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PUMP CLINIC 12

RADIAL & AXIAL THRUST IN CENTRIFUGAL PUMPS

Radial Thrust

A centrifugal pump consists of a set of rotating vanes, enclosed within a housing or casing and used to impart energy to a fluid through centrifugal force. Thus, stripped of all refinements, a centrifugal pump has two main parts:

1. A rotating element including an impeller and shaft
2. A stationary element made up of a casing, stuffing box and bearings

In a centrifugal pump the liquid is forced by atmospheric or other pressure into a set of rotating vanes. These vanes constitute an impeller which discharges the liquid at its periphery at a higher velocity. This velocity is converted to pressure energy by means of a volute *(Fig 1.1)* or by a set of stationary diffusion vanes *(Fig 1.2)* surrounding the impeller periphery. Pumps with volute casings are generally called volute pumps, while those with diffusion vanes are called diffuser pumps.

Diffuser pumps were once quite commonly called turbine pumps, but this term has recently been more selectively applied to the vertical deep-well centrifugal diffuser pumps usually referred to as vertical turbine pumps.

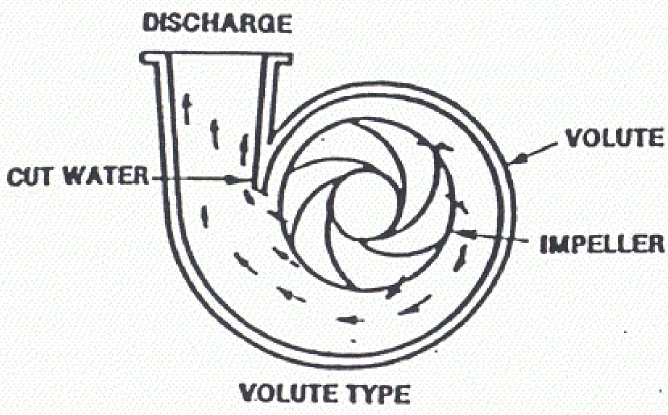

Fig.1.1

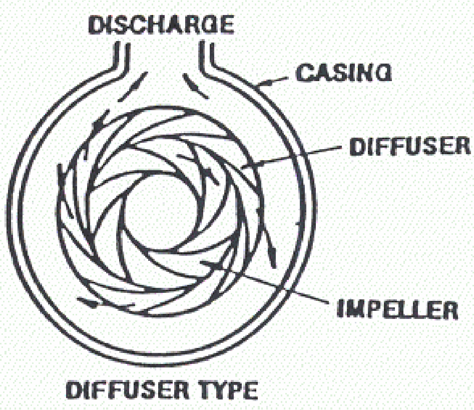

Fig.1.2

The diffuser is seldom applied to a single-stage radial-flow pump. Except for certain high-pressure multi-stage pump designs, the major application of diffusion vane pumps is in vertical turbine pumps and in single-stage low-head propeller pumps.

In a single-volute pump casing design *(Fig 1.3)* uniform or near-uniform pressures act on the impeller when the pump is operated at design capacity (which coincides with the best efficiency). At other capacities, the pressures around the impeller are not uniform and there is a resultant radial reaction. A graphical representation of the typical change in this force with pump capacity is shown in *(Fig 1.4)* - NOTE that the force is greatest at shut-off.

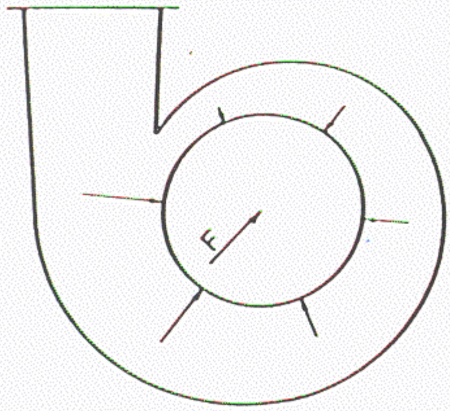

Fig.1.3 **Radial reaction in a single-volute casing**

Uniform pressures do not exist at reduced capacities.

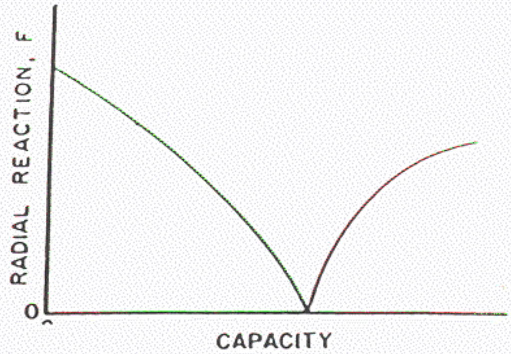

Fig.1.4 **Magnitude of radial reaction in single-volute casing**

F decreases from shut-off to design capacity and then increases with over-capacity. With over-capacity, the reaction is roughly in the opposite direction from that with partial capacity.

For any percentage of capacity, radial reaction is a function of total head and of the width and diameter of the impeller. Thus a high-head pump with a large-diameter impeller will have a much greater radial reaction force at partial capacities than a low-head pump with a small-diameter impeller.

A zero radial reaction is not often realised; the minimum reaction occurs at design capacity. In a diffuser-type pump which has the same tendency for over-capacity unbalance as a single-volute pump, the reaction is limited to a small arc repeated all around the impeller with the individual forces cancelling each other.

In a centrifugal pump design, shaft diameter and bearing size can be affected by allowable deflection as determined by shaft span, impeller weight, radial reaction forces and the torque to be transmitted. Formerly, standard designs compensated for reaction forces if maximum-diameter pump impellers were used only for operations exceeding 50% of design capacity.

For sustained operations at lower capacities, the pump manufacturer, if properly advised, would supply a heavier shaft, usually at a much higher cost. Sustained operation at extremely low flows, without informing the manufacturer at the time of purchase, is a much more common practice today. The result is broken shafts, especially on high-head units.

Because of the increasing operation of pumps at reduced capacities, it has become desirable to design standard units to accommodate such conditions. One solution is to use heavier shafts and bearings. Except for low-head pumps in which only a small additional load is involved, this solution is not economical. The only practical answer is a casing design that develops a much smaller radial reaction force at partial capacities. One of these is the double-volute casing design, also called twin-volute or dual-volute.

The application of the double-volute design principle to neutralise reaction forces at reduced capacity is illustrated in *(Fig 1.5).* Basically, this design consists of two 180º volutes; a passage external to the second, joins the two into a common discharge. Although a pressure unbalance exists at partial capacity through each 180º arc, forces F1 and F2 are approximately equal and opposite, thereby producing little, if any, radial force on the shaft and bearings.

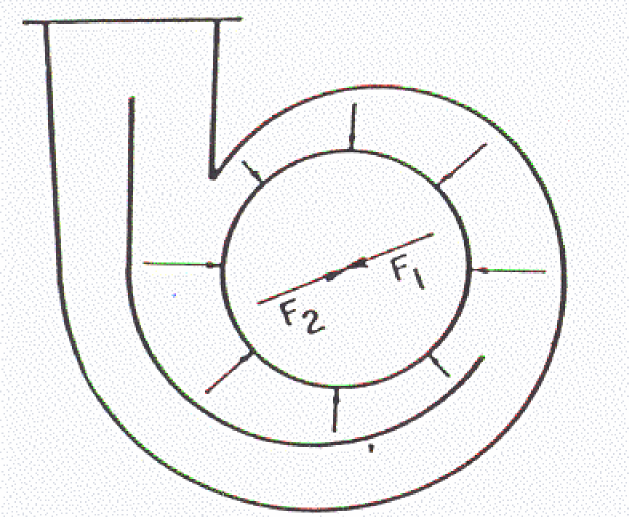

Fig.1.5 Radial reactions in double-volute pump

Axial Thrust in Single-Stage Pumps

The pressures generated by a centrifugal pump exert forces on both its stationary and rotating parts. The design of these parts balances some of these forces, but separate means may be required to counter-balance others. Axial hydraulic thrust is the summation of unbalanced impeller forces acting in the axial direction. As reliable large-capacity thrust bearings are not readily available, axial thrust in single-stage pumps remains a problem only in larger units.

Theoretically, a double-suction impeller is in hydraulic axial balance with the pressures on one side equal to, and counter-balancing the pressures on, the other *(Fig 2.1).* In practice, this balance may not be achieved for the following reasons:

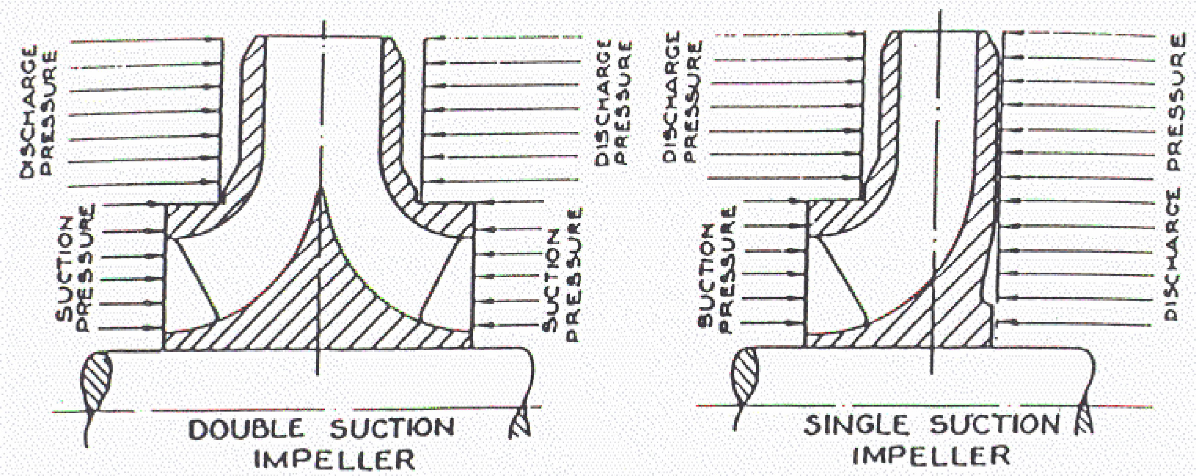

Fig. 2.1 Origin of pressure acting on impeller shrouds to produce axial thrust

The suction passages to the two suction eyes may not provide equal or uniform flows to the two sides.

1. External conditions such as an elbow being too close to the pump suction nozzle may cause unequal flows to the suction eyes.
2. The two sides of the discharge casing may not be symmetrical, or the impeller may be located off-centre. These conditions will alter the flow characteristics between the impeller shrouds and casing, causing unequal pressures on the shrouds.
3. Unequal leakage through the two leakage joints will tend to upset the balance.

Combined, these factors create definite axial unbalance. To compensate for this, all centrifugal pumps, even those with double-suction impellers, incorporate thrust bearings.

The ordinary single-suction radial-flow impeller with the shaft passing through the impeller eye *(Fig 2.1)* is subject to axial thrust because a portion of the front wall is exposed to suction pressure, thus exposing relatively more back wall surface to discharge pressure. If the discharge chamber pressure were uniform over the entire impeller surface, the axial force acting towards the suction would be equal to the product of the net pressure generated by the impeller and the unbalanced annular area.

Actually, pressure on the two single-suction impeller walls is not uniform. The liquid trapped between the impeller shrouds and casing walls is in rotation and the pressure at the impeller periphery is appreciably higher than at the impeller hub. Although we need not be concerned with the theoretical calculations for this pressure variation, *(Fig 2.2)* describes it qualitatively. Generally speaking, axial thrust towards the impeller suction is about 20% to 30% less than the product of the net pressure and the unbalanced area.

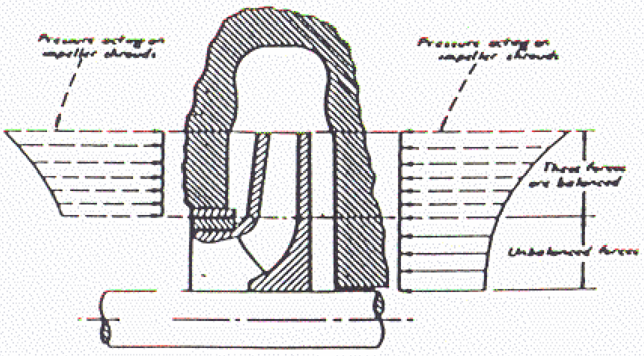

Fig.2.2 **Actual pressure distribution on front and back shrouds of single-suction impeller with shaft through impeller eye**

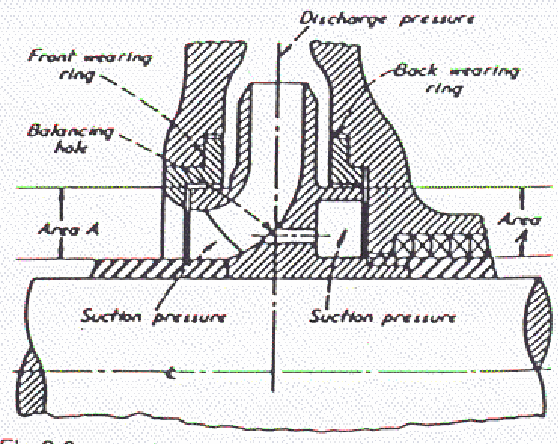

Fig.2.3 **Balancing axial thrust of single-suction impeller with wearing ring on the back and balancing holes**

To eliminate the axial thrust of a single-suction impeller, a pump can be provided with both front and back wearing rings. To equalise thrust area, the inner diameter of both rings is made the same *(Fig 2.3).*   Pressure approximately equal to the suction pressure is maintained in a chamber located on the impeller side of the back wearing ring by drilling so-called balancing holes through the impeller. Leakage past the back wearing ring is returned into the suction area through these holes.

However, with large single-stage suction pumps, balancing holes are considered undesirable because leakage back to the impeller suction opposes the main flow, creating disturbances. In such pumps, a piped connection to the pump suction replaces the balancing holes.

Another way to eliminate or reduce axial thrust in single-suction is by use of pump-out vanes on the back shroud. The effect of these vanes is to reduce the pressure acting on the back shroud of the impeller *(Fig 2.4).* This design, however, is generally used only in pumps handling gritty liquids where it keeps the clearance space between the impeller back shroud and the casing free of foreign matter.

So far, the discussion of the axial thrust has been limited to single-suction impellers with a shaft passing through the impeller eye and located in pumps with two stuffing boxes, one on either side of the impeller. In these pumps, suction pressure magnitude does not affect the resulting axial thrust. On the other hand, axial forces acting on an overhung impeller with a single stuffing box *(Fig 2.5)* are definitely affected by suction pressure.

In addition to the unbalanced force found in a single-suction, two-box design *(Fig 2.2)* there is an axial force equivalent to the product of the shaft area through the stuffing box and the difference between suction and atmospheric pressure. This force acts towards the impeller suction when the suction pressure is less than the atmospheric, or in the opposite direction, when it is higher than the atmospheric.

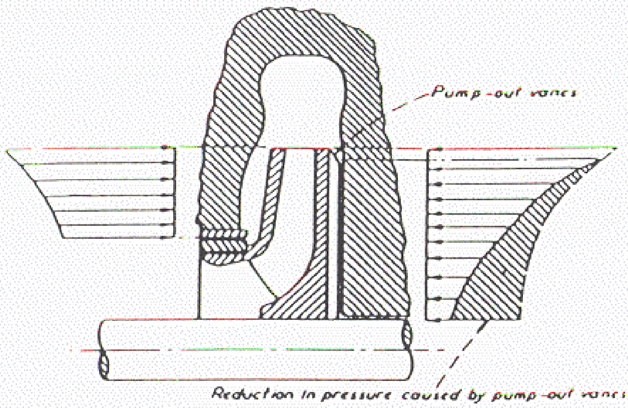

Fig 2.4 **Reducing axial thrust of single-suction impeller with pump-out vanes**

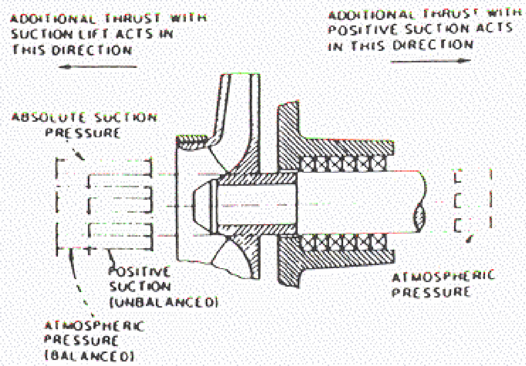

Fig 2.5 Axial thrust problem with single suction overhung impeller and single stuffing box

When an overhung impeller pump handles a suction lift, the additional axial force is very low. For example; if the shaft diameter through the stuffing box is 2" (area = 3.14 sq.in) and if the suction lift is 20ft of water (absolute pressure – 6.06 psia), the axial force caused by the overhung impeller and acting towards the suction will be only 27lb.

On the other hand, if the suction pressure is 100 psi, the force will be 314lb and acts in the opposite direction. Therefore, as the same pump may be applied for many conditions of service over a wide range of suction pressures, the thrust bearing of pumps with single-suction overhung impellers must be arranged to take thrust in either direction. They must also be selected with sufficient thrust capacity to counteract forces set up under the maximum suction pressure established as a limit for that particular pump.

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Axial Thrust in Multistage Pumps

It might seem that the advantages of balanced axial thrust and greater available suction area in a double-suction impeller would warrant applying such impellers to multistage pumps. But there are definite shortcomings to this practice. The average multistage pump has relatively low capacity when compared to the entire range covered by modern centrifugal pumps.

It is seldom necessary, therefore, to use double-suction impellers just to reduce the net positive suction head (NPSH) required for a given capacity. Even if a double-suction impeller is desirable for the first stage of a large capacity multistage pump, it is hardly necessary for the remaining stages. As to the advantage of the axial balance it provides, it must be considered that a certain amount of axial thrust is actually present in all centrifugal pumps and the necessity of a thrust bearing is therefore not eliminated.

Most important, the use of double-suction impellers in a multistage pump adds needless length to the pump shaft span. Additional space is required for the extra passage leading to the second inlet of each successive stage. In a pump with four or more stages *(Fig 2.6)* this increase becomes quite appreciable and causes additional casting difficulties. If shaft diameter is increased to compensate for the longer span so as to maintain reasonable shaft deflection, the impeller inlet areas are correspondingly reduced.

The result is that the advantage of superior suction conditions usually offered by double-suction impellers is considerably reduced. Finally, as it is impractical to arrange the various double-suction impellers in any but the ascending order of the stages, the impeller at one end of the casing becomes the last stage impeller and the pressure acting on the adjacent stuffing box becomes the discharge pressure on the next-to-last stage.

To reduce this pressure, a pressure-reducing bushing must be interposed between the last-stage impeller and the stuffing box and this bushing further increases the overall length. The result of all these considerations is that most multistage pumps are built with single-suction impellers.

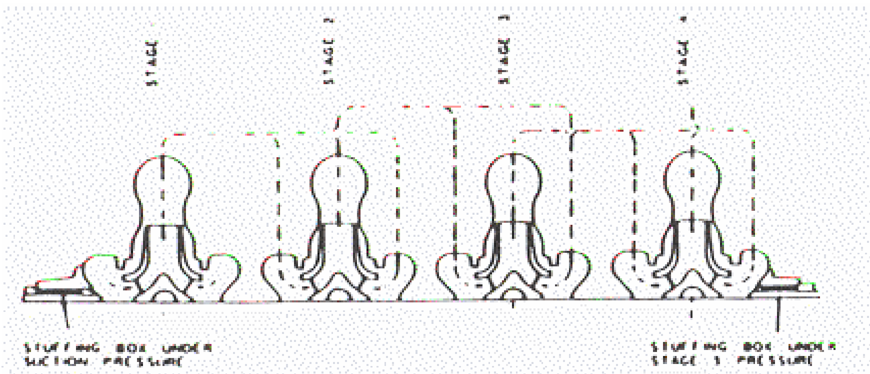

Fig.2.6 Four-stage pump with double-suction impellers

Two obvious single-suction impeller arrangements for a multistage pump are as follows:

Several single-suction impellers may be mounted on one shaft, each having its suction inlet facing in the same direction and its stages following one another in ascending order of pressure *(Fig 2.7).* The axial thrust is then balanced by a hydraulic balancing device.

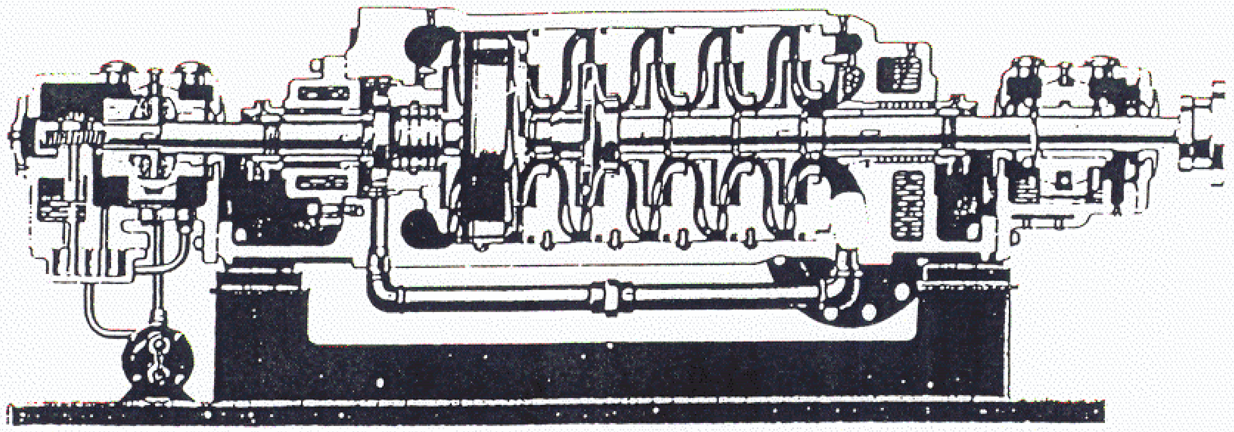

Fig.2.7 Multistage pump with single-suction impellers facing in one direction and hydraulic balancing device

An even number of single-suction impellers can be mounted on one shaft, one half of these facing in an opposite direction to the second half. With this arrangement, axial thrust on the one half is compensated by the thrust in the opposite direction on the other half *(Fig 2.8).* This mounting of single-suction impellers back-to-back is frequently called 'opposed impellers'.

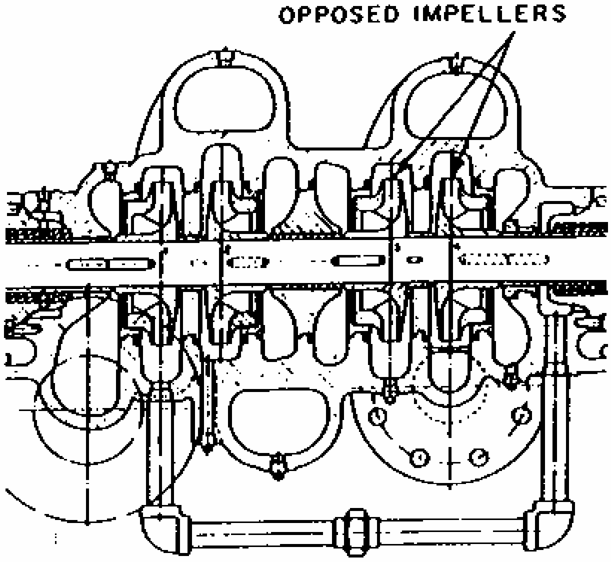

Fig.2.8 Four-stage pump with opposed impellers

An uneven number of single-suction impellers may be used with this arrangement, provided the correct shaft and interstage bushing diameters are used to give the effect of an hydraulic balancing device that will compensate for the hydraulic thrust on one of the stages. It is important to note that the opposed impeller arrangement completely balances axial thrust only under the following conditions:

1. The pump must be provided with two stuffing boxes.
2. The shaft must have a constant diameter.
3. The impeller hubs must not extend through the interstage portion of the casing separating adjacent stages.

Except for some special pumps that have an internal and enclosed bearing at one end, and therefore only one stuffing box, most multistage pumps fulfil the first condition. But because of structural requirements, the last two conditions are not practical. A slight residual thrust is usually present in multistage opposed-impeller pumps, unless impeller hubs or wearing rings are located on different diameters for various stages.

Because such a construction would eliminate axial thrust only at the expense of reduced inter-changeability and increased manufacturing costs, this residual thrust, being relatively small, is usually carried on the thrust bearing.

Hydraulic Balancing Devices

If all the single-suction impellers of a multistage pump face in the same direction, the total theoretical hydraulic axial thrust acting towards the suction end of the pump will be the sum of the individual impeller thrusts. The thrust magnitude (in pounds) will be approximately equal to the product of the net pump pressure (in pounds per square inch) and the annular unbalanced area (in square inches). Actually, the axial thrust turns out to be about 70% to 80% of this theoretical value.

Some form of hydraulic balancing device must be used to balance this axial thrust and to reduce the pressure on the stuffing box adjacent to the last-stage impeller. This hydraulic balancing device may be a balancing drum, a balancing disk or a combination of the two.

Balancing Drums

The balancing drum is illustrated in *(Fig 3.1).* The balancing chamber at the back of the last stage impeller is separated from the pump interior by a drum that is either keyed or screwed to the shaft and rotates with it. The drum is separated by a small radial clearance from the stationary portion of the balancing device, called the ‘balancing drum head’ which is fixed to the pump casing.

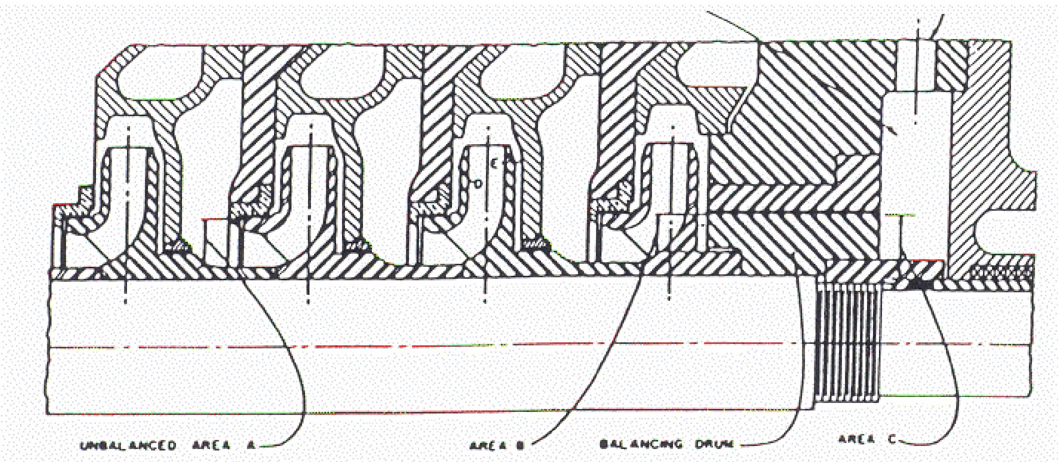

Fig.3.1 Balancing drum.

The balancing chamber is connected either to the pump suction or to the vessel from which the pump takes its suction. Thus the back pressure in the balancing chamber is only slightly higher than the suction pressure, the difference between the two being equal to the friction losses between this chamber and the point of return. The leakage between the drum and the drum head is, of course, a function of the differential pressure across the drum and of the clearance area.

The forces acting on the balancing drum in *(Fig 3.1)* are the following:

1. Toward the discharge end; the discharge pressure multiplied by the front balancing area (Area B0) of the drum.
2. Toward the suction end; the back pressure in the balancing chamber multiplied by the back balancing area (Area C) of the drum.

The first force is greater than the second thereby counter-balancing the axial thrust exerted upon the single-suction impellers. The drum diameter can be selected to balance axial thrust completely or within 90% to 95% depending on the desirability of carrying any thrust-bearing loads.

It has been assumed in the preceding simplified description that the pressure acting on the impeller walls is constant over their entire surface and that the axial thrust is equal to the product of the total net pressure generated and the unbalanced area. Actually, this pressure varies somewhat in the radial direction because of the centrifugal force exerted upon the water by the outer impeller shroud *(Fig 2.4).*

Furthermore, the pressures at two corresponding points on the opposite impeller faces *(D and E Fig 3.1)* may not be equal because of variation in clearance between the impeller wall and the casing section separating successive stages. Finally, pressure distribution over the impeller wall surface may vary with head and capacity operating conditions.

This pressure distribution and design data can be determined by test quite accurately for any one fixed operating condition and an effective balancing drum could be designed on the basis of the forces, resulting from this pressure distribution. Unfortunately, varying head and capacity conditions change the pressure distribution, and as the area of the balancing drum is necessarily fixed, the equilibrium of the axial forces can be destroyed.

The objection to this is not primarily the amount of the thrust but rather that the direction of the thrust cannot be pre-determined because of the uncertainty about internal pressures. Still, it is advisable to pre-determine normal thrust direction as this can influence external mechanical thrust-bearing design. Because 100% balance is unattainable in practice, and because the slight but predictable unbalance can be carried on a thrust bearing, the balancing drum is often designed to balance only 90% to 95% of total impeller thrust.

The balancing drum satisfactorily balances the axial thrust of single-suction impellers and reduces pressure on the discharge side stuffing box. It lacks, however, the virtue of automatic compensation for any changes in axial thrust caused by varying impeller reaction characteristics. In effect, if the axial thrust and balancing drum forces become unequal, the rotating element will tend to move in the direction of the greater force.

The thrust bearing must then prevent excessive movement of the rotating element. The balancing drum performs no restoring function until such time as the drum force again equals the axial thrust. This automatic compensation is the major feature that differentiates the balancing disk from the balancing drum.

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Balancing Disks

The operation of the simple balancing disk is illustrated in *(Fig 3.2).* The disk is fixed to, and rotates with, the shaft. It is separated by a small axial clearance from the balancing disk head which is fixed to the casing. The leakage through this clearance flows into the balancing chamber and from there, either to the pump suction or to the vessel from which the pump takes its suction.

The back of the balancing disk is subject to the balancing chamber back pressure, whereas the disk face experiences a range of pressures. These vary from discharge pressure at its smallest diameter to back pressure at its periphery. The inner and outer disk diameters are chosen so that the difference between the total force acting on the disk face and that acting on its back will balance the impeller axial thrust.

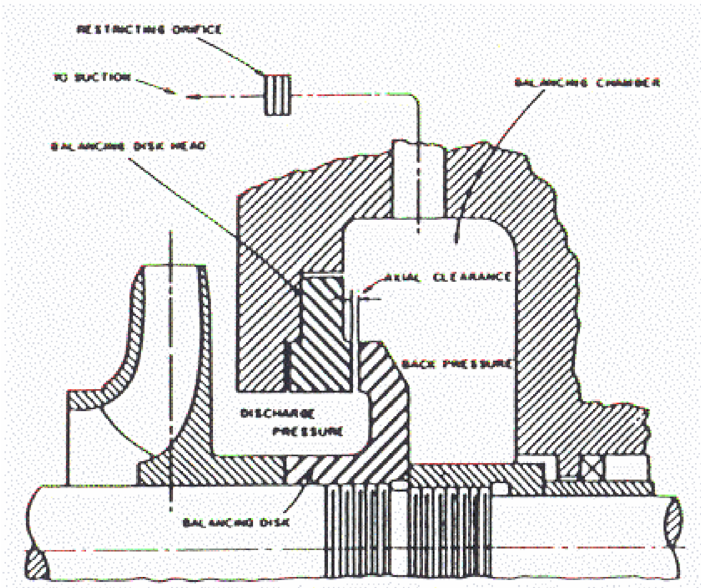

Fig.3.2 Simple balancing disk

If the axial thrust of the impellers should exceed the thrust acting on the disk during operation, the latter is moved towards the disk head, reducing the axial clearance between the disk and the disk head. The amount of leakage through the clearance is reduced so that the friction losses in the leakage return line are also reduced lowering the back pressure in the balancing chamber.

This lowering of pressure automatically increases the pressure difference acting on the disk and moves it away from the disk head, increasing the clearance. Now the pressure builds up in the balancing chamber and the disk is again moved towards the disk head until an equilibrium is reached.

To assure proper balancing disk operation, the change in back pressure in the balancing chamber must be of an appreciable magnitude. Thus, with the balancing disk wide open with respect to the disk head, the back pressure must be substantially higher than the suction pressure to give a resultant force that restores the normal disk position. This can be accomplished by introducing a restricting orifice in the leakage return line that increases back pressure when leakage past disk increases beyond normal.

The disadvantage of this arrangement is that the pressure on the stuffing box packing is variable - a condition that is injurious to the life of the packing and therefore to be avoided. The higher pressure that can occur at the packing is also undesirable.

Combination Balancing Disk and Drum
For the reasons just described, the simple balancing disk is seldom used. The combination balancing disk and drum *(Fig 3.3)* was developed to obviate the shortcomings of the disk while retaining the advantage of automatic compensation for axial thrust changes.

CENTRIFUGAL PUMPS

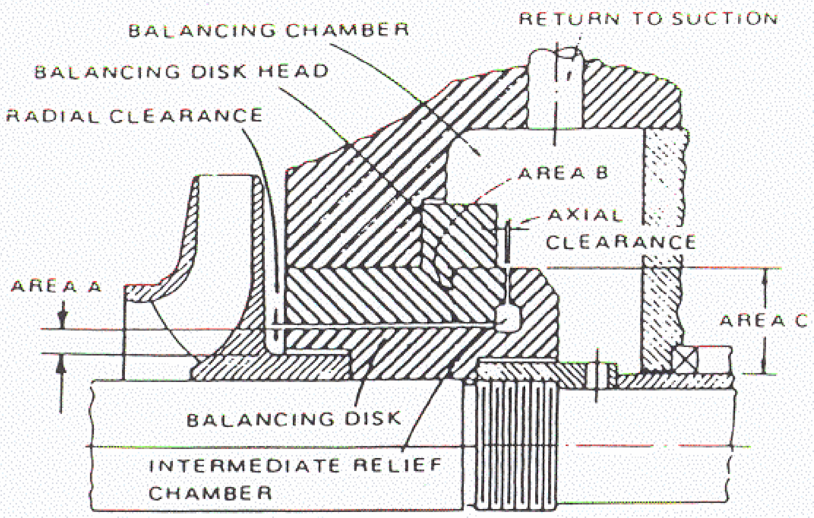

Fig.3.3 Combination balancing disk and drum

The rotating portion of this balancing device consists of a long cylindrical body that turns within a drum portion of the disk head. This rotating part incorporates a disk similar to the one previously described. In this design, radial clearance remains constant regardless of disk position, whereas the axial clearance varies with the pump rotor position.

The following forces act on this device:

1. Towards the discharge end; the sum of the discharge pressure multiplied by Area A, plus the average intermediate pressure multiplied by Area B.
2. Towards the suction end; the back pressure multiplied by Area C.

Whereas the position-restoring feature of the simple balancing disk required an undesirably wide variation of the back pressure, it is now possible to depend upon a variation of the intermediate pressure to achieve the same effect.

Here is how it works. When the pump rotor moves towards the suction end (to the left in *Fig 3.3)* because of increased axial thrust, the axial clearance is reduced and pressure builds up in the intermediate relief chamber, increasing the average value of the intermediate pressure acting on Area B. In other words, with reduced leakage, the pressure drop across the radial clearance decreases, increasing the pressure drop across the axial clearance.

The increase in intermediate pressure forces the balancing disk towards the discharge end until equilibrium is reached. Movement of the pump rotor towards the discharge end would have the opposite effect, increasing the axial clearance and the leakage, and decreasing the intermediate pressure acting on Area B.

There are now in use numerous hydraulic balancing device modifications. One typical design separates the drum portion of a combination device into two halves, one preceding and the second following the disk *(Fig 3.4).* The virtue of this arrangement is a definite cushioning effect at the intermediate relief chamber thus avoiding too positing a restoring action which might result in the contacting and scoring of the disk faces.

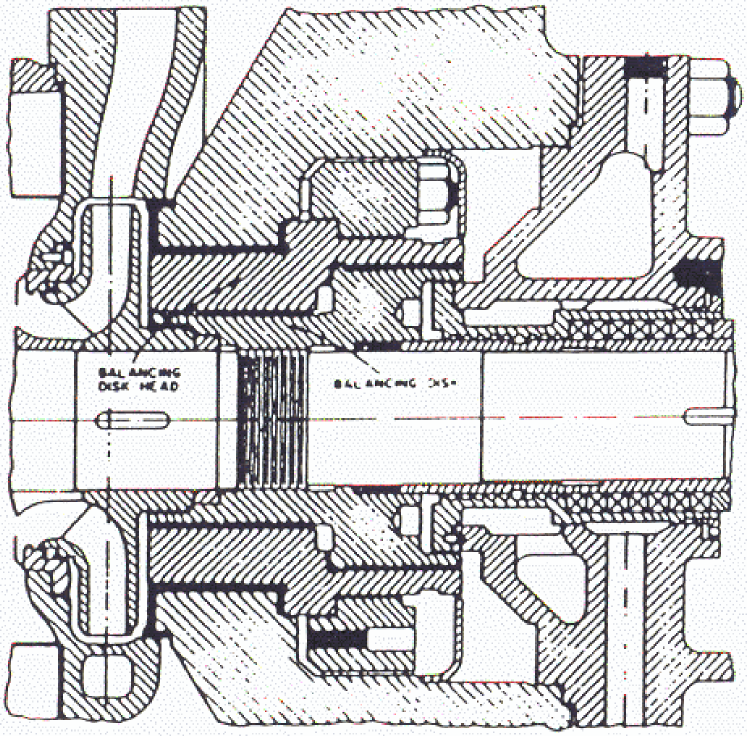

Fig.3.4 Combination balancing disk and drum with disk located in center portion of drum Worthington Pump

Pump Handbook ed.Igor J. Karassik, William C. Krutzsch, Warren H. Fraser, Joseph P. Messina (USA:1986)
Igor J. Karassik, Roy Carter, Centrifugal Pumps (USA:1960)