

Nov. 30, 1943.

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2,335,284

ROTARY FLUID PRESSURE DEVICE

Filed Dec. 6, 1939

2 Sheets-Sheet 1

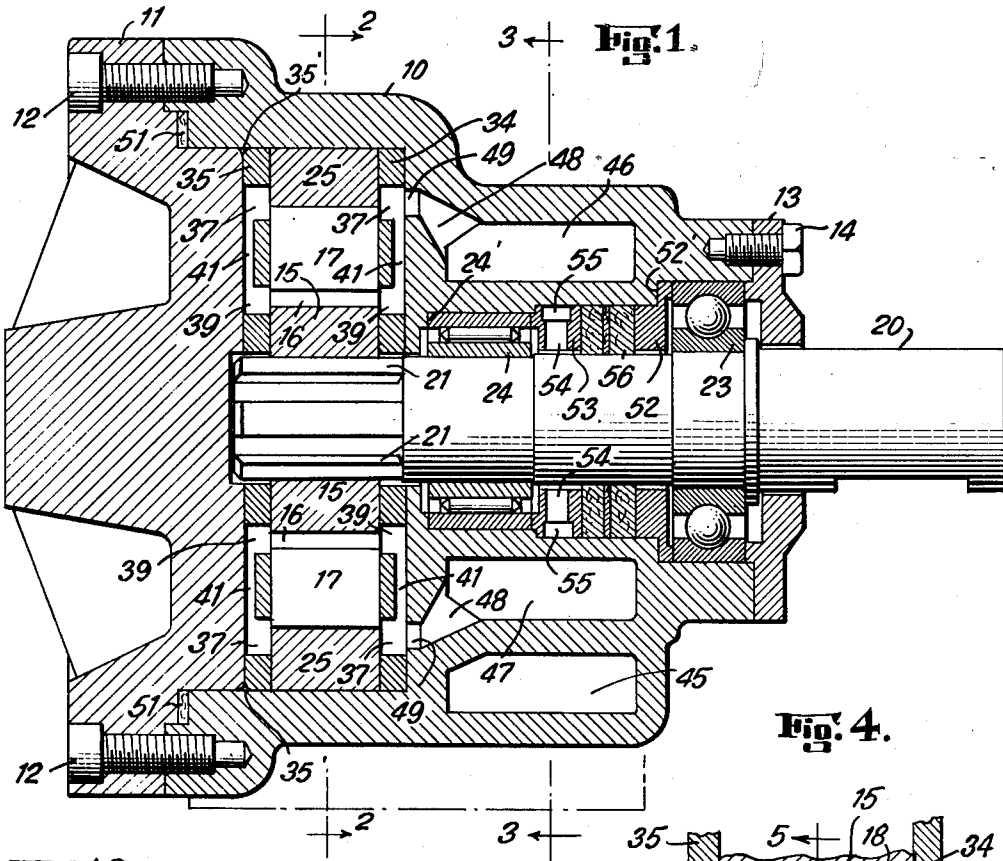


Fig. 1.

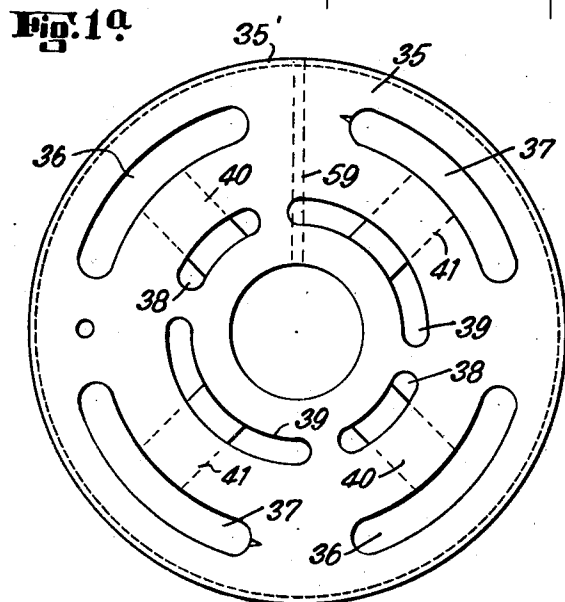


Fig. 1a.

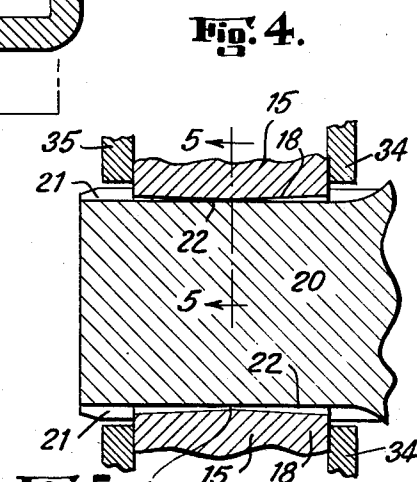


Fig. 4.



Fig. 5.

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2 Sheets-Sheet 2

Fig. 2.

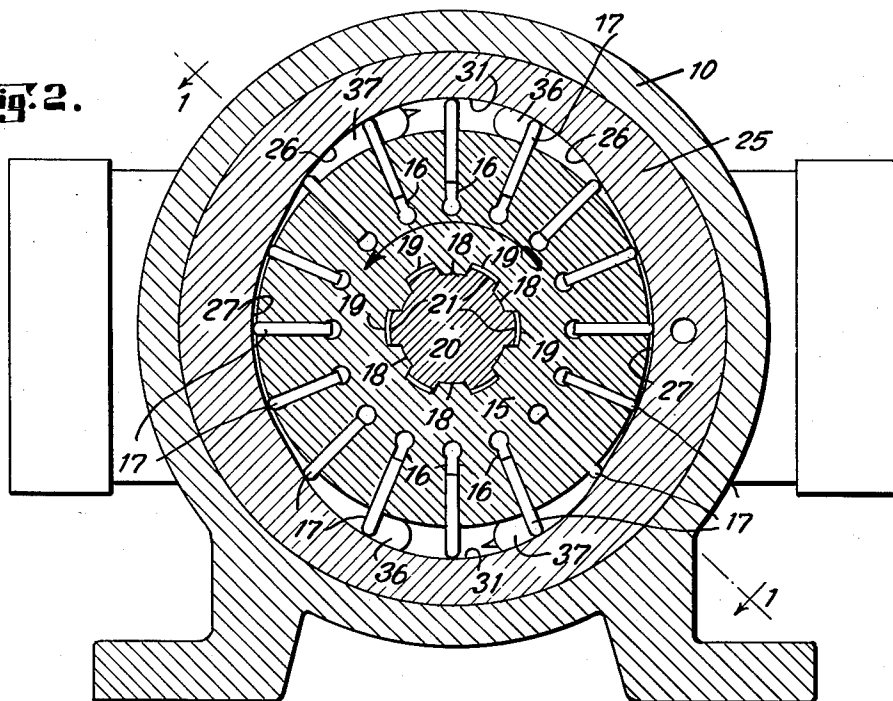
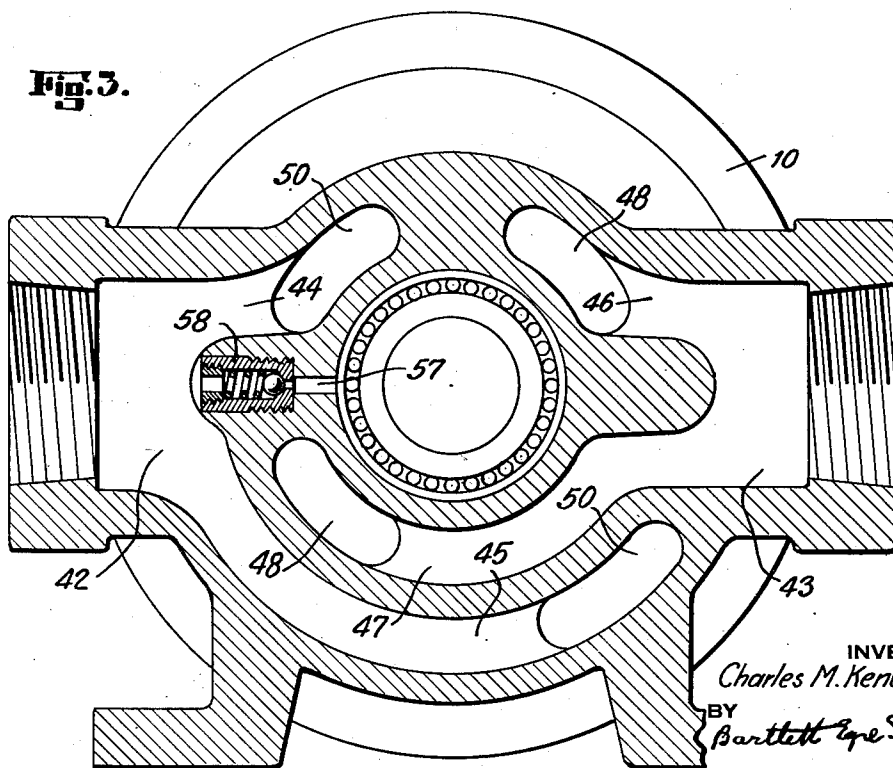


Fig. 3.



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UNITED STATES PATENT OFFICE

2,335,284

ROTARY FLUID PRESSURE DEVICE

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Application December 6, 1939, Serial No. 307,755

12 Claims. (Cl. 103—136)

This invention relates to vane type rotary fluid pressure devices such, for example, as pumps and motors and is concerned more particularly with devices of this sort in which the rotor is provided with a plurality of vanes arranged to move inwardly and outwardly thereof in a substantially radial direction during the operation of the device.

Fluid pressure devices of this general class find their widest use at present as hydraulic devices, that is, devices for handling or whose motive fluid is a liquid, such, for example, as oil, and the device of the present invention will be described in connection with such use. It will be understood, however, that certain features of the invention are applicable to pumps or motors operating with elastic fluids.

One object of this invention is an improved and simplified rotary vane type fluid pressure device providing for decreased cost, improved operation and greater ease of assembly. In one aspect the invention resides not only in the novel features per se but in the combination of co-operating elements, some of which are known in the art, which together provide the improved and simplified device.

Another object of the invention is to provide simple and improved driving and supporting means between the rotor and shaft of the fluid pressure device.

Another object of the invention is to provide simple and improved driving and supporting means between the rotor and shaft of the device permitting axial and angular relative movements between these parts, with the guide plates determining the plane of rotation of the rotor and vanes, and embodying certain other novel and improved features to be more fully explained hereinafter.

A further object is to provide simple, positive and improved means for preventing the entry of air and improved means for causing the vanes to move radially outward during intake when the device is used as a pump.

Other and more specific objects will hereinafter appear.

The invention will be understood from consideration of the accompanying drawings which illustrate, by way of example, the invention embodied in a pump of constant capacity. It will also be understood, however, that the invention is likewise applicable to pumps of variable capacity and to other fluid pressure devices, such, for example, as vane type fluid motors of either constant or variable capacity.

In the drawings:

Fig. 1 is a longitudinal sectional view taken along the line 1—1 of Fig. 2.

Fig. 1a shows an inner elevation of one of the members, for convenience termed the "end plate" or "cheek plate."

Fig. 2 is a view in vertical transverse section taken along the line 2—2 of Fig. 1.

Fig. 3 is also a vertical transverse section but with the shaft removed and is taken along the line 3—3 of Fig. 1, looking in a direction opposite to that of Fig. 2.

Figs. 4 and 5 are enlarged fragmentary sectional views illustrating in somewhat exaggerated manner the method of mounting of the rotor and the operative connection between the rotor and the shaft, Fig. 5 being taken along the line 5—5 of Fig. 4.

Referring now to the drawings, as shown in Figs. 1 and 2 the pump includes a casing 10 formed with an open-ended rotor cavity for the rotor 15 and associated parts. The rotor cavity is closed (Fig. 1) by an end head or cover member 11 which is attached to the casing 10 as by cap screws 12. The rotor 15 is provided with a plurality of vanes 17 which are movable in a substantially radial direction inward and outward in the vane slots 16 (Fig. 2). A vane track ring 25 surrounds the rotor and vane assembly and its inner circumferential surface 26 forms a track adapted to contact the radially outer ends of the vanes 17 and to guide the vanes in their inward and outward movement; the surface 26 will hereinafter be referred to as the "vane track."

The rotor 15 is hydraulically balanced with respect to all forces imposed thereon by fluid pressure. Hydraulic balance of forces acting on the rotor in a radial direction is obtained by dividing the space intermediate the periphery of the rotor 15 and the vane track 26 into two equal and oppositely positioned fluid sections, each fluid section comprising a pumping or working chamber flanked by an inlet area and an outlet area. As shown in Fig. 2, the division between the two fluid sections is effected by co-operation of the rotor 15 and the outer ends of the vanes 17 with the vane track 26 at the regions of the vane track's least diameter which in the present embodiment is adjacent the horizontal centerline. The vane track 26 is preferably provided at each of these points of division with an arc 27, for convenience termed the "sealing arc," substantially concentric with the rotor 15 and extending in a circumferential direction for a distance equal to at least the angular distance between a pair

of adjacent vanes 17 in order that there shall be substantially no radial movement of the vanes while passing thereacross. Clearance is preferably provided between the periphery of the rotor 15 and the sealing arcs 27 in order to permit slight eccentric movement or displacement of the rotor in a substantially radial direction, as will be more fully explained presently.

The pumping or working chambers of the two fluid sections are formed by means of two diametrically positioned arcs 31, preferably concentric with the rotor 15 and termed "pumping arcs" or "working arcs", which are located in the regions of greatest diameter of the vane track 26. Each pumping or working chamber extends in a circumferential direction for a distance equal to the space between two adjacent vanes 17 which at any given instant are moving in contact with the corresponding pumping or working arc 31. Fluid is admitted between the vanes as they move through the inlet areas toward the pumping chambers and is discharged as the vanes recede therefrom through the outlet areas.

The surface of the vane track 26 intermediate the sealing arcs 27 and pumping arcs 31 may be given any suitable curvature producing satisfactory rates of inward and outward movement of the vanes 17 as the rotor revolves. In the embodiment illustrated in Figs. 1 and 2, the vanes 17 are urged radially outward during intake and are kept in contact with the vane track during this portion of their rotary travel by centrifugal force alone. Under this condition it is important and necessary that the contour of the vane track 26 intermediate the sealing arcs 27 and pumping arcs 31 be such that the vanes 17 will remain continuously in contact therewith under the influence of centrifugal force alone and must be such as to require radially outward acceleration which said centrifugal force can produce. For this purpose each of these intermediate vane track surfaces may be given a curvature approximating a segment of a parabola, whereby there is substantially constant acceleration and deceleration of the vanes 17 in their inward and outward movement.

The sides or axial ends of the pumping chambers are closed by a pair of disc-shaped members 34 and 35 (Figs. 1 and 1a), for convenience termed "end plates" or "cheek plates," which are provided with holes at their centers for the shaft 20. The outer surfaces of the cheek plates 34 and 35 fit snugly against the wall surfaces of the casing 10 and end head 11 respectively and form substantially fluid tight fits with the several ports and passages to be presently described. The inner or opposing faces of the cheek plates 34 and 35 form fluid tight fits with the sides of the vane track ring 25 by which they are axially positioned with respect to the rotor 15 in such manner that the rotor is permitted to turn freely while its sides and the sides of the vanes 17 form substantially fluid tight running fits with the adjacent faces of the cheek plates 34 and 35. The cheek plate 34 will hereinafter be termed the "casing cheek plate" and the cheek plate 35 will be termed the "end head cheek plate."

The cheek plates 34 and 35 are each provided with coextensive ports (Figs. 1, 1a and 2), the ports of one cheek plate being axially opposite the ports of the other cheek plate when the parts are in position in the casing 10 so that all forces exerted upon the rotor and vanes in an axial direction by fluid pressure are thus completely balanced. The ports in the cheek plates 34 and 35

will be best understood from Fig. 1a which shows an inner elevation or the rotor face of the end head cheek plate 35. Referring to Fig. 1a, each cheek plate is provided with a pair of diametrically opposed arcuate inlet slots or ports 36 and a similar pair of outlet slots or ports 37; these ports are also partially shown in Fig. 2 and the outlet ports 37 are shown in the sectional view of Fig. 1. Fluid is admitted to the outer ends of the vanes through the inlet ports 36 of the casing cheek plate 34 and, similarly, fluid discharged by said vanes passes out through the outlet ports 37 of the same cheek plate. The ports 36 and 37 of the end head cheek plate 35 function chiefly as "balance ports" to contain fluid under the same pressure as that in the corresponding and axially opposite ports of the casing cheek plate 34 in order to produce hydraulic balance of the rotating parts, as already stated. Fluid may, however, also be admitted to and discharged from the inner ends of the vanes 17 through the ports 36 and 37 respectively of the end head cheek plate 35 as will be presently explained.

Each of the cheek plates 34 and 35 is also provided with two pairs of arcuate recesses or vane slot ports 38 and 39 which are positioned to register successively with the inner ends of the vane slots 16 as the rotor revolves. The vane slot ports 38 are connected with the inlet ports 36 by radial grooves or passages 40 on the outer faces of the cheek plates 34 and 35, as shown by the dotted lines of Fig. 1a; the vane slot ports 39 are similarly connected with the outlet ports 37 by the radial grooves or passages 41 as shown by the dotted lines of Fig. 1a and by the full lines in the sectional view of Fig. 1. The arrangement is such that the inner ends of the vane slots are connected with fluid under the same pressure as the fluid acting upon the outer ends of their corresponding vanes 19 while said vanes are passing between the sealing arcs 27 and pumping arcs 31, and vice versa, and hence the vanes 17 are substantially in hydraulic balance when they are moving radially while passing along the intermediate portions of the vane track 26. This arrangement materially reduces the wear on the vane track and frictional losses; it also provides maximum delivery from the pump due to the fact that fluid is also pumped by the inner end of each vane as it reciprocates in its vane slot as well as by the radially outer ends of the vanes as fully explained in co-pending application, Serial No. 271,874, filed May 5, 1939, since matured into Patent No. 2,255,782. The vane slot ports 39 are made of such length that they also connect with the inner ends of those vane slots 16 whose vanes 17 are in contact with the sealing arcs 27 and pumping arcs 31 so that fluid from the outlet ports 37, which are the high pressure ports, is supplied to the inner ends of such vanes to assist in holding their outer ends firmly in contact with the vane track surfaces at these points.

As shown in Fig. 3, the casing 10 is provided with an inlet conduit 42 and an outlet conduit 43. The inlet conduit 42 is adapted to convey fluid to the inlet ports of the two fluid sections of the pump and for this purpose is provided with a short branch 44 and a longer branch 45. The outlet conduit 43 is similarly adapted to convey fluid from the outlet ports of the pump's two fluid sections and is also formed with a short branch 46 and a longer branch 47. The end portions of the branches 46 and 47 connect with slanted passages 48 (Figs. 1 and 3) formed in the casing 10 which lead to and connect with arcuate slots

or ports 49 in the side wall of the rotor cavity substantially in register and co-extensive with the diametrically positioned outlet ports 37 of the casing cheek plate 34. The end portions of the branches 44 and 45 of the inlet conduit 42 similarly connect with slanted passages 50, as indicated in Fig. 3, similar to the slanted passages 48, and which lead to and connect with arcuate slots or ports (not shown) in the side wall of the rotor cavity substantially in register and co-extensive with the diametrically positioned inlet ports 36 in the casing cheek plate 34; this arrangement is similar to that illustrated and described in connection with the branches of the outlet conduit 43.

As also shown in Fig. 3, each branch of the conduits 42 and 43 is of a length just sufficient to connect with its corresponding slanted passages and does not extend beyond such connections. As also shown in Fig. 3, the shorter branches 44 and 46 are positioned at the same radial distance from the center of the pump casing and are also located in the same transverse planes. The longer branches 45 and 47 are also located in the same transverse planes but are positioned at different radial distances from the axial center of the pump and effect the "cross-over" from one side of the pump to the other. The present construction eliminates all pockets in which air or the like can become trapped and provides a fluid circuit all portions of which are in direct line of fluid flow. Positioning the two short branches 44 and 46 as shown and described also reduces the size and weight of the pump and gives greater latitude in design of the exterior of the pump.

Referring now to Fig. 1, the pump further includes a shaft 20 which extends through the central bore of the pump casing 10. One end of the shaft 20 projects outward beyond the casing 10 for connection to the source of power and its other end extends into the rotor cavity. The shaft 20 is supported by a pair of bearing elements, both of which are carried by the casing 10, these bearing elements being here shown as a ball bearing 23 and a needle-type roller bearing 24. The shaft is axially positioned by a pressed fit into the inner race of the ball bearing 23 and the outer race of the ball bearing 23 is held in position by the bearing cover 13, fastened to the casing 10 as by cap screws 14, and the rim of the packing gland 52 which is positioned intermediate the outer race of said ball bearing 23 and the wall surface 52' of the bearing cavity in the casing 10. The needle bearing 24 is held in place against the shoulder 24', at the inner end of the central bore of the casing 10, by means of an annular spacer member 53 having a hole at its center through which the shaft 20 passes and which is maintained in position against the outer race of the needle bearing 24 by the shaft packing 56 which is intermediate and pressed by said spacer member 53 and the packing gland 52.

The alignment of the shaft 20 is thus fixed in the casing 10 by the bearings 23 and 24, and the rotor 15 is operatively connected with and supportedly mounted on the end of the shaft 20 which projects into the rotor and cheek plate opening in such manner that the cheek plates determine the axial and angular positions of the rotor on the shaft and the plane of rotation of the rotor, without vibration and undue rubbing, together with the other advantages hereinafter set forth. One embodiment is shown in Figs. 1, 2, 4 and 5.

The end of the shaft 20 is formed with axially

extending splines 21 and the rotor 15 is formed in its central opening with mating splines 18 as best shown in Figs. 2, 4 and 5. The inner surfaces of the rotor splines 18 are adapted to engage or contact the bottoms of the recesses or slots 22 between the shaft splines 21; the depths of the recesses or slots 19 between the rotor splines 18 is greater, however, than the height of the shaft splines 21 for the purpose of assuring that there will be no interference with the engagement or contact between the inner surfaces of the rotor splines 18 and the bottoms of the recesses 22 and for the further purpose of permitting leakage fluid to pass axially through the openings thus provided from one side of the rotor to the other. The shaft splines 21 and the rotor splines 18 are both somewhat narrower than the corresponding slots or recesses into which they fit, as shown in exaggerated form in Fig. 5, thus providing circumferential clearance between the splines of the rotor and shaft, this clearance extending the full axial length of the splines. The splines are shown in Fig. 5 as positioned centrally with respect to their recesses but it is understood that the shaft splines 21 move into engagement with the rotor splines 18 when the pump is operated. The rotor opening diameter and the shaft diameter vary relatively to each other to a slight extent in the axial direction, and in the particular embodiment shown the rotor opening diameter is varied. To this end the rotor opening at the inner surfaces of splines 18 is of a varying diameter in the axial direction and the minimum diameter is made slightly greater than the diameter of the shaft at the bottoms of the shaft recesses 22. The arrangement employed in the present embodiment will be best understood by reference to the enlarged and exaggerated view of Fig. 4 in which it will be observed that the rotor opening diameter at the splines 18 is a minimum at a point midway between the sides or radial faces of the rotor 15 that is to say, at substantially the axial mid-point of the splines 18; this minimum diameter may be limited to virtually a point, or may extend axially for a short distance such as $\frac{1}{8}$ " or $\frac{1}{4}$ ", the preferred arrangement. There is thus a small clearance between the rotor splines 18 and the bottoms of the shaft recesses 22 at this midpoint, as can be observed at the bottom of the shaft 20 and as indicated at A; this clearance is preferably kept as small as practical, such, for example, as .001" to .003" or the like. The depth of the rotor splines 18 decreases (and the rotor opening thus increases) progressively from the midpoint toward both sides or radial faces of the rotor 15, as also shown in Fig. 4; the difference between the maximum and minimum depths of the rotor splines may be any preferred amount but in practice it has been found that a very small difference (such, for example, as .005") is adequate and it is usually desirable to keep this difference in diameters to a small amount in order to provide maximum contact areas on the sides of the rotor splines 18 that contact the sides of the shaft splines 21 in driving relation therewith. The above specifications are found to be desirable for a rotor of approximately $3\frac{1}{2}$ " diameter, and of approximately 1" axial width, the rotor having a rotor opening diameter of approximately 1" at the inner surfaces of the six splines 18. This arrangement permits limited radial movement or displacement of the rotor relative to the shaft, with the vanes 17 adjusting their protrusion from

the rotor conformably with such radial displacement or movement of the rotor to maintain the outer ends of said vanes in contact with the vane track 26. In practically all instances, the above described arrangement provides driving contact between one or more of the rotor splines and mating shaft splines, during at least a part of each rotation, which contact extends for substantially the full axial length of the rotor splines.

With this construction and arrangement the cheek plates 34 and 35 guide and determine the plane of rotation of the rotor 15, and not the shaft 20, due to the limited free relative axial, angular and radial movements between the rotor and the shaft. For example, the rotor 15 is free to move axially on the shaft 20 to its position between the opposing faces of the cheek plates 34 and 35 and its location axially on the shaft is solely determined by said cheek plates; also the rotor 15 is free to rock to a limited extent with respect to the shaft 20, to adjust itself to any slight variation in the position of said shaft 20 from a truly right angle relation to the opposing faces of the cheek plates 34 and 35, and as indicated, the faces of said cheek plates determine the angular position of the sides or axial faces of said rotor 15 relative to the axis of the shaft 20. The cheek plates 34 and 35 thus function in part as guide plates for the rotor and vane assembly. With the minimum difference in diameters of the rotor opening and the shaft support located at the axial midpoint of the rotor (as is preferred) the weight of the rotor 15 is equally distributed on each side or axial end of the point of support so that the rotor tends and maintains a substantially balanced condition of the weight thereof even when said rotor is rocked a small amount out of truly vertical relation to the axis of said shaft.

This construction and arrangement of the device is of great practical and commercial importance. It provides excellent operation as has been proved with both pumps and fluid motors embodying this construction and arrangement in tests extending over hundreds of hours at pressures of 1000 lbs. per sq. in. and more. The construction is extremely simple, and the cost of manufacture is low. For example, the flat-sides of the rotor and cheek plates can be readily finished to close limits, in groups if desired, and all associated parts are simple and easy to make accurately. Tendency toward binding of the rotor is eliminated or substantially reduced and the cap screws which fasten the end head to the casing may be pulled up hard without gripping the rotor, thus firmly positioning the cheek plates 34 and 35 and the vane track ring 25 in substantially fluid tight relation with each other and with the side walls of the casing 10 and head 11. Less time is required for assembly and the cost thereby reduced because, as already stated, the cap screws may be pulled up hard and no time need be lost in delicate adjustments.

It will be noted that several elements cooperate to provide the successful functioning of this construction and arrangement. For example, in the embodiment illustrated the shaft 20 is supported by bearings, both of which are carried by the casing 10 so that the shaft may be readily and accurately positioned at right angles to the side wall of the rotor cavity, thus reducing the possibility of improper alignment. Further, in this embodiment alignment and position of the shaft 20 is independent of alignment or position

of the end head, which is one of the reasons why the cap screws that attach the end head to the casing may be pulled up hard without delicate adjustment. The end head of the present embodiment, in fact, serves principally to close the rotor cavity and to hold the cheek plates and vane track ring in the necessary fluid tight relation with each other and the side walls of the rotor cavity and end head. Hydraulic balance of the rotor is also an important co-operating factor because it frees the shaft of all loads due to action of pressure fluid on the rotor, and this is particularly important when the device is to be operated at high pressures such as 1000 lbs. per sq. in. or the like. The action of the pressure fluid on the faces of the vanes passing across the two diametrically opposed pumping arcs (see Fig. 2) form a couple when the pumping arcs are equidistant from the center of rotation as is the case in the present instance. The shaft is thus freed of substantially all load except the torque and the load on the bearings is correspondingly reduced. In fact, this method of mounting the rotor directly upon the shaft adds no load to be carried by the shaft except the weight of the rotor and vanes and this is negligible. Hydraulic balance of fluid imposed forces acting on the rotor in an axial direction also leaves the rotor free to assume its operating position between the opposing faces of the cheek plates 34 and 35 so that there is no heavy rubbing action of the rotor thereon.

No difficulty is usually encountered in preventing the entry of air when devices of this character are operated as fluid motors because in almost all instances the fluid in both the inlet and outlet passages is under pressure above atmospheric. When these devices are used as pumps, however, the intake or inlet is usually under suction so that the fluid therein is under pressure less than atmospheric and much difficulty has been experienced in preventing the entry of air into the inlet portion of the pump's fluid circuit. This air is sucked in past the shaft and end head packings and enters the inlet conduit through the conventional passage employed to return thereto the fluid that leaks internally in the pump. While both end head and shaft packings have given trouble, the principal difficulty has been encountered in connection with the shaft packing. Stuffing boxes capable of functioning as an adequate seal against the entry of all air usually add a large amount of friction which causes heating and loss of power and are also usually too large for use in compact devices of the character here involved; low-friction compact sealing devices capable of giving satisfactory results are relatively expensive and add more to the cost of the device than is practical from the commercial standpoint. At the same time it is absolutely necessary to prevent the entry of all air as any air whatsoever entering the inlet conduit will cause noise and unsatisfactory operation of the pump and of the device that is operated by the pressure fluid delivered by the pump. Simple, inexpensive and positive means are provided by the pump of the present invention for preventing the entry of all air past both the end head and shaft packings, and will now be described.

Referring now to Figs. 1 and 2, a certain amount of internal leakage takes place in any device of this character as it is physically impossible to provide fits of the rotor and vanes with the cheek plates that permit the rotor and

vanes to move freely and at the same time prevent all leakage of fluid, particularly when high pressures are employed. Fluid leaking past the vanes and the portion of the rotor that extends radially outward beyond the vane slot ports 38 and 39 in the cheek plates 34 and 35 will pass directly into the inlet areas of the pump. Fluid leaking past the sides of the rotor and radially inward toward the shaft 20 will collect in the holes at the centers of the cheek plates 34 and 35 (Fig. 1) and fluid collecting in this center hole of the end head cheek plate 35 will pass axially through the rotor recesses 19 and between the shaft and rotor splines to the side of the rotor adjacent the casing cheek plate 34. Fluid leaking radially inward on both sides or axial faces of the rotor 15 thence passes axially through the central bore of the casing 10, along the shaft 20, through the needle bearing 24 and into the space between the shaft 20 and the annular spacer member 53. The shaft packing 56 prevents or resists further axial travel of the leakage fluid and it is forced through the radial holes 54 (Fig. 1) in the annular spacer member 53 and into the outer annular groove 55 thereon. The annular groove 55 connects with a passage 57 formed in the casing 10 and leading from the central bore of said casing 10 to the inlet conduit 42 as shown in Fig. 3. A pressure relief or spring-loaded check valve 58 is positioned intermediate the ends of the passage 57 and functions to prevent the passage of fluid from the inlet conduit 42 into the central bore of the pump casing 10 but is adapted to open to permit the passage of fluid from said central bore to said inlet conduit 42 whenever the pressure of the fluid in said central bore exceeds the pressure in said inlet conduit by a small predetermined amount.

With this arrangement the pressure of the fluid in the central bore of the casing 10 is kept substantially constant at a predetermined value which is low but which is above atmospheric pressure. The shaft packing 56 thus functions to prevent the escape of low pressure fluid from the pump and this presents a relatively simple sealing problem as it is much easier to seal against the outward leakage of fluid such as oil (which is the fluid usually circulated by these devices) than to seal against the entry of absolutely all air.

A radial groove 59 on the outer face of the end head cheek plate 35 connects the hole at the center of said cheek plate with the chamfer 35' at the circumference thereof, as shown by the dotted lines of Fig. 1a. Fluid under low pressure but above atmospheric pressure is thus supplied to the small annular space formed by the chamfered edges of the end head cheek plate 35 and the wall surface of the end head 11, and will either seal out air at this point or will pass to the end head packing 51 and there effect such seal, depending upon how tightly the end head is fitted into the bore of the rotor cavity. The groove 59 on the outer face of the end head cheek plate 35 preferably extends to the point of maximum elevation or top of said cheek plate in order to maintain fluid (oil) in the chamfered passage when the pump is not in operation. The groove 59 also permits the escape of fluid, if any, that may leak radially outward between the cheek plates and the vane track ring 25 or between the cheek plates and the wall surfaces of the rotor cavity and the end head 11; the vane track ring 25 and cheek plates are usually fitted into the rotor cavity with sufficient clearance to permit

the passage of leakage fluid in an axial direction along the circumference of the rotor cavity.

The shaft packing 56 (Fig. 1) is shown as consisting of three pieces of packing material and forms a very inexpensive seal. The use of multiple pieces of packing material for the shaft packing also facilitates the obtaining of a fluid tight seal as the edge of each piece offers resistance to the passage of fluid along the shaft in somewhat the same manner that conventional leakage "guard-grooves" prevent the leaking of fluid past a tightly fitted piston. Any other type of suitable shaft packing may be used, however, such, for example, as the well known oil seals made by the Chicago Rawhide Company and others.

The arrangement just described has given excellent results and no difficulty has been experienced in preventing the entry of air into pumps in which it has been employed. It thus eliminates a problem which in the past has proved serious and annoying and this result is accomplished in a very simple and inexpensive way. This arrangement is likewise capable of use in fluid motors but, as already stated, is not usually required for purposes of air exclusion.

While the device has been described as a pump it will also function as other types of fluid pressure device; for example, it will function as a fluid motor if supplied with pressure fluid for its operation. When functioning as a motor the high pressure areas are then the inlet areas and the low pressure areas are the exhaust or outlet areas. When employed as a motor it is necessary to introduce means tending to hold the vanes radially outward, particularly if the motor is to be operated at low rotative speeds. The term fluid pressure device as used in the claims is therefore to be understood to include both motors and pumps as well as all other forms of fluid pressure devices to which the invention is applicable unless otherwise specifically stated.

It is to be understood that the foregoing is merely an exemplifying disclosure and that changes may be made in the apparatus without departing from the invention which is defined in the appended claims.

I claim:

1. In a rotary fluid pressure device having a rotor provided with a plurality of vanes movable inward and outward thereof in a substantially radial direction, a casing therefor including a track for guiding said vanes in their in and out movement, a revoluble shaft extending into a central opening in said rotor and means for supporting said rotor directly upon said shaft for rotation therewith and for limited radial movement and rocking motion with respect thereto, including power-transmitting contacts between said rotor and said shaft extending for substantially the full axial length of said rotor.

2. In a rotary fluid pressure device of the multi-fluid-section type, a rotor having a plurality of vanes movable inward and outward thereof in a substantially radial direction, a casing having an open-ended cavity, a revoluble shaft supported entirely upon said casing and having an end projecting into said cavity, a vane track in said cavity for guiding said vanes in their in and out movement, means for balancing hydraulic forces acting on said rotor in axial and radial directions, means for supporting said rotor upon said shaft for rotation therewith and limited universal movement with respect thereto, and an end head for closing said cavity.

3. In a rotary fluid pressure device of the multi-fluid-section type, a casing having a cavity open at one of its ends and having a closed end provided with a wall surface, an end head closing said cavity and having a wall surface disposed opposite and substantially parallel with said first named wall surface, a pair of cheek plates positioned in said cavity, one of said cheek plates being positioned adjacent the wall surface of said cavity and the other adjacent the wall surface of said end head, the opposing faces of said cheek plates forming the side walls of the rotor chamber, a rotor having a plurality of vanes movable inward and outward thereof, said rotor being positioned between the opposing faces of said cheek plates, a revoluble shaft extending into a central opening in said rotor, a vane track for guiding said vanes in their in and out movement, means for balancing all hydraulic forces acting on said rotor and means for supporting said rotor directly upon said shaft for rotation therewith, axial movement thereon and limited universal movement with respect thereto, whereby the axial position of said rotor on said shaft and the plane of rotation thereof are determined by the opposing faces of said cheek plates in conjunction with centrifugal force, said last named means comprising axially extending splines on said shaft and mating splines on said rotor adapted to contact the bottoms of the recesses intermediate said shaft splines, said rotor splines having a minimum diameter adjacent the axial midpoint thereof and a varying diameter increasing toward their axial ends, said minimum diameter providing a clearance with respect to the diameter of the bottoms of said recesses intermediate said shaft splines, said rotor splines having a circumferential width less than the circumferential distance between said shaft splines and extending substantially the full axial length of said rotor to thereby provide power-transmitting contact between said rotor and said shaft for substantially the entire axial length of said rotor.

4. In a rotary vane type hydraulic pump, a rotor having a plurality of vanes movable inwardly and outwardly thereof, a two part casing therefor including a vane track for guiding said vanes in their in and out movement and also including a chamber having liquid-leaking seals with the side faces of said rotor, packing intermediate the parts of said casing, a central bore in one of said casing parts, a revoluble shaft in part disposed in said central bore and projecting outward beyond the casing part having said bore, said shaft having operative connections with said rotor, packing surrounding said shaft, fluid inlet and outlet passages in said casing for conveying liquid to and from said vanes, means for collecting in said central bore liquid leaking radially inward thereto, fluid connections between said central bore and the side of said shaft and casing packing adjacent said bore, fluid connection between said central bore and said fluid inlet passage, and a valve regulating the passage of collected leakage liquid from said central bore to said inlet passage to thereby maintain the fluid in said central bore and on the side of said packings adjacent said bore at a substantially constant relatively low pressure greater than atmospheric pressure.

5. In a rotary vane type fluid pressure device, a rotor provided with a plurality of vanes movable inward and outward thereof, a casing therefor including axially spaced and radially disposed guide surfaces determining the plane of rotation

of the rotor, a revoluble shaft extending into a central opening in said rotor, and supporting and driving connection between said shaft and the rotor comprising a group of splines on the rotor and a group of mating splines on said shaft arranged to transmit rotary power therebetween and to support said rotor on said shaft for free rotation in the plane determined by said guide surfaces, the splines of one of said groups having a varying diameter, and the splines of both groups being arranged to permit limited radial movement of said rotor relative to said shaft and to provide circumferential clearance between the mating splines of both groups throughout substantially the entire length thereof.

6. In a rotary vane type fluid pressure device, a rotor having a plurality of vanes movable inwardly and outwardly thereof in a substantially radial direction, a casing therefor including a vane track surrounding the rotor and vane assembly for guiding the vanes in their in and out movement, said casing also being provided with inlet and outlet fluid areas adjacent the periphery of the rotor, said vanes contacting said vane track to radially separate said fluid areas from one another, a pair of cheek plates having radially disposed guide surfaces determining the plane of rotation of the rotor, a revoluble shaft extending into a central opening in said rotor and arranged to provide support for said rotor, and means for supporting said rotor directly upon said shaft for rotation therewith, axial movement thereon and limited tilting with respect thereto for rotation in the plane determined by the guide surfaces of said cheek plates, said last named means comprising a group of axially extending splines and slots on said rotor and a group of axially extending mating splines and slots on said shaft, the splines of one group having a varying diameter to thereby provide limited axial rotor supporting contact with the bottoms of the slots of the other group, the splines and slots of both groups being spaced to provide circumferential clearance between the mating splines of both groups and the splines and slots of both groups also permitting limited movement of the rotor generally radially with respect to said shaft and to said vane track, with the vanes adjusting the extent of their inward and outward movement relative to the rotor to compensate for radial movement of the rotor relative to said vane track.

7. In a rotary vane type fluid pressure device, a rotor provided with a plurality of vanes movable inward and outward thereof, a casing therefor including axially spaced and radially disposed guide surfaces determining the plane of rotation of the rotor, a revoluble shaft extending into a central opening in said rotor and arranged to provide support for said rotor, axially extending splines in said central opening of said rotor and mating splines on said shaft cooperating with the splines in said central opening to provide supporting and driving connections between said shaft and the rotor comprising limited axial rotor-supporting contact between said rotor and said shaft and power-transmitting contact between said rotor and said shaft extending in an axial direction for a distance greater than the axial length of said rotor-supporting contact, said rotor supporting and power-transmitting contacts being arranged to permit free rotation of said rotor in the plane determined by said guide surfaces irrespective of limited departures in the angular relation of said plane relative to a plane normal to the axis of said shaft.

8. In a rotary vane-type fluid pressure device comprising a casing and a rotor with radially movable vanes, a vane track surrounding the rotor and vane assembly, a pair of guide plates disposed on axially opposite sides of said rotor and closely adjacent the axial sides thereof, means for balancing hydraulic forces acting on said rotor in axial and radial directions, a revoluble shaft extending into a central opening in said rotor, and driving and supporting means between said shaft and said rotor whereby said rotor is supported entirely upon said shaft for free tilting movement with respect thereto with said plates guiding and determining the plane of rotation of the rotor and vanes, said last named means comprising a group of axially extending splines and slots on said rotor and a group of mating splines and slots on said shaft, the splines of one group cooperating with the bottoms of the slots of the other group to provide limited axial rotor-supporting contact therebetween and the splines of both groups having throughout substantially the entire axial length thereof a substantially non-varying circumferential width less than the circumferential width of the corresponding mating slots therefor to provide clearance in a generally circumferential direction between the splines of the two groups.

9. In a rotary vane-type fluid pressure device comprising a casing and a rotor with radially movable vanes, a vane track surrounding the rotor and vane assembly, a pair of guide plates disposed on axially opposite sides of said rotor and closely adjacent the axial sides thereof, means for balancing hydraulic forces acting on said rotor in axial and radial directions, a revoluble shaft extending into a central opening in said rotor, and driving and supporting means between said shaft and said rotor permitting free axial and angular movements of said rotor on said shaft with said plates guiding and determining the plane of rotation of the rotor and vanes, said last named means comprising a group of splines and slots on said rotor extending axially thereon for substantially the full axial length thereof and a group of mating splines and slots on said shaft providing power-transmitting contact with the splines on said rotor for substantially the entire length of said rotor splines, the splines of one of said groups cooperating with the bottoms of the mating slots therefor to form limited axial rotor-supporting contact therebetween while permitting limited radial movement of the rotor relative to the shaft and the splines of the other group having radial clearance with respect to the bottoms of the mating slots therefor, the splines of both groups having throughout substantially the entire axial length thereof a circumferential width less than the circumferential width of the corresponding mating slots therefor to provide clearance in a generally circumferential direction between the splines of the two groups.

10. In a rotary fluid pressure device comprising a rotor provided with vanes movable inwardly and outwardly thereof, a casing therefor including a track for said vanes provided with two

working chambers each having inlet and outlet areas on opposite circumferential sides thereof adjacent the periphery of the rotor, means for substantially balancing fluid-imposed forces acting on said rotor in axial and radial directions including arrangement of said chambers and areas to provide substantially equal and opposite forces imposed on said rotor by action of fluid pressure in said chambers and areas, a pair of guide plates disposed on opposite axial sides of said rotor and in close proximity thereto, a revoluble shaft extending into a central opening in said rotor, and driving and supporting means between said shaft and said rotor permitting free axial and angular movements of said rotor on said shaft with the guide plates guiding and determining the plane of rotation of the rotor and vanes, said driving and supporting means also providing limited radial clearance between said shaft and the central opening of the rotor.

11. In a rotary vane-type fluid pressure device, a rotor having a plurality of vanes movable inwardly and outwardly thereof in a substantially radial direction, a casing therefor, a track in said casing for guiding the vanes in their in-and-out movement and provided with two working chambers each having inlet and outlet areas on the opposite circumferential sides thereof, said chambers and areas being arranged to provide substantial balance of forces imposed on said rotor in a radial direction by pressure fluid in said chambers and areas, a revoluble shaft extending into a central opening in said rotor, and means for supporting said rotor directly upon said shaft for rotation therewith, axial movement thereon and limited tilting movement with respect thereto, said last named means comprising a group of axially extending splines on said shaft and a group of mating splines formed in the central opening of said rotor with the splines of one group having a varying diameter, the splines of both groups being arranged to permit limited movement of said rotor relative to said shaft in a substantially radial direction and to also provide, throughout the axial length of the splines in the central opening of said rotor, substantially circumferentially disposed clearance between the mating splines of said two groups.

12. In a rotary fluid pressure device comprising a rotor provided with vanes movable inwardly and outwardly thereof, a casing therefor including a track for said vanes, means for substantially balancing fluid imposed forces acting on said rotor, a pair of guide plates disposed on opposite axial sides of said rotor and in close proximity thereto, a revoluble shaft extending into a central opening in said rotor and driving and supporting means between said shaft and said rotor permitting free axial and angular relative movements between said rotor and said shaft with the guide plates guiding and determining the plane of rotation of the rotor and vanes, said driving and supporting means also providing limited radial clearance between said shaft and the central opening of the rotor.

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