June 15, 1965 D. FIRTH ETAL 3,188,973

HYDRAULIC PUMPS AND MOTORS

Original Filed April 14, 1960 3 Sheets-Sheet 1

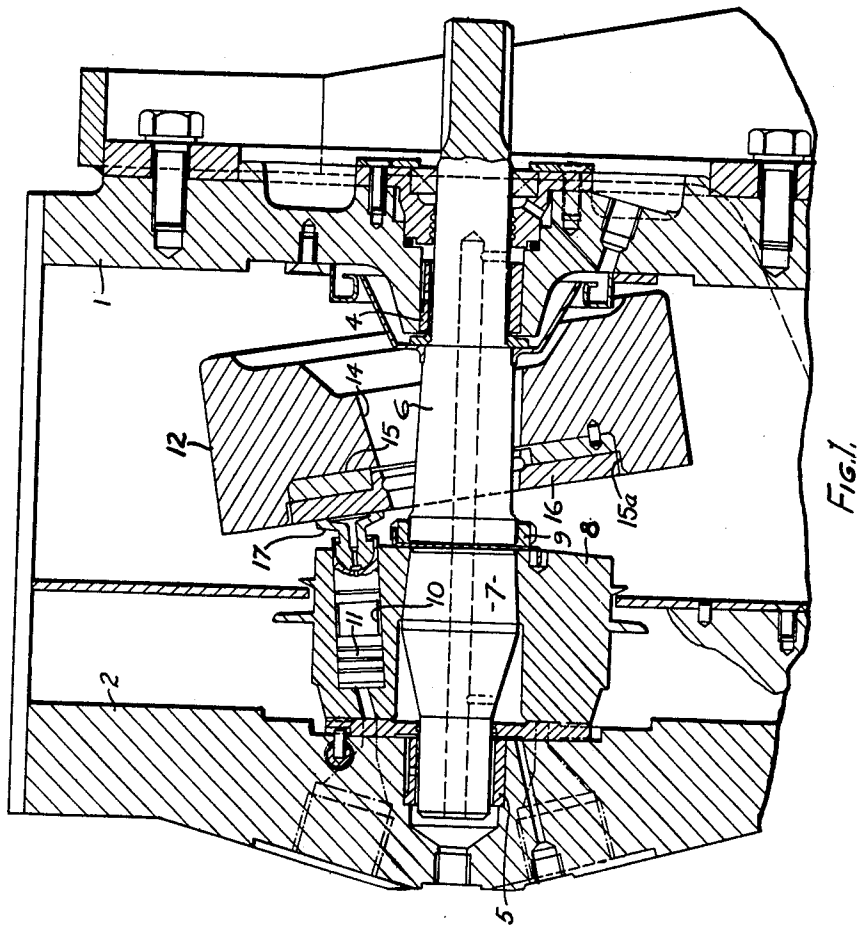


FIG. 1.

INVENTORS

Donald Firth

James D. Hamilton

BY Kenneth O. Flocks

ATTORNEY

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HYDRAULIC PUMPS AND MOTORS

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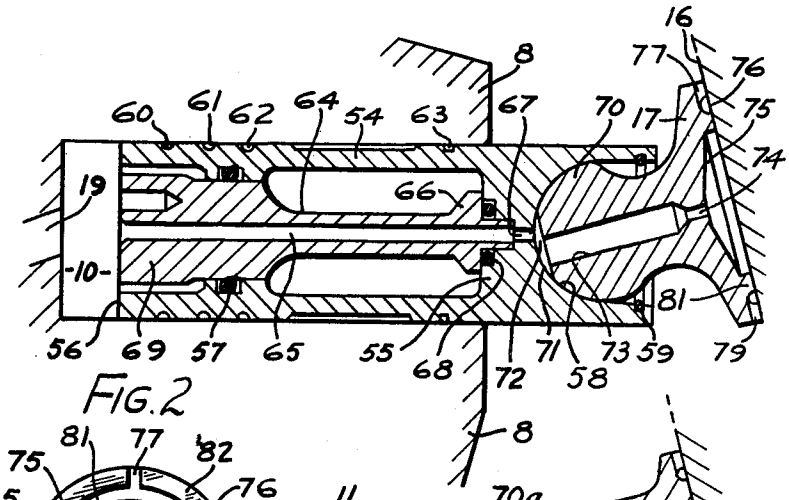


FIG. 2

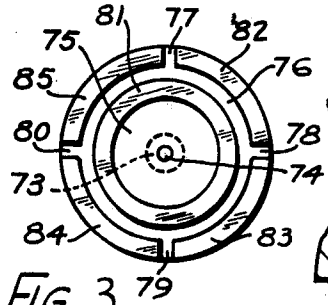


FIG. 3.

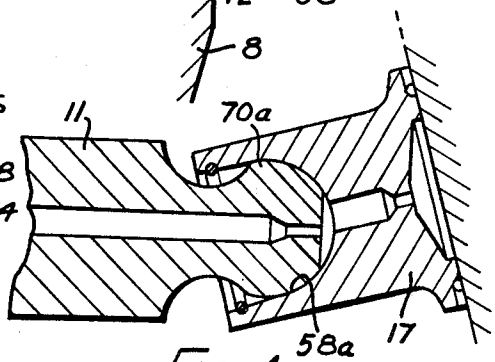


FIG. 4.

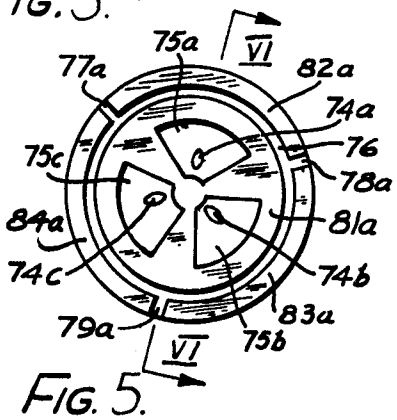


FIG. 5.

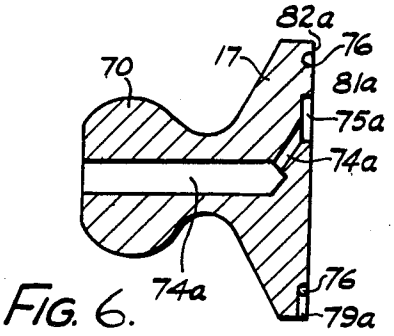


FIG. 6.

INVENTORS

Donald Firth
James D. Hamilton

BY Karl W. Flocks

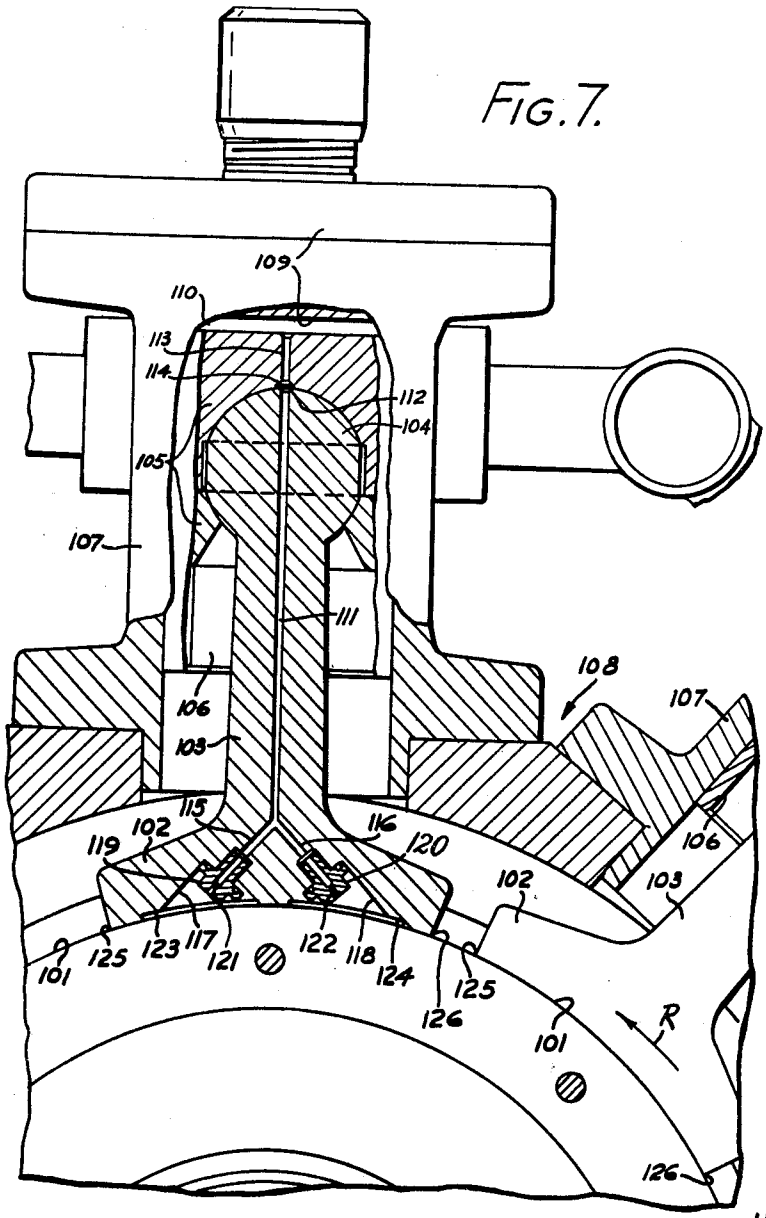
ATTORNEY

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FIG. 7.



INVENTORS

Donald Firth

James D. Hamilton

BY Karl W. Flocks

ATTORNEY

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HYDRAULIC PUMPS AND MOTORS

Donald Firth and James D. Hamilton, Glasgow, Scotland, assignors to Council for Scientific and Industrial Research, London, England, a body corporate of the United Kingdom

Original application Apr. 14, 1960, Ser. No. 22,335, now Patent No. 3,120,816, dated Feb. 11, 1964. Divided and this application Oct. 24, 1963, Ser. No. 318,679
Claims priority, application Great Britain, May 1, 1959, 14,890/59
1 Claim. (Cl. 103—174)

This application is a division of application Serial No. 22,335 of Donald Firth and Roger H. Y. Hancock filed April 14, 1960, now U.S. Patent No. 3,120,816 issued February 11, 1964, entitled Hydraulic Pumps and Motors.

This invention relates to hydraulic machines of the positive displacement type, whether pumps or motors, in which the reaction to the thrust exerted on a piston by the fluid pressure in a cylinder is taken through a slipper or shoe on the piston rod working on a tilted or eccentric reaction surface. The machine may be a swash plate machine, in which the reaction surface is constituted by the inclined surface of the swash plate, or a radial cylinder machine in which the reaction surface is a relatively fixed cam or eccentric on the axis of the machine shaft.

One source of loss of efficiency in positive displacement pumps and motors in which the piston rod bears through a slipper against an inclined or eccentric surface is that, especially at low speeds, the slipper tends to tilt, and this materially increases the risk of metal-to-metal contact and high friction.

A positive displacement hydraulic motor or pump according to the present invention has each piston and slipper assembly designed to operate with a high degree of hydrostatic balance of pressures between the piston and the slipper bearing surface, i.e. the pressure of a lubricant film at the slipper surface is substantially a direct function of the bearing load. In this way, metal-to-metal friction and the tendency of the slipper to tip or tilt are materially reduced or even eliminated.

To this end, the piston and slipper assembly is conveniently designed so that oil at the working pressure in the cylinder at any given instant is available at the slipper bearing surface which engages the reaction member-eccentric, swash plate, or the like—and which controls the reciprocatory motion of the piston.

Preferably the piston and the slipper, together with any intervening connecting rod, are drilled through in register to provide a continuous oil duct which includes a metering restriction at or near the end adjacent the working face of the slipper, and the latter had oil-retaining recesses adjacent its normally leading and trailing edges from which oil can leak at a controlled rate dependent on the resistance of the metering restriction.

Practical embodiments of the invention will now be particularly described, by way of example only, with reference to the accompanying drawings in which:

FIG. 1 is an axial cross-section of a swash plate pump or motor, all but one piston and cylinder being omitted for convenience;

FIG. 2 is an enlarged sectional view of a piston and cylinder;

FIG. 3 is a plan view of the slipper in FIG. 2;

FIG. 4 is a fragmentary sectional view similar to FIG. 2 showing a detail modification;

FIG. 5 is a view similar to FIG. 3 of a modified form of slipper;

FIG. 6 is a sectional view on the line VI—VI of FIG. 5 and

FIG. 7 is a radial fragmentary section through a radial cylinder eccentric type pump or motor.

The pump illustrated in FIG. 1 consists of a main frame having front and back end plates 1, 2 clamped by four pillars (not shown). Each plate 1, 2 carries a journal bearing 4, 5 respectively for a short rigid drive shaft 6 Adjacent the bearing 5 in the back end plate 2, shaft 6 is formed with a locking taper section 7 on which is locked a cylinder block 8. This block is drawn up on the taper by a back-nut 9 on the shaft. The cylinder block 8 contains a number of cylinders 10 whose axes are mutually inclined inwards towards the back end plate 2. A piston 11 in each cylinder is reciprocable under the control of a normally fixed swash plate 12 carried on trunnions (not shown) by which it can be angularly adjusted on an axis normal to the shaft 6. The swash plate 12 has a central conical aperture 14 through which the shaft 6 passes, the dimensions of this aperture being sufficient to allow for adjustment of the angle of the swash plate to the shaft 6.

The working face of the swash plate is recessed at 15*a* to accommodate an annular bearing pad 15 and an annular slipper plate 16. The latter is free to rotate under the frictional drag of slippers 17 each of which is engaged with a respective piston 11. For clarity of illustration in FIG. 1, only one cylinder 10, piston 11 and slipper 17 is shown. The bearing pad 15 is locked against rotation by means of a dowel. Thus, the friction between the piston slippers 17 and the slipper plate 16 causes the latter to tend to follow the slippers 17 around the shaft 6, while the eccentricity of the plate, due to the tilt of the swash plate 12, with respect to the axis of the shaft 6, causes the slippers 17 to trace a path over the working surface of the slipper plate which is not of constant configuration. Thus, wear of the plate is distributed over an area greater than the annulus which would be traced by a single slipper 17 if the plate 16 were stationary.

Each piston 11 consists of a hollow sleeve 54 (FIG. 2) closed at its outer end to form a crown 55 and which is a snug fit in the cylidner 10 and has four external oil control grooves 60, 61, 62, 63. Into the outer or working end 56 of the sleeve 54 is screwed a headed stem 64 having an axial oil feed bore 65 passing therethrough. The outer end of this stem is shouldered at 66 and has a spigot which passes through an oil seal 68 in the piston crown 55. An oil seal 57 prevents leakage of oil between the head 69 of the stem and the end 56 of the sleeve. Coaxially with the bore 65, the crown 55 is drilled through to form an oil duct 67 which opens into a hemispherical socket 58 which forms a seating for a hemispherical pivot 70 integral with the slipper 17. The pivot is retained in the socket 58 by means of a spring ring 59.

The hemispherical pivot 70 is flattened at 71 to leave a small pocket 72 around the end of the oil duct 67 within the seating 58. A radial oilway 73 is drilled through the pivot and leads into a constriction 74 which communicates with a central recess 75 in the face of the slipper 17 (see also FIG. 3). This recess can thus receive oil from the cylinder 10 through the bore 65 in the stem 64, the duct 67 in the piston crown 55, the pocket 72, the oilway 73 in the hemispherical pivot 70, and the constriction 74. Thus it will be seen that the pocket 72 must always be proportioned so that the pivot 70 never closes the duct 67 at a limit position of its angular deflection.

The sliding surface of the slipper 17 also has an annular groove 76 adjacent its periphery, this groove having no communication with the central recess 75, but communicating with the atmosphere through four radial notches 77, 78, 79, 80. The sliding surface of the slipper 17 thus effectively consists of five lands—an inner annulus 81 and four peripheral part-annuli 82, 83, 84 and 85. For convenience of identification on the drawing, these lands are horizontally shaded in FIG. 3.

The purpose of the above-described arrangement of oil recess and grooves is to combat the tendency of a slipper **17** to tip on the swash plate due to friction between the spherical pivot **70** and its seating **58**. This tendency increases with increase of thrust between the slipper and the swash plate, which occurs over approximately one half of each revolution of the particular cylinder concerned, and is coextensive with the oil delivery stroke of the piston. Since all the coacting thrust surfaces are fed with oil at delivery pressure (neglecting any pressure drop due to flow resistance in the circuit **65, 57, 72, 73, 74**) the load will be transmitted at each bearing point through a film of oil at a pressure which is a direct function of the axial thrust on the piston. Thus, there will be a film of oil between the lands **81 . . . 84** and the slipper plate **16**, and also between the pivot **70** and its seating **58**. The whole piston and slipper assembly **11, 17** is thus substantially hydrostatically balanced.

In practice, there will be a pressure drop in the circuit **65 . . . 74** due to leakage between the slipper lands **81 . . . 84** and the slipper plate **16**, but this can be kept to a very low value by careful design and accurate machining of the bearing surfaces. Furthermore, if the clearance between the bearing surfaces at any point increases so that the pressure in the central recess **75** falls, the hydrostatic balance of the piston and slipper assembly is upset. The pressure on the piston **11** thereupon tends to reduce the clearance until equilibrium is restored. If the slipper **17** tends to tilt about a point on its leading edge, so that the adjacent arc of the slipper tends to make metal-to-metal contact with the slipper plate **16**, the moment of the piston thrust about this point opposes this tilting, so that the system is self-compensating to a considerable degree.

The degree of self-compensation is partly dependent on the cross-sectional area of each notch **77–80** which is presented to the escaping oil, and by careful design of these notches and their interconnecting annular groove **76** a dashpot action can be induced in which the oil pressure changes are slower than the changes in mechanical thrust, so that oscillation of the slipper **17** is minimized.

FIGS. 5 and 6 illustrate a modified oil film control arrangement. Here there are three symmetrically arranged recesses **75*a*, 75*b*, 75*c*** fed with oil through metering ducts **74*a*, 74*b*, 74*c***. The recesses are of sector shape, and of equal radius and angle, and are surrounded by a separate concentric groove **76** which communicates with atmosphere through three equiangularly spaced notches **77*a*, 78*a*, 79*a***. This arrangement of recesses and groove forms four lands, the central land **81*a*** resembling in shape a three-spoked wheel and the outer lands **82*a*, 83*a*, 84*a*** being part-annular in shape.

The operation of the modified slipper, however, is somewhat different from that of the slipper described with reference to FIG. 3, in that tilting is now opposed by differential hydraulic pressures across the face of the slipper.

For example, let it be assumed that the slipper **17** in FIGS. 5 and 6 is moving over the slipper plate **16** along the line of the section plane marked VI—VI in FIG. 5, the notch **79*a*** leading. There will be a tendency for the ends of the lands **83*a*** and **84*a*** on either side of the notch **79*a*** to dig into the surface of the slipper plate **16**, while points on the lands **81*a*, 82*a*** diametrically opposite the notch **79*a*** will tend to lift. As soon as any such displacement commences, the clearance between the land **81*a*** and the slipper plate **16** will increase and oil will escape into the annular groove **76** and out through the notches **77*a*, 78*a***. Its pressure at this zone will accordingly be reduced, while at the diametrically opposite side of the annular land **81*a*** the clearance will be reduced and the oil film pressure will be increased. This change in pressure produces a restoring couple so that the modified system is hydrostatically self-compensating.

In both forms of slipper, the surface areas of the lands, together with their shapes and positions on the slipper surface, are chosen so that the forces between the slipper **17** and the slipper plate **16** on the swash plate **12** during a delivery stroke of the associated piston **11**, which forces tend to promote metal-to-metal contact between the parts, are counterbalanced by forces in the oil films which separate them, these latter forces being directly derived from the instantaneous pressure of oil in the cylinder **10**.

A similar hydrostatic balance is not quite achieved by the hemispherical pivot **70** and coacting seating **58** as shown in FIG. 2, since the projected area of the latter on a plane normal to the piston axis is less than the area of the working face **56** of the piston. Consequently, if hydrostatic balance is to be achieved at this point also of the piston-slipper assembly, the arrangement of FIG. 4 is adopted. In this figure, the hemispherical male pivot **70*a*** is formed on the piston **11** and the coacting socket seating **58*a*** is formed on the slipper **17**, the projected area of the seating on a plane normal to the piston axis being made equal to the area of the working face **56** of the piston. By this arrangement the surfaces can be proportioned so that a similar balance between mechanical and hydrostatic forces is achieved as at the working face of the slipper **17**.

It is to be understood that, with appropriate design modifications, the pump described above may be used as a motor.

FIG. 7 shows the same principle of oil film control applied to the slipper of an eccentric type radial pump or motor.

The motor shaft carries a main eccentric **101** on whose periphery bear the slippers **102** of the piston rods **103**. Each piston rod has a part-spherical "little end" **104** which works in a complementary little end bearing assembly **105** in the piston **106**. The latter reciprocates in a cylinder **107** in a cylinder block **108**, and each cylinder has a detachable head **109** which defines, with the piston **106** a compression space **10**.

Each slipper **102** has the surface which bears on the main eccentric **101** lubricated by oil under pressure from the working space **110** above the piston **106**. For this purpose, each piston rod **103** is drilled axially at **111** from the crown of the part-spherical little end **104**. Opposite the point **112** at which the drilling **111** opens through the crown of the little end, the crown of the piston **106** is pierced by a coaxial bore **113**. The mouth **114** of this bore at the part-spherical bearing surface **105** is enlarged to ensure continuity of the oil flow path between the bore **13** and the drilling **111** at all working angles between them during each stroke of the piston **106**. Alternatively, if preferred, a flat may be machined on the crown of the little end **104** concentric with the drilling **111** for the same purpose.

The lower end of the drilling **111** is bifurcated, each branch drilling **115, 116** leading to a respective shouldered cavity **117, 118**. This cavity has a metering plug **119, 120** screwed thereinto, the plug being provided with an axial duct which includes a short metering section **121, 122** of reduced diameter. Each cavity **117, 118** opens into a respective relatively shallow recess **123, 124** on the working face of the shoe **102**. The recesses **123, 124** extend over a relatively large proportion of the projected area of the working surface of the shoe, and are aranged substantially symmetrically about both the longitudinal and transverse centre lines of this surface.

When the motor is running, each recess **123, 124** becomes filled with oil pumped into it from the working space **110** of the cylinder by way of the drilling **115** or **116** and metering duct **121** or **122**. Since the hydrostatic pressure of the oil in the recess at any instant is of the same order as the pressure in the working space **110** of the cylinder, the shoe **102** tends to float on an oil film trapped between it and the bearing surface of the main eccentric **101**. As the oil leaks away through the work-

ing clearance between the shoe and the eccentric, a pressure drop is established across the metering duct **121** or **122** tending to reduce the pressure of oil forming the lubricating film. The thickness of this film therefore tends to be reduced, thus reducing the leakage flow and re-establishing the pressure equilibrium across the metering duct **121** or **122**.

This automatic balance mechanism serves not only to maintain a constant thickness of oil film at the working surfaces but also to oppose the tendency for the shoe **102** to tilt. Thus, assuming that at a given point in the travel of a shoe on the main eccentric **101** in the direction of the arrow R, leading edge **125** of the shoe tends to rise, the clearance between the shoe and the eccentric is increased locally and the rate of leakage of oil from the adjacent recess **123** is increased. As explained above, this brings about a reduction in the hydrostatic pressure of oil in that recess and a consequent reduction in pressure of the oil film. At the same time, the trailing edge **126** of the shoe will tend to bite into the eccentric, the clearance between the bearing surfaces on the shoe and the eccentric being thereby reduced. The resistance to the leakage of oil between these surfaces from the recess **120** therefore increases, and the pressure of oil in the recess **120** consequently rises as the pressure drop across its metering duct **118** falls. Hence, a couple is exerted on the shoe **102** which tends to oppose the couple causing the initial tilting, and the system tends to be self-correcting as well as floating. Hence, the friction between the working parts is kept to a minimum.

Provided that the actual area of the bearing surface on the shoe **102** is kept relatively small, the total friction between the shoe and the eccentric **101** can be kept to a low value. This is partly due to the fact that the area exposed to friction is less, and partly to the fact that the resistance of the oil leakage path between the shoe and the eccentric is reduced, so that a greater volume of oil leaks away from the recesses **123**, **124**—subject only to the control exercised by the metering duct **121** or **122**. The higher rate of leakage leads to the establishment of a thicker film of oil which requires less energy to shear than a thin film, and the loss of thrust at the piston due to this leakage is offset by the reduced absorption of power at the friction surface, especially at "creep" speeds.

The hydrostatically balance floating feature of the shoe thus leads to increased motor efficiency.

The machine illustrated in FIG. 7 is equally capable of acting as a pump when the eccentric **101** is externally driven.

It will be obvious to those skilled in the art that various changes may be made without departing from the scope of the invention and therefore the invention is not limited to what is shown in the drawings and described in the specification but only as indicated in the appended claim.

What is claimed is:

In a positive displacement hydraulic machine, in combination; a radial cylinder; a piston reciprocable in said cylinder; a slipper articulated to said piston; a piston thrust reaction member on which said slipper is slidable; said piston thrust reaction member comprising an eccentric; at least two separate oil retaining recesses symmetrically disposed with respect to the midpoint of the sliding face of said slipper; each of said recesses extending from a point adjacent the center of said sliding face towards the periphery thereof and bounded on all sides by a continuous land; separate ducts communicating between the pressure side of said piston and said recesses; and a metering constriction in each duct for controlling the supply of working fluid to said recess whereby said slipper is restored to its normal operating position by at least one of said recesses when said slipper is moved from said reaction member.

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LAURENCE V. EFNER, *Primary Examiner.*