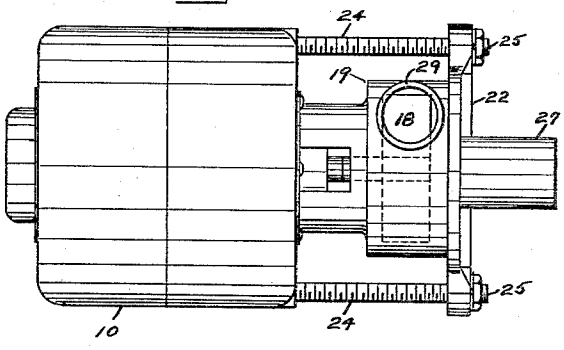
July 20, 1954 H. M. MONTGOMERY ET AL 2,684,033

CENTRIFUGAL PUMP

Filed Dec. 30, 1949

FIG. 1



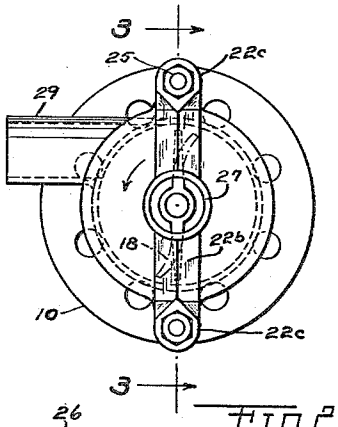


FIG. 2

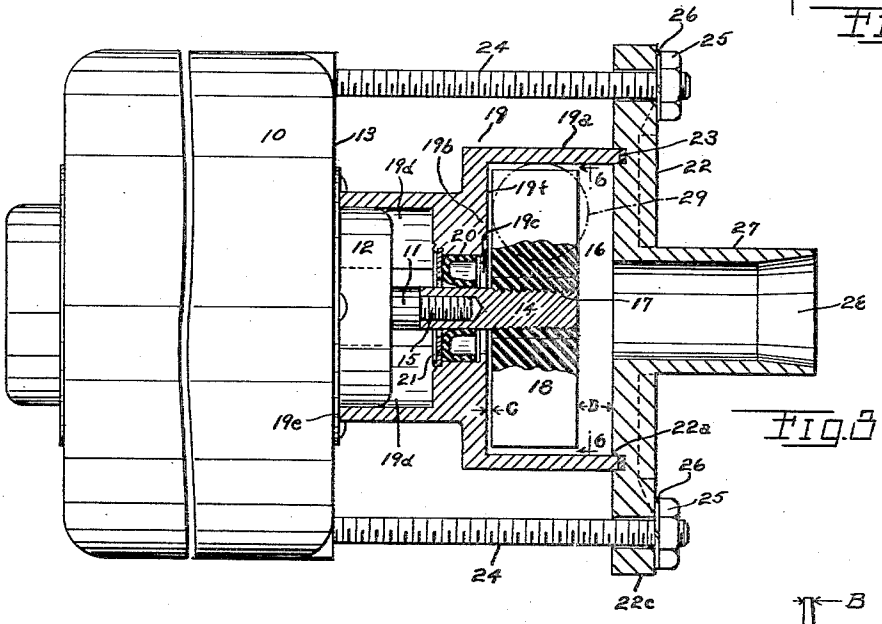


FIG. 3

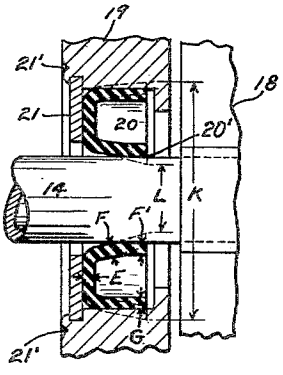


FIG. 4

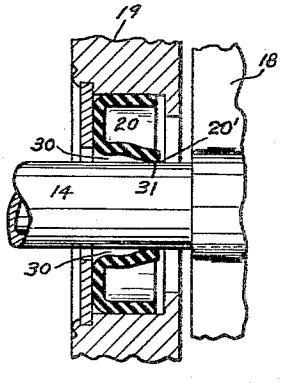


FIG. 5

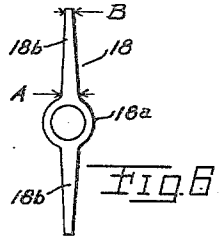


FIG. 6

INVENTORS.
HORACE M. MONTGOMERY
EDWARD M. SMITH
BY
Hyde, Myers, Baldwin & Doran
ATTORNEYS

Patented July 20, 1954

2,684,033

UNITED STATES PATENT OFFICE

2,684,033

CENTRIFUGAL PUMP

Horace M. Montgomery and Edward M. Smith, Mansfield, Ohio, assignors to Barnes Manufacturing Co., Mansfield, Ohio, a corporation of Ohio

Application December 30, 1949, Serial No. 135,996

2 Claims. (Cl. 103—103)

1

This invention relates to improvements in a centrifugal pump and more particularly to a pump adapted to be driven by a motor of very small horsepower such as one-fiftieth horsepower.

It has long been desirable to provide centrifugal pumps for very light duty, such as dishwashing, and adapted to be driven by such small motors as those of say one-fiftieth horsepower. One of the difficulties heretofore encountered in providing such a pump, is the reduction of friction load both at starting and at full load, and one of the objects of the present invention is to so reduce the friction of such a centrifugal pump both by vane clearances and improvements in the seal around the shaft which extends into the pump chamber.

Another object of the present invention is to provide a centrifugal pump of the type described with critical clearances between the blades and the housing which contribute to high efficiency in pumping, and at the same time permits our pump to handle effectively small solids in the stream, such as particles of food in dish washing apparatus.

Another object of the present invention is to provide a seal about a shaft entering a pump chamber so constructed and arranged that the pump chamber is well sealed against leakage when the pump is idle and the embracing action of the seal on the shaft is automatically reduced as the pump load increases.

Another object of the present invention is to provide a pump having vanes of resilient but fairly stiff rubberlike material so constructed and arranged that the vanes extend substantially in radial direction under low loads but which tend to take a reverse involute bend when the pump is under heavy load thereby automatically maintaining the efficiency of the pump under various loads.

Another object of the present invention is to provide a novel combination of an electric motor having a bearing boss concentric with the drive shaft thereof and shell means rigid with the pump housing and concentric with the cylindrical pump chamber, these parts so coacting when assembled that the motor bearing boss centers and alines the pump shaft and the pump housing or pumping chamber.

A further object of the present invention is to provide, in combination with the benefits of the preceding paragraph, an end face on the motor exteriorly of the boss lying in a plane normal to the motor shaft together with an end edge of

2

the shell means also lying in a plane normal to the shaft whereby engagement of the shell means against the motor end face accurately positions the pump impeller in the pump housing.

Still another object of the present invention is to provide an end plate for the pump housing which is a bolting bracket for securing the pump housing to the motor and at the same time provides a seal for closing the outer end of the pump housing, this end plate also carrying the suction flange.

Other objects and advantages of our invention will be apparent from the accompanying drawings and description and the essential features thereof will be set forth in the appended claims.

In the drawings, Fig. 1 is a side elevational view of our novel pump assembled to its driving motor;

Fig. 2 is an end elevation of the same taken at the right-hand end of Fig. 1;

Fig. 3 is an enlarged view similar to Fig. 1 with the right-hand portion thereof in central sectional view to more clearly show the construction;

Fig. 4 is an enlarged fragmental sectional view through the seal portion of Fig. 3;

Fig. 5 is a fragmental sectional view similar to Fig. 4 but showing the position of the seal, somewhat exaggerated, at top motor speed; while

Fig. 6 is an end elevational view of the vanes of the pump taken along the line 6—6 of Fig. 3 and slightly reduced in size therefrom.

The drawings show our improved pump secured to an electric motor 10 which in the present instance is a one-fiftieth horsepower A. C. motor having two poles and operating at 110 volts and 3000 revolutions per minute. This motor drives a shaft 11. Rigid with the motor 10, and concentric about the shaft 11 is a boss 12 which normally houses the bearings for the shaft 11. The end face 13 of the motor exteriorly of the boss 12 is flat which cooperates with the pump housing in a manner later described. The pump shaft has an extension 14 which has a threaded connection at 15 with the end of shaft 11. The outer end of the shaft extension 14 extends into the pump chamber 16 and is provided with means for bonding to the shaft 14 the pump impeller 18, this in the present instance, comprising a fine knurling at 17.

A pump housing 19 is provided having a generally cylindrical wall 19a which provides a cylindrical pump chamber 16. One end of this chamber is substantially closed by the end 19b of the housing. There is a central opening 19c through the end 19b where the shaft portion 14

enters the pump chamber. A seal 20 is provided between the shaft 14 and the pump housing for the purpose of closing the opening 19c against egress of fluid from the pump or ingress of air to the pump. A retaining washer 21 secures the seal in position and the washer is held in place by peening a portion of the pump housing end wall over on the outer face of the washer as by means of a center punch as indicated at 21'.

Shell means 19d is provided at the motor end of the housing 19 for positioning the housing 19 relative to the motor. This consists of upper and lower arcuate shells, each of about 135° in extent, rigid with the housing 19 and extending toward the motor 10 and having an end at 19e terminating in a plane at right angles to the shaft 11. The inside diameter of the portions 19d is adapted to snugly embrace the boss 12. The shell portions 19d are concentric with the inside diameter of the housing portion 19a and, since the boss 12 is concentric about the shaft 11, when the parts are assembled as shown in Figs. 1 and 3 the parts 12 and 19d serve to hold the pump chamber properly alined in both horizontal and vertical planes with respect to the motor 10 and its shaft 11. Since the impeller 18 is fixed on the shaft 14, which in turn is fixed to the shaft 11, the abutment of the end 19e of the pump housing against the motor face 13 properly positions the impeller with respect to the face 19f of the pump chamber.

An end plate 22 serves to close the open outer end of the pump housing and at the same time serves as a clamping bracket to secure the housing to the motor 10. The walls 19a terminate at the open end of the pump housing in a plane at right angles to the axis of the pump housing. An annular recess 22a in the end plate 22 snugly receives the outer ends of the walls 19a. A sealing gasket 23 is provided between the parts to make a liquid tight seal. On the outer face of the end plate there is provided a stiffening rib 22b extending across the end plate and providing outstanding ears 22c through which pass bolts 24 which are threaded into the motor 10. Nuts 25 and lock washers 26 clampingly secure the end plate in the positions shown in Figs. 1 and 3 so as to close the open outer end of the pump housing and to secure the housing to the motor 10 in proper position. Centrally of the end plate there is provided a suction flange 27 which carries the fluid inlet 28 communicating with the pump chamber 16 centrally thereof. A fluid outlet 29 is provided tangentially of the chamber 16 and communicating therewith.

The pump impeller 18, as clearly seen in Fig. 6, comprises a hub portion 18a having an internal diameter of about ¼ inch and an external diameter of about ⅜ inch. This hub portion is bonded to the finely knurled portions 17 of the shaft 14. The impeller also comprises two radially extending vanes 18b extending diametrically opposite each other away from the hub portion 18a. In this form of our invention, where the pump chamber 16 has an internal diameter of 1⅞ inches, the vanes 18b extend 1¾ inches from tip to tip, leaving a clearance all around between the vanes and the pump housing walls 19a of about $\frac{1}{16}$ inch. In this form of our invention, each vane has a thickness A near the hub of about ⅛ inch and a thickness B at the outer end of about $\frac{1}{16}$ inch. The vane thickness tapers gradually from the base to the outer tip. In this form of our invention, the dimension from the root to the tip of each vane is about $\frac{13}{16}$ of an inch. Our invention comprises the use of rubberlike material which is resilient but fairly stiff. We find that a hardness of 75 to 80 durometer is about right in the present instance to give maximum efficiency. It results from this construction that as the pump is turning in the direction of the arrow of Fig. 2, the vanes 18b at low loads hold a substantial radial position as shown in Fig. 6 and is indicated in dotted lines in Fig. 2. However, as the load increases, the vanes bend backwardly at the tip and the graduated thickness of the vanes tends to cause them to take the form of an involute curve bending backwardly with respect to the direction of rotation as indicated in dot-dash lines in Fig. 2. The greater the speed of the pump and the greater the load on the pump, the greater the curvature of the vanes so that the efficiency is automatically increased as the load increases. The pump here shown is designed to pump about eight gallons per minute of water at the top capacity of the one-fiftieth horsepower motor.

As previously mentioned, one of the purposes of the present invention is to reduce the frictional drag of the pump on the motor while at the same time retaining pumping efficiency. We have found that in a pump of the size described, the dimension C as shown in Fig. 3 between the side edge of the vane nearest the motor end of the pump housing should be about $\frac{1}{32}$ inch. We have also found that the dimension D as shown in Fig. 3 between the other side of the vane and the end plate 22 should be about ¼ inch for best efficiency. It has previously been taught by many that centrifugal pumps should have the minimum clearance between the vanes and the housing. In the present instance, we have obtained very high efficiencies with the construction as clearly shown in Fig. 3 where there are clearances on all sides of the vanes.

Another advantage of the clearance D and the clearance between vane tips and the housing, coupled with the resilient quality of the vanes, is that we are enabled to pump liquids containing a considerable proportion of solids such as particles of food, toothpicks, etc. found in dish washing applications.

We have found that the seal 20 between the shaft and the pump housing is one of the very critical spots in this design and therefore it will be shown and described in great detail. The seal is of cup-shape and comprises an uninterrupted annular ring of resilient rubberlike material of 45 to 50 durometer hardness. The seal is U-shape in section as clearly shown in Figs. 4 and 5. The radially inner lip of the U-shape section is adjacent shaft 14 and has a free edge 20' which sealingly engages shaft 14. In dot-dash lines in Fig. 4, there is shown the unstressed condition of the seal. The thickness at the base of the U-shape section as indicated at E in Fig. 4 is about .040 inch. The thickness at F and G is approximately .025 to .030 inch. In unstressed condition, as shown in dot-dash lines in Fig. 4, the inner and outer leg of the U-shape section flare outwardly at about 3½° from an axial position. The outside diameter between the lips as indicated at K in Fig. 4 in the unstressed condition is approximately .716 inch. This is compressed inwardly to the full line position of Fig. 4 to a dimension of about .670 inch. The inside dimension L as indicated in Fig. 4 in unstressed condition is approximately .226 inch. This is extended outwardly by engagement with the shaft portion 14 to a dimension of .250 inch.

To show how critical this seal is, it may be mentioned that when the dimension L in unstressed condition was .236 inch, the seal was unsatisfactory because it was too loose. When the dimension L was .220 inch, the seal was unsatisfactory because it was too tight. In Fig. 4, the seal 20 is shown in its approximately correct position in full lines when the pump is idle. In Fig. 5, the position of the seal at top speed of the pump is shown slightly exaggerated. It will have been noted that the vanes of the pump impeller have no shrouding and therefore there is free access of the pump suction near the shaft 14 on both sides of the vane. In other words, the suction at the hollow side of the cup-shape seal 20 is approximately the same as the suction at the suction flange 27. This suction inside the seal combined with the fact that the outer portion of the seal as indicated at 30 in Fig. 5 is open to atmosphere causes a differential pressure on opposite sides of the zone 31 as indicated in Fig. 5. With greater speed of rotation, the vacuum inside the housing is greater, thus causing a greater differential relative to atmospheric pressure outside. The tendency is to bend the resilient lips very slightly where the inner leg of the U-shape joins the base so that less of the lip is in engagement with the shaft portion 14 at top speed as indicated in Fig. 5 than at idle or lower speed as indicated in Fig. 4. When the dimension L (unstressed) changed to .226 inch, the lips at F' hug the shaft sufficiently at the zone 31 at top speed of this model to prevent the intake of air at this point. The U-shape form of the seal also takes care of slight misalinements of the shaft portion with respect to the housing. It results from this construction, that the seal contact area on the shaft is decreased as the suction increases, thus reducing the load on the motor. As a matter of fact, at the very start of pump operation, which is the critical period as regards load on the low starting torque motor, the friction load due to the seal 20 drops off as the fluid pumping load begins to build up at the impellers or vanes.

While the embodiment shows the pump close coupled to an electric motor, other drive means may be used and the pump characteristics remain the same.

What we claim is:

1. A pump comprising a housing having a generally cylindrical chamber and end closure members therefor, there being a fluid outlet positioned near the periphery of said chamber, one of said closure members having a fluid inlet centrally of said chamber, the other of said closure members having a central through opening, a driven shaft extending through said opening axially of said cylindrical chamber, an impeller fixed on said shaft and having radially extending vanes free of shrouding, said vanes being free of engagement with the walls of said housing by reason of a clear space provided between the sides and ends of said vanes and said housing, sealing means closing said opening between said shaft and said housing, said sealing means comprising an uninterrupted resilient annular lip extending about said shaft and generally parallel thereto and having a free edge extending toward the pump chamber, the interior of said annular ring communicating with the central interior portion of said chamber, said free edge sealingly engaging said shaft, the resiliency of said lip normally urging said free edge against said shaft and providing the sole bias for said edge, and the face of said lip toward said shaft and spaced from said free edge being spaced slightly from said shaft, there being an opening communicating between said face and atmosphere, whereby the greater the speed of rotation of said impeller, the greater is the differential pressure outside and inside of the housing and the less friction said lip exerts against said shaft.

2. A pump comprising a housing having a generally cylindrical chamber and end closure members therefor, there being a fluid outlet positioned near the periphery of said chamber, one of said closure members having a fluid inlet centrally of said chamber, the other of said closure members having a central through opening, a driven shaft extending through said opening axially of said cylindrical chamber, an impeller fixed on said shaft and having radially extending vanes free of shrouding, said vanes being free of engagement with the walls of said housing by reason of a clear space provided between the sides and ends of said vanes and said housing, sealing means closing said opening between said shaft and said housing, said sealing means comprising a resilient annular cup of U-shape section surrounding said shaft with the open side thereof toward said chamber and in communication with the central portion of said chamber, the radially innermost wall of said cup surrounding said shaft and having its free edge sealingly engaging said shaft, said innermost wall away from said free edge being slightly spaced from said shaft, there being an opening communicating between the outer face of said wall and atmosphere, whereby the greater the speed of rotation of said impeller, the greater is the differential pressure outside and inside of the housing and the less friction said lip exerts against said shaft.

References Cited in the file of this patent

UNITED STATES PATENTS

Number	Name	Date
1,328,061	Smith	Jan. 13, 1920
1,621,272	Raymond	Mar. 15, 1927
2,047,329	Petersen	July 14, 1936
2,292,125	Ifield	Aug. 4, 1942
2,382,839	Wuensch	Aug. 14, 1945
2,464,936	McConaghy	Mar. 22, 1949
2,508,678	MacDougall	May 23, 1950
2,509,436	Isenbarger	May 30, 1950
2,517,233	Peters	Aug. 1, 1950
2,570,862	Rosenkrans	Oct. 9, 1951