

April 9, 1963

K. HENRICHSEN
HYDRAULIC PUMP OR MOTOR

3,084,633

Filed Sept. 9, 1957

6 Sheets-Sheet 1

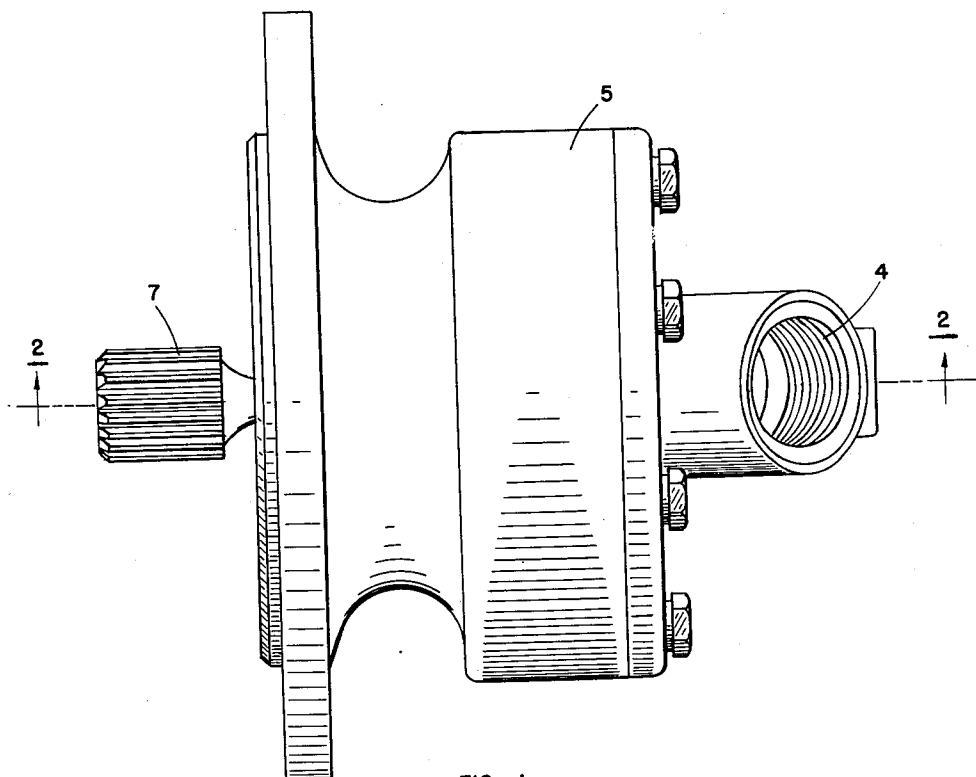


FIG. 1

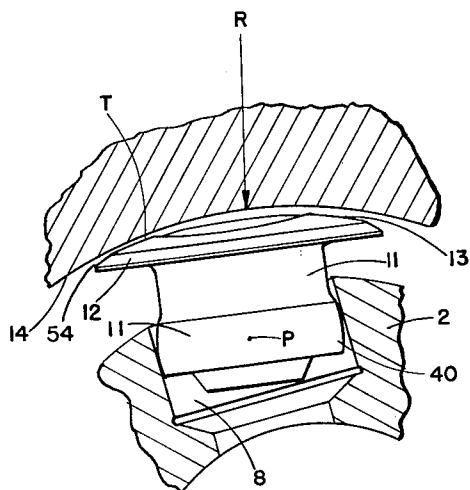


FIG. 13

INVENTOR.
KNUT HENRICHSEN

BY

Richard F. Carr

ATTORNEY

April 9, 1963

K. HENRICHSEN
HYDRAULIC PUMP OR MOTOR

3,084,633

Filed Sept. 9, 1957

6 Sheets-Sheet 2

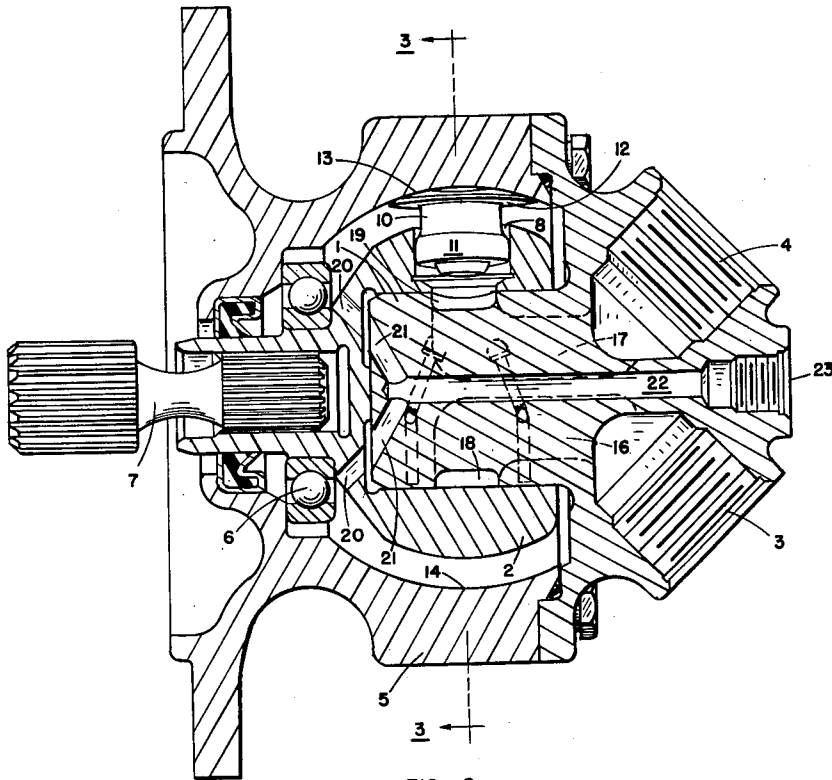


FIG. 2

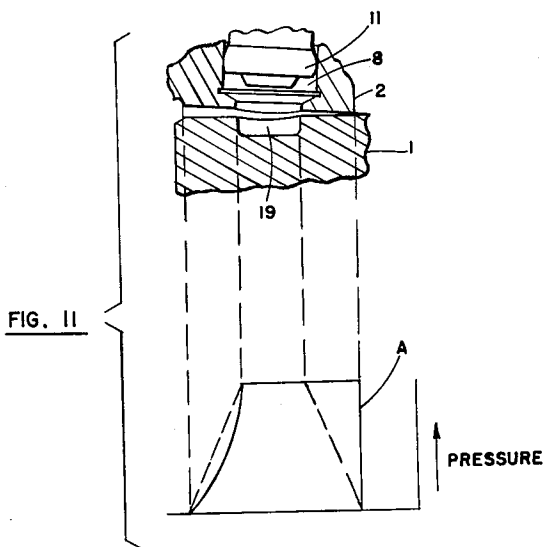


FIG. 11

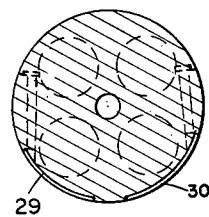


FIG. 10

INVENTOR.
KNUT HENRICHSEN

BY *Richard F. Carr*
ATTORNEY

April 9, 1963

K. HENRICHSEN
HYDRAULIC PUMP OR MOTOR

3,084,633

Filed Sept. 9, 1957

6 Sheets-Sheet 3

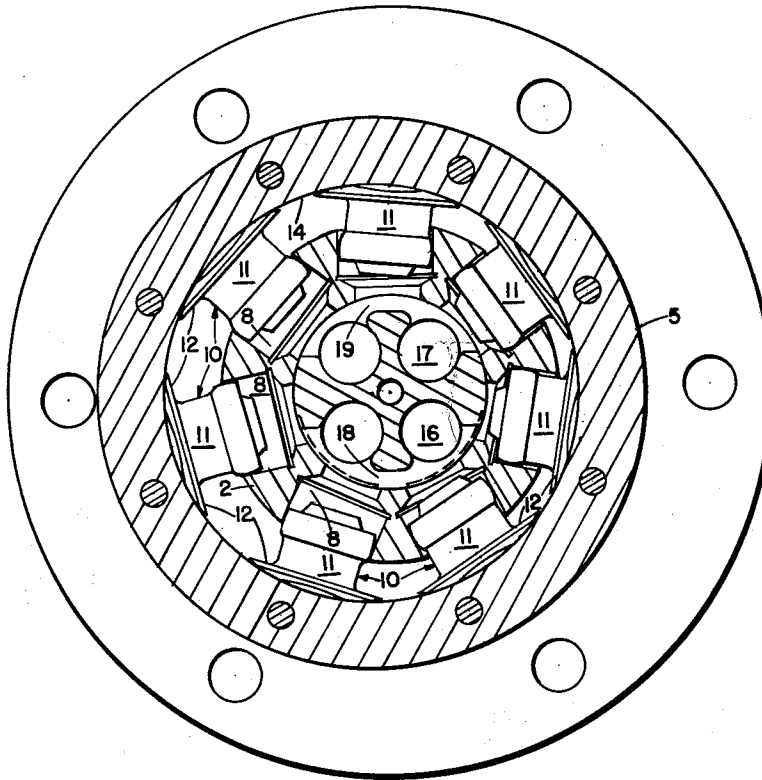


FIG. 3

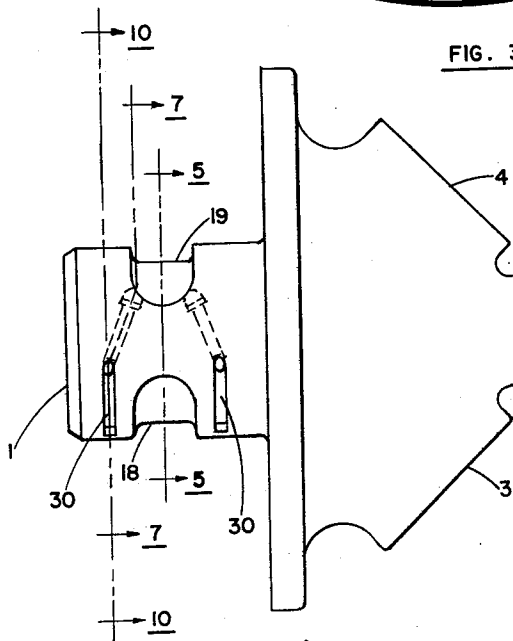


FIG. 4

INVENTOR.
KNUT HENRICHSEN

BY *Richard F. Carr*

ATTORNEY

April 9, 1963

K. HENRICHSEN
HYDRAULIC PUMP OR MOTOR

3,084,633

Filed Sept. 9, 1957

6 Sheets-Sheet 4

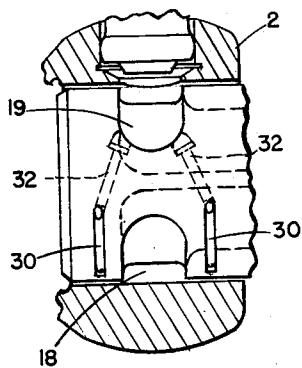


FIG. 8

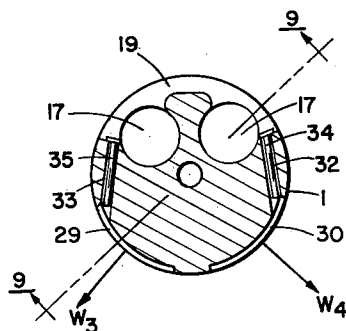


FIG. 7

FIG. 6

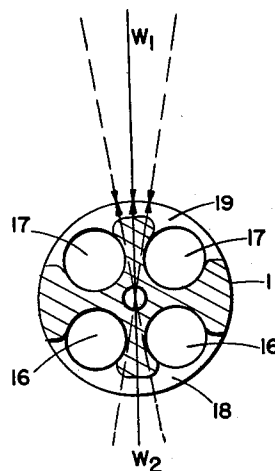
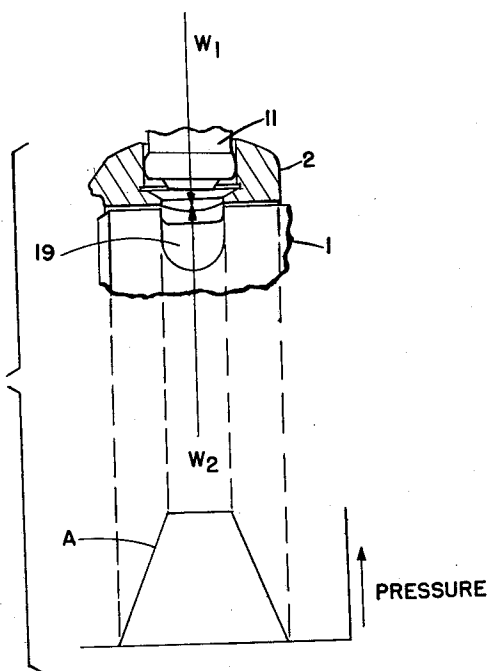


FIG. 5

INVENTOR.
KNUT HENRICHSEN

BY *Richard F. Carr*

ATTORNEY

April 9, 1963

K. HENRICHSEN
HYDRAULIC PUMP OR MOTOR

3,084,633

Filed Sept. 9, 1957

6 Sheets-Sheet 5

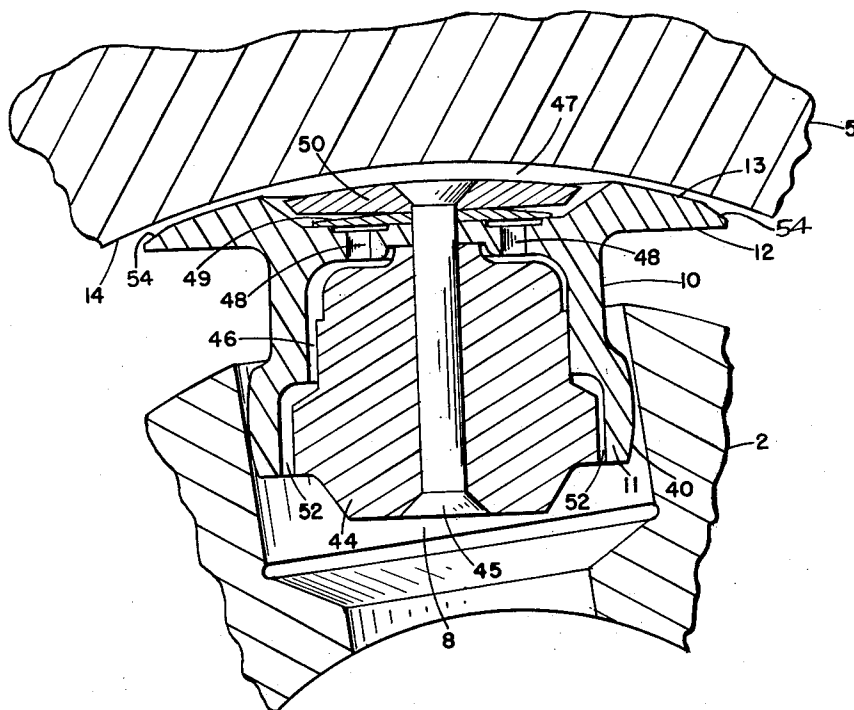


FIG. 12

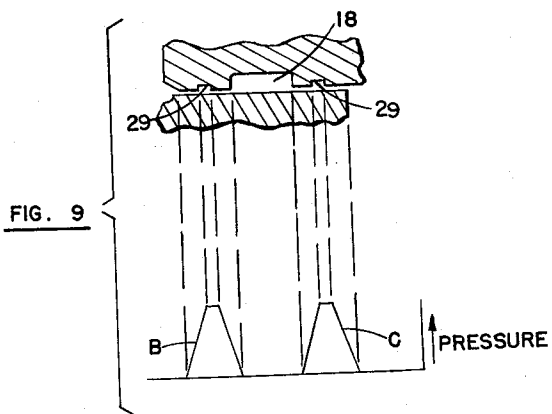


FIG. 9

INVENTOR.
KNUT HENRICHSEN

BY *Richard F. Carr*

ATTORNEY

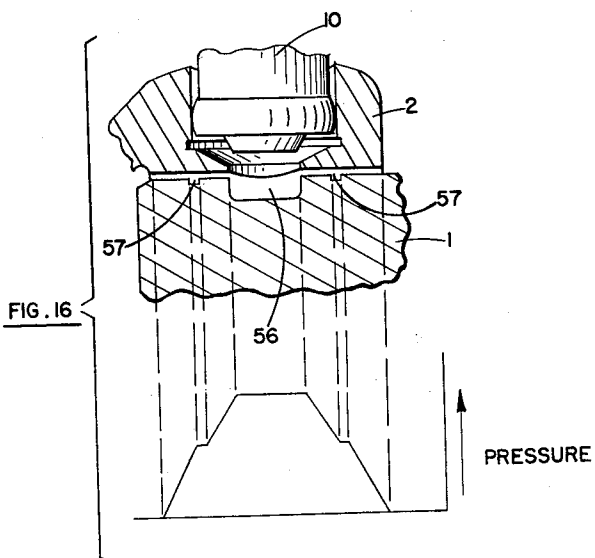
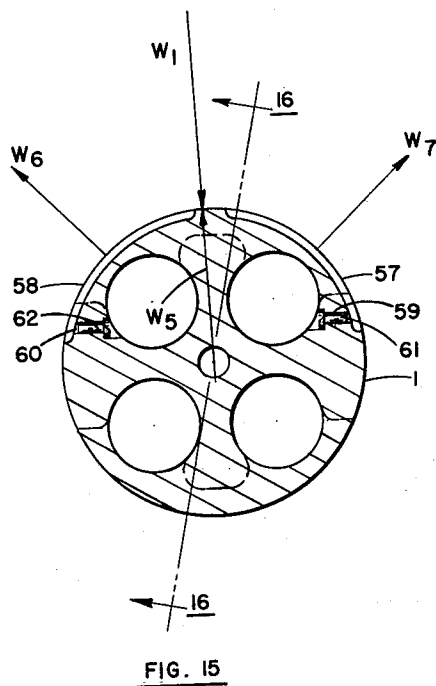
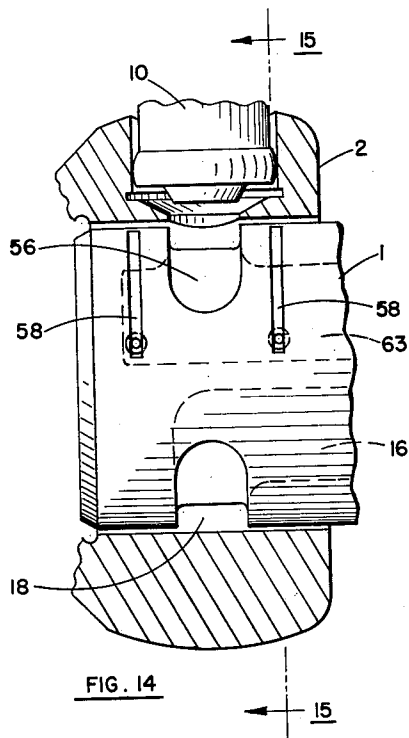
April 9, 1963

K. HENRICHSEN
HYDRAULIC PUMP OR MOTOR

3,084,633

Filed Sept. 9, 1957

6 Sheets-Sheet 6



INVENTOR.
KNUT HENRICHSEN
BY *Richard F. Carr*
ATTORNEY

1

3,084,633

HYDRAULIC PUMP OR MOTOR

Knut Henriksen, Los Angeles, Calif., assignor to

North American Aviation, Inc.

Filed Sept. 9, 1957, Ser. No. 682,981

8 Claims. (Cl. 103-161)

This application is a continuation in part of my copending application Serial No. 651,240, filed April 8, 1957, and now abandoned.

This invention pertains to a hydraulic pump or motor operable at high speeds and fluid pressures.

The device of this invention is a pump or motor of the pintle valve type of considerably improved efficiency, providing a compact lightweight unit of large capacity. The invention includes a means for balancing pressures around the outer surface of the pintle valve to assure that a fluid film is provided between the valve and the rotatable cylinder block. This pressure balance arrangement may include passageways metering fluid from the high pressure port to the surface of the valve adjacent the low pressure port. The piston units are of thin walled, hollow construction having spherical exterior portions in the cylinder. An axial flow passage to a step bearing at the slipper is provided, supplying lubrication for slower pump speeds. A check valve preferably is included preventing reverse flow and allowing the slipper to act also as a fluid wedge bearing at higher speeds. Additionally a temperature compensating means inside the piston in the form of an aluminum plug restricts the axial passage as the temperature of the unit increases.

Therefore, it is an object of this invention to provide a pump or motor of compact, lightweight design of high capacity. A further object of this invention is to provide a pump or motor having provisions for minimizing friction and wear. Another object of this invention is to provide a pump or motor having means to control lubricating fluid flow in accordance with temperature. A still further object of this invention is to provide a pump or motor having means to balance pressures around the pintle valve to preclude metal-to-metal contact. Yet another object of this invention is to provide a pump or motor incorporating a thin walled piston capable of providing large area contact with the cylinder wall for minimizing wear. An additional object of this invention is to provide a slipper operable both as a step bearing and as a fluid wedge bearing. These and other objects will become more apparent when taken in connection with the following detailed description and the accompanying drawings in which

FIG. 1 is a top plan view of the exterior of the device of this invention;

FIG. 2 is a sectional view of the invention taken along line 2-2 of FIG. 1;

FIG. 3 is a sectional view taken along line 3-3 of FIG. 2, illustrating the relationship of the bearing race, piston-slipper assemblies and pintle valve;

FIG. 4 is a side elevational view of the pintle valve, with the cylinder block shown removed for purposes of clarity;

FIG. 5 is a sectional view taken along line 5-5 of FIG. 4, illustrating the forces acting on the pintle valve;

FIG. 6 is a fragmentary view, partially in section, showing how the fluid pressures in the high pressure port provide a force on the pintle;

FIG. 7 is a sectional view taken along line 7-7 of FIG. 4, illustrating the balancing groove and metering pin arrangement;

FIG. 8 is a fragmentary elevational view, partially in section, further illustrating the balancing groove arrangement;

2

FIG. 9 is a sectional view taken along line 9-9 of FIG. 7, illustrating the fluid pressures resulting from the balancing grooves;

FIG. 10 is a sectional view of the pintle valve taken along line 10-10 of FIG. 4;

FIG. 11 is a fragmentary sectional view illustrating the fluid forces resulting from angular misalignment of the cylinder block and pintle valve;

FIG. 12 is an enlarged fragmentary sectional view, taken generally as in FIG. 3, illustrating in detail the construction of a piston-slipper assembly;

FIG. 13 is a fragmentary view illustrating the operation of the dynamic fluid wedge bearing formed by the slipper face on the intake stroke and during high speed operation;

FIG. 14 is a fragmentary elevational view partially in section similar to FIG. 8, showing the balancing groove arrangement for an underbalanced valve;

FIG. 15 is a sectional view taken along line 15-15 of FIG. 14 further illustrating the balancing groove and metering pin design; and

FIG. 16 is a fragmentary sectional view taken along line 16-16 of FIG. 15 illustrating the fluid pressure distribution.

Referring in particular to FIGS. 1, 2 and 3 of the drawing the device of this invention includes a pintle valve 1 about which cylinder block 2 rotates. The pintle valve includes openings 3 and 4 which serve as the inlet and outlet, respectively, when the unit acts as a pump. The pintle valve in the embodiment illustrated also serves as one end of the housing, connecting to main housing section 5 which surrounds the cylinder block. The pintle valve acts as the main bearing for the pump, while bearing 6 also serves to support the cylinder block axially where shaft 7 mates with splines in the block and acts as a power input when the device is used as a pump, or as a power takeoff when employed as a motor. The cylinder block is provided with a plurality of radial cylinders 8 in which piston-slipper assemblies 10 reciprocate. Each of these assemblies includes a piston portion 11 for engagement with a cylinder, while slipper portion 12 projects beyond the block and includes a spherical outer face 13 which engages complementary spherical bearing race 14 on the inner surface of the main housing. The unit illustrated is of fixed displacement type with bearing race 14 disposed eccentrically with respect to the pintle valve to effect reciprocation of the piston-slipper assemblies as the cylinder block rotates.

Pump inlet 3 and outlet 4 connect with passages 16 and 17 which in turn communicate with diametrically opposed ports 18 and 19. The latter extend circumferentially around portions of the lower side and of the upper side of the pintle valve, respectively. Ports 18 and 19 are dimensioned to correspond to the diameter of the cylinder ports for cooperation therewith in pumping the fluid. With the embodiment shown operating as a pump the cylinder block rotates clockwise in the showing of FIG. 3, thereby drawing fluid from inlet 3 into port 18 and thence into the cylinders on the lower half of the pintle valve. The pistons on the upper portion of the pintle valve move inwardly and force the fluid into port 19, through passage 17 and outlet 4. Leakage fluid in the pump case passes through ports 20 and 21, to central passage 22, the outlet 23 of which may be connected to the reservoir of the hydraulic system. The axial location of this return passage assures that any air in the case will be immediately exhausted as the pump rotates.

An important consideration in providing an efficient pump capable of large capacities, high rotation speeds and long life is to assure that a fluid film is maintained between the outer surface of the pintle valve and the inner surface of the cylinder bore. In the design illustrated there is only .0002 inch clearance between the valve and the cyl-

inder block, yet a fluid film must be maintained at all times between these two elements. The pump is subject to certain loads and pressures which complicate the problem of providing such a fluid film. As shown diagrammatically in FIG. 5, the cylinders on the top half of the pintle valve provide a downward load having a resultant W_1 passing through the pintle center. This load moves back and forth angularly between the two dotted line positions shown in FIG. 5 as the cylinder block rotates. This occurs because either three or four cylinders will be in communication with port 19, depending upon the rotational position of the cylinder block. The midpoint is offset slightly from the vertical because of the pump eccentricity.

An upward force is exerted on the top half of the cylinder block opposing W_1 . This results from the fluid pressure within port 19 as illustrated in FIG. 6 (in which clearances have been exaggerated) where curve A depicts the pressure distribution across the top of the pintle. Thus, full pump pressure is exerted in port 19 while the pressure drops off as a straight line across the area between the pintle and the cylinder block to the pump case pressure which will be substantially zero. The resultant W_2 of this pressure always opposes W_1 and moves back and forth in the same manner. In the design shown the high pressure port 19 is sufficiently large so that W_2 is about one-fourth greater than W_1 , thereby causing a pintle overbalance urging the lower surfaces of the cylinder block and pintle into contact.

Additional forces are present tending to cause metal-to-metal contact between the pintle and the cylinder block. The eccentricity of the slipper race with respect to the pintle causes a force tending to move the cylinder block to the left from the position in FIG. 3, which would cause metal-to-metal contact on the right-hand portion of the pintle. A force urging the cylinder block to the right results from acceleration of the pistons within the cylinders, but this may be either greater or less than the leftward force depending upon the pump design and rotational speeds. Additionally, if the pump is installed in a rapidly moving vehicle such as an aircraft, gyroscopic forces resulting from high speed maneuvering will tend to misalign the cylinder block and pintle.

The features best seen in FIGS. 7, 8 and 9 overcome these unbalancing forces and assure that a fluid film is maintained. To this end, a pair of balance grooves 29 is provided in the surface of the pintle on one side of the midpoint of port 18 (the vertical line of the pump) arranged to straddle port 18. A similar pair 30 is located on the other side of the vertical of the pintle. Passageways 32 and 33 interconnect high pressure port 19 with the balance grooves. These ports are provided with suitable restrictions such as metering pins 34 and 35 which hold the flow to a predetermined value, depending upon the clearance in the passageways. These metering passageways and the grooves permit fluid from port 19 to provide an additional force at the bottom of the pintle as illustrated in curves B and C of FIG. 9. The groove pressure drops off substantially linearly to case pressure at the edge of the port, and at the edge of the adjacent surfaces of the cylinder block and pintle. The pressures within these grooves provide resultants W_3 and W_4 . With proper proportioning of the metering passageways, these resultants can be made to counteract the overbalance caused by the excess of W_2 over W_1 and thus maintain the cylinder block vertically concentric with the pintle.

Additionally, forces W_3 and W_4 counteract other unbalancing forces on the pintle. For example, if the cylinder block tends to move to the left from the position of FIG. 3, which may result from pump eccentricity, this tends to close up grooves 29, while opening up grooves 30. This means that the flow out of grooves 29 will decrease and the pressure therein will rise, while the pressure in grooves 30 decreases. This will move the cylinder block back toward the central position. A similar balancing

action takes place if the cylinder block tends to move to the right.

Forces tending to misalign the cylinder block and pintle are counteracted in the manner illustrated diagrammatically in FIG. 11, as well as by the balancing grooves in the bottom of the pintle. If the cylinder block tilts to a position with one edge in contact with the pintle as illustrated in FIG. 11, the pressure curve A will be altered to the shape illustrated from its original symmetrical form shown in dotted lines. By reason of the tilted position of the cylinder block, substantially full pressure will be maintained out to the contacting edge causing the right-hand portion of the curve to increase in area. The raised portion of the block at the left causes a tapered clearance at the left which causes the pressure to drop off more rapidly and reduces the area under that portion of the curve. Resultant W_2 therefore moves to the right and provides a moment tending to move the cylinder block back toward its properly aligned position. Additionally, the tilting of the cylinder block will tend to close up grooves 29 and 30 on the left-hand side of port 18 while opening up the grooves on the right-hand side. This increases the pressure in the grooves on the left of the port 18 while decreasing the pressure in the grooves on the right-hand side of the port, thereby also providing a righting moment for returning the cylinder block to a position of alignment.

It is apparent, therefore, that the balancing grooves are located so that they provide fluid forces in directions to overcome any of the unbalancing forces which may be encountered.

By the design described above, it is possible to obtain forces which will maintain a fluid film between the valve and cylinder block for all rotational speeds and conditions so that virtually no wear results. In tests pencil marks made on the surface of the pintle valve have remained intact after millions of rotations of the cylinder block.

This is possible because the high pressure port 19 is proportioned to give an overbalanced condition for the pintle, which is counteracted by the pressure in the balancing grooves located remote from the high pressure port. In this manner, not only may the pintle overbalance be corrected, but the additional unbalancing forces such as may arise from pump eccentricity, acceleration of the pistons and gyroscopic conditions, also may be offset. If an unbalanced condition were not created deliberately between the high pressure port and the cylinder block, the latter unbalancing forces could not be overcome. In other words, if the high pressure port were dimensioned so that the resultant of its fluid pressure exactly equaled the downward load from the cylinders (i.e., if W_1 equaled W_2) balancing grooves could not be used because they would upset the equilibrium between these two loads. As a result, the additional unbalancing forces from eccentricity, piston acceleration and gyroscopic effect would meet no opposition and metal-to-metal contact would be inevitable.

The principles of this invention may be applied with equal facility to a design where the pintle valve is underbalanced as illustrated in FIGS. 14, 15 and 16. Here the size of the high pressure port 56 is less than the size of port 19 of the previously described embodiment. It is reduced sufficiently in width so that resultant force W_5 from the fluid pressure in port 56 is smaller than resultant force W_1 of the cylinder load on the pintle. Preferably also the width of the cylinder block at the valve is less than in the previous embodiment so that there is a more rapid drop off in pressure and consequently a reduced resultant force. The excess of W_1 over W_5 in this design urges the top of the cylinder block inner surface against the top portion of the valve at the area around the high pressure port.

The unbalanced condition is corrected again by means of balancing grooves connecting with the high pressure port. A pair of grooves 57 is provided on the surface of

the pintle on one side of the midpoint of port 56, arranged to straddle that port. A similar pair 58 is provided on the other side of the midpoint of port 56. Passageways 59 and 60 interconnect port 56 and the balance grooves, while metering pins 61 and 62 restrict the flow there-through to a predetermined value. This connection is made through outlet passage 63 which, of course, has the same pressure as port 56. The connection from port 56 to the balancing grooves 57 and 58 allows an additional force to be exerted between the upper portion of the pintle valve and the cylinder block. This is illustrated graphically in FIG. 16 where it may be seen that the grooves result in a greater area beneath the pressure curve. By proper proportioning of the metering pins 61 and 62, the additional force so provided, plus the resultant W_6 , may be caused to balance W_1 and prevent metal-to-metal contact between the block and the pintle.

In a manner similar to that for the previously described embodiment, the resultant forces W_6 and W_7 from the balance grooves 57 and 58 extend angularly with respect to the midpoint of the high pressure port and return the pintle to a concentric position with respect to the cylinder block whenever an unbalancing force tends to move it to one side or to cause relative tilting. If the clearance at the grooves is changed the pressure therein will also vary as with the previously described embodiment to counteract the unbalancing force.

The basic concept, therefore, in providing a fluid film between the cylinder block and the valve block is first to create an unbalance between the load imposed by the cylinders on the valve and the opposing load from the fluid in the high pressure port. Then fluid is metered from the high pressure port to the valve surface at a location where additional fluid pressure can be applied to overcome the unbalance between those two loads. This location is selected also so that the additional fluid pressure will counteract any other unbalancing forces present. Without this additional fluid pressure the latter unbalancing forces cannot be offset and metal-to-metal contact will result. Positioning the additional pressure outlets at the valve surface adjacent the inner perimeter of the cylinder block means that when the clearance at the pressure outlet decreases the force exerted will become greater. When the clearance becomes larger, the force is less. Thus the magnitude of the force from the additional pressure outlets adjusts itself to the requirements at hand.

The piston-slipper assemblies of this improved pump or motor are likewise designed to minimize wear and improve the efficiency of the unit. It is preferred to make piston portion 11 and slipper portion 12 of the assembly a unitary hollow shell as seen in FIG. 12. Portion 11, which reciprocates within the cylinder, includes a spherical exterior segment 40 which allows the assembly to float freely within the cylinder as the cylinder block rotates. The spherical contours of face 13 and bearing race 14 assure that no misalignment will occur. The shell, like the cylinder block, is made of steel so that expansion of the parts due to heat will not open up excessive clearances.

Within the hollow shell is an aluminum plug 44 held in place by aluminum pin 45. The plug is eccentrically mounted with respect to the interior of the shell, substantially filling the space within the shell and providing a restricted passageway 46 through the piston-slipper assembly. This passageway empties into basin 47 in the center of slipper face 13, from which the fluid from the cylinder flows outwardly across the dam, or step bearing, formed by the annular slipper face. This fluid precludes metal-to-metal contact between the slipper and bearing race, and by proper proportioning of passageway 46 may be held to a predetermined rate of flow. Because of its higher coefficient of thermal expansion, the aluminum plug expands more rapidly under heat than the steel piston, thereby decreasing the size of passageway 46

as the temperature of the unit rises. This compensates for the increased flow which would otherwise result from the decreased viscosity of the oil being pumped at the higher temperature.

Outlet openings 48 of passageway 46 are covered by a relatively thin, flexible steel disk 49 held by pin 45 and aluminum washer 50. The disk 49 acts as a check valve in passageway 46, yielding to pressure within the passageway to allow flow from the cylinder, but preventing reverse flow from basin 47. At low and medium r.p.m., on the pressure stroke of the piston, cylinder pressure will exceed the pressure in basin 47, opening the check valve and causing lubricating fluid to flow.

On the intake stroke of the piston, and at relatively high speeds of rotation, the pressure within basin 47 will exceed the cylinder pressure so that no flow will take place through passageway 46 from the cylinder to the basin. The check valve prevents any reverse flow under these conditions. When this occurs the slipper of this invention provides a dynamic fluid wedge bearing which prevents metal-to-metal contact with the slipper race. The manner in which this occurs may be seen by reference to the diagrammatic showing of FIG. 13. As the pump rotates and the parts become heated the slipper bearing face will tend to increase in its convexity. This results from the flow of heat from the portions adjacent the slipper race at higher temperature to the relatively cooler portions of the slipper flange opposite from the bearing race. Similarly, the slipper race decreases its concavity due to heat flow from the relatively hot slipper race to the cooler regions remote therefrom. This lifts the periphery of slipper face 13 from the bearing race and permits the wedging of fluid film beneath the leading edge of the slipper as the pump rotates. This effect is enhanced by elastic deflection of the slipper resulting from fluid pressure between the slipper and the race. Additionally, there is a slight bevel 54 provided at the periphery of the slipper surface which also tends to permit lubricating fluid to enter the area beneath the slipper at the bearing face. Frictional forces on the piston in the cylinder also tend to cock the piston so as to elevate the leading edge of the slipper at all rotational speeds.

As the slipper moves relative to the slipper race, pressure builds up beneath the slipper from its leading edge to point T where the slipper is nearest the slipper race, this being the point of tangency between the bearing surface of the slipper and a surface parallel to the slipper race. In back of point T the pressure decreases sharply and theoretically reaches a negative value. The pressure beneath the leading edge of the slipper causes the slipper to cock slightly relative to the slipper race, elevating the leading edge. This causes the slipper to pivot until point T, where it is nearest the slipper race, has shifted well behind point P which is the center of spherical portion 40 of the piston. This pivoting of the piston-slipper assembly about point P progresses until resultant R of the pressure beneath the leading edge of the slipper passes through point P, thereby precluding further pivoting of the slipper and causing stable operation of the piston-slipper assembly. This fluid film is maintained at all times during relatively high speed rotation of the pump and on the intake stroke. If the speed drops the cylinder pressure will again force fluid through the piston-slipper assembly to provide lubrication as a step bearing. Thus the check valve permits a combination of step bearing and dynamic lubrication avoiding metal-to-metal contact for all rotational speeds.

The piston-slipper assembly of this invention includes further wear reducing features. Passageway 46 includes an annular portion 52 within spherical segment 40 of the thin walled piston shell. This permits full cylinder pressure to be exerted on the interior of the piston tending to expand portion 40. This helps avoid localized contact and spreads wear over a greater area.

The thin walled piston construction gives an additional

advantage in minimizing wear. The side load on the piston flattens it against the cylinder wall in a relatively wide area, thereby increasing the area of contact. In FIG. 12, for example, where the block rotates clockwise, the piston is forced to the left against the cylinder wall. The distortion of the piston from the side load is much the same as for a pneumatic tire where it contacts the ground.

Localized wear is further avoided by the symmetrical design of the piston-slipper assembly. The slipper bearing is stable for its operating conditions in all aspects except yaw. This permits the slipper to rotate about its axis so that any wear will occur evenly on the slipper face and piston wall.

It may be seen by the foregoing, therefore, that I have provided an improved pump incorporating features which materially reduce the wear of the unit, and greatly increase its efficiency and capacity. Unique features of pressure balancing ports and slipper and piston design, permit these results to be obtained.

The foregoing detailed description is to be clearly understood as given by way of illustration and example, the spirit and scope of this invention being limited only by the appended claims.

I claim:

1. A hydraulic device comprising a valve member; a cylinder block in engagement with and rotatable relative to said valve member, said cylinder block having a plurality of cylinders therein, said valve member having inlet and outlet ports cooperating with said cylinders; a piston-slipper assembly reciprocative in each of said cylinders; a bearing race engaged by the slipper portions of said piston-slipper assemblies, said bearing race being positioned to cause reciprocation of said piston-slipper assemblies upon such rotation of said cylinder block, each of said piston-slipper assemblies having a metering passageway therethrough extending from the cylinder with which the same is associated to the slipper thereof at the location of said bearing race for providing lubricating fluid for said slipper; and check valve means in each of said passageways for permitting flow only from said cylinders to said bearing race while preventing reverse flow therethrough, said piston comprising a relatively thin-walled hollow member and a plug member within said thin-walled member dimensioned to substantially fill the space therein while providing a predetermined clearance therebetween for providing said passageway and controlling the fluid flow therethrough, said plug being of a material having a greater coefficient of thermal expansion than that of the material of said piston for thereby decreasing said clearance at elevated temperatures.

2. A hydraulic device comprising a valve member; a cylinder block in engagement with and rotatable relative to said valve member, said cylinder block having a plurality of cylinders therein, said valve member having inlet and outlet ports cooperating with said cylinders; a piston-slipper assembly reciprocative in each of said cylinders; a bearing race engaged by the slipper portions of said piston-slipper assemblies, said bearing race being positioned to cause reciprocation of said piston-slipper assemblies upon such rotation of said cylinder block, each of said piston-slipper assemblies having a metering passageway therethrough extending from the cylinder with which the same is associated to the slipper thereof at the location of said bearing race for providing lubricating fluid for said slipper; and check valve means in each of said passageways for permitting flow only from said cylinders to said bearing race while preventing reverse flow therethrough, said piston portion of each of said piston-slipper assemblies comprising a relatively thin steel shell and an aluminum plug filling a major portion of the volume of said shell while providing a predetermined clearance between said shell and said aluminum plug thereby defining the passageway for the passage of lubricating fluid from said cylinder to said bearing face.

3. A device as recited in claim 2, in which the clearance around said aluminum plug includes an annular portion for permitting fluid in said passageway to exert an expansive force on said steel shell.

4. A hydraulic device comprising a valve member; a cylinder block in engagement with and rotatable relative to said valve member, said cylinder block having a plurality of cylinders therein, said valve member having inlet and outlet ports sequentially communicating with said cylinders; a unitary piston-slipper assembly reciprocative in each of said cylinders; a slipper race engaged by the slipper portions of said piston-slipper assemblies, said slipper race being positioned to cause reciprocation of said piston-slipper assemblies upon rotation of said cylinder block, each of said piston-slipper assemblies including a spherical segmental hollow piston portion reciprocative in said cylinders and a spherical sector slipper face for sliding cooperation with the slipper race, said hollow spherical piston portion being of a unitary thin-wall construction elastically deformable under operational side loads to increase the area of contact of its spherical surface with the cylinder wall and thereby minimize localized wear, said hollow piston communicating with said cylinder and the pressurized fluid therein to thereby pressurize the interior of said thin-walled hollow piston and cause the same to deflect and increase the area of contact of the piston spherical exterior surface with the cylinder wall to thereby decrease localized piston wear.

5. In combination in a hydraulic device; a casing with inlet and outlet ports; a cylinder block having cylinders therein rotatably mounted in said casing; a valve means providing communication between said cylinders and the inlet and outlet ports; a piston-slipper assembly for cooperation with each of said cylinders having a piston portion; a slipper race engaged by said slipper portion, said slipper race being positioned relative to said cylinder block to effect reciprocation of said piston slippers upon rotation of the cylinder block, each of said piston-slipper assemblies including a spherical segmental hollow piston portion reciprocative in said cylinders and a spherical sector slipper face for sliding cooperation with the slipper race, each of said piston slipper assemblies being angularly displaceable from the cylinder axis to tilt said slipper face relative to the slipper race and thereby permit the formation of a lubricating dynamic fluid wedge between said slipper face and said slipper race, said piston-slipper assembly having a passageway therethrough extending from the cylinder with which the piston is reciprocally associated to the slipper face for providing pressurized lubricating fluid to said slipper face; temperature sensitive means for restricting said metering passageway with increasing temperature; and check valve means in said passageway for permitting flow only from said cylinder to the slipper face while preventing reverse flow therethrough.

6. A unitary piston-slipper adapted for use in a radial piston hydraulic device having a cylinder and an eccentric curved slipper race wherein reciprocation of the pistons is affected by rotation of the slippers on the eccentric slipper race comprising a thin-walled hollow piston having a spherical exterior surface adapted to contact the cylinder wall of such a hydraulic device, and having a hollow interior adapted to communicate with the interior of the cylinder and the pressurized fluid therein whereby the spherical surface of said piston may be slightly deformed by the interior pressure to increase the area of contact thereof when installed in such cylinder with the cylinder wall; and an integral spherical sector slipper face remote from said piston adapted for sliding cooperation with the curved slipper race, said hollow piston being of a unitary thin-wall construction elastically deformable under operational side loads to increase the area of contact of

its spherical surface with the cylinder wall and thereby to minimize localized wear.

7. A unitary piston-slipper adapted for use in a radial piston hydraulic device having a cylinder and a slipper race engageable by the slipper portions of said piston wherein reciprocation of the pistons is effected by rotation of the slipper portions on the eccentric slipper race comprising a first portion forming a spherical segmental piston adapted for reciprocation in the cylinder and an integral second portion forming a slipper face adapted for sliding cooperation with the associated slipper race, said piston-slipper having a passageway therethrough adapted to extend from the cylinder with which the piston is adapted to be reciprocally associated to the associated slipper face for providing pressurized lubricating fluid to such slipper face; temperature sensitive means for restricting said metering passageway with increasing temperature; and check valve means in said passageway adapted for permitting flow only from the cylinder to the slipper face while preventing reverse flow through said passageway.

8. A rotary hydraulic device comprising a valve member; a cylinder block in engagement with and rotatable relative to said valve member, said cylinder block having a plurality of radial cylinders therein, said valve member having inlet and outlet ports cooperating with said cylinders; an inflexible unitary piston-slipper assembly for cooperation with each of said cylinders having a piston portion including an arcuate segmental surface and an integral slipper portion including an arcuate segmental slipper face; an annular slipper race engaged by said arcuate segmental slipper face and cooperable therewith to form a hydrodynamic fluid wedge bearing therebetween, said annular slipper race being positioned to cause reciprocation of said piston-slipper assemblies upon rotation of said cylinder block, said piston portion arcuate segmental surface contacting the cylinder wall and arcuate segmental slipper face being slidably cooperable to form a fluid wedge bearing with said annular slipper race whereby said piston-slipper assembly is freely movable to a position wherein the slipper portion is angularly displaced from the normal to the cylinder axis to permit the formation of a lubricating fluid wedge between said

slipper portion and the slipper race, each of said piston-slipper assemblies having a metering passageway therethrough extending from the cylinder with which the same is associated to the slipper face for providing lubricating fluid between said slipper face and the slipper race; and check valve means in each of said passageways for permitting flow only from said cylinders to said slipper face while preventing reverse flow therethrough whereby said slipper may operate as a hydrostatic step bearing when high pressure fluid is available in said cylinder and as a hydrodynamic fluid wedge bearing when such high pressure fluid is not available for pressurized lubrication.

References Cited in the file of this patent

UNITED STATES PATENTS

1,584,897	Skinner et al. -----	May 18, 1926
1,710,567	Carey -----	Apr. 23, 1929
1,775,892	De Salardi -----	Sept. 16, 1930
1,813,122	Moore -----	July 7, 1931
1,878,862	Landenberger -----	Sept. 20, 1932
1,890,953	Smith -----	Dec. 13, 1932
1,920,725	Wallgren -----	Aug. 1, 1933
1,931,969	Thoma -----	Oct. 24, 1933
1,972,907	Shaw -----	Sept. 11, 1934
2,055,602	Dodge -----	Sept. 29, 1936
2,101,732	Benedek -----	Dec. 7, 1937
2,139,387	Schweiss -----	Dec. 6, 1938
2,147,515	Benedek -----	Feb. 14, 1939
2,205,913	Stacy -----	June 25, 1940
2,380,907	Hall -----	July 31, 1945
2,427,224	Morton -----	Sept. 9, 1947
2,528,739	Carey -----	Nov. 7, 1950
2,674,956	Hilton -----	Apr. 13, 1954
2,675,763	Muller -----	Apr. 20, 1954
2,679,210	Muller -----	May 25, 1954
2,710,137	Arnouil -----	June 7, 1955
2,721,519	Henrichsen -----	Oct. 25, 1955
2,752,214	Ferris -----	June 26, 1956
2,820,473	Reiners -----	Jan. 21, 1958
2,956,845	Wahlmark -----	Oct. 18, 1960

FOREIGN PATENTS

246,097	Germany -----	July 1, 1909
---------	---------------	--------------