core	Hot Hydrocarbons Max temperature 200°c with lead or white metal 500°c with aluminium.
Aromid fibre	Strong acids and alkalis (pH 2-12) Max temperature 250°c Max sliding speed 15 m/s
Exfoliated graphite	Strong oxidising acids (eg Nitric acid) Max temperature 500°c Max sliding speed 10 to 15 m/s

[Note that asbestos impregnated with PTFE or graphite was used for steam and strong acids but is now prescribed for health reasons]

2.2.5 Shaft Bearings

Two types of bearing are used; rolling element (ball and roller bearings) and journal or hydrodynamic designs.

Rolling bearings are extensively used in smaller pumps, and are available in a wide range of sizes. Single row roller and ball bearings are widely used in a roller plus ball bearing combination, with the ball bearing acting as the location for the shaft assembly and also taking residual axial thrust. Where the thrust load is larger a double row ball bearing is used and this also can take thrust reversal. In some designs opposed taper roller bearings are used for thrust and location. They are available as complete items ready for use and are tolerant of lubricant so they can be placed inside machines filled with liquid without the need for seals; they can be supplied sealed for life avoiding the need for lubrication systems, and can be grease lubricated. The choice of bearing lubrication is usually a matter of user preference based on operating experience.

The criterion for bearing choice is the L10 life (figure 21). It is defined as the number of cycles (revolutions) that 90% of a set of identical bearings will operate without suffering fatigue failure. The average life, the period when 50% of the bearings will have failed is about 5 times the L10 life, and some may last for very much longer. ISO 281 gives calculation method for rating the bearing life. The L10 life may be calculated from

$$L10 = \left(\frac{C}{P}\right)^{x} \textit{(revolutions)}$$

P = radial load,
C = basic dynamic capacity of bearing,
x = 3 for balls and 10/3 for rollers

$$L10\,(hours) = \frac{L10}{60N}\left(\frac{C}{P}\right)^{x}$$

N = rotational speed in rpm

C appears in the maker's catalogue together with the method for finding the radial load P. For pumps in continuous use the L10 hours figure is 25000 to 50000, and for high reliability 100000.

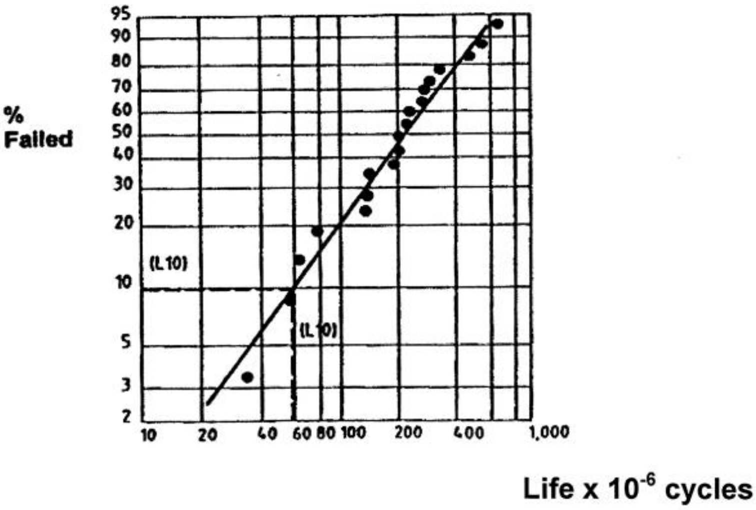


Figure 21 The L10 Life of Bearings

Normal internal clearances are specified by the makers and are a function of size. The usual fitting instruction for fits between races and the housing/shaft assumes that this applies; clearances are obtained by selective fitting and are used for high temperature pumps to allow for increased thermal gradients and thus ensure a better life. The other grades are difficult to obtain.

Lubrication may be by grease or oil, with grease giving a limited life. Grease life can be found by using empirical formulae, and if this is less than about 1000 hours this method is not feasible and oil lubrication should be used.

As will be commented later in the lecture on applications, over greasing can give premature failure. An oil system provides both lubrication and bearing cooling to some extent but either the housing will need cooling or the oil circuit must be fitted with a cooler. Care has to be taken that overcooling does not occur as this may close up clearances and lead to failure. If there is a water laden atmosphere a turbine oil needs to be used so that water in suspension in the oil can be removed easily.

2.2.5.2 Plain Bearings

These are usually hydrodynamic in operation with the static and moving surfaces separated by a thin oil film. A typical bearing under and pressure distribution under load is shown in Figure 22. As can be seen the shaft centreline is eccentric to the bearing axis and the rotation of the shaft creates a pumping action so that the oil film supports the shaft loading. The load bearing capacity varies as the speed of rotation. Usually the bearing in a multi-stage machine is made in two halves, the split being on the horizontal centre line. The bearings are provided with oil grooves, which may be axial (Figure 23a), the normal method or if the load rotates with a central groove (figure 23b) in both cases the oil is fed from behind the shell of the bearing. Dirt gutters are provided with axial grooved bearings to ensure that any dirt in suspension is flushed out by oil, Figure 24.

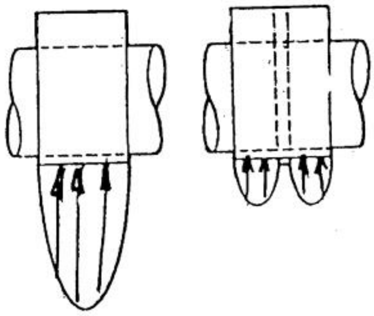


Figure 23

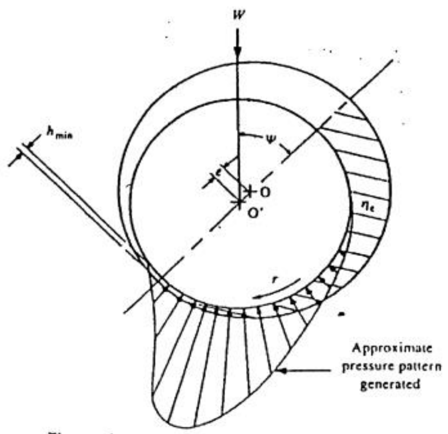


Figure 22

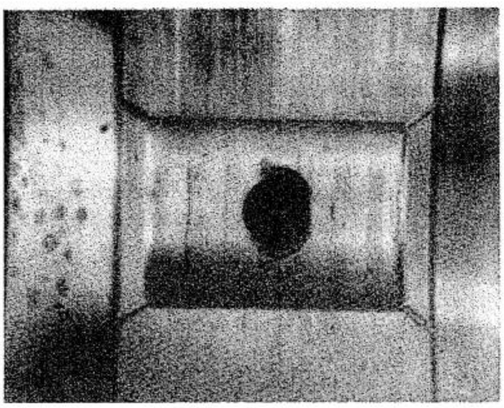


Figure 24 Oil Groove with dirt gutters

Referring back to Figure 22, the displacement of the shaft centre from the bearing axis is known as the eccentricity e. The usual ratio used is the eccentricity $\varepsilon = e/c_r$ and c_r, the radial clearance between shaft and bearing, is of the order of 0.1% of the journal diameter. Figure 25 is a plot of the locus of the journal eccentricity with the attitude angle, ψ . The higher the speed and the viscosity, the smaller is e and the converse applies with increasing load.

These parameters are not independent, the viscosity changes with increased load which brings more heat into the oil.

Typical design parameters:

Length/diameter = 0.5-0.75
Diametral clearance to journal diameter = 1 x 10^{-3} to 2 x10^{-3}
h_{min} = 0.025mm
Peak pressure in the oil film 10N/mm^2

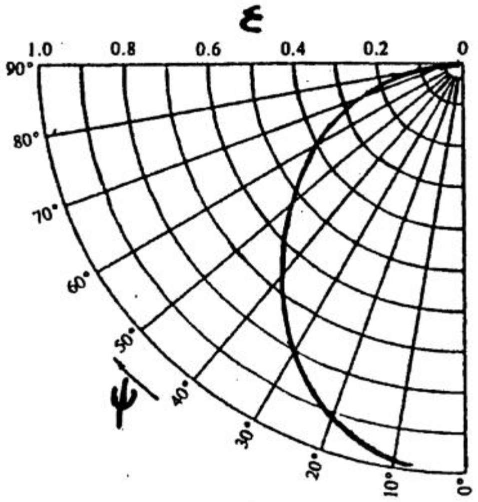


Figure 25 Locus of shaft centre showing relationship between eccentricity ratio and attitude angle, ψ

The designer and the operator of multi-stage pumps has to remember that the oil film acts as a "spring" so that if the load is dynamic due to vibration or out of balance it will act as a damper or resonate with other "springs" in the system to cause instability. If the shaft and bearing are almost concentric the oil film acts as a soft spring and if eccentricity is high it act as a stiff spring. Resonance can destroy bearings in high speed machine. Two modes of running are of interest. Firstly, synchronous whirl where the shaft orbits round the equilibrium position causing a vibration at shaft frequency whose amplitude depends on film damping and stiffness. (If run up is fast critical frequencies are passed through quickly and oil film damping helps). Secondly, half speed whirl (so called) can occur when a small perturbing force is applied and then removed (hydraulic effects perhaps): e increases, the attitude reduces and the oil film acts to restore to equilibrium but at a different angle causing the shaft to orbit around point O in Figure 22. When this latter orbiting occurs a concentric film is set up which cannot support load and metal to metal contact takes place. Recently developed bearings as shown in Figure 26 counteract this problem. In all designs the shaft should be finished to 0.4 μm CLA (centre line average).

Journal bearing types: left two-lobe bearing; right tilting pad bearing.

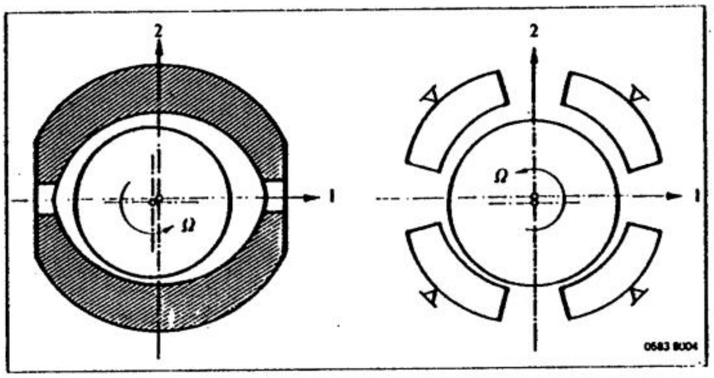


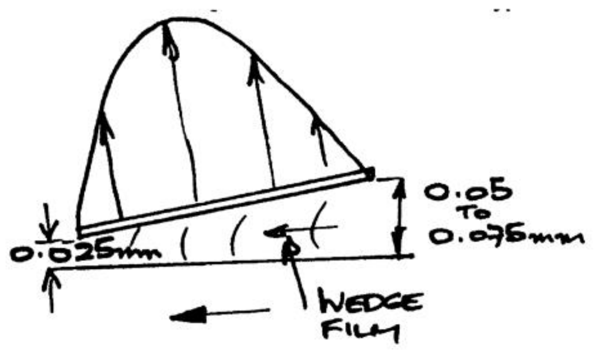
Figure 26 Journal bearing type

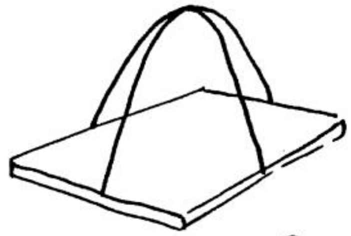
Modern journal bearings give long times between overhauls given the right conditions, as is required by codes like API 610. They are required to cope with start up and occasionally to run with no oil film present. Materials and the bearing designs have been developed to deal with the range of conditions applied to them. An authorative source of information is ESDU data item 84031. In some applications water or product lubrication is used. If water is used a special grade of rubber is used for the bearing and the shaft sleeve is suitably hardened, and care has to be taken that the system is never run dry. In some cases a resin fabric bearing is used and here care has to be taken to allow for swelling due to water absorption. If a corrosive is present suitably resistant materials must be used for both shaft sleeve and the bearing. For oil lubricated journals the normal bearing is formed with a steel body coated with a tin layer on to which Babbitt (or white metal) is deposited. This gives a surface that will conform to the shaft finish and cope with hard particles without serious scoring of the mating surfaces and it will withstand 3.5N per square millimetre at start up and 10-15 N per square millimetre when running. For higher loadings, lead bronze compounds are used and use is made of aluminium tin mixtures of some applications, as these do not require hardened shaft surfaces. For more detailed discussion tribology text should be consulted.

2.2.5.3 Thrust Bearing

Thrust bearings that use tilting pads utilise the oil wedge principle, Figure 27. They may either be of the Michel type or the Kingsbury design. In the Michel bearing the pads rest on step type pivots that are placed about 60% off the pad width from the inlet edge ensuring that they may not be placed the wrong way round or at 50% in some cases. In the Kingsbury design the pads rest on central buttons ensuring (it is argued) that they have more freedom to adjust to the load and will cope with reverse running if it occurs. For detail discussion, see Summersmith (8).

The surface finish of the pads is high and the thrust collar attached to the shaft is ground on both sides. Figures 28 and 29 illustrate typical bearing blocks.





A SKETCH OF PRESSURE DISTRIBUTION ON A PLATE

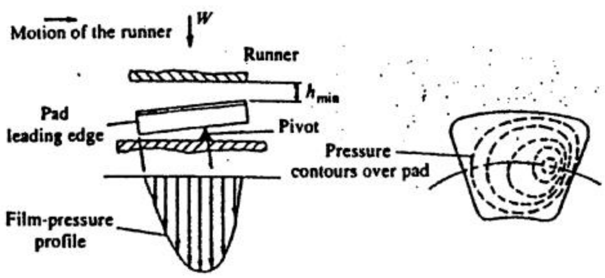


Figure 27 Generation of hydrodynamic pressure in tilting-pad thrust bearing

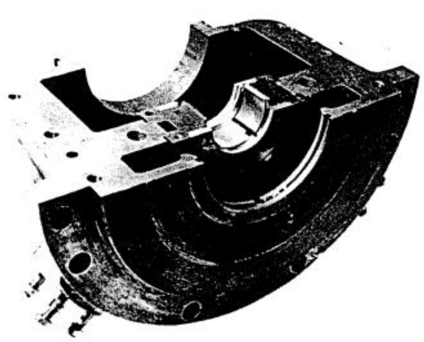


Figure 28 Bearing block with radial bearing (lower part)

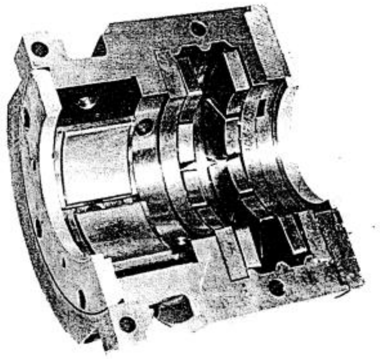


Figure 29 Combined radial/thrust bearing (lower shell)

2.2.5.4 Thrust Loads on Rotating Assemblies

2.2.5.4.1 Radial Trust Loads

In centrifugal designs fitted with a volute casing the pressure distribution round the impeller periphery can produce large forces at flows away from best efficiency point which have a large fluctuating component, see figure 30. This effect can cause shaft breakage.

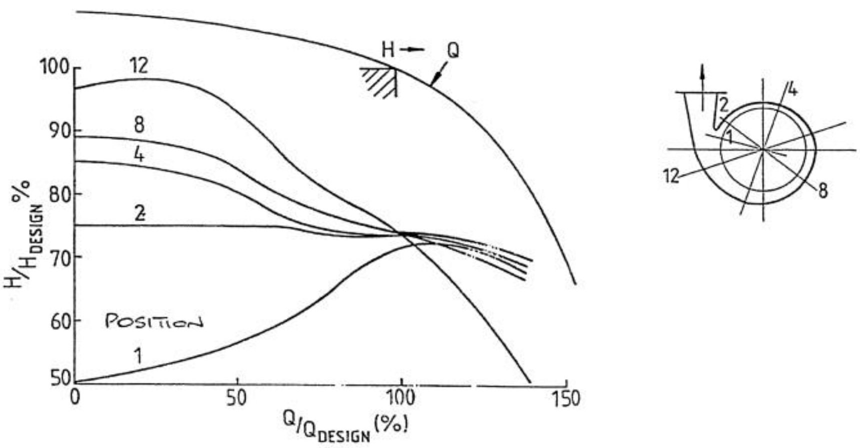


Figure 30 Variation of pressure around the periphery of an impeller at several different flow rates

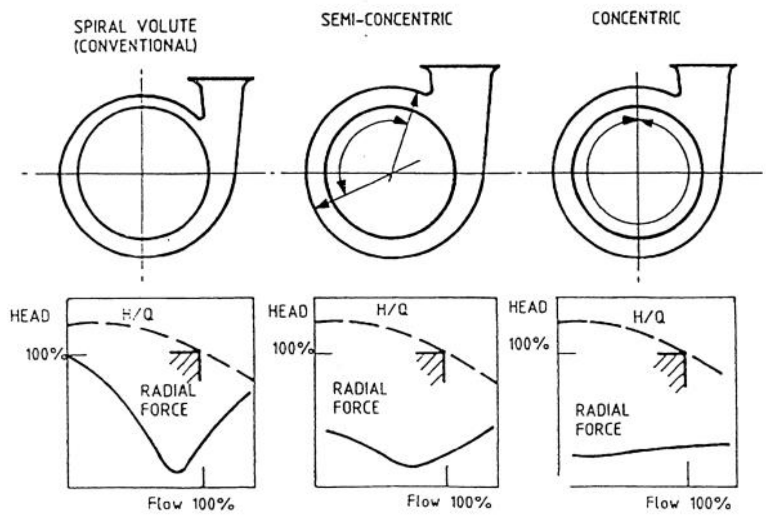


Figure 31 The effect of casing change on radial load

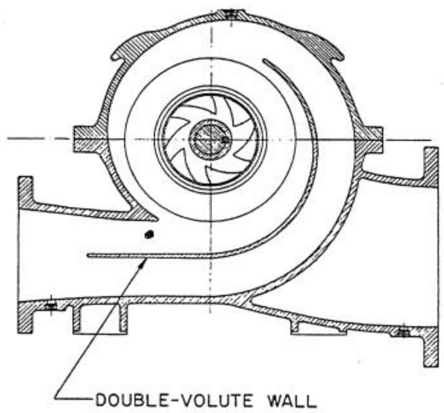


Figure 32 Traverse section of a double volute casing pump

Figures 31 compares three volute designs and show how the pressure distribution round the impeller and radial force is changed by providing a semi-concentric or concentric casing. The latter method is found in small pumps like those used in garden fountain systems. Figure 32 illustrates a method of balancing out most radial load by providing a double volute. This generates two opposing radial forces on the impeller and so reduces the radial force produced. This method will obviously reduce the efficiency because friction losses will be increased in the volute, but a number of manufacturers provide this design as their standard and others will provide it if requested. Stepannof (9) provides an empirical equation for a radial impeller (sketched in Figure 33) that predicts the radial force at zero flow which produces the maximum force effects.

$$F_{radial} = k\, P\, D_2\, B_2$$

where K = 0.36, P is the delivery pressure, D and B are defined in the figure.

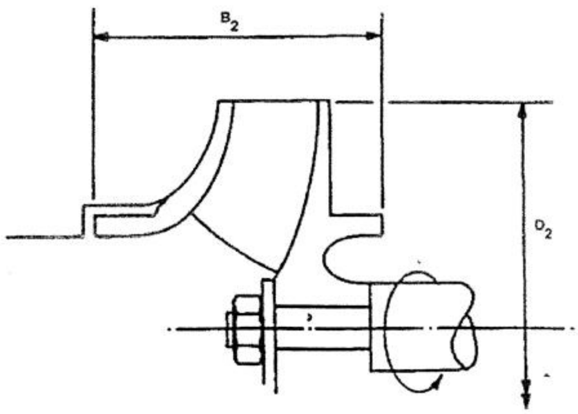


Figure 33

2.2.5.4.2 Axial Thrust Load

Axial thrust loads arise particularly for shrouded impellers due to the pressure distribution on the shroud and the back plate being out of balance, Figure 34 shows in idealized way the pressures and their sizes assuming that those on the shroud and back plate are effectively at delivery level. The fact that the pressure in the suction is less than the other pressures creates a force towards the suction flange for an end suction machine but it can be seen that for a double suction impeller the pressure are in rough balance.

Figure 35 defines the unbalanced forces for a single stage end suction impeller and also how the pressure in the spaces between impeller surfaces and the casing reduce towards the centre of rotation.

Figure 36 shows a very common solution to the problem; the impeller back plate is provided with a wear ring at almost the same diameter as that on the shroud to provide a balance chamber connected to the suction zone by balance holes which provide a larger flow area than that through the back wear ring and ensure that the pressure in the chamber is close to the suction pressure and thus provides force balance. Leakage flow will occur so there is an efficiency penalty. An alternative provided by some manufacturers is to replace the balance chamber with radial vanes (called pumpout vanes) which reduce the pressure close to the shaft and so reduce the axial thrust.

These methods are not really possible in multi-stage pumps so figure 37 illustrates two methods, and balance piston (a) which is the most common and (b) the balance disc. The in board face of the piston or disc is held at discharge pressure and the other face is vented to suction pressure thus generating a force opposing the pressure forces on the rotating assembly and so reducing the load on the thrust bearing.

With mixed flow and axial machines pressure forces are generated partly by pressure produced by the dynamic action of the blading as well as the sort of pressures just discussed but all tend to need balance by one of the methods examined above.

Figure 38 shows a simple end suction machine rotating assembly and summarises the force and torque system applied to the shaft and support system. The design of shafts is not included in this module, but for an approach to the problems to be solved, Steppanof (9) and the handbook by Karassik (3) may be consulted.

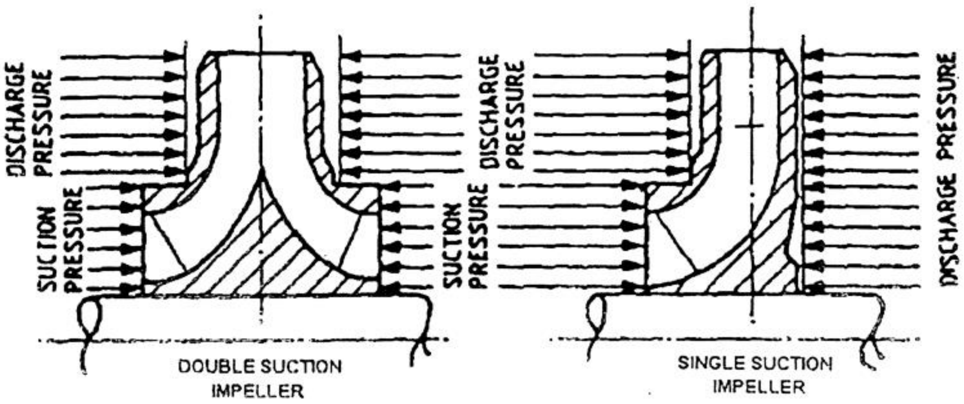


Figure 34

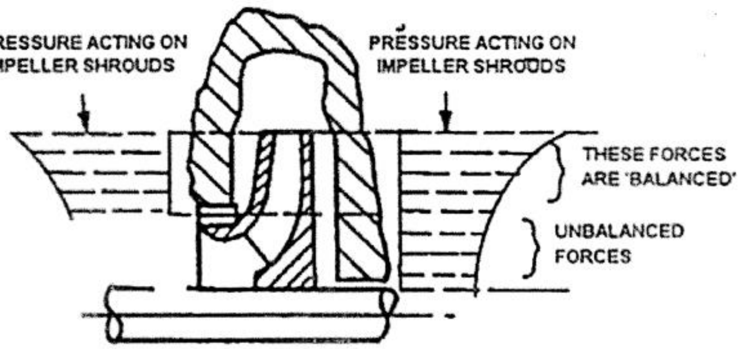


Figure 35

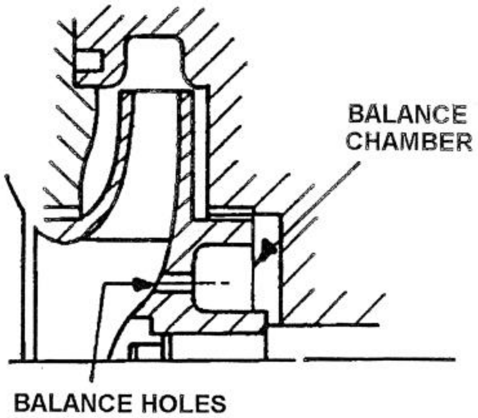


Figure 36

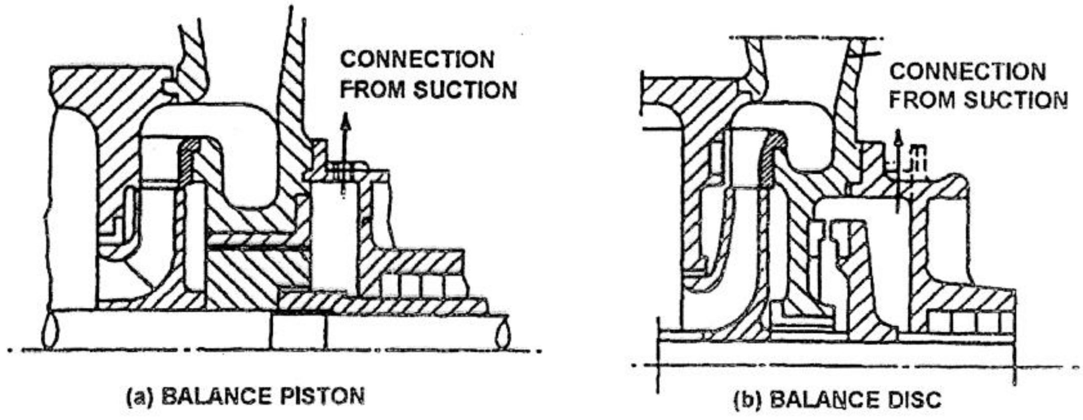


Figure 37

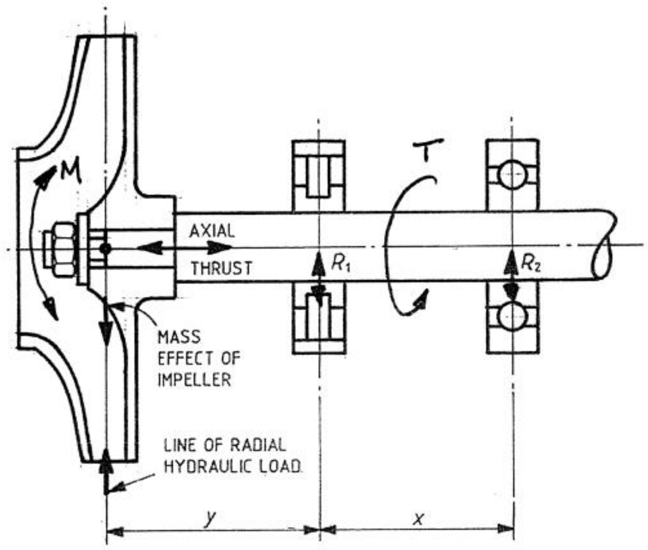


Figure 38

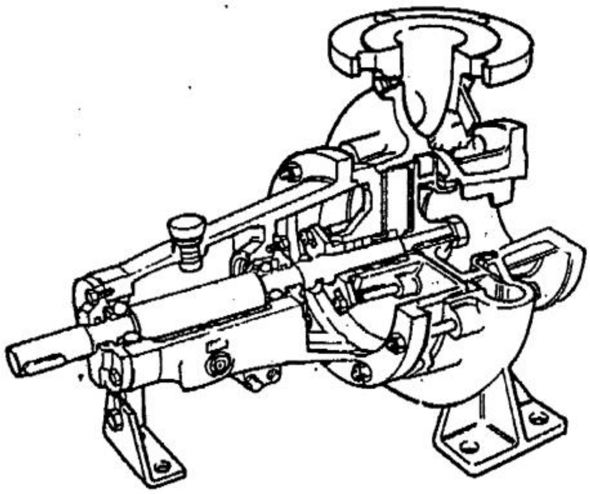
REFERENCES

1. Pfleiderer, C., Die Kreiselpump, 1961, Springer Verlag
2. Stanitz, J.D., 'Some theoretical aerodynamic investigations of impellers in radial and mixed flow machines', Trans, ASME, 1952:74: May pp 473-97
3. Karrasik I.., Pump Handbook, 1976, McGraw Hill, New York
4. Wiesner, F.J., 'A review of slip factors for centrifugal impellers', ASME, J. Eng for Power, 1967, Oct pp 558
5. Turton, R.K., Rotodynamic Pump Design, Cambridge University Press, 1994
6. Worster, R.C., 'The flow in volutes and its effects on centrifugal pump performance', Proc I Mech E, 1963
7. Summers-Smith, J.D., Mechanical Seal Practice for Improved Performance, I Mech E Guide, Mechanical Publications Ltd, 1988
8. Summers-Smith, JD, An Introductory Guide to Industrial tribology, Mechanical Engineering Publications, 1994
9. Stepannof, I, Centrifugal and axial flow pumps, 1976, John Wiley, New York

ROTODYNAMIC PUMP DESIGN

1. ROTODYNAMIC PUMP TYPE AND CONSTRUCTION

The following is a brief review of pump types commonly found with a description of their leading features.



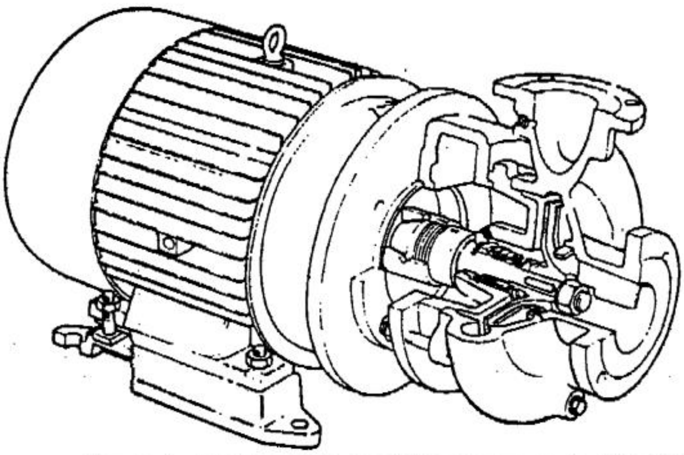


Figure 1 - End suction centrifugal pumps to ISO 2858

The above pumps are used for basic water circulation duties and are available in a "back pull out" design where the rotating assembly is removed from the drive side for maintenance. The pumps are made to a series of standard dimensions and should be interchangeable between manufacturers. There are other standards available extending the pressure range and chemical compatibility and including American dimensions. The above diagrams show long and close coupled options and illustrate the use of packed gland and mechanical seals respectively.

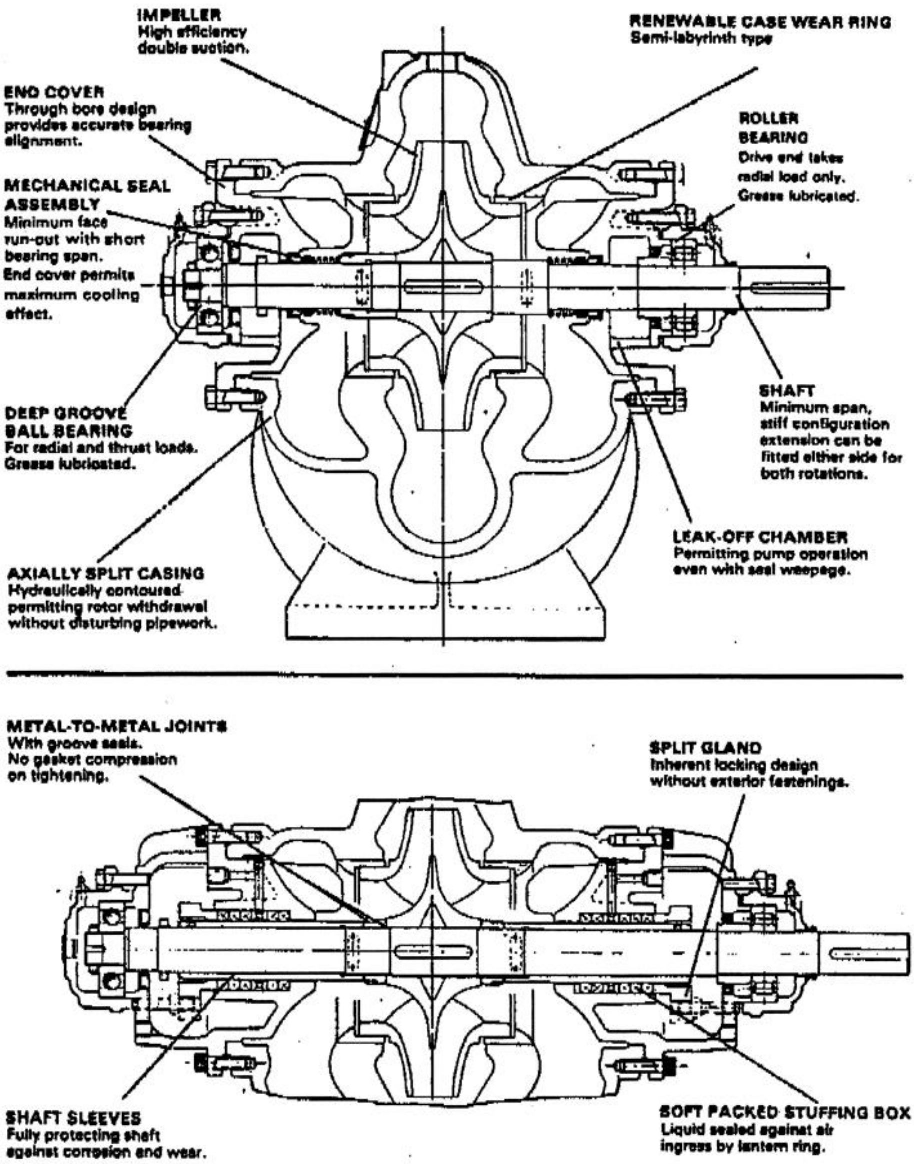


Figure 2- Cross sections through double entry split case pumps.

Double entry split case pumps are another widely used geometry of pump. The use of two entry paths to the impeller virtually eliminates end thrust and also improves NPSH performance. The unit has the added advantage that removal of the top casing half exposes the entire rotating assembly which can be removed for maintenance. Once again, mechanical seal and soft packed gland options are shown. Notice the shorter shaft span required by mechanical seals. Also note that packed glands require a lantern ring and flushing water supply from the pump volute to lubricate the gland packing.

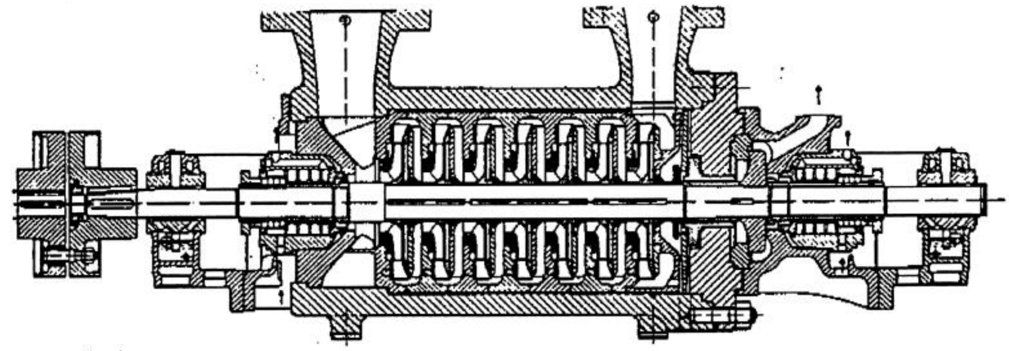


Figure 3 - Barrel type diffuser ring casing pump.

A single stage pump is limited in maximum speed (due to centrifugal stress limits on the impeller) and hence in maximum head that can be generated. Therefore, in order to obtain higher heads, pumps are multi staged. The above figure shows such a pump having a diffuser style casing. The diffuser casing comprises a number of small throats each working as a small "volute". This type of pump would be applied to high lift clean water duties. On lower head rises, the outer barrel casing may be dispensed with and the diffusers would be retained with tie rods along the length of the stack. Axial thrust becomes an important issue with multiple stages, and in the example shown thrust is balanced by a balance disk.

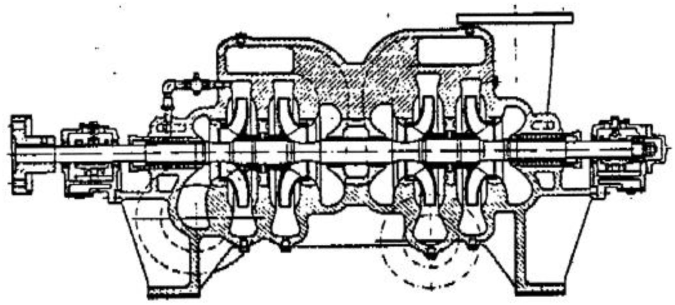


Figure 4 - Multi stage axially split volute casing with back to back impellers.

The above illustration shows an alternative approach to multi staging. Here the impellers are placed "back to back" to balance thrusts and a volute style axially split casing is used. The disadvantage of this casing is the long shaft span, presence of radial loads and a complex casing.
The process and power generation industries have moved toward faster running, smaller pumps with fewer stages for a given head rise.

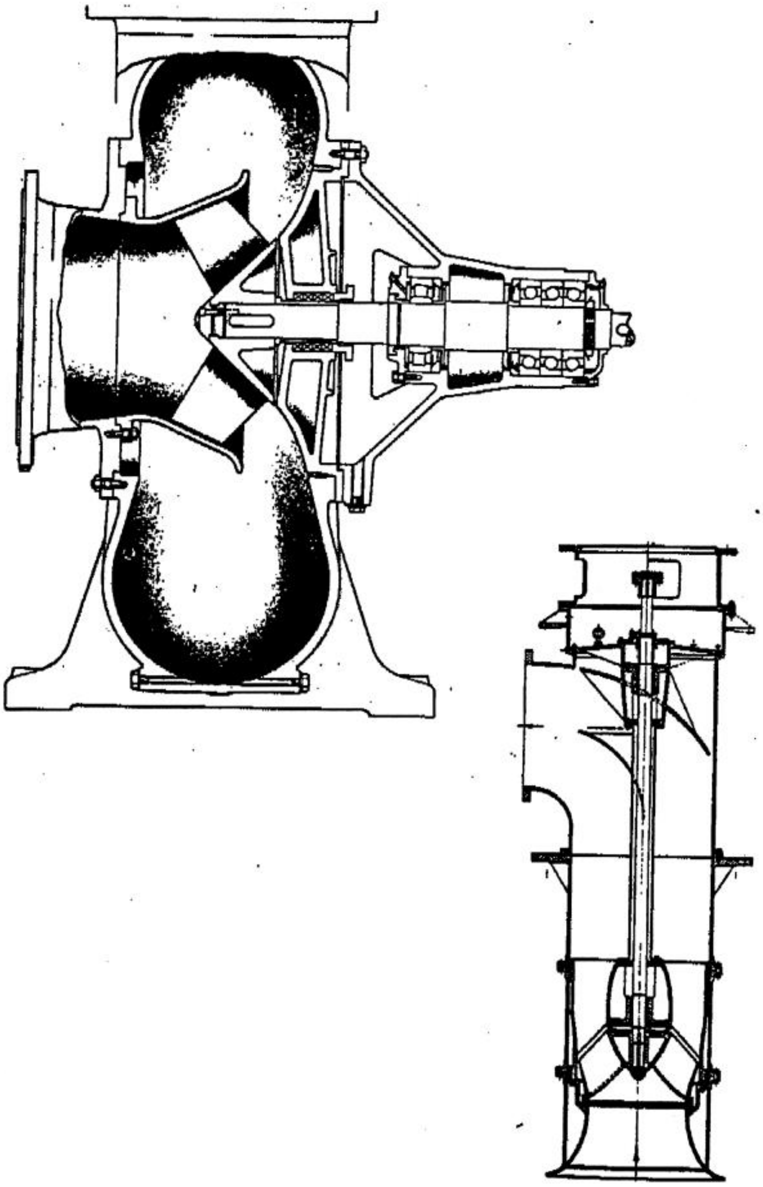


Figure 5 - Mixed flow pumps of diffuser and bowl consttruction.

The previous examples have all been of pumps of relatively low type number. The above two examples are both of medium type number and have similar impeller geometries. However, the outlet element is different being volute in the horizontal mounted machine and a diffuser bowl in the vertical one. Note that provision of bearings to resist the axial thrust becomes an important issue in this range of type number. Typical applications for this type of pump would include low lift sewage, activated sludge or similar for the volute machine and low lift raw water, filter backwash or final effluent for the diffuser bowl unit.

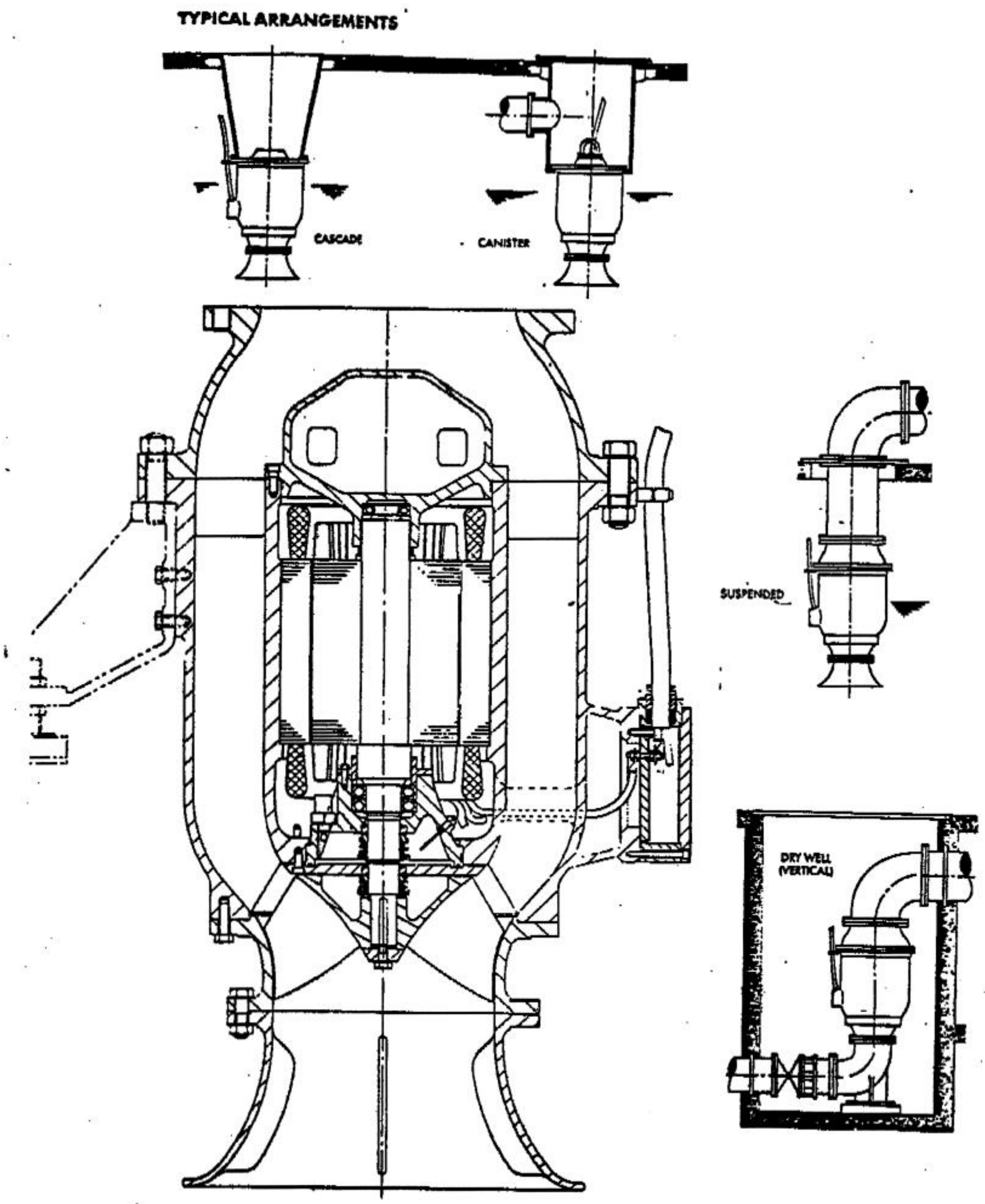


Figure 6 - Mixed flow diffuser bowl pump with submersible motor.

A more modern arrangement of the diffuser bowl pump is shown above. Hydraulically, this is identical to the previous diffuser bowl pump, but the drive is now from a submersible motor which is supported by the diffuser blades and includes the pump bearings within the motor housing. The motor internals are air filled, the pumped medium being kept out by twin mechanical seals and an oil bath. Bearings are of the rolling element type, grease lubricated. The adoption of this technology allows a number of installation options which are also shown above.

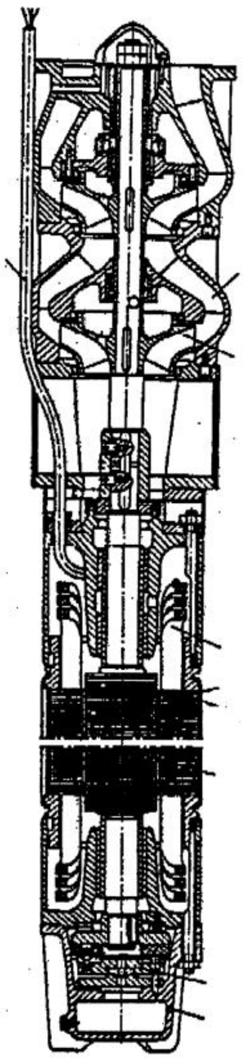


Figure 7 - Vertical multi stage pumps for boreholes.

The mixed flow – diffuser bowl style of pump can be multi staged to produce a pump of small external diameter suitable for fitting down a borehole for ground water abstraction. The diagram on the left shows an older style of shaft driven unit while that on the right shows a submersible motor driven unit. These pumps may have up to 20 or more stages depending on head rise required. This style of submersible motor is either water or oil filled. The motor incorporates substantial sleeve bearings and a plain or tilting pad thrust bearing at the bottom.

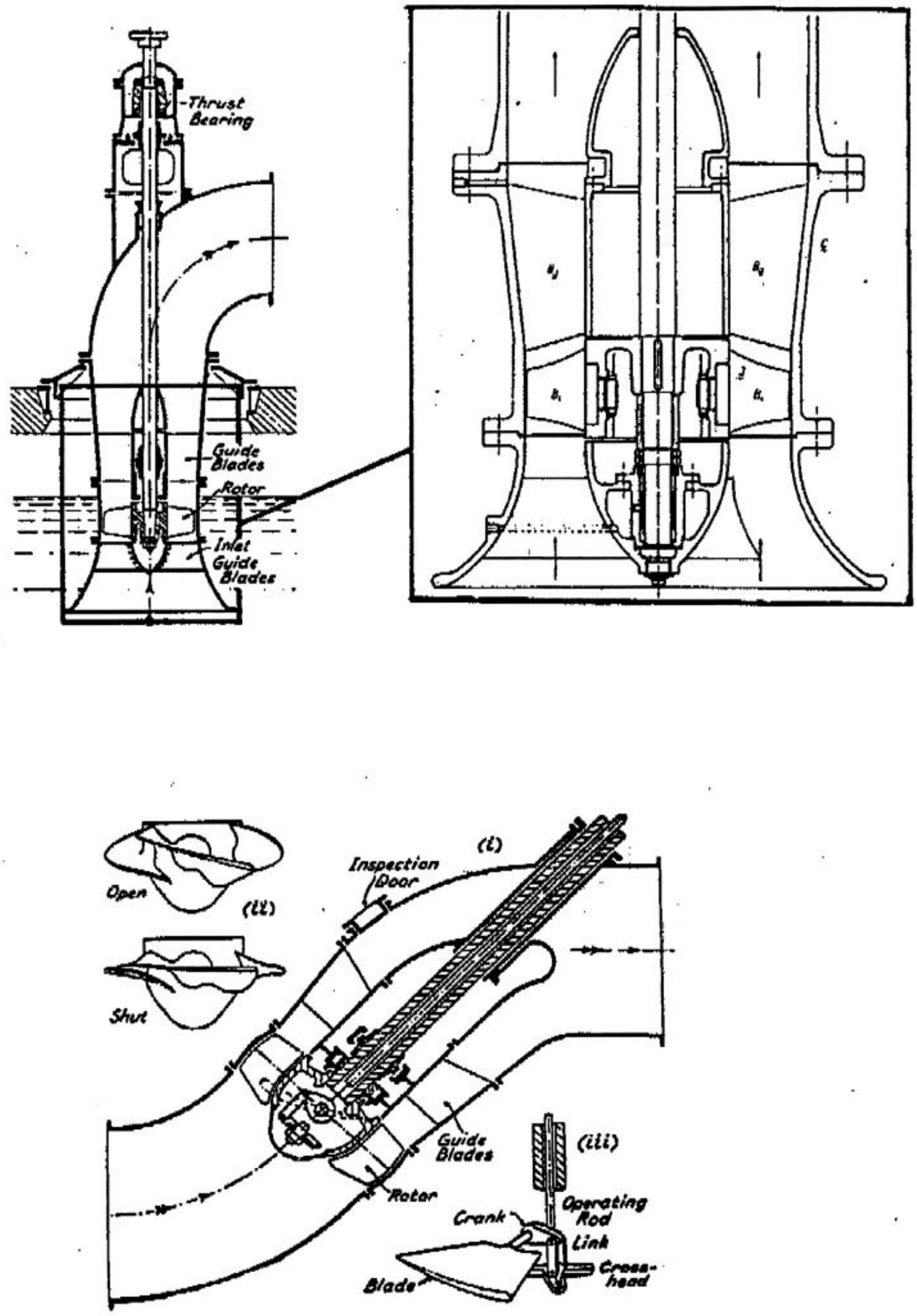


Figure 8 - Axial flow pumps

The axial flow pump gives a further increase in the type number and is available in the typical formats shown above. The submersible motor option is also available. The lower diagram shows a variable pitch impeller which allows some variation of delivered flow and can also ease the possible starting problems associated with this type of pump.

This class of pump is used for high volume, low lift duties such as raw water abstraction, final effluent lifting, land drainage and storm water disposal. Axial flow pumps are not suitable for liquids with significant quantities of rags.

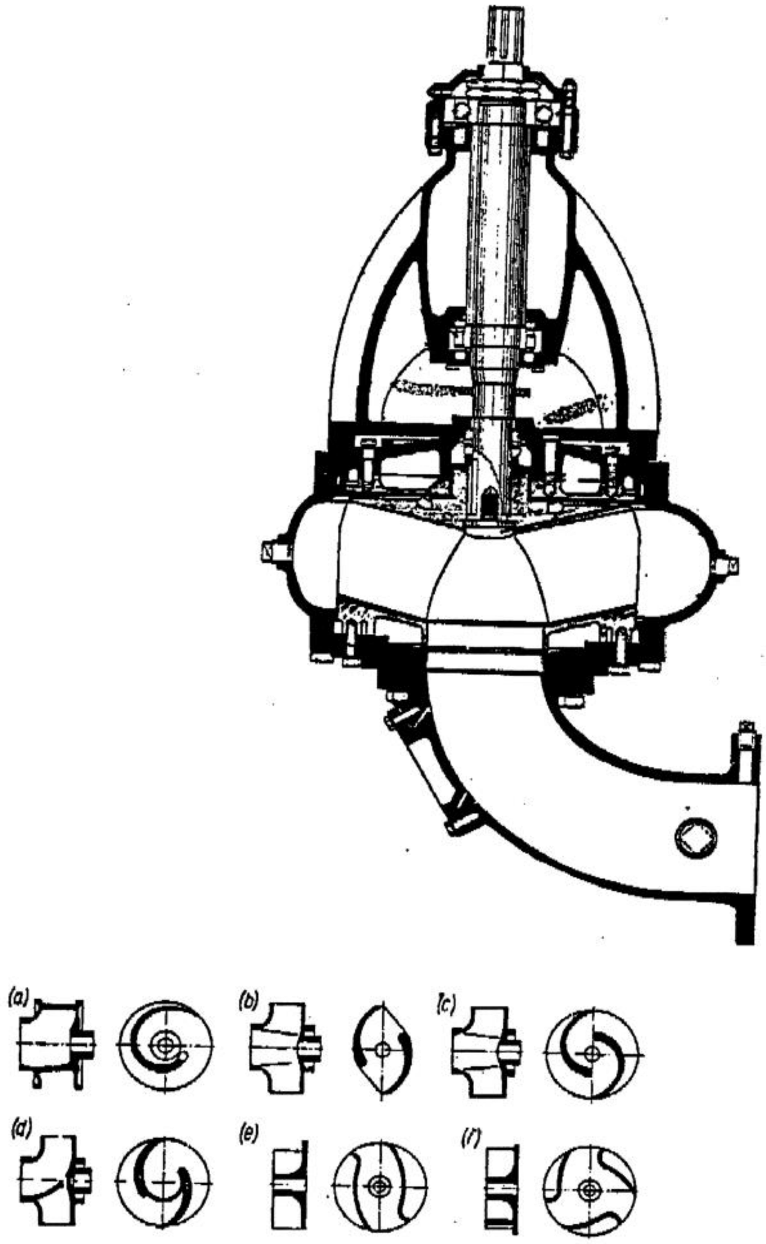


Figure 9 - Sewage pump

All the foregoing examples have been pumps for reasonably clean liquids. For handling heavily polluted waters such as sewage and sewage sludge, the design is modified somewhat. The hydraulic end has an impeller of broad outlet width and a few thick blades. Various designs of "non clog" impeller are available, in both shrouded and semi shrouded format but none are absolutely choke free. Mechanical design aspects include a hand hole for unblocking, replaceable wear plates, special seals or flushed glands arrangements, substantial shaft and bearings. The above example is shown as a vertical shaft type, but similar units are available for horizontal mounting.

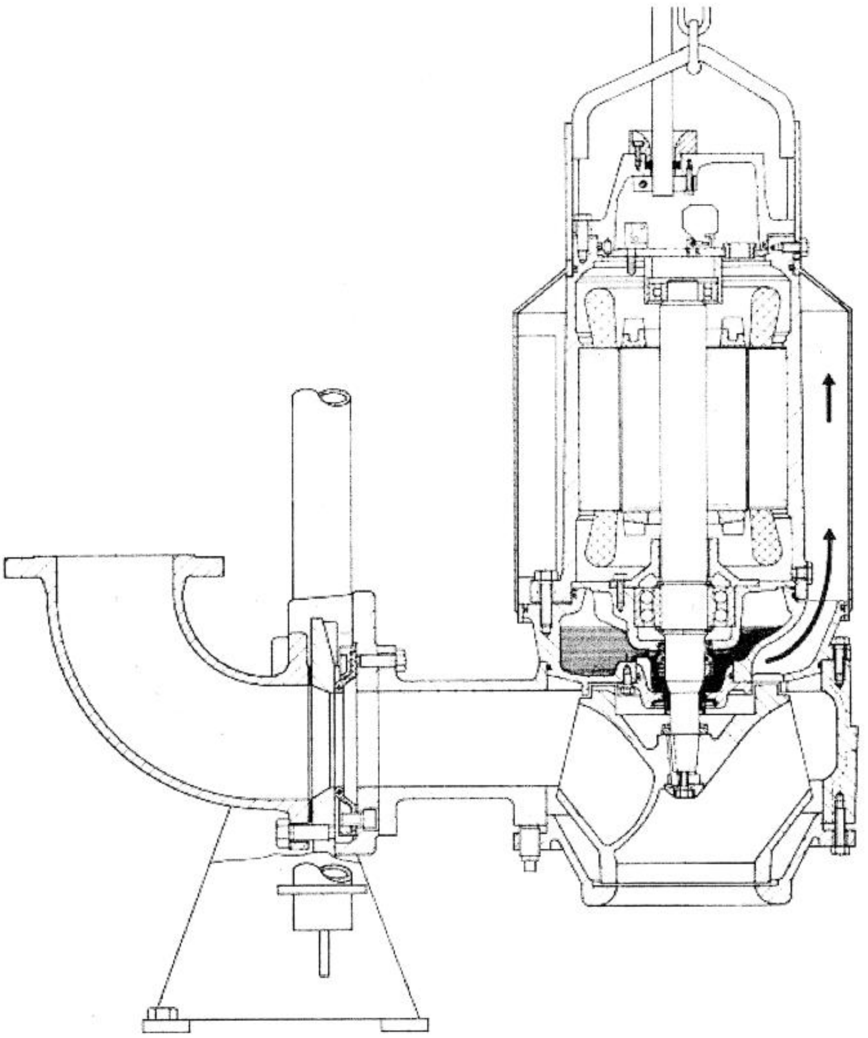


Figure 10 - Submersible sewage pump

The dry well style of sewage pump illustrated in the previous figure has been almost completely overtaken by the submersible style for pumps up to around 250 mm branch size and in a lot of cases for much larger pumps. The above figure shows a typical arrangement which includes a quick release duckfoot bend allowing the complete pump and motor to be hoisted out of the sump. The motor is an air filled unit and has grease lubricated rolling element bearings. There are twin seals with an oil filled buffer space between. Some pumps include a water filled cooling space around the motor, allowing the motor casing to be clear of the pumped medium without overheating problems.

2. DESIGN OF DRIVES

Pumps in the water treatment industry are often large, built into a civil work structure and incorporate long drive shafts, flywheels or other accessories to minimise flooding or to alleviate pressure surge problems.

The design of these associated structures must take account of both steady and fluctuating loads to ensure the structures are free from vibration. This can be a complex task and includes design of items such as:

- Floors supporting pumps
- Floors supporting motors
- Flywheels
- Vertical Drive shafts (Hardy Spicer style)
- Enclosed vertical drive shafts ("Tunnel tubing")
- Pipework systems
- Pipework restraints, thrust blocks etc.
- Access platforms & walkways

3. ROTODYNAMIC PUMP CHARACTERISTICS

Most manufacturers literature includes a range chart (sometimes known as "tombstone chart") to aid the customer in assessing what pumps in the range might be suitable for a given application. However, these really only show broad areas of application and a more detailed assessment of the suitability of the pump for a given system must be made from the performance curve for the particular pump model.

Pump characteristics are not usually complicated, being perhaps four graphs plotted against flow (for rotodynamic machines) or pressure (for most PD machines). However, experience shows that most pump application problems start with a failure to fully investigate the interaction of these parameters with the system curve.

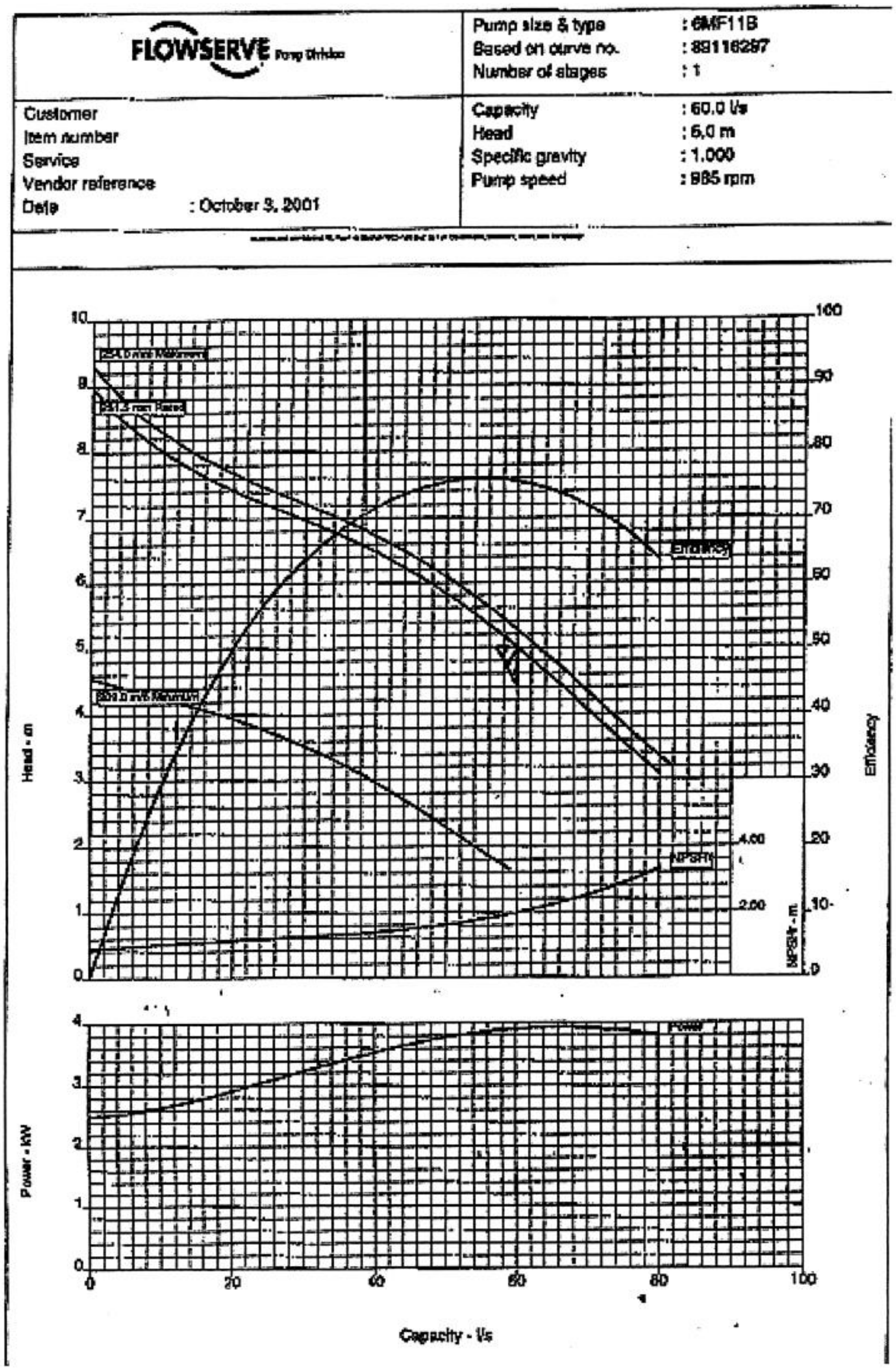


Figure 11 - Typical rotodynamic pump curve

The above figure shows a typical rotodynamic pump curve for a large pump of medium type number. The presentation shows three pump speeds and presents pump efficiency as iso-efficiency lines and as separate curves. Rotodynamic pump curves would not usually be issued with both presentations of efficiency. A similar presentation (and pattern of performance) would be shown for multiple impeller diameters, for example.

The following examples will highlight some features of pump curves to be aware of:

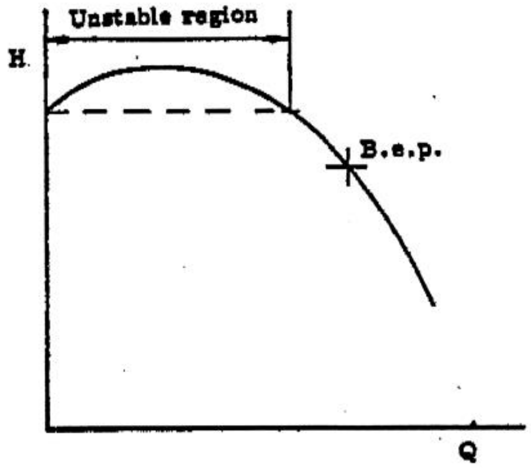


Figure 12 - Head – Flow Curve Instability at Low Flows

This type of instability might be expected with pumps having a high head ratio which tend to be smaller and cheaper pumps. Pumps designed to ISO 2858 dimensions and some solids handling pumps exhibit this problem. Many purchaser specifications exclude this type of characteristic from use on multiple parallel pump systems although the pumps can be used with care in the design and selection of the system. The potential danger is that the operating point will drift into the unstable region marked above making flow heavily dependant on minor changes in system head. In extreme cases, the system static head can exceed the zero flow head of the pump.

The amount of instability (the degree to which the curve curves over at low flows) can be heavily dependant on pump inlet conditions. Systems that allow significant inlet swirl will exacerbate the situation (Refer to lecture on sumps and intakes for a discussion on inlet swirl).

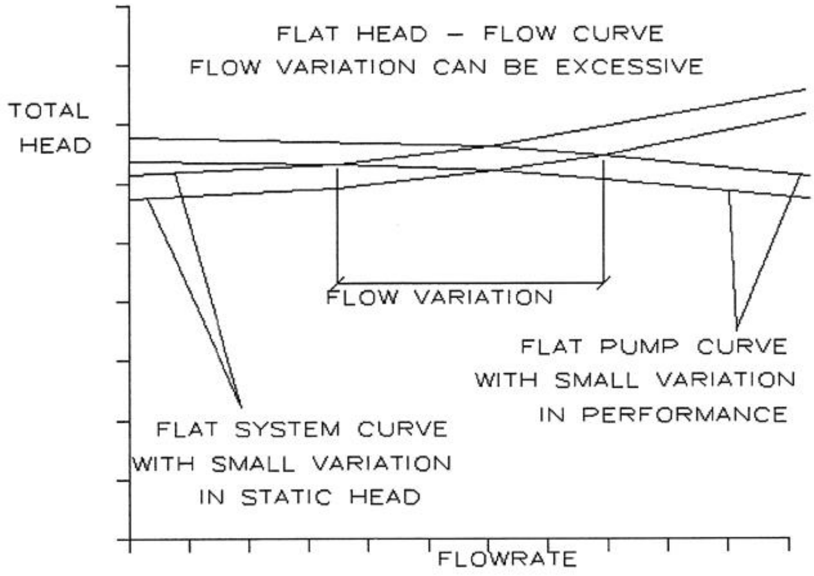


Figure 13 - Combination of flat pump H-Q curve and flat system curve

The above figure shows a stable (but very flat) pump curve combined with a very flat (high proportion of static head) system curve. This combination gives a large change in flow rate for small changes in the pump curve or system curve. The pump curve could change due to wear, manufacturing and testing tolerances, for example. The system curve could change due to errors in estimating system resistance or due to sump level changes, for example. However, the resulting change in flow from such small errors is obviously unacceptable.

Pumps with very flat H-Q characteristics will have been designed to provide the maximum performance from a given impeller diameter and hence cost (as for the previous unstable curve) or the design may be restricted by solids handling size criteria.

Some specifications will place limits on the head rise to closed valve from the operating point in an effort to exclude such pumps. However, they may be quite acceptable on a steeper system curve.

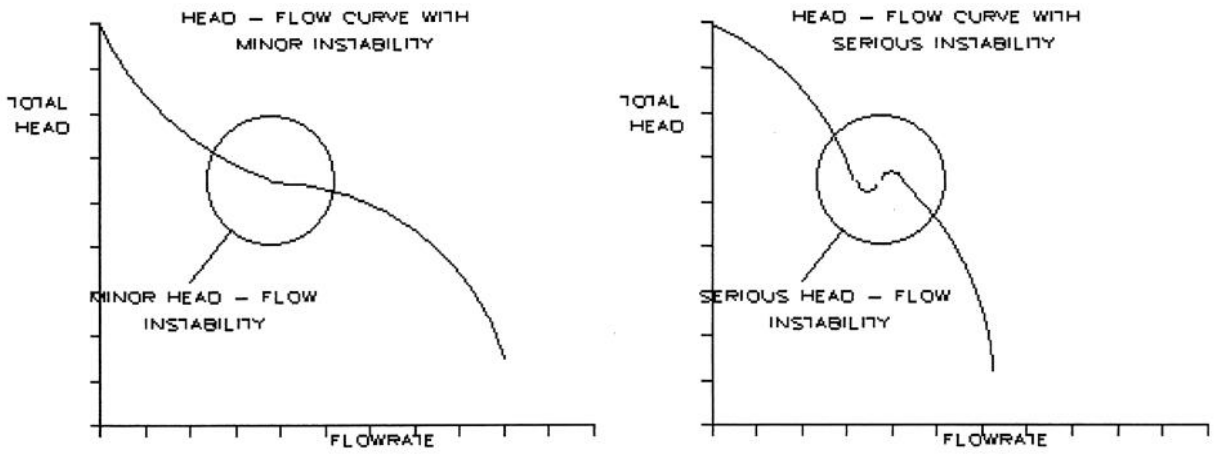


Figure 14 - Mid range instability of medium to high type number pump.

The above instability is shown to some extent by all pumps of medium to high type number and becomes more severe as type number increases. It is caused by changes in the flow pattern internal to the impeller (in essence, a stalling effect of the impeller blades). Operation within the unstable region is likely to cause noise, vibration and erratic pump behaviour and should be avoided.

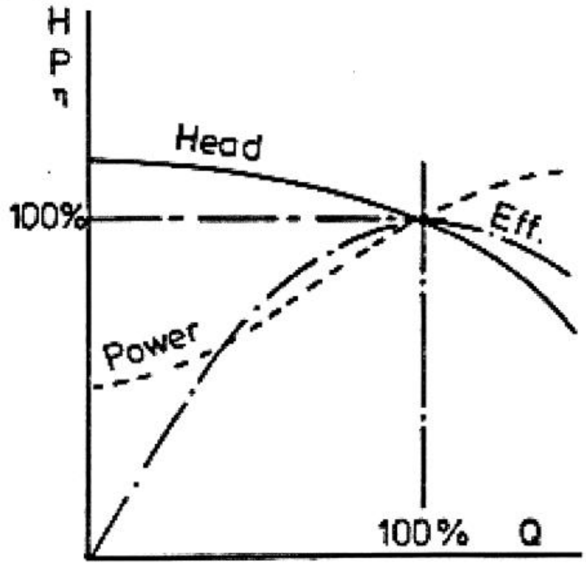


Figure 15 - Radial Flow, Overloading Power Curve

This and the next three examples are normalised around best efficiency conditions and show patterns of the power consumption curve. The above figure shows an overloading power curve which is typical for most pumps of low to medium type number and high head coefficient (large performance for a given impeller diameter). Always ensure that the available drive has sufficient power to operate at the MAXIMUM flow condition.

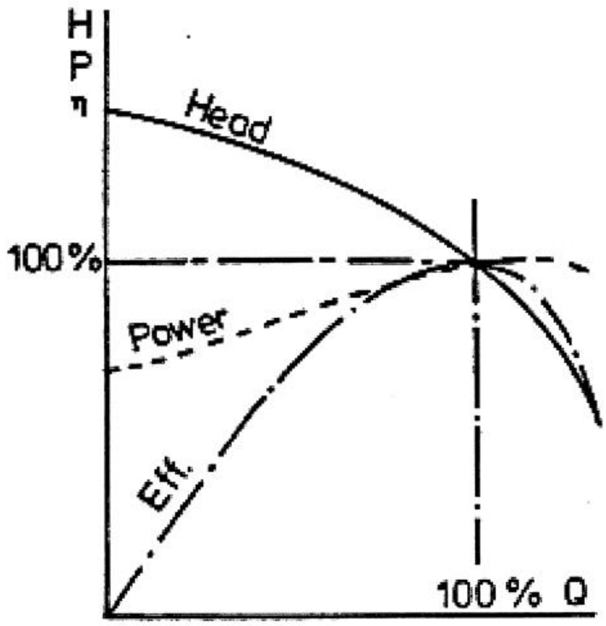


Figure 16 - Radial Flow, Non-overloading Power Curve

The above type of performance can be designed into pumps of reasonably low type number at the expense of a larger impeller (hence more cost) for a given duty. Power consumption reaches a maximum at or just above the best efficiency flow making the selection of a driver relatively simple.

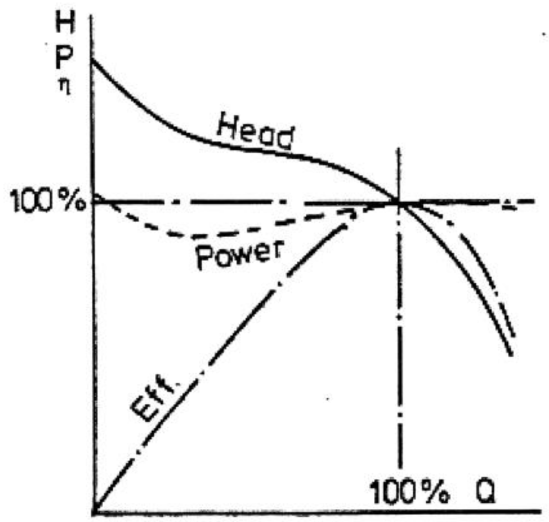


Figure 17 - Mixed Flow, Non Overloading

The above characteristic would be typical for a medium type number mixed flow pump. The power reaches a maximum at around best efficiency flow. Power demand at low or zero flow is approximately equal to rated demand so pump starting arrangements can need some careful attention.

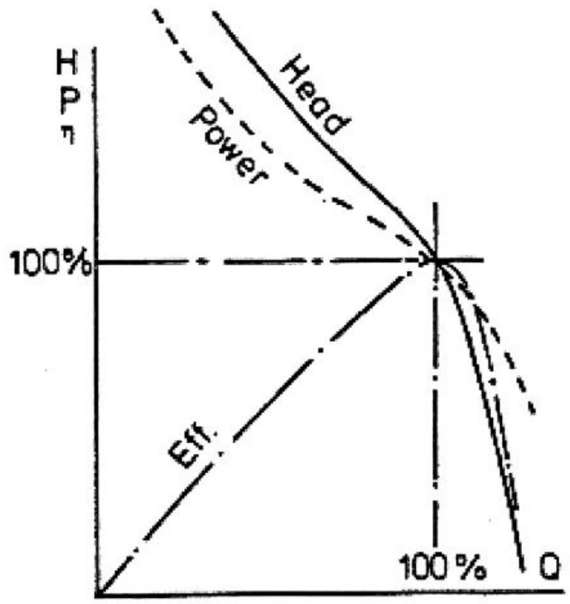


Figure 18 - Axial Flow – Falling Power Curve

The above characteristic is typical of high type number pumps. The highest power consumption occurs at the lowest flow. Selection of the driver must be based upon the LOWEST flow, HIGHEST head condition. Starting the pump will also require careful selection of starting method, to limit the power demand of the pump. This type of pump is unlikely to be capable of running at zero flow and the driver and shaft may suffer damage if this is attempted.

Pump Basics II

Cranfield UNIVERSITY

Mr Tony Salisbury

www.cranfield.ac.uk

Rotodynamic Pumps 2

Cranfield UNIVERSITY

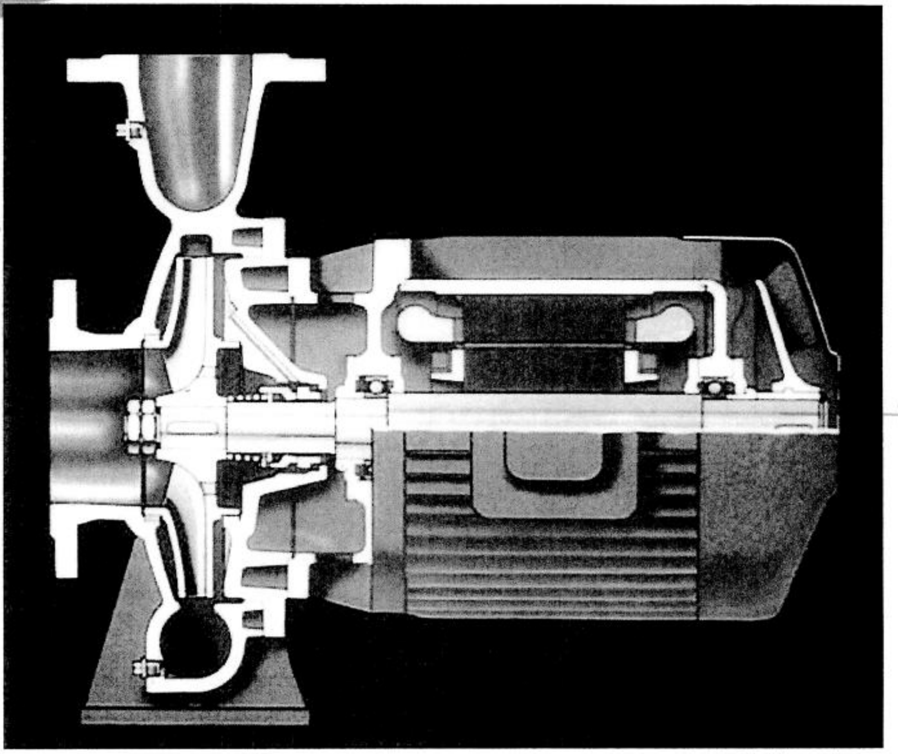
Presented by

Tony Salisbury

www.cranfield.ac.uk

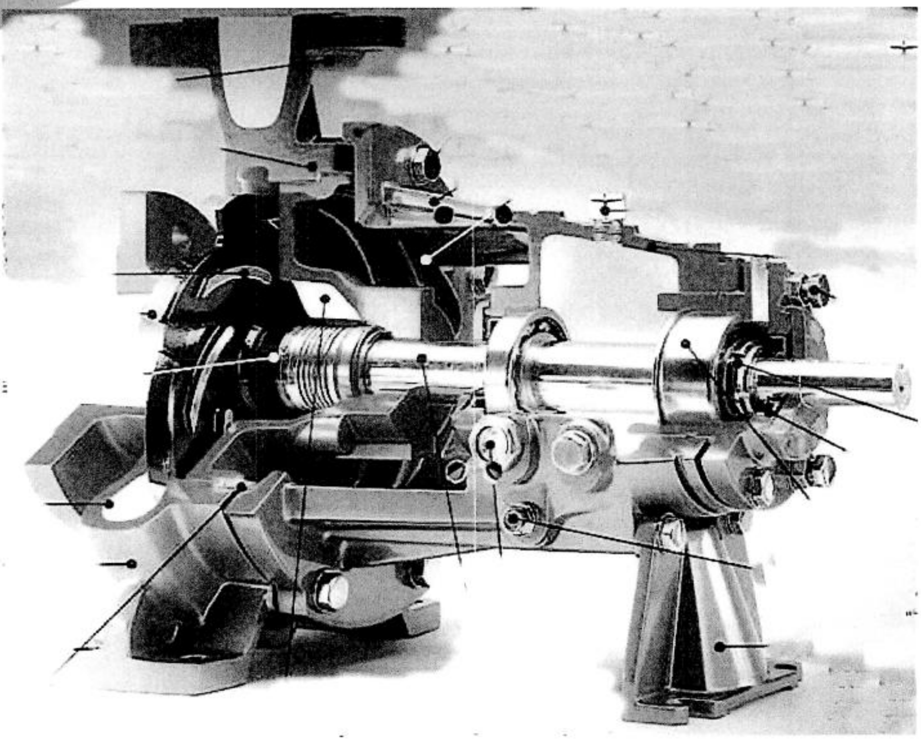
Cranfield UNIVERSITY
Process Systems Engineering

" SINGLE STAGE END
SUCTION BACK PULL-OUT PUMP"



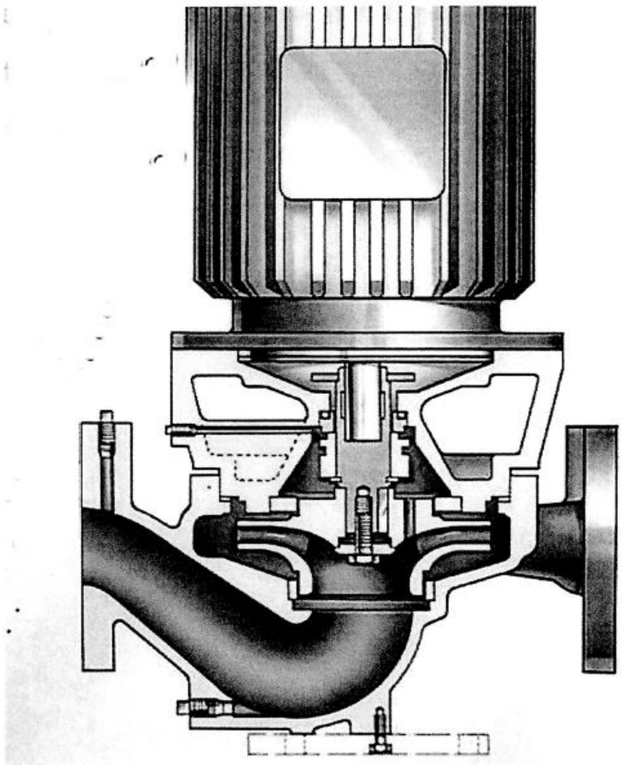
SHAFT

Cranfield UNIVERSITY
Process Systems Engineering



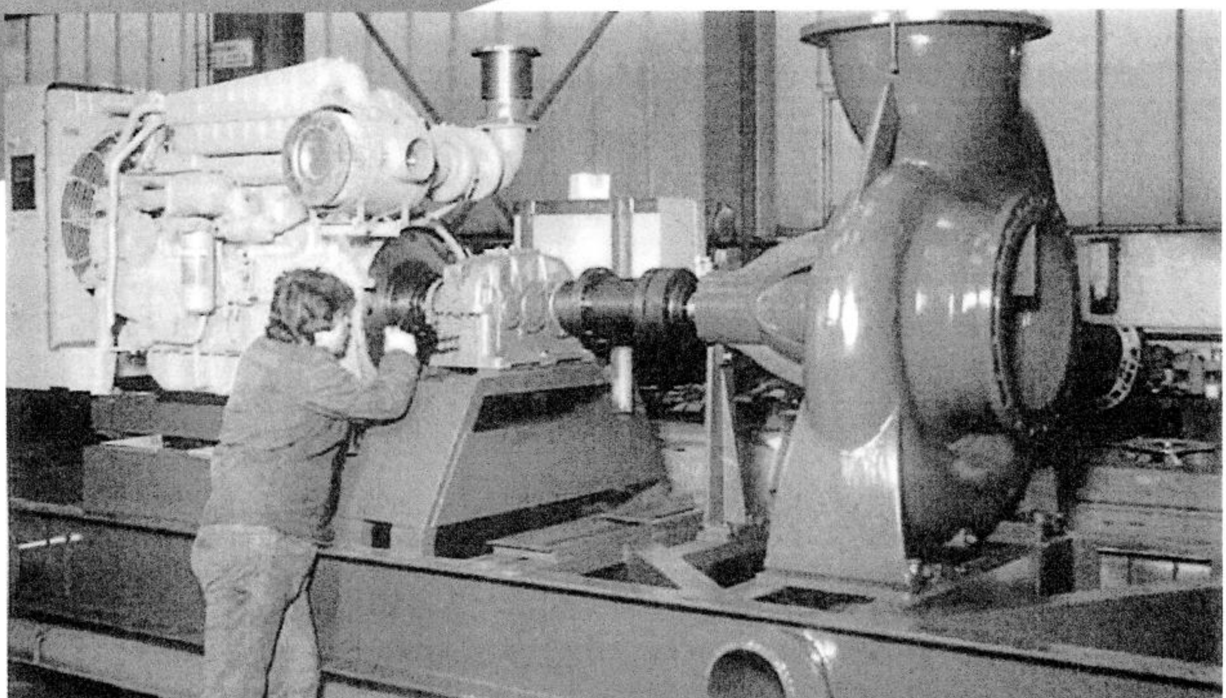
Cranfield UNIVERSITY

Process Systems Engineering



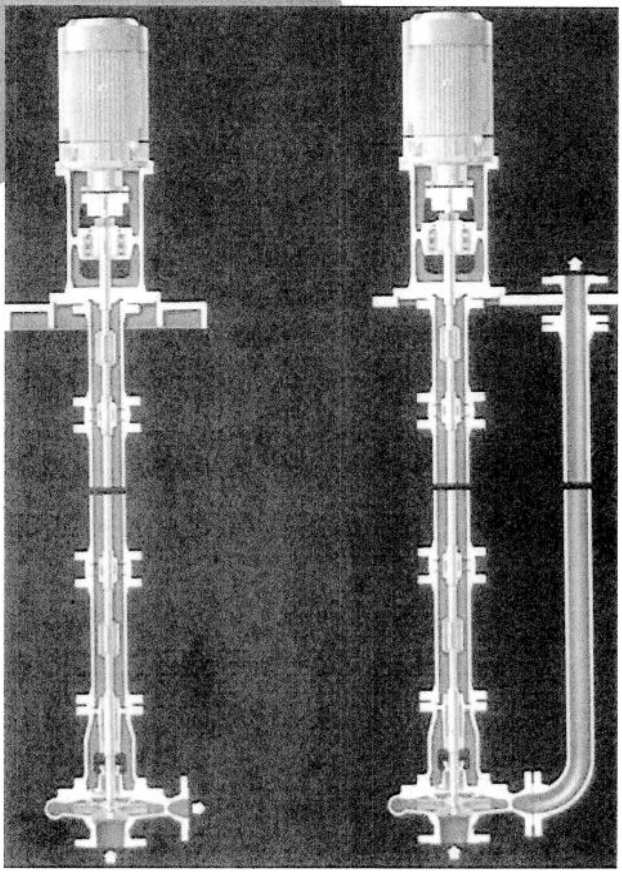
Cranfield UNIVERSITY

Process Systems Engineering



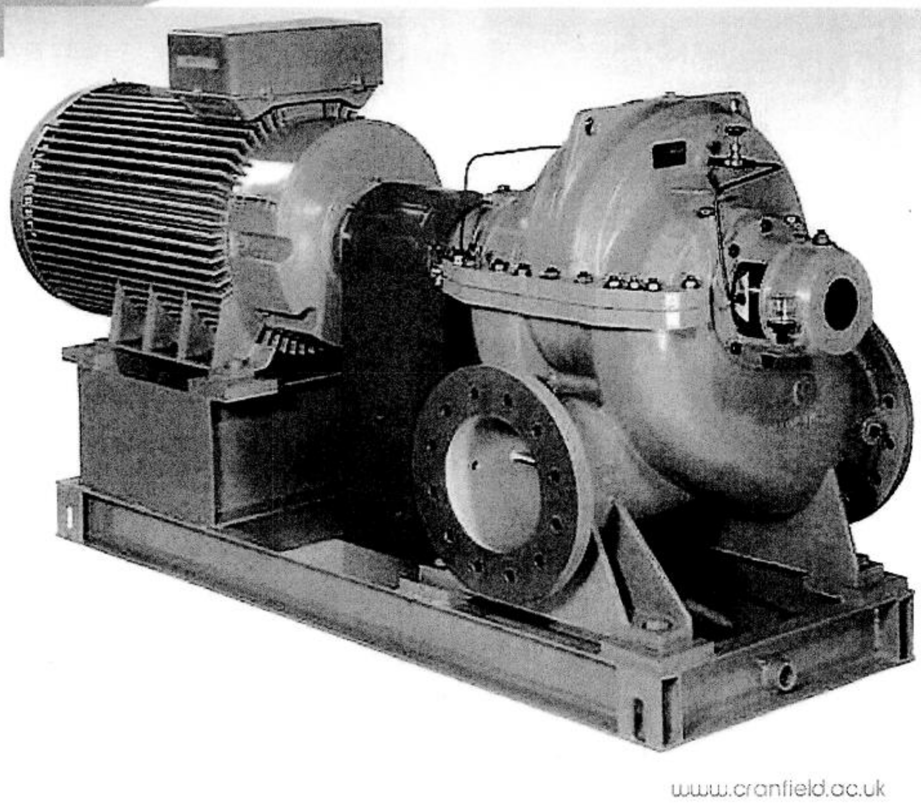
Cranfield UNIVERSITY

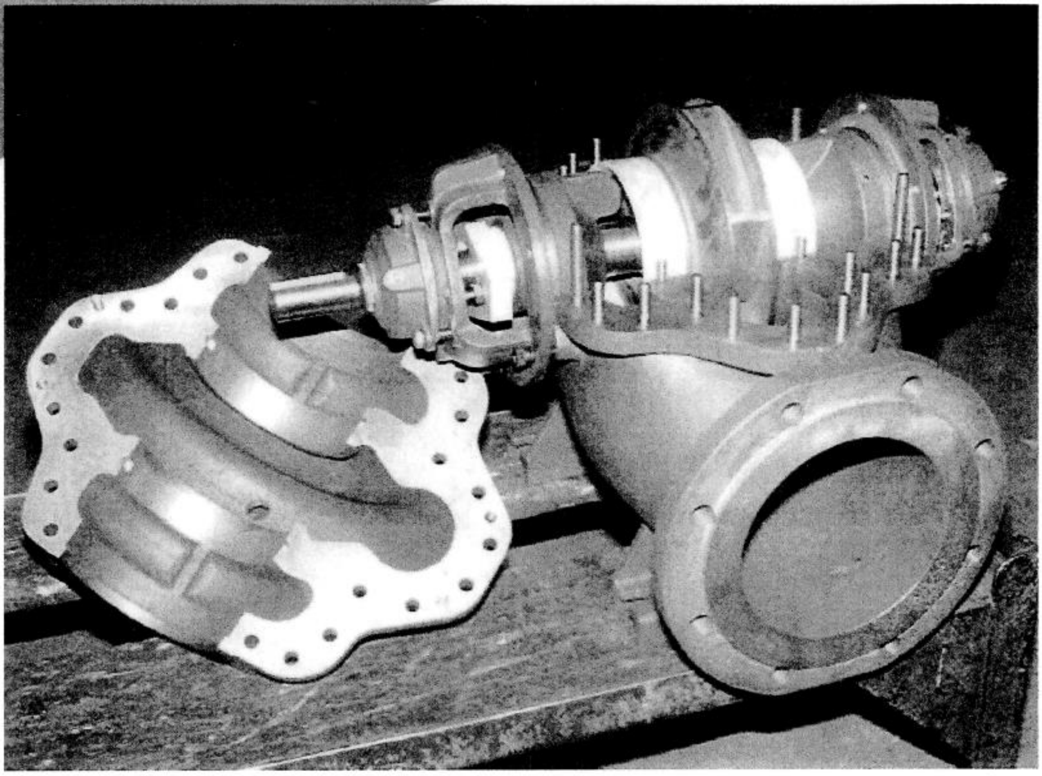
Process Systems Engineering



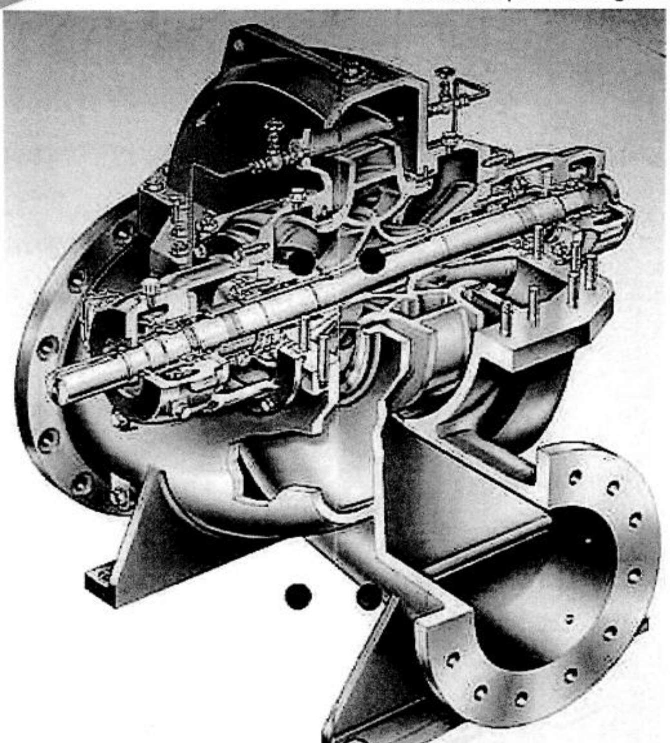
Cranfield UNIVERSITY

Process Systems Engineering



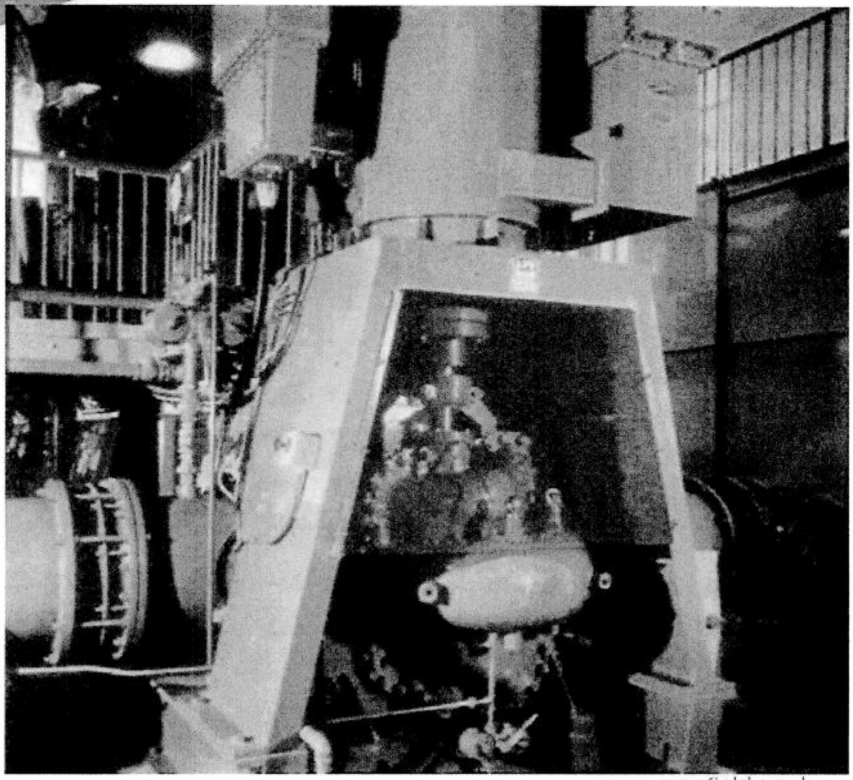


Cranfield UNIVERSITY
Process Systems Engineering



PSE 12-2008
www.cranfield.ac.uk

Cranfield UNIVERSITY
Process Systems Engineering

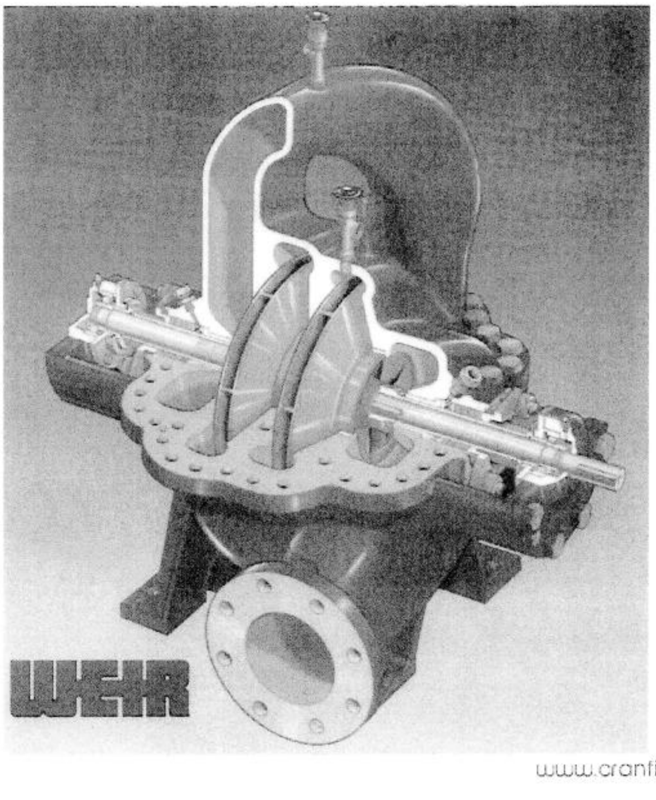


PSE 12-2008

www.cranfield.ac.uk

Two Stage Split Case Pump

Cranfield UNIVERSITY
Process Systems Engineering

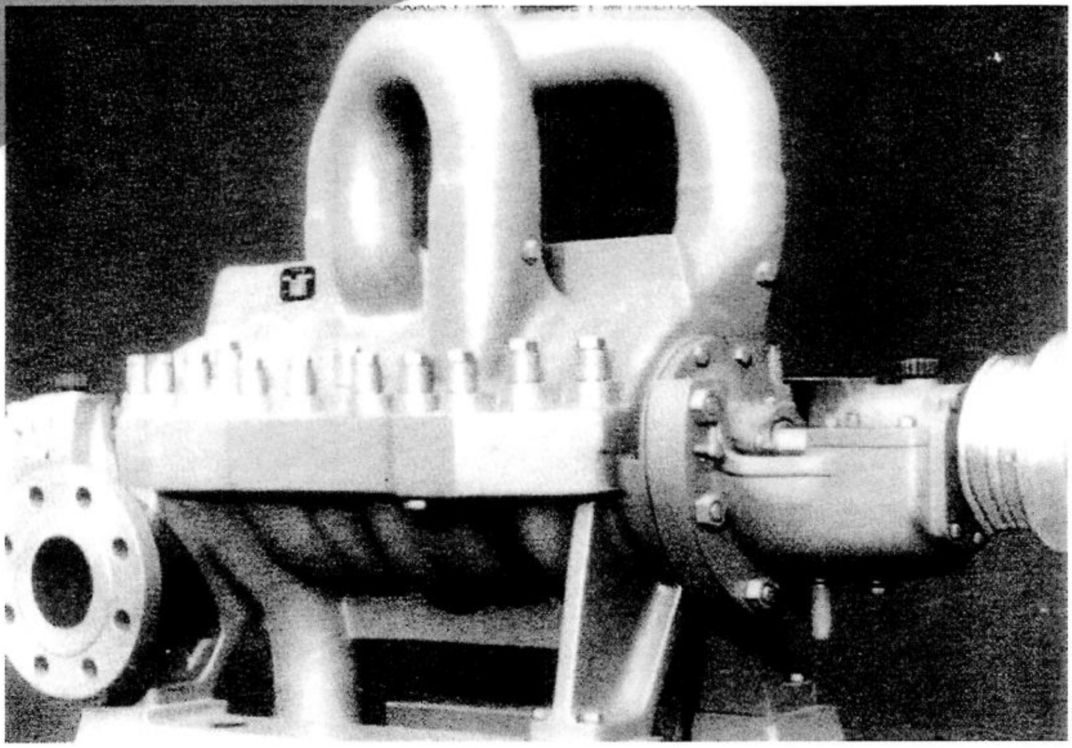


PSE 12-2008

www.cranfield.ac.uk

Cranfield UNIVERSITY

Process Systems Engineering

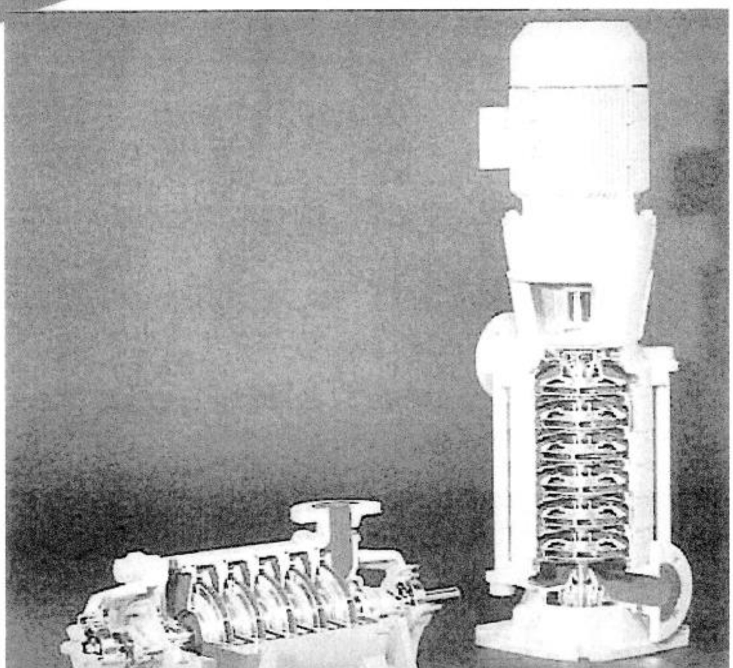


PSE 12-2008

www.cranfield.ac.uk

Cranfield UNIVERSITY

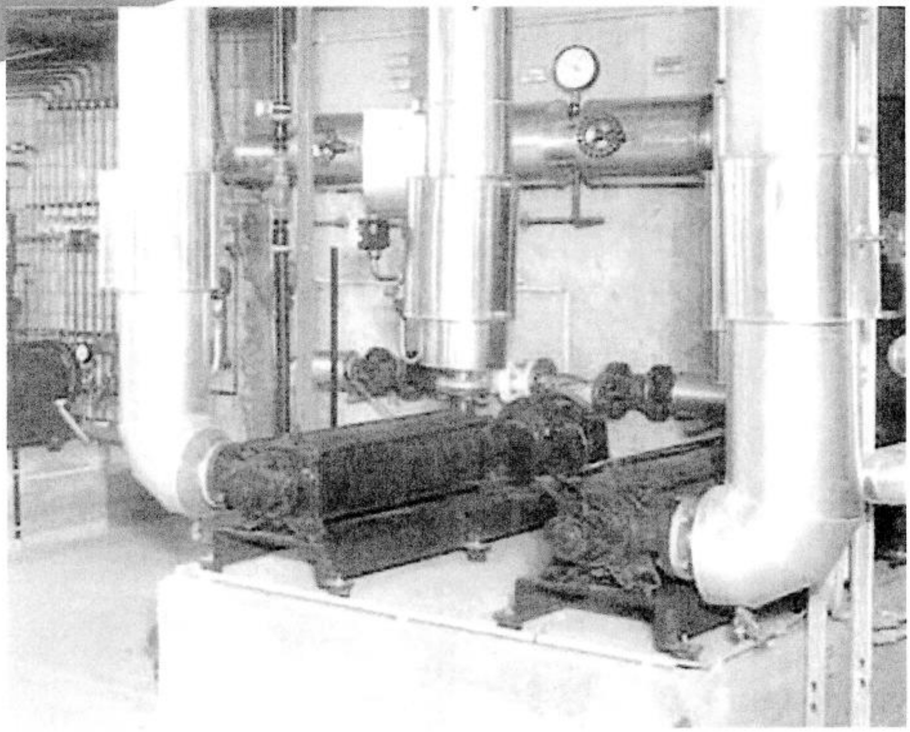
Process Systems Engineering



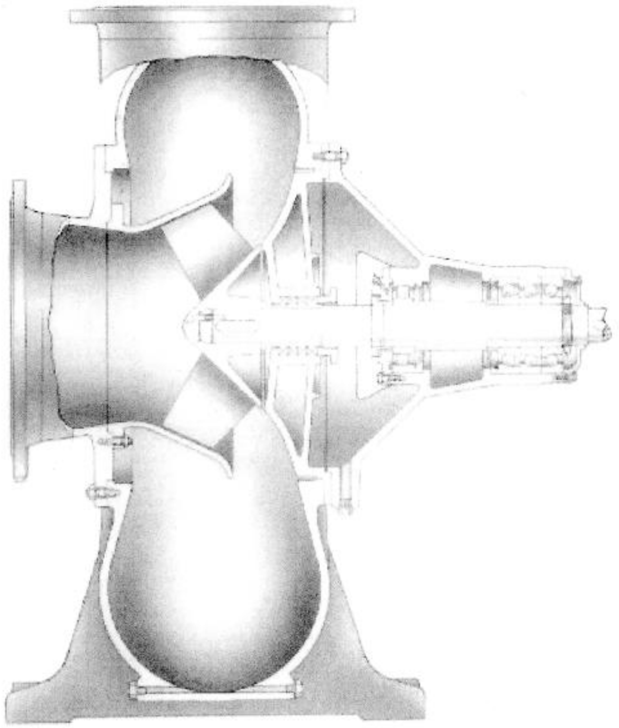
PSE 12-2008

www.cranfield.ac.uk

Cranfield UNIVERSITY
Process Systems Engineering



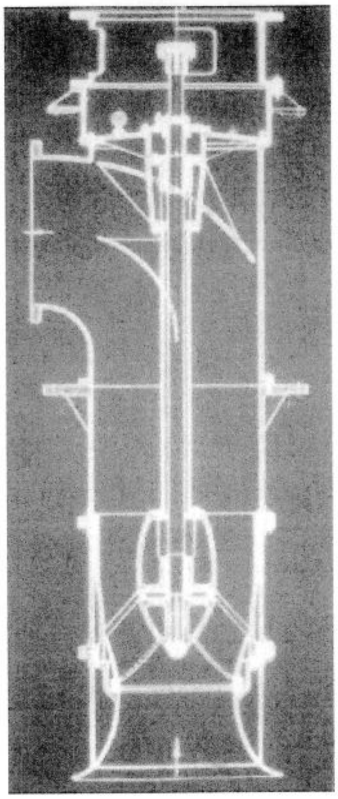
Cranfield UNIVERSITY
Process Systems Engineering



PSE 12-2008
www.cranfield.ac.uk

Cranfield UNIVERSITY

Process Systems Engineering

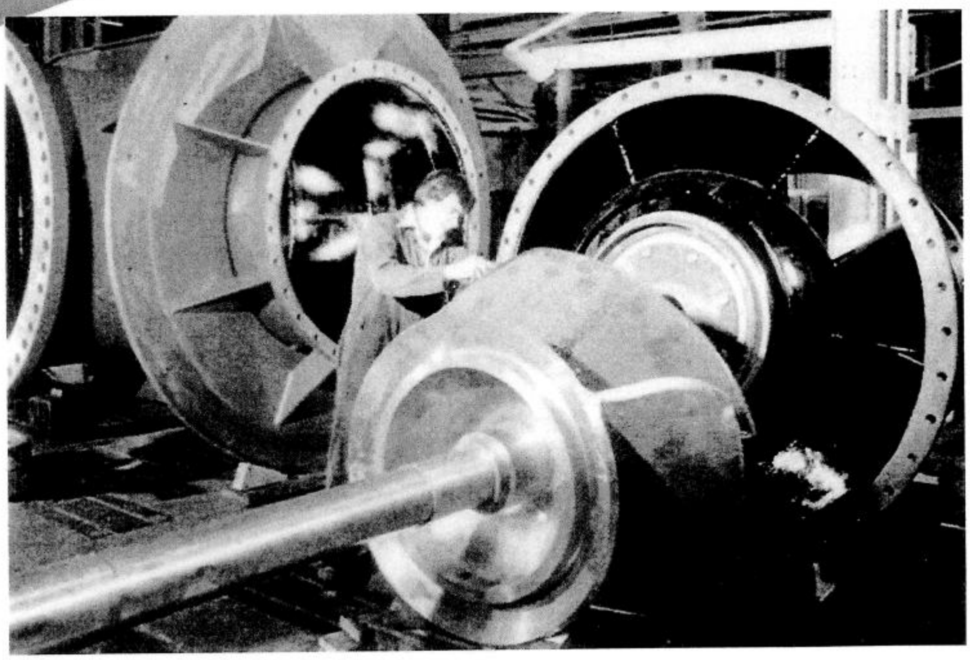


PSE 12-2008

www.cranfield.ac.uk

Cranfield UNIVERSITY

Process Systems Engineering



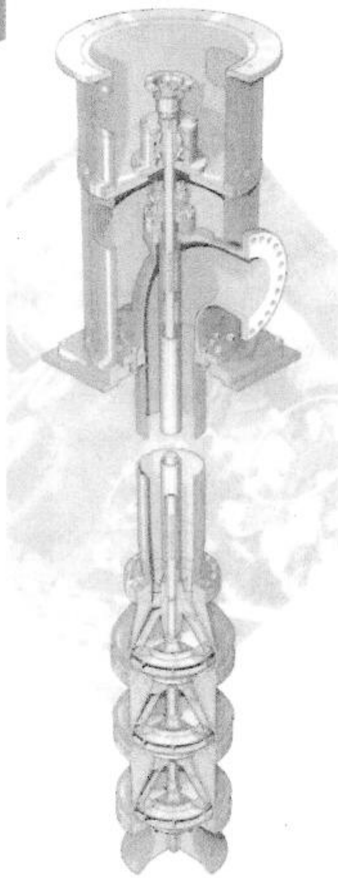
PSE 12-2008

www.cranfield.ac.uk

Vertical Mixed Flow Bowl (Shaft Drive)

Cranfield UNIVERSITY

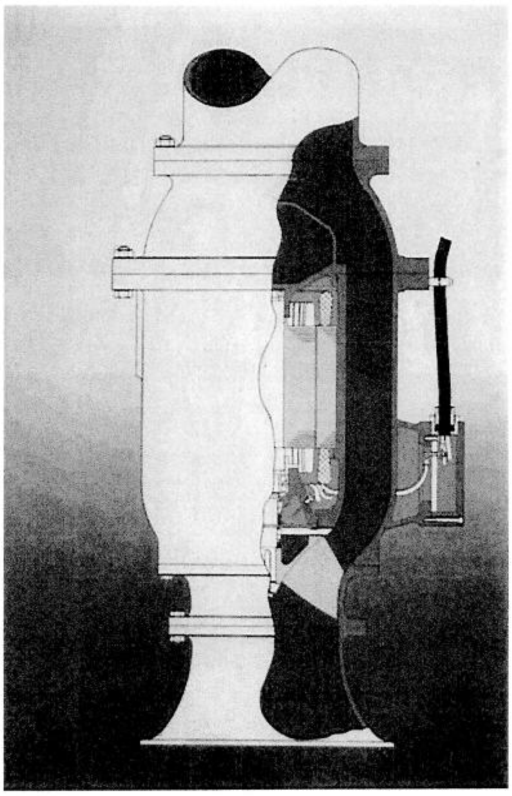
Process Systems Engineering



Vertical Mixed Flow Bowl (Submersible Drive)

Cranfield UNIVERSITY

Process Systems Engineering

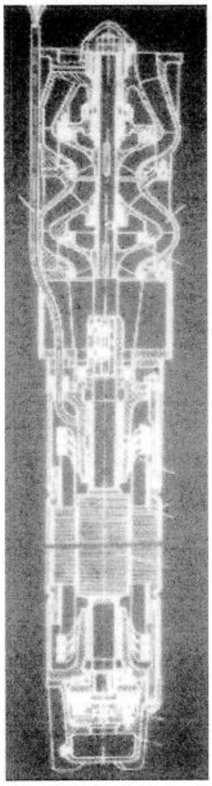


PSE 12-2008

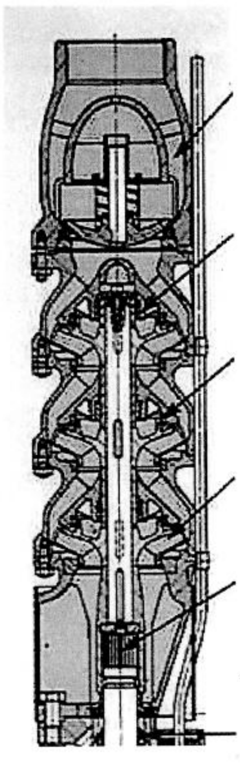
www.cranfield.ac.uk

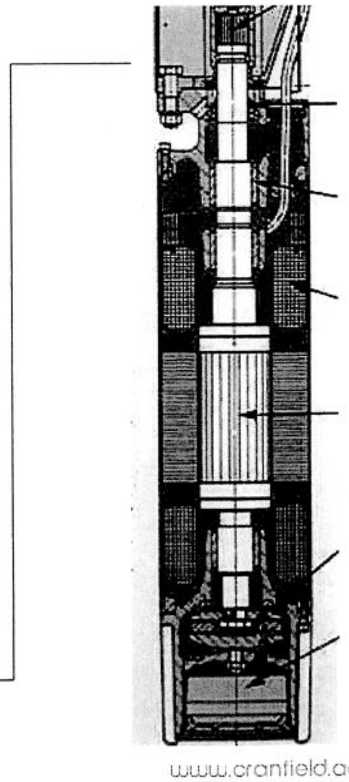
Borehole Pumps

Cranfield UNIVERSITY
Process Systems Engineering

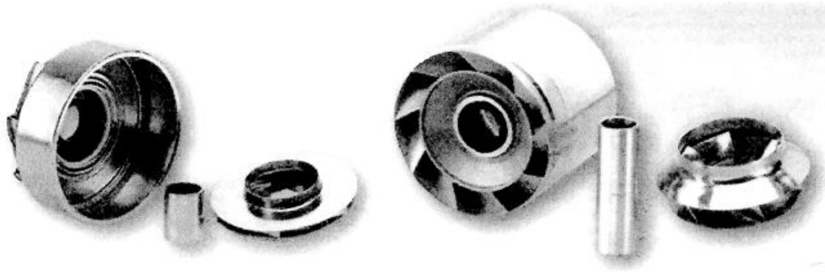


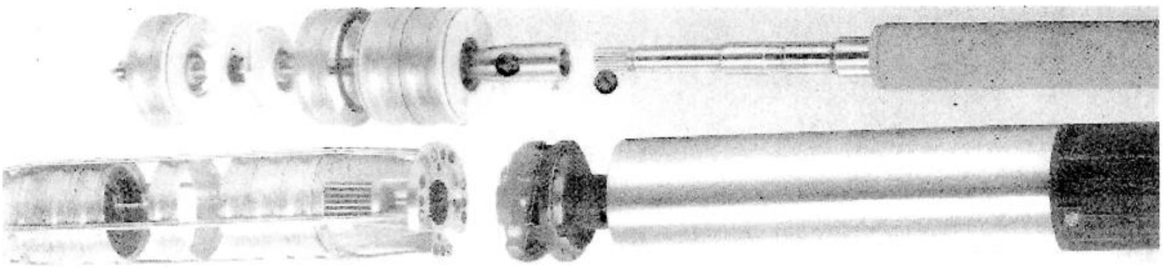
Cranfield UNIVERSITY
Process Systems Engineering



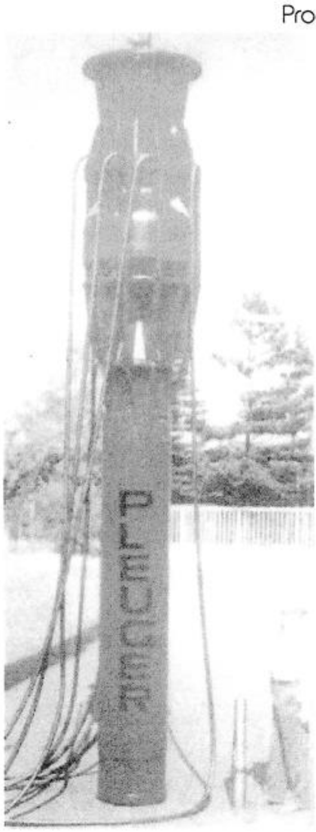


BOREHOLE PUMP COMPONENTS





Cranfield UNIVERSITY
Process Systems Engineering



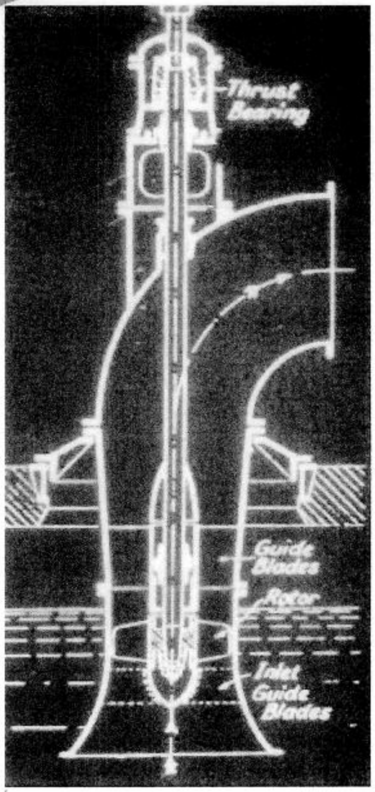
PSE 12-2008

www.cranfield.ac.uk

Axial Flow Shaft

Cranfield UNIVERSITY

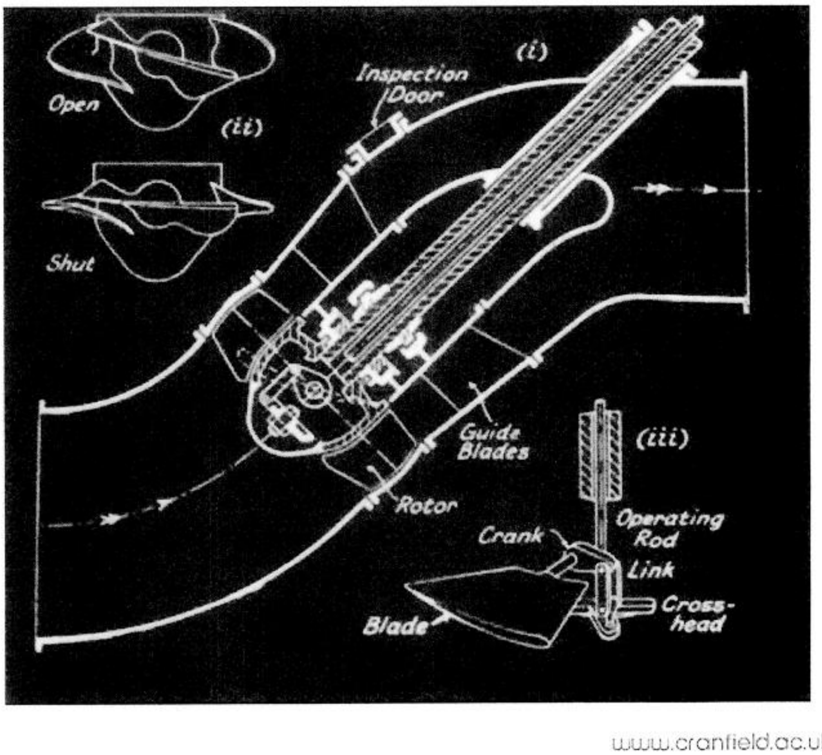
Process Systems Engineering



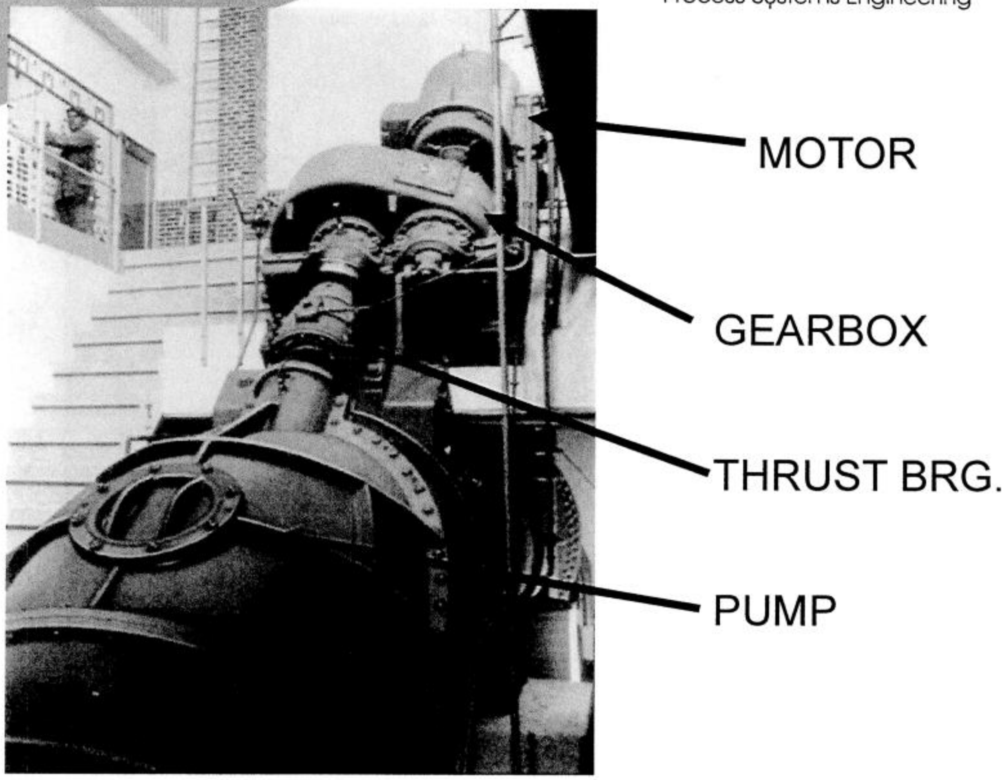
Axial Flow (Feathering Blades)

Cranfield UNIVERSITY

Process Systems Engineering

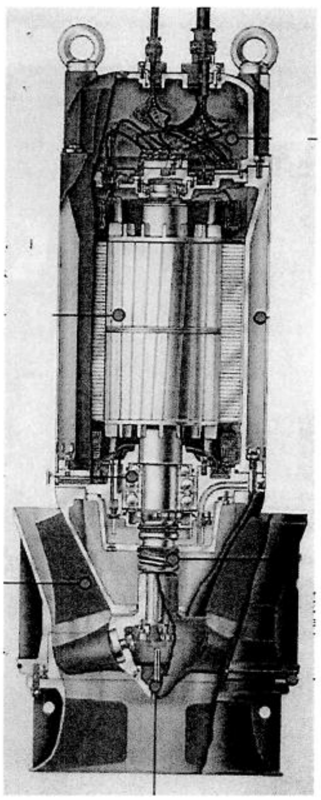


Cranfield UNIVERSITY
Process Systems Engineering



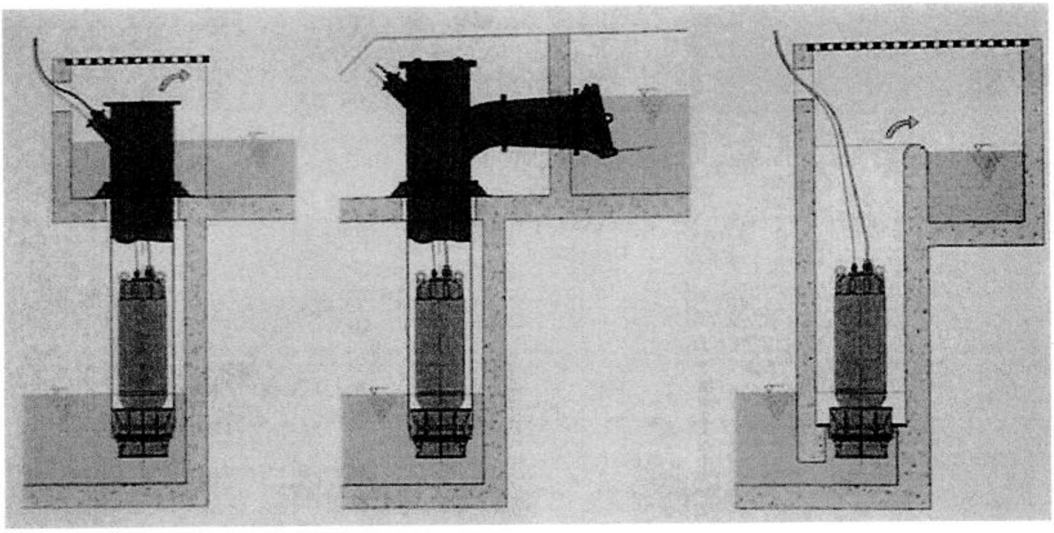
AXIAL FLOW SUBMERSIBLE MOTOR

Cranfield UNIVERSITY
Process Systems Engineering



Cranfield UNIVERSITY

Process Systems Engineering



PSE 12-2008

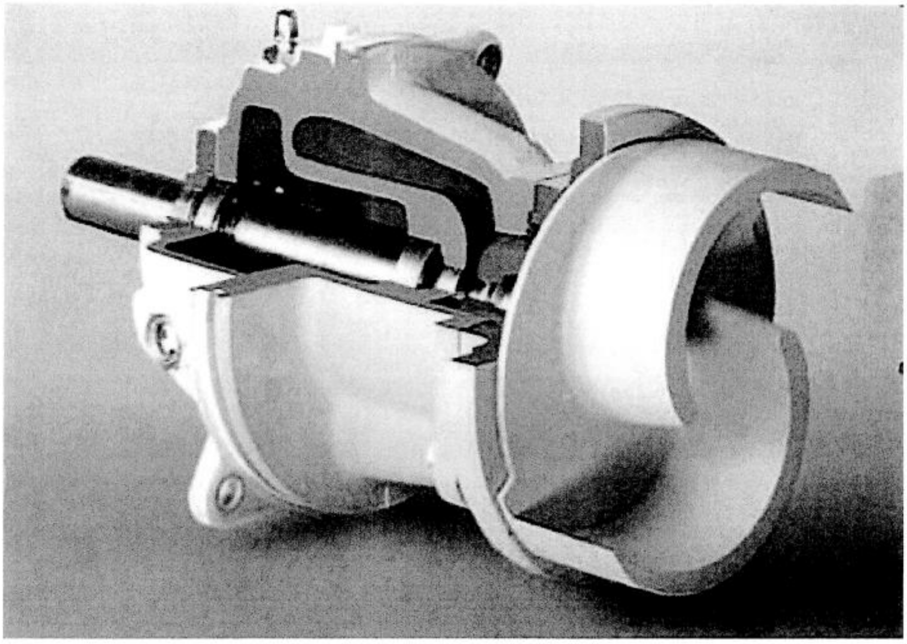
www.cranfield.ac.uk

SEWAGE PUMPS - THE PROBLEM

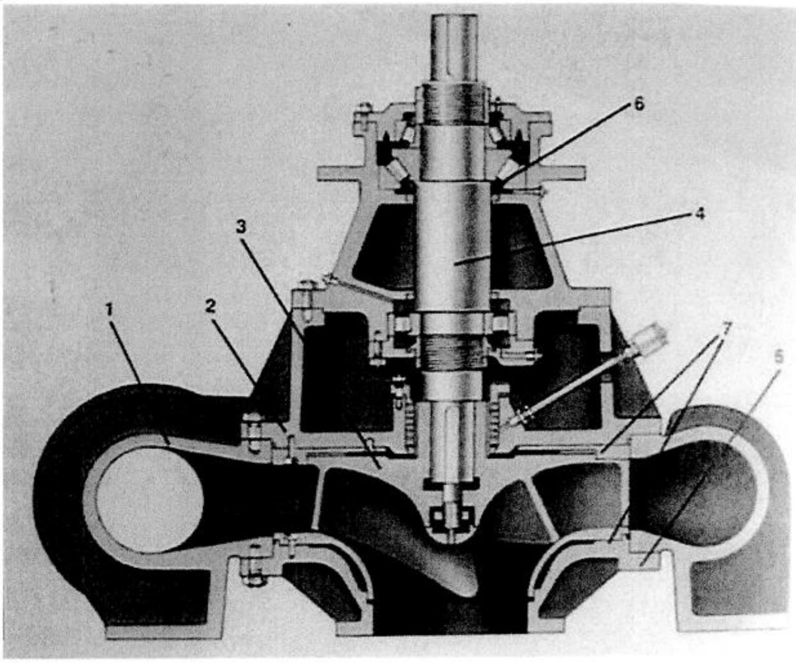
Cranfield UNIVERSITY

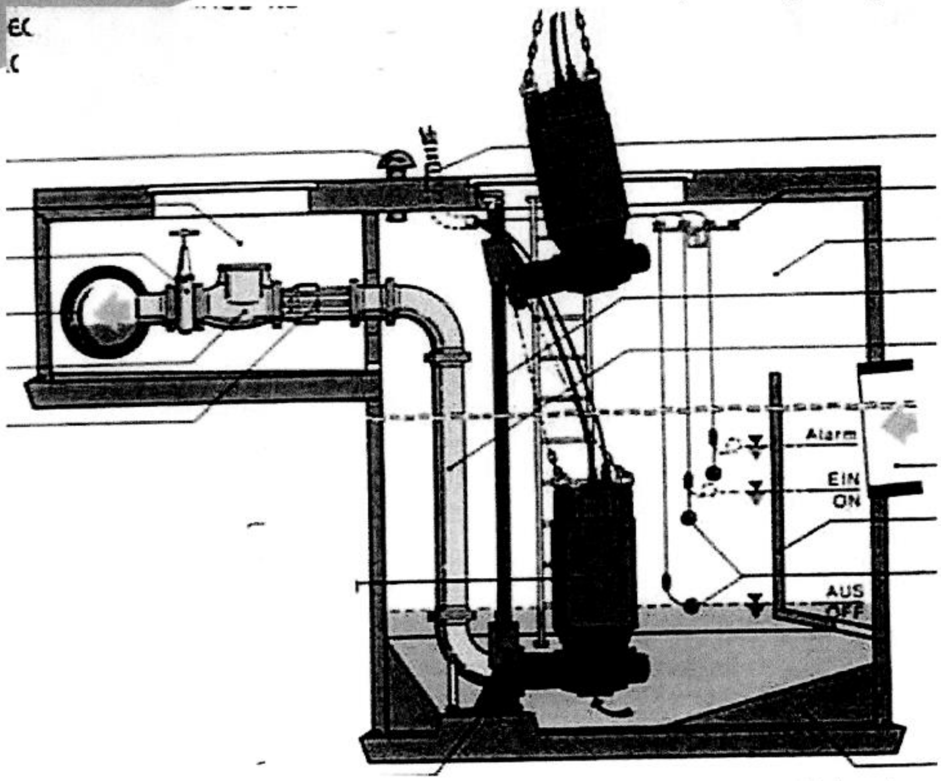
www.cranfield.ac.uk

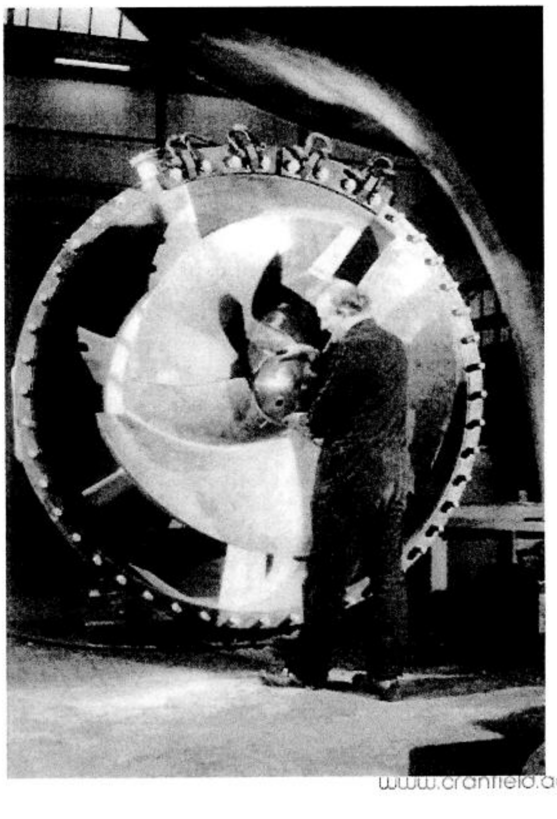
Cranfield UNIVERSITY
Process Systems Engineering



Cranfield UNIVERSITY
Process Systems Engineering




Alarm
EIN
ON
AUS
OFF



Cranfield UNIVERSITY

Pump Basics II

Mr Tony Salisbury

www.cranfield.ac.uk

Centrifugal Pump Design

Pumps and Pumping Systems

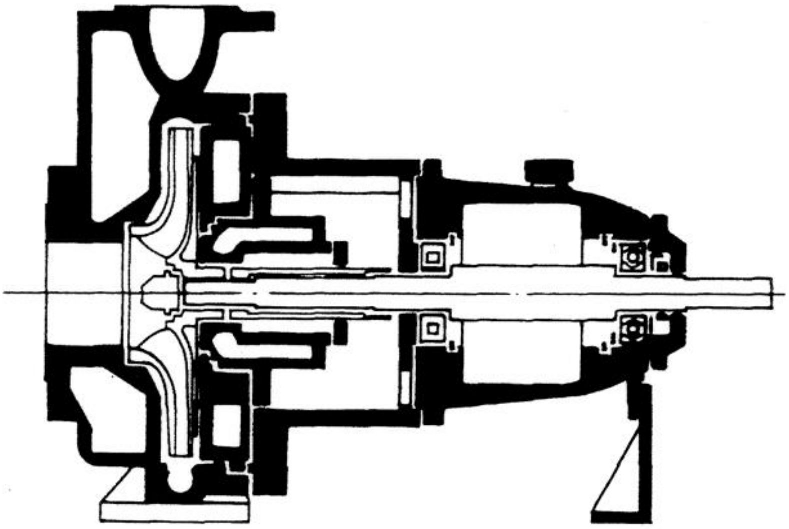
Cranfield UNIVERSITY

هيئة مياه وكهرباء أبوظبي
Abu Dhabi Water & Electricity Authority
ACADEMY

7city LEARNING

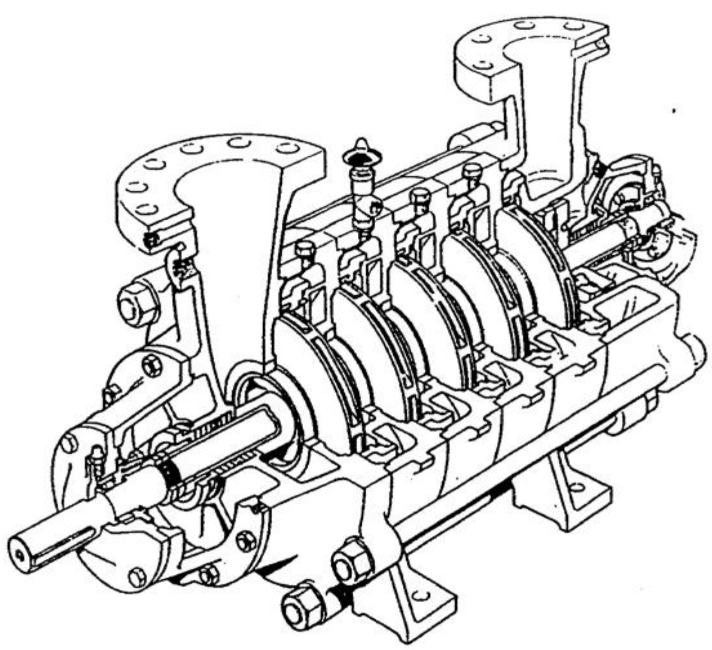
A Single Stage End Suction Back Pull-out Pump

Cranfield UNIVERSITY
Process Systems Engineering



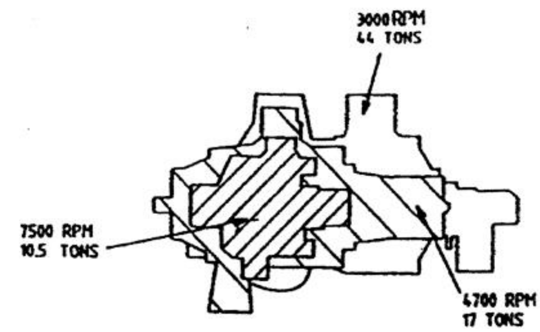
A multi-stage pump

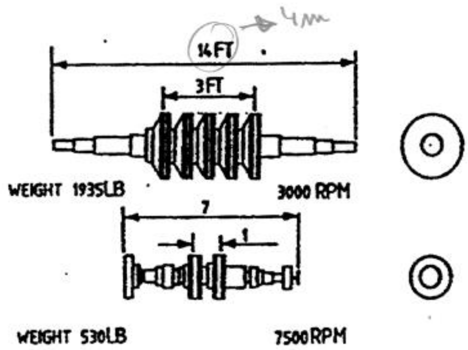
Cranfield UNIVERSITY
Process Systems Engineering



Effect of speed on size

Cranfield UNIVERSITY
Process Systems Engineering



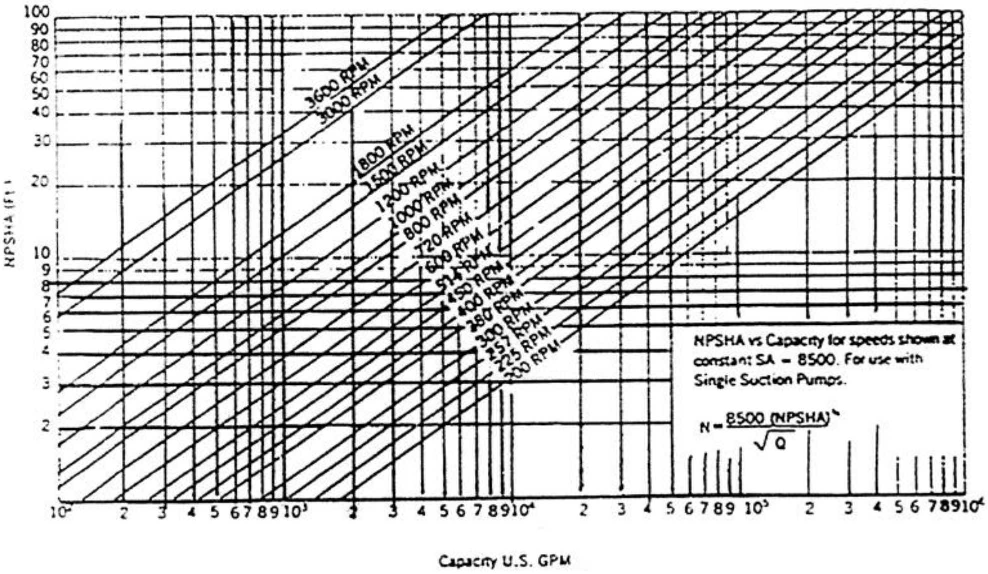


Recommended rotational speed

Cranfield UNIVERSITY
Process Systems Engineering

Q, H & NPSHa Given by the system

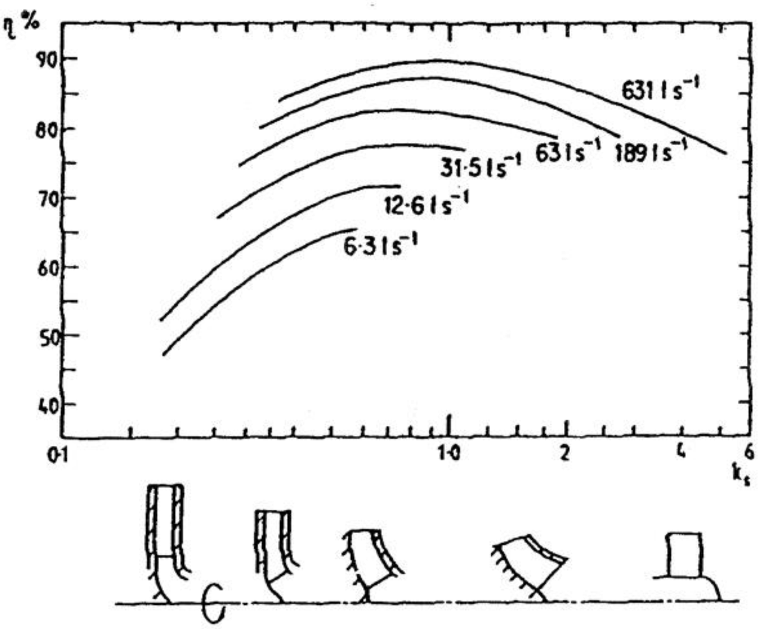
Q, NPSHa => (RPM / N)



RECOMMENDED MAXIMUM OPERATING SPEEDS FOR SINGLE SUCTION PUMPS

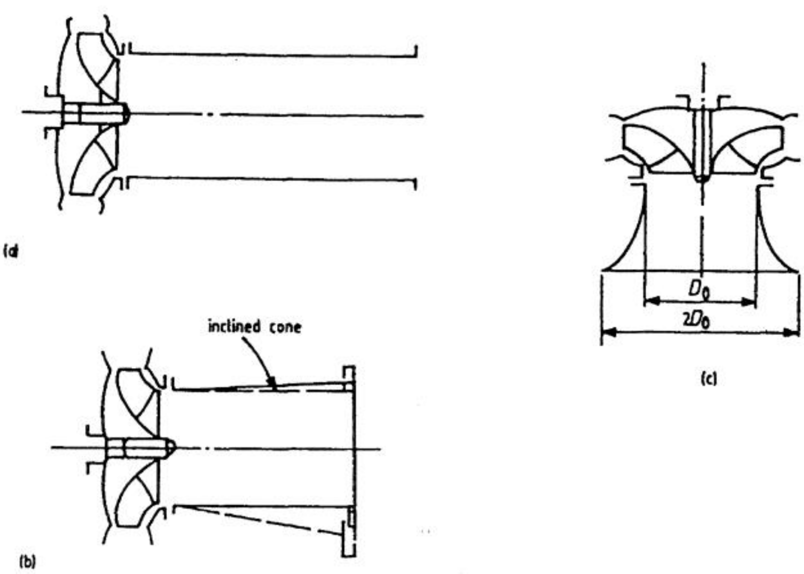
Efficiency and Type number (specific speed)

Cranfield UNIVERSITY
Process Systems Engineering

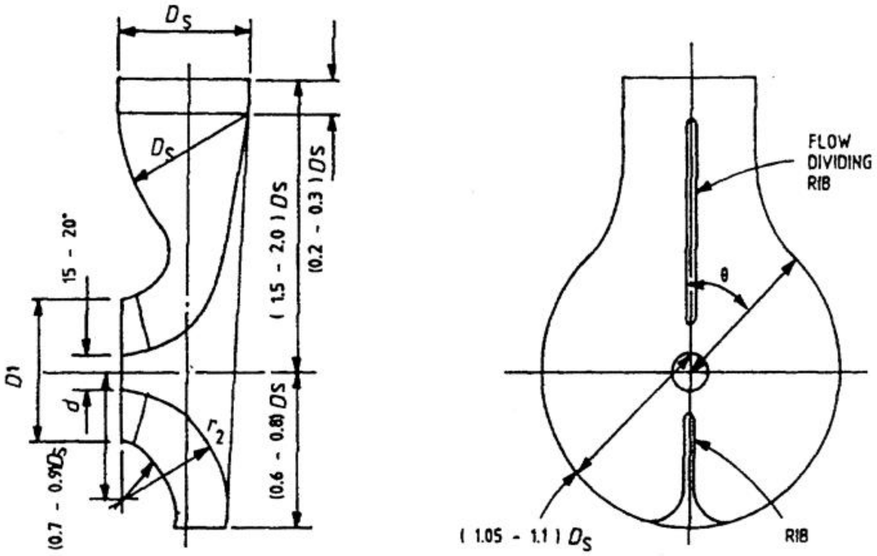


Alternative suction arrangements

Cranfield UNIVERSITY
Process Systems Engineering

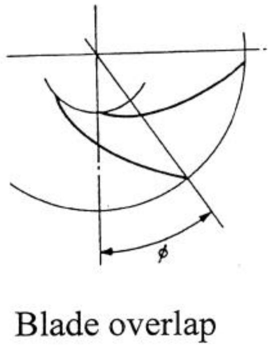


Typical suction duct showing dividing rib

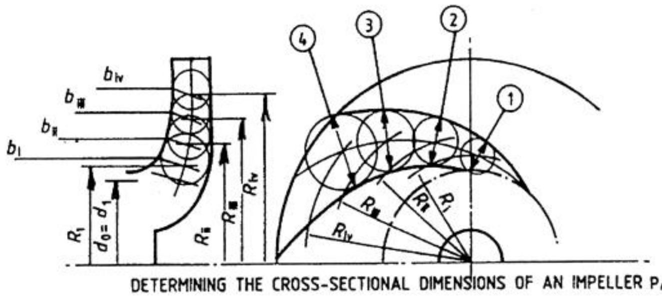


Blade Design

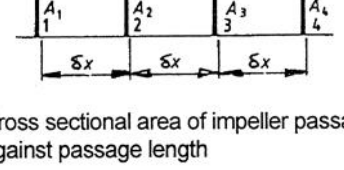
Cranfield UNIVERSITY
Process Systems Engineering



Blade overlap



DETERMINING THE CROSS-SECTIONAL DIMENSIONS OF AN IMPELLER PASSAGE

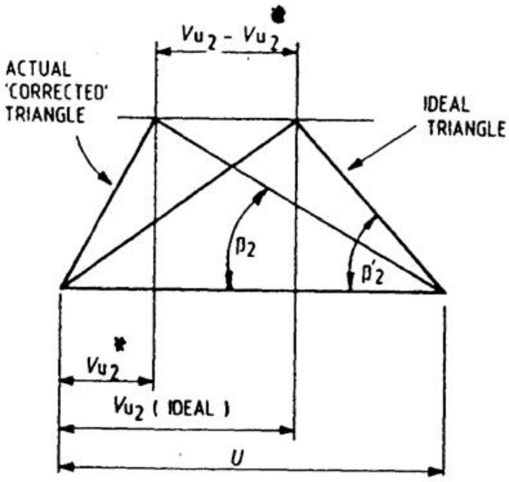


STATIONS ALONG PASSAGE

Cross sectional area of impeller passage plotted against passage length

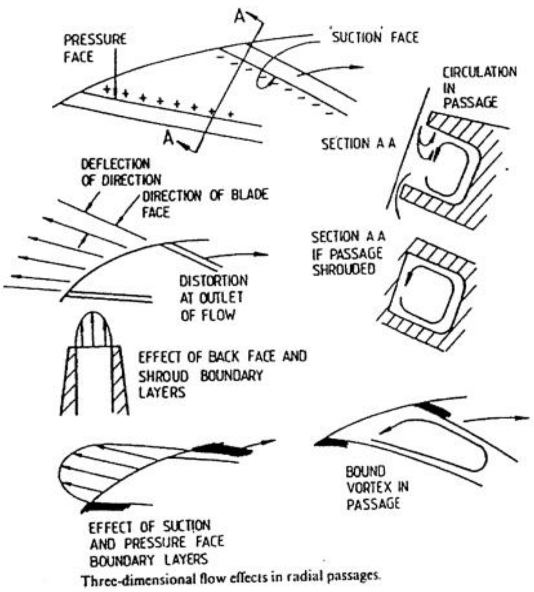
Velocity triangle

Cranfield UNIVERSITY
Process Systems Engineering

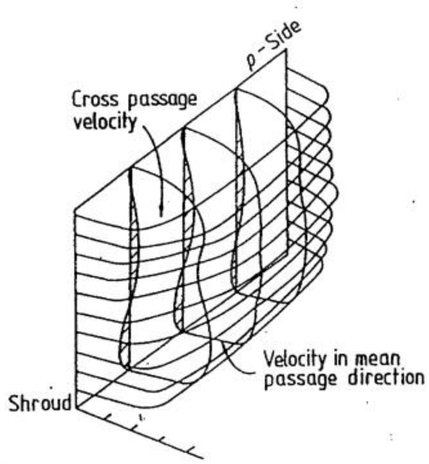


Three dimensional flow pattern

Cranfield UNIVERSITY
Process Systems Engineering



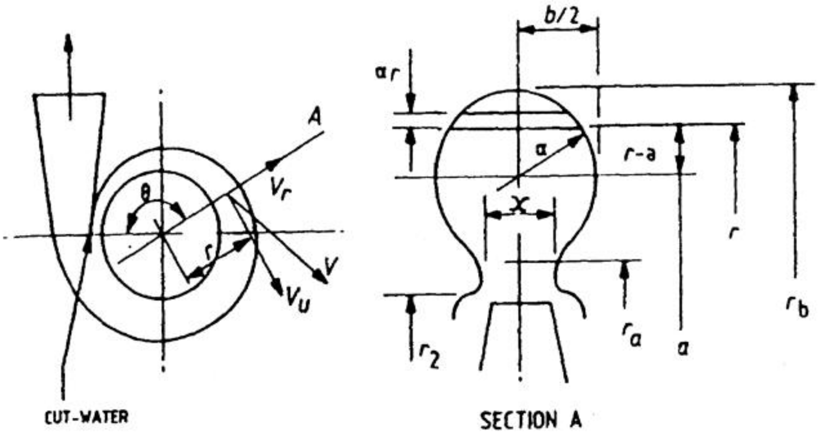
Three-dimensional flow effects in radial passages.



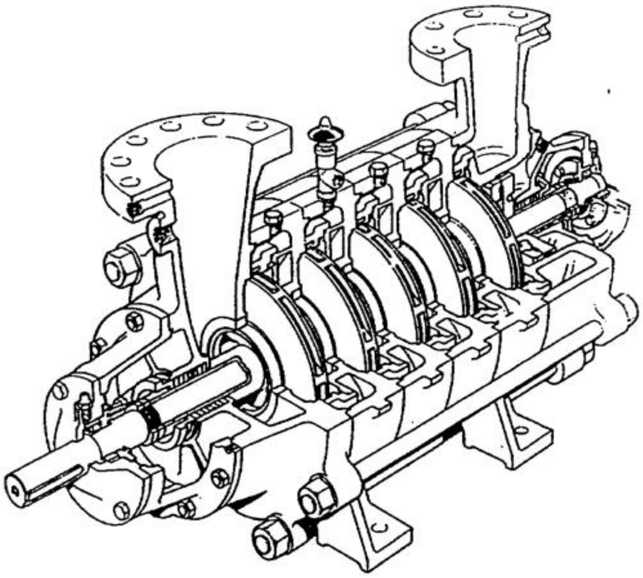
Typical velocity distribution in flow leaving a rotor passage.

A typical pump volute

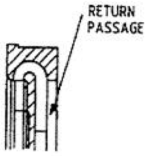
Cranfield UNIVERSITY
Process Systems Engineering

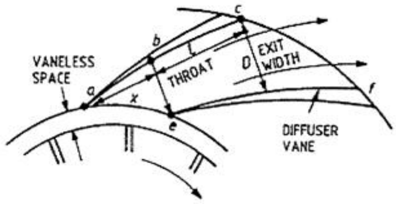


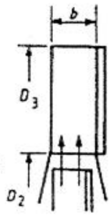
Cranfield UNIVERSITY
Process Systems Engineering

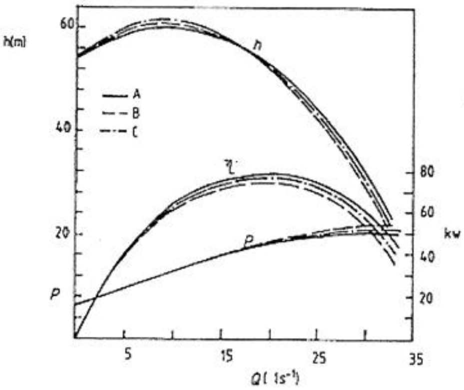


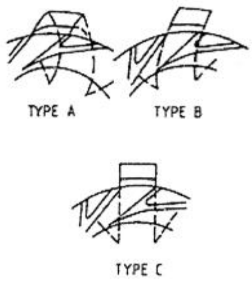
Vane diffuser and cross over duct design

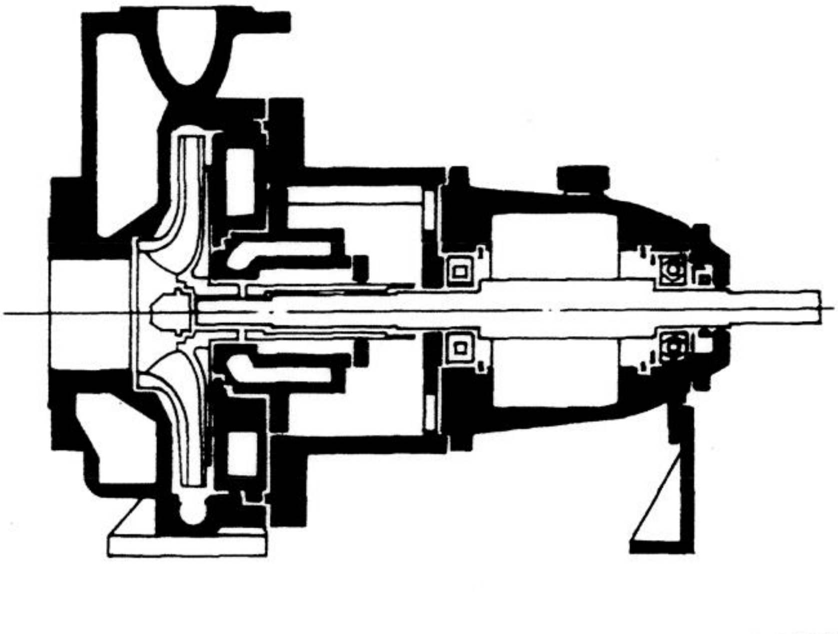






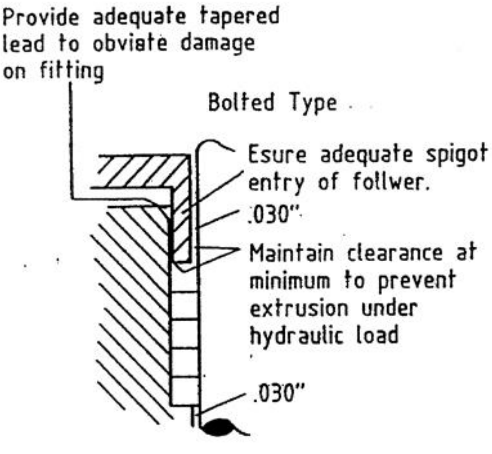




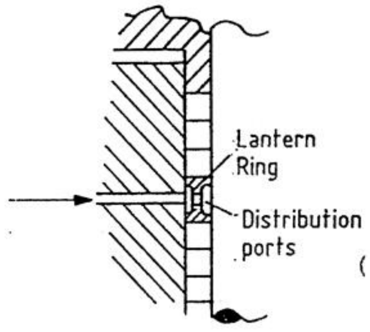


Packed gland detail

Cranfield UNIVERSITY
Process Systems Engineering



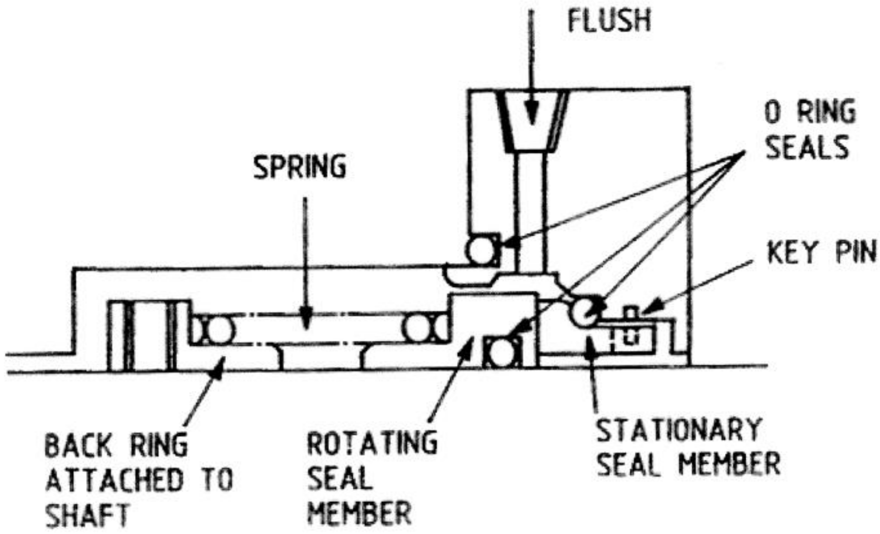
Design considerations



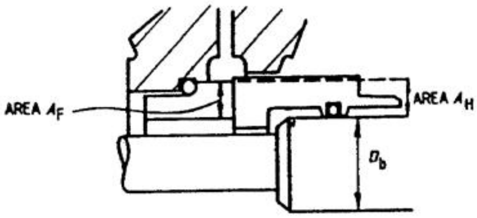
Typical lantern ring arrangement

A simple mechanical seal

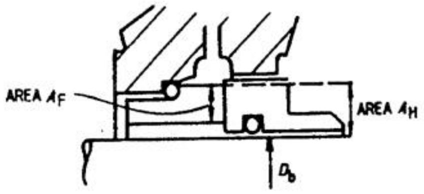
Cranfield UNIVERSITY
Process Systems Engineering



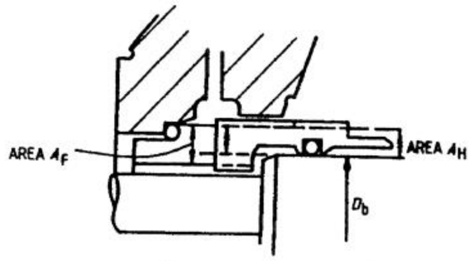
Force balance in a mechanical seal



(a) BALANCE RATIO $B = \frac{A_H}{A_F} = 1$

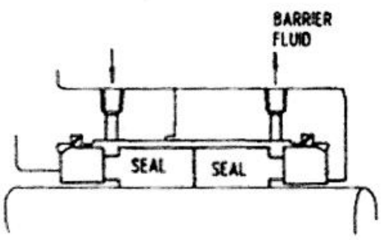


(b) $B = \frac{A_H}{A_F} > 1$

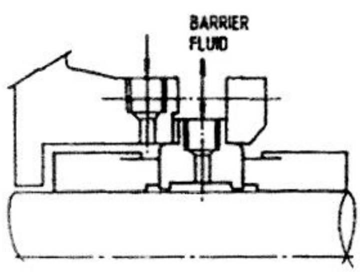


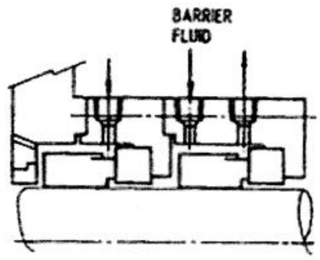
(c) $B = \frac{A_H}{A_F} < 1$

Double mechanical seals

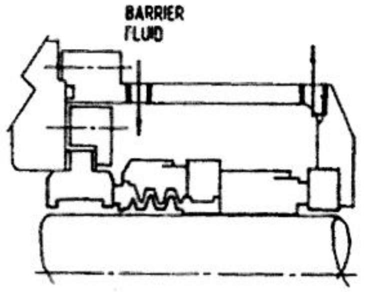


(a)





(c)

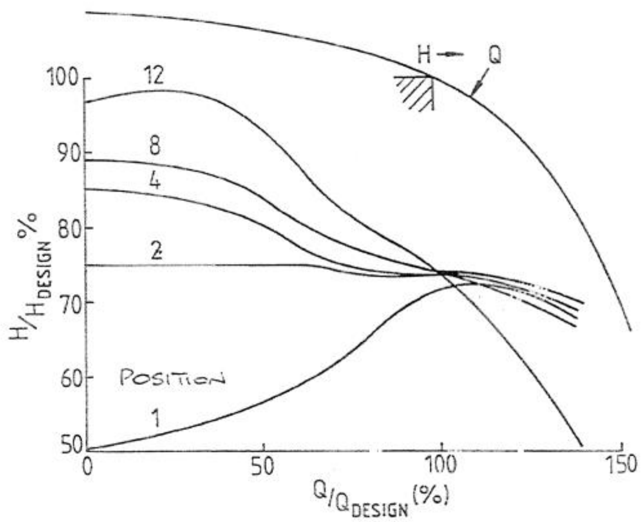


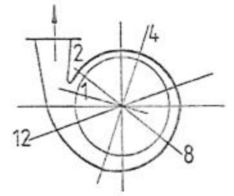
(d)

Pressure around the periphery of an impeller

Cranfield UNIVERSITY

Process Systems Engineering

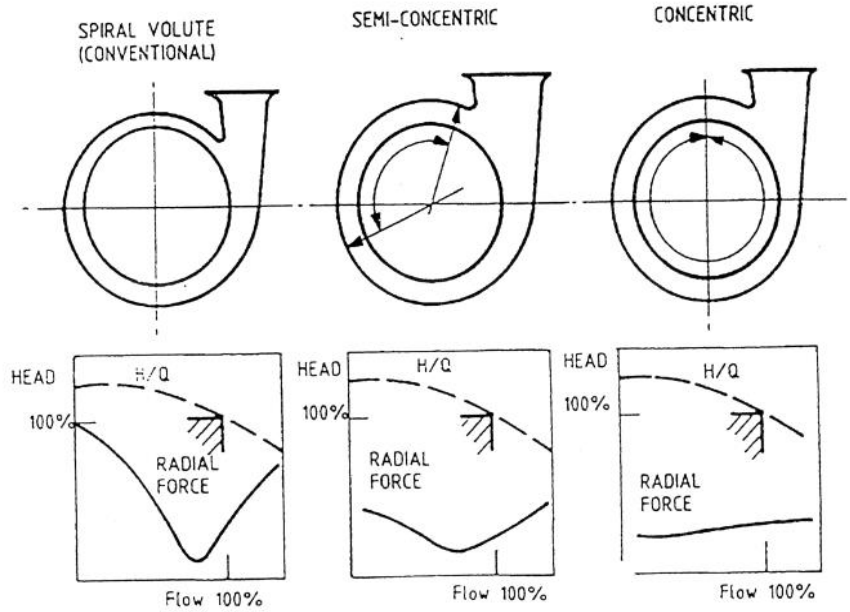




Effect of casing change on radial load

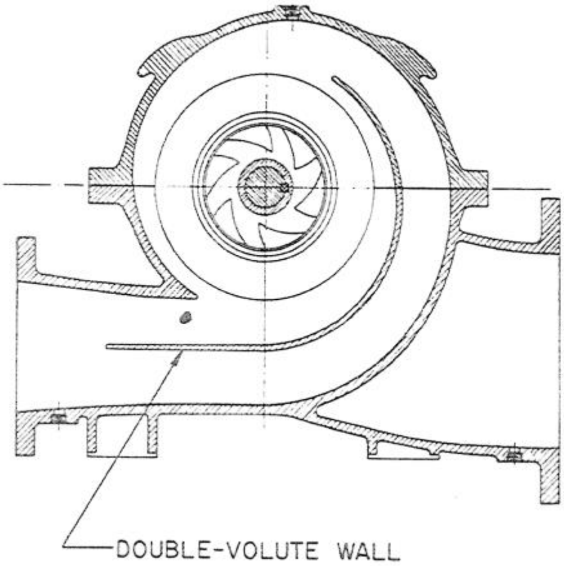
Cranfield UNIVERSITY

Process Systems Engineering



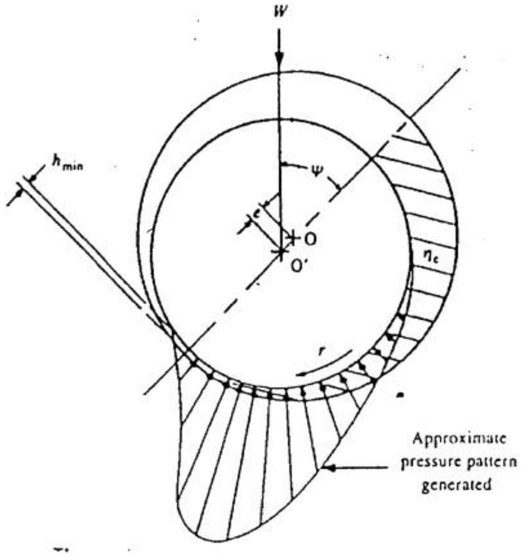
Double Volute Casing

Cranfield UNIVERSITY
Process Systems Engineering



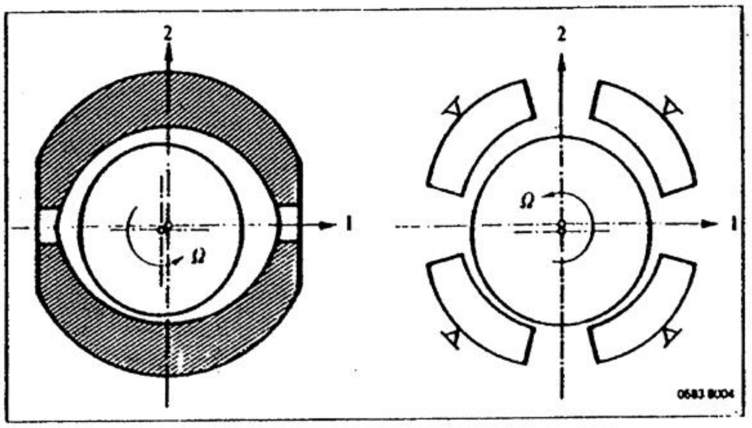
Pressure distribution of a typical cylindrical journal bearing under load

Cranfield UNIVERSITY
Process Systems Engineering



Journal bearing

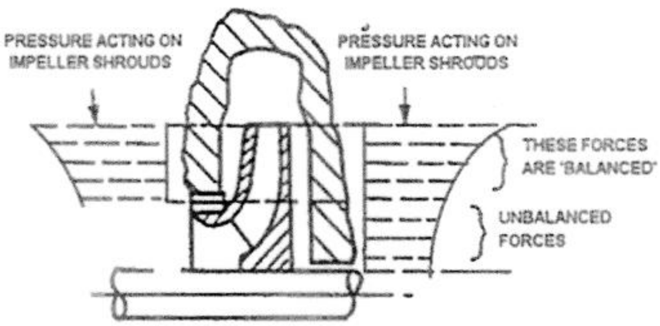
Cranfield UNIVERSITY
Process Systems Engineering

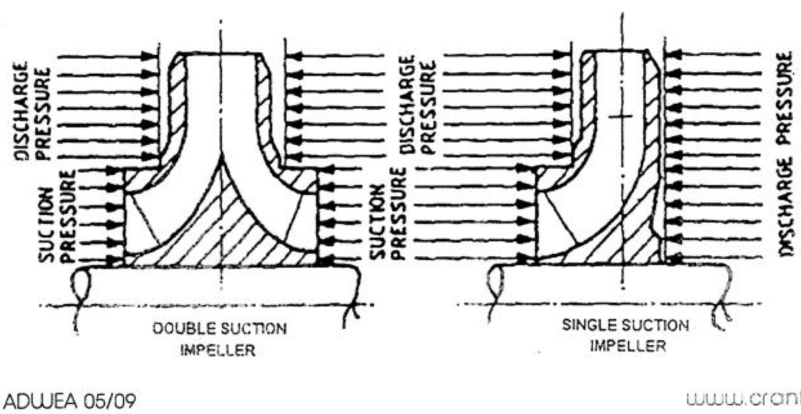


Journal bearing types: left two-lobe bearing; right tilting pad bearing.

Axial Forces on Impeller

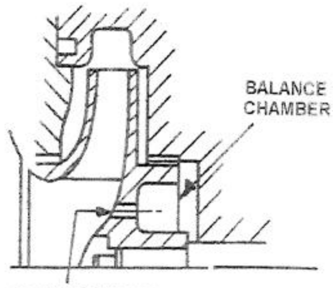
Cranfield UNIVERSITY
Process Systems Engineering

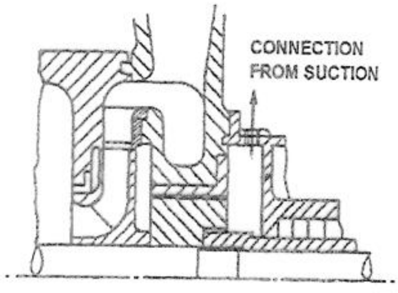




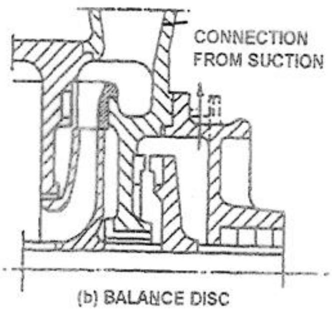
Balancing Axial Hydraulic Forces

Cranfield UNIVERSITY
Process Systems Engineering





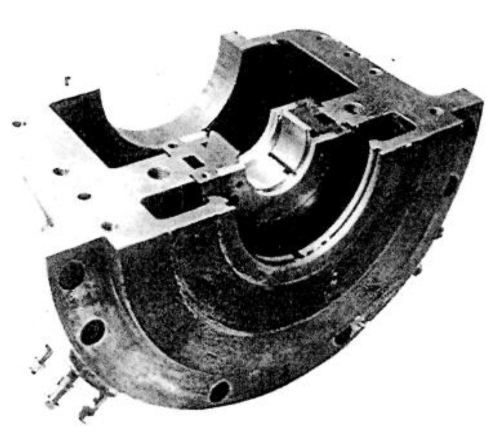
(a) BALANCE PISTON



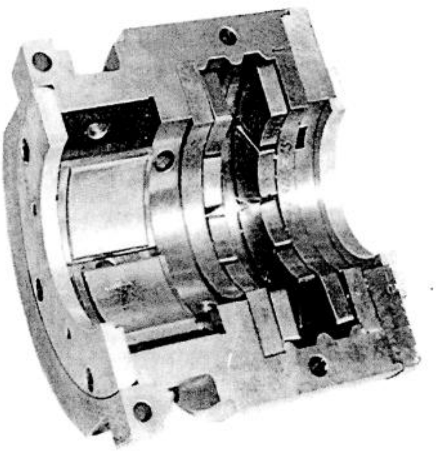
(b) BALANCE DISC

Tilting Pad Hydrodynamic Bearings

Cranfield UNIVERSITY
Process Systems Engineering



Bearing block with radial bearing



Combined radial and trust bearing

Forces on Rotor