Nov. 2, 1943. W. T. LIVERMORE 2,333,323

PUMP

Filed Aug. 10, 1940

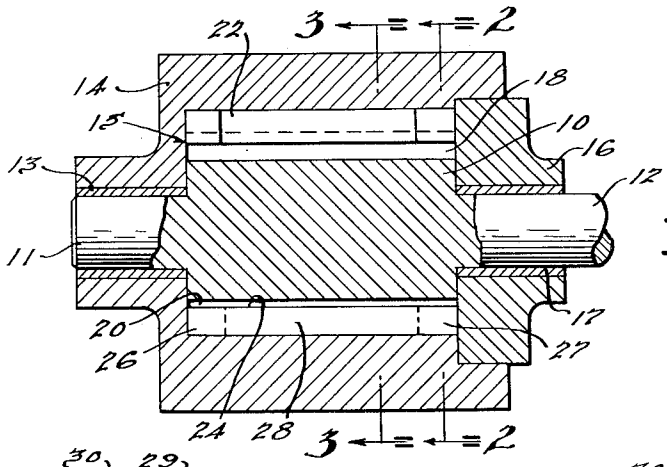


FIG. 1.

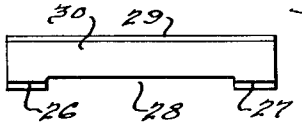


FIG. 4.

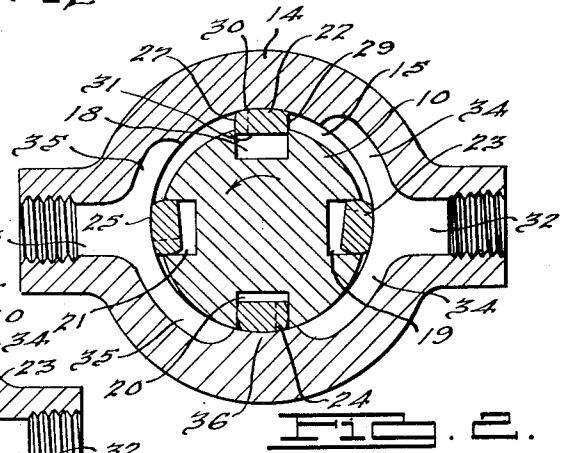


FIG. 2.

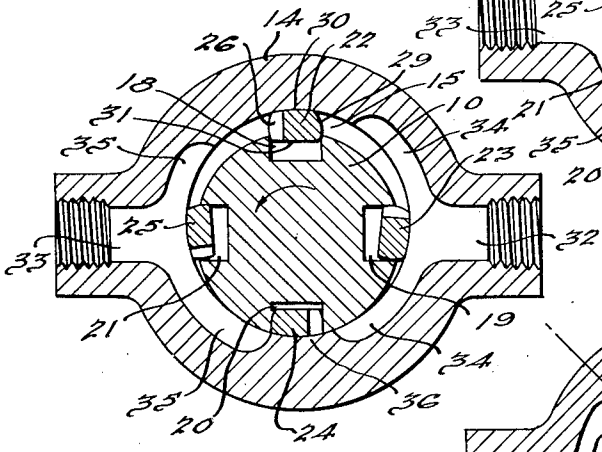


FIG. 3.

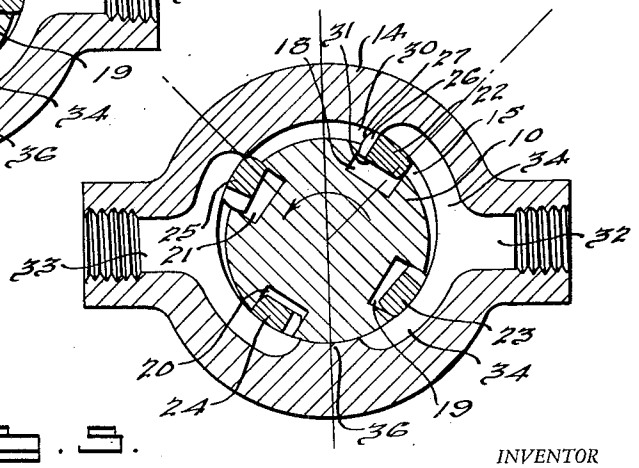


FIG. 5.

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PUMP

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Application August 10, 1940, Serial No. 352,081

2 Claims. (Cl. 103—136)

This invention relates to pumps of the so-called vane type.

There are two types of rotor propelled vane pumps which have come into the most common commercial use. In one of these, the vanes are formed by a member which extends completely through the rotor which carries the vanes for rotation, the vane member being allowed to slide in the diametrically positioned opening in the rotor so that the projecting ends of the vane will contact the wall of the pump chamber to carry fluid from an inlet to an outlet and thereby create a pumping action, and in this so-called through type of vane pump, the pump chamber is so shaped that the ends of each vane are continuously in contact with the wall of the pump chamber. The other more common commercially used type of construction is that of two short vanes projecting from the openings in the rotor with a spring in the center of the rotor to hold each of the two opposed vanes in contact with the wall of the pump chamber. It is noted that in both of the above types the vanes have contacting surfaces diametrically opposed to each other and the vanes are held in the openings in the rotor with relatively close clearance.

In my co-pending applications, Serial Nos. 220,308, filed July 20, 1938, and now abandoned, and 239,199, and now issued as Patent 2,278,131 of March 31, 1942, constructions for vane type pumps are disclosed wherein the shape of the bore is entirely independent of the action of the vanes, centrifugal force and pressure being utilized to position the vanes for pumping action, the vanes in both cases being free to move relative to the rotor and to be positioned by centrifugal force and pressure. In the application, Serial No. 220,308, the vanes are of the so-called through type but of a length less than the smallest diameter across the bore, while in the application, Serial No. 239,199 the vanes are of the so-called slipper type mounted in slots in the periphery of the rotor, having a limited freedom of both circumferential and radial movement and retained for driving action by the slots in the rotor, but positioned by the action of centrifugal force and pressure, there being circumferential looseness of the vanes allowing fluid pressure to act under the vanes to allow positioning by fluid pressure. As the result of experiment on the slipper type vane construction, it has been found that a certain shape of the vane limiting its circumferential movement while still making possible the entry of fluid under the vane because of the shape of the leading edge of the vane and with driving contact of the trailing edge acting as a seal results in a marked advantage in operation of the pump.

It is therefore a primary object of the present invention to provide a construction of rotor and fluid impelling vane or slipper of relative shape such as to retain the vane from circumferential movement relative to the slots or notches in the rotor and at the same time allowing free passage of fluid under the vane thereby leaving said vane free to assume a radial position determined by centrifugal force and pressure while at the same time affording a substantial seal by driving contact of said vanes with said notches.

It is a further object to afford exceptional smoothness of operation and pumping efficiency by specific relative positioning and arrangement of the several parts of the pump of the type herein disclosed.

The above and other objects of the invention will appear more fully from the following more detailed description and by reference to the accompanying drawing forming a part hereof and wherein:

Fig. 1 is a vertical cross section through the pump rotor and housing.

Fig. 2 is a section taken on the line 2—2 of Fig. 1.

Fig. 3 is a section taken on the line 3—3 of Fig. 1.

Fig. 4 is a plan view of one of the slipper type vanes.

Fig. 5 is a section similar to Fig. 3 but showing the vanes and rotor in a position different from that shown in either Fig. 2 or 3.

Referring to the drawing, the pump shown comprises a rotor 10 circular in section and having axial shaft extensions 11 and 12. The shaft extension 11 is rotatively journaled in a bearing 13 on the pump body 14 which has a circular pump chamber in which the rotor 10 is mounted for rotation. The circular pump chamber 15 is eccentric with respect to the center of the rotor 10 and the chamber 15 is closed by a cover plate 16 provided with a bearing 17 in which the shaft extension 12 is journaled. The shaft extension 12 extends beyond the end of the bearing 17 and is adapted to have driving means connected thereto for the purpose of rotating the shaft 12 and the rotor 10 and thus actuating the pump. It is understood that it would be possible to make the rotor 10 as a separate ring secured to a through shaft, in which case the pump could be easily adapted for assembly on a shaft available

in a motor or other unit without the provision of additional gearing or other driving means.

The rotor 10 has a plurality of notches in its periphery, here shown as the four notches 18, 19, 20 and 21. Each of the notches 18, 19, 20 and 21 has a slipper mounted in the notch, the four slippers here shown as 22, 23, 24 and 25. One of the slippers is shown in plan view in Fig. 4, and the other views of the drawing serve the purpose of details of other views of the slipper, and on consideration of the several views, it is apparent that each of the slippers 22, 23, 24 and 25 are mounted in the notches 18, 19, 20 and 21 in such a way that the slippers are free to slide radially in the notches and are held from circumferential movement by projections 26 and 27 which contact the wall of the notch with a clearance sufficient to allow radial movement in the notch, but only what might be termed a working clearance therewith. The space between the two projections 26 and 27 provide an opening 28 for the purpose of allowing fluid to circulate around and under the slipper. In considering the relative construction of the slipper and notches in the rotor, it is understood that as viewed in Figs. 2 and 3 the rotor rotates in a counterclockwise direction as shown by the arrow on the rotor, and the projections 26 and 27 are therefore on the leading edge of the fluid impelling slipper or vane, the slippers 22 being selected for illustration, while the trailing edge 29 serves to contact what might be termed the trailing edge and driving edge of the notch. It is also apparent on inspection of the Figs. 2 and 3 that the leading edge made up of the projections 26 and 27 are curved in cross section as well as the trailing edge 29, and the leading and trailing surfaces form parts of a cylinder in cross section. The notches are preferably formed by substantially parallel faces spaced apart on the periphery in amount slightly greater than the diameter of the cylindrical cross section of a slipper which might be termed the width of a slipper. It is also to be noted that the top surface of the slipper 30 is curved but of a radius approximately equal to the radius of the wall of the pump chamber, thus allowing the slipper to make contact with the chamber wall. The bottom surface 31 is here shown as a flat surface but might be provided with projections to prevent cohesive contact with the bottom of the notch. The radial thickness of each slipper is such that there is always a space between the bottom of the notch and the slipper when the slipper is in contact with the wall of the pump chamber.

An intake port 32 and an outlet port 33 are provided in the pump body preferably at substantially diametrically opposite points, the intake port 32 having extension 34 and the outlet port having extension 35, all extending circumferentially of the pump chamber 15. The intake and outlet extensions 34 and 35 are separated adjacent the point of tangency 36 between the rotor 10 and the pump chamber 15 as shown by the shape of the pump chamber at this point, the arcuate width of said separation being slightly greater than the width of any notch 18, 19, 20 or 21. The opposite ends of extensions 34 and 35 are separated by the circular wall of the chamber an arcuate distance equal to the arcuate distance between any two slippers, and this space constitutes the working zone.

As the shaft extension 12 is driven to rotate the rotor 10, centrifugal force and pressure act upon the slippers 22, 23, 24 and 25 and force them into contact with the wall of the pump chamber 15, and they are impelled around the pump chamber by contact of the trailing edges of the notches with the trailing edges 29 of the slippers. The slippers are free to move radially and to tilt under the influence of centrifugal force and pressure and thus are forced against the wall of the pump chamber. The slippers are thus caused to move the fluid from the intake port 32 to the outlet port 33, and the pressure under which the fluid is placed during movement through the working zone acts on the inner faces of the slippers to augment the action of centrifugal force in holding the slippers in contact with the wall of the pump chamber, and although the slippers are prevented from relative circumferential movement by the circumferential location in the slots imposed by the contact of the projections 26 and 27 with the leading edge of the slot and the contact of the trailing edge of the slot with the trailing edge 29 of the slipper, the fact that the remainder of the leading edge between the projections 26 and 27 is cut out to form the opening 28 makes it possible for the fluid to circulate and enter under the slipper, thus allowing the fluid pressure to act under the slipper and thus produce a uniform action thereof depending upon pressure conditions and the action of centrifugal force. Obviously the projections and relieved portions may take varied shapes and sizes as well as number without change in principle. The driving action of the notch on the trailing edge of the slipper produces a substantial fluid seal since there is uniform contact of these edges.

As will be noted on reference to Figs. 2 and 3, the slippers are circumferentially long as compared with the greatest radial distance between the point at which slipper 22 contacts the notch in the rotor and the wall of the pump chamber. Such a construction assures that there will be a sufficient exposed inner surface on each slipper so that the fluid pressure action on the slipper will always counteract the tendency of the driving force applied by the notch 18 to tilt the slipper away from seating contact with the bore. The arcuate shape of the leading and trailing edges of the slipper making a portion of a cylinder as shown in cross section makes it possible for the slipper to have a certain freedom of tilting movement and to be balanced, so to speak, by the effect of its shape and the action of pressure and centrifugal force to have its curved outer face held against the chamber wall. The particular cross sectional shape of the slipper as shown has been found to give satisfactory results, and the fact that the slippers are sufficiently wide, i. e., measured circumferentially, the fluid pressure action on the exposed inner surface of each slipper will be sufficient to overcome the tilting action of the driving force of the notches, and as a result, the slippers are continuously being held in sliding surface contact with the pump chamber.

As previously mentioned, and as shown in Figs. 2 and 3, the upper extensions 34 and 35 of the inlet and outlet ports are separated by the working zone or arc formed on the circular wall of the pump chamber, this arc having a circumferential length equal to the distance between any two adjacent slippers measured circumferentially along the same arc. To assure smoothness of operation it is necessary to equalize the rate of fluid displacement between a slipper just entering the working zone with the slipper just leaving it. Such a condition can be visualized

by considering Fig. 2 or 3 and assuming the pump to be in position such as Fig. 5 wherein the trailing edge of slipper **25** is just leaving the left hand end of the working zone **30** and the leading edge of slipper **22** is just entering the right hand end of working zone **30**. If a line is drawn through the point of tangency **36** and through the centers of both the circular pump chamber and the center of rotation of the rotor **10**, the geometry of the figure will show that the distance between the surface of the rotor **10** and the circular wall of the pump chamber, measured at the trailing edge of the slipper **25** will be equal to the distance between said rotor and pump chamber measured at the trailing edge of the slipper **22**. Furthermore that the points where these measurements are taken will be symmetrical or, in other words, equally distant on the arc of the pump chamber wall from the previously mentioned extended line drawn through the point of tangency and the center of rotation and the center of the circular pump chamber. It is, as above stated, desired that the rate of displacement of fluid by each of the adjacent slippers be equal when the leading slipper, i. e., **25** is leaving the working zone and the next slipper **22** is entering a working zone. And this equal rate of displacement is accomplished when the distances between the rotor and the circular surface of the pump chamber measured at the trailing edges of both slippers are equal for the two slippers under consideration, since the area at the trailing edge of a slipper between the rotor and the pump chamber wall is the effective displacement producing area thereof. It is obvious that this occurs with circular pump chamber and rotor when trailing edges of the wall contacting faces of the slippers are symmetrically positioned about the line above mentioned. However, since there is a definite width of slipper to be considered, the working arc must be displaced in the direction of rotation so that the trailing edge of one slipper will be just leaving the working arc, and the leading edge of the following slipper will be just entering the working arc when the above equal displacement occurs. From the geometry of the figure, it is obvious that this displacement of the working arc from symmetry is necessarily equal to ½ the width of a slipper measured at the narrowest section, as at **28** (Fig. 4), which width is the effective circumferential width of a slipper. Such a displacement has been shown on the drawing, Figs. 2, 3 and 5. This equalization of the rate of displacement by offsetting the working zone to compensate for slippers of material width has been found to effect great improvement in the smoothness and quietness of operation of the pump.

Although the principles of the invention have been illustrated by the disclosure of a specific mechanism, it is understood that various changes and modifications may be made without departing from the fundamental principles as indicated by the scope of the following claims.

I claim:

1. In a pump, a pump chamber having a circular wall and an inlet port and an outlet port, a rotor of substantially circular cross section with notches in its periphery mounted for rotation in said pump chamber with its axis of rotation eccentric to the center of said circular pump chamber and with its circumferential surface tangential to said circular wall of said pump chamber, a plurality of vanes carried in said notches, the arc of said circular wall between said inlet and outlet ports which bounds the working zone of said pump being equal in length to the distance measured along said arc between the adjacent port opening and closing edges, respectively, of two circumferentially adjacent vanes, and the position of said working arc relative to said rotor, said vanes and said circular pump chamber being such that said arc is displaced circumferentially in the direction of rotation of said rotor out of symmetry with a line drawn thru the point of tangency of said rotor pump chamber and said axis of rotation of said rotor in amount substantially equal to one-half the effective circumferential width of one of said vanes.

2. In a pump of the character described, a pump chamber having a circular wall and an inlet port and an outlet port, a rotor mounted within said chamber eccentrically with respect thereto, said rotor having a plurality of equally spaced vanes for exerting a pumping action when rotated between said rotor and said circular pump chamber wall, that portion of the arc of said pump chamber wall which bounds the working zone of said pump having a circumferential length equal to the distance measured along said arc between two circumferentially adjacent vanes, and said arc being circumferentially positioned relative to said inlet and outlet, said pump chamber wall, said rotor and said vanes so that the effective displacement producing area of a vane whose leading edge is entering said working arc is equal to the effective displacement producing area of a vane whose trailing edge is leaving said working arc when said areas are measured at the trailing edges of the wall contacting faces of said vanes.

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