Nov. 6, 1945. G. ROESSLER 2,388,644

HYDRAULIC PUMP, FLUID MOTOR OR COMPRESSOR

Filed May 8, 1943 8 Sheets-Sheet 1

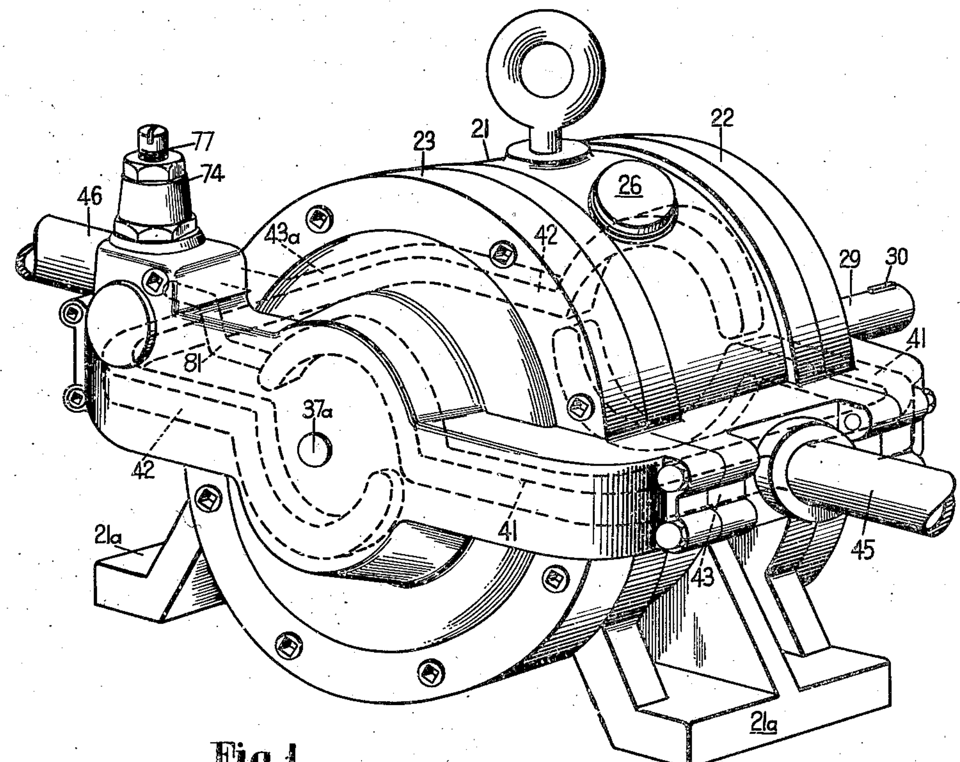

Fig. 1

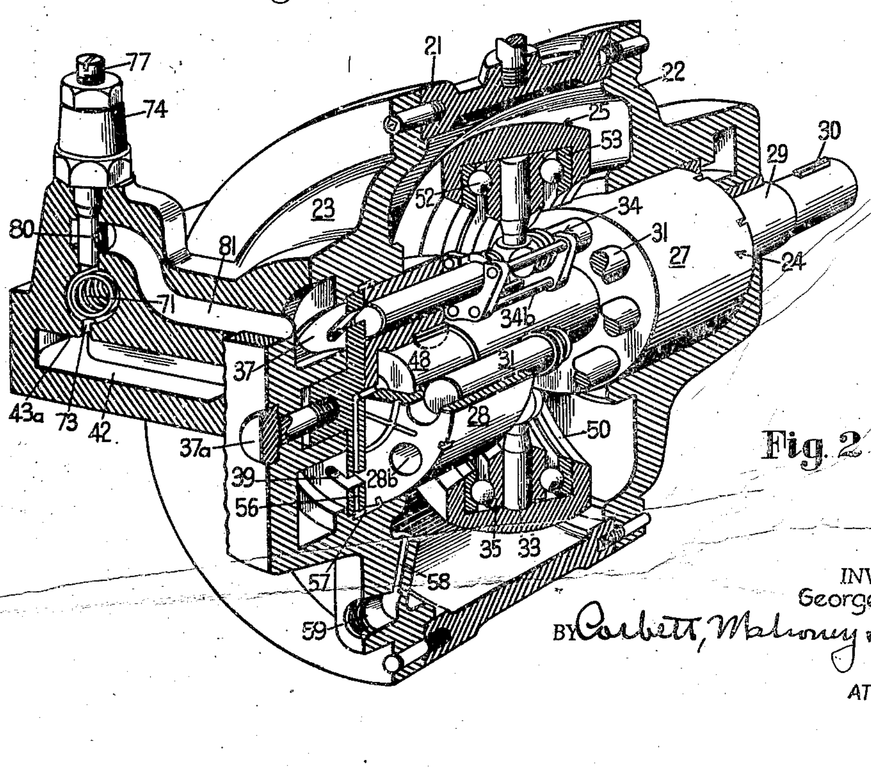

Fig. 2

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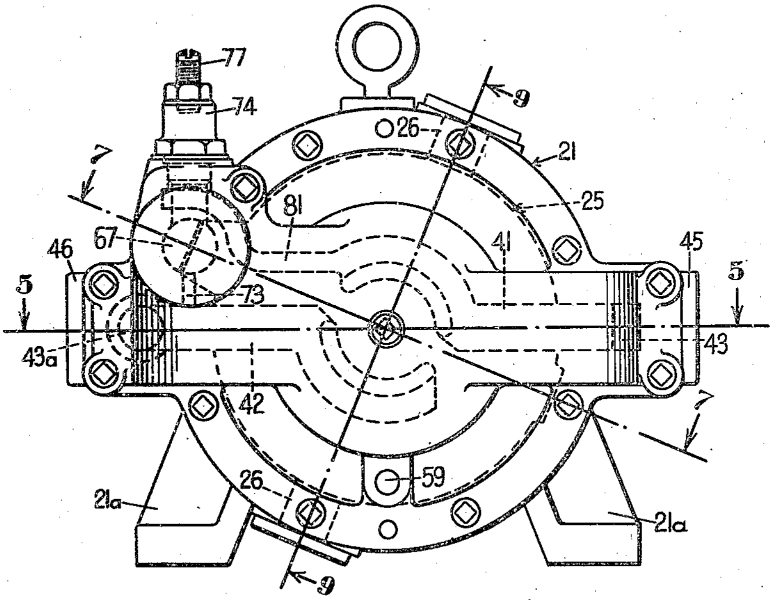

Fig. 3

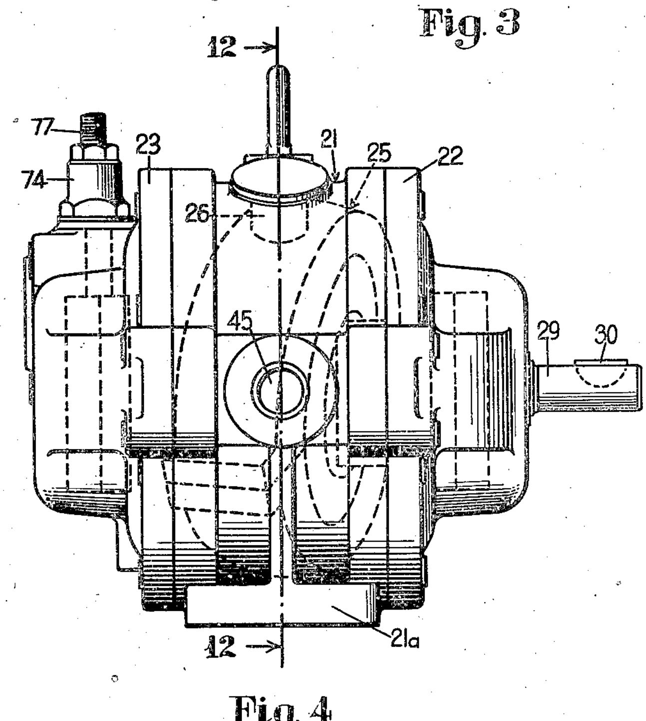

Fig. 4

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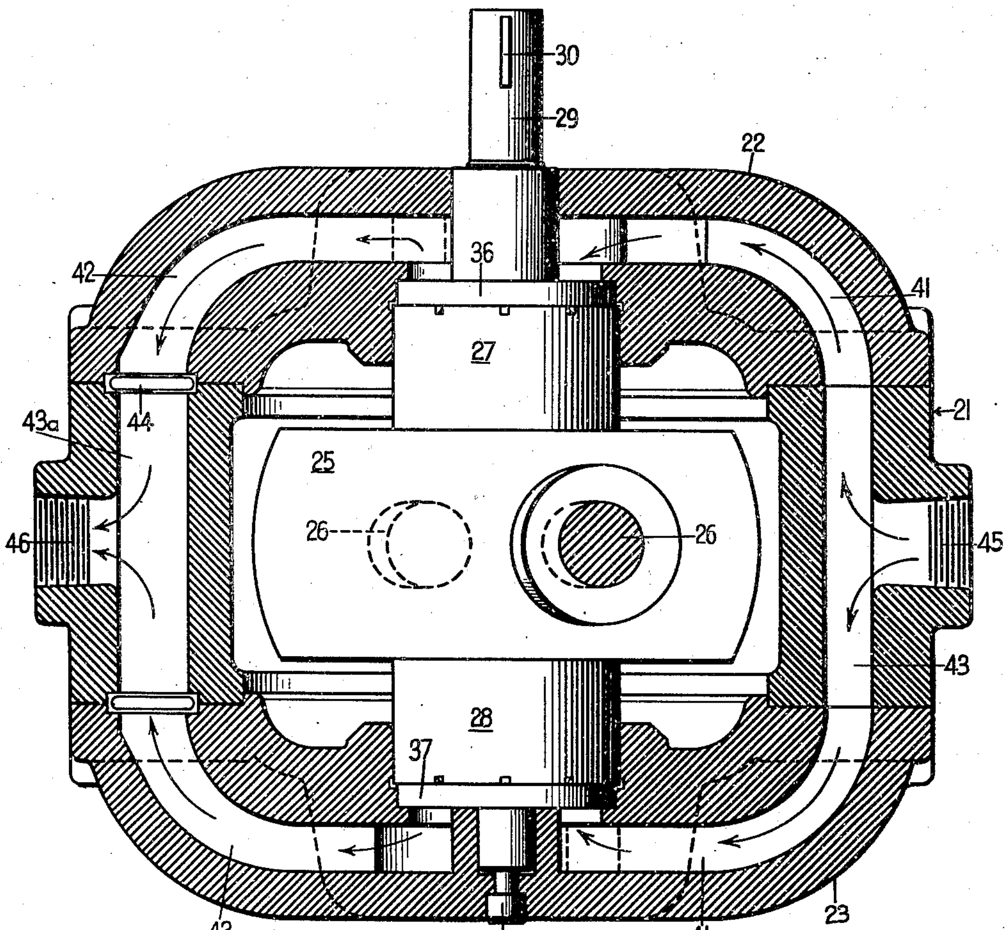

Fig. 5

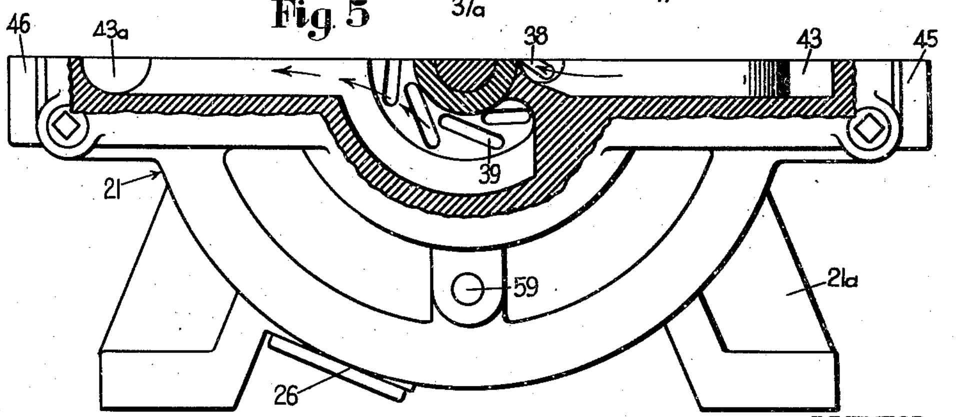

Fig. 6

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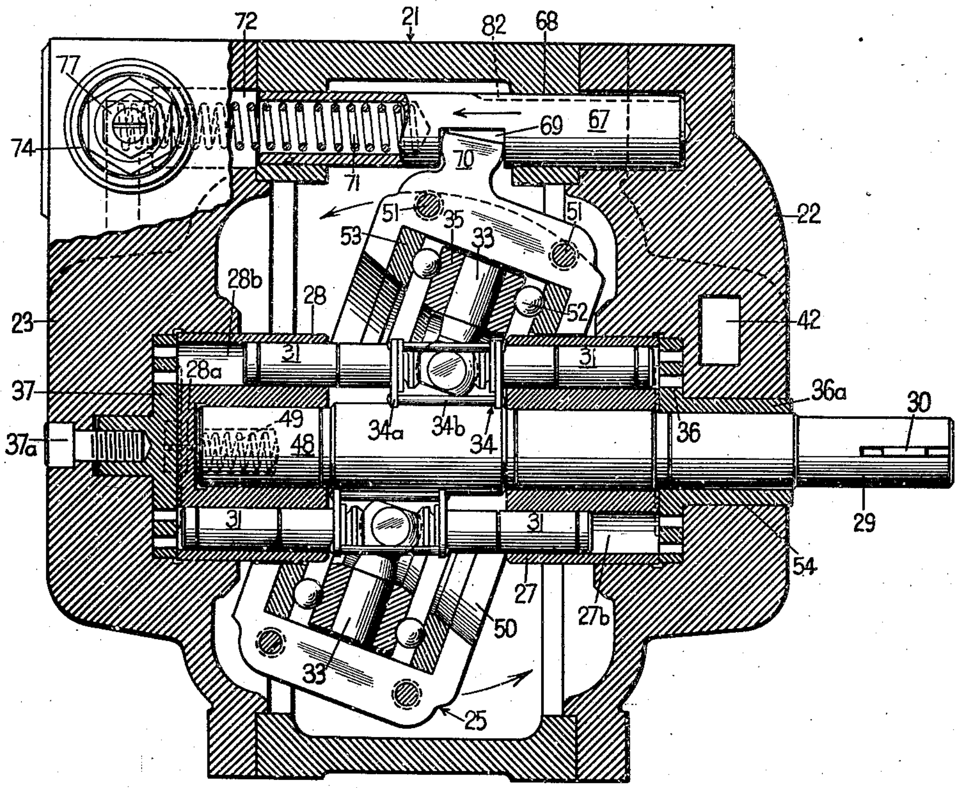

Fig. 7

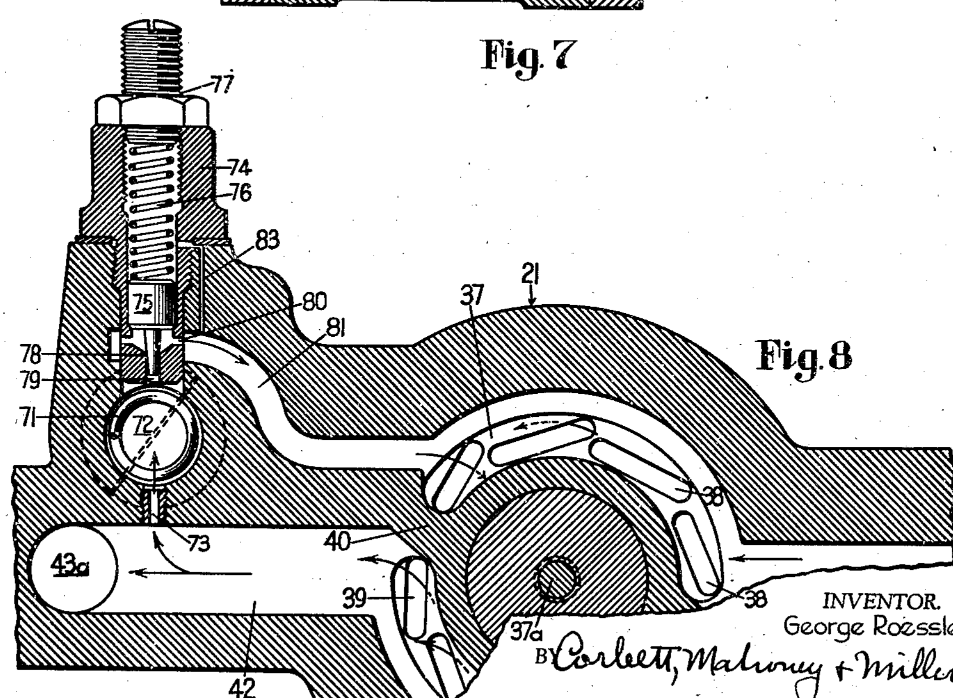

Fig. 8

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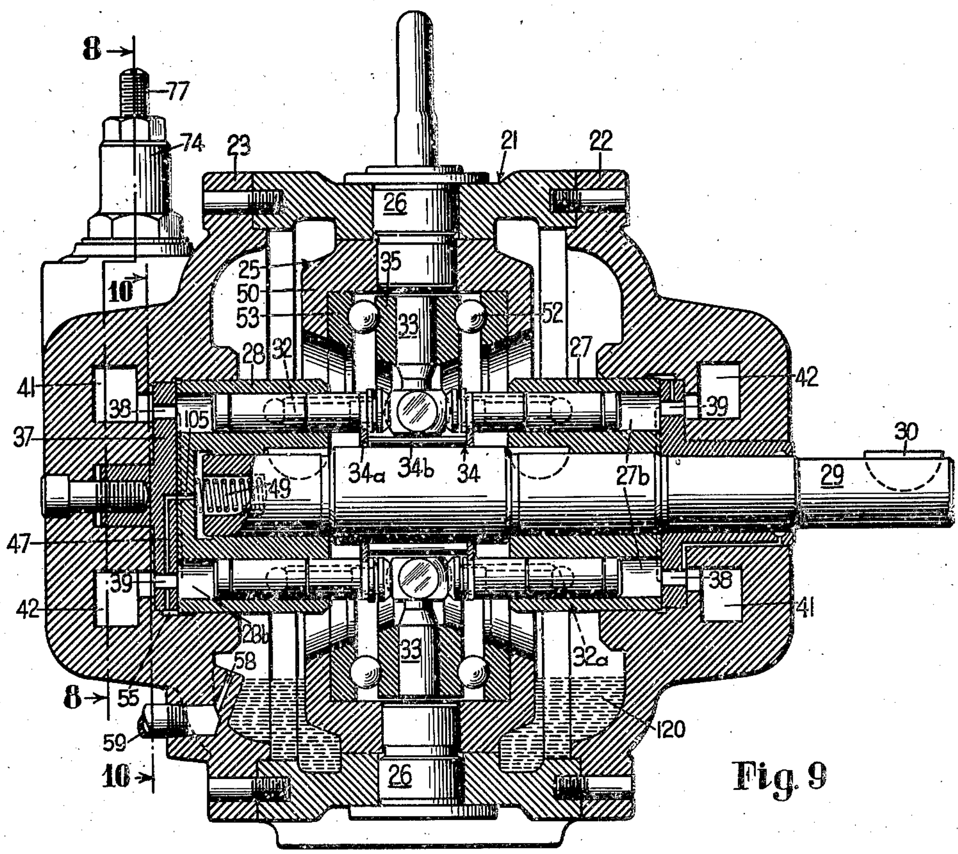

Fig. 9

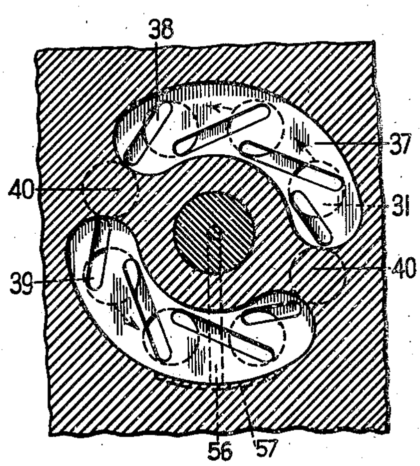

Fig. 10

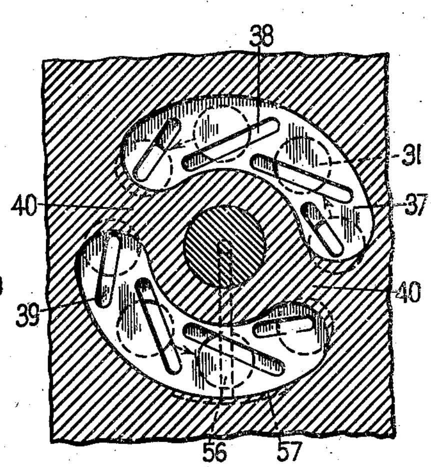

Fig. 11

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HYDRAULIC PUMP, FLUID MOTOR OR COMPRESSOR

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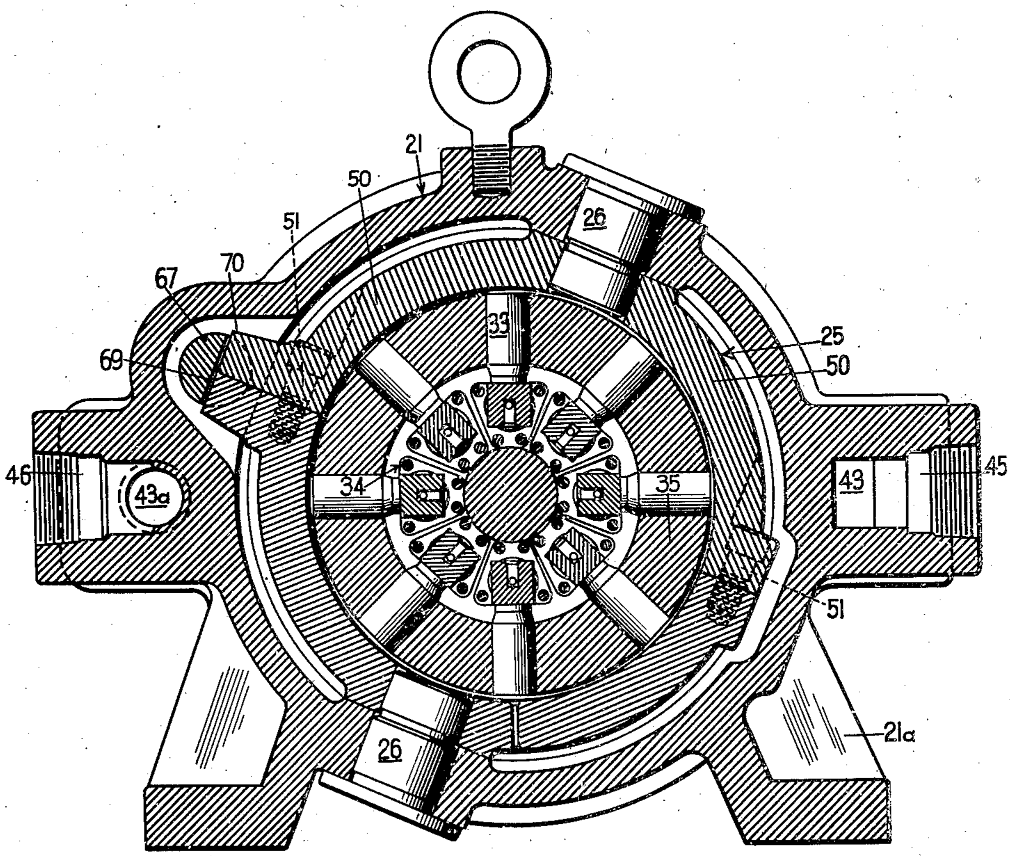

Fig. 12

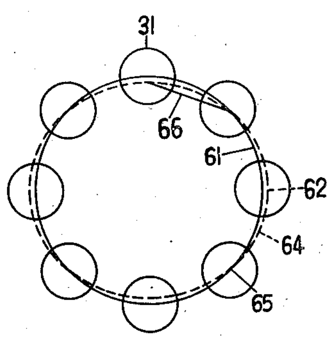

Fig. 13

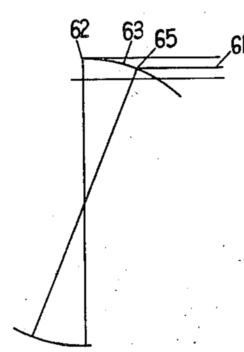

Fig. 14

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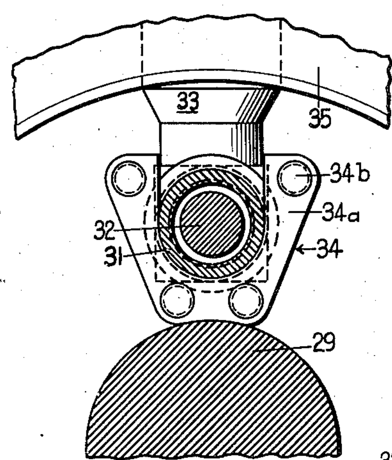

Fig. 15

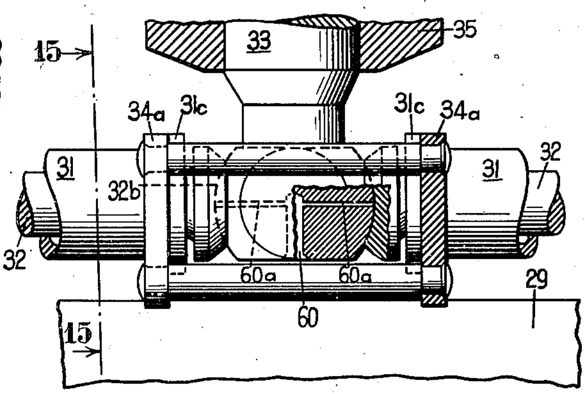

Fig. 16

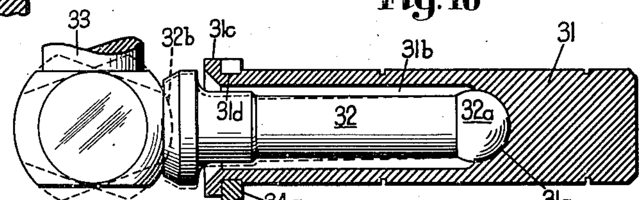

Fig. 17

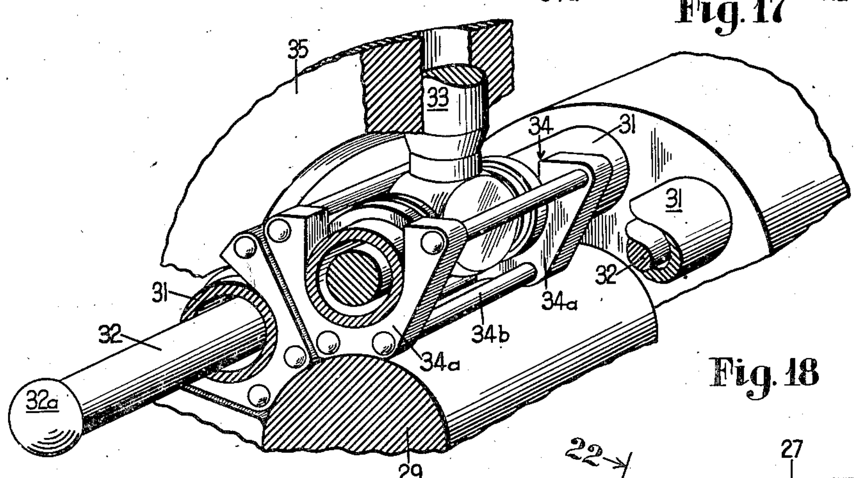

Fig. 18

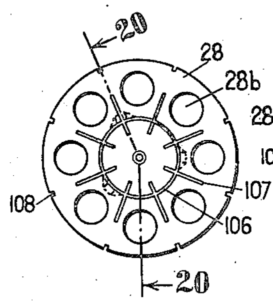

Fig. 19

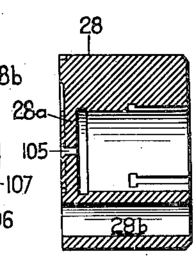

Fig. 20

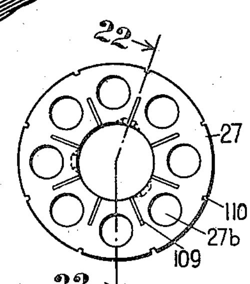

Fig. 21

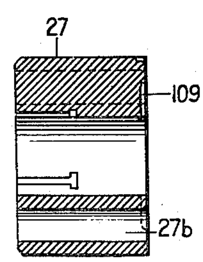

Fig. 22

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HYDRAULIC PUMP, FLUID MOTOR OR COMPRESSOR

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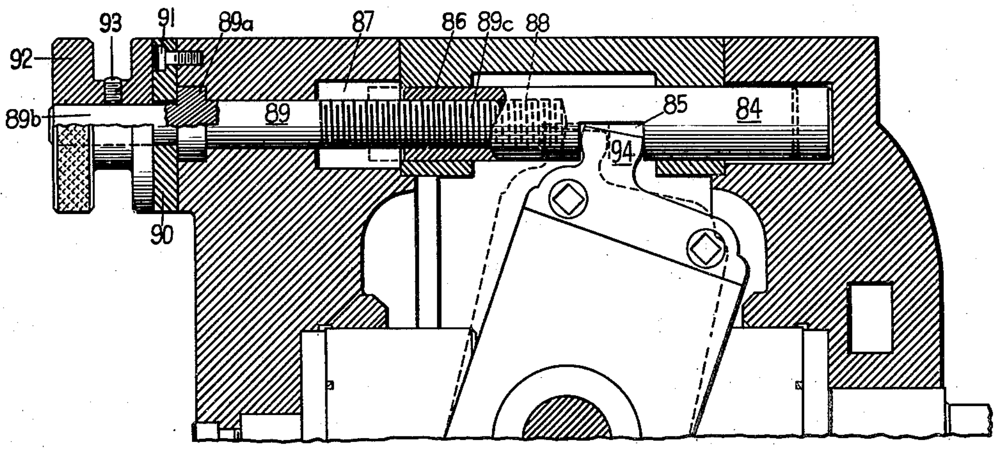

Fig. 23

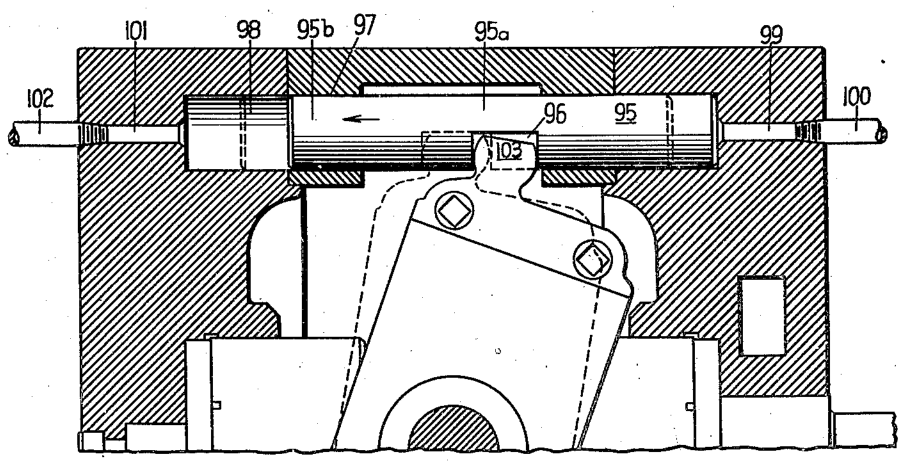

Fig. 24

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Patented Nov. 6, 1945 2,388,644

UNITED STATES PATENT OFFICE

2,388,644

HYDRAULIC PUMP, FLUID MOTOR, OR COMPRESSOR

George Roessler, Teaneck, N. J.

Application May 8, 1943, Serial No. 486,186

14 Claims. (Cl. 103—162)

My present invention relates to high speed and high pressure hydraulic pumps, fluid motors or compressors. It has to do with an improved high speed and pressure pump having a wide variety of uses which will operate effectively and efficiently with liquids, air or gases. Its field of use is practically unlimited and while it finds particular usefulness in connection with the aircraft and automotive fields, it is in no wise restricted to use in these fields. It is also specifically adaptable for use in connection with power presses, automatic machinery, pneumatic applications, liquefaction of gases, multi-stage compression, variable speed power transmission, remote controls of many types, in the refrigeration and air conditioning arts, and in marine applications, such as gun turret controls and submarines.

There is at this time a definite and obvious need, in what might be termed the high pressure division of the hydraulic field, for a satisfactory and efficient high pressure pump. With war requirements in the aviation branch pointing directly to the adoption of hydraulic transmissions and hydraulic revolution stabilizers, etc., there is a present and urgent need for a satisfactory new pump and fluid motor which has high qualifications as to strength, mechanical feasibility and simplicity of construction and production. To meet the requirements it is believed that such a pump should be capable of high speed to permit compactness; that it should be of such mechanical construction that it will operate at high speeds for a minimum of 250 hours under pressure loads of 3000 pounds; that it should be free from hydraulic lock and that the construction should have freedom of movement of the working parts so that it will be non-heating in operation and have a low initial power factor. It should have the facility for adjustment so that when a predetermined and selected pressure output is reached, it will cease to produce further output or consume power until the circuit pressure therein has been dissipated by useful work or otherwise has fallen below the pressure identified with the adjustment. It should have means for making volume adjustments to suit the exact circuit requirements according to a prescribed timing. It should have means for quick adjustment from high volume and low pressure, on the one hand, to low volume and high pressure on the other hand, and such means should be arranged so as to be responsive to regular circuit controls.

It is, therefore, one of the objects of my present invention to provide an improved high speed variable volume hydraulic pump having all of the foregoing features and characteristics which is capable of operating at high speeds and pressures for long periods of time without overheating; one which is of sturdy, compact and relatively simple construction, containing comparatively few working parts, and one which is capable of being manufactured by mass production methods.

Another object of my invention is to provide an improved pump of the foregoing character which is capable of volume adjustment to suit any particular circuit requirements; and one which is capable of quick adjusment from high volume and low pressure to low volume and high pressure.

Another object sought in the new and improved design embodying my invention is to provide an assembly and operating principle wherein the pressure-making members are opposed or double-acting, thus splitting the working load and back pressure and placing one half against the other to effect a counterbalance of the pressure.

A further object of the invention is to provide an improved hydraulic pump capable of being set to operate at a given or predetermined speed and pressure output and which will automatically maintain said speed and pressure output regardless of any variation in the speed of operation of the motor, engine, or other source of motive power unit to which the pump is drivingly connected.

Generally speaking and as shown in the drawings, by way of illustration and not of limitation, the improved high speed and pressure hydraulic pump, fluid motor, or compressor embodying my present invention consists of a housing or casing containing a rotor mounted on a shaft having an extension for connection to a source of power, the rotor including opposed cylinders and sets of pistons therein, a thrust assembly suspended within the housing and shiftable into an angular or tilted position, said assembly including a thrust ring or member which is rotatable with said rotor and being operatively connected with said pistons to reciprocate the same upon rotation of the rotor and the thrust ring or member, and a regulating device capable of being set or adjusted to automatically maintain the speed and pressure of the pump constant.

The foregoing and other objects and advantages of my invention will appear from the following description and appended claims when considered in connection with the accompanying drawings forming a part of this specification, wherein like reference characters designate corresponding parts in the several views.

In said drawings:

Figure 1 is a perspective view of my improved hydraulic pump, motor or compressor.

Figure 2 is a view similar to Figure 1, partly in section and partly broken away.

Figure 3 is an end elevational view of the pump embodying the invention.

Figure 4 is a side elevational view of my improved pump structure shown in the preceding views.

Figure 5 is an enlarged horizontal sectional view taken substantially along the line 5—5 of Figure 3, looking in the direction of the arrows.

Figure 6 is an enlarged elevational view, partly broken away, of the lower portion of the structure shown in Figure 3.

Figure 7 is an enlarged section taken substantially along the line 7—7 of Figure 3, looking in the direction of the arrows, and showing the parts in operative pumping position.

Figure 8 is an enlarged fragmentary vertical sectional view taken substantially along the line 8—8 of Figure 9, looking in the direction of the arrows.

Figure 9 is an enlarged section taken substantially along the line 9—9 of Figure 3, looking in the direction of the arrows and showing the parts in neutral position.

Figure 10 is an enlarged detail vertical section taken substantially along the line 10—10 of Figure 9, looking in the direction of the arrows.

Figure 11 is a view similar to Figure 10, showing the parts in different positions.

Figure 12 is an enlarged vertical sectional view taken substantially along the line 12—12 of Figure 4, looking in the direction of the arrows.

Figures 13 and 14 are diagrammatic views illustrating the kinematic system governing the actuating elements of the structure embodying the present invention.

Figure 15 is an enlarged detailed sectional view, partly in elevation, taken substantially along the line 15—15 of Figure 16, looking in the direction of the arrows.

Figure 16 is a side elevational view, partly in section, of one of the keepers or cages for maintaining a pair of opposed pistons and one of the ball pins in assembly.

Figure 17 is an enlarged detail sectional view, partly in elevation, of one of the pistons and ball pin assemblies of the invention.

Figure 18 is a fragmentary detail perspective view, partly in section and partly broken away, showing somewhat more in detail the structure disclosed in Figures 15, 16 and 17.

Figure 19 is an end elevational view of one of the cylinder shells of the rotor.

Figure 20 is a sectional view taken substantially along the line 20—20 of Figure 19, looking in the direction of the arrows.

Figure 21 is an end elevational view of the other cylinder shell of the rotor.

Figure 22 is a sectional view taken substantially along the line 22—22 of Figure 21, looking in the direction of the arrows.

Figure 23 is a fragmentary sectional view of a hydraulic pump embodying the invention, similar to that of the preceding views, but showing a modified form of regulating device or mechanism for the thrust bearing assembly; and

Figure 24 is a view similar to Figure 23 but showing a further modified form of piston and regulating means for the thrust bearing assembly, embodying the present invention.

Referring now to the drawings, and particularly to Figures 1 to 18 thereof, the improved hydraulic pump, fluid motor, or compressor of my present invention comprises, as best seen in Figures 1 and 3, a compact structure or unit which includes a central housing or casing 21 provided with a supporting base or feet 21a and end plates or closures 22 and 23. The housing and end plates support a rotor assembly shown as a whole at 24, Figure 5, with which is associated a thrust assembly or thrust bearing assembly shown as a whole at 25. The thrust bearing assembly is supported to permit turning movement about its axis by means of a pair of diametrically opposed trunnions 26, 26 which are mounted in the central housing 21.

The rotor 24 consists of a pair of cylinder shells 27 and 28 both of which are mounted, in tandem relation, upon and keyed to a shaft 29 journaled in the end plates or walls 22 and 23 and which has an extension projecting through and beyond the end plate 22, said extension carrying a key 30 permitting the shaft to be coupled to a suitable source of power, such as an engine or motor (not shown).

The cylinder shell 28, while being keyed to portion 48 of shaft 29 to prevent relative rotation of the shell and shaft, has a snug sliding fit on the shaft. Moreover, this shell 28 has a closed or solid end wall 28a and forms a hydraulic cylinder. A vent 47 through the end wall 28a and the disk valve 37 opens into the pressure port and provides pressure to said cylinder and over the end of the shaft portion 48. Since there are always four pistons on the pressure side which are subject to operating pressure, there is a tendency for this pressure to break through the seal between the cylinder shell and the valve 37. In other words, the back pressure tends to force the cylinder shell away from the valve and to increase the filament of lubricant between the valve and cylinder shell, in accordance with the viscosity of the lubricant, to an extent which might be considered leakage. By providing a shaft portion 48 of an area which is slightly greater than the combined area of four of the pistons to be described below, a hydraulic differential is provided which can be calculated and will provide the correct pressure of the cylinder shell 28 against the valve 37 to assure a seal under any conditions of circuit pressure or viscosity of hydraulic oil. This sealing pressure automatically increases as the circuit pressure increases and any wear on the faces of the disk valve 37 and the cylinder shell 28 is automatically compensated for. A compression spring 49 is provided to obtain the sealing effect at low pressures and before there is enough circuit pressure to make the cylinder hydraulically operative. The pressure of this spring becomes a negligible factor almost immediately after starting the pump in operation and this occurs at very low circuit pressures.

Each of the cylinder shells of the rotor is provided with a plurality of bores, eight such being shown in the present instance for each shell, which provide cylinders for sixteen reciprocable pistons 31. One of these pistons is clearly shown in Figure 17 of the drawings and, as shown, is provided with a spherical socket 31a formed in the inner end of the piston head which defines the inner end of a cylindrical opening or space formed in the piston. The lower or inner end of the piston is provided with an annular flange 31c and its outer periphery, adjacent the flange, with an annular groove 31d. The piston is provided with a universal type connecting rod 32 whose inner end 32a is ball shaped to fit the spherical socket 31a. The opposite end of the connecting or universal rod 32 is enlarged and is cupped to provide a socket 32b. The socket 32b is adapted to engage and form a seat or bearing for the spherical portion or end of a ball pin 33. It will be understood that the pistons 31 are arranged in pairs and in tandem relation in opposed pairs of cylinders formed in the cylinder shells 27 and 28.

Each pair of opposed pistons 31 is assembled with a ball pin 33. For this purpose, as shown, see particularly Figures 15, 16 and 18, I provide a so-called keeper or cage which is shown as a whole at 34. This cage comprises a pair of spaced substantially U-shaped members or yokes 34a which seat in the grooves 31d of the opposed piston and which embrace the flanges 31c formed on the pistons. The yoke members 34a are held together and in binding relation with the opposed pistons by means of a plurality of rods or pins 34b, four such being shown, whose outer ends are flattened or turned down to cause the yoke members to bind the pair of opposed pistons 31 and ball pin 33 together as a unit or assembly.

While I have shown and described the cylinder shells 27 and 28 as each containing eight pistons 31, it is to be understood that the shells may be formed to contain any desired number of pistons and that either an odd or even number of pistons may be located in each of said shells.

The thrust bearing assembly 25 includes, in addition to its non-rotatable outer ring-like portion, a thrust ring 35 which receives and supports the stems or shanks of the ball pins 33, one such ball pin being provided for each pair of pistons located in the shells 27 and 28.

Located in suitable sockets in the end plates 22 and 23 are disk valves 36 and 37, respectively. The disk valve 36 has a tubular hub portion 36a which extends through a bearing formed in the end plate 22 and which surrounds a portion or section of the shaft 29. The disk valve 37 is provided with a hub portion which is internally threaded to receive a cap screw or bolt 37a provided for the purpose of holding the valve 37 in proper position with relation to the rotor cylinder shell 28.

Each of the disk valves is provided with two sets of ports 38 and 39, ports 38 serving as intake ports and ports 39 as discharge ports, there being four such ports in each set, the two sets being separated by means of solid portions 40 which serve to separate the pressure of exhaust from the negative pressure of intake and provide a cut-off between these two at the face of the valve and as the cylinder shells 27 and 28 revolve. The valve ports 38 and 39 register with the cored passages 41 and 42 formed in the end plates 22 and 23, which passages are in communication with the cored passages 43 and 43a formed in the central housing 21 adjacent and communicating with the inlet or intake opening 45 on the one hand and the discharge or outlet opening 46 on the other hand, see particularly Figure 5 of the drawings. As seen in this figure, all of the joints between the casing sections or portions in the region of these passages are sealed by means of suitable annular pressure seals of conventional nature, as shown at 44.

Since the inner end portion of the intake passage 41 formed in the end plate 22 extends below the shaft 29 and the opposite end portion thereof located in end plate 23 extends above or over said shaft, and since the inner end portion of the discharge passage 42 located in the end plate 22 extends over the shaft 29, with its opposite end portion formed in end plate 23 extending below or beneath shaft 29, it will be understood that the ports or slots 38 and 39 formed in the disk valves 36 and 37 must be arranged or located accordingly so that they will register properly with the passages 41 and 42. Thus, the valve 36 has its set of four discharge ports 39 located in the upper half of the valve to register with the passage 42, whereas the set of four intake ports 38 is located in the lower half of the valve 36. At the opposite end the sets of ports in valve 37 are reversed. The set of four discharge ports 39 is located in the lower half of valve 37, whereas the set of four inlet or intake ports 38 is located in the upper half of said valve. The inner adjacent ends of the passages 41 and 42 are not connected but are in communication with one another through the medium of the valves 36 and 37 and the various pistons 31.

By providing, as shown, four intake ports or slots 38 and four outlet or discharge ports or slots 39 in each of the disk valves, instead of employing a single continuous intake slot and a single continuous outlet slot, the valve has greater strength than if two continuous slots were used. The effect is the same with the two sets of four slots, which, see particularly Figures 10 and 11, are staggered and overlap, so to speak, as if two separate and continuous slots or ports were employed.

Referring particularly to Figure 9, it will be seen that a vent or passageway 47 is provided in the disk valve 37 and communicates with a port or opening 105 through the cylinder shell 28. The portion 48 of the shaft 29, as seen in Figures 7 and 9, and as stated above, has an area which is slightly greater than the combined area of four of the pistons 31 in the cylinder shell 28. The compression spring 49 is located in a cylindrical recess formed in the shaft portion 48 for the purpose of forcing the cylinder shell against the face of the disk valve 37 to thus provide a sealing effect at low pressures.

Referring now to Figure 12 of the drawings it will be seen that the thrust bearing assembly is provided with a split yoke or housing 50, the two halves of which are held together by screws or bolts 51.

With particular reference to Figures 7 and 9, it will be seen that the split yoke or housing 50 of the thrust bearing assembly is provided with a pair of hardened steel ball races 52 and with a ball bearing assembly 53 with which the thrust ring 35 is in engagement and that the portions or surfaces 54 of the end plates 22 and 23 provide bearings for the rotor assembly 24, these bearings being lubricated at points 55 by means of lubricant supplied through the lubricating vents 56 from the pressure port. At point 57 there is provided a hydraulic cushion which is fed lubricants by the vent 56 to provide a counter-balancing force, see also Figures 10 and 11 which clearly show the location of the lubricating vent 56 and the space or recess where the hydraulic cushion 57 is created. In Figure 9 there is shown a drain orifice 58 communicating with an oil drain pipe 59 provided for the purpose of maintaining a constant oil level within the basin or lubricant containing reservoir 120 of the pump. Splash lubrication is provided for the internal working mechanism of my pump. The oil supply could be introduced through a suitable oil hole provided for that purpose but ordinarily the pressure seepage within the pump and from the circuit is relied upon to fill the oil basin at the bottom of the housing 21.

As best seen in Figure 16, each of the ball pins 33 is drilled at 60. This drilled hole 60 is in communication with a pair of lateral orifices or ducts 60a which extend through the ball pin and supply oil collected by the hole 60 to the surface of the ball which seats in the sockets 32b formed in the inner ends of the connecting rods 32, thus lubricating the contacting surfaces of the ball and the sockets in the opposed connecting rods 32. The centrifugal force of rotation of the parts furnishes the pressure.

Referring now particularly to the diagrammatic views in Figures 13 and 14 in which there is illustrated the kinematic system governing the actuating elements of the device, the circular orbit of the pistons is shown at 61. The focal point of the ball pins 33 is shown at 62 and this focal point describes the arc 63 when the thrust assembly 25 and the ball pins are swiveled on the trunnions 26. The focal point 62 is placed somewhat beyond the circular orbit 61 of the pistons so that when the angular position of the focal point orbit is projected as shown at 64, the orbit becomes elliptical and has both major and minor axes. At the axes the focal points travel above and below the circular orbit of the pistons and the distances above and below said orbit are the same. At intermediate points such as at 65, the orbits cross each other and the focal points of the ball pins together with the ball pins 33 are in exact alignment with the pistons.

The distance between the focal points as measured along the chord 66 is fixed and invariable. Since, however, the rotation is defined by and confined to an elliptic orbit, when they are at or near the minor axis, the chord 66 is less than the similar chord as measured between the centers of the pistons. The ball pin chord, at or near the major axis, is greater than the piston chord. Therefore, when the ball pins and pistons are rotating on their respective orbits as described, there is a sidewise movement of the ball pins relative to the pistons. The movement of the focal points above and below the circular orbit of the pistons and the relative sidewise movement are maximum at the major and minor axes, but at an intermediate position as shown at 65, the points coincide with the circular orbit of the pistons. These movements can be so calculated that the opening 31b in the piston 31 and the diameter of the connecting rod 32 can be made to provide a bearing contact which will drive the thrust ring 35 at the four places on the orbit as located by the axes. As the focal points progress around the orbit to a point midway between the intermediate point and the axis, one set of four bearing contacts give up their driving function to the other set of four bearing contacts and this occurs eight times during one revolution of the parts about the orbit, thus conforming to a mechanical principle which I might term, for convenience, an improved universal joint.

Referring again particularly to Figure 7 of the drawings and with particular reference to the regulating device or mechanism of my improved pump structure, there is shown a piston 67 which is adapted to have reciprocating movement in bored holes in the central housing 21 and the end plates 22 and 23. The piston 67 has a pressure-tight fit in the portion of the bore designated by the reference character 68. The piston 67 is provided with a notch or cut-out portion 69 which is adapted to receive a projection or lug 70 formed on the yoke member 50 of the thrust bearing assembly 25. By providing such means, reciprocating movement of the piston 67 will cause movement of the thrust bearing assembly 25 resulting in a change of its angular position about the trunnions 26. One end of the piston 67 is socketed to receive and support a compression spring 71 whose tendency is to move the thrust bearing assembly to its angular position as shown in Figure 7. The end plate 23 is provided with a cylinder cavity or chamber 72 which serves to provide hydraulic pressure to the piston 67. By reference to Figures 2 and 8 an orifice 73 is provided which connects the cylinder cavity 72 with the cored pressure passage 42 so that the cylinder cavity can be and is supplied with circuit pressure.

The particular improved type of regulator device or valve structure embodying my invention is perhaps best shown in Figure 8. The valve structure, as shown, comprises a body portion 74, needle valve 75 and a compression spring 76 which engages at its lower end the valve 75 and at its upper end, an adjusting screw and lock nut 77, the spring tending to normally maintain the valve upon its seat 78 to close the passage 79 from the valve seat. The lower end of the body portion 74 is provided with vents 80 which communicate with a cored passage 81. When the needle valve is on its seat 78 the orifice 79 is seated but when sufficient hydraulic pressure is exerted in the lower part of the orifice or passage 79 to compress the spring 76, the needle valve is lifted from its seat to permit hydraulic pressure to escape through the vents 80, cored passage 81 and back through the intake port. The escapement orifice thus becomes a metering or measuring orifice with high pressures serving to compress the spring to a greater extent, lifting the needle valve further away from its sealing position and permitting a greater flow of escapement. When this escapement flow is greater than the incoming flow through the orifice 73, the pressure in the cylinder 72 (Figure 7) becomes dissipated and the piston 67 will move toward the left of Figure 7 and into the cylinder cavity.

It will be understood that the thrust assembly 25 swivels freely on its supporting trunnions 26 and that the thrust assembly must be held in an angular position with relation to the axis of the rotor 24 such, for example, as that shown in Figure 7, to produce an effective pumping stroke. It is to be further understood that when the thrust assembly 25 is in a central or neutral position, see Figure 9, there is no pumping stroke and therefore there cannot be any hydraulic output by the pump. Furthermore, when the full energy of hydraulic back pressure as related to the area of eight pistons is applied according to the principles of force resolution and this energy tends to turn or swivel the thrust assembly 25 to the neutral position, the calculated tendency for the thrust assembly to return to the neutral position finds its countering force in the hydraulic effort which issues from the cylinder cavity 72 and an overbalance of hydraulic effort is provided to hold the angular position in accordance with the functioning of the needle valve 75 of the regulator device.

The piston 67 has a calculated area which provides a force which is slightly greater than the calculated force of the thrust assembly 25 which tends to return said assembly to the neutral position. When there is no circuit pressure within the cored passages, the compression spring 71 moves the thrust assembly 25 to maximum pumping position as seen in Figure 7 and when there is circuit pressure the hydraulic effort supplied by the cylinder cavity 72 is always greater than the force which tends to move the thrust assembly to a neutral position. When, however, the circuit pressure becomes great enough to raise the needle valve 75 off its seat to the extent which permit the metering off of a flow equal to or greater than the flow through the inlet passage or orifice 73, then the pressure in the cylinder cavity 72 drops and the thrust assembly 25 moves toward its neutral position (Figure 9) thus reducing the hydraulic output to the circuit. As this hydraulic output is thus being reduced, the circuit pressure is likewise being reduced correspondingly and the needle valve 75 is revising the differential flow through the orifices and the consequent pressure in the cylinder cavity 72. These reactions follow in rapid succession. Thus, the regulator valve measures the circuit pressure and regulates the hydraulic output of the pump so that it never exceeds, within certain operational limitations, the maximum pressure which has been determined by the strength of the valve spring 76. The strength or power of this spring can be adjusted within a wide range by virtue of the provision of the adjusting screw and lock nut 77.

As seen in Figure 7, the head portion of the piston 67 is provided with a longitudinal slot or groove 82 in its periphery to dispose of the piston effect. Moreover, see Figure 8, a communicating vent or passageway 83 is provided to assure freedom from hydraulic lock which might be caused by pressure seepage past the needle valve 75.

In accordance with my invention I have provided means to supply and maintain a film of lubricant between the adjacent faces of the cylinder shells 27 and 28 and the valves 36 and 37, while at the same time preventing the film from becoming, during rotation of the parts, a so-called pressure break-out. Such means are best shown in Figures 19 to 22, inclusive, of the drawings. In Figures 19 and 20, the end wall 28a of the pressure cylinder 28 is provided with a centrally disposed port or opening 105 which communicates with the passage 47, see Figures 7 and 9. The face of the pressure cylinder 28, as shown, is provided with an annular groove 106 and a series or plurality of intersecting and intercommunicating radial slots or grooves 107. The slots or grooves 107, as shown, terminate at their outer ends at points between and opposite the approximate centers of the cylinder cavities or piston chambers 28b. The face of the cylinder shell 28 is also provided with a series of relatively small notches or grooves 108 which are located around the peripheral edge of the cylinder shell face and are diametrically disposed to and spaced from the outer ends of the radial grooves 107.

The face of the cylinder shell 27 is provided with a series or plurality of radial grooves or slots 109, similar to the grooves 107 but which, as shown, are somewhat shorter than the grooves 107. These grooves 109 are located between the cylinders or piston chambers 27b and their outer ends terminate at points between and opposite the approximate centers of said chambers. The face of the cylinder shell 27 is also provided with a plurality of relatively small notches or grooves 110, corresponding in number to the slots 109 and being located at the peripheral edge of the face at points diametrically opposite and spaced from the radial slots 109.

As the cylinder shells or cylinders 27 and 28 rotate over and in contact with the adjacent faces of the valves 36 and 37, the grooves 109 and 107 carry the film of oil and wipe it across the adjacent faces of the cylinder shells and valves to properly lubricate them. The intake or suction side is supplied with the necessary film of lubricant and this film is wiped into the suction cylinder through the suction or intake valve ports and returns to the hydraulic circuit. The pressure within the grooves 106, 107 and 109 is considerably less than the circuit pressure of the pump but is sufficient to adequately lubricate the suction or intake side. The pressure within these grooves is a neutral pressure. Thus, with one of the grooves 107 or 109 containing lubricant having a neutral pressure sweeping, during rotation, after each pressurized cylinder or piston cavity 27b or 28b, the excess film pressure, which might normally build up and separate the face of the cylinder shell from its seat against the valve, is dissipated and the lubricating grooves or slots on the suction side are automatically provided with enough pressure to insure a lubricating film on that side. By virtue of my improved means or method of lubricating the contacting faces or surfaces referred to above, the lubricant is maintained in circulation and thus a fresh film of lubricant will be available at all times. Such method or means reduces friction to a minimum, minimizes wear, and provides for substantially noiseless operation.

In Figure 23 of the drawings, I have shown a hydraulic pump structure which is identical in all respects to the improved pump of the preceding views with the exception of the fact that in the present form I do not employ the automatically operable needle valve regulating device of Figure 8. In lieu thereof I have provided a piston 84 having a notch 85 formed therein and with its skirt portion having a sliding fit at 86 with the bore formed in the central housing 21. The end plate 23 is provided with a cylinder cavity 87 to provide a recess for the piston to move into. The piston is provided with an internally threaded socket or bore 88 to receive the threaded end 89c of an adjusting screw 89. The screw has a thrust collar 89a and a shank or outer end portion 89b. A keeper plate 90 is adapted to be slipped over the end portion 89b of the shaft and engages the thrust collar 89a to hold the screw in place. The keeper plate is held in place on the member 23 preferably by means of several screws or bolts 91, only one being shown. A knurled knob or handle 92 is pushed onto the shank portion 89b and held in place by means of a screw 93. The thrust bearing assembly 25 is provided with a projection or lug 94 which engages the notch or cut-out portion 85 of the piston. It will be seen that by turning the handle or knurled knob 92 in either direction, the threaded end 89c of the adjusting screw will move or reciprocate the piston 84 in the bore formed in the central housing and the end plates. By so adjusting the piston the angular position of the thrust bearing assembly 25 may be changed to increase or decrease the amount of pressure and consequent output of the pump. Adjustment is entirely by manual control, provides selective volume output, and can be changed at any time during operation to meet precision requirements. Such change or adjustment will increase or decrease the power ratio from a theoretically infinite position near the neutral position of the thrust bearing assembly to the minimum at approximately a 20 degree position.

In Figure 24 I have shown a further modified form of hydraulic pump embodying my invention. In this form of the invention I have provided a piston shown as a whole at 95 which, at its head portion, is substantially like the piston 84 of Figure 23. The left end portion of the piston 95, as viewed in Figure 24, comprises, in effect, a dashpot 95b, the head 95 and the dashpot portion 95b of the piston being interconnected by a central portion 95a which provides, in effect, an integral connecting rod for these portions. The piston at about its central point is provided with a notch or cut-out portion 96 to receive a projection or lug 103 formed on the thrust bearing assembly 25. The dashpot portion or end section 95b of the piston has a pressure-tight sliding fit with the bearing portion 97 formed in the central housing 21 and beyond this portion of the piston the end plate 23 is provided with a cylinder cavity 98, similar to the cavities 72 and 87 of the preceding forms of my invention.

The end plate 22 is provided with a cap bore 99 to receive a pipeline or pilot line 100. The bore 99 communicates with the head of the piston 95. The end plate 23 is likewise provided with a tapped bore 101 which communicates with the cylinder cavity 98 and to which is connected a pipeline or pilot line 102. It will be understood that the device as constructed in accordance with the present form of my invention may operate as a two-stage pump, having high volume-low pressure for one stage and high pressure-low volume for the other stage. These stages can be automatically selected by a suitable valve arrangement in the circuit. In the present form the orifice or passage 73 of the form of my invention shown in Figures 1 to 18, inclusive, and the automatic pressure regulator of Figure 8, are not used. The end portion 95b of the piston 95 may be designed, if desired, so as to serve as a dashpot for cushioning any hammering effect which might occur at the end of the piston stroke. With the exception of the exclusion of the orifice 73 and the automatic pressure regulator, as best seen in Figure 8, and the redesigned piston structure 95, the pump of Figure 24 is substantially identical with that shown in Figures 1 to 18, inclusive.

The improved pump in its several forms as described above, including its principle of operation, its mechanical movements, elements, devices and assemblies, can be used as a fluid motor by connecting it to a pressure circuit and it will rotate in either direction in accordance with the direction of flow of the circuit. Such a fluid motor can be regulated as to its revolutions per minute by throttling the flow or, in other words, by metering the flow of fluid to the motor. Moreover, by the application of pressure and flow, the pump of my invention can be used as an air motor and regulated as to pressure and output as desired.

Pumps made in accordance with my invention can be manufactured from various materials such as those which are non-corrosive, rust-proof and acid-resistant and can be used for pumping any liquids which it appears desirable to pump. In addition, by adding a lubricating circuit, the pump can be used for the compressing of gases. Thus, it will be seen that my improved pump is adapted for various and diversified uses.

It is believed that the failure in operation of previously known pumps has been due largely to misconceptions and errors in design particularly with respect to the fluid circuits of these pumps. These errors in design, the use of faulty and improper accessories, the presence of hydraulic lock, vibration and chatter were reflected in the pump, since in an assembly or group of mechanical apparatus or units, the pump is the unit which receives the application of the power in the form of shock or hydraulic hammer. With my improved pump structure including the automatic pressure-limiting device, this shock or hammer factor is substantially reduced, if not entirely eliminated. Furthermore, the variable volume requirement of any given cycle of circuit operation will be translated precisely to the power source instead of building up a pressure to break through a relief valve which must always be adjusted to exceed the maximum pressure requirement. In a pump structure provided with a relief valve, the maximum power requirement occurs when the circuit is idle.

Operation

The operation of my improved hydraulic pump as shown in Figures 1 to 18, inclusive, of the drawings is substantially as follows:

Let us assume that the pump structure is to be used to deliver a constant supply of fluid, such as oil, under pressure to one or a plurality of fluid transmissions, or the like, or for any other use wherein it is desirable to maintain a constant supply or volume and a given pressure of oil or other fluid. Let us assume also that the exposed end of the shaft 29 is keyed or otherwise drivingly connected to a source of motive power, such as an internal combustion engine, electric motor, or the like (not shown).

The working parts of the hydraulic pump, as shown in Figure 7, are in their idle position prior to the application of power to the shaft 29. When power is applied to the shaft 29 the thrust assembly 25 is held in its tilted or angular position by virtue of the force of the spring 71 which maintains the piston 67 to the right of Figure 7, the notch 69 in the piston holding the lug or projection 70 of the thrust assembly in the position shown in this figure. Rotation of the shaft will likewise cause rotation of the rotor 24 and the simultaneous rotation of the thrust ring 35 and ball pins 33 of the thrust assembly. It will be understood that the rotatable parts of the thrust assembly will rotate as a unit with the rotor and by reason of the fact that each pair of opposed pistons 31 of the rotor, through the connecting rods 32, is directly connected with a ball pin 33 by means of the cage or keeper 34, the pistons will be caused to reciprocate in their respective cylinders 27 and 28 as the rotor revolves.

There are eight of the pistons 31 located in each of the cylinder shells 27 and 28. Each pair of opposed pistons reciprocates or operates as a unit so that while piston 31 in cylinder shell 27 is performing its suction stroke, the other piston 31 of the pair located in cylinder shell 28 is performing its discharge stroke. The pairs of opposed pistons follow each other in action and operation in rapid succession and continuously. There are always four pistons in one of the cylinder shells operating at various stages of their pressure stroke and four opposed pistons in the other cylinder shell operating at various stages of their suction stroke. It will be understood that the reciprocation of the pairs of pistons as described occurs while the entire rotor is revolving with the shaft 29. While the rotor is revolving certain of said pistons function to draw or suck oil or other fluid in through the intake opening 45 in the casing, through the cored passages 43 and 41 and also through the intake ports 38 in each of the disk valves 36 and 37. At the same time that this action is taking place other pistons in each of the cylinder shells 27 and 28 are performing their discharge strokes to force the oil or other fluid through the ports 39 in both of the disk valves 36 and 37 and discharge it through the cored passages 42 and 43 and the discharge opening 46 located in the opposite side of the casing or housing. It will be understood that the pressure sides of the rotor are catawampous.

The disk valves 36 and 37 remain stationary during the operation of the pump and their separate sets of four intake and four exhaust ports 38 and 39, respectively, remain stationary whereas the various pistons during their rotation alternate in performing their intake and discharge strokes.

As the rotor 24 and the thrust ring 35 and ball pins 33 rotate and cause the pistons 31 to build up pressure, the tendency of the thrust assembly is to assume or move toward its vertical position as shown in Figure 9. This is resisted, however, by the oil pressure in the cylinder cavity 72 at the end of piston 67. This cylinder cavity is in direct communication through an orifice or passageway 73 (see Figures 2 and 8) with the oil in the cored discharge passage 42.

So as to be able to adjust the hydraulic pump in such a manner as to regulate its pressure output, I have provided the regulator device previously described which includes the spring-pressed needle valve 75 and associated parts, as best seen in Figure 8. The valve spring 76 may be adjusted to apply any desired force to maintain the needle valve 75 on its seat 78 through the medium of the adjustment screw and nut 77 located at the top of the valve body 74. For example, if it is desirable to discharge oil through the discharge opening 46 at a pressure of say 3000 pounds per square inch and to maintain this pressure, the spring 76 is adjusted so as to cause the needle valve 75 to remain upon its seat against the action of the pressure within the cylinder cavity 72 for all pressures up to 3000 pounds p. s. i. If, however, the pressure in the discharge passages 42 exceeds 3000 pounds, p. s. i., the valve 75 is forced or lifted from its seat against the action of the spring 76 which permits the escape of pressure through the vents 80 and cored passage 81 back to the intake passage 41. When this escapement through the metering needle valve 75 exceeds the flow of fluid through the orifice 73, the pressure in the cylinder cavity is reduced permitting the thrust assembly 25 to move gradually in the direction toward its vertical position against the action of spring 71 to slow down the pumping action and allow the pressure to be reduced to the predetermined maximum of 3000 pounds per square inch for which the regulating device has been adjusted.

It will be understood that the needle valve 75 meters the flow of fluid, that the flow through the valve 75 and the orifice 73 is on a differential basis and that the pressure, therefore, in the cylinder cavity 72 is constantly rising and falling to accomplish a precise regulation of the pump. As a result of the action of the needle valve the force tending to drive the thrust assembly 25 to neutral or vertical position and the force of the piston 67 are being constantly regulated hydraulically to produce a volume which, in turn, will hold to a norm of pressure as determined by the needle valve 75.

It will be understood that movement of the thrust assembly from its 20 degree angular position toward its inactive or neutral vertical position gradually reduces the pressure output of the pump. The tendency of the spring 71 is to move the thrust assembly 25 toward its angular or working position at all times and serves to counteract the tendency of the thrust assembly to move or swing back, due to the back pressure on the pistons, toward its vertical or inactive position. To assist the spring 71 to counteract this tendency on the part of the assembly 25, the pressure in the cylinder cavity 72 acts upon the piston 67 to force it toward the right of Figure 7 and maintain the thrust assembly in its proper angular or pumping position in accordance with the degree of pressure for which the pump is adjusted as determined by the pressure or force exerted on the needle valve 75 by its spring 76.

After the pressure in the discharge passages 42 has been reduced to 3000 pounds p. s. i. or less, the force or power of the spring 76 will return the valve 75 to its seat, close passage 79 and allow the pressure in cylinder cavity 72 to build up to normal, for example 3000 pounds p. s. i.

If at any time the thrust assembly should assume its vertical position as shown in Figure 9, no pumping action of the pistons 31 will occur even though the source of motive power continues to drive the rotor shaft 29.

The hydraulic pump of my invention is designed to deliver a steady or constant output or flow of fluid at a given or predetermined pressure regardless of the speed at which the engine, motor, or the like which drives it, is operating. The regulator valve 75 measures the circuit pressure and regulates the hydraulic output of the pump so that it never exceeds, within certain operational limitations, a maximum pressure as determined by the strength of the valve spring 76.

Combinations of my improved pump and fluid motor can be used for many new and useful purposes.

The pump and fluid motor can be combined into a unit to be used as a speed variator wherein the pump is driven at constant R. P. M. and the fluid motor is metered to produce a different R. P. M. to suit a given requirement; or wherein the angle of the thrust assembly of the pump or of the fluid motor, or both, may be adjusted to obtain the described speed change. This may also be done when the motor is a remote location with relation to the pump.

A combination may be used wherein a pressure limit pump is used with a fluid motor as a torque safety device in such a way that the speed of the fluid motor will slow down or even come to a stop when torque loads occur.

A pump and fluid motor combination may be used to produce the greatest R. P. M. of output with a constant power input when the torsional load is extremely variable.

Moreover, the combination may be used as a revolution stabilizer for obtaining a constant output of R. P. M. from a source of power which is highly variable as to R. P. M.

Many other combinations may be used within the scope of my present invention.

It is to be understood that the type of mechanical structure herein described, particularly as it applies to hydraulic pumps and the motors, can be designed for use with any pressure up to the limiting strength of the confining materials from which the structure is formed. With a kinematic system which prescribes effective mechanical advantages, freedom of motion of the working parts, full mechanical counterbalance and full hydraulic counterbalance, there is no component of this mechanical combination which discloses any inherent weakness and the limitations of pressure created are restricted only by the strength of the materials used. The present pump will give a lasting performance for the reason that the wearing parts are reduced to a minimum, the bearing surfaces are ample and because the thrust of the hydraulic load is carried directly by a ball bearing assembly.

For the foregoing reasons and because full use is made of pressure lubrication, the improved pump structure of the present invention can be operated at speeds which are substantially higher than could be obtained from previously known pumps and because of the fact that the pump casing or housing is provided with cored passageways of relatively large area, there is a minimum tendency to produce or generate heat during pressure operation.

Having thus described my invention, what I claim is:

1. A hydraulic pump comprising a housing having intake and discharge openings, a rotatable shaft journaled in said housing, a rotor including opposed cylinder shells and sets of coaxial opposed pistons rotatable with said shaft, a thrust bearing assembly suspended within said casing having a portion rotatable with said rotor, means for operatively connecting said assembly to said pistons for reciprocating the same, said thrust bearing assembly being capable of tilting movement about its points of suspension between a vertical position and an angular position, spring means for normally maintaining said thrust bearing assembly in its angular position, and pressure means for controlling the angular movement of said assembly.

2. A hydraulic pump of the axially rotatable type comprising a housing having intake and discharge openings, a rotatable shaft journaled in said housing, a rotor including opposed cylinder shells and sets of coaxial opposed pistons rotatable with said shaft, a thrust bearing assembly suspended within said casing having a portion rotatable with said rotor, means for operatively connecting said assembly to said pistons for reciprocating the same, said thrust bearing assembly being capable of tilting movement about its points of suspension between a vertical position and an angular position, spring means for normally urging said thrust bearing assembly toward angular position, and metering valve means in operative communication with the assembly for automatically controlling the angular movement of said assembly whereby to govern the speed and pressure output of said hydraulic pump.

3. A hydraulic pump comprising a closed housing having intake and discharge openings, a rotatable shaft journaled in said housing, a rotor including opposed sets of coaxial reciprocable pistons rotatable with said shaft, disk valves having ports formed therein located in opposite walls of the housing and cooperable with said pistons, a thrust assembly suspended within the housing surrounding said rotor and having members rotatable with said rotor, certain of said members being associated with said pistons for reciprocating the same, said thrust assembly as a whole being automatically shiftable about its points of suspension between a vertical position within the housing and an angular position of approximately 20 degrees, and automatic means including said thrust assembly for regulating the output of said pump.

4. A hydraulic pump comprising a housing having intake and discharge openings, a rotatable shaft journaled in said housing and having a portion projecting outwardly through one of said walls for connection to a source of power, a rotor including opposed cylinder shells and sets of coaxial opposed pistons rotatable with said shaft, a thrust bearing assembly suspended within said housing having parts thereof rotatable with said rotor and operatively connected to said pistons for reciprocating the same, said thrust bearing assembly being capable of tilting movement about its points of suspension between a vertical position and an angular position, spring means for normally maintaining said thrust bearing assembly in its angular position, pressure means for controlling the angular movement of said assembly, and adjustable fluid metering means for relieving the pressure within said pump whereby to maintain the pressure output of the pump constant.

5. A hydraulic pump comprising a housing having intake and discharge openings, a rotatable shaft journaled in said housing, a rotor including opposed cylinder shells and sets of axially aligned opposed pistons rotatable with said shaft, a thrust bearing assembly suspended within said casing having a portion rotatable with said rotor and operatively connected to said pistons for reciprocating the same, said thrust bearing assembly being capable of tilting movement about its points of suspension between a vertical position and an angular position, spring means for normally maintaining said thrust bearing assembly in its angular position, pressure means for controlling the angular movement of said assembly, and a fluid metering needle valve for relieving the pressure within said pump when it exceeds a predetermined amount whereby to maintain the pressure output of the pump constant.

6. A hydraulic pump of the axially rotatable type comprising a housing having intake and discharge openings, a rotatable shaft journaled in said housing, a rotor including opposed cylinder shells and sets of opposed pistons rotatable with said shaft, a thrust bearing assembly suspended within said casing including a thrust ring rotatable with said rotor and having means operatively connected to said pistons for reciprocating the same, said thrust bearing assembly being capable of tilting movement about its points of suspension between a vertical position and an angular position, a spring-pressed plunger for normally maintaining said thrust bearing assembly in an angular position, a pressure chamber associated with said spring-pressed plunger for resisting the movement of said assembly from angular position toward vertical position, and metering means including a needle valve for relieving the pressure in said chamber at predetermined times whereby to maintain the pressure output of the pump constant.

7. A hydraulic pump of the axially rotatable type comprising a housing having intake and discharge openings, a rotatable shaft journaled

in said housing, a rotor including opposed cylinder shells and sets of opposed pistons rotatable with said shaft, valves with which said cylinder shells and pistons cooperate, a thrust bearing assembly suspended within said casing having parts rotatable with said rotor, certain said parts being operatively connected to said pistons for reciprocating the same, said thrust bearing assembly being capable of tilting movement about its points of suspension between a vertical position and an angular position, a spring-pressed plunger for normally maintaining said thrust bearing assembly in an angular position, a pressure chamber associated with said spring-pressed plunger for resisting the movement of said assembly from angular position toward vertical position, and a manually adjustable regulator including a needle valve for automatically relieving the pressure in said chamber at predetermined times whereby to reduce the pump pressure when it exceeds a predetermined limit.

8. A hydraulic pump comprising a housing having intake and discharge openings, a rotatable shaft journaled in said housing, a rotor including opposed cylinder shells and pairs of axially aligned opposed pistons rotatable with said shaft, valve means associated with said cylinders, a thrust bearing assembly suspended within said casing having parts thereof rotatable with said rotor and operatively connected to said pistons for reciprocating the same, said thrust bearing assembly being capable of tilting movement about its points of suspension between a vertical position and an angular position, and means for normally maintaining said assembly in its angular position but permitting movement thereof toward vertical position when the pressure in the pump exceeds a predetermined amount.

9. A hydraulic pump of the axially rotatable type comprising a housing having end plates, a rotatable shaft journaled in said end plates, a rotor including opposed sets of reciprocable pistons rotatable with said shaft, the pistons of each set being arranged in axial alignment with one another, valves located in the end plates and cooperable with said pistons, trunnions located at diametrically opposite points in the housing, and a thrust assembly carried by said trunnions surrounding said rotor and having parts thereof rotatable with the rotor and operatively associated with said pistons, said thrust assembly being movable between a vertical position within the housing and an angular position whereby to cause reciprocation of said pistons upon rotation of said rotor, each of said valves having a series of intake and a separate series of discharge ports whereby upon rotation and reciprocation of said pistons certain thereof of both sets draw in fluid from the induction side of the pump through the intake ports while others thereof of both sets are simultaneously discharging fluid through the discharge ports to the discharge side of said pump.

10. A reversible hydraulic pump comprising a housing having intake and discharge openings, a rotatable shaft journaled in said housing, a rotor including opposed cylinder shells and opposed pairs of pistons rotatable with said shaft, a thrust bearing assembly suspended within said casing having rotatable parts operatively connected to said pistons for reciprocating the same, said thrust bearing assembly being capable of tilting movement about its points of suspension between a vertical position and an angular position, and means for maintaining said thrust bearing assembly in angular position, said means comprising a fluid actuated piston having a portion engaging said assembly for tilting the same, said housing having a cylinder cavity located behind said piston, one end of said piston cooperating with said cylinder cavity and together providing a dashpot for cushioning the piston at the end of its stroke.

11. A reversible hydraulic pump of the axially rotatable type comprising a housing having end walls, a rotatable shaft journaled in said casing, a rotor including opposed cylinders and sets of reciprocable pistons located in said cylinders and rotatable with said shaft, valves located in the end walls and cooperable with said pistons, trunnions located at diametrically opposite points in the housing, a thrust assembly carried by said trunnions surrounding said rotor having a portion rotatable therewith and operatively associated with said pistons, said thrust assembly being movable between a vertical position within the housing and an angular position whereby to cause reciprocation of said pistons upon rotation of said rotor, and means at the forward end of said shaft for applying pressure to the cylinder at said shaft end to maintain said cylinder in intimate contact with the adjacent valve.

12. A hydraulic pump comprising a closed housing having intake and discharge openings, a rotatable shaft journaled in said housing and having a portion projecting outwardly through one of the walls for connection to a source of power, a rotor including spaced cylinders and opposed sets of reciprocable pistons rotatable with said shaft, disk valves having ports formed therein located in opposite walls of the housing and cooperable with said pistons, a thrust assembly suspended within the housing surrounding said rotor and having members thereof rotatable with said rotor, said thrust assembly having means associated with said pistons for reciprocating the same, said thrust assembly as a whole being automatically shiftable about its points of suspension between a vertical position within the housing and an angular position, means for automatically maintaining said cylinders in intimate contact with said disk valves, and means for maintaining drifting films of lubricant between the adjacent faces of the cylinders and the disk valves.

13. A reversible hydraulic pump of the axially rotatable type comprising a housing having intake and discharge openings, a rotatable shaft located within the housing, a rotor including a pair of opposed spaced cylinders journaled in bearings formed in opposite walls of said housing, one of said cylinders having open ends and the other of said cylinders having a centrally disposed wall at one end thereof, a plurality of pairs of opposed and articulated pistons reciprocably mounted in said cylinders, said open ended cylinder being fixed to said shaft and said other cylinder having a sliding fit on said shaft but rotatable therewith, said shaft being journaled at one of its ends in a wall of said housing and having its opposite end journaled in said walled cylinder, disk valves located in the housing walls and cooperating with the outer adjacent ends of said cylinders, a thrust bearing assembly suspended within said housing and having a ring rotatable with said rotor, means operatively connecting said ring and said pairs of pistons for reciprocating the latter, said thrust bearing assembly being capable of tilting movement about its points of suspension between a vertical position and an angular position to cause reciprocation of said pistons upon rotation of said rotor,

means for normally maintaining said assembly in its angular position, means for automatically controlling the angular movement of said thrust bearing assembly whereby to govern the speed and pressure output of said pump, means between the centrally disposed cylinder wall and the adjacent end of said shaft for maintaining the face of said cylinder in intimate contact with the face of the adjacent disk valve, and means for providing and maintaining a mobile film of lubricant between said last-named cylinder and the face of said adjacent disk valve.

14. A reversible hydraulic pump of the axially rotatable type comprising a housing having intake and discharge openings, a rotatable shaft located within the housing, a rotor including a pair of opposed spaced cylinders journaled in bearings formed in opposite walls of said housing, one of said cylinders having open ends and the other of said cylinders having a centrally disposed wall at one end thereof, a plurality of pairs of opposed and articulated pistons reciprocably mounted in said cylinders, said open ended cylinder being fixed to said shaft and said other cylinder having a sliding fit on said shaft but rotatable therewith, said shaft being journaled at one of its ends in a wall of said housing and having its opposite end journaled in said walled cylinder, disk valves located in the housing walls and cooperating with the outer adjacent ends of said cylinders, a thrust bearing assembly suspended within said housing and having a ring rotatable with said rotor, means operatively connecting said ring and said pairs of pistons for reciprocating the latter, said thrust bearing assembly being capable of tilting movement about its points of suspension between a vertical position and angular position to cause reciprocation of said pistons upon rotation of said rotor, means for normally maintaining said assembly in its angular position, means for automatically controlling the angular movement of said thrust bearing assembly whereby to govern the speed and pressure output of said pump, and mechanical and hydraulic pressure means between the central wall of said cylinder and the adjacent end of said shaft for maintaining the outer face of said cylinder in intimate contact with the face of the adjacent disk valve.

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