Nov. 28, 1967 J. F. BEDFORD 3,354,786

HYDRAULIC MOTORS

Original Filed June 25, 1964 9 Sheets-Sheet 1

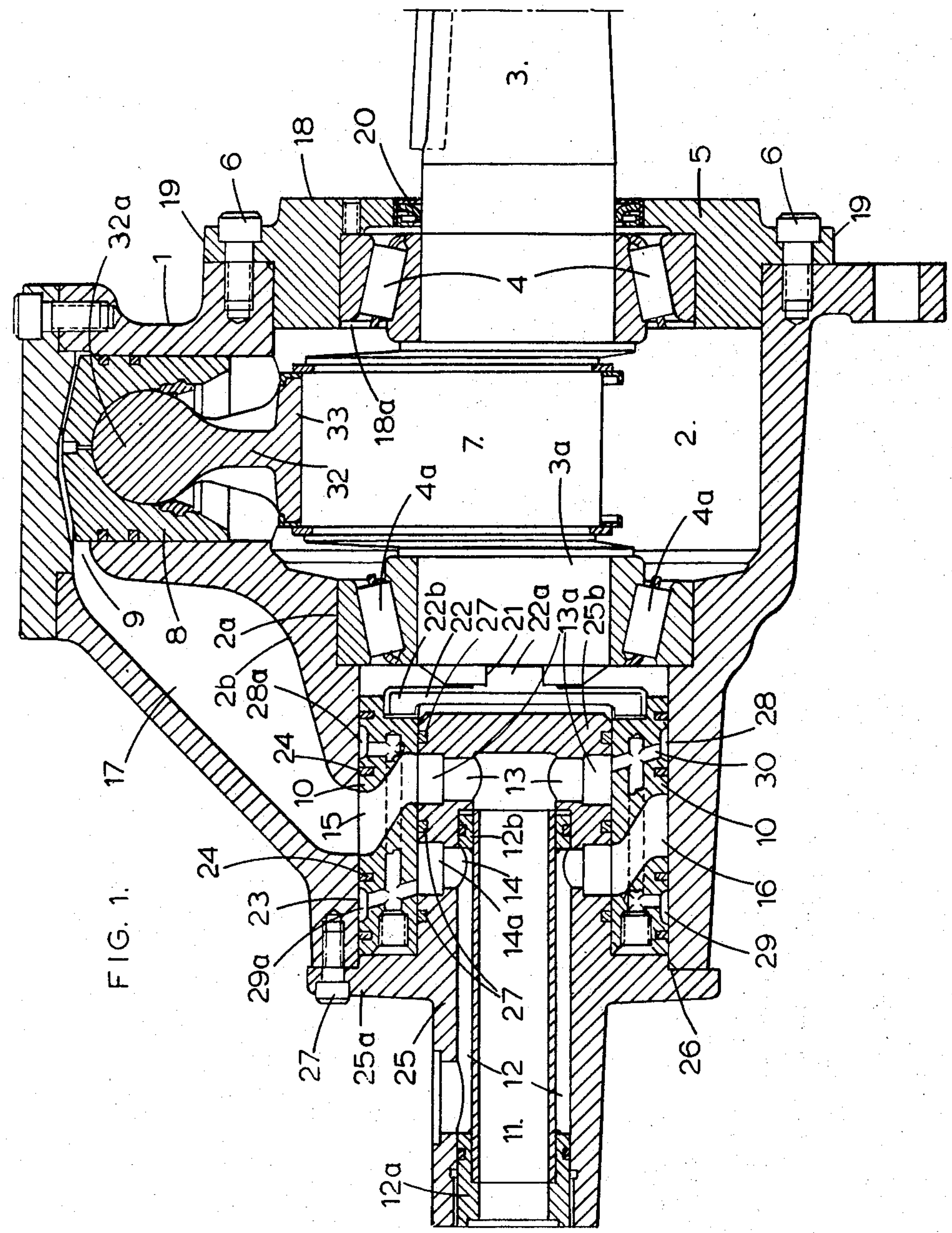

FIG. 1.

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Nov. 28, 1967 J. F. BEDFORD 3,354,786

HYDRAULIC MOTORS

Original Filed June 25, 1964 9 Sheets-Sheet 2

FIG. 2.

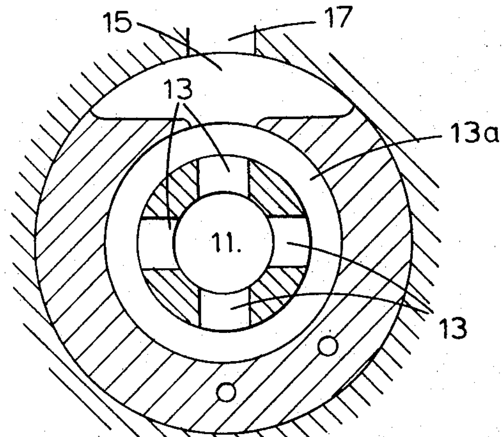
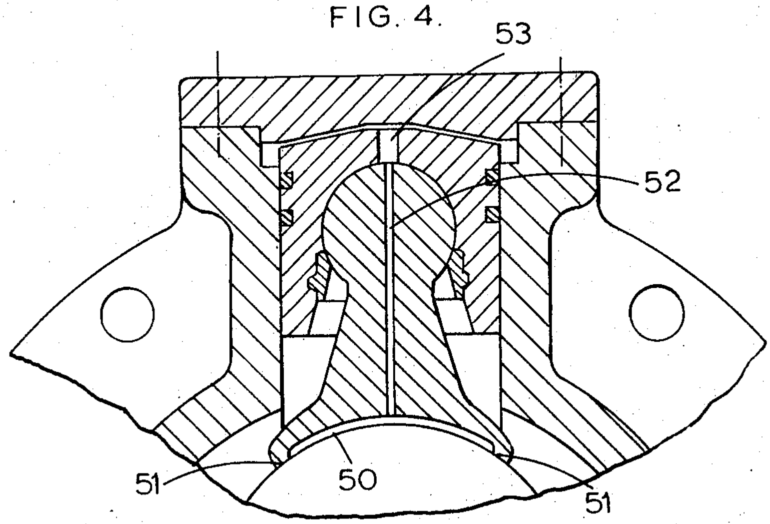


FIG. 4.



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HYDRAULIC MOTORS

Original Filed June 25, 1964 9 Sheets-Sheet 3

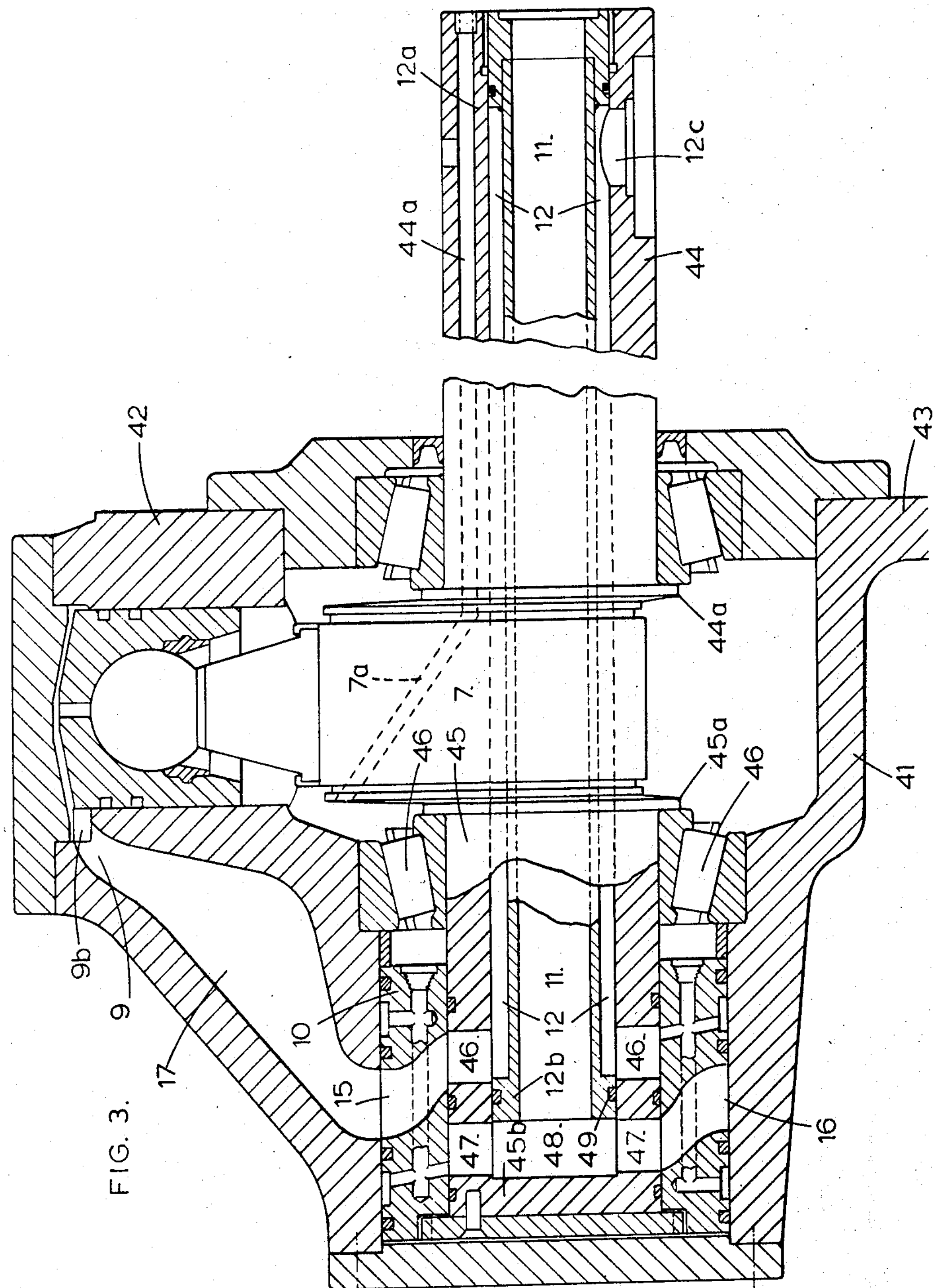

FIG. 3.

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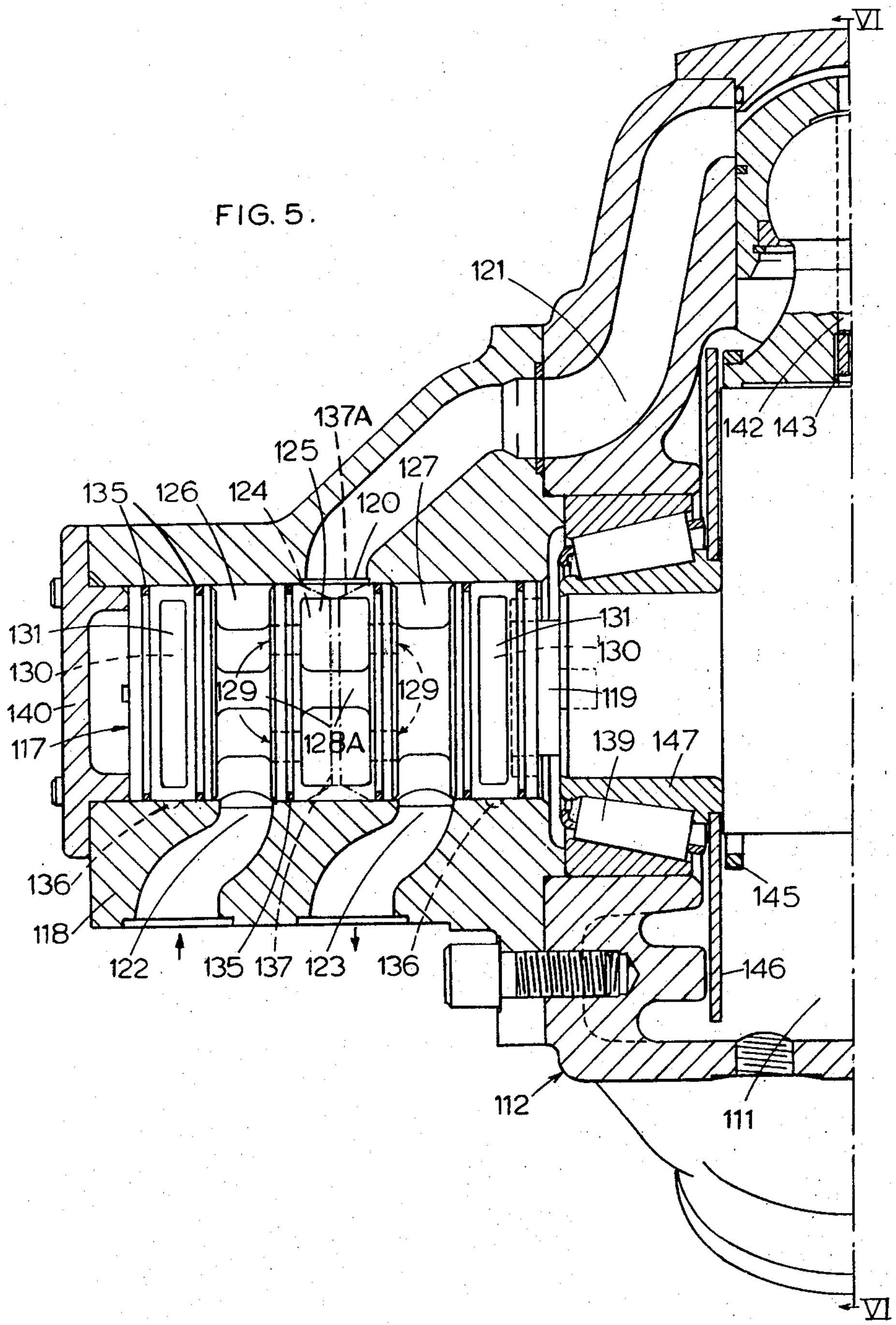
HYDRAULIC MOTORS

Original Filed June 25, 1964 9 Sheets-Sheet 4

FIG. 5.



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HYDRAULIC MOTORS

Original Filed June 25, 1964 9 Sheets-Sheet 5

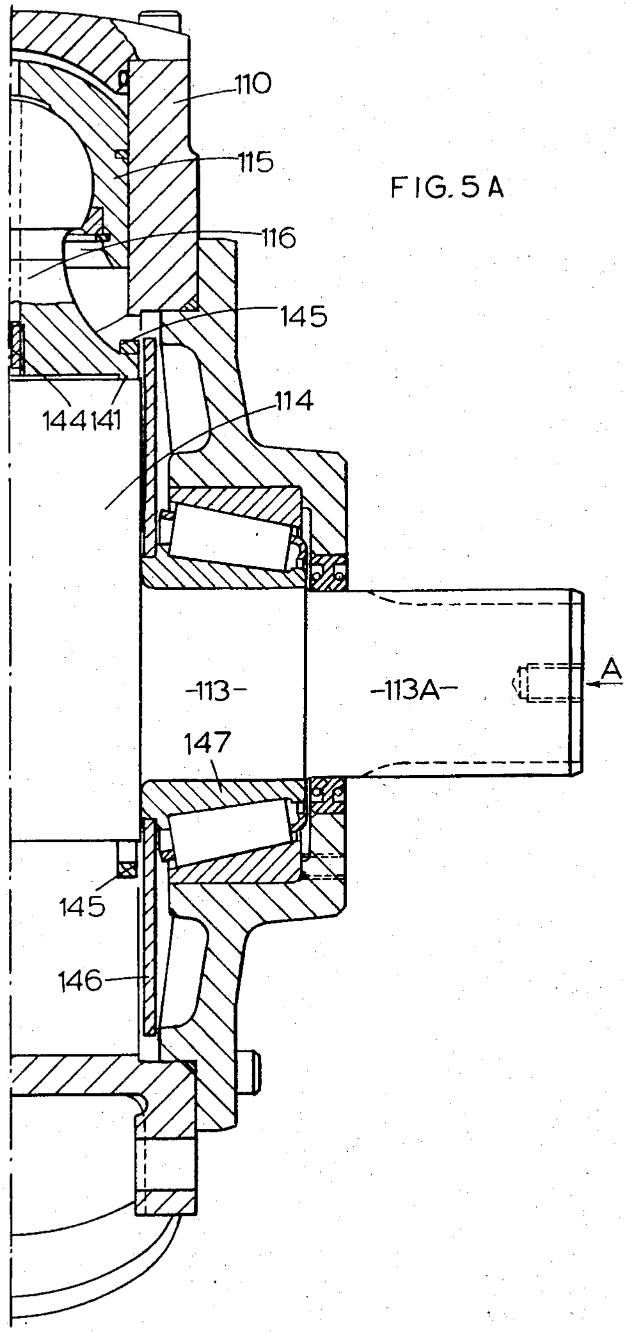

FIG. 5A

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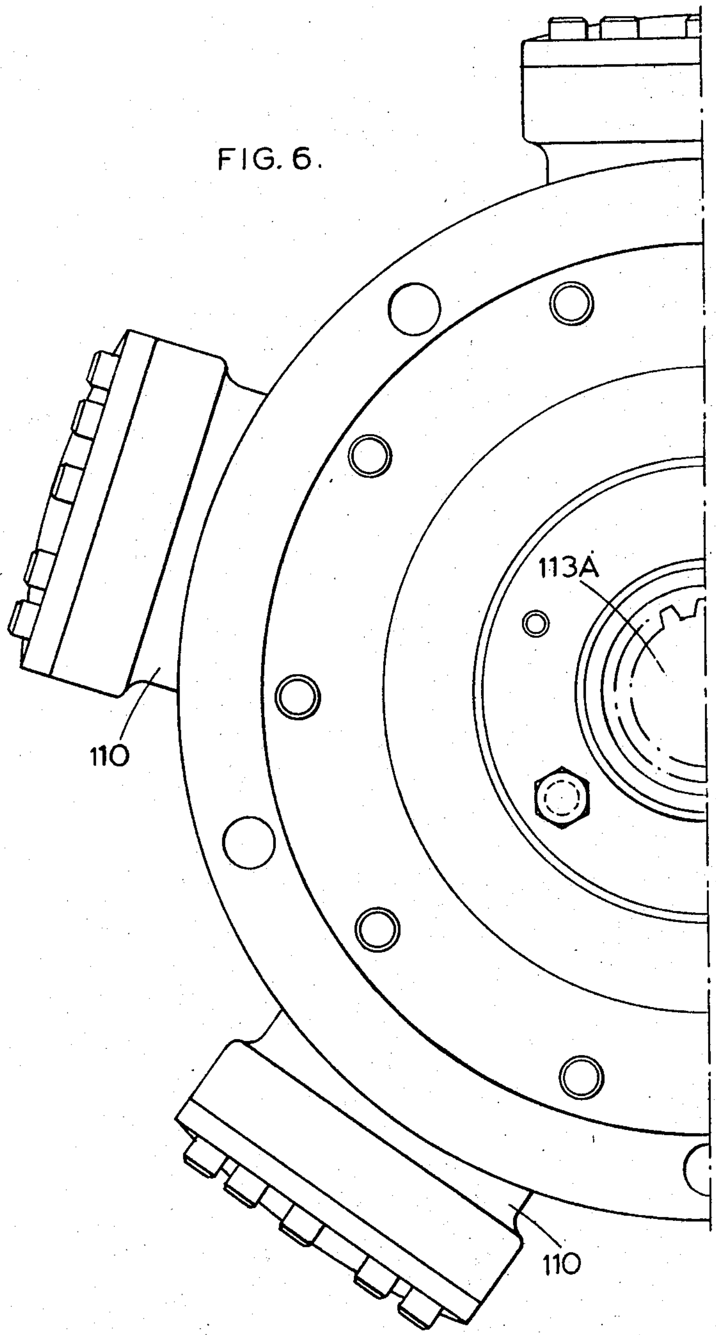
FIG. 6.



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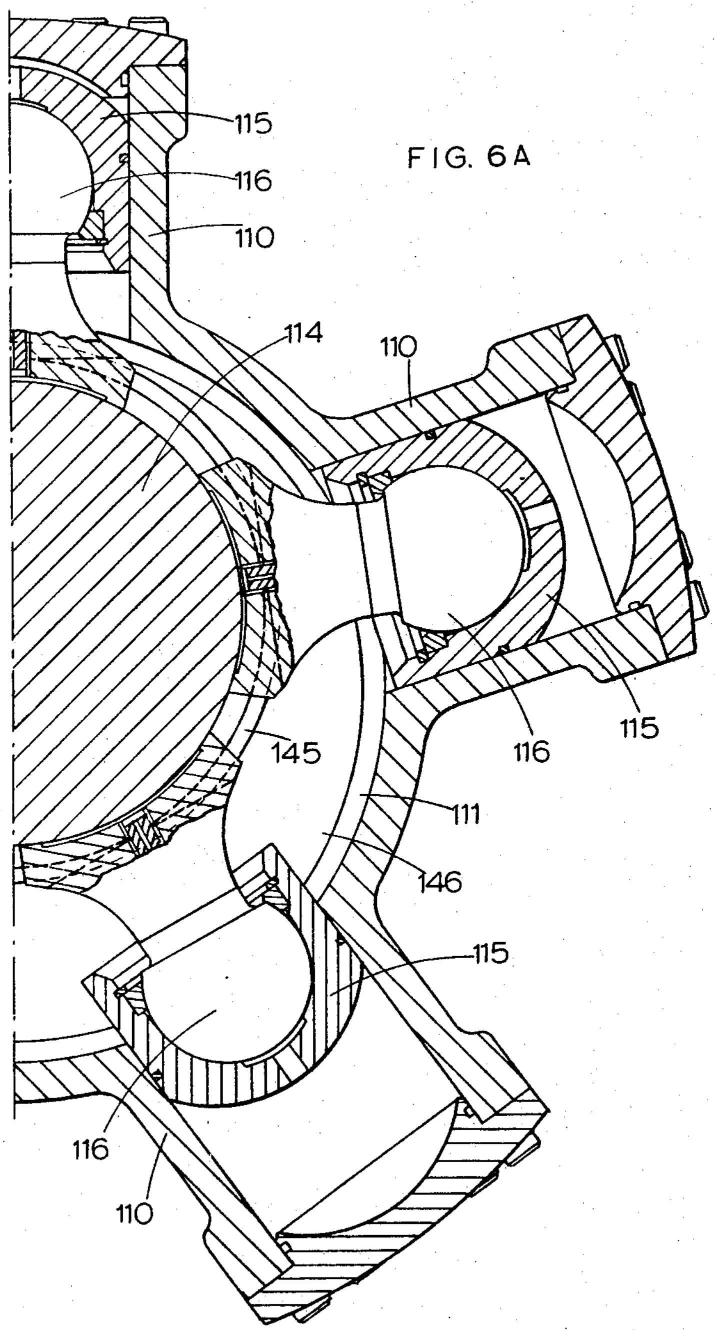
Nov. 28, 1967 J. F. BEDFORD 3,354,786

HYDRAULIC MOTORS

Original Filed June 25, 1964 9 Sheets-Sheet 7

FIG. 6A



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HYDRAULIC MOTORS

Original Filed June 25, 1964 9 Sheets-Sheet 8

FIG. 7.

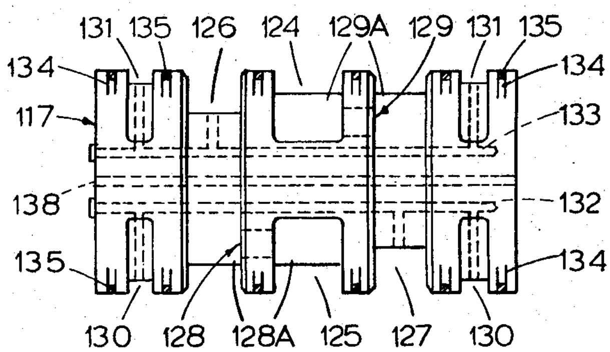


FIG. 8.

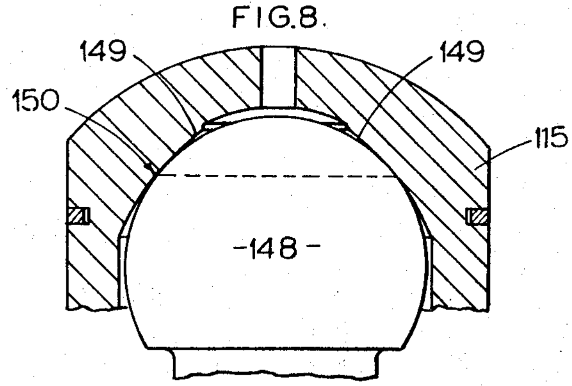


FIG. 9.

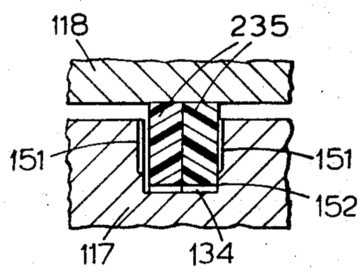
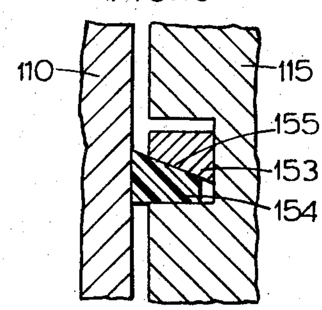


FIG. 10.



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HYDRAULIC MOTORS

Original Filed June 25, 1964 9 Sheets-Sheet 9

FIG. 11.

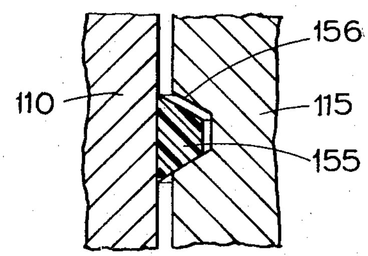
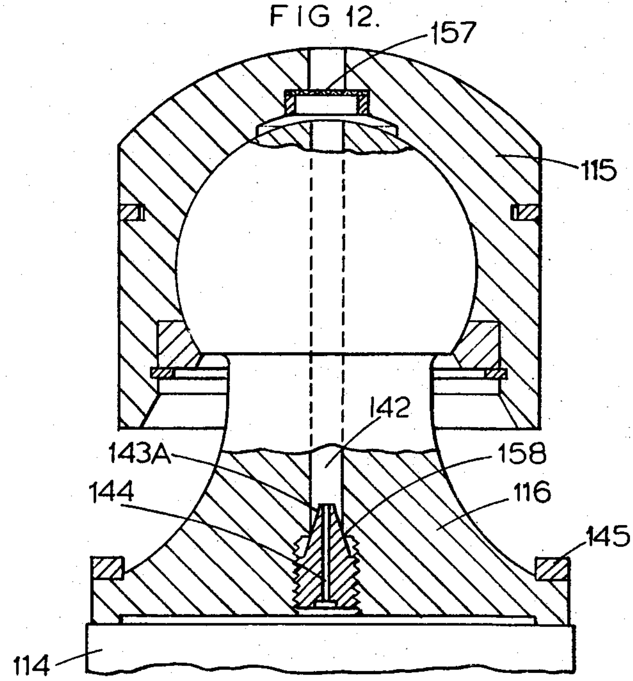


FIG 12.



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3,354,786

HYDRAULIC MOTORS

John Ferguson Bedford, London, England, assignor to Chamberlain Industries Limited

Continuation of application Ser. No. 380,977, June 25, 1964. This application Feb. 21, 1966, Ser. No. 540,118
Claims priority, application Great Britain, Dec. 21, 1960, 43,889/60; June 25, 1963, 25,285/63; Mar. 25, 1964, 12,759/64
29 Claims. (Cl. 91—180)

This application is a continuation of my copending application Serial No. 380,977 filed June 25, 1964, now abandoned, which last-mentioned application was a continuation-in-part of my now-abandoned application Serial No. 158,683 filed December 12, 1961.

The present invention relates to low-speed hydraulic motors.

The hydraulic motor of the invention comprises a series of radially arranged cylinders each having a piston axially movable therein. The pistons are connected by piston rods to act simultaneously on an eccentric portion of a central shaft. In one embodiment, the shaft rotates and the cylinders are stationary. In another embodiment, the cylinders rotate and the shaft is held stationary.

A feature of the invention resides in the provision of a hydraulically balanced rotary control valve which places each cylinder alternately in communication with a supply of pressure fluid and a discharge outlet for spent pressure fluid. The valve passages are so arranged that hydraulic forces acting axially on the valve are balanced as well as hydraulic forces which act laterally on the valve members. The freedom from extraneous forces acting on the valve is further enhanced by the provision of a self-aligning coupling which prevents the transmission of bending forces from the shaft to the rotary element of the valve.

The shaft comprises an eccentric smooth cylindrical drive surface which is simultaneously engaged by complementarily shaped slipper portions of all of the piston rods. Each slipper comprises a marginal portion which directly engages the eccentric drive surface of the shaft and a hollow central portion which is spaced from the drive surface and laterally enclosed by the marginal portion. The central portion of the slipper is connected by a longitudinal passage in the piston rod to receive pressure fluid acting on the piston and thereby produce a hydraulic balancing force acting against the drive surface which counteracts the working force produced by the piston. This reduces the net force of engagement between the peripheral portion of the slipper and the drive surface, thereby minimizing friction between the slippers and the eccentric drive surface.

Another feature resides in the provision of special piston rings which have frusto-conical surfaces arranged to increase the pressure of engagement against the cooperating cylinder wall when the working pressure of the fluid increases.

Embodiments of the invention will now be described, by way of example, with reference to the accompanying drawings in which:

FIG. 1 is a sectional side elevation of an embodiment in which the shaft of the motor is rotated;

FIG. 2 is a diagrammatic sectional end elevation showing the arrangement of control of the motor;

FIG. 3 is a sectional side elevation of another embodiment in which the shaft of the motor is stationary and the cylinders rotate about the shaft;

FIG. 4 is a diagrammatic sectional elevation showing a preferred means for minimizing friction by maintaining pressure balance with the transmission of force from the pistons to the eccentric drive surface of the shaft;

FIGS. 5 and 5A when placed side by side constitute a longitudinal section through a radial cylinder hydraulic motor having a modified form of rotary valve;

FIGS. 6 and 6*a* when placed side by side constitute an end elevation looking in the direction of arrow A in FIG. 5A, the view being partly broken away and shown in section on the line VI—VI of FIG. 5;

FIG. 7 is a plan view separately showing the control valve of the hydraulic motor illustrated in FIGS. 5 and 6;

FIG. 8 is an enlarged fragmentary sectional view separately showing one form of connection between a piston and its connecting rod;

FIG. 9 is a fragmentary sectional view showing a valve ring arrangement;

FIGS. 10 and 11 are fragmentary sectional views showing two different piston ring constructions; and

FIG. 12 is an enlarged fragmentary sectional view showing a connecting rod arrangement.

Referring to FIGURES 1 and 2 of the drawings the radial type hydraulic motor comprises an integral set of symmetrical radial cylinders **1** radiating from a common rotor chamber **2** and forming with the chamber a stator part of the motor, an output shaft **3** journalled in thrust-journal bearings **4** in an end wall **5** of said chamber and secured by studs **6** to the chamber **2**, the shaft having its axis intersecting all of the axes of the cylinders, a cylindrical rotor member **7** integral with but eccentric with respect to said output shaft, pistons **8** in said cylinders **1** and driving connection with the rotor, a port **9** at the outer end of each cylinder, a rotary distributor sleeve valve **10** arranged to rotate in unison with said shaft, inlet and return conduits **11** and **12** respectively concentric with said rotary distributor valve, radial transfer passages **13** and **14** respectively establishing communication between the conduit **11** and conduit **12** respectively and supply ports **15** and return ports **16** respectively in the sleeve valve **10**. These passages **13** and **14** communicate with continuous annular channels **13*a*** and **14*a*** respectively machined in the cylindrical periphery of a tubular end fitting **25** hereinafter described, the passages **13** and **14** being radial borings in the fitting **25** and the conduit **12** consisting of a number of longitudinal slots in a sleeve **12*a*** also hereinafter described. The ports **9** comprise the outer ends of passages **17** the inner ends of which where they align with the outer ends of the ports **15** and **16** and said outer ends of the ports **15** and **16** all having their centres in a common plane normal to the axis of the shaft **3**.

In the embodiment illustrated in FIGURE 1 the stator or housing part **1** of the motor comprises a casting with five symmetrically disposed radial cylinders **1** the bores of which at their inner ends open into the cylindrical chamber **2** which might be regarded as the equivalent of a crank chamber, this chamber containing the rotor **7** which is eccentric and integral with the output shaft **3** journalled opposite sides of the rotor **7** in roller type foregoing thrust-journal bearings **4** and like bearings **4*a*** the outer rings of which are mated in, one a recess **2*a*** in an integral lateral extension **2*b*** of the stator casting, and the other in a like recess **18*a*** of a cover plate an annular end ring **18** mated in a side opening of the stator casting and comprising with the bearing unit 4 a closure for one side of the chamber 2 by being secured to the stator casting by a flange or radial lugs 19 abutting against an annular end wall of the casting and receiving the studs 6 threaded into the casting. A sealing ring or packing assembly 20 is fitted about the shaft and fixed in the said end ring 18.

The shaft is extended beyond the end ring 18 to receive the hub of a road wheel or other driven member, and at its other end it projects beyond the rotor as a stub shaft 3 in the thrust-journal bearing 4*a*. This stub shaft part is recessed diametrically or radially as at 21 to receive a complementary projection 22*a*, or projections, on a coupling disc 22, which may be of shallow dished form, mating as at 22*b* with a positive driving connection in the opposed annular end face of the ported sleeve 10. The coupling disc 22 operates as a self aligning coupling or universal joint. Bending or other transverse stresses applied to the output shaft 3 by the load connected thereto will tend to disturb the axial alignment between the sleeve valve 10 and the shaft 3, 3*a*. The coupling disc 22 permits such misalignment without imparting undesirable stresses to the sleeve valve 10.

The sleeve 10 is in the form of a rotatable sleeve valve member which at its periphery bears rotatably in a cylindrical bore 23 of the stator casting, which bore is concentric with the shaft 3 but larger in diameter than the stub axle part 3*a*. Sealing rings 24 in the periphery of the sleeve provide the necessary seal between the sleeve and the bore in which it rotates. The bore 23 at its outer end is closed by a flange 25*a* of the aforesaid end fitting 25 which whilst serving to close the end of the housing 1 containing the sleeve valve 10 also serves as a housing for the sleeve 12*a* along which are formed four or other convenient number of slots comprising the conduiting 12, this sleeve 12 being fixed about a tube 11*a* the bore of which comprises the supply conduit 11. The flange 25*a* is stepped to mate closely in said bore 23, a sealing ring 26 seated against a bevelled end of the bore 23 seating against the step, studs 27 being passed through the flange 25*a* into opposed annular outer ends of the stator casting containing the said bore.

The said end fitting or housing 25 extends concentrically along the said bore 23 nearly to the said coupling member 22 where the housing 25 is closed by an inner end wall 25*b* and its cylindrical periphery forms a bearing fit in the said valve sleeve 10 and is provided with three sealing rings 27 by which the two axially spaced sets of radial transfer passages 13 and 14 in communication with the conduits 11 and 12 respectively are sealed off from communication with each other. There is one port 15 and likewise one port 16, these two ports being diametrically opposed and at their inner ends are of substantially square cross-section but at their outer ends each occupying about 150° of the circumference of the sleeve 10, so that they are flared almost on a chord (relative to the sleeve 10) from their inner ends to their outer ends as shown in FIGURE 2, these two ports establishing the necessary sequence communication with the supply and return conduits 11 and 12 and the cylinders 1, the communication from the ports of the sleeve to the cylinders being via the said passages 17 formed in the stator casting and terminating at their outer ends with the relatively narrow ports 9 in the outer ends of the cylinders. It will be evident that the outer ends of the transfer passages 15 and 16 are oblong in cross-section, whereas the opposed ends of the passages 17 are much smaller in area and approximately square in cross-section, these ends of the passages 17 each occupying an angle of about 20° measured from the axis of the shaft 3 and as there is only one of said passages for each cylinder port 9 the ends opposed to the sleeve 10 are positioned so that their centres are contained in a common plane as aforesaid this plane being midway between the axially spaced two sets of ports 13 and 14. It will be understood that the ports 15 and 16 alternate "two and three" with respect to the number of cylinders each communicates with, with a progressive change up from "two to three" in one case and change down from "three to two" in the other case. Thus looking at FIGURE 2 the port 15 is open to two cylinders and about to bring in a third cylinder, it being apparent that at this moment the port 16 is in communication with three cylinders and just about to cut off one of them. In order to bring the centres of the outer ends of the ports 15 and 16 and the centres of the inner ends of the passages 17 in the said common plane, the ports 15 and 16 are inclined relative to the axis of the motor in opposite directions as shown in FIGURE 1.

It will be apparent that the sleeve valve 10 is subjected to forces due to hydraulic fluid pressure through the ports 15 and 16, and that these forces occur in the zone containing the aforesaid plane common to the centres of the ports comprising the inner ends of the passages 17. Accordingly in order to balance these forces and also, at the same time ensuring good lubrication of the sleeve 10, there are provided in balanced relationship with and on opposite sides of the outer ends of the ports 15 and 16 peripheral channels, on one side two symmetrical channels 28, 28*a* each occupying about 160°, and two like channels 29, 29*a* on the other side, the total area of these channels balancing the total area of the ports comprising the inner ends of the passages 17. Hydraulic fluid pressure is admitted to the channels 28 and 29 and to the channels 28*a* and 29*a* by radial bores 30 and 31 in constant communication with the annular channels 13*a* and 14*a* respectively. The channels communicate with each other by axial bore 30*a* from the bore 30 and axial bore 31*a* from bore 31. This arrangement ensures complete balancing and proper lubrication of the sleeve.

In the embodiment shown in FIGURE 3 the main casting 41 carrying the cylinders 42 is arranged to operate as the rotor, e.g. by mounting it by radial lugs 43 in a road wheel or other unit to be driven hydraulically. For this purpose the casting 41 rotates relative to an axle 44 adapted to be secured fixedly in position, e.g. to a road vehicle chassis when the motor is mounted in a road wheel. Accordingly, the ported sleeve 10 will have to be stationary and the connection of the hydraulic fluid pressure to the conduit 11 will have to be made from one end of the axle and for this purpose the tube comprising the conduit 11 is carried along the axle through the member 7 (which in this case is not a rotor) and is adapted as at 11*a* to be connected to the source of hydraulic fluid pressure, and likewise the conduit sleeve 12*a* is carried through the rotor and the outlet therefor is arranged as at 12*c* at the supported end of the axle 44. Instead of the stub axle 3*a* of the previous embodiment the equivalent of the aforesaid housing member 25 is substituted by a sleeve 45 which may be regarded as a part of the axle of the motor. This sleeve 45 is formed with a flange 45*a* opposed to a similar flange 44*a* on the hollow axle 44, and between these flanges is fixed the member 7 which is maintained stationary to ensure that the main housing 41 rotates about it. Desirably the axle 44, sleeve 45 and member 7 may be machined as a one-piece member, but in any event the main axle of the hydraulic motor shown in FIGURE 3 may be regarded as comprising the shaft 44 and extension 45 integral effectively with the member 7, the conduits 11 and 12 passing through this assembly, the main housing accommodating bearings 46 and carrying relatively fixedly the sleeve valve 10, the ports in which communicate with the conduits 11 and 12 via the member 45.

The ports 15 and 16 in the sleeve 10 are in registration with radial holes 46 and 47 respectively in the shaft extension 45, the holes 46 communicating with the annular cross-section discharge conduit 12 and the radial holes 47 communicate with the conduit 11 via a clearance 48 between an end wall 45*b* of the sleeve 45 and the flanged end 12*b* of the conduit 11, such flanged end carrying a sealing ring 49. A lubricating oil duct 7*a* in the member

7 communication with a duct 44a extending along the shaft 44, and the duct 44a may be a cored hole along the shaft 44.

With the foregoing arrangement for balancing the forces upon the distributor valve, it is desirable to ensure that at all running conditions the mid-balancing of the forces is not detracted from by loss of torque and pressure balance in the transmission of the forces from the pistons to the output rotary unit, i.e. the shaft in FIGURE 1 or the cylinder assembly of FIGURE 3. In this connection in radial type hydraulic motors in which the piston rods 32 carry slippers 33 engaging the eccentric on the main shaft the piston force is transmitted to the said shaft via the eccentric by means of white metal slippers on the segmental faces of the piston rods and generally the area of each slipper is less than the cross-sectional area of the associated piston. Provided that the co-efficient of friction between the sliding surfaces is low, no difficulties in terms of loss of torque, heat generation and wear occur. Such conditions pertain only when the slipper is supported by a hydrodynamic oil film, but at slow speeds this film cannot form adequately.

Referring now to FIGS. 5, 6 and 7 of the drawings, the motor comprises a stationary bank of five radial cylinders 110. The cylinders radiate from a common chamber 111 in a body part 112 of the motor, and a shaft 113 extends through the chamber. In the chamber 111 the shaft is provided with an eccentric 114 which is connected to pistons 115 through short connecting rods 116. The outer end 113A of the shaft has an external driving connection.

The control valve comprises a rotary cylindrical valve member 117 arranged within a valve housing 118. The valve member is driven by the shaft 113 through an Oldhams coupling 119 to accommodate misalignment and avoid transmission of bending force.

The valve housing 118 is formed with central circumferentially-spaced ports 120 which open into conduits 121 which extend outwardly to communicate with the outer ends of the radial cylinders. The ports 120 function as fluid transfer ports. The housing is also formed with a supply port 122 and a return port 123 which are spaced on either side of the transfer ports and communicate with hydraulic supply and return conduits (not shown). It is of advantage that the function of these ports is exchangeable to permit simple reversal of the motor drive.

The valve member 117 is formed with a central pair of arcuate channels 124 and 125 each of which extends through an angle which may be of approximately 150°. These channels are arranged to communicate sequentially with the transfer ports 120; the size of the ports 120 determines the circumferential length of the valve lands between the channel ends and thus determines the length of the channels. It is of importance that the valve lap is arranged to give smooth fluid control and low fluid leak across the lands; a small lap gives shock-free running but a high degree of leakage, whereas a large lap reduces leakage but increases shock.

The member 117 is also formed with a pair of circumferential annular channels 126 and 127 aligned with the supply and return ports. A first pair of axial passageways 128 connects channel 126 with channel 125 and a second pair of axial passageways 129 connects channel 127 with channel 124; the passageways are located adjacent strengthening webs 128A and 129A.

The valve member is further formed with two pairs of arcuate balancing channels 130 and 131 spaced axially outwardly from and diametrically opposed to channels 124 and 125, respectively. Each balancing channel subtends an angle of about 150°, i.e. the same as that subtended by channels 124 and 125 and the channel width is approximately half that of the channels 124 and 125. Channels 130 are connected to channel 124 through an end-plugged branched bore 132, channel 127 and passageways 129, and channels 131 are connected to channel 125 through an end-plugged branched bore 133, channel 126 and passageways 128.

Six annular grooves 134 are formed in the valve member 117 to accommodate sealing rings 135. The rings are at each end of the member 117 and between the various channels.

It will be noted that the various channels and grooves are positioned symmetrically about the effective centre of the valve member and that the axial space between the rings which bound channels 130 and 131 is half that between the rings which bound channels 124 and 125; the rings 135 act to prevent longitudinal leakage and as the clearance between the rings and the valve housing is considerably less than the clearance between the valve lands and the housing, pressure drop takes place effectively across the valve rings.

In operation, as the shaft 113 rotates, the member 117 rotates to bring the cylinders 110 sequentially into communication with the hydraulic supply and return conduits thus providing the rotary drive for the shaft. As channel 126 is in communication with the supply conduit, the hydraulic pressure in channel 125 will exceed that in channel 124 and act to cause an out-of-balance force on the valve member; this force is, however, balanced by the forces created by the hydraulic supply pressure in channels 131 which are equal in arcuate length to channel 125. Similarly, the force created by the fluid pressure in channel 124 is balanced by the provision of channels 130.

It will be appreciated that the symmetrical construction of the valve member gives balanced running irrespective of the difference between inlet and outlet pressures, and irrespective of the direction of rotation.

To balance the valve member further, compensation can be made for the effect of hydraulic pressure from the ports 120 on the lands between the arcuate channels 124 and 125. This may be effected by providing scallops (e.g. at 136) in the valve housing in the planes of the corresponding balancing channels 130 and 131, the effective area of the two scallops being equal to that of the transfer ports 120. Alternatively, the lands may be recessed as at 137A and the recesses interconnected by a transverse bore 137, angled to avoid bore 138 (mentioned below). The axial width and the depth of the grooves or scallops should be reduced as far as is practically possible so that the compressibility effects will not seriously affect the valve balance.

To balance the valve member axially, a central axial bore 138 is provided to extend through the valve member to communicate through shaft bearing 139 with the central chamber 111 in the body part of the motor and so prevent build-up of pressure between the member 117 and end plate 140 of the valve housing.

Due to the balancing of the valve, frictional forces are reduced and this promotes high efficiency and rapid acceleration; the torque losses are further minimized by an improved connecting rod arrangement according to the invention. Referring to FIG. 5 the big end of each connecting rod has a large area slipper with a peripheral bearing skirt 141. The skirt is continuous and encloses a space which is connected to pressure fluid through a radial bore 142 which extends through the rod and piston. This results in a floating action of the slipper which greatly reduces torque losses. However, as the effective slipper area is larger than or even comparable with the piston area the floating slipper will tend to lift. To control this tendency to lift, the inner end of the bore is constricted by means of a removable part 143 formed with a metered orifice 144 to restrict oil flow. Thus, when excessive lifting of the slipper occurs the oil pressure under the slipper drops due to the restricted oil flow and the slipper returns. In fact, the slipper should float at a predetermined distance from the surface of the crankshaft eccentric and to this end the orifice 144 (FIG. 12) is designed in accordance with the slipper and piston dimensions to minimize torque loss and fluid leakage. Also, if excessive leakage develops due to scores in the slipper skirt, the resulting pressure drop under the slipper causes the skirt to clamp down on the eccentric to increase wear and consequently effect resealing between the mating surfaces.

In an alternative construction, the space enclosed by the peripheral bearing skirt is sufficiently large to reduce appreciably friction caused by metal-to-metal contact, although the space may not be sufficiently large to cause the floating action described above. Again the bore **142** is constricted to provide a safety device against the consequences of skirt rupture.

The connecting rods are, in operation, urged into engagement with the eccentric **114** by hydraulic pressure. However, positively to prevent disengagement due to, for example, centrifugal force or pressure build-up under the slipper, a pair of retaining rings **145** is located in recesses in the big ends of the connecting rods. Disc-like plates **146** are mounted on bearing members **147** to prevent accidental displacement of the rings and the plates are themselves prevented from displacement by a suitably shaped housing interior.

Alternatively, the skirts are grooved on their outer faces and retaining rings, such as **145**, are located in these grooves; it will be appreciated that the retaining rings are fitted in position before the rods **116** are moved outwardly into their positions and the shaft is introduced into the motor body part **112**.

FIG. 8 shows the connection between the small end of the connecting rod **116** and the piston **115** in greater detail and illustrates a further feature of the invention. In the production of hydraulic motors it is known to be difficult to produce satisfactory ball-and-socket joints for the piston assemblies. It is, of course, impossible to produce a ball-and-socket with part-spherical co-operating surfaces which are geometrically perfect and, inevitably, high spots and other imperfections exist. To overcome this difficulty the small end of the connecting rod comprises a ball **148** which is as nearly spherical as possible and the piston socket is shaped to have central vertical sections of which the outer parts comprise a pair of arcs **149** having a radius greater than that of the ball **148** to provide ring contact, at **150**, with the ball. The two arcs may be joined by a recessed area, as shown, or by a transition curve. It will be appreciated that the radius of the arcs is only slightly greater than that of the ball and that the difference is exaggerated in the drawing.

It is envisaged that this arrangement may be inverted by providing a piston socket of part-spherical shape and forming the connecting-rod ball with a symmetrical proud ring in its outer part. In this case the central vertical sections of the ball would comprise a pair of arcs each having a radius slightly smaller than that of the spherical part of the socket, the arcs again being joined by a transition curve.

It is also envisaged that the desired ring contact may be obtained by other means such as a combination of off-spherical parts or non-spherical parts such as a frusto-conical piston socket.

The valve rings **135** may be formed of plastic material, for example P.T.F.E., but when such material is employed it is important that the rings rotate relative to the smooth-finish surface of the housing **118** but remain stationary relative to the relatively rough surface of the grooves **134**; thus wear on the relatively soft plastic material is minimized. To this end, as shown in FIG. 9, each groove **134** is formed with at least one relief flute or like passage **151** which extends from the mouth of the groove to within a short distance from the base of the groove to provide a relief passage to prevent build-up of fluid pressure on the low-pressure side of the ring. Preferably, a series of, say, four regularly spaced flutes is provided.

In use, each ring is held against rotation in its groove by high-pressure fluid forcing the ring axially against the groove face on the low-pressure side of the ring; however, the ressulting frictional force tends to be reduced due to leakage of high pressure fluid under the ring to the low-pressure face of the groove. The flutes **151** are radially arranged to relieve pressure caused by such leakage, but it will be appreciated that "lands" **152** at the roots of the flutes prevent free leakage from under the ring.

The fluid pressure under the rings also gives the beneficial effect of urging the rings outwardly into sealing contact with the surrounding housing **118**.

Flutes **151** may, as shown, be provided in both faces of the groove. This is of importance in the case of rings where high pressures and low pressures exist alternatively on each side of the groove. Also, the flutes are relatively easy to produce in pairs, by drilling.

In the construction shown in FIG. 9, two similar rings **235** are provided in each groove **134**, each ring width being approximately half the groove depth or, more specifically, the depth of penetration of the rings into the groove; thus, the axial force exerted on each ring is considerably greater than the radial force exerted. More than two rings may be provided in each groove.

As an alternative to the flutes **151**, suitably produced machining marks can be employed to give the same effect. It is also possible that flutes be found in the appropriate faces of the rings instead of in the groove walls.

According to a further feature of the invention, as illustrated in FIG. 10, each piston **115** is provided with at least one composite piston ring. The composite piston ring comprises two ring parts **153** and **154** which are formed with co-operating frusto-conical faces **155** so arranged that on axial compression of the composite ring, part **153** will contract diametrically and the other part **154** will expand diametrically as the contacting faces **155** of the ring parts slide over each other. Thus, the action of the working hydraulic pressure will effect close sealing engagement of ring part **153** with the piston **115** and of ring part **154** with the cylinder **110**.

The ring parts **153** and **154** are preferably formed of plastics material, for example P.T.F.E., as such rings usually undergo radial expansion and contraction more satisfactorily than metal rings.

Preferably, also, the ring parts **153** and **154** are formed with gaps which comprise slots angled in opposite senses to prevent interference between the rings. This form of gap may also be employed in the valve rings **235** of FIG. 9.

In the alternative arrangement shown in FIG. 11, a single piston ring **155** is arranged in each groove **156** of piston **115**. The piston ring **155** is of trapezoidal section and the groove **156** is of complementary shape. In use, hydraulic pressure urges the ring downwardly into engagement with the lower face of the groove **156** and the slope of this face urges the ring to expand diametrically into close contact with the adjacent wall of the cylinder **110**.

Referring now to FIG. 12, the connecting rod **116** is formed at its small end with a recess which accommodates a filter button **157**. The filter button comprises a circular disc of bronze gauze supported across a steel ring, and prevents the passage of dirt through bore **142**. It is important that the metered orifice **144** remains free, and to supplement the action of the filter button **157** a conical outer end **143A** is formed on the removable part to provide an annular dirt trap around the orifice mouth. The conical end **143A** also acts, by engaging shoulder **158** of the bore **142**, as a means for locking the removable part which is screw-threaded.

I claim:

1. A motor comprising a main body part defining a set of symmetrical radial cylinders radiating from a common chamber in said body part; a shaft with its axis intersecting all of the axes of the cylinders, said shaft comprising an eccentric member in said chamber, said body part and said shaft being rotatable relative to each other; pistons in all of said cylinders in simultaneous driving connection with said eccentric member; a port at the outer end of each cylinder; a distributor valve; coupling means connecting said valve with said shaft; hydraulic fluid pressure inlet and return conduits; a pair of diametrically opposed distributor ports in said valve and with which said conduits are in continuous communication; individual passages connecting said cylinder ports with said conduits via said distributor ports in said valve; axially spaced peripheral channels in said valve, equalizing passages in said valve continuously connecting said channels with said hydraulic fluid pressure conduits to create at the periphery of said valve hydraulic fluid pressure to balance the forces due to the hydraulic fluid pressure acting upon said valve at its ports and circumferential grooves which receive sealing rings formed of resilient plastic, the circumferential grooves being positioned so that the rings define the axial limits of the effective areas of the valve member on which the hydraulic forces act.

2. A motor according to claim 1, wherein said body part comprises a housing member which is an element separate from and in axial alignment with said shaft.

3. A motor according to claim 1, wherein said conduits are disposed concentrically relative to each other and to said valve said valve being a sleeve valve disposed externally relative to the outer one of said conduits, the inner conduit being open adjacent one end to communicate continuously with said one end of said distributor ports and the outer conduit being open at a position axially spaced from said one end of the inner conduit for continuous communication with said one end of the other distributor port.

4. A motor according to claim 1, wherein said driving connection comprises a piston rod which extends between each piston and said eccentric member; and a slipper on each piston rod in driving engagement with said eccentric member, the area of each slipper adjacent to the eccentric member being greater than the cross-sectional area of its associated piston, the face of each slipper in engagement with said eccentric member having a cavity formed therein which occupies a predominant area of said face and leaves a continuous marginal surface for direct engagement with the surface of said eccentric member, interconnecting hydraulic pressure fluid supply passages being formed in each piston and in each piston rod providing communication between the said cavity and the working space in the associated cylinder.

5. A motor according to claim 1, wherein said distributor valve comprises a sleeve coaxially connected to said shaft to be held against rotation thereby, said motor further comprising a valve housing member with which the external surface of said sleeve is engaged.

6. A motor according to claim 1, wherein said valve is in permanent driving connection with said shaft by means of a self-aligning coupling means which transmits only rotary driving force.

7. A motor according to claim 1, wherein said coupling means comprises means for the transmission of torque, said means for torque transmission being operative to prevent the transmission of bending moments.

8. A motor according to claim 1, wherein said distributor ports in said valve are separated by lands with which said individual passages come into registry, said motor further comprising scallops formed in said body part and registering with said peripheral valve channels for balancing forces acting on said lands.

9. A motor according to claim 1, wherein said distributor ports in said valve are separated by lands with which said individual passages come into registry, said motor further comprising recesses formed in said lands and a transverse bore interconnecting said recesses.

10. A radial type hydraulic motor comprising a main body part incorporating a set of symmetrical radial cylinders radiating from a common chamber in said body part; a shaft with its axis intersecting all of the axes of the cylinders, said shaft and said body part being rotatable relative to each other and comprising eccentric means intermediate its ends, said eccentric means having a smooth cylindrical drive surface the axis of which is spaced from and parallel to the longitudinal axis of said shaft; a piston in each of said cylinders; a port at the outer end of each cylinder; a rotary distributor valve in axial alignment with said shaft; self-aligning coupling means connecting said valve to said shaft; means defining a hydraulic fluid pressure supply conduit and a return conduit; a pair of diametrically opposed distributor ports in said valve and with each of which one of said conduits is in continuous communication; individual passages connecting said cylinder ports with said conduits via said distributor ports; equalizing means by which the hydraulic fluid pressure in the fluid pressure supply and return conduits is utilized to balance forces due to said pressure upon said distributor valve; a piston rod extending between each piston and said eccentric means; a slipper on each piston rod in driving engagement with said drive surface, the area of each slipper adjacent to said drive surface being greater than the cross-sectional area of its associated piston; means defining a cavity in the drive surface engaging face of each slipper occuping a predominant area of the slipper and leaving a continuous marginal surface for direct contact with said drive surface; and means defining a hydraulic pressure fluid supply passage in each piston and each piston rod providing communication between said cavity and the working space in the associated cylinder.

11. A motor according to claim 10, further comprising flow restricting means included in said means defining a hydraulic pressure fluid supply passage between said cavity and said working space.

12. A motor as claimed in claim 11, further comprising a filter included in said means defining a hydraulic pressure fluid supply passage between said cavity and said working space.

13. A multi-cylinder hydraulic motor in which a plurality of cylinders are brought sequentially into communication with hydraulic fluid pressure supply and discharge conduits through passages communicating with the supply and discharge conduits through a set of ports sequentially opened and closed by a control valve, the valve comprising a shaped cylindrical valve member mounted within a housing and formed with supply and return channels and balancing channels and circumferential grooves which receive sealing rings formed of resilient plastic, the circumferential grooves being positioned so that the rings define the axial limits of the effective areas on which the hydraulic forces act.

14. A motor as claimed in claim 13, in which at least one of said circumferential grooves is formed with at least one flute or the like which extends from the mouth of the groove to within a short distance from the base of the groove to provide a relief passage to prevent build-up of hydraulic pressure on the low-pressure side of the rings.

15. A motor as claimed in claim 14, in which said flutes are constituted by machining marks in the groove faces.

16. A motor as claimed in claim 13, in which at least one flute or the like is formed in at least one of said rings to provide a relief passage to prevent build-up of hydraulic pressure in the low-pressure side of the ring.

17. A motor as claimed in claim 13, in which the depth of penetration of the rings into the grooves is effectively greater than the width of the rings whereby the force exerted on the rings by hydraulic pressure in the axial direction is effectively greater than that exerted in the radial direction.

18. A motor as claimed in claim 13, in which two rings are provided in each groove, the double-ring width

being approximately equal to the depth of penetration of the rings into the groove.

19. A motor as claimed in claim 13, in which a piston is contained in each of said cylinders, each piston being provided with a composite ring formed of resilient material and comprising two co-operating ring parts which are formed with engaging frusto-conical faces which co-operate circumferentially so that, on axial compression of the composite ring, one of the parts expands diametrically and the other contracts diametrically.

20. A motor as claimed in claim 13, in which a piston is contained in each cylinder and each piston is connected to the small end of its connecting rod through a joint comprising a ball on the connecting rod and a piston socket which is so shaped as to provide ring contact between the ball and the socket.

21. A motor as claimed in claim 13, in which a piston is contained in each cylinder and each piston is provided with a ring groove formed with a frusto-conical sloping face, and a ring of resilient material formed with a similarly sloping face for engagement with the groove face is arranged in said groove, the sloping faces of the ring and groove coacting under hydraulic pressure to urge the ring to expand outwardly into engagement with the co-operating cylinder wall.

22. A motor as claimed in claim 21, in which said ring is of trapezoidal transverse section and the ring groove is of complementary shape.

23. A hydraulic control valve, comprising: a valve housing; a cylindrical valve member in said housing, said valve member and said housing being arranged for relative rotation therebetween; first and second ports formed in said housing, said ports being axially spaced with respect to said valve member, one of said ports being connectable to supply pressure fluid to said valve member, the other being connectable to discharge spent pressure fluid therefrom; first and second axially spaced circumferential fluid flow passages formed in said valve member and extending completely therearound, said first passage being in continuous communication with said first port and said second passage being in continuous communication with said second port in all positions of said relative rotation; a circularly arranged series of utilization ports formed in said housing and spaced circumferentially around said valve member the centers of said utilization ports lying in a common plane perpendicular to the axis of said valve member; first and second arcuate fluid flow passages formed in said valve member, said arcuate passages being circumferentially aligned and laterally radially outwardly open for registry successively with said utilization ports during the course of said relative rotation, said first arcuate passage being in continuous communication with said first circumferential passage and said second arcuate passage being in continuous communication with said second circumferential passage; and equalizing means by which the hydraulic fluid pressure in said first and second ports is utilized to balance forces due to said pressure acting on said valve member.

24. A valve according to claim 23, wherein said equalizing means comprises a pair of third arcuate passages formed in said valve member axially spaced from said second arcuate passage at opposite sides thereof and located circumferentially opposite said first arcuate passage, said third passages being in continuous communication with said first arcuate passage; and a fourth pair of arcuate passages formed in said valve member, said fourth passages being separate from said third passages and circumferentially aligned therewith, said fourth passages being located circumferentially opposite said second arcuate passage and in continuous communication therewith, said third and fourth passages being laterally radially outwardly open for communication with said housing for applying lateral pressures to said valve member which balance the pressures applied thereto by said first and second arcuate passages, respectively.

25. A valve according to claim 24, wherein said third and fourth arcuate passages are located axially outwardly of said second and first arcuate passages, respectively, and in which said first and second circumferential passages are located axially at opposite sides of said first and second arcuate passages intermediate said first and second arcuate passages and said third and fourth arcuate passages.

26. A valve according to claim 24, wherein said first and second circumferential passages are located axially at opposite sides of said first and second arcuate passages.

27. A valve according to claim 24, wherein said valve member has a fluid flow passage formed therein which extends between opposite end portions thereof.

28. A motor according to claim 11, wherein said flow restricting means comprises a removable member having a metering orifice formed therein, said removable member being positioned in said fluid supply passage in said piston rod for operation of said metering orifice as said flow restricting means.

29. A hydraulic motor comprising: a main body part defining a symmetrical set of cylinders radiating outwardly from a common chamber in said body part; a shaft extending through said chamber, the axes of all of said cylinders being perpendicular to the axis of said shaft, said shaft comprising an eccentric member in said chamber, said body part and said shaft being rotatable relative to each other; a piston in each cylinder, each piston being in continuous driving connection with said eccentric member; a circularly arranged series of working ports in said body part, the centers of said working ports lying in a common plane spaced from and parallel to the axes of said cylinders, said series of working ports being concentric with the axis of said shaft; a plurality of duct means each connecting one of said working ports individually with the working space within one of said cylinders; a cylindrical valve member in said body part coaxial with said shaft and connected thereto for said relative rotation therewith; supply and discharge ports formed in said body part, said supply and discharge ports being axially spaced with respect to said valve member, said supply and discharge ports being interchangeably connectable to supply pressure fluid to said valve member and to discharge spent pressure fluid therefrom for reversing the direction of said relative rotation; first and second axially spaced circumferential fluid flow passages formed in said valve member and extending completely therearound, said first passage being in continuous communication with said supply port and said second passage being in continuous communication with said discharge port in all positions of said relative rotation; first and second arcuate fluid flow passages formed in said valve member, said arcuate passages being circumferentially aligned and laterally radially outwardly open for registry successively with said working ports during the course of said relative rotation, said first arcuate passage being in continuous communication with said first circumferential passage and said second arcuate passage being in continuous communication with said second circumferential passage; a pair of third arcuate passages formed in said valve member axially spaced from said second arcuate passage at opposite sides thereof and located circumferentially opposite said first arcuate passage, said third passages being in continuous communication with said first arcuate passage; and a fourth pair of arcuate passages formed in said valve member, said fourth passages being separate from said third passages and circumferentially aligned therewith, said fourth passages being circumferentially opposite said second arcuate passage and in continuous communication therewith, said third and fourth passages being laterally radially outwardly open for communication with said body part for applying lateral pressures to said valve member which

balance the pressures applied thereto by said first and second arcuate passages, respectively.

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