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2 Sheets-Sheet 1

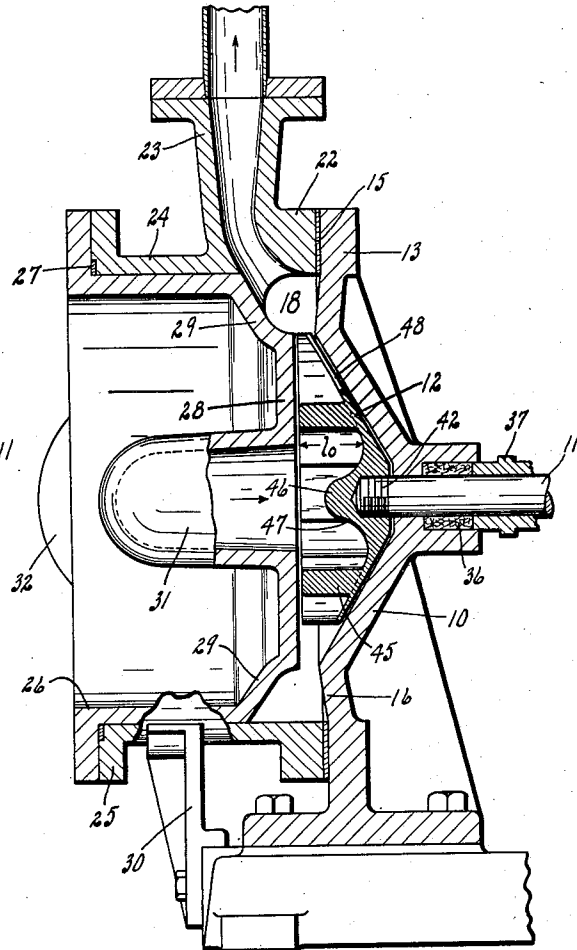


FIG.-2

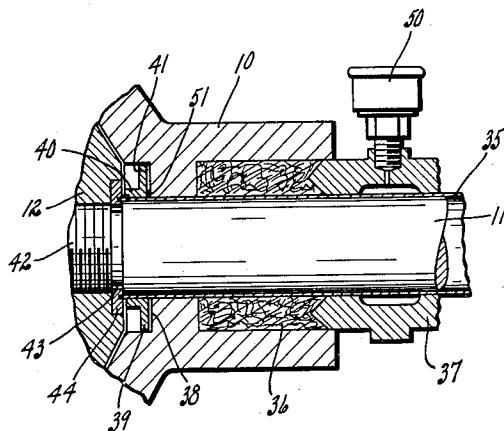


FIG.- 4

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CENTRIFUGAL PUMP

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2 Sheets-Sheet 2

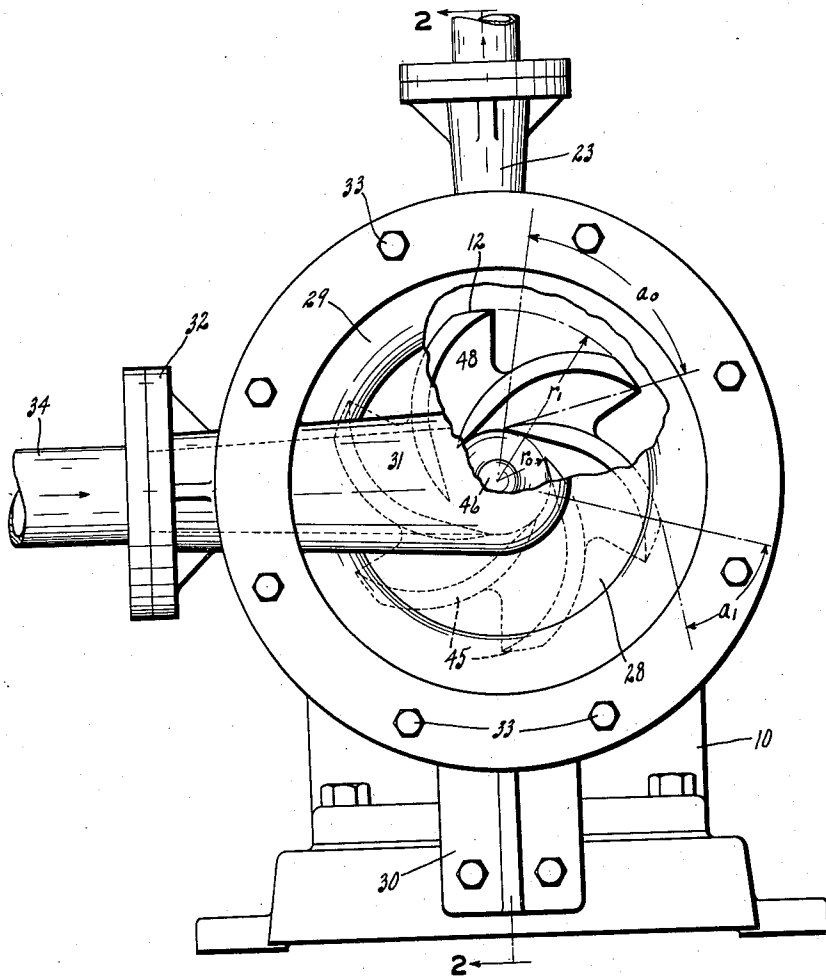


FIG.-3

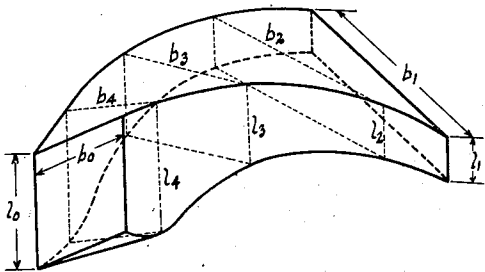


FIG.- 6

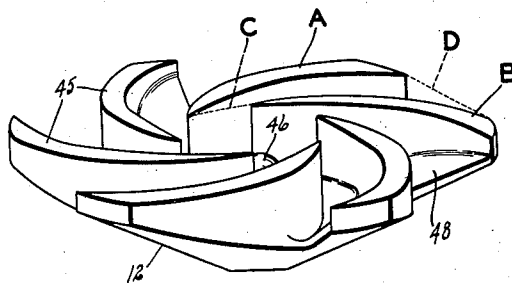


FIG.-5

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CENTRIFUGAL PUMP

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Application December 13, 1934, Serial No. 757,366

6 Claims. (Cl. 103—103)

This invention relates to centrifugal pumps and more particularly to pumps of that type especially designed for the handling of corrosive and erosive fluids or fluids containing occluded gases.

5 The use of centrifugal pumps in chemical industries has been limited because of various disadvantages which have been regarded as inherent in this type of pump. In operation with fluids of the nature specified, it is a common fault of 10 centrifugal pumps that a great deal of wear occurs due to cavitation or pitting and eating away of the impeller which is due largely to the formation of eddies within the impeller and pump casing and to the high velocities employed. It is one of the 15 objects of the present invention to provide a centrifugal pump in which the parts are so constructed and proportioned that eddies between the impeller vanes and within the impeller casing are greatly minimized and the velocities kept 20 at a low value.

A further difficulty in the application of centrifugal pumps in chemical industries has been the lack of sufficient suction lift so that the usual centrifugal pump when employed in these locations first shows a great loss of pumping capacity as attempt is made to increase the pump output, whereupon the pump loses its prime and is operated in unstable equilibrium.

Our invention provides a pump so constructed that an extremely high suction lift is developed, actual practice having shown suction of twenty-seven inches of mercury or higher.

It is inherent in any pump handling corrosive and erosive liquids that the impeller may require frequent change. It is a further object of the present invention to provide a pump casing so arranged that access is provided for changing 35 impellers with the removal of very few casing parts and with the parts so arranged that suction lines need not be disturbed.

Our improved centrifugal pump also provides novel means for protecting the drive shaft and associated parts from corrosive fluids and for sealing the shaft against leakage.

45 A further object of the present invention is to provide a pump having a back plate supporting a rotatable shaft and impeller thereon and which is adapted to form a part of a standard pump having the usual front housing but which can quickly be converted to a pump of our improved type in which the front housing is so arranged as to permit easy access for changing impellers and without breaking the long suction lines.

Other objects and advantages of our invention will appear from the drawings and the accompanying description and the essential features will be set out in the claims.

60 In the drawings, Fig. 1 is a sectional view through a pump embodying our invention and provided with a substantially standard front

housing; Fig. 2 is a similar sectional view, cut away to more clearly disclose the impeller structure and showing the back plate of Fig. 1 as forming part of a pump constructed with the front housing permitting easy access to the impeller,—this view is substantially along the line 2—2 of Fig. 3; Fig. 3 is a front elevation of the pump of Fig. 2, partly broken away to show the impeller structure; Fig. 4 is an enlarged sectional detail showing the shaft and packing construction which is only indicated generally in Figs. 1 and 2; Fig. 5 is a perspective view of the impeller removed from the pump casing; while Fig. 6 is an enlarged diagram illustrating the formation and dimensions of the fluid passageway between adjacent vanes of the impeller.

The pump shown in Fig. 1 comprises a back plate 10 suitably mounted upon a base and in which is rotatably mounted a drive shaft 11 to which is rigidly secured an impeller 12. The 20 drive shaft is adapted to be connected to any suitable source of driving power, the usual construction being an electric motor mounted upon a common base with the back plate 10. The back plate has a flange 13 extending radially outward beyond the impeller and to this is attached a front housing 14 by means of bolts or other securing means not shown. It will be understood by those skilled in the art that suitable gaskets are used at 15 to provide a fluid-tight joint. In 30 the type of pump here disclosed the back plate flange 13 has a portion 16 which forms part of the wall of an annular discharge passageway which surrounds the impeller. The front housing provides the remaining walls of this annular 35 discharge passageway and carries a discharge conduit fitting 17 which communicates with the annular discharge passageway 18. The conduit 17 would normally be supplied with a pipe flange 19 for connection with the discharge conduits. 40 The front housing 14 carries an inlet conduit connection 20 adapted to communicate with the axis or eye of the impeller and provided with a suitable flange 21 by which it may be secured to the incoming pipe line.

45 The pump casing of Fig. 1 generally conforms to standard practice but this is often unsuitable for economical applications of the pump because of the necessity of frequent access to the interior of the pump for changing the impeller or for 50 other purposes and we have therefore provided the structure shown in Figs. 2 and 3 to permit easy access to the interior of the pump. In these figures the back plate, shaft and impeller are the same or similar to those in Fig. 1, but the front 55 housing comprises two main casing members, one of which is an annular member 22 which carries the discharge conduit member 23 and also forms a portion of the wall of the annular passageway 18. The casing member 22 is preferably provided 60

with a forwardly extending cylindrical wall 24 having an outwardly extending flange 25 within which a central casing member 26 is adapted to fit in a fluid-tight joint and be sealed by a gasket 27. This central casing member has a portion 28 adapted to closely overlie the face of the impeller and has other portions 29 which complete the wall of the annular passageway 18. These casing members are all secured together by suitable bolts at the joints shown, as will be readily understood, but these connections have been omitted for the sake of clearness. Because of the heavy overhanging portion of the housing member, we have provided a bracket member 30 secured to the base plate in a manner to support the front housing. The central casing member 26 carries an inlet conduit member 31 which registers with an inlet conduit carried by wall 24, this inlet conduit having a pipe flange connection 32 so that the member 26 may be lifted out of place without disturbing the suction line 34 or the discharge line 23. This is a valuable feature in locations where the suction line is of considerable length because the constant breaking of joints in such a line for access to the pump militates against the maintenance of a high suction in that line. The inlet conduit passageway 31 is integral with the member 26 and leads the inlet fluid by an easy curve to the eye of the impeller.

To overcome the difficulties due to leakage of fluid along the drive shaft, we have provided the improved structure best shown in Fig. 4. The shaft 11 is of a size and material suitable to carry the structural strains transmitted through it. As a protection against corrosion and erosion this shaft is closely surrounded by a cylindrical sheath 35 of corrosion-resisting material and between this sheath and the back plate 10 is the usual packing 36 held in place by a packing nut 37. As an additional protection against leakage at this point the casing back plate 10 is provided at the point where the shaft enters the casing with an annular flat shoulder 38 surrounding the shaft and in a plane normal to its axis. To the shaft is secured a flexible sealing member 39 preferably L-shape in section and having a cylindrical portion 40 secured upon the shaft and another portion 41 extending radially outward therefrom in a manner to lie substantially parallel and with slight clearance from the shoulder 38 when the shaft is in rotation. At this time the rotation of the sealing member 39 will serve to throw the fluid being handled outwardly in the impeller casing so that no leakage occurs along the shaft. When the pump is idle, any pressure in the casing will press the flange portion 41 of the sealing member against the shoulder 38 thus providing an additional seal against leakage from the casing along the shaft. It will also be noted that the shaft has a threaded connection 42 with the impeller 12 and adjacent the point where the shaft enters the impeller it is provided with an annular recess 43. The sheath 35 extends beyond the main portion of the shaft to overlie the recess 43 and a gasket 44 is embedded in the impeller and enters the recess 43. When the impeller is turned down upon the threads 42 the sheath 35 is forced into the gasket 44 forming a tight joint at this point, proof against leakage from the casing into the connecting joint between the shaft and impeller.

Referring now to Figs. 2, 3, and 5, the impeller 12 is provided with vanes 45 bent backward from the direction of rotation and extending from the eye of the impeller to the periphery thereof.

These vanes have their greatest depth at a point adjacent the center or eye of the impeller from which point they taper to a point of least depth near the periphery thereof. The vanes as shown are cut away at their inner ends to form a recess at the eye of the impeller having a radius which we have designated as r_0 in Fig. 3. In the design of a pump for specific requirements this radius of the eye of the impeller is a function of the required suction lift. Pumps as usually designed have a liquid velocity at the inlet of five or six feet per second or more, but we prefer to design our pump to keep this suction velocity low, in the nature of one to three feet per second. With this restricted inlet velocity we find that it is desirable, to obtain suitable pumping characteristics, that the depth of the recess at the eye of the impeller, which we have designated l_0 in Figs. 2 and 6, be made 1.3 to 1.5 times the radius of the eye r_0 . If the depth of this recess l_0 be not made large enough, the suction lift will be limited due to strangulation at this point. This restriction will also cause cavitation with resulting rapid corrosion of the impeller. The radius of the impeller which we have designated r_1 in Fig. 3 is selected with regard to the desired pressure head to be developed as will be understood by those skilled in the art.

Referring now to Fig. 6, we have attempted to illustrate the development of the passageway between adjacent vanes of our impeller. Fig. 6 represents the space bounded by the vanes A and B in Fig. 5 and as limited by the lines C and D at the eye and periphery of the impeller respectively. The depth of the vane adjacent the eye of the impeller is indicated by l_0 , as previously described, and the depth at the periphery of the impeller is designated l_1 . The circumferential distance between the inner ends of adjacent vanes is indicated as b_0 and the circumferential distance between the outer ends of adjacent vanes at the periphery of the impeller is designated b_1 . Similar dimensions at intermediate points have been indicated by the letters l and b with suitable subscripts. Preferably we design this passageway between adjacent vanes to provide a constant cross sectional area from the eye to the periphery of the impeller. In other words, the areas represented by $l_0 \times b_0$, $l_1 \times b_1$, $l_2 \times b_2$, etc., give a substantially constant area.

Usually in pumps of this type the impeller surfaces are designed to conduct the incoming fluid in an easy curve along the vanes of the impeller toward its periphery. We find, however, that a pump designed as here described may be formed with a flat face on the side toward the front housing, that is to say, the vanes on the side toward the inlet end in a plane normal to the shaft axis. The entire difference in depth of the vanes from the center to the periphery is then taken care of by varying the depth of the impeller with reference to this plane on the inlet side. When the fluid passageway between adjacent vanes is designed to give a substantially constant cross-sectional area throughout, it gives to the bottom of the passageway a somewhat parabolic shape as best seen in Fig. 2, although it will be understood that this is not the true shape of this passageway as Fig. 2 is a diametric section whereas the bottom of the passageway follows a curved path more like that shown in Fig. 6. We find that shock is minimized and best results are obtained if the bottom of the impeller is raised in a substantially conical point at the center of the eye as indi-

cated at 46, then curved downwardly in all directions away from the direction of inlet flow as shown at 47, reaching its greatest depth adjacent the inner ends of the vanes at which point we have designated the dimension l_0 .

Ordinarily pressure is below atmospheric at the suction inlet and to prevent this suction reaching the back of the impeller at the point where the shaft enters the casing and for strengthening the structure, the impeller has been provided with a solid or closed back extending slightly beyond the eye of the impeller as shown at 48. Our pump is thus designed so that the impeller is always subject to the suction pull at the inlet side thereof and this axial pull is taken care of by suitable thrust bearings on the pump shaft.

As is well known to the designers of pumps, the outside diameter of the impeller r_1 is a function of the pressure head to be developed and the assumed speed of the pump, while the depth l_1 of the vanes at the periphery of the impeller is likewise a function which depends upon the desired capacity of the pump. It results from the structural characteristics of our design that our impeller may be cut to various radii r_1 without changing the inherent characteristics of the pump, that is to say, the passageways between adjacent vanes will always be of constant cross-section from the eye to the periphery of the impeller. It also follows that the flat face of the impeller may be milled off to change the depth of the vanes to provide the desired pump characteristics and in such case it will still be true that the passageway between the vanes will be of constant cross-sectional area throughout. It is thus possible to use one impeller casting for many different types of pumps, it only being necessary to mill off the flat face or to turn down the circumference to meet various capacity characteristics desired.

A pump designed as above described makes possible the curving of the impeller blades backward from rotation to a far greater angle than has been possible in the past without having short circuit losses due to eddies in the passages between the vanes. This construction improves the ability of our pump to handle emulsions and liquids more viscous than water. We have found that we can employ vanes of great angularity, such that the angle α_0 at the impeller eye and α_1 at the periphery of the impeller (see Fig. 3) may be greater than 60° .

The principles here disclosed are valuable in either closed or open-type impellers. For our purpose we prefer the open type impeller so that in handling liquids containing solids, the tendency to clog the passages between vanes is obviated as the relative rotation of the vanes with respect to the housing causes a shearing effect on the solid particles which tends to keep the passages clear.

When desired lubricant may be supplied to shaft 11 by a fitting 50 communicating with the shaft through the packing nut 37 or by other suitable means. In any case we find that lubricant so supplied, or lubricant from packing 36, will follow inwardly along shaft 11 until stopped by gasket 39 at 51. The gasket at this point prevents lubricant from being washed away by the fluid in the pump.

While we have disclosed a preferred embodiment of our invention, it will be understood that the principles of pump design here set forth

may be applied to various types of centrifugal pumps and we do not desire to be limited therein except by the appended claims as interpreted in view of the prior art.

What we claim is:

1. In a centrifugal pump for a substantially incompressible fluid, a casing, a rotatable shaft, a circular impeller on said shaft within and closely conforming to said casing, there being a fluid inlet opening through said casing and communicating with the eye of said impeller and an annular discharge chamber of substantially uniform section about the periphery of said impeller, said inlet opening being of a size sufficient to provide a fluid inlet velocity not substantially over three feet per second, said impeller having vanes curved backward from the direction of rotation, said vanes being entirely cut away at their inner ends to form a recess at the eye of said impeller substantially equal in diameter to said inlet opening, said vanes adjacent their inner ends being of a depth not substantially less than 1.3 times the radius of said inlet opening, and said vanes decreasing in depth from the eye to the periphery of said impeller in a manner to provide a substantially constant cross-sectional area of the fluid passageways between adjacent vanes.

2. A pump as in claim 1, and in which the bottom of the said impeller at the center of the eye is raised in a substantially conical point.

3. A pump as in claim 1, and in which the angle between the radius of said impeller cutting the curve of one of said vanes adjacent the periphery and a tangent to the curve at the same point is greater than sixty degrees.

4. A pump as in claim 1, and in which the vanes are open at the face of the impeller and there end in a plane normal to the shaft axis, whereby a cut may be easily taken from the face of said impeller to form an impeller of equal efficiency having vanes of reduced depth.

5. A pump as in claim 1, and in which the vanes are open except for a closed back extending slightly beyond the eye of the impeller, and said vanes end at the face of the impeller in a plane normal to the shaft axis, whereby the inlet suction is limited to the face of the impeller, and whereby a greater vane depth is provided adjacent the periphery of said impeller without increasing the depth of the casing.

6. In a centrifugal pump for a substantially incompressible fluid, a casing, a rotatable shaft, a circular impeller on said shaft within and closely conforming to said casing, there being a fluid inlet opening through said casing and communicating with the eye of said impeller and an annular discharge chamber of substantially uniform section about the periphery of said impeller, said inlet opening being of a size sufficient to provide a fluid inlet velocity not substantially over three feet per second, said impeller having a back member against which the inlet fluid impinges and vanes on the inlet side of said back member and extending from the eye to the periphery of said impeller, said vanes at their inner ends terminating in a circle substantially equal in diameter to said inlet opening, and said impeller parts being arranged to provide a substantially constant cross-sectional area of the fluid passageways between adjacent vanes.

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