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RECIPROCATING PUMP

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2 Sheets-Sheet 1

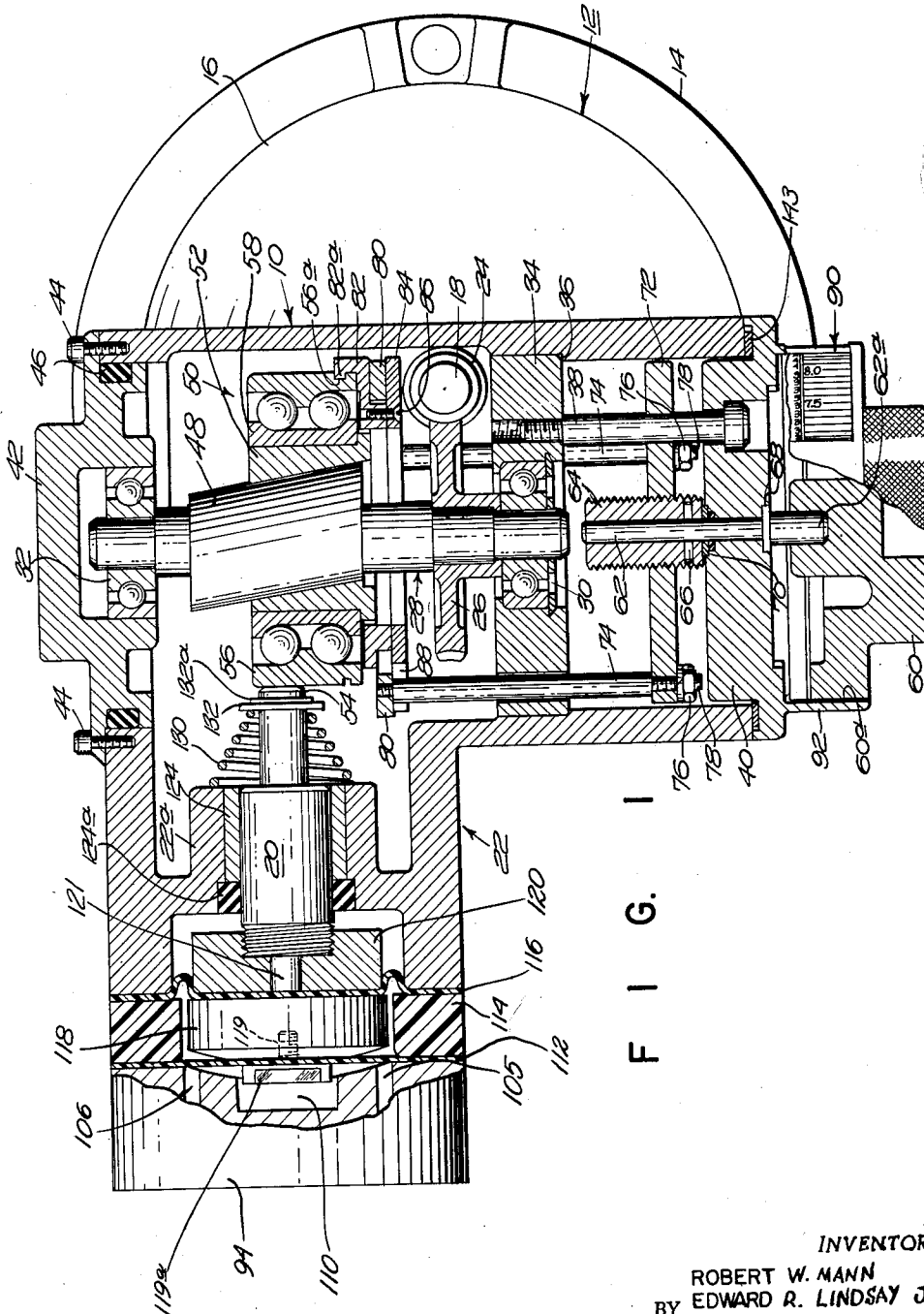


FIG. 1

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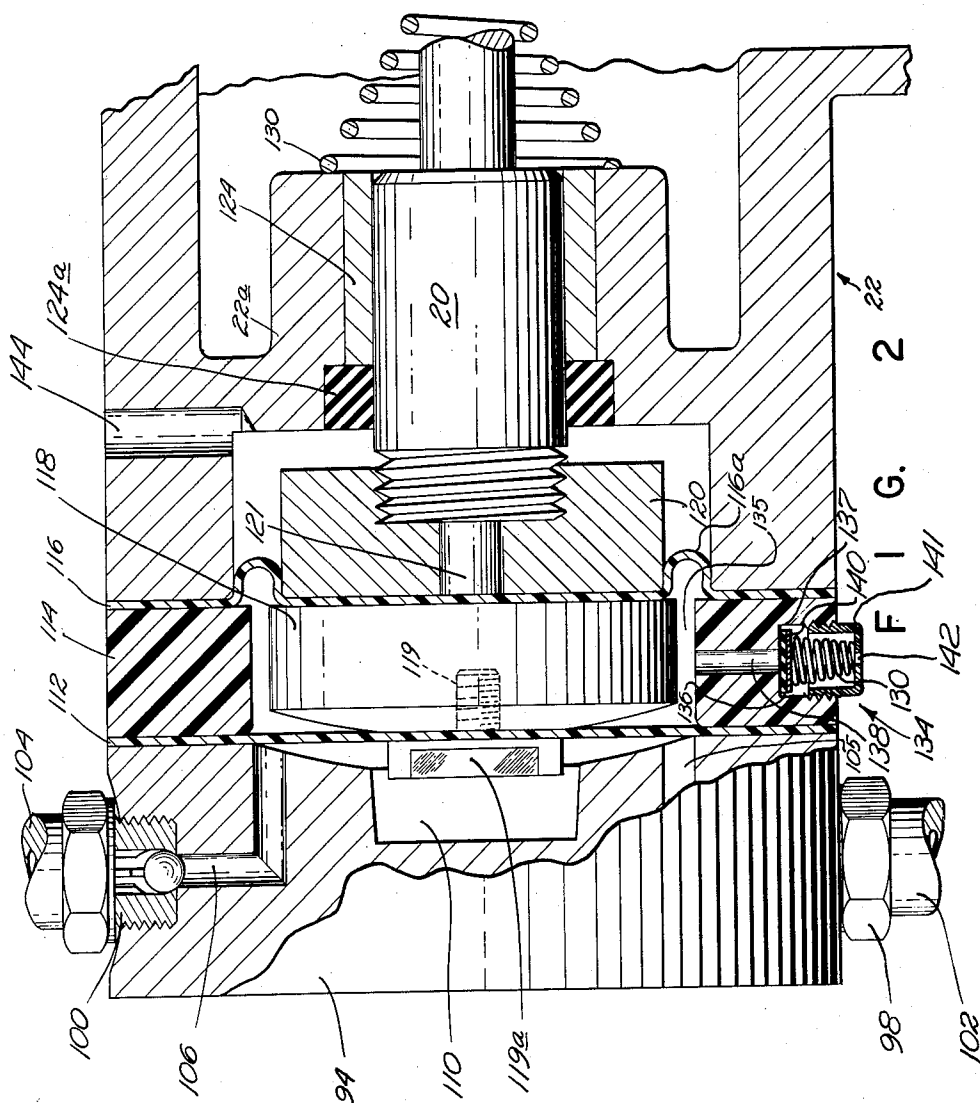
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RECIPROCATING PUMP

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1 Claim. (Cl. 74-571)

This invention relates to an improved diaphragm pump provided with such features as substantially complete protection of the drive system from pump head leakage and simple, yet accurate adjustment of the pump stroke. The pump head includes an unstressed barrier diaphragm disposed behind a conventional pumping diaphragm so that fluid leaking through the latter member is prevented by the barrier from escaping further into the interior of the pump housing. The drive mechanism includes an axially adjustable eccentric bearing mounted on a skewed cylinder, the position of the bearing on the cylinder determining the length of the pumping stroke.

A principal object of our invention is to provide a diaphragm pump with improved protection against leakage of the pumped fluid into the drive mechanism of the pump. This feature of the pump materially reduces corrosion attributable to seepage of pumped fluids onto vulnerable mechanical parts.

Another object of the invention is to provide an improved pump head for a pump of the above character.

A further object of the invention is to provide an improved drive mechanism for a reciprocating pump particularly a diaphragm pump of the above type.

Yet another object of our invention is to provide a drive mechanism of the above type having an accurate stroke adjustment readily operable by unskilled personnel.

Other objects of the invention will in part be obvious and will in part appear hereinafter.

The invention accordingly comprises the features of construction, combination of elements, and arrangement of parts which will be exemplified in the construction hereinafter set forth, and the scope of the invention will be indicated in the claim.

For a fuller understanding of the nature and objects of the invention, reference should be had to the following detailed description taken in connection with the accompanying drawing, in which:

FIGURE 1 is a horizontal plan view, partly in section, looking down into the interior of a pump incorporating the features of our invention, and

FIGURE 2 is a side elevation, partly in section, of the portion of the pump containing the pumping diaphragm and associated parts.

Referring to FIGURE 1, a pump embodying the features of our invention may be enclosed within an upper housing generally indicated at 10, mounted on a lower housing, generally indicated at 12, integral with a base flange 14. As shown in the drawing, the housing 10 may take a roughly cylindrical form with a horizontally disposed axis, while the housing 12 may be a vertically disposed cylinder. A motor 16, disposed within the lower housing 12, has a shaft 18 extending upwardly into the housing 10 to actuate a drive mechanism for the pump. In a manner to be described, the drive mechanism converts the rotary motion of the shaft 18 into a reciprocating motion imparted to a pump rod generally indicated at 20. The rod 20 extends into an extension of the housing, generally indicated at 22, where it actuates a pumping diaphragm operating in conjunction with other parts in the manner of a conventional diaphragm pump.

The drive mechanism includes a worm 24 fastened to the shaft 18 and meshing with a worm wheel 26. The worm wheel 26 is keyed to a shaft generally indicated at 28, rotating in bearings 30 and 32. The bearing 30 is

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mounted in a plate 34 held against a shoulder 36 in the housing 10 by a series of shoulder screws 38 working against a cover plate 40. The bearing 32 is mounted in an end plate 42 secured to the housing 10 by bolts 44 and preferably provided with an O-ring seal 46.

A skewed cylinder 48 is preferably formed integrally with the shaft 28 for rotation about the axis of the shaft. A bearing generally indicated at 50 is mounted on the cylinder 48 by means of a bushing 52. The interior surface of the bushing 52 is preferably canted at the same angle as the cylinder 48, and, thus, the axis of the bearing 50 is parallel to the axis of rotation of the shaft 28. The two axes do not necessarily coincide, and, when they do not, rotation of the shaft 28 imparts an eccentric motion to the bearing 50. When projected onto the horizontal plane (the plane of the drawing), this motion is reciprocatory in a direction perpendicular to the axis of the shaft 28. The reciprocation is imparted to the rod 20, which has an end 54 in contact with the outer race 56 of the bearing 50.

More specifically, for the purpose of the describing the eccentric motion of the bearing 50, the inner race 58 thereof, together with the bushing 52 and cylinder 48, may be considered as one solid member. If the position of the bushing 52 on the cylinder 48 is such that the axis of the bearing 50 coincides with the axis of the shaft 28, the inner race 58 will undergo a perfectly circular motion upon rotation of the shaft 28. In other words, the shaft, in effect, passes through the center of a disk comprising the inner race, bushing and skewed cylinder.

In the position illustrated, however, the bearing 50 is located on a portion of the cylinder 48 where the axes of the bearing and the shaft do not coincide. In effect, the shaft 28 passes through an off-center portion of a disk, and the periphery of the disk (the inner race 50) undergoes an eccentric rather than circular motion. The amount of eccentricity, i.e., the amplitude of the oscillatory motion described by inner race 50, increases with the distance between the axis of shaft rotation and the axis of the bearing 50, and this, in turn, depends on the position of the bushing 52 longitudinally along the cylinder 48. In the drawing, the position of the bushing corresponds to maximum eccentricity. If the bushing is moved to the left along the cylinder 48, the two axes move closer together, and the eccentricity is thereby reduced. Thus, the length of the pump stroke may be varied by moving the bushing 52 and bearing 50 thereon longitudinally along the cylinder 48.

Adjustment of the pumping stroke is effected by rotation of a knob 60 connected to the bearing 50 by way of a screw thread and linkage, which converts the rotary motion of the knob into transverse motion along the axis of the shaft 28. The knob 60 is affixed to a shaft 62 fastened to a threaded slug 64 by means of a pin 66. An oversized portion 62a of the shaft 62 bears against a bearing 68, and the slug 64 is in contact with a bearing 70, thereby preventing axial movement of the slug.

The thread on the slug 64 meshes with the thread in a plate 72 secured between sleeves 74 and nuts 76 on bolts 78. The bolts 78, in turn, are suitably fastened to a ring 80 fastened to the bearing 50. More particularly, the ring 80 is sandwiched between a segmented clamp 82 and an annular plate 84, the plate being secured to the clamp by means of screws 86. The clamp 82, which is in two or more segments, has a projecting rim 82a extending into a circumferential groove 56a in the outer race 56. Radial displacement of the segments forming the clamp 82 is prevented by their attachment to the plate 84, which maintains them as a unified structure. The plate 84 is also provided with clearance holes 88 permitting relative motion between the bearing 50 and the

clamping structure affixed thereto, on the one hand, and the ring 80 and bolts 78 on the other hand.

Accordingly, rotation of the knob 69 provides axial displacement of the bearing 50 and bushing 52 by way of the plate 72, bolts 78 and the clamping structure enclosing the ring 80. For accurate calibration of the stroke, the circumferential surface 60a of the knob 69 may be provided with suitable indicia, generally indicated at 90, accessible through a window in a tube 92 preferably integral with the cover plate 40. Adjustment of the pump stroke is linear with respect to rotation of the knob 69. That is, rotation of the knob through a given angle provides the same increment in stroke length regardless of the position of the knob. This makes calibration of the indicia 90 a relatively simple procedure and also facilitates adjustment of the pump stroke by servo or other automatic means connected to the shaft 62. Linear control of the stroke will, of course, also be obtained if such means are connected directly to the bearing 50.

While the outer race 56 of the bearing 50 does not follow the rotary motion of the inner race 58 about the axis of the bearing, it does undergo the eccentric motion of the bearing about the axis of rotation of the shaft 28. In other words, its motion is described by the rotation of the bearing axis about the shaft axis. This results in a sliding engagement of the clamp 82 and plate 84 with the ring 80, which is stationary with respect to rotation of the shaft 28. However, for several reasons, the frictional wear between these parts is minimal. In the first place, the amount of the relative motion between them for each revolution of the shaft 28 is quite small. In the second place, the interior of the housing 10 is preferably partially filled with a suitable lubricant, thereby minimizing friction. Furthermore, the angle between the axis of the cylinder 48 and that of the shaft 28 is kept small, so that minimal axial forces on the bearing 50 are developed by its movement against the pump rod, and the frictional forces between the ring 80 and the members engaging it are negligible insofar as the problem of wear is concerned.

It should also be noted that, with the construction described above, there is a slight rolling contact between the outer race 56 and the end of the rod 20. However, there is essentially no sliding contact between these parts, and, thus, wear on the engaging surfaces thereof is also at a minimum.

The housing extension 22 supports a pump head 94 affixed by means of screws (not shown) extending into an annular spacer 114. The spacer 114, in turn, is affixed to the housing extension 22 by means of further screws (not shown) isolated from the screws holding the head 94. As seen in FIGURE 2, a pair of check valves 98 and 100 are disposed between inlet and output lines 102 and 104 and passages 105 and 106 communicating with a pumping chamber 110. A pumping diaphragm 112, secured to the pump rod 20, reciprocates with the rod to provide a pumping action in the chamber 110.

The outer edge of the pumping diaphragm 112 is clamped between the pump head and the spacer 114, while the outer edge of a barrier diaphragm 116 is disposed between the spacer and the extension 22. The inner edge of the diaphragm 112 is clamped against a backing member 118 by the head 119a of a screw 119 threaded into the backing member. The member 118, in turn, has an integral stud 121 extending through a support disk 120 and threaded into the pump rod 20. The inner edge of the diaphragm 116 is thus clamped between the member 118 and the disk 120.

The pump rod 20 rides in a sleeve 124 secured in the housing extension 22. Leakage of lubricant from the housing 10 is prevented by a shaft seal 124a. A spring 130, bearing against an inner annular portion 22a of the housing extension 22 and a washer 132 and snap ring 132a affixed to the rod 20, urges the rod 20 against the outer race 56 of the bearing 50 and thus causes the rod

to follow the projected motion of the outer race in the horizontal plane.

The backing member 118 preferably has a diameter only slightly less than the interior diameter of the pump head 22, and, thus, its reciprocation thereof by the rod 20 provides an essentially piston-like action within the chamber 110, particularly with the pressure compensation described below.

Should the pumping diaphragm rupture, fluid leaking rearwardly thereof will encounter the barrier diaphragm 116, which prevents it from seeping into the interior of the housing 10. It will be noted that the diaphragm 116 is of a rolling fold type. Thus, the barrier diaphragm is under no tension during operation of the pump, and, thus, combined with its limp nature, minimizes fatigue and extends its life materially beyond that of the pumping diaphragm 112. Thus, assuming that both diaphragms are replaced whenever the pumping diaphragm deteriorates, there is complete insurance against leakage of the pumped fluid into the drive mechanism. The spacer 114 is provided with a valve generally indicated at 134 which vents the space between the pumping and barrier diaphragms and thus prevents a build-up of fluid pressure against the barrier diaphragm 116 upon rupture of the diaphragm 112. The valve 134 also controls the air pressure in the chamber 135 between the diaphragms 112 and 116 in such manner as to increase the output of the pump.

More specifically, the valve 134 includes an elastomeric disk 136 seated in a counterbore 137 communicating with the chamber 135 by way of a passage 138. The disk 136 is urged in place, to seal the passage 138 by a weak spring 139 acting between a backing plate 140 and a cup 141 threaded into the counterbore 137. A hole 142 vents the interior of the cup 141 to the atmosphere.

During the forward stroke of the pump rod 20 and backing member 118, i.e., toward the pump head 94, the diaphragm 112 tends to wrap around the curved forward surface of the member 118 in response to the increased pressure in the chamber 110. On the return stroke, however, there is a partial vacuum in the chamber 110, and, with atmospheric pressure in the chamber 135, this would pull the diaphragm 112 away from the member 118, thereby shortening the effective length of the pumping stroke.

Operation of the valve 134 rests on the fact that the volume of the chamber 135 is less during the forward stroke of the pump than on the return stroke. This is due largely to the configuration of the barrier diaphragm 116, which has a fold 116a extending rearwardly in the housing extension 22. Thus, during the forward stroke there is a tendency for the pressure in the chamber 135 to rise, and this opens the valve 134 to maintain the pressure at almost atmospheric value.

Then, the valve closes, and, on the return stroke, the pressure in the chamber 135 drops. This keeps the pressure differential across the diaphragm 112 at a low level, thereby minimizing the pull of this diaphragm away from the member 118. Thus, use of our construction lengthens the effective stroke of the pumps, with a resultant increase in output of as much as one third.

Since the drive mechanism is preferably immersed in a lubricant, it is desirable to seal the interior of the housing 10. This may be accomplished by the use of a gasket 143 (FIGURE 1) between the cover plate 40 and the housing, shaft seal 124a, and the length of the sleeve 124 along the pump rod 20, as well as the O-ring seal 46 described above. A passage 144 (FIGURE 2) in housing extension 22 vents the space behind the barrier diaphragm 116 and prevents a build-up in pressure of such lubricating fluid as might leak between rod 20 and sleeve 124 and seal 124a from the housing 10.

From the above, it will be apparent that we have provided an improved diaphragm pump with greater reliability than prior units of this type. In particular, leakage of the pumped fluid into the drive mechanism has been

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virtually eliminated. Furthermore, the drive mechanism is simple in construction and largely free of wear problems. Yet, it may be adjusted with a high degree of accuracy even during operation of the pump.

It will also be apparent that many changes may be made in the structure recited above without departing from the principles of our invention. Thus, the cylinder 48 need not have a circular cross section; for example, it may be polygonal or fluted, with the bushing 52 shaped to fit. Furthermore, it is not necessary that the shape of the bushing be such as to maintain the axis of the bearing 50 parallel to the axis of the shaft 28. However, this last condition is desirable, since there is then a slight rolling contact between the outer race 56 and the end of the stub 54, but absolutely no sliding engagement between these parts. Therefore, wear on the engaging surfaces thereof is at a minimum. To maximize this advantage, the pump rod 20 should move perpendicular to the bearing 50 and shaft 28 axes.

It will thus be seen that the objects set forth above, among those made apparent from the preceding description, are efficiently attained and, since certain changes may be made in the above construction without departing from the scope of the invention, it is intended that all matter contained in the above description or shown in the accompanying drawings shall be interpreted as illustrative and not in a limiting sense.

It is also to be understood that the following claim is intended to cover all of the generic and specific features of the invention herein described, and all statements of the scope of the invention which, as a matter of language, might be said to fall therebetween.

We claim:

In a reciprocating pump of the type comprising a pump head, a pump rod extending into said pump head and mounted for axial reciprocation therein and a drive mechanism for reciprocating said pump rod, the improvement

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in which said drive mechanism comprises a shaft mounted for rotation about a first axis, a cylinder on said shaft and rotating therewith, said cylinder having an axis angled to said first axis, a bearing unit mounted on said cylinder and extending around the circumference thereof, said pump rod being in contact with said bearing unit along a circumferential surface thereof, adjusting means for moving said bearing unit axially of said shaft, means for rotating said shaft, said bearing unit including a frictionless bearing having an inner race conforming to the circumferential surface of said cylinder and an outer race mounted for relative rotation with respect to said inner race about an axis parallel to said first axis, said adjusting means including an annular member disposed around and spaced from said shaft, means clamping said annular member to said outer race in such manner as to permit relative motion of said annular member and said outer race radially with respect to said shaft and prevent relative motion axially with respect to said shaft, said annular member being radially fixed with respect to said shaft, and means for moving said annular member axially along said shaft.

References Cited in the file of this patent

UNITED STATES PATENTS

540,843	Richards	June 11, 1895
2,083,073	Loeber	June 8, 1937
2,323,950	Wade	July 13, 1943
2,326,474	Malsbary et al.	Aug. 10, 1943
2,356,993	Glasner et al.	Aug. 29, 1944
2,414,806	Finney	Jan. 28, 1947
2,544,055	Staats	Mar. 6, 1951
2,612,837	Midgett	Oct. 7, 1952
2,776,573	Willi	Jan. 8, 1957
2,900,839	Mackintosh	Aug. 25, 1959