A centrifugal pump is provided with an improved impeller sealing arrangement which substantially reduces the likelihood of seal failure or leakage of the pump. The drive shaft of the impeller is sealed on one side of the pump housing by a mechanical seal carried in a non-pressurized reservoir of liquid, and a reduced pressure area is created by the pump impeller on an opposite side of the pump housing so that any tendency to leak will be from the reservoir to the pumped fluid. The mechanical seal which may be of a single or double face type is disposed within a reservoir containing an aqueous mixture of ethylene glycol which is under atmospheric pressure so that the normal direction of flow is from the reservoir through the seal faces and into the low pressure region. The above described pump may be provided with either straight centrifugal or self priming inlet housing sections and may be provided with a bearing pedestal or the pump shaft may be directly connected to the output shaft of an electric motor or other driving device.

6 Claims, 3 Drawing Figures
FIG. 1

INVENTOR

DOUGLAS JOHNSTON

BY

Cushman, Darby & Cushman
ATTORNEYS
1 CENTRIFUGAL PUMP

BACKGROUND OF THE INVENTION

1. Field of the Invention

This invention relates to centrifugal pumps and, more particularly, to centrifugal pumps utilized to transfer, apply, agitate and otherwise handle fertilizing materials in a liquid form.

2. Description of the Prior Art and Summary of the Invention

During the early 1950's fertilizer manufacturers began the promotion of liquid fertilizers instead of solid dry fertilizers. Since liquids could be applied more accurately, could be mixed with herbicides and insecticides, and, by the use of pumps, liquids eliminated the hard manual labor necessary in handling sacked dry fertilizers.

The only pumps available at a practical price and quantity range were the contractor's centrifugal pumps, designed primarily for pumping out ditches and excavations. In pumping out excavations a high suction lift is necessary, but the pump rarely is required to develop any discharge pressure, and therefore the pump's sealing means is subjected to little or no pressure against it.

Pumps used in liquid fertilizer service, such as for applications through nozzles, for agitation through sparging tubes, and for transfer from one tank to another at higher elevation, usually operate with a positive suction head and a relatively high discharge pressure, which puts their seals under heavy pressure, tending to force abrasive crystals contained with the liquid fertilizer through the high speed sealing surfaces which, in turn, causes seals to fail rapidly, frequently in a matter of a few days. In addition to seal failure caused by abrasive crystals getting between the sealing surfaces, conventional contractor's pumps will rapidly burn out the seal if operated dry, since they normally are mounted on impeller shafts which operate at about 3,450 rpm.

There have been many modifications developed in the last 20 years in attempt to eliminate excessive seal failure on these pumps. An early modification was the addition of a grease cup to force grease through holes drilled in the stationary sealing element. This method enabled the pump to run dry without ruining the seal faces, provided the grease cup maintained a steady flow of grease, but in operation the grease cup quickly became exhausted, and the usual type of seal failures again occurred even more rapidly as the residual grease adjacent to the sealing surfaces collected abrasive crystals to form a cutting or lapping compound.

Another modification provided a seal reservoir filled with oil and containing two seals with a single spring holding the two rotating elements against the pump pressure on one side and against atmospheric pressure on the other side. Several arrangements were provided for pressurizing the reservoir to a level above that of the pump side so that no abrasive materials carried in the liquid being pumped could cross the sealing faces. One arrangement provided an air tank to maintain air pressure at a value higher than the maximum pump pressure, usually 120 psi, so that the pressure in the seal reservoir prevented the pump discharge liquid from tending to flow through the seal faces into the reservoir. The air tank arrangement, however, required frequent refilling of the tank, a chore the operator found inconvenient and at times a source of sufficient pressure was unavailable. Consequently, the air pressure ultimately became too low and the abrasive suspension worked between the sealing surfaces and caused failure. Another arrangement provided a diaphragm between the pump discharge and the oil reservoir so that the oil in the reservoir was pressurized equal to the pump discharge pressure. The diaphragm arrangement was not satisfactory for a number of reasons. No seal is completely leak proof, and as soon as a small amount of oil leaked from the reservoir, the diaphragm being limited in movement, became taut, having to withstand the full pump pressure, and the pressure in the seal reservoir became less than the fluid pressure against the seal and failure resulted. Enhancing the leaking problem is the fact that, due to the many kinds of chemicals and solvents being mixed with liquid fertilizers, it is practically impossible to obtain diaphragm fabrics resistant to all liquid fertilizer mixes.

Further the pressurized double oil filled seal reservoir has some other inherent disadvantages which does not depend upon the manner in which it is pressurized. Pressurizing the reservoir puts the atmospheric seal under heavy pressure, increasing wear and encouraging leakage. If a pump with a long suction line is pumping at a high rate of flow and a spring loaded chemical nozzle valve in the discharge line snaps shut, as occasionally happens, the surge or "water hammer" may build up very high pressure (200 to 250 psi) in the pump, forcing the seal off the seat or bursting the diaphragm, and thereby contaminating the reservoir with abrasive crystals, which will cut out the seals almost immediately.

A more recent modification of the contractor's pump provides a double liquid filled seal chamber with the suction entrance between the pump seal and the pump impeller, with the entrance to the impeller on the shaft side. The location of the suction entrance between the seal and the impeller requires an impeller shaft of such great length it is too long to clamp over an engine or motor shaft, as a very slight misalignment between the two shafts is greatly magnified due to the distance to the impeller. At the high operating speed of 3,450 rpm an impeller operating off center with the driving shaft builds up high centrifugal forces that tend to force it further and further off center until failure is likely to result in a short time. Furthermore, a design that provides the suction entrance between the seal and the impeller holds the liquid against the seal at suction pressure, which may be below atmospheric pressure, but if the suction is connected to a vertical tank or an elevated tank the seal will very likely have a positive pressure against it, tending to force abrasive crystals between the sealing forces, thereby causing seal failure.

Representative prior art on this subject includes U.S. Pat. Nos. 981,763; 1,141,556; 1,516,822; 1,715,944; 1,967,316; 2,580,347; 2,684,033; 2,700,338; 2,951,449; 3,045,603; 3,077,161; 3,228,342; and 3,487,784.

The present invention provides a double seal reservoir chamber at atmospheric pressure with an impeller design that maintains lower than atmospheric pressure on the pump side of the seal so that any tendency to leak will be from the reservoir to the pump. Both seals are operating under little, if any, pressure, and as will be seen from the following description no auxiliary diaphragm or air tanks, that may be a source of trouble, are required. One feature of the invention is to provide critically placed openings through an impeller disc so
as to provide for a substantially reduced pressure zone adjacent the reservoir chamber. Although openings through impeller discs are generally known, as shown for example in U.S. Pat. No. 1,967,316, it has been found that a particular placement and angular disposition of such openings results in substantially improved performance for pumps of the types contemplated herein.

Also, the suction entrance to the pump is opposite the shaft side, in the conventional manner, so that the impeller shaft is short enough for the pump to be "short coupled" with the impeller shaft clamped on a motor or engine shaft.

This invention further provides that the seal reservoir be filled with an aqueous solution of ethylene glycol so that in the event of a pressure surge (as explained above) forcing suspended crystals, usually potash (KCl); into the seal reservoir the water in the solution will immediately dissolve the crystals into solution and the return to vacuum against the seal face will wash the seal faces clean by drawing a small amount of water across them. The purpose of the ethylene glycol, which is provided with a rust inhibitor, is to prevent freezing in cold climates.

As will be seen from the description, this invention also can be built with a single seal design, with the seal operating in the pumped fluid in the conventional manner, but operating with the fluid against the seal under a pressure much lower than the suction pressure, and with the hydraulically balanced impeller as with the double seal and water filled reservoir design. The purpose of offering the single seal design is to provide a less expensive pump for pumping fluids containing little, if any, abrasive crystals. Should the seal in this model eventually wear out the pump will still operate without leakage as the vacuum against the seal will draw in air, instead of leaking fertilizer, and the pump can be operated without shutdown until a new seal can be obtained. Obviously, drawing air into the pump will reduce its efficiency, but a conventional single seal pump would have to be completely stopped as liquid fertilizer, which is corrosive and expensive, would be spraying out under high pressure and would ruin the engine or motor and any other controls or machinery in the same area.

Either the single seal or the double seal model can be built in a self priming design or a straight centrifugal design, and may be bolted to an engine or electrical motor, with the impeller shaft clamped over the engine or motor shaft, eliminating the need for bearings, or either type may be mounted on a pedestal bearing shaft support for being driven by a belt or chain or for connecting to an engine or motor with a flexible coupling.

Another advantage of the present invention, as will be seen from the following description, is the fact that the design provides a hydraulically balanced impeller, which relieves the shaft bearings of the usual heavy thrust loads, and thereby greatly increases bearing life or else allows the use of lighter and less expensive bearings.

Another feature which allows the impeller to be hydraulically balanced and eliminate leakage from the high pressure discharge side of the impeller to the low pressure suction side includes a pair of O-rings which are mounted between the housing and respective sides of the impeller so as to completely seal the suction from discharge side of the housing. Unlike conventional O-rings which are over-sized so as to be in engagement with a running surface even when the pump is running dry and which therefore are subject to rapid wear, the applicant's design presents O-rings which are sized so as to contact only the housing or impeller in the undistorted state, i.e., when the pump runs dry. These O-rings are exposed to the discharge side of the pump and thus they are distorted when the pump is running wet and discharging to cause a seal between the housing and the impeller. Alternatively, the impeller and casing portions of the pump assembly can be designed and machined to close tolerances to achieve substantially the same sealing and balancing characteristics just discussed.

These and other features and advantages of the present invention will become apparent in the more detailed discussion which follows. In that discussion reference will be made to the accompanying drawings as briefly described below.

BRIEF DESCRIPTION OF DRAWINGS

FIG. 1 is a vertical cross-section of a self-priming centrifugal pump having a double seal and constructed in accordance with the invention.

FIG. 2 is an enlarged detail of a portion of FIG. 1 showing an O-ring sealing arrangement; and FIG. 3 is a cross-sectional view along the lines 3—3 in FIG. 1 showing the impeller design of this invention.

DETAILED DESCRIPTION OF THE INVENTION

Referring to FIG. 1, a self-priming centrifugal pump 10 is illustrated in cross section. The features which will be discussed below can be applied as well to a straight centrifugal pump design.

The centrifugal pump 10 includes a housing having a suction inlet passage defined at 12 and a discharge passage defined at 14. An impeller means, generally indicated at 18, is completely enclosed within the pump housing by a shroud 20 and other portions of the housing structure. The impeller means 18 is mounted for rotation within the housing through a drive shaft means 16. The drive shaft means 16 is connected at one end to an impeller disc portion of the impeller means 18, and the impeller means 18 includes a plurality of vanes or blades 36 which are of a known design for pumping a liquid.

A seal chamber 29 is positioned adjacent the pump housing to provide a reservoir area around a double-acting mechanical seal 26 carried around the drive shaft 16. The double-acting mechanical seal 26 is of a known design which includes a pair of rotating carbon rings 28a and 28b urged apart by a spring 30 so as to press against stationary rings of ceramic material 32a and 32b, respectively. The rings 32a and 32b are, in turn, seated in rubber or elastomeric composition annular cups 34a and 34b, respectively, to complete the seal. Other forms of double-acting mechanical seals may be used, and even single-acting seals can be substituted for what is shown in FIG. 1.

The seal chamber 29 functions to protect the mechanical seal 26 by surrounding the mechanical seal with a liquid bath maintained at atmospheric pressure. A suitable filling inlet and sight gauge are provided in the chamber 29 for maintaining the chamber in a filled condition. Preferably, the liquid bath comprises water or a mixture of water with another liquid such as an anti-freeze solution of ethylene glycol. Unlike prior art
arrangements, there is no requirement for pressurizing the contents of the seal chamber 29 because of a novel pressure differential producing means which will be discussed with reference to the design of the impeller means 18. Another function of the seal chamber 29 is to protect the pump during surges of pressure within the pump housing by allowing a movement of the mechanical seal 26 out of its sealing relationship with the housing structure of the pump for a sufficient time to allow limited surging of pumped liquid into the seal chamber 29. However, during normal operation of the pump, there is a very slight back flow of liquid from the seal chamber 29, through an annular space 62 around the drive shaft and into the inlet passage area of the pump housing. This keeps the mechanical seal washed and free of contact with harmful suspensions of materials or chemicals within the liquid being pumped. This is especially advantageous when the pump is used to handle certain fertilizer solutions containing abrasive suspensions or undissolved particles. In the event of a surge of pressure in the pumping zone, there is a movement of a small quantity of liquid back into the seal chamber 29, but the relatively large volume of water in that chamber assures the dissolving of any abrasive crystals that may enter the chamber during such a surging event. The volume of water in the chamber 29 and the fact that water is relatively incompressible, prevents movement of any large amounts of pumped liquid back into the seal chamber during a surge. Water or an aqueous solution is a suitable lubricant in fertilizer pumping situations because many of the crystals contained within the fertilizers such as potassium chloride, ammonium nitrate, and ammonium phosphate are readily soluble in water so that even if such crystals pass between the seal faces and into reservoir 29 they will be dissolved therein and cause no further damage to the seal faces. With the arrangement just described, there is no undue pressure on the mechanical seal components within the seal chamber 29, as has the impeller means 18, with prior art designs. Further, the mechanical seal is maintained free of harmful effects of pumped liquid by a slight washing of an aqueous solution over the seal parts which communicate with the pump housing itself. The slight washing action is obtained by a pressure differential established between the mechanical seal and the inlet passageway of the housing, and this is accomplished by forming critically placed openings 64 through the impeller disc of the impeller means, as will be discussed later.

The flow of liquid fertilizer, or the like, through centrifugal pump 10 is as follows: Upon entering through the suction inlet passage 12, the fertilizer passes through orifice 24 and into the impeller means 18 which spins the liquid outward in a known manner by vanes 36 to the shroud 20 which is annularly disposed about the impeller 18. As can best be seen in FIG. 3, the shroud 20 opens at region 42 into an outlet section of the housing, which, in turn, communicates with discharge outlet 14. In order to prevent a flow of fertilizer from the shroud 20 past the fronts of vanes 36 and into the relatively low pressure region existing near inlet 24, an O-ring of standard construction and material 46 is fitted into a groove formed in an annular shoulder on the outer face of impeller 18. As can best be seen from FIG. 2, O-ring 46 is so sized as to avoid a sealing contact with shroud 20 when impeller 18 is stationary or when no liquid is being pumped therethrough. This prevents undue wear on O-ring 46, especially in the event the pump is operated dry. When a liquid is pumped, however, pressure of the liquid from the discharge passage upon O-ring 46 causes it to distort to the position shown in dotted lines in FIG. 2 to seal the annular opening 52 which exists between shroud 20 and impeller 18. It should further be noted that the compression of O-ring 46 forms an effective seal even after a portion of O-ring 46 has been worn away in contradistinction to standard O-ring applications where only a complete O-ring provides a seal.

Another O-ring 54 is carried in a groove formed in a second annular shoulder 55 of the impeller. As with the O-ring 46, O-ring 54 is sized to avoid sealing engagement with the impeller 18 until a liquid is pumped. As impeller 18 rotates and pumps a liquid through pump 10, a portion of the liquid from the discharge outlet passage passes into annular opening 58, and the pressure of this liquid distorts O-ring 54 in a manner similar to O-ring 46 to effectively seal chamber 58 from chamber 60. The liquid in chamber 58, under approximately discharge pressure, is balanced by liquid in chamber 58a, on the opposite side of the impeller, by liquid at approximately discharge pressure. The vacuum at impeller orifice 24 is approximately balanced by the vacuum on the left side of the impeller in chamber 60. The vacuum in chamber 60 is actually greater than the vacuum at the impeller orifice entrance 24, tending to pull the impeller to the left (FIG. 1), but the pressure in chamber 58a is slightly less than in chamber 58, due to less rotating clearance, so that the lower pressure (higher vacuum) in chamber 60 is compensated for by slightly less pressure in chamber 58a, compared to chamber 58, and the impeller is hydraulically balanced so that the usual heavy thrust load on its bearings is eliminated.

Chamber 60 communicates with ceramic ring 32b of the mechanical seal through an annular area 62 which extends around the drive shaft and between the ceramic ring 32b and the drive shaft (the ring 32b being of a slightly larger internal diameter than the outside diameter of the drive shaft). Chamber 60 also communicates with the low pressure region in front of impeller 18 by means of a plurality of apertures 64 which are formed through impeller 18. As best seen in FIG. 3, the apertures 64 are placed adjacent the trailing edges of vanes 36. Due to the high velocity of the vanes 36, relative to the fluid entering the impeller 18, the region behind the vanes 36 has a very low pressure so that a vacuum of between 8 and 20 feet for the normal output range of the pump exists in this area and is transferred via apertures 64 to seal face 62a (between the ring seal 32a and 32b). Even with a zero suction head on the pump, the vacuum at the suction inlet 24 runs about 3 to 9 feet. In addition, since apertures 64 are disposed nearer the axis of rotation of impeller 18 than is the circumference of suction inlet 24, the centrifugal force of the liquid being rotated by impeller 18 causes the pressure at the circumference of inlet 24 to be higher than that at the entrance holes 64 so that the seal faces are exposed to even lower pressure than the normal inlet or suction pressure of pump 10. It is also to be noted from FIG. 1 that the holes 64 are drilled at an angle of about 45° with respect to the impeller shaft so that they emerge from impeller 18 even closer to the axis of the impeller shaft on the rear side of impeller 18. Thus, the
angular disposition of holes 64 and the rotating of impeller 18 further enhances the vacuum created in region 60 by forcing fluid out from region 60 in centrifugal manner. The point of emergence of the holes 64 behind and adjacent to the vanes and close to the inside radius of the vanes is very important, since this location is in a region of extremely low pressure in relation to the suction vacuum. For highest theoretical efficiency the vane thickness should be infinitely thin, but practical considerations require a thickness of the order of three-sixteenths to one-fourth inches thick, with a relatively blunt leading edge. With an infinitely thin vane the entrance angle of both the front and rear of the vane would be equal, and streamlined flow would be possible, but with a thick, blunt vane the rear entrance angle of the vane cannot be the same as the entrance angle at the front of the vane, and the fluid will not follow the rear of the vane at the entrance, which causes a high vacuum to be formed in this region. Thus the location of the exits of holes 64 still further enhance the vacuum in region 60, communicating with the seal face. The vacuum created in region 60 and therefore applied to the seal face 62a insures that a small but steady trickle of water or the like will flow from reservoir 29 toward the impeller and thus, preserve the seal face 62a between ceramic and graphic faces from crystalline structures and other solid impurities in the liquid fertilizer.

It is also of note that the impeller 18 is mounted facing the suction inlet 24 and thus shaft 16 may be a short shaft which can be directly coupled to electric motors or other prime movers which are not shown, thus alleviating the necessity for bearings in the housing structure of the pump itself. Of course, if so desired, conventional pedestal mounted bearings can be attached to the adapter "A" of pump 10.

Also, the O-rings 46 and 54 may be omitted in pumps designed with closer tolerances between the impeller and housing parts, or in pumps which can operate at less than maximum efficiency.

Many other known modifications may also be made to the preferred embodiment within the scope of the invention. For example, instead of the self-priming housing shown in FIG. 1, a straight centrifugal type housing may be used in which the inlet housing is not bent but rather is coaxial with the impeller. Also, automatic replenishing means may be utilized to maintain a constant volume of water and ethylene glycol or other suitable lubricant in the seal chamber reservoir 29.

What is claimed is:

1. A centrifugal pump comprising:
   a housing provided with an interior pumping cham-
   ber having a liquid inlet passage and a liquid outlet passage;
   impeller means mounted for rotation within said chamber for receiving liquid from said inlet passage and pumping it to said outlet passage, said impeller means including an impeller disc having one side thereof facing said inlet passage and a plurality of vanes carried by said one side and extending from adjacent the axis to the periphery thereof;
   drive shaft means connected to said impeller means and extending from the other side of said disc outwardly through a wall of said housing;
   mechanical seal means for said drive shaft means including two relatively-rotatably-engaged parts, one carried by said housing wall and the other by said drive shaft means;
   means defining a seal chamber exterior of said housing adapted to receive and hold a liquid at about atmospheric pressure in contact with the interengagement between said parts for cooling and cleaning said seal means;
   and means for maintaining a pressure less than atmospheric on the pumping chamber side of said seal means to induce a flow of liquid from said seal chamber between said parts into said pumping chamber, comprising at least one aperture extending through said impeller disc at a position close to the axis thereof and behind and closely adjacent to the trailing surface of one of said vanes.

2. The structure defined in claim 1 including first annular sealing means between the housing and the side of the impeller means opposite the inlet passage for sealing the mechanical seal means from discharge pressure.

3. The structure defined in claim 2 including second annular sealing means between the housing and the other side of the impeller means for sealing the inlet passage from discharge pressure, the diameters of said first and second annular sealing means being such as to substantially axially balance the hydraulic pressures on said impeller means.

4. The pump as claimed in claim 1 wherein said seal chamber reservoir contains an aqueous solution.

5. The pump as claimed in claim 4 wherein said aqueous solution includes ethylene glycol.

6. The pump claimed in claim 1 wherein said inlet passage is generally circular in cross section and said aperture is formed nearer to the axis of rotation of said impeller means than is the periphery of said suction passage.

* * * *