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THREE AREA VANE TYPE HYDRAULIC PUMP HAVING FORCE
MODULATING FLOW RESTRICTOR MEANS

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2 Sheets-Sheet 1

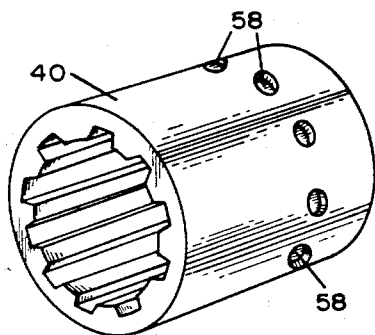
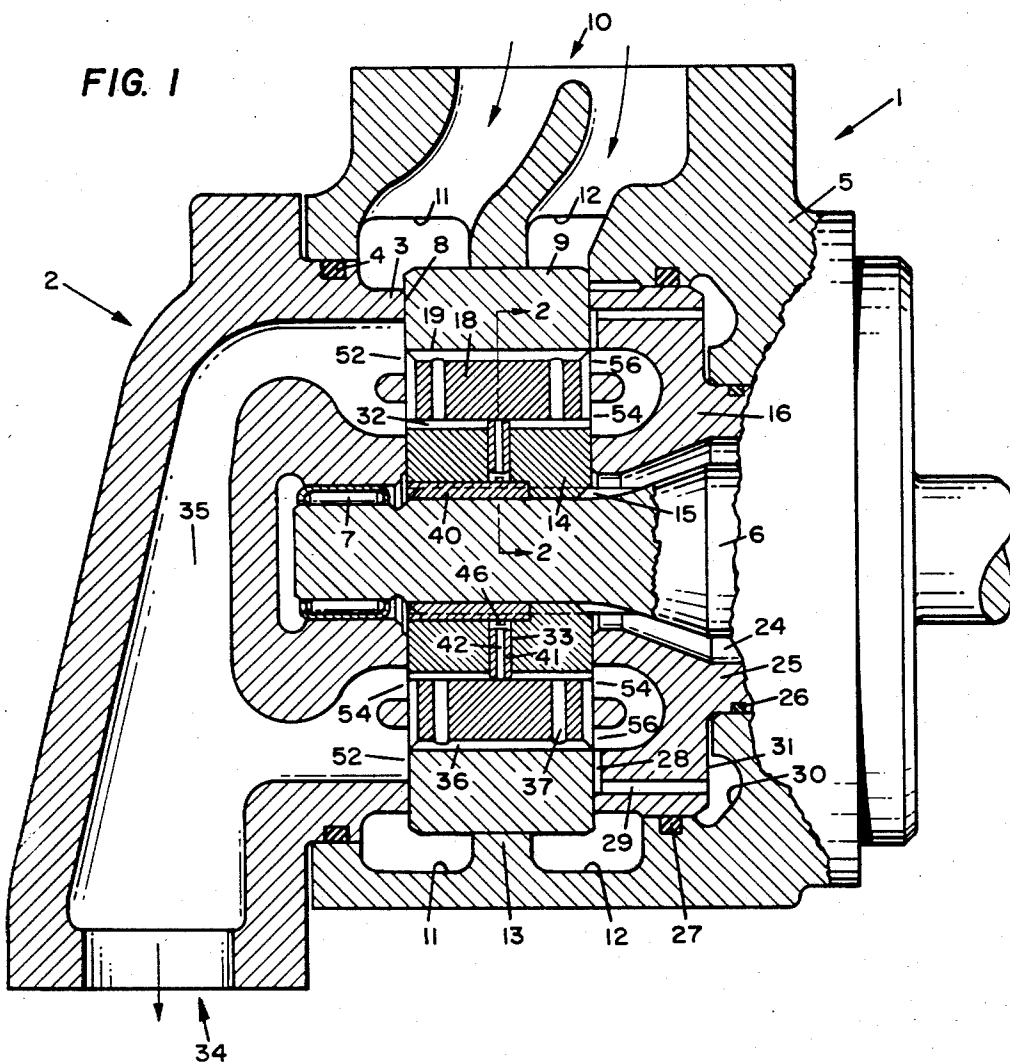


FIG. 3

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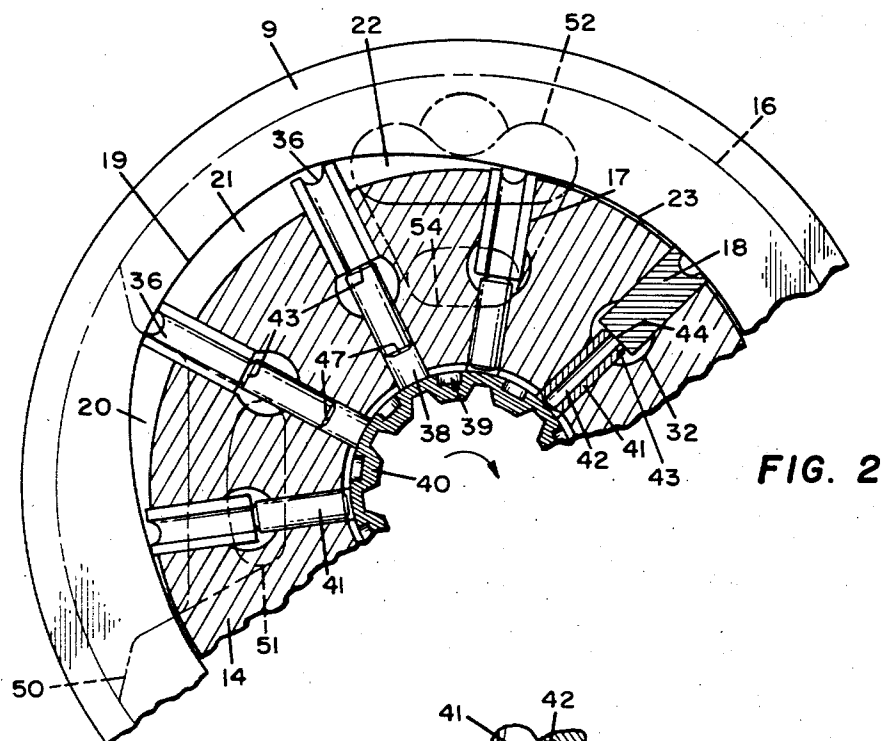


FIG. 2

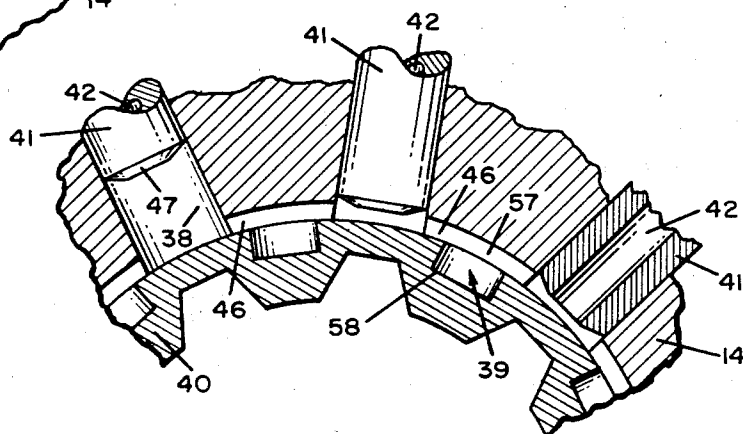


FIG. 4

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THREE AREA VANE TYPE HYDRAULIC PUMP HAVING FORCE MODULATING FLOW RESTRICTOR MEANS

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ABSTRACT OF THE DISCLOSURE

In a vane pump of the type which includes hydraulically operated pistons for vane control and wherein pressure to urge the pistons outwardly is applied to the pistons from a pressure zone through an interconnecting pressure chamber, flow restricting means are provided in the pressure chamber between each two adjacent piston cylinders, to reduce the pressure acting on the pistons as flow through the restrictors increases with pump speed. This automatically reduces the pressure force component on the pistons when the centrifugal force component is increased, so that the total force on the vanes at high operating speeds is modulated.

This invention is directed to improvements in hydraulic pumps of the vane type which include hydraulic means for vane control. More specifically, the invention relates to means for preventing damage to the vane tips and the cam ring such as is normally encountered due to increased total force acting outwardly on the vanes as the rotational speed of the pump increases.

A typical hydraulic pump of the vane type includes a rotor having a plurality of radially movable vanes carried in slots spaced around its periphery. These vanes engage the cam surface of a fixed stator or cam ring which surrounds the rotor. Inlet and outlet ports open at spaced positions into the area between the periphery of the rotor and the cam surface and are swept or traversed sequentially by the vanes as the rotor turns, whereby fluid received at the inlet port is transferred by the vanes to the outlet port.

In a vane pump of the three-area type, fluid pressures act on three areas associated with each vane, and the forces resulting from these pressures cooperate to urge each vane into operative engagement with the cam surface, i.e., to maintain a dynamic seal between the vane tip and the cam surface. By utilization of the three area vane control principle, a limited predetermined hydraulic force for urging or moving the vanes outwardly is obtained.

The fluid pressures acting on two of the areas associated with each vane are substantially equal but act in opposite directions, and the forces resulting from these pressures thus tend to counteract each other. The first area comprises a surface on the radially outer end of the vane and is subjected to pressure which urges the vane inwardly in its slot. The second vane area comprises a surface on the radially inner end of the vane. The second area is subjected to a pressure equal but opposed to that acting on the first area, which pressure urges the vane outwardly in its slot.

A third area associated with each vane is also subjected to a fluid pressure which urges the vane outwardly. Pressure on this third area provides a controlling hydraulic force which, in addition to centrifugal force, urges the vane outwardly to effect and maintain a fluid seal between the outer end of the vane or vane tip and the cam ring.

Three area pumps of the type to which this invention relates include a rotor assembly having an internal

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pressure chamber and a pressure differential operated pin or piston associated with each vane, the piston being slidable in a bore intersecting the pressure chamber and leading radially to the vane. Fluid pressure in the chamber acts on the pistons and provides the third area force holding the vanes outwardly. Pressure is supplied into this chamber through passageways from a high pressure zone of the pump whenever the pressure in the chamber tends to drop sufficiently below the pressure in the high pressure zone.

As shown in Adams et al. Patent No. 3,223,044, issued Dec. 14, 1965, of which one of the present applicants is a coinventor, the third area means may, for example, comprise a piston having an axial passage and an outer end which forms a check valve with the inner end of the respective vane for regulating the flow of pressure fluid from the pressure port into a pressure chamber interconnecting the inner ends of all the pistons. Whenever the external pressure acting on a piston exceeds the pressure in the internal chamber tending to hold the piston out, fluid from the vane slot will flow into the chamber through the longitudinal passage in the piston until a substantial pressure equilibrium is reached. This pressure acts equally on all of the pistons and on the vanes to hold the vanes against the cam surface.

While three-area pumps of the type shown in the Adams et al. patent have effected a significant improvement in operating performance, such pumps have occasionally been subject to internal damage in operation. In particular, operation of such pumps under high speed conditions has been found to result, over a period of time, in undesirable wear of the vane tips and of the cam ring. This damage occurs when the pump is operated at high speeds and the vane tips are subjected to a high total force caused by the large centrifugal forces arising from the high speed operation, in conjunction with a high fluid pressure force arising from the third area means.

As mentioned, the total force acting on a vane to force it outwardly in its slot has two components: a fluid pressure force component dependent on the output pressure of the pump, and a centrifugal force component arising from and dependent on the speed of operation of the pump. It has long been accepted as normal by those skilled in this art that the total outward force on the vanes of a pump increases as the speed of the pump increases.

This invention arises from our determination that, at constant pump output pressure, the total force urging the vanes outwardly against the cam ring at higher speeds should be lower than the total force urging the vanes outwardly at the minimum operating speed of the pump. We have also found that the total outward force acting on the vanes can be reduced or modulated by the use of flow restrictors or orifices in the passage means for conducting hydraulic liquid to the third areas on the inner ends of the piston pins. These flow restrictors or orifices function to progressively reduce the hydraulic component of the total force acting to urge the vanes outwardly against the cam ring as the speed of the pump increases.

In carrying out the foregoing, it has been an objective of this invention to provide means by which wear of the vane tips and cam ring will be reduced by modulating the fluid pressure force component, and hence, the total force acting outwardly on the vanes during high speed pump operation, thus lengthening the useful life of the pump.

Another object of this invention has been to modulate by reducing the fluid pressure force component acting on the vanes as the speed of operation of the pump increases, i.e., to reduce the total outward force on the vanes as the speed of the pump increases, pump output pressure remaining constant, so that the total force, i.e., the sum of

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the fluid pressure force and the centrifugal force on the vanes, decreases with pump speed.

The further objects and advantages of the present invention will be apparent from the following description, reference being made to the accompanying drawings wherein a preferred embodiment of the invention is disclosed.

In the drawings:

FIGURE 1 is an axial section of a three-area type vane pump incorporating a preferred embodiment in accordance with the principles of this invention;

FIGURE 2 is a partial transverse or radial section taken along line 2—2 of FIGURE 1;

FIGURE 3 is a perspective view depicting the sleeve structure of the preferred embodiment; and

FIGURE 4 is an enlarged view of a portion of the sleeve, pressure chamber, rotor and pistons shown in FIGURE 2.

The three-area vane pump shown by way of illustration in the drawings includes a housing or casing formed by a body casting 1 having a generally cylindrical interior chamber, and an end cap 2 having a cylindrical boss 3 which telescopes into the end of the body and is sealed thereto by an O-ring 4.

The end wall 5 of the body opposite end cap 2 includes a bore through which the pump operating shaft 6 extends. Shaft 6 is supported for rotation in this bore by a bearing (not shown) which is secured against axial movement in the bore. Shaft 6 extends from the body 1 into end cap 2 and is carried for rotation therein by a needle type roller bearing 7 mounted within a central bore in the end cap.

Cylindrical boss 3 of the end cap is finished to form a flat inner surface which is clamped against a side or radial face 8 of a cam ring 9. It may be mentioned here that the cam ring itself as well as the housing and cam ring together are sometimes referred to in the art as a stator.

A fluid intake passageway 10 extends radially into body 1 and communicates with a pair of internal annular channels 11, 12 which encircle the internal cavity within the body. These annular channels 11, 12 distribute fluid from the intake passageway 10 to suction ports to be described.

The cam ring 9 is supported radially by an annular rib 13 formed in the body 1 between the annular channels 11, 12. The cam ring 9 encircles a rotor 14 which is connected to shaft 6 through a motion permitting spline joint 15 that permits proper running alignment between the rotor, the flat surface of the cylindrical boss 3, and a movable check plate 16. Rotor 14 is provided with a plurality of radial vane slots 17 in each of which a vane 18 is mounted.

The cam ring 9 has a cam surface 19 that is contoured to provide a balanced or symmetrical pump construction in which there are diametrically opposite low pressure or suction zones 20, fluid transfer zones 21, high pressure or exhaust zones 22, and sealing zones 23 formed between the cam surface and the rotor 14 (see FIG. 2). In order to provide the opposed zones, the cam surface 19 is formed in part from a first pair of arcs of equal radii which extend across the fluid transfer zones 21 and, in part, by a second pair of arcs of shorter radii than the first pair of arcs which extends across the sealing zones 23. These pairs of arcs are interconnected by cam surfaces which extend across the low and high pressure zones 20 and 22 respectively.

Cheek plate 16 is finished to provide a smooth flat radial surface on the inner side thereof which abuts the cam ring 9, and has a central bore 24 surrounded by a cylindrical boss 25 which extends into the bore in the wall 5 of the body 1 and is sealed thereto by an O-ring 26. The outer cylindrical surface of cheek plate 16 is sealed to body 1 by an O-ring 27.

The cheek plate 16 is movable axially in the body 1 and

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is urged toward rotor 14 by fluid pressure supplied from the high pressure zone 22 through passageways 28 and 29 to a pressure chamber 30 formed between the body and the outer face 31 of the cheek plate. The cheek plate functions in the nature of an axially movable, non-rotatable piston under the pressure supplied by the fluid in chamber 30 to maintain it in engagement with the adjacent side face of the cam ring 9.

Intake passageway 10 communicates through annular channels 11, 12 around the cam ring 9 to suction ports spaced 180° apart. Two suction ports, one of which is shown at 50 in FIG. 2, are formed in cheek plate 16 and are fed by channel 12, and two additional suction ports (not shown) are formed in end cap 2 and are fed by channel 11. These suction ports in the end cap and cheek plate are identical in shape and are axially aligned with the suction zones 20 between rotor 14 and cam surface 19. Each suction port has a branch passage, the opening of one of which is designated at 51, whereby the suction port communicates with the inner ends 32 of vane slots 17 in the rotor 14 as well as with the inlet zones 20.

As shown in FIG. 1, the end cap 2 includes two diametrically opposed crescent-shaped exhaust or pressure ports 52, 52 which are spaced substantially 90° from the suction ports. Similarly, pressure ports 56, 56 are formed in cheek plate 16 which are axially aligned with the pressure zones 22 and with ports 52, 52 in the end cap. Each pressure port 52 and 56 communicates with the inner ends 32 of the vane slots 17 in the rotor as the vane slots pass the ports through branch ports 54. Pressure ports 52, 52 are connected with a fluid outlet or delivery chamber 34 by a passageway 35 in the end cap 2.

In the direction of rotor movement (clockwise as shown by the arrow in FIG. 2), the cam surface 19 progressively recedes from the periphery of the rotor 14 across the suction zones 20. In the transfer zones 21 cam surface 19 has a nearly constant radial spacing from the rotor, and across the exhaust zones 22 the cam surface progressively approaches the rotor 14 as it comes into close proximity with the periphery of the rotor 14 in the sealing zones 23. Fluid from the suction ports 50 is drawn into the fluid transport pockets defined between the successive vanes as those pockets become larger when the vanes 18 move through the suction zones 20. The fluid is positively displaced from the pockets as the volume thereof diminishes when the vanes move through the pressure zones 22, to thereby effect a pumping action.

Each vane 18 is provided with grooves 36 which are formed in its outer edge and opposite side edges. One or more channels or bores 37 are also provided in each vane which communicate between the outer groove 36 of the vane and the inner end 32 of the vane slot. The grooves 36 and channels 37 insure that the fluid pressure acting on the first area or outer end surface or tip of any given vane will be substantially balanced at all times by the pressure acting on the second area or inner end surface of that vane.

For the pump to operate at high efficiency it is necessary to maintain a continuous sealing engagement between the vane tips or outer end surface with the cam surface 19, regardless of changes in the arcuateness of the cam surface. To provide the hydraulic pressure for this purpose, one or more radial bores or piston cylinders 38 is formed in the rotor 14, extending inwardly from the inner end 32 of each vane slot 17. The bores 38 are interconnected at their inner ends with an annular pressure chamber 39 having fluid under pressure therein. Thus, fluid can flow into and out of pressure chamber 39 only through radial bores 38.

As best seen in FIG. 4, the pressure chamber 39 communicates with the bores 38 through orifices or flow restrictors 46 having cross-sectional dimensions which are small relative to the diameter of the bores 38. Within the pressure chamber 39, and intermediate the flow restrictors 46, are fluid velocity reducing chambers 58. The pressure

chamber 39 is constructed, in part, by defining in rotor 14 an annular groove 57 having the same cross-sectional dimensions as the flow restrictors 46. The chambers 58 are formed in a sleeve 40, as best seen in FIG. 3. Thereafter the sleeve is fitted and sealed in an axial recess in rotor 14 with the chambers 58 intermediate the flow restrictors 46, thus forming the pressure chamber 39. Hence, the pressure chamber 39 includes the flow restrictors 46, the chambers 58, and the groove 57.

A generally cylindrical pin or piston valve element 41 is received in each radial bore 38. Each piston 41 includes an axial bore 42 and is slidable in its cylinder 38 with which it is closely fitted so that leakage of fluid along the external walls of the piston is negligible. The outer end of each piston 41 is conically tapered as at 43 and forms a valve with the flat inner edge surface 44 of each vane 18. The lower end surface of each piston is preferably chamfered, as at 47, and presents a surface with which the fluid pressure force within the chamber 39 may cooperate to force the piston outwardly against the inner edge surface 44 of each vane. The admission of fluid to the radial bores 38 is regulated by the balance of forces between the fluid pressure force acting inwardly upon the conical taper 43, tending to open the valve, and the opposing force arising from the fluid pressure in chamber 39 and the centrifugal force, tending to close the valve. The length of the piston 41 is such as to permit it to move into and out of engagement with the flat inner edge surface 44 of the vane 18, regardless of the position of the vane in its slot.

In operation, valve 41, 44 at the outer end of the piston functions in the manner of a check valve. When fluid pressure at the inner end 32 of a vane slot acting upon the conical taper 43 of the piston 41 sufficiently exceeds the pressure in chamber 39, the piston is moved inwardly in its bore 38 and valve 41, 44 opens. Pressure fluid in the inner end 32 of vane slot 17 flows inwardly through bore 42 toward chamber 39 and restores, maintains, or increases the fluid pressure in the pressure chamber as necessary to balance the fluid pressure acting to open the valve 41, 44. This action occurs when the vane slot 17 traverses a pressure zone 22, for fluid pressure in the zone 22 is usually the highest in the pump and the pressure in chamber 39 is somewhat lower. When a vane 18 is traversing a suction zone 20, the fluid pressure in chamber 39 exceeds the opposing fluid pressure on end 43 of piston 41, and the piston is held against the inner end 44 of the vane, to close valve 41, 44 and to urge the vane 18 radially outwardly in its slot 17.

As previously mentioned, there are two force components which make up the total force acting on each vane 18 urging it outwardly against the cam ring 9. These two force components are the force arising from fluid pressure within chamber 39 acting on piston 41 and the centrifugal force resulting from rotary movement of each vane 18 and its associated piston 41. As pump speed increases, the centrifugal force component increases and the total force on each vane will therefore increase with it if the other force component remains constant. The increased centrifugal force acting outwardly on the vanes 18 at high pump speeds can, in the absence of this invention, cause excessive or undesirable wear of the vane tips and/or the cam ring 9. By this invention, total force is modulated as the pump speed increases. The force modulation takes place through means which reduce the fluid pressure force component progressively as pump speed increases.

Reduction of the fluid pressure force component occurs with the higher fluid flow rates into and out of the pressure chamber 39 through the flow restrictors 46 during higher speed pump operation. Each time a piston 41 moves outwardly, as its vane 18 passes through a suction zone 20, fluid flows into its bore 38 from chamber 39 through restrictors 46 with a resultant pressure drop across the restrictors 46. Since at higher speeds of rotation each vane passes through the suction zones 20 more

often per unit of time, the flow through the restrictors 46 is greater per unit of time. Hence the pressure drop across the orifices is greater at higher speeds. At higher speeds this establishes a relatively lower pressure at the inner end 47 of each piston in the suction zones.

Moreover, flow into the chamber 39 through the bores 38 from the high pressure zones 22 occurs at a higher rate at higher speeds, and tends to cause a reduction of pressure in the chamber relative to pump outlet pressure. During low speed pump operation the fluid flow rate into and out of the pressure chamber 39 through the flow restrictors 46 is lower, and, therefore, the pressure drop as the fluid passes through the flow restrictors 46 is lower. Thus, for a given pump output pressure, at low pump speed the fluid pressure force component is relatively high with the centrifugal force component being low, and at high pump speed the fluid force component is relatively low (due to the increased pressure drops caused by the flow restrictors) with the centrifugal force component being high, thereby providing a modulating or control effect on the total force. This force modulation reduces vane tip and cam ring wear at high speed operation which would otherwise result from the greater force.

While the embodiment of the invention above disclosed is preferred, we do not desire to be limited thereby as the invention is equally applicable to other structural embodiments. For instance, instead of providing a flow restrictor on each side of an intermediate chamber 58 and the respective piston bores 38, the objectives of the invention can be obtained through the use of a single restrictor between each pair of bores 38. It will also be seen that the annular groove 57 may be formed in the sleeve 40 rather than in the rotor 14, if desired.

Those skilled in the art will also recognize that the chambers 58 may be omitted, and that the pressure chamber may wholly be formed by narrow groove 57 itself. However, the pressure drop across such a relatively long thin orifice tends to vary rather widely with the viscosity of the hydraulic fluid, with the result that pump performance varies with temperature. For that reason we prefer the embodiment shown, which is less sensitive to fluid viscosity changes.

The invention can also be utilized in solid pin three area pumps of the type shown in Adams et al. Patent No. 2,832,293.

What is claimed is:

1. In a hydraulic pump of the vane type: a rotor having vanes mounted in vane slots for reciprocatory movement therein, each vane having associated with it a pressure operated piston slidable in a bore in said rotor, the said piston being movable into abutting contact with said vane in said slot to urge said vane outwardly, passage means interconnecting the inner ends of said pistons with one another for supplying fluid under pressure from a pump delivery chamber to the inner ends of said pistons in said bores, said fluid tending to urge the same outwardly, said passage means including relatively restricted fixed area orifice means between each two adjacent bores, whereby the operating pressure at the pistons is reduced with respect to the pressure in the pump delivery chamber as the speed of the pump increases.
2. The pump of claim 1 wherein said passage means comprises an annular internal chamber within said rotor intersecting each said bore, said orifice means being formed in said annular chamber between said bores as areas of relatively restricted cross sectional area.
3. The pump of claim 2 wherein said annular chamber has two orifice means in series relation between each two adjacent bores, said two orifice means being spaced in said annular chamber between said two bores by an enlarged area within said annular chamber.
4. The pump of claim 2 wherein said annular chamber is formed in part by said rotor and in part by a sleeve

secured within a central shaft bore in said rotor, said sleeve being adapted to engage an operating shaft.

5. A rotor assembly for a hydraulic pump of the vane type comprising:

- a rotor having vanes mounted in vane slots for reciprocatory movement therein,
- a piston associated with each vane, each piston being movable in a radial bore extending inwardly in the rotor from the vane slot,
- a conduit for applying pressure fluid to all of said pistons tending to urge the same outwardly into contact with said vanes, said conduit interconnecting said radial bores in series relation, said conduit entering said bores inwardly of the pistons therein,
- a plurality of fixed flow restrictors spaced along the length of said conduit between said bores, at least one flow restrictor being provided in said conduit between each two adjacent bores through which fluid in said conduit must pass to flow into and out of said bores,
- and means for supplying fluid under pressure into said conduit.

6. The assembly of claim 5 wherein said conduit contains a pair of flow restrictors separated by a plenum between each pair of adjacent bores.

7. A rotor assembly for a hydraulic fluid pressure energy translating device of the vane type, comprising:

- a rotor having vanes mounted in vane slots for reciprocatory movement therein,
- a piston associated with each vane, each piston being movable in a radial bore extending inwardly in the rotor from the vane slot,
- a generally circular conduit formed within said rotor for applying pressure fluid to all of said pistons tending to urge the same outwardly into contact with said vanes, said conduit interconnecting all said radial bores in the rotor inwardly of said pistons, said conduit restricting free fluid flow therethrough between said bores in operation,
- said conduit between each pair of bores having an enlargement therein.

8. In a hydraulic pump of the vane type, structure comprising:

- a rotor having vanes mounted in vane slots for reciprocatory movement therein,
- each vane having associated with it a pressure operated piston means, the said piston means being arranged to urge said vane outwardly,
- passage means for supplying fluid from a pump delivery chamber to each said piston means thereby tending to move said piston means and urge the vane outwardly in its slot,
- said passage means including pressure dropping orifice means through which fluid must flow to reach said piston means for reducing the operating pressure at the piston means with respect to the pressure in said delivery chamber as the rotational speed of the pump increases.

9. The structure of claim 8 wherein said passage means comprises a chamber connected with each of said piston means, said pressure dropping orifice means being formed between said chamber and said piston means as a passage of relatively restricted cross-sectional area through which fluid must flow to reach said piston means.

10. The structure of claim 9 wherein said chamber is formed in part by said rotor and in part by a sleeve secured within a central bore in said rotor, said sleeve being adapted to engage in operating shaft.

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