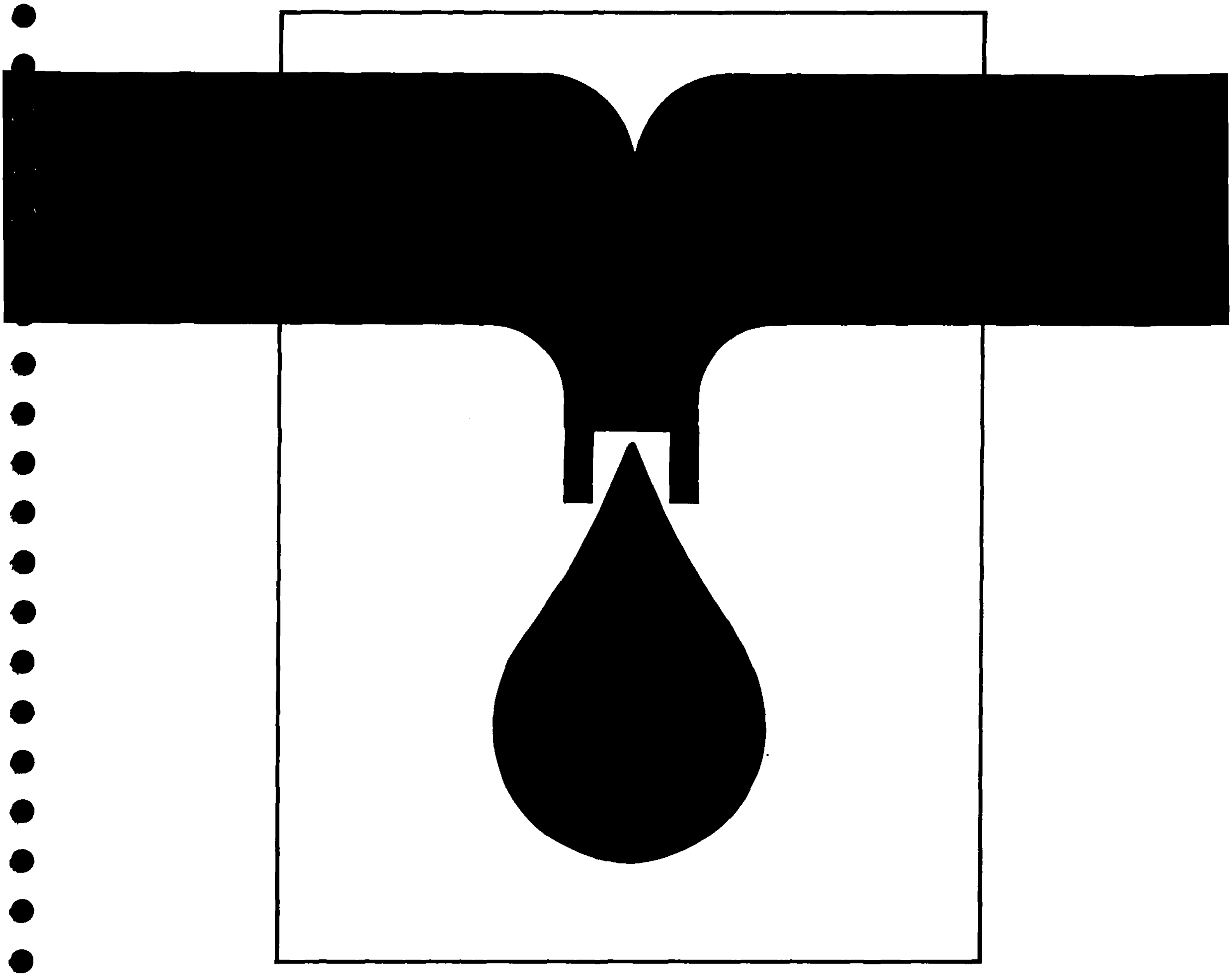
TRAINING MODULES
FOR
WATERWORKS PERSONNEL



Special Knowledge

2.3 e

Maintenance and repair of pumps

Foreword

Even the greatest optimists are no longer sure that the goals of the UN "International Drinking Water Supply and Sanitation Decade", set in 1977 in Mar del Plata, can be achieved by 1990. High population growth in the Third World combined with stagnating financial and personnel resources have led to modifications to the strategies in cooperation with developing countries. A reorientation process has commenced which can be characterized by the following catchwords:

- use of appropriate, simple and – if possible – low-cost technologies,
- lowering of excessively high water-supply and disposal standards,
- priority to optimal operation and maintenance, rather than new investments,
- emphasis on institution-building and human resources development.

Our training modules are an effort to translate the last two strategies into practice. Experience has shown that a standardized training system for waterworks personnel in developing countries does not meet our partners' varying individual needs. But to prepare specific documents for each new project or compile them anew from existing materials on hand cannot be justified from the economic viewpoint. We have therefore opted for a flexible system of training modules which can be combined to suit the situation and needs of the target group in each case, and thus put existing personnel in a position to optimally maintain and operate the plant.

The modules will primarily be used as guidelines and basic training aids by GTZ staff and GTZ consultants in institution-building and operation and maintenance projects. In the medium term, however, they could be used by local instructors, trainers, plant managers and operating personnel in their daily work, as check lists and working instructions.

45 modules are presently available, each covering subject-specific knowledge and skills required in individual areas of waterworks operations, preventive maintenance and repair. Different combinations of modules will be required for classroom work, exercises, and practical application, to suit in each case the type of project, size of plant and the previous qualifications and practical experience of potential users.

Practical day-to-day use will of course generate hints on how to supplement or modify the texts. In other words: this edition is by no means a finalized version. We hope to receive your critical comments on the modules so that they can be optimized over the course of time.

Our grateful thanks are due to

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and
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for their committed coordination work and also to the following co-authors for preparing the modules:

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It is my sincere wish that these training modules will be put to successful use and will thus support world-wide efforts in improving water supply and raising living standards.

Dr. Ing. Klaus Erbel
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Eschborn, May 1987

Title: Design, working principle, use, maintenance and repair of pumps

Table of contents:

1 Pump varieties

There is no universally valid system of nomenclature for pumps, since a wide range of different criteria are used to describe and name them. The following outline can therefore cover only those aspects which are most often used to classify the different types of pumps. In this way, pumps may be distinguished according to their ...

1.1 Working principle:

Centrifugal pumps. These work through a transfer of energy inside the vaned impeller resulting from deflections of the direction of flow; a method of operation which stands in contrast to the displacement principle by which displacement pumps work. The delivery head is proportional to the square of the pump's rotational speed.

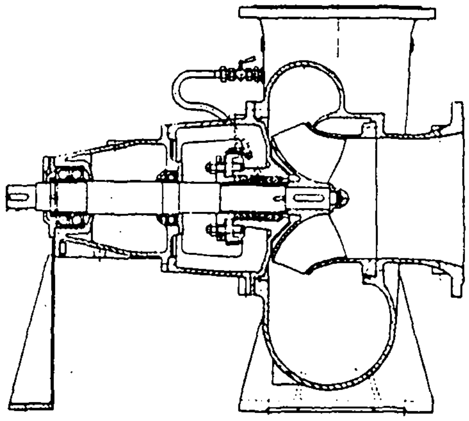


Fig. 1: Radially split volute casing pump with screw-type impeller

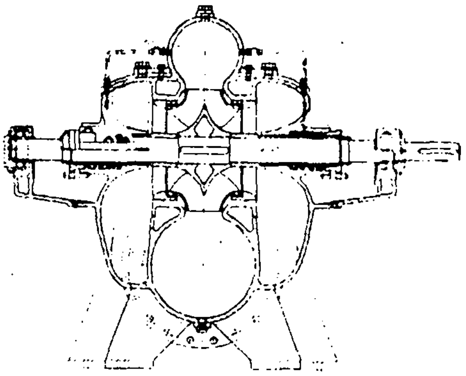


Fig. 2: Longitudinally split double radial-flow pump

Displacement (piston) pumps. These function through a periodic alteration of the volumes of chambers which are isolated from the suction and discharge branches by valves. Their delivery head is not dependent on the speed of the pump. A distinction is made between oscillatory and rotary displacement pumps. For instance, reciprocating, diaphragm and semi-rotary pumps are oscillatory. Examples of rotary displacement pumps are radial cylinder, eccentric spindle drag, gear, screw jack, multi-stage impeller and liquid seal pumps. In addition, the expansion of vapours or gases may be utilized to effect displacement.

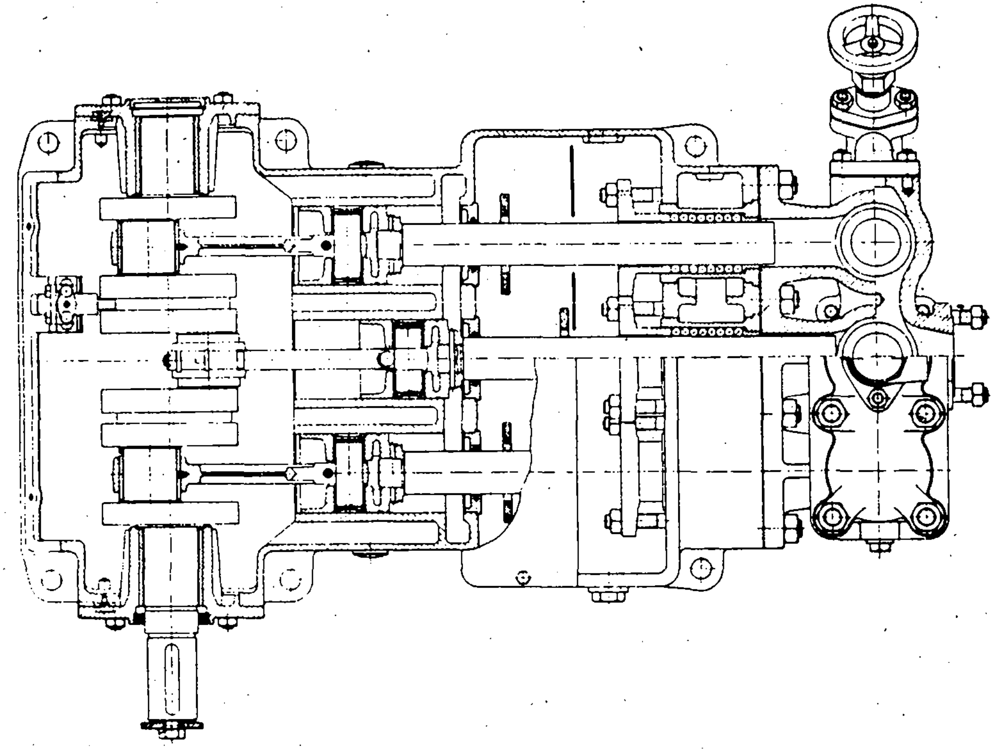


Fig. 3: Triple plunger pump

Jet pumps (deep suction devices). These are pumps in which a moving jet of a propellant medium delivered through a restricted throat is used to raise a liquid. The medium may be a gas, liquid or vapour. Such pumps have no moving parts and are thus of very simple design, although performances are limited.

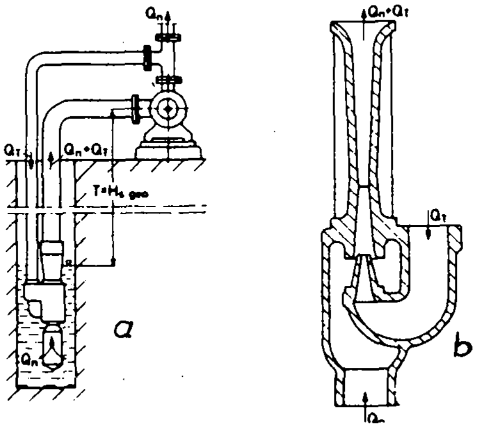


Fig. 4: a - Centrifugal pump with deep suction device
b - Jet pump as deep suction device

<u>Air lift (mammoth) pumps</u>. The operating principle of these pumps is based on the buoyancy effect of a mixture of liquid and gas.

<u>Hydraulic rams</u>. These use the kinetic energy of a flowing column of liquid, transforming this energy via periodic sudden checking of the flow into other energy forms (e.g. pressure).

<u>Raising devices</u>. These raise liquids to a higher level with the aid of scoop wheels, bucket devices, Archimedean screws and similar means.

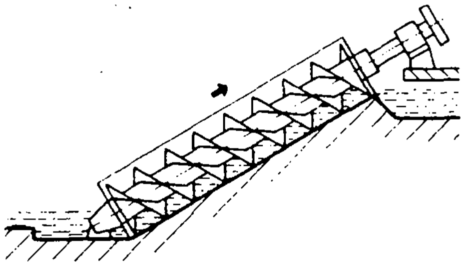


Fig. 5: Screw trough pump

Electro-magnetic pumps. This type functions as a result of the direct action of a magnetic field on a ferromagnetic medium. Use of such pumps is thus limited to the pumping of liquid metals, so that they play virtually no part in water-supply systems.

1.2 Construction:

The fundamentally different working principles of the varieties of pumps described above clearly dictate quite different constructions. When pumps are designated on the basis of some aspect of their construction, this is only to distinguish a variant within one of the main categories. The following comments are limited to describing variations in the design of centrifugal pumps, since these are the chief pumps used in water-supply systems.

Impeller design. Depending on the specific rotational speed of a centrifugal pump, its impeller may be constructed for radial flow, semi-axial (mixed) flow (also known as a helical or screw impeller) or axial flow (propeller-type). Radial and semi-axial impellers may have open passages (without covering plates) or closed passages (with covering plates). Propellers may be cast all in one piece with their hubs, or the blades may be fixed adjustably to the hub for better correction of the characteristic curve, or they may, for control purposes, be adjustable in operation. The blades of the propeller do not have to be at right angles to the pump's centre of rotation, especially if the propeller is used, whilst retaining its control advantages, for greater delivery heads and the medium therefore flows through it with radial velocity components.

There are, further, special impellers which are used in the pumping of certain media - e.g. single-vane, double- and triple-passage, free-flow and peripheral impellers.

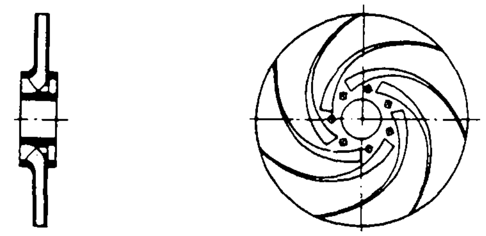


Fig. 6: Radial-flow impeller

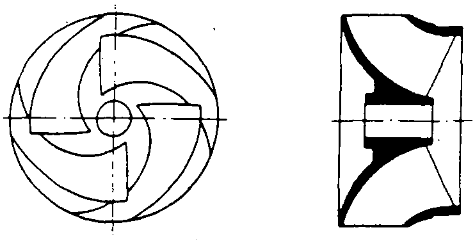


Fig. 7: Mixed-flow impeller

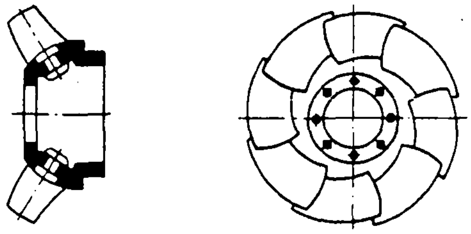


Fig. 8: Mixed-flow propeller

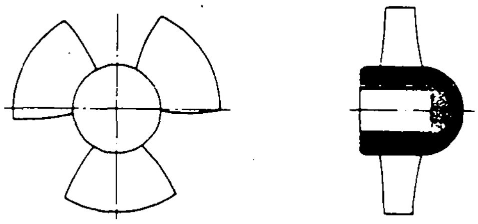


Fig. 9: Axial-flow impeller

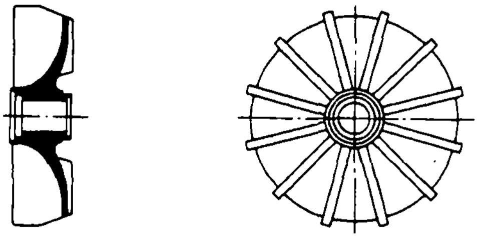


Fig. 10: Free-flow impeller

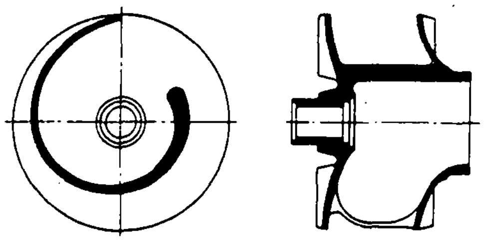


Fig. 11: Single-vane impeller

Impeller configuration. The impeller of a centrifugal pump can be mounted either on both sides or on one side only (overhung). Mounting on one side means that one less shaft seal is needed, but it increases shaft deflection under otherwise comparable conditions. If large volumes are to be pumped, bilaterally mounted impellers can be constructed for a double flow of liquid, whereby the axial pressure can be simultaneously equalized. High delivery heads can be achieved through a multi-stage arrangement of impellers.

If equalization of the axial pressure is required here too, an even number of impellers can be used in back-to-back arrangement. Combinations of multi-stage and multi-flow constructions are also possible.

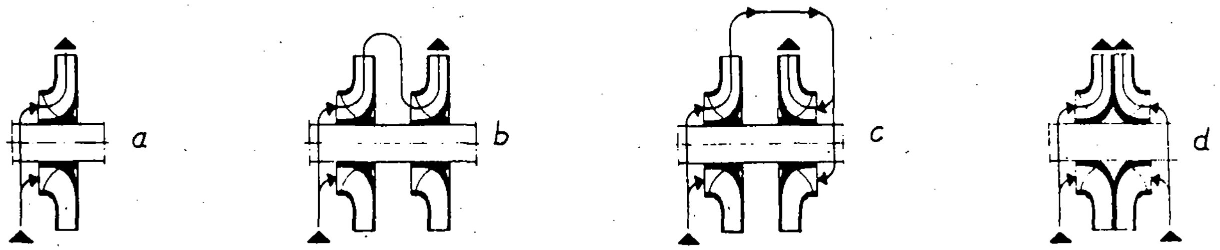


Fig. 12: a - single-flow, single-stage
b - single-flow, two-stage
c - single-stage, two-stage (back-to-back)
d - double-flow, single-stage

Design of the pump casing. Most constructional differences result from variations in the design of the pump casing. With increasing specific speeds, the volute type of casing is chosen to house the pump. With the aim of equalizing the radial thrust, the volute casing is sometimes constructed in the form of a double helix, with two spirals beginning opposite each other. Further possible designs are the twisted spiral, with a strongly asymmetrical cross-section of the spiral at the vertical centre, the annular-type casing, with a uniform cross-section over the circumference, and the tubular casing, which guides the liquid delivered by the impeller in axial direction - in a similar way to the elbow casing pump, the impeller of which does not deliver into a coaxial rising pipe, however, but into a pipe bend.

The shape of the casing may also be determined by the range of pressures: low-pressure pumps require different constructions from high- or maximum-pressure pumps. In the case of volute and multi-stage pumps, the external geometrical form becomes simpler as the pressure level increases, approaching cylindrical, conical or spherical designs. The advantages of these constructions - i.e. reduction of stress, ease of installation - are more or less concomitant with a loss of efficiency or of capacity of the casing - thus also with higher costs.

Further variations in the construction of the pump casing arise from its separation into two parts horizontally or vertically, at right angles or parallel to the shaft - which are nearly always practised for greater convenience of installation and servicing.

Finally, the position of the suction and delivery connections on the pump also has an effect on the design of its casing. The axial intake - i.e. the suction branch - of single-stage volute pumps is, for instance, a particular characteristic of this type of pump - in contrast with inline pumps, where intake and delivery are located opposite each other, or refinery pumps with both connections on the top. Where the pump shaft is mounted on one side only, even its alternative mounting in a bearing flange or bearing block is sometimes taken as a distinguishing feature.

In segmented pumps, the casing sections are designated according to their function - i.e. suction casing, stage casing, several of which may be lined up in series, and delivery casing. When assembled, the casings, including the anchor bars which hold the suction casing, stage casings and delivery casing together as an airtight unit, are often surrounded by a jacket made of sheet metal to give the pump its external form. In the case of hot-water pumps, the jacket contains the insulating materials (glass fibres, high-gloss foil etc.)

If the casing joint is in the same plane as the shaft, e.g in centrifugal pumps with horizontal shaft the casing consists of a lower section, which contains the suction and delivery connections and the pump feet, and a simpler upper section.

Both sections are provided with a flange at the joint, which extends right round the casing, including the two stuffing-box housings, and which is used to fasten the two parts of the casing together, with the aid of many screws, so that it is completely air- and water-tight.

In longitudinally divided pumps with a vertical shaft, the impeller often runs in a submerged bearing, so that the second shaft packing is not required. The two casing sections in this case are called the back section - this contains the suction and delivery connections and the pump foot - and the front section.

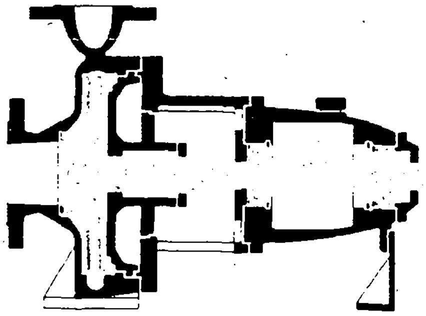


Fig. 13: Simple volute casing with axial suction branch

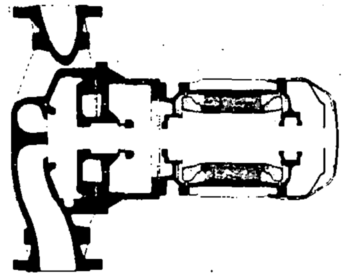


Fig. 14: Inline casing

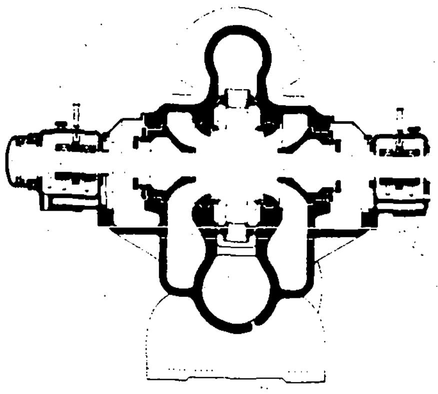


Fig. 15: Horizontally divided casing

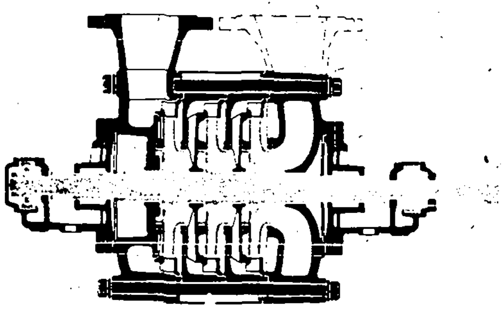


Fig. 16: Multi-stage segmented pump

Methods of installation. Methods of installation vary quite widely. A basic distinction is made between centrifugal pumps with a horizontal shaft and those with a vertical shaft. Pumps with a vertical shaft may be installed in wet or dry surroundings. Centrifugal pumps which can be completely immersed are also known as submersible pumps - most pumps with tubular casings, for instance, come under this heading. Centrifugal pumps with their shafts positioned at an angle are occasionally found in pumping stations for pumping large quantities of water at low heads.

A second characteristic which varies to some extent is the method of mounting the centrifugal pump on its foundation. It may rest on its own feet, or it may be flanged onto the prime mover in the form of a self-contained unit. The connection between the casings of pump and prime mover, e.g. in the case of flanged motors or motor lanterns, provides further distinguishing features, as also does the design of the base plate (one for pump and prime mover together, or separate bases for motor and pump).

Finally, in contrast with permanent methods of installation, pumps may also be mounted on wheels or skids, or be provided with a carrying handle.

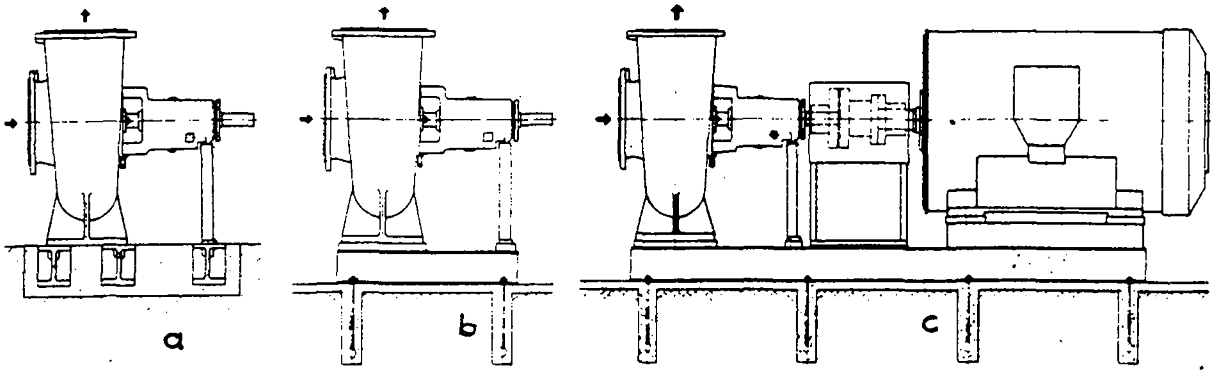


Fig. 17: Horizontal, dry installation
a - pump and motor mounted separately on foundation blocks
b - pump and motor mounted separately on base plates
c - pump and motor mounted on a common base plate

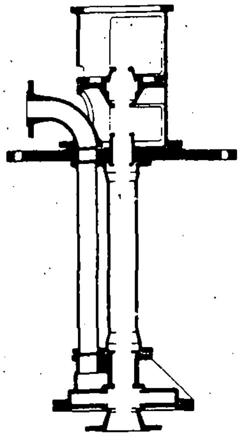


Fig. 18: Vertical shaft, submerged installation

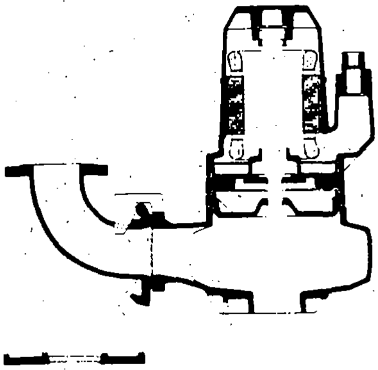


Fig. 19: Submersible motor, submerged installation

1.3 Field of application:

Pumps are also commonly classified according to their application. Mode of operation, exact use and substance(s) the pump is designed to handle may provide an additional component of the pump's designation in this context. Such terms are usually easily understood without further explanation.

The different modes of operation can be categorized roughly as follows: in additon to the operational pumps forming the main category, there are also standby and substitute pumps. The principal pump may occasionally be supported by a feed or "booster" pump. Terms such as full-load, base-load, part- or weak-load, auxiliary, starting or emergency pump are all derived from the pump's specific application.

Many names derive from the plant served by the pump. Only the most common of these are given below.

In the field of water supply and distribution, including irrigation, drainage and sewage disposal, pumps may be referred to as waterworks, hydrophor, deep-well of wellpoint, irrigation, sprinkler, dewatering, flood-water of storm-water pumps, to name only a few of the most common types.

In power stations and heating installations, the pumps used will be known as boiler-water feed, condensed-steam, reactor, stored-water (these can often operate both as pumps and as turbines), or heating-water circulating pumps.

In the chemical and petro-chemical industries: chemical, pipeline, refinery, process, inline, loading, mixing and return-flow pumps.

In marine engineering: ship's pumps, oil pumps for loading and unloading tankers, ballast and bilge pumps and dock pumps for filling and draining docks.

Other applications require: trench drainage pumps, dredging, fire-control, injection, jetting, scavenging, petrol and vent/vacuum pumps.

The designation of a pump is also often derived from the medium it is designed to handle.

Most pumps transport fluids the main component of which is water: pure-water, drinking-water, hot- and cold-water pumps, cooling-water, sea-water and brine pumps, condensate and feed-water pumps; sewage and faeces pumps, liquid manure, sludge and solids-handling (non-chokeable) pumps, cellulose and wood-pulp pumps.

Named after other fluids are oil pumps (heating oil, lubricating oil, hot oil), fuel pumps, heat-transfer oil pumps, coolant, liquid gas, grease, acid, lye, beverage, fish, beet, fruit and concrete pumps.

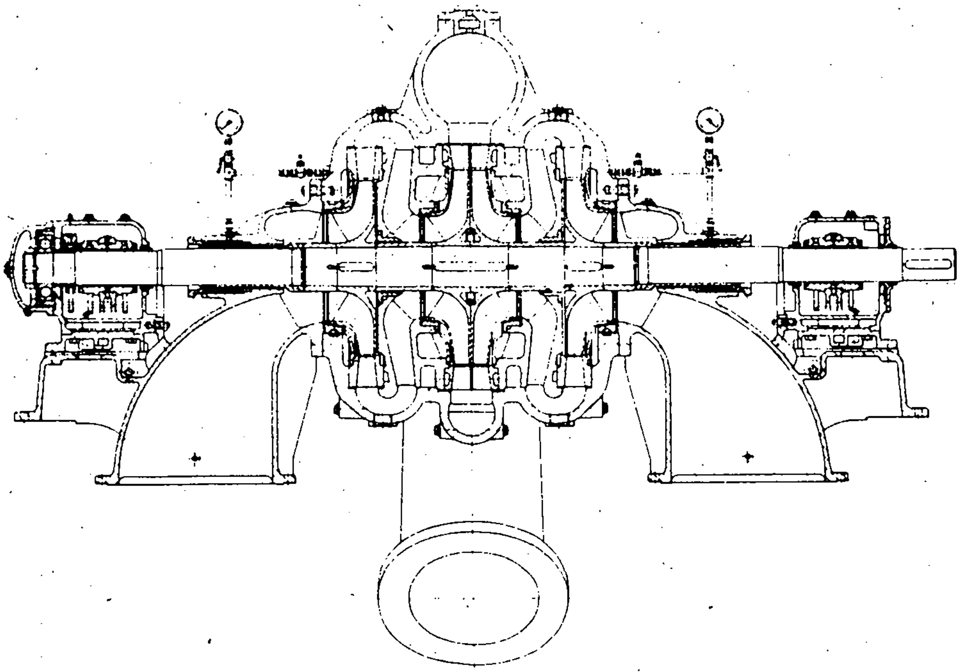


Fig. 20: Typical waterworks pump; double-flow, two-stage volute casing pump, longitudinally split

1.4 Prime mover:

It is quite common to name pumps after their prime mover; these terms are also easily understood: hand-operated an motor-driven pumps, turbine, gear-type and electric pumps, flange-mounted motor pumps, submersible-motor, wet-motor, canned-motor and magnetic induction pumps (these are pumps without stuffing boxes in which the shaft torque is transmitted via magnetic induction).

1.5 Material:

When centrifugal pumps are named after the material used, this nearly always refers to the casing only, since the various pump components are made of the material best suited to their function: i.e. there is not one material common to the complete pump. The materials are only loosely classified, e.g.:

Cast-iron, nodular-iron, bronze, cast-steel, stainless-steel, plastics and ceramic (stoneware or china) pumps, concrete casing pumps.

The parts of a centrifugal pump which come into contact with the pumped fluid are often given a protective coating, thus: rubber-lined, enamelled and plastics-lined pumps, also armoured pumps (surfaces in contact with the liquid are built up by welding).

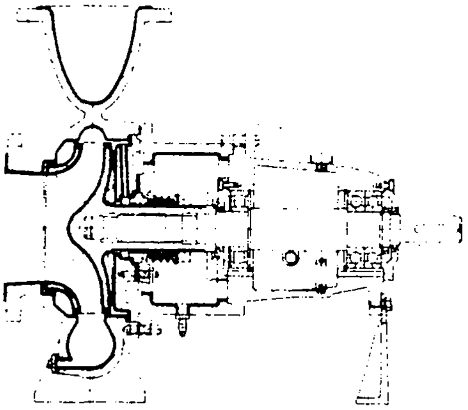


Fig. 21: Volute pump with rubber lining

2 Centrifugal pumps

The centrifugal pump is a device which utilizes the centrifugal force imparted to the liquid by a rapidly-rotating impeller. The transfer of force ends as soon as the fluid is discharged at the periphery of the impeller, and results in increased pressure and velocity of the fluid. To convert the velocity head (actually an unwanted side-effect, since only an increase of pressure in the pump is desired) into pressure head, the fluid leaving the impeller is guided through stationary passages which gradually expand, so that velocity is converted into pressure. These passages, which are attached permanently to the casing, are known collectively as the diffusor. Sometimes there is only one passage arranged as an annular chamber round the impeller, which may then take the form of a spiral tube.

The most important component of a centrifugal pump is the impeller. In most centrifugal pumps, the fluid enters the impeller near the centre of rotation and discharges from it with a radial motion at its periphery, This is the most common arrangement because the centrifugal forces then result in a continuous increase in pressure in the direction of motion.

With increasing delivery head and a certain discharge, a point is eventually reached where it is no longer expedient to operate with one impeller only. Although it is technically possible to design an impeller to handle any delivery head - however great - and any velocity, this impeller could operate beyond a certain point only at a low rate of efficiency; moreover the size and shape of the pump would reach impractical dimensions. For this reason, it may be necessary in order to achieve a certain required delivery head to connect a number of impellers in series, i.e. in the form of a multi-stage device, whereby the delivery head of each impeller is proportionately lower. The impellers run on a common shaft.

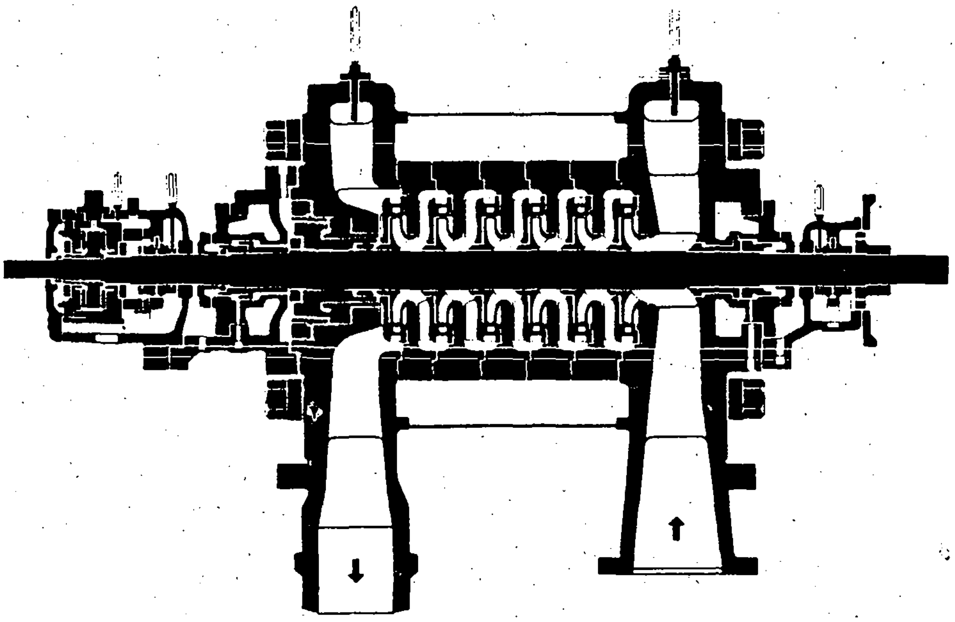


Fig. 22: Multi-stage boiler feed pump

Just as it is expedient to split up the pressure in the case of extreme delivery heads, where pumpage is very high the volume of liquid may have to be distributed amongst a number of impellers connected in parallel. This results in the multiple-flow arrangement - whereby, generally speaking, single impellers are used with the liquid entering on both sides. This concept has today largely been replaced by very rapidly rotating impeller types, however.

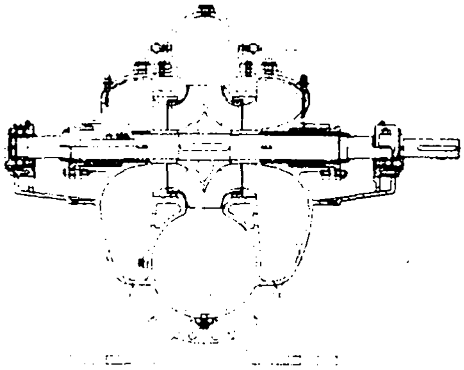


Fig. 23: Double-flow volute pump

A variant of the centrifugal pump is the peripheral pump. Here, energy is imparted to the fluid in a passage next to the impeller, by means of a star-shaped impeller rotating concentrically in the casing. The impeller, with straight, radial vanes without side walls, is very simply constructed and is narrowly enclosed on the outside and on both sides by the casing. The casing has a passage on one or on both sides which continues almost completely round the circumference but is interrupted at one point (between the entry and the exit slits). In the course of rotation, the fluid is recirculated many times between the vanes of the impeller and the peripheral passage, whereby an intensive energy transfer takes place by means of pulse exchange between the fluid rotating at the high circumferential velocity of the impeller and the fluid flowing at a lower speed through the peripheral passage. This pulse exchange is repeated many times over at the impeller's periphery and results in an extremely high delivery head, which may be 5 to 15 times higher than that of radial-flow impellers of the same size rotating at the same velocity.

In addition to this extremely high delivery head - which can be increased still further by using peripheral pumps in multi-stage arrangement - this type of pump has the advantage of being self-priming. As in liquid seal pumps, the water remaining in the pump casing forms, with rotating impeller, a ring of water with a free surface if air is drawn into the peripheral pump through the suction branch. This liquid seal would circulate concentrically and therefore ineffectively in the casing if the passage were not interrupted at one point on the circumference. At this point, the inner radius of the ring is smaller due to the displacement effect, so that the empty spaces filled with air between the radial vanes of the impeller are also reduced and the peripheral pump thus acts as a compressor on the displacement principle, in the same way as a liquid seal pump.

2.1 Characteristic curves

In the case of a centrifugal pump being driven at a constant speed, the delivery head H, power consumption P - and thus also the degree of efficiency - and the required NPSH are all functions of the capacity Q. These relationships are shown by characteristic curves.

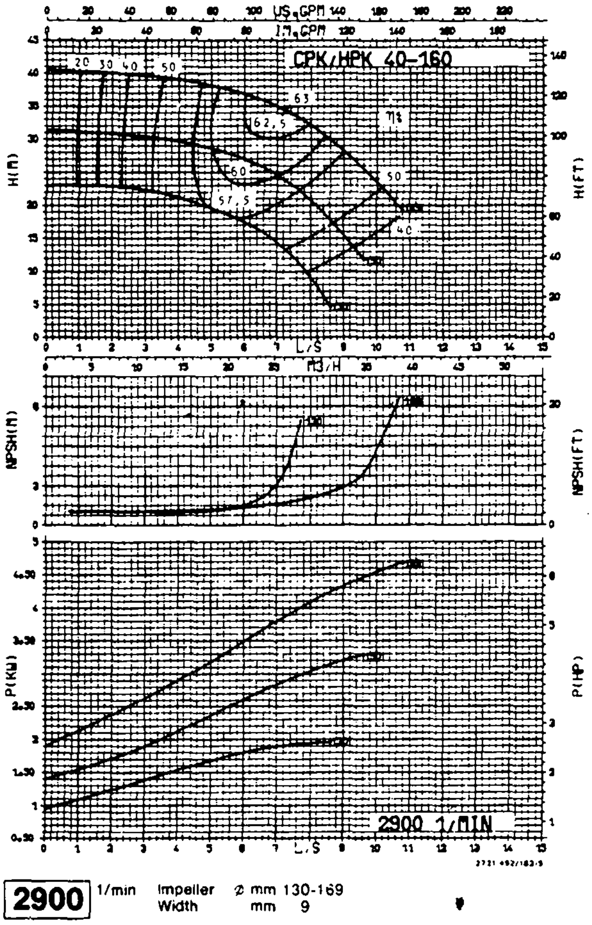


Fig. 24: Characteristic curve of a single-stage centrifugal pump at speed n = 2900 1/min

The path of these curves, which characterize the action of a centrifugal pump in operation, is determined largely by the specific speed - cf. fig. 25.

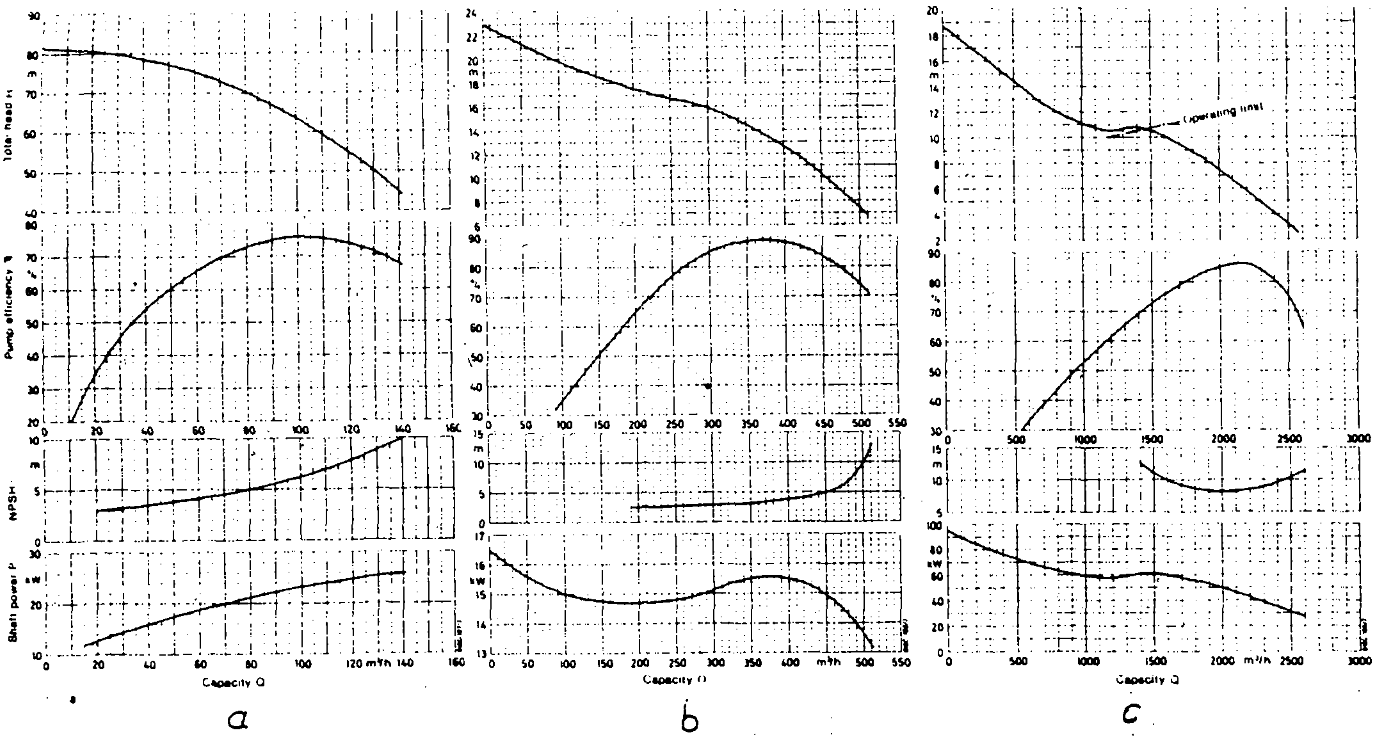


Fig. 25: a - characteristic curve of a radial-flow centrifugal pump, specific speed nq = 20 min^{-1}
b - characteristic curve of a mixed-flow centrifugal pump, specific speed nq = 80 min^{-1}
c - characteristic curve of an axial-flow centrifugal pump, specific speed nq = 200 min^{-1}

The H/Q curve shows how the delivery head and the capacity of a centrifugal pump alter. Generally speaking, the delivery head drops with increasing capacity. The delivery head ratio, i.e. the steepness of the curve, is determined by the type of pump and the shape of the impeller and is thus not freely adjustable.

H/Q curves in which the delivery head lessens with increasing discharge are referred to as "stable". In this case, each delivery head has only one corresponding discharge (cf. fig. 26-a). In contrast, a curve which has sections in which the delivery head increases with growing discharge is called "unstable". In such cases, two or sometimes more discharges belong to one delivery head (cf. fig. 26-b).

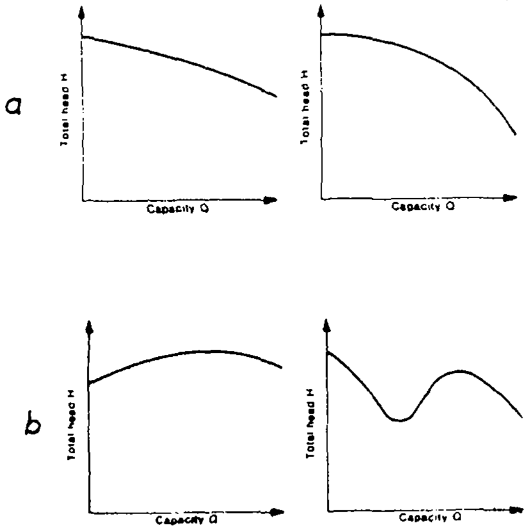


Fig. 26: a - stable curve
b - unstable curve

The path of the power consumption curve P_{Input} of a centrifugal pump also depends on its specific speed (cf. fig. 25). Peripheral pumps (displacement principle) have their highest power requirement at Q = 0. In the case of radial-flow pumps, the power consumption rises from P_0 = approx. 0.4 to 0.6 x $P_{Optimum}$ with the discharge. The highest power consumption of semi-axial pumps occurs at approx. $Q_{Opt.}$; at higher discharges P_{Input} drops again. The power consumption of axial-flow pumps is at its highest at Q = 0, falling with increasing discharge. Radial-flow pumps are therefore usually started up against closed stop valves, peripheral and axial-flow pumps, on the other hand, against open stop valves, in order to avoid overloading of the prime mover when the pump starts running.

The efficiency curve η rises with the capacity from zero up to a maximum level, then drops again at higher discharges. Unless other considerations outweigh, it will be expedient

to select the pump which has its highest efficiency $_{Opt.}$ near the rated capacity Q_N, i.e. $Q_N = Q_{Opt.}$
The path of the curve of the required NPSH, $NPSH_{req.}$, is also largely dependent on the specific speed.

2.2 NPSH (net positive suction head)

The NPSH is an important factor in the analysis of the suction action of a centrifugal pump, i.e. it allows an assessment of the relative likelihood of cavitation occurring to be made.

Cavitation is the occurrence of cavities or hollow spaces, resulting from the formation of gas bubbles, in a liquid in motion, followed by their sudden dispersal. If, in a flowing liquid, the static pressure drops to the vapour pressure belonging to the liquid's temperature, e.g. due to an increase in the absolute flow velocity or to a change in the geodetic head, gas bubbles form in the interior of the liquid at this point. They are transported along by the flowing liquid and collapse suddenly when the static pressure is again higher than the vapour pressure. This collapse occurs at high speed in the form of an implosion. At the start of the implosion, an indentation forms, in cavities nearer to the walls on the side farthest from the wall, and in cavities nearer to the centre of the flowing liquid on the side where the pressure is highest. As the indentation grows, a "micro-jet" of liquid forms, which then bursts the deformed bubbles into two or more parts. Where cavities are adhering to a wall or are near to one, this micro-jet then hits the surface of the wall at high speed, causing considerable damage to it. These pressure impacts are to be seen as the mechanical cause of cavitation erosion, whereby chemical aggression may add to the effect. On examination, the eroded areas can be seen to have a pitted, sponge-like structure.

In centrifugal pumps, cavitation may be caused especially by local drops in pressure in the entrance to the impeller, due to the higher flow velocity at this point. Even before cavitation occurs, the gas bubbles both lead to a drop in delivery head and efficiency and also, through their sudden collapse, cause the pump to run unevenly and noisily. To avoid, or at least reduce, the risk of cavitation in centrifugal pumps, static pressure must be in excess of the vapour pressure of the liquid at the entrance to the impeller. This difference between the absolute static pressure and the vapour pressure, plus the dynamic losses inside the pump, are determined on a test stand and given in the pump characteristic. Thus the $NPSH_{req.}$ of a pump indicates to what minimum extent the total pressure head in the plane of reference for the NPSH value, usually the impeller entry side, must be greater than the vapour pressure of the liquid in order to guarantee fault-free running of the pump at the rated velocity, rated discharge and with the liquid on which the calculations were based.

2.3 Action

The operating point of a pump is the point at which the delivery heads of pump and plant coincide - i.e. at which the characteristic curves of pump and plant intersect.

The plant (i.e. pipe) characteristic is the delivery head of the plant plotted against capacity. It usually consists

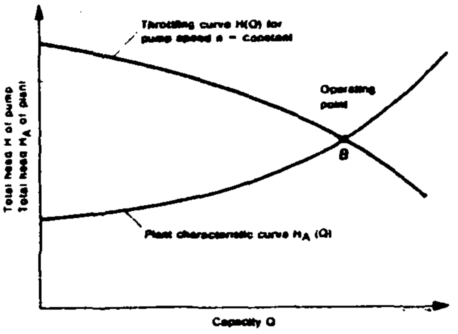


Fig. 27: Operating point of a centrifugal pump

of one part which is independent of discharge, the static head (the vertical distance between the water level on the suction side and on the delivery side) and another part which increases in proportion with the square of the discharge volume, the dynamic head - friction losses in the pipes. In special cases (e.g. circulating plants), the static head may equal zero.

The point of intersection thus gives the volume of liquid being pumped through the plant and the corresponding values for power consumption, efficiency and the required NPSH of the pump.

Generally speaking, the design and dimensioning of a pump depend primarily on discharge; the delivery head of the plant (= delivery head of the pump) is then calculated on the basis of the given conditions.

2.4 Operation in parallel/in series

In the case of centrifugal pumps operating in parallel (see fig. 28), the discharge volume Q_{I+II} is the sum of the individual discharges of pumps I and II at the same delivery head.

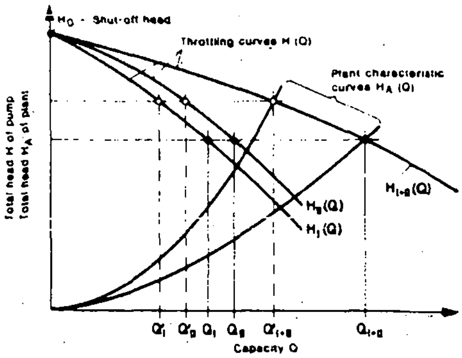


Fig. 28: Parallel operation of two centrifugal pumps I and II with stable H/Q curves

Centrifugal pumps operate in parallel without problems provided they have stable characteristic curves and, if possible, the same shut-off head. With regard to discharge, the pumps may well have different H/Q curves, however. Fig. 28 shows that when Q_{I+II} drops to Q'_{I+II}, the individual discharges Q_I and Q_{II} also drop to Q'_I and Q'_{II}. Fig. 29 shows that at different shut-off heads H_0 of the two pumps I and II, pump I is rapidly forced to shut-off head whilst pump II continues to deliver.

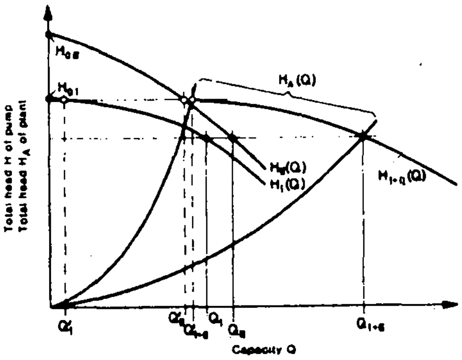


Fig. 29: Parallel operation of two centrifugal pumps I and II with different shut-off heads

Fig. 30 shows two unstable centrifugal pump characteristic curves H_I/Q and H_{II}/Q with the same shut-off head H_0. Pumps I and II can be operated in parallel in the range between 4 and 5, and any centrifugal pump of the same type may be added without problems. Due to the shut-off head H_0 in 4, one more pump of similar type can still just be connected. Between points 4,3,2,1 and nearly up to point 0, however, this is no longer possible. This pump, due to its small shut-off head H_0 as compared with H_{I+II}, would not be able to open the non-return valve with the pressure exerted by the other pumps on it.

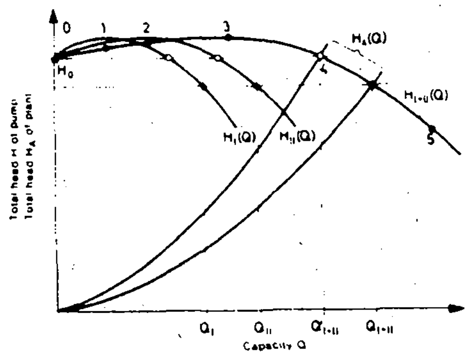


Fig. 30: Parallel operation of two centrifugal pumps I and II with unstable characteristic curves and the same peak delivery heads

Fig. 31 shows an instance of the parallel operation of two pumps I and II with unstable characteristic curves. Let the peak delivery head H_p of pump I be greater than that of ump II. If both centrifugal pumps now move from operating point 5 via 4 into the partial-load zone, discharge from pump II is abruptly checked from the point $H = H_{PII}$ down to even lower discharges, to $Q_{II} = 0$, provided there is a non-return valve in the pipe of pump II or, if this is not the case, to negative discharges Q_{II}, i.e. to a reverse flow through pump II. In the event e.g. of a further throttling of the plant characteristic curve down to a lower discharge in the range $H_I = H_{PII}$, in the small area to $Q_I = Q_{II} = 0$, pump II again operates in parallel with pump I. Successful operation in parallel is possible in the area of the characteristic curve H_{I+II}/Q between operating point 5 and H_{PII} only. Below this, down to even lower discharge, considerable fluctuations in Q_{II} and Q_{I+II} occur. Smooth running in parallel is no longer possible in this case.

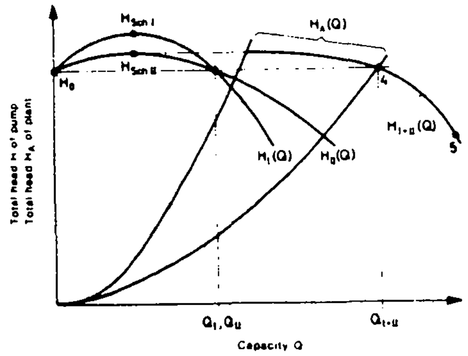


Fig. 31: Parallel operation of two centrifugal pumps I and II with unstable characteristic curves and unequal peak delivery heads

Where a piston pump is connected in parallel with a centrifugal pump, the individual discharge volumes are added together, i.e. the piston pump's virtually constant discharge Q_K is added to the centrifugal pump's discharge Q_I, as shown in fig. 32.

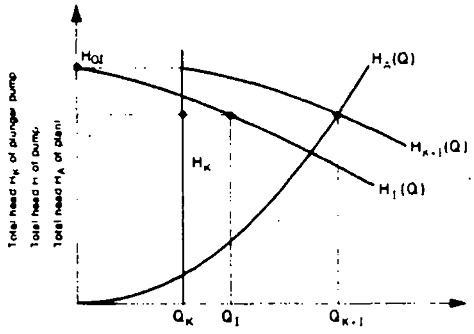


Fig. 32: Parallel operation of a piston pump with a centrifugal pump

Where the characteristic curve of the plant is very steep, the discharge can be increased more by connecting pumps in series than in parallel. Here, the discharge being the same, the delivery heads are added together. The same principle is utilized in multi-stage centrifugal pumps.

2.5 Adjustment to altered operation conditions

If a different capacity from that resulting as under 2.4 is required in the plant, there are various possible ways of moving the operating point. These methods are based either on an alteration of the plant characteristic curve H_A/Q, e.g.

by throttling
by introducing a bypass

or on an alteration of the pump characteristic curve H/Q, e.g.

by adjusting the speed
by altering the diameter of the impeller
by adjusting the angle of a grid in front of the impeller
by producing an initial spin directly in front of the impeller by means of a directed bypass flow
by adjusting the impeller vanes
by blocking the impeller flow by means of a certain cavitation volume
by sharpening the ends of the vanes.

3 Principles of the installation and operation of pumping equipment

3.1 Design of suction and gravity feed pipes, intake structures

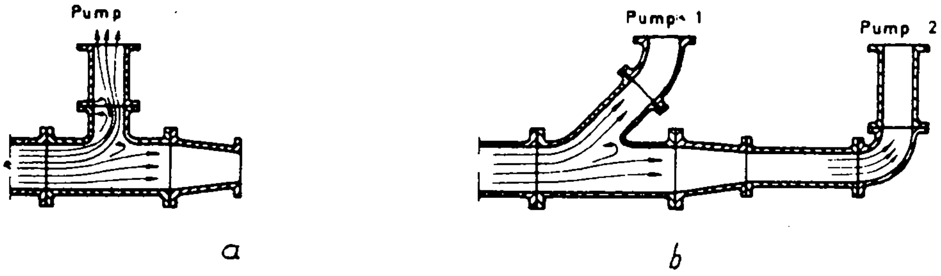
Suction pipes must always be laid horizontally or else sloping steadily upwards to the pump, in order to avoid accumulations of gas or air. The pipes must be completely water-tight and it should be possible to release the air from them without difficulty. If conical connecting pieces are necessary, these should be of non-concentric design. Gravity feed pipes should be laid either horizontally or sloping steadily downwards to the pump.

Sudden alterations of cross-section and sharp bends should be avoided. Poorly-designed suction or feed pipes (e.g. where there are bends on several planes directly before the pump's suction nozzle) can have a considerably detrimental effect on the pump's performance.

In the case of double-flow pumps, care must be taken to ensure that admission to both sides of the impeller is of equal volume. There must therefore be a straight pipe run at least 2 x DN long between a necessary bend and the pump's suction nozzle, to balance the flow.

If several pumps of the same type are connected to a common suction or feed pipe, it must be ensured, through appropriate design, that flow conditions are the same before each pump.

Right-angled branches are undesirable, even if followed by a straight run of pipe before the pump is reached. Better flow conditions are obtained with branches curving gently away from the pipe run (sweep tees).



a

b

Fig. 33: a - Poorly-designed branch
b - Satisfactory connection of 2 pumps

Flow velocities (approximate):
in the suction pipe v_s approx. 1 to 2 m/s
in the gravity feed pipe v_s approx.
1.5 to 2.5 m/s
} max. 3 m/s

Stop valves in the suction or feed pipe must remain fully open during operation and may not be used for control purposes.

Stop valves in the suction pipe are best installed with a horizontal spindle or a spindle projecting vertically downwards, to avoid the formation of air pockets in the spindle hood.

The spindle of the stop valve should further be sufficiently well sealed to reliably prevent the admission of air.

If the pump is drawing the liquid out of a tank and there is no suction strainer with foot valve, a trumpet-shaped inlet housing (inlet nozzle) must be provided. The floor- and wall-clearance of the suction strainer or inlet nozzle should be such that the liquid can enter the suction pipe freely and without hindrance from all sides.

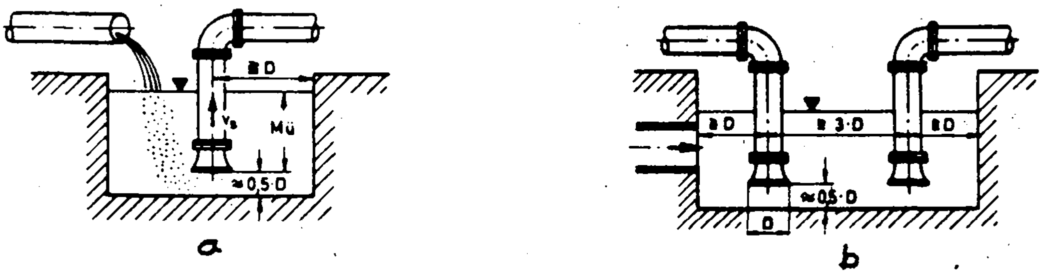


Fig. 34: a - Suction tank with open feedline
b - Suction tank with two suction pipes

If the feed pipe does not end underneath the surface of the liquid in the tank but above it, as shown in fig. 34a, there is a risk of the liquid entraining air. This could have a detrimental effect on the pump's performance and should therefore be avoided if possible. If not possible, however, it is advantageous to increase the distance the liquid travels between its point of entry and the suction opening, either by installing bulkheads or selecting a relatively large minimum cover of water, so that the air can escape again - this may involve higher construction costs, however. At a flow velocity v_s of approx. 2 m/s, for instance, a minimum cover of approx. 1.3 m is necessary.

The intake chamber of a vertically installed centrifugal pump should be designed so that a continuous flow of liquid to the pump can be guaranteed for every operational state and whatever the level in the chamber. This is especially important in the case of specifically fast-running pumps (mixed-flow and axial/propeller), since these are more affected by an uneven flow of liquid than radial-flow pumps.

Smooth operation of the pump is achieved when the flow of liquid to the pump's impeller has low spin and a uniform speed-profile over the complete cross-section of the intake nozzle. In addition, the formation at lowest levels of the liquid of air-drawing vortices in the intake chamber must be avoided. If these conditions are not met, a drop in the pump's discharge and efficiency must be expected. Under particularly unfavourable conditions, damage resulting from vibration or cavitation may occur.

Open intake chambers. Fig. 35 gives recommendations on the principle dimensions of tanks where only one pump is to be installed. These must be taken as approximate figures only. The channel should have a uniform cross-section for at least 5 x D before the pump. The flow velocity in the intake channel should not exceed 0.5 m/s. The reference quantity D is equivalent to the outside diameter of the inlet casing of a vertical mixed-flow or axial-flow impeller pump.

The minimum cover is measured as the distance between the lower edge of the inlet casing and the lowest possible low-water level. When vertical pumps are used, there is no generally valid figure that can be given for this minimum cover. It must be given by the pump manufacturer individually in each specific case.

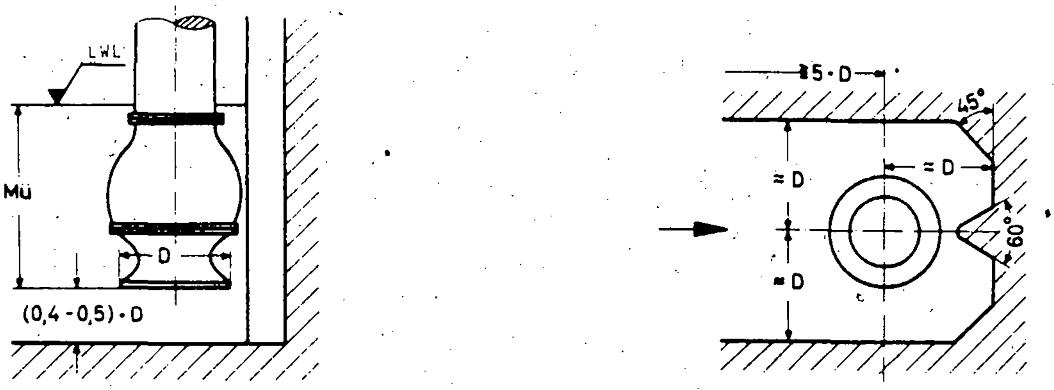


Fig. 35: Approximate dimensions of open intake chambers for single pumps

If several pumps have to be installed in one chamber, the best solution is to divide the chamber into several compartments, one for each pump.

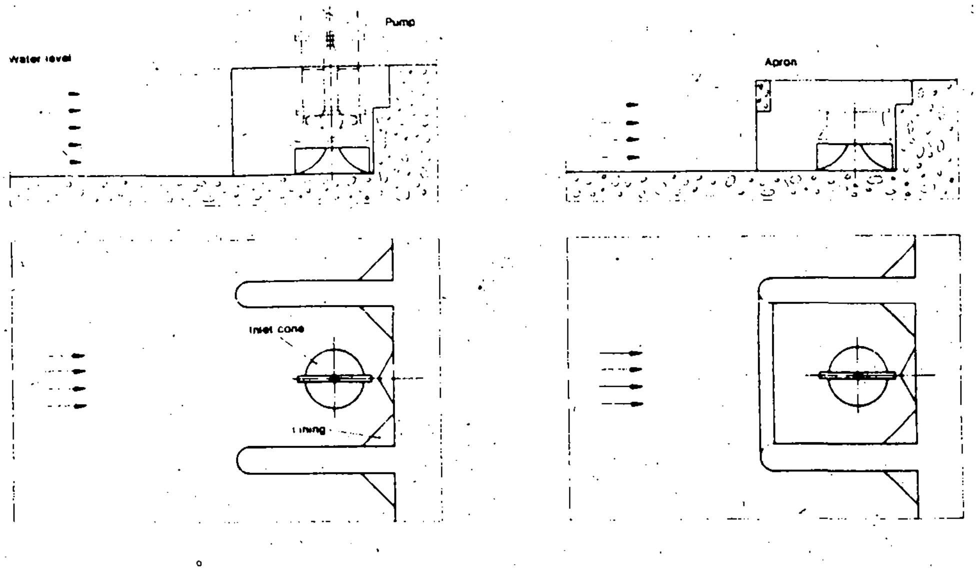


Fig. 36: Open intake chambers with several pumps in one chamber

If this arrangement is not possible, bulkheads (baffles) must be provided or else certain minimum clearances between the inlet casings observed. The measures taken in each case should always first be discussed with the pump manufacturer.

Common mistakes in the design of intake chambers:

Constructions as in fig. 37, where the fluid is fed in at one end of the tank. Flow to the individual pumps is then uneven; also the pumps have a reciprocal effect on each other.

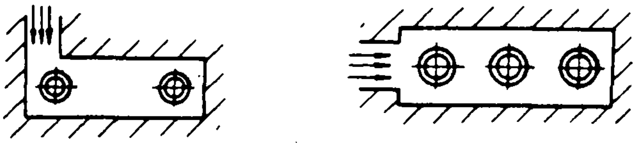


Fig. 37: Incorrect arrangement of the pumps in an intake chamber

Several pumps arranged asymmetrically in a chamber.
Sudden widening or narrowing of the feed channel.
Feed channel too short with a uniform cross-section.
Pipes or stages on the tank floor just in front of the pump.
Not enough floor clearance.
Open feeding of fluid into tank, so that entrained air may cause pump to run unevenly.

Covered intake chambers. If the general design of the plant does not allow the feed channel to be made long enough to ensure smooth running of the pump (i.e. 1 ≥ 5 x D), one possible solution is to provide the chamber with a cover. Such covers have a highly successful spin-reducing effect. General guidelines on the principal dimensions are given in fig. 38. The exact dimensions in each specific case should be obtained from the pump manufacturer.

If it is not possible to avoid an expansion of the open intake chamber (e.g. sloping side walls, inclination of the floor running towards the inlet nozzle), an appropriately-designed cover can help to produce the acceleration of the intake flow necessary for an equalization of the velocity profile.

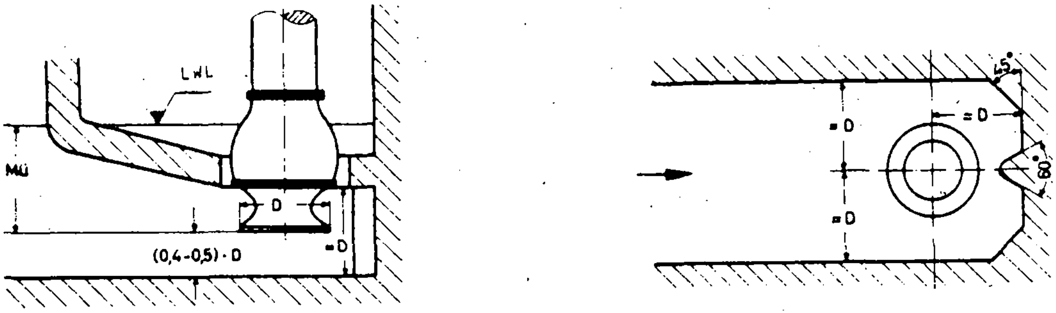


Fig. 38: Approximate principal dimensions of a covered intake chamber

Intake elbows. The most compact dimensions can be achieved with the use of "intake elbows" which, like turbine elbows, function as accelerators. At an acceleration of the flow velocity of 4 to 5 times the initial speed, a length of approx. 4 times the diameter of the impeller entrance is sufficient to achieve uniform velocity before the impeller.

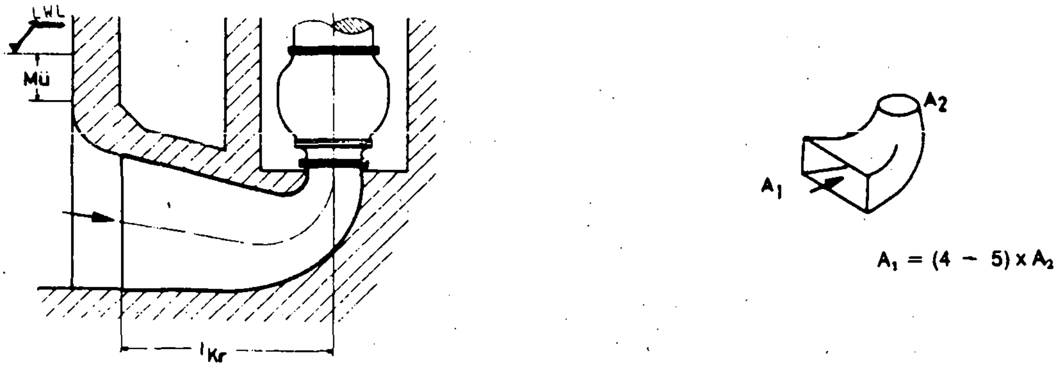


Fig. 39: Principal dimensions of an intake elbow

The cross-sectional area of the entry into the elbow (A_1) should be large enough (and thus the admission velocity of the liquid low enough) to prevent air funnels forming and air being sucked in.

However, it is advisable to carry out a cost/benefit analysis for each separate case, to determine whether the higher construction costs of this alternative, as compared with an intake chamber, are justified.

3.2 Installation of centrifugal pumps

Generally speaking, the installation of a centrifugal pump covers mounting of the pumping unit (pump plus prime mover) at its point of application, together with all the connecting pipework necessary for its functioning. The unit must be installed in such a way that all external forces can reliably be carried by the foundation, columns, frame, bases, or even the pipes.

Depending on the type of pump and often also on the field of application, installation will fall into one of the following categories: installation on or without a foundation, dry or submerged, indoor or outdoor.

Where pumps are installed on a foundation, a further distinction must be made between pumps with horizontal and with vertical shaft. Decisive for safe and reliable running of the pump are the foundation's strength and vibrational properties (high-frequency/low-frequency). A general distinction can be made between concrete foundations on which a base common to centrifugal pump and prime mover is mounted; and concrete or steel console foundations mounted on special elements which provide optimum vibration insulation. Such elements include e.g. cork mats, rubber or spring components. Each machine is attached separately to the console by its own base or frame.

Horizontal pumps and their prime movers, with the exception of block pumps, are supplied mounted together on one base. Larger units, and sometimes also gear driven/feed pump units have bases which are constructed in several sections. The bases are then mounted on the foundation and the spaces between them filled in with a sealing compound, e.g. cement mortar. The foundation screws must not be tightened until the compound has set. This procedure gives the bases the necessary strength to resist any undesired deformation under loading, e.g. due to pipe expansion forces. The next step is to align the couplings of the separate parts of the unit by inserting adjustment plates underneath the feet of the machines on the base. There follows stress-free connection of the pipes to the pump, installation of accessories, lubricating devices, filters etc. The process of installing the pump ends with an alignment check.

Vertical pumps and their prime movers are installed by a similar procedure. The coupling does not have to be aligned, however, if pump and drive are directly connected by means of a drive lantern. In such cases the pump is fixed to the foundation by means of a base flange only. The drive lantern then determines the exact position of the drive.

Installation without a foundation is the rule wherever the weight of the machines and the anticipated stresses from the pipes are low, or wherever the pump has to remain mobile.

Submersible-motor pumps, most inline pumps and glandless pumps, especially circulating pumps, are installed without a foundation. Larger units are occasionally provided with an additional simple column foundation.

Mobile pumps, e.g. fire-fighting or mobile pipeline pumps, are fixed either directly or together with their bases to the frame (wheel- or skid-mounted). This construction requires the connected pipes to be flexible. Where larger units are mounted in this way, the bases must be rigid enough to keep distortions down to an acceptable level.

Portable pumps (e.g. for use in gardens or for pumping water out of cellars), are always constructed as self-contained units, needing no adjustment of the coupling. Since such pumps are always relatively small and connected to flexible hoses rather than to rigid pipework, no foundations are necessary. The pump can be stood on any firm, level surface.

Bases without a foundation are sometimes necessary in the case of acid pumps, to allow any aggressive fluid which may have leaked under the base to be removed.

In the case of vertical pumps in particular, a distinction is made between submerged (wet) applications and dry installation.

The main advantage of a submerged installation lies in the lower installation and construction costs. The pump is directly immersed in the liquid to be pumped - e.g. submersible-borehole and most tubular-casing pumps. Tubular-casing pumps can be installed either with the weight of pump and motor together supported by one floor of the construction or by different floors; also the delivery pipe may be laid above or below the floor of the pump house.

If it is necessary to inspect the pump - e.g. the seals - regularly from the outside, dry installation must be chosen.

The advantage of dry installation is that the pump chamber can then also be used for other purposes as well - e.g. the erection of other machines. This is e.g. the rule for the most common types of vertical ship's pumps. These are installed either in a standing arrangement, with the pump foot mounted on a steel frame in the floor of the engine-room (with pump and motor casing connected by a drive lantern), or in a wall-mounted construction, where the motor lantern is mounted on one of the engine-room walls. Dry installation is also the rule in the case of volute pumps used in other applications - e.g. for sewage pumping or in the petrochemical industry, also in the case of some other tubular-casing pumps.

Installation in the open, i.e. without the protection of a building or roof, requires certain other conditions besides those already given to be met. Climatic features such as rain (protection against corrosion, motors protected against spray), solar radiation (unilateral heat expansion, overheating of the drive unit), frost, flying sand, wind forces etc. have to be taken into account. Because conditions vary so widely, there are no stanardized rules for the outdoor installation of pumps.

3.3 Venting and priming of centrifugal pumps

The suction pipe of non-self-priming pumps and the pump itself must be filled with fluid before starting up. If the impeller or suction impeller is not surrounded by water when the pump is not running, it is expedient to provide a foot or non-return valve below the lowest level of the liquid. In this arrangement, which is restricted to low nominal diameters only, the foot valve, pump casing, pipes and other fittings must be dimensioned with an additional safety margin over and above the highest static pressure load. The reason for this is that the foot or non-return valve may suddenly be pulled closed by a return flow when the pump is switched off, so that pressure surges (water hammer) may occur. For this reason another non-return valve is often provided in the discharge pipe, to protect the pump. This arrangement only functions smoothly if the valve on the discharge side closes before that on the suction side.

Centrifugal pumps are considered to be self-priming if they are suitable for pumping liquids, gases and mixtures of liquids and gases. They must be able to extract air from the pump suction pipe without the aid of additional external extraction devices.

Pumps which are not self-priming and where the installation of a foot or non-return valve in the suction pipe is not possible are filled before the pump is started by an extraction of the air from the suction pipe and the pump. Borehole pumps are provided with a foot valve in spite of their submersion provided the shaft is mounted in water-lubricated rubber bearings and - where the pump is installed at a greater depth - provided some bearings are above the surface of the water. These bearings also need water during the starting-up phase. The foot valve keeps all bearings constantly surrounded by water.

In general, centrifugal pumps have to be filled with liquid prior to working. In plants where the liquid runs to the pump by gravity, care must be taken to ensure that the pump casing is adequately vented. The situation is different in plants with a geodetic suction lift. Unlike displacement pumps, standard-type centrifugal pumps can produce only a very low negative pressure if they are not filled with liquid. This means that they are not always able to vent their suction pipe without help. In suction operation, therefore, special measures must be taken before starting the pump to extract the air from pump and suction pipe. Centrifugal pumps installed above the water-level on the suction side are filled with the aid of venting equipment. In the case of an automatic venting system, the pipe to be vented is connected to the control tank by means of a pipe sloping steadily upwards. This contains two level-control switches which switch the vacuum pump on and off. To vent the pump casing and if necessary the discharge pipe up to the non-return valve, these chambers are connected to the air chamber of the control tank through intermediate switching of an air-relief valve. A double-acting ball check valve both maintains the vacuum after the venting pump has been switched off and also prevents water from overflowing into it if the control system should fail. The switch-on level in the control tank should be approx. 0.3 m above the highest level in the system to be vented.

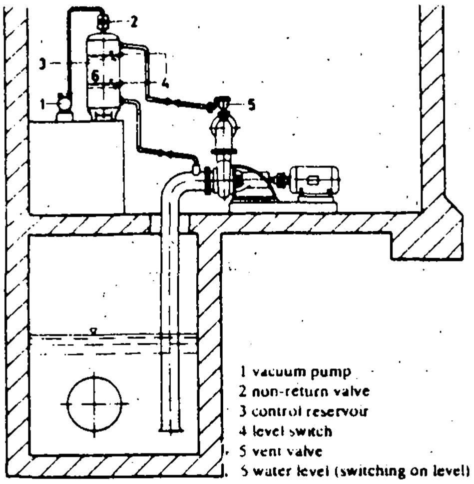


Fig. 40: Automatic venting system

To prevent the water level in the control tank from being too strongly affected when the centrifugal pump is started up, or if the level of the suction water alters, the height of the air chamber should be as low as possible (approx. 0.2 to 0.3 m). The difference in levels of the level switches should be once to twice the height of the air chamber.

Where gas-free liquid is being pumped, the diameter of the control tank is governed only by the accommodation of the level-control switches. If the exudation of dissolved gases has to be anticipated at negative pressure, the amounts of these must be determined and the necessary diameter of the control tank calculated under consideration of the permissible number of switchings for the vacuum pump.

Any number of pumps and suction pipes can be connected to one control tank, provided valves are installed in the connection pipes which are opened to evacuate air before each group of pumps is started up. In calculating the length of time required to release the air from suction pipes and siphons, the air volumes must be determined separately for the rising and for the horizontal sections of the pipe and then added together.

3.4 Alignment of couplings

Couplings must be properly aligned to ensure smooth running of the pump units and also to prevent damage to the transmission elements. First of all it must be ensured that the two halves of the coupling are spaced as specified (fig. 41).

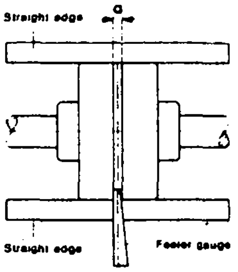


Fig. 41: Alignment of flexible couplings with straight edge and feeler gauge

Secondly, the centre lines of the shafts must align exactly at the coupling, without a break. Any lateral, vertical or angular non-alignment must be corrected.

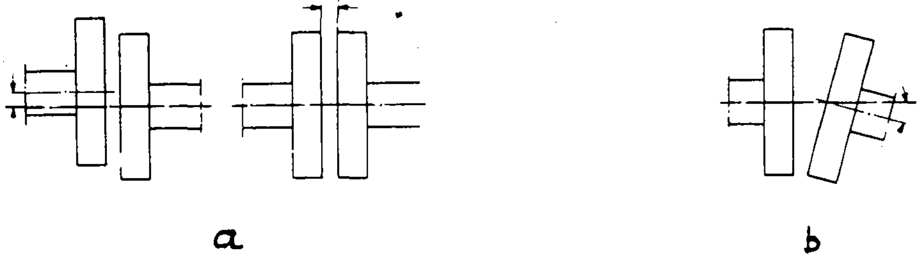


Fig. 42: a - Lateral/vertical non-alignment
b - Angular non-alignment

The easiest way of checking proper alignment is with a straight edge and a feeler gauge. A coupling is properly aligned if the straight edge, when laid across both halves of the coupling parallel to the shaft, is always the same distance away from the shaft, round the complete perimeter. Also, the axial spacing between the two halves of the coupling must be the same all round the perimeter of the coupling. A precise and rapid alignment can be achieved with the aid of an aligning device.

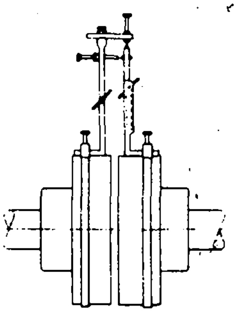


Fig. 43: Alignment of flexible couplings using aligning devices

The necessary accuracy of alignment of centrifugal pumps depends chiefly on the type of coupling and on the rotational speed.

When aligning couplings of hot-water pumps, the special instructions given by the manufacturer because of the possibility of thermal distortions must be followed with care.

After alignment of the coupling, it is advisable to fasten pump and prime mover to the base or foundation to prevent the machines from altering their position in operation.

3.5 Starting up

The process of starting up a machine assembly consisting of electric motor and pump - e.g. centrifugal pump - is affected by the following factors:

1. Type of electric motor: D.C. motors have good starting properties and seldom cause problems in this phase. Asynchronous motors with slip-ring rotors which are started up with starting resistors are also problem-free. Asynchronous motors with squirrel-cage rotors have various starting torque characteristics, depending on design and type of rotor, and always require adjustment to the specific application.

2. Type of pump: piston pumps (displacement pumps) necessitate high breakaway torques. Radial-flow centrifugal pumps require a relatively low torque. The counter-torque- i.e. the torque developed by the pump which opposes the motor torque - increases, however, with the square of the rotational speed. The extent of the counter-torque depends on whether the pump is started up with open or closed throttle.

3. Type of switching on and type of current: with direct switching to a "rigid grid", the full starting and run-up torque is available. Starting current is approx. 4.5 to 6 times the rated current. With direct switching to a "flexible grid", the mains voltage drops under the effect of the switching-on current consumed. The torque curve of the motor falls in an approximately quadratic

ratio to the drop in voltage. After run-up, the mains voltage recovers its full potential as soon as the current taken up by the motor has dropped back to the rated value.

The starting torque T_p is the torque required by the pump during starting to maintain the speed achieved. It climbs from zero at standstill up to the rated torque.
In contrast, the torque T_a taken up during starting via the coupling is primarily dependent on the motor. This driving torque is higher than the starting torque until the rated torque is reached. The difference between the two torques serves to accelerate all rotating masses in the complete machine assembly.

Since every possible operating point has a corresponding specific rotational speed and specific torque in the graph, the development of the starting torque is decisively influenced by the path along which the operating point of the pumps moves from zero (at standstill) to operating point B (at the rated speed). Four main possibilities can be disringuished (taking the example of a radial-flow pump with low specific speed):

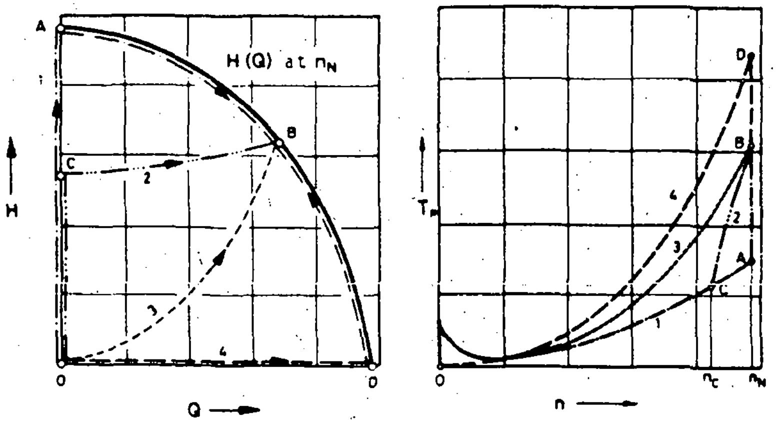


Fig. 44: H/Q characteristic curve and starting torque curve of a centrifugal pump with low specific speed

1. Starting up with stop valve closed, opening of stop valve after rated speed has been reached (curve 0-A-B). In the section 0-A, the hydraulic torque of the pump increases quadratically with the speed. This torque is increased through the superposition of the frictional torque of bearings and shaft seals, which represents a relatively large part of the starting torque at low speeds. At n = 0, frictional torque is particularly high due to the static friction, also known as the breakaway troque. Normally it makes up approx. 5 to 10 % of the rated torque. In the case of overhung-mounted pumps with a high feed pressure, it may also reach the magnitude of the rated torque.

In section A-B, the starting torque rises or falls - now at approximately constant speed - depending on the pump's power intake. Peripheral and axial-flow pumps, where at Q = 0 the power requirement, and so also the starting torque, are greater than at the operating point, will therefore not be started up by this method.

2. Starting up with open stop valve against a purely static delivery head exerted against a non-return valve (curve 0-C-B): here the development of the starting torque up to point C is the same as with a closed stop valve, since the non-return valve cannot be opened by the pump pressure (thereby beginning delivery) before this point. The further development of the torque curve is determined by plotting intermediate curves for the section C-B for various speeds and calculating the torque from the power input.

3. Starting up with open stop valve against a purely dynamic delivery head of the plant (curve 0-B): this path results only if the pipe is very short. The starting torque T_P increases - apart from the frictional torque - quadratically with the speed from zero up to the rated torque. If the pipe is very long, however, the time required to accelerate

the water mass is much longer than the starting-up time of the pump. The water mass at rest then functions as a closed stop valve and the development of the starting torque is more similar to that in case 1.

4. Starting up with stop valve open and delivery pipe empty (curve 0-D-B): as long as the delivery pipe remains empty, the pump is required to produce virtually no delivery head. If the time needed for the pipe to fill up is considerably longer than the pump's starting-up time, the starting-up process will follow the path 0-D and the torque thereby increase in proportion to the square of the speed.

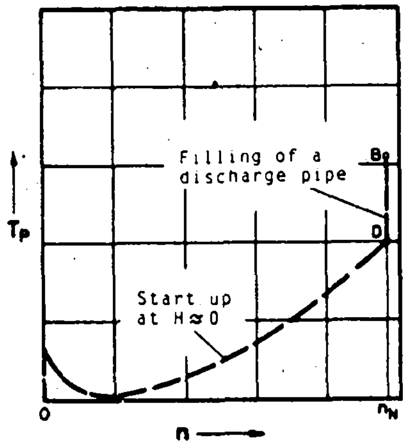


Fig. 45: Starting torque curve with open stop valve and empty delivery pipe - axial-flow pump

Section D-B will again depend on the pump's power intake. For this reason, starting-up along the line 0-D-B is favourable for pumps where the power input decreases with increasing discharge volume: If the delivery pipe already fills to a considerable extent as the speed increases, the path of the torque will be similar to that in case 3.

If an axial-flow pump is to start running with the delivery pipe full, the use of a bypass pipe parallel to the pump, open during the starting-up process, can mean that discharge into the delivery pipe is avoided - i.e. the pump is started up against minimum delivery head.

When starting peripheral pumps in plants with an empty delivery pipe, the stop valve should not be completely opened, so that discharge is restricted and cavitation in the pump avoided.

4 Maintenance of pumps

Pumps must be inspected and serviced at certain set intervals. The operating instructions for each pump give exact maintenance schedules, including the type and extent of the work to be carried out. Sometimes service contracts are drawn up for certain pieces of equipment.

Regular maintenance of pumps and motors prolongs their service life and saves costs in the long run.

External servicing of pumps and prime movers: the machines must be cleaned externally at regular intervals. Note that pumps and motors may on no account be cleaned with a water jet (i.e. hosed down). Any rust should be removed and the affected area treated. Uncoated parts should be coated with a preservative grease.

Maintenance of bearings: bearings should be serviced regularly. Since lubricants are consumed with varying rapidity, these must either be topped up or renewed (oil change, topping-up or complete renewal of grease). The intervals between lubrication and the type and grade of lubricant to use are given in the operating instructions.

Maintenance of shaft seals: this depends on the type of seal. In the case of packed stuffing boxes, it must be remembered that these should always drip slightly in operation. Packed stuffing boxes which seal completely cause damage to the sealing elements - shaft, shaft sleeves. In this case, the nuts of the stuffing-box screws should be loosened until the stuffing box drips slightly. The same is true of metal, sleeve and plastic packings.

In the case of mechanical (floating ring) seals, leakages are usually not externally visible. Separate servicing is not necessary. If comparatively extensive leakages occur, the sealing elements should be replaced. Further details are given in the operating instructions.

Other maintenance work:

Checking to see that centrifugal pump and prime mover are running smoothly; where appropriate tightening of the V-belt.

Checking engine performance with regard to power consumption.

Inspection of the coupling alignment.

Inspection of the flexible transmission elements for wear (coupling plate, coupling bolts, coupling buffers).

Checking the supply of flushing and sealing water.

Inspection of pressure-relieving equipment (balancing disc, high-pressure pumps) where appropriate.

Checking functioning of the automatic grease guns and grease pipes, lubricant pumps.

4.1 Checklist

This list is intended as a general guide only and does not claim to cover all eventualities:

- Check to see that pump and prime mover are running smoothly.
- Check performance data, measuring and entering current consumption.
- Check oil or grease supply (bearings).
- Inspect shaft seal (packed stuffing box, mechanical seal, shaft sealing rings).
- In the case of packed stuffing boxes, re-pack if necessary. Mechanical seals: replace carbon ring seal and rotating slide ring if necessary.

- Inspect alignment of coupling.
- Check flexible transmission elements for wear (coupling plate, bolts, buffers).
- Check stop and non-return valves for correct functioning.
- Check outside of equipment for corrosion, interior pump parts for erosion and cavitation.
- Carry out a trial run after servicing.
- Determine spare parts requirement and order where appropriate.
- Check supply of flushing and sealing water.
- Check installed solenoid valves, filters, screens etc.
- Check pressure-relieving equipment for wear.
- Inspect electric control devices.
- Check functioning of automatic grease guns and pipes.
- Look for wear through hand-hole cover.
- Measure rotor clearance.
- Check throttle slit.

5 Repair of pumps

The following list is intended to help in recognizing and dealing with any faults which may occur. It chiefly covers problems with centrifugal pumps.

Table 1: Fault - Cause - Remedy

Fault	Key to cause/remedy
Pump discharge too low	1, 2, 3, 4, 5, 6, 7, 8, 9, 10, 11, 18, 28
Prime mover overloaded	12, 13, 14, 15, 20, 27, 28
Pump discharge pressure too high	15

Bearing temperature too high	22, 23, 24, 25, 26
Pump leaking	16, 29
Too much leakage from shaft seals	17, 18, 19, 20, 21, 22, 23, 33
Pump running unevenly	3, 6, 11, 12, 22, 23, 25, 30, 31, 32
Undue increase of temperature in the pump	3, 6, 32

See below for list of causes and remedies.

Cause/remedy

1. Pump delivering against excessive pressure.
- Open stop valve wider until operating point is adjusted.

2. Counter-pressure too high.
- Install a larger impeller (consult manufacturer first).
- Increase speed (turbine, internal combustion engine, gearbox mechanism).

3. Pump or pipe not fully vented/not primed.
- Vent or fill.

4. Feed pipe or impeller blocked
- Remove deposits/foreign matter from interior of pump and/or pipe.

5. Formation of air-pockets in the pipe.
- Alter pipe lay-out.
- Install air-relief valve.
- Inspect screens, suction strainer.

6. NPSH of the plant (feed) too low.
- Check level of the liquid and correct if necessary.
- Open stop valve in the feed pipe fully.
- Possibly alter feed pipe if resistances in it are too high.

7. Suction lift too high.
- Clean suction strainer and suction pipe.
- Correct level of liquid.
- Alter design of the suction pipe.

8. Air being drawn in at the stuffing box.
- Clean sealing liquid channel, if necessary introduce other sealing liquid or increase its pressure.
- Replace shaft seal.

9. Wrong rotational direction.
- Change over 2 phases of the current (alternating current). Observe indicator of pressure-gauge (higher pressure = correct rotational direction).

10. Speed too low.
- Increase rotational speed.
- Increase voltage.
- This problem may also be solved by installing an impeller with a larger diameter (consult manufacturer).

11. Interior parts worn.
- Replace worn parts.

12. Counter-pressure of the pump is lower than that specified or calculated when designing the plant.
- Adjust operating point precisely by means of the stop valve in the delivery pipe.
- If overloading is constant, possibly trim the impeller (consult manufacturer first).

13. Liquid has a higher density or viscosity than was assumed when designing the plant.
- Consult the manufacturer and re-dimension the equipment.

14. Gland too tight or not straight.
- Adjust.

15. Speed too high.
- Reduce speed (turbine, internal combustion engine etc.).
- Consult manufacturer - diameter of impeller may have to be altered.

16. Defective seal.
- Replace seal between cooling chamber cover and stuffing-box casing.

17. Shaft seal worn.
- Inspect shaft seal, replace if necessary.
- Check pressure of flushing/sealing liquid.

18. Shaft sleeve scored or rough.
- Replace.

19. Not enough coolant, or dirt accumulation in coolant chamber.
- Increase amount of coolant.
- Clean coolant chamber and tubes.
- Clean/replace coolant.

20. Gland, end cover, seal cover wrongly tightened, wrong sealing material used.
- Alter, replace.

21. Pump running unevenly.
- Adjust suction conditions.
- Align pump correctly.
- Re-balance rotor.
- Increase pressure on suction side.

22. Poor alignment.
- Inspect coupling, re-align if necessary.

23. Stress distortion of pump.
- Inspect pipe connections and pump mounting.

24. Undue axial thrust.
- Clean pressure-relieving holes in the impeller.
- Replace split rings.

25. Too much, not enough or unsuitable lubricant.
- Top up, reduce or replace lubricant.

26. Coupling spacing incorrect.
- Adjust spacing as specified in installation plan.

27. Operating voltage too low.

28. Running on two phases.

- Replace faulty fuse.
- Inspect lead connections.

29. Connecting screws too loose.

- Tighten.
- Replace washers.

30. Rotor running out of true.

- Clean rotor.
- Rebalance.

31. Damaged bearing.

- Replace.

32. Insufficient discharge.

- Increase minimum discharge.

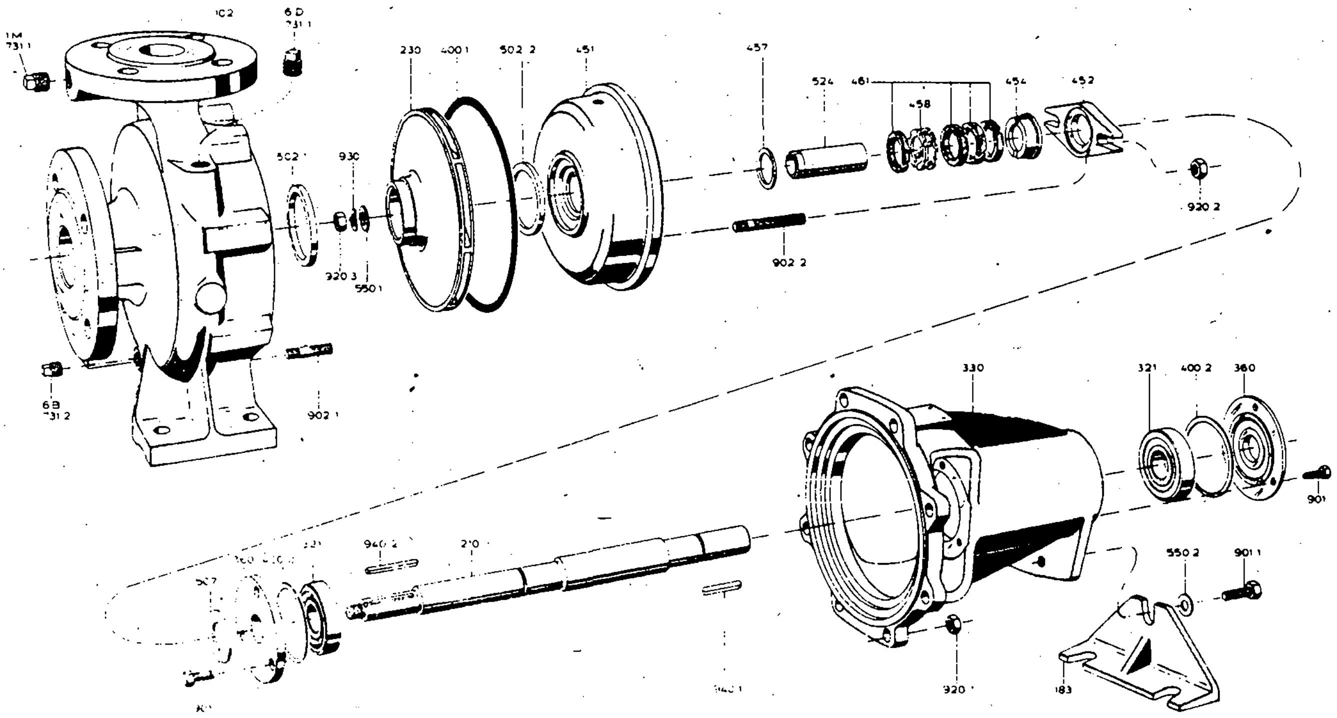
33. Fault in the feed of circulation liquid.

- Increase free cross-section.

Further details are given in the operating instructions provided by the pump manufacturer.

The components of a single-stage, horizontal volute pump with e.g. radial-flow impeller are shown in the drawing below.

Fig. 46: Components of a volute pump with non-cooled packed stuffing box



Part No.	Nomenclature	Part No.	Nomenclature
102	Volute casing		
		502.1	Split ring (only in impeller with relief holes)
183	Supporting foot		
210	Shaft		
230	Impeller	507	Oil ring
321	Grooved ball bearing	524	Shaft sleeve
330	Bearing flange		
360	Bearing cover	550.1/.2	Disc
400.1/.2/.3	Gasket	731.1/.2	Threaded plug
		901/.1	Hexagon-head screw
433	Slide ring seal		
451	Stuffing-box housing	902.1/.2/.3	Stud bolt
452	Gland	920.1/.2/.3	Hexagonal nut
454	Stuffing-box ring, divided	930	Spring washer
		940.1/.2	Key
457	Base ring		
458	Locking ring, divided		
461	Stuffing-box packing		

Revised:

Fig. 34: Mü = Min. cover

Fig. 35: Lowest water level Minimum cover

Fig. 38: Lowest water level

Fig. 39: Lowest water level Minimum cover

Fig. 44: H/Q at n_N

Fig. 45: Filling of discharge pipe
Starting up at H = 0

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