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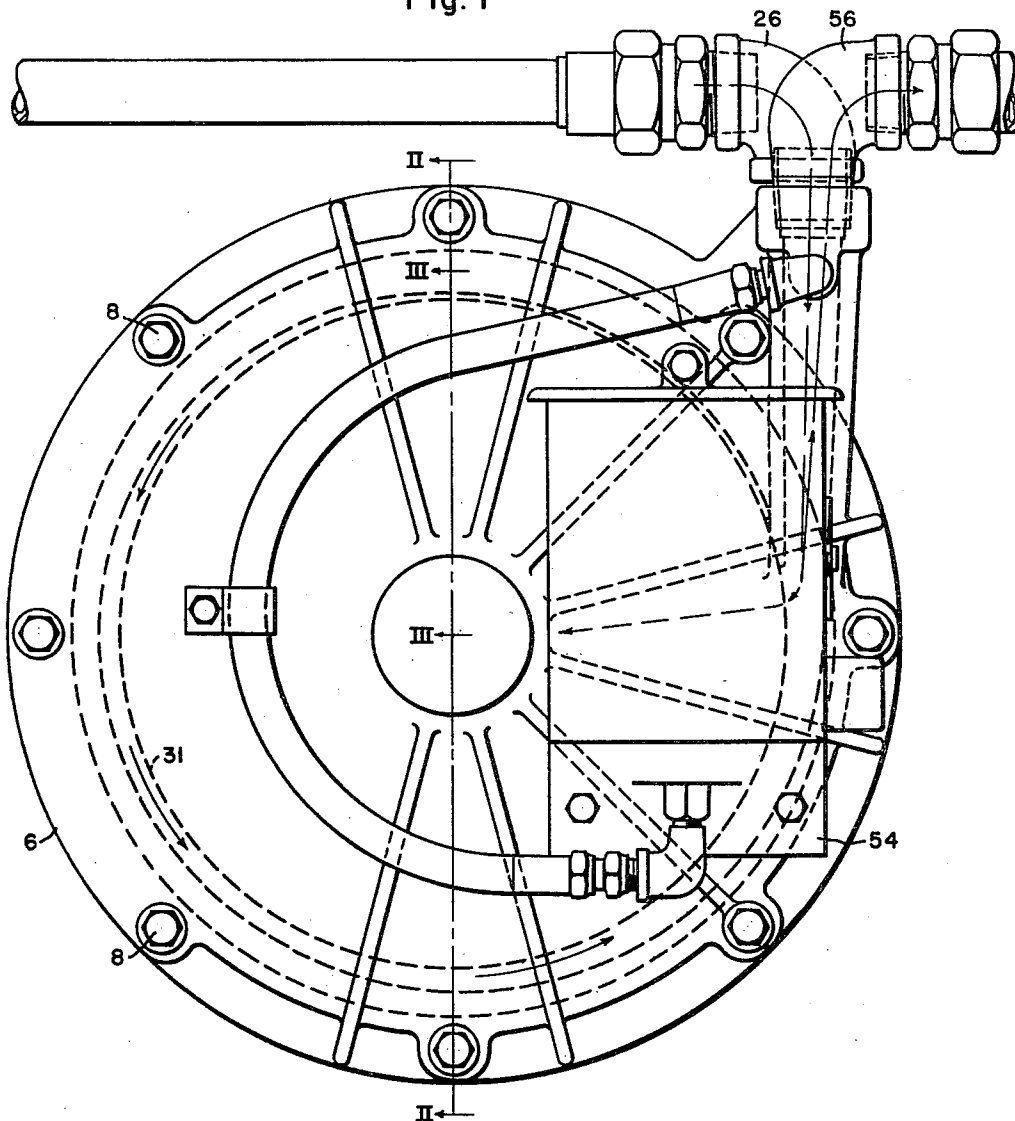
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SEALING MEANS FOR CENTRIFUGAL TYPE PUMPS

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2 Sheets-Sheet 1

Fig. 1



WITNESSES

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Fig. 2

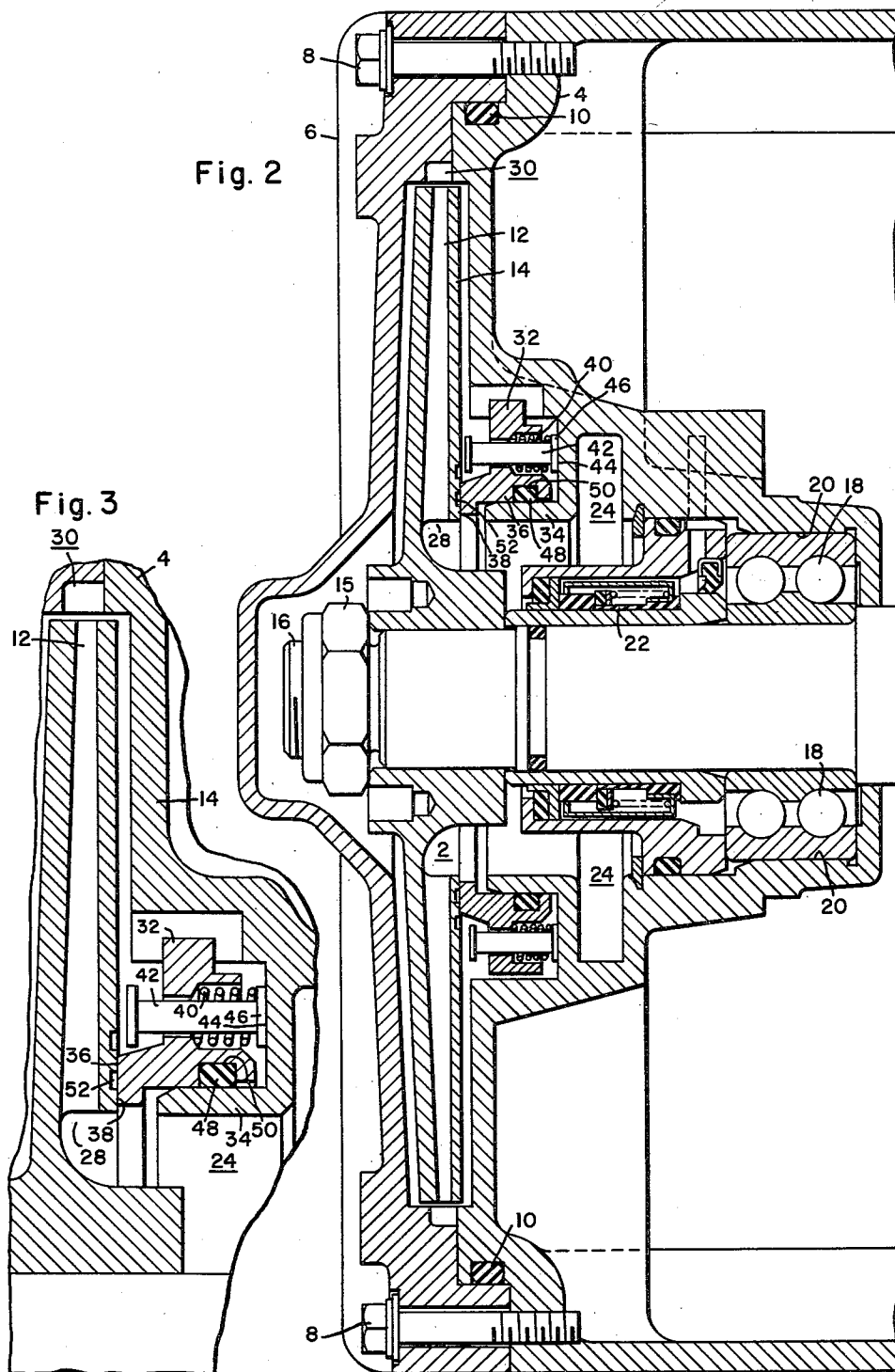
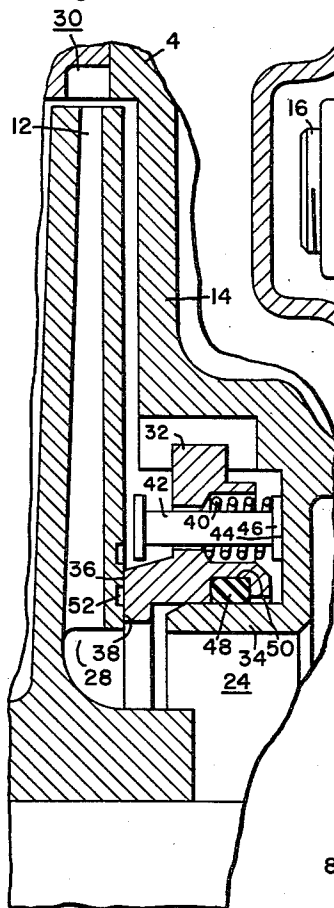


Fig. 3



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SEALING MEANS FOR CENTRIFUGAL TYPE PUMPS

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1 Claim. (Cl. 103-103)

The invention relates generally to centrifugal type pumps, and more particularly, relates to a construction of a centrifugal type pump having a positive seal.

The invention is particularly suitable for circulating pumps used for cooling ignitron rectifiers of self propelled railway cars. A high pressure low capacity flow of cooling liquid is required of the pump. The pump must be driven by means available in the car which car is limited in space and power. The 25 cycle power frequently available in railway cars results in a motor drive of relatively low speed for the usual pump application making the attainment of high pressure liquid doubly difficult.

In the usual centrifugal type pump, a conventional rotating seal is used to prevent recirculation of the working liquid from the high pressure discharge chamber to the low or negative pressure intake chamber. The close running clearance provided by the usual rotating seal is quite adequate for standard centrifugal type pumps. However, where a relatively low capacity and relatively high pressure centrifugal type pump is desired, the usual solution is not practical since even the close running clearances provided by the conventional rotating seal still provide a large bypass area allowing leakage from the discharge chamber to the intake chamber, thereby reducing the pump's efficiency. Further, the close running gap provided by the conventional seal becomes larger due to wear during operation and the gap loss can become large compared to the useful liquid flow. If such gap loss of liquid should become very large compared to the useful liquid flow from the discharge chamber of the pump, the centrifugal type pump can be retarded and even prevented from creating the desired liquid pressure in the discharge chamber.

The object of my invention is to provide an efficient centrifugal pump having a relatively low capacity but a relatively high discharge pressure.

Another object of my invention is to provide a centrifugal pump capable of providing relatively high pressure even though rotating at lower than the usual operating speeds of conventional pumps.

Another object of my invention is to provide a centrifugal pump having negligible recirculation from the high pressure to the low pressure chamber.

A further object of the invention is to provide a centrifugal pump for relatively low speed, low capacity and relatively high pressure in which the liquid seal between the bearing for the drive shaft and the pump housing is subjected only to low negative suction pressure.

Further objects and advantages of my invention will be more readily apparent from the following detailed description, taken in conjunction with the drawings, in which:

FIGURE 1 is an end elevation view of a centrifugal type pump and motor embodying my invention;

FIG. 2 is a sectional view along the lines II-II in FIG. 1; and

FIG. 3 is an enlarged fragmentary view of a portion of FIG. 2.

The embodiment chosen to illustrate my invention is a motor pump which provides cooling liquid for power rectifiers located on the same traction equipment as the pump itself. In the illustrative embodiment chosen a

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pump housing 2 is formed between a motor end bracket 4 and an end cap 6. The motor itself has not been shown. Any suitable type may be used. The end cap 6 is secured to the end bracket 4 by any suitable means shown as bolts 8 disposed around the outer periphery of the end cap 6 and end bracket 4, with an O-ring 10 providing sealing means so as to contain the liquid and allow no leakage to the outside of the housing 2. An impeller 12, which may be of any suitable type and is shown as having generally radial blades and a shroud 14, is mounted on a shaft 16, which may be the motor shaft, and retained there by a lock nut 15. It is to be understood, however, that any suitable means for driving the rotating shaft 16 on which the impeller 12 is located may be used.

The rotating shaft 16 is positioned within the pump housing 2 by means of a bearing 18 located in a bearing seat 20 within the motor end bracket 4. The bearing 18 is lubricated in any suitable manner, and a conventional liquid seal 22 separates the lubricant for the bearing 18 from the fluid within the pump housing 2. A liquid intake chamber 24 is located adjacent the drive side of the impeller 12, so that the conventional liquid seal 22 between the liquid within the chamber 24 and the lubricant of the bearing 18 is subjected only to the low or negative suction pressure of the intake chamber 24. The intake chamber 24 receives liquid from the intake pipes 26 in an area close to the rotating shaft 16. Where desirable, a removable strainer may be built into the motor end bracket 4 to screen the liquid entering the intake chamber 24 from the inlet pipes 26. The liquid entering the intake chamber 24 is directed to the impeller 12 at its entrance ducts 28 located close to the impeller hub which is mounted on drive shaft 16. The liquid is then forced by the impeller 12 to a discharge chamber 30 with sufficient force to create a high pressure. The discharge chamber 30 is located generally radially outward from the impeller 12 while the intake chamber 24 is located radially close to the shaft 16. The discharge chamber 30 is formed to have a volute passage 31 which connects to the discharge line 56 leading from the pump housing 2. A face type sealing ring 32 mounted on an annular ledge 34 formed within the end bracket 4 separates the discharge chamber 30 from the intake chamber 24. The sealing ring 32 will be more fully described below.

The impeller 12 is of relatively large diameter and narrow in width to be capable of creating the high pressure desired with relatively low volume of flow. The narrow width of the impeller 12 allows a compact pump construction of relatively small size. The relatively large diameter of the impeller 12 overcomes the lower than conventional speed at which the pump must operate.

It can be seen that with the relatively high pressure and low capacity provided by the impeller 12, means are required for preventing recirculation from the discharge chamber 30 to the intake chamber 24. Leakage or recirculation of the liquid is detrimental to obtaining the desired liquid flow from the pump and requires greater horsepower input to obtain the desired liquid output.

As mentioned previously, the conventional rotating seal does not provide a practical solution for a low capacity pump since even the very close clearances provided by a rotating seal are still relatively large compared to the volume of liquid to be pumped and provide a large bypass area allowing leakage from the discharge chamber 30 to the intake chamber 24. This area increases as the close running clearances become larger due to wear. The gap loss or feedback resulting can prevent the pump from creating the required liquid pressure.

In accordance with my invention, the face-type sealing ring 32 prevents recirculation of liquid from the discharge chamber 30 to the intake chamber 24. As can be seen in FIG. 3 the discharge chamber 30 of the pump housing

2 has an annular ledge 34 formed therein. The annular ledge 34 is shown coaxial with the shaft 16. Upon this ledge is mounted the face-type sealing ring 32. The ring 32 is free to move axially along the ledge 34 to insure adequate engagement of the ring 32 against the impeller shroud 14. The sealing ring 32 has a sealing face 36 which is ground flat. This face 36 engages a sealing portion 38 of the impeller shroud 14 which portion also is ground flat as a contact face to assure a proper tightness of the engagement between the seal face 36 and the impeller shroud portion 38.

Helical compression springs 40 are shown mounted on the circumferential portions of the sealing ring 32 to urge the sealing face 36 into engagement with the impeller shroud 14 at the portion of the shroud shown at 38. The helical springs 40 are mounted on rivets 42 which have a flange 44 on each end. The flange 44 on one end of the rivet 42 abuts a wall 46 of the discharge chamber 30. This end of the rivet also serves to anchor the helical spring 40 in such a manner that the helical spring 40 urges the ring 32 to move axially on the rivet 42. Since the sealing face 36 abuts the impeller shroud portion 38, the spring 40 will be unable to urge the sealing ring 32 to the opposite end of the rivet 42 but will, instead, insure a tight engagement or seal between the sealing face 36 and that portion 38 of the impeller shroud 14 which has been purposely ground flat. An O-ring 48, disposed in a groove 50 in the sealing ring 32 furnishes a slidable contact between the ring 32 and the annular ledge 34 and maintains the separation of liquids between the discharge chamber 30 and the intake chamber 24.

The sealing ring 32 shown has been made to be partly unbalanced. That is, the ring 32 presents opposite surface areas to the discharge chamber 30 which are unequal. The summation of surface area on the side of the sealing ring 32 away from the shroud 14 is greater than the summation of surface area on the side immediately adjacent the shroud 14. In other words the side of the ring 32 away from the seal has the greater area. The ratio of unequal surface areas presented to the discharge chamber 30 creates a force unbalance acting upon the sealing ring 32 urging the seal face 36 into engagement with the impeller shroud 14. The seal face 36 will be urged into engagement with the impeller shroud 14 by a force which increases in proportion with the ratio of the unbalanced areas as the liquid pressure within the discharge chamber 30 increases. Of course, the sealing ring 32 can be made force balanced, if desired, by making the summation of surface on each side of the ring equal. In this case the helical springs 40 can be made of sufficient strength to urge the ring 32 into engagement regardless of the pressure in the discharge chamber 30.

The sealing ring 32 may be made of any suitable material but it is preferred to be cast in a simple mold from an epoxy resin with a high percentage of suitable filler materials to assure good wearing qualities and low friction. Rotation of the ring 32 is prevented in any suitable manner as by a tongue on the ring (not shown) which fits with clearance into a groove in the end bracket. When the pump is in operation the ring 32 will thus be prevented from rotating with the shaft 16.

An eccentric lubricating groove 52 is cut into the portion 38 of the impeller shroud which has been ground flat so as to permit entrance of liquid between the seal face 36 and the impeller shroud 14. The groove allows liquid to enter therein between and lubricate the seal face 36 assuring low friction and long seal life.

A pressure switch 54 may be mounted on the outer portion of the end cap if desired to measure the pressure of the liquid in the discharge line 56. Upon measurement of a predetermined liquid pressure, the switch 54 may initiate a contact or perform any desired control function such as energizing the ignitron rectifiers.

In operation, flow of liquid to the impeller 12 is as indicated by the dotted arrows in FIG. 1. Flow from the

impeller through the volute passage 31 to the discharge line 56 is indicated by solid arrows. Upon start up of the pump, the sealing ring 32 is urged into engagement with the impeller shroud 14 by the spring loaded rivets 42 as described previously. As the pressure difference between the intake chamber 24 and the discharge chamber 30 increases the unequal surface areas of the ring 32 presented the discharge chamber 30 will result in a pressure unbalance urging the ring into tight engagement or seal with the impeller shroud 14 thus positively preventing any substantial leakage.

It can be seen that a high pressure, low volume centrifugal pump has been provided with a seal between the discharge chamber and the intake chamber which is capable of preventing any substantial recirculation of the liquid. As the sealing face 36 wears due to operation of the pump, the compression springs, as well as the unbalanced surface areas presented by the sealing ring 32, will not allow a clearance gap to develop between the sealing ring and the impeller shroud. The sealing face will be continually urged to engage the impeller shroud and allow no recirculation of liquid. Should the shaft 16 be subjected to end play, the seal 32 will automatically adjust itself for a proper engagement with the shroud 14.

In this way, there is no recirculation of liquid being pumped and even at low speeds the pump efficiency is high. Relatively high pressure at a low volume flow is attained with little power input required to drive the pump. The compactness of the pump saves valuable space within the railway equipment.

While this invention has been described with a certain degree of particularity, it is to be understood that the invention is not limited to these specific arrangements and in its broadest aspects, includes all equivalents, embodiments and modifications which come within the spirit and scope of my invention.

I claim as my invention:

A centrifugal pump for a liquid, including a rotatable shaft, a centrifugal impeller mounted on said shaft, lubricated bearing means on the drive side of said impeller, a housing for containing said impeller and said bearing means, said housing having an intake chamber adjacent the drive side of the impeller and a discharge chamber communicating with said impeller, said impeller having a shroud, said shroud having a sealing face, sealing means between said intake chamber and said bearing means, a sealing ring between said intake chamber and said discharge chamber having a radial portion extending into said discharge chamber with its opposite surfaces exposed in said discharge chamber, said ring having a sealing face engaging said first mentioned sealing face, said radial portion having a greater total surface area exposed on the surface remote from said first mentioned sealing face than the total surface area on the side adjacent said impeller shroud whereby said second mentioned sealing face is biased against said first mentioned sealing face with a force proportional to the magnitude of the pressure of the liquid within the discharge chamber to form a seal between said intake chamber and said discharge chamber, auxiliary resilient means for urging said second mentioned sealing face against said first mentioned sealing means, and an eccentric lubricating groove on said first mentioned sealing face for permitting entrance of lubricant between said sealing faces.

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