

Dec. 8, 1953

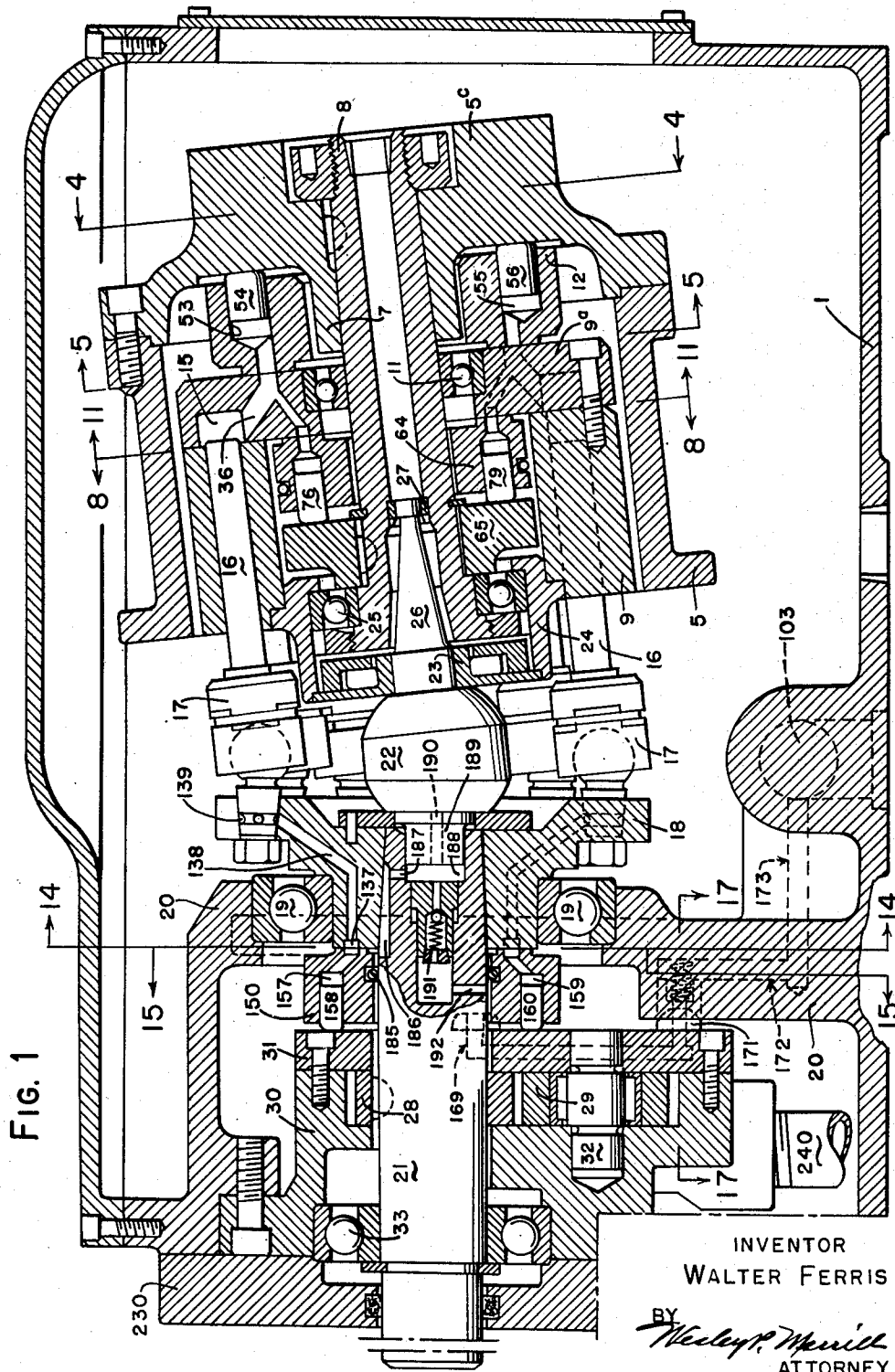
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2,661,701

AXIAL TYPE HYDRODYNAMIC MACHINE

Filed Oct. 3, 1947

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AXIAL TYPE HYDRODYNAMIC MACHINE

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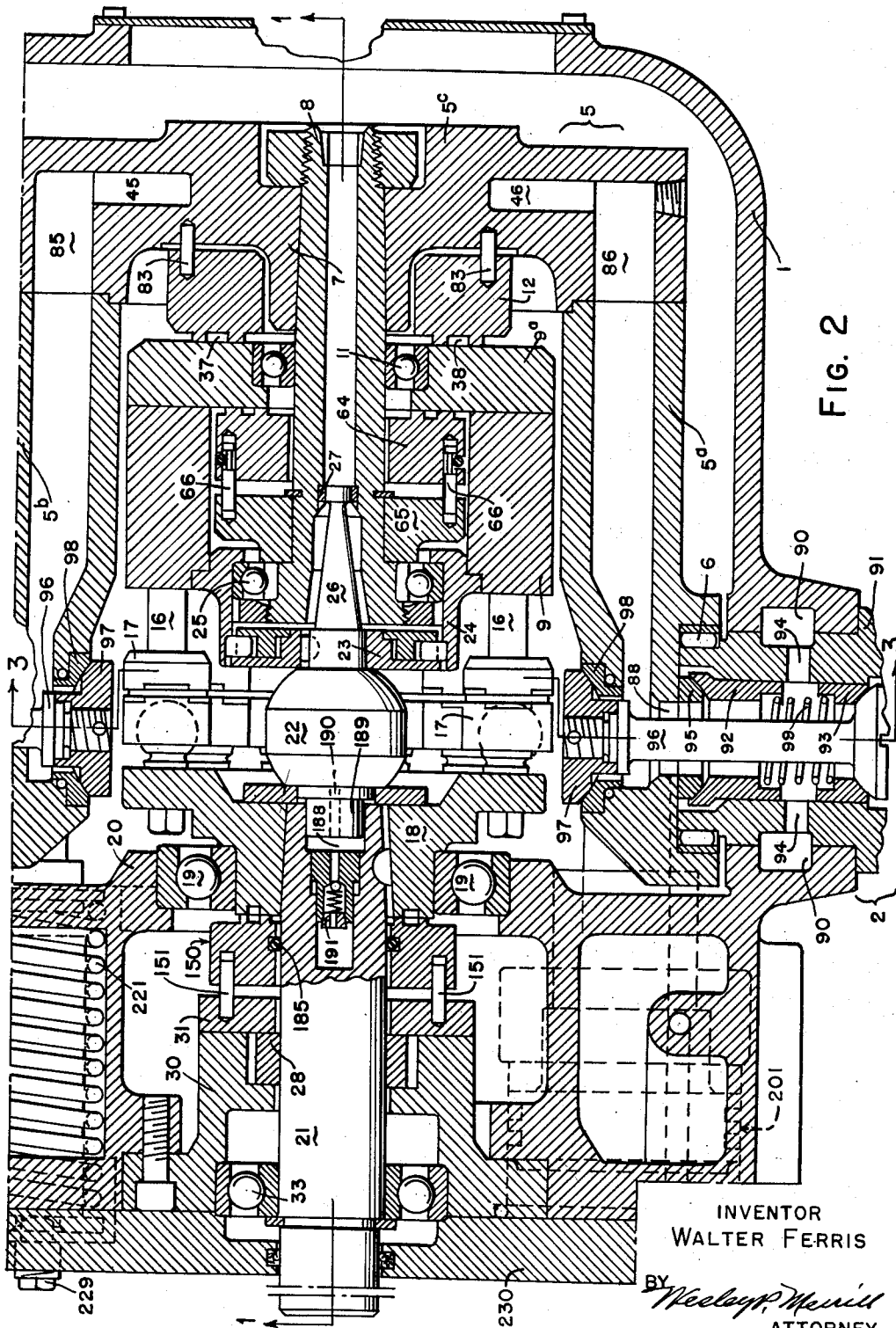


FIG. 2

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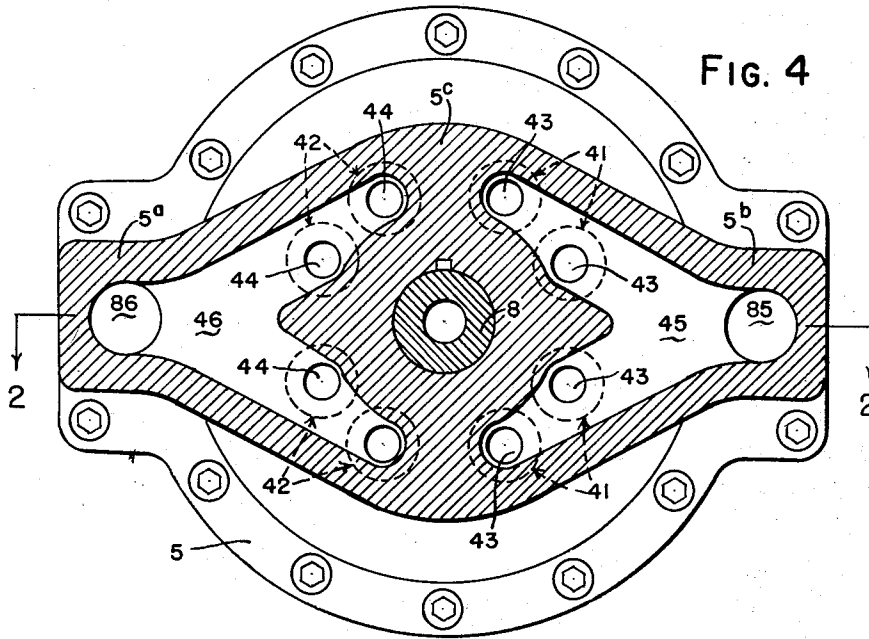
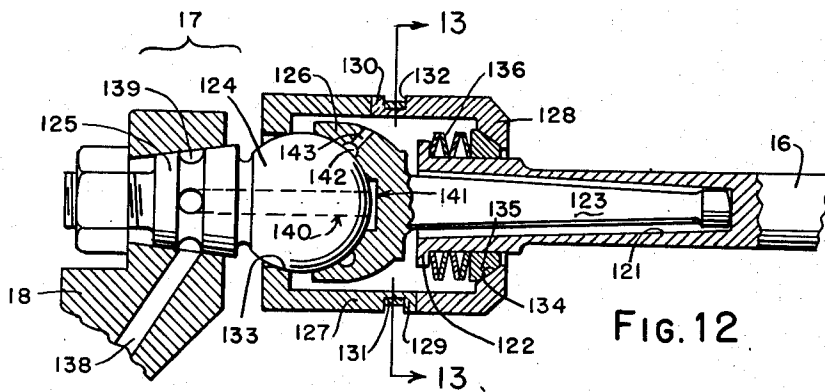
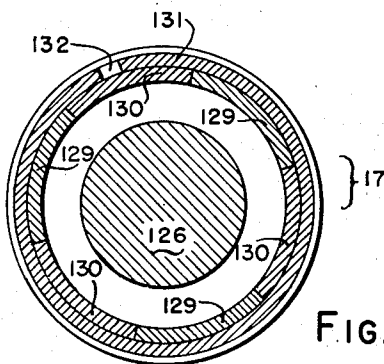


FIG. 12

FIG. 13

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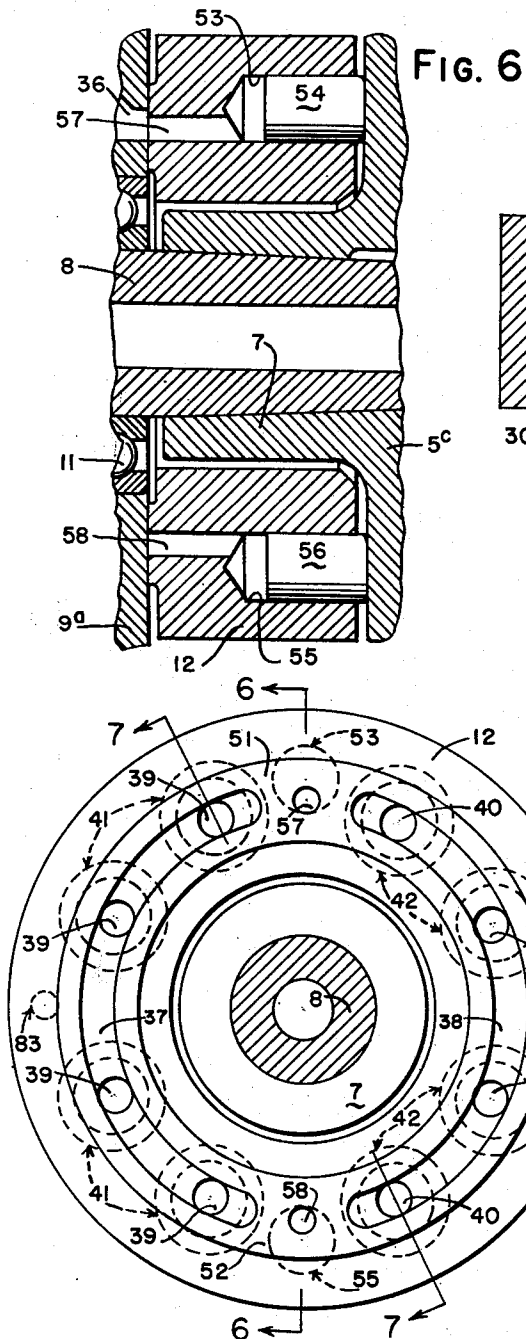
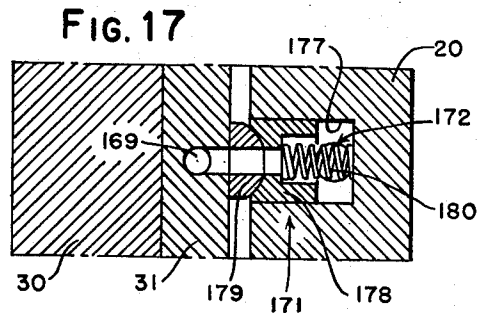
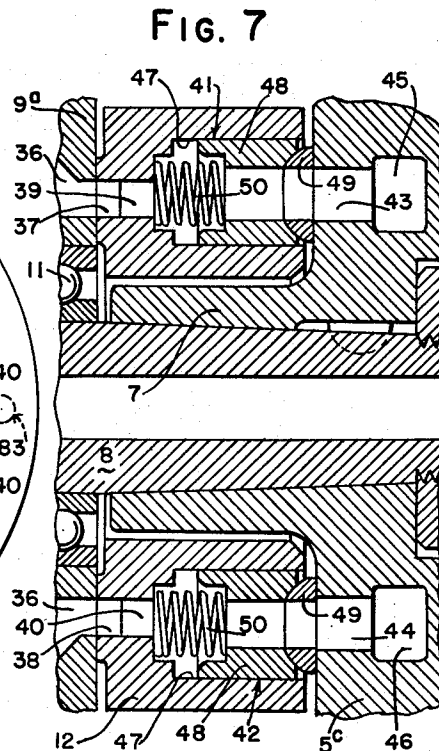
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AXIAL TYPE HYDRODYNAMIC MACHINE

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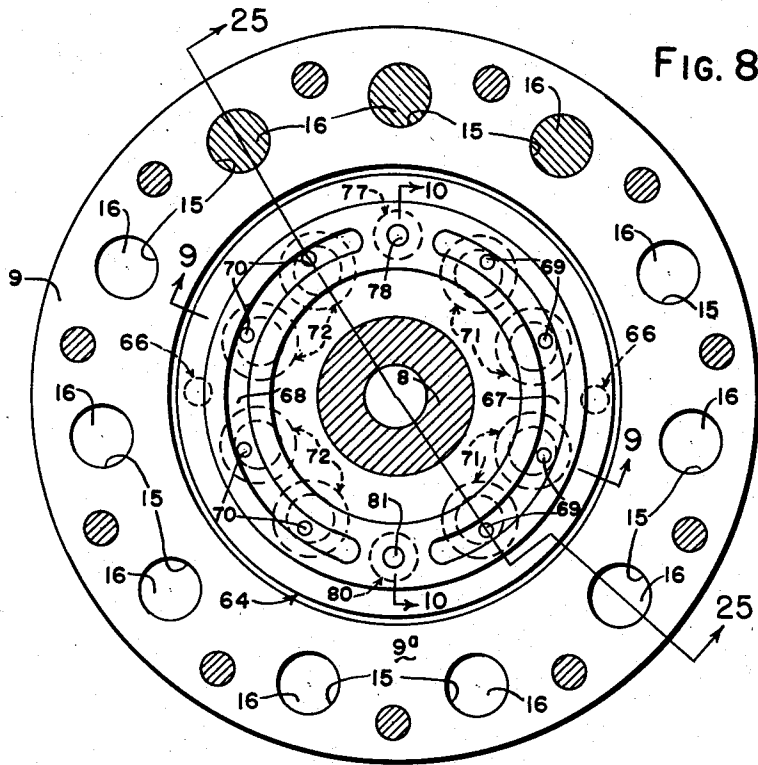


FIG. 8

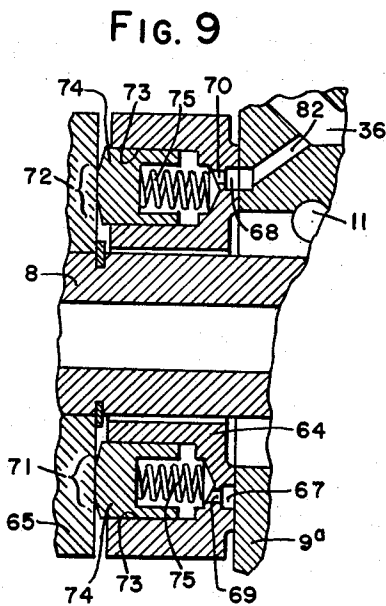


FIG. 9

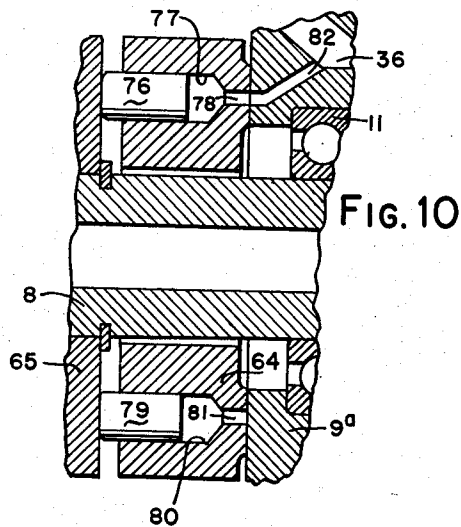


FIG. 10

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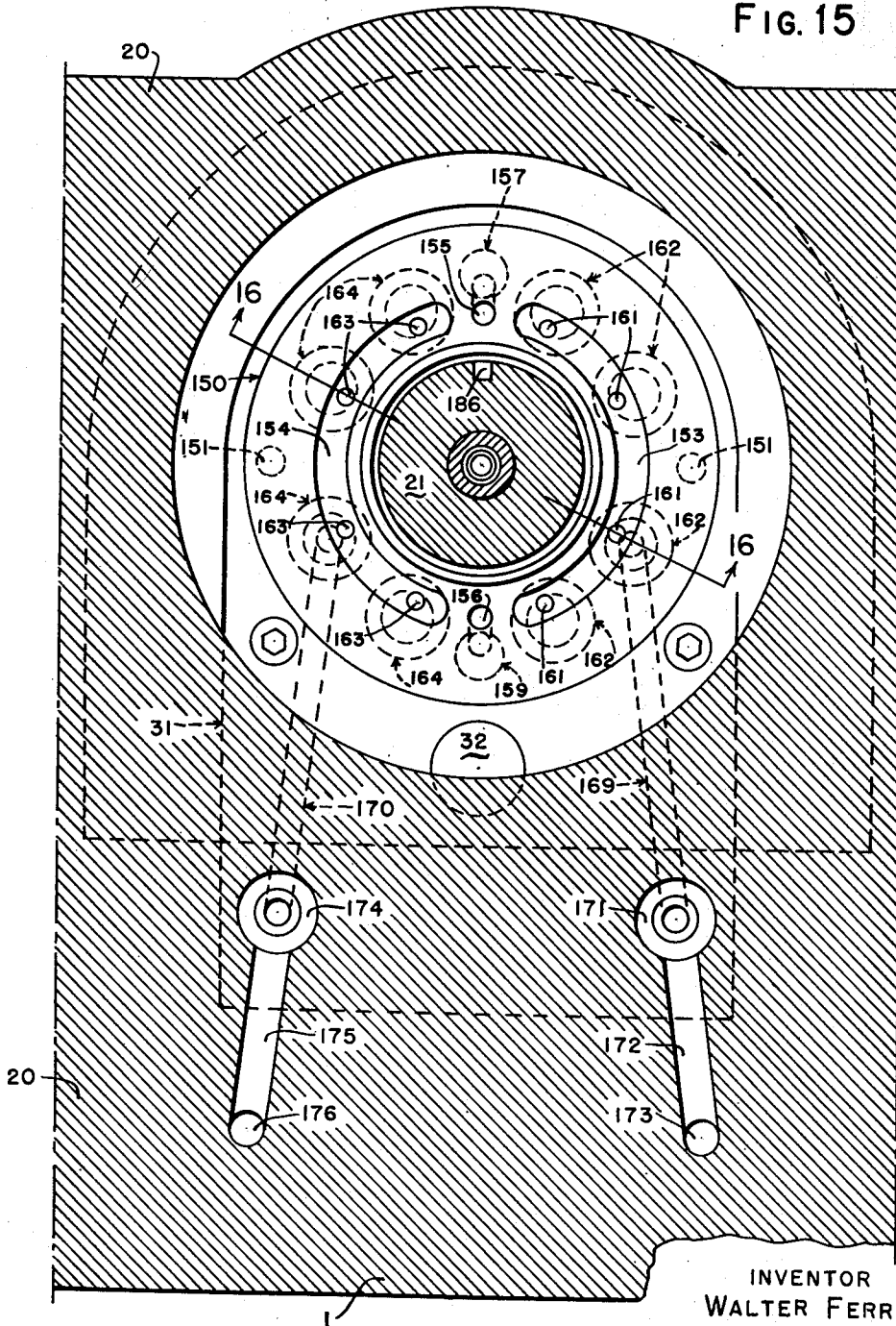
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FIG. 15



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FIG. 21

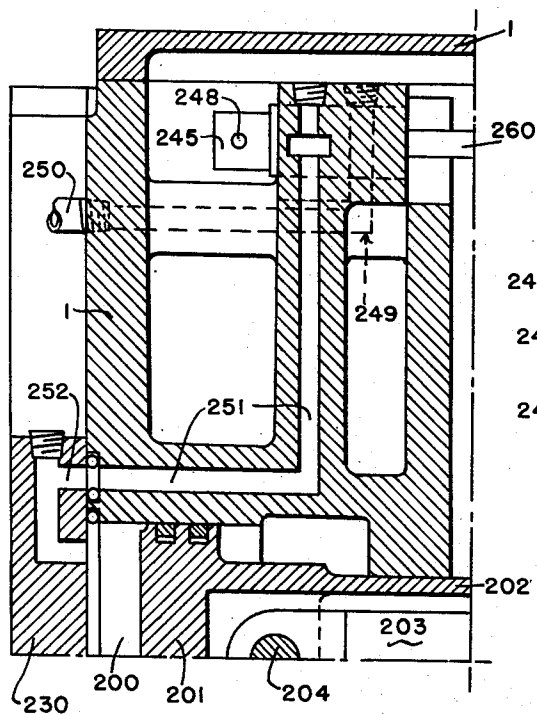


FIG. 22

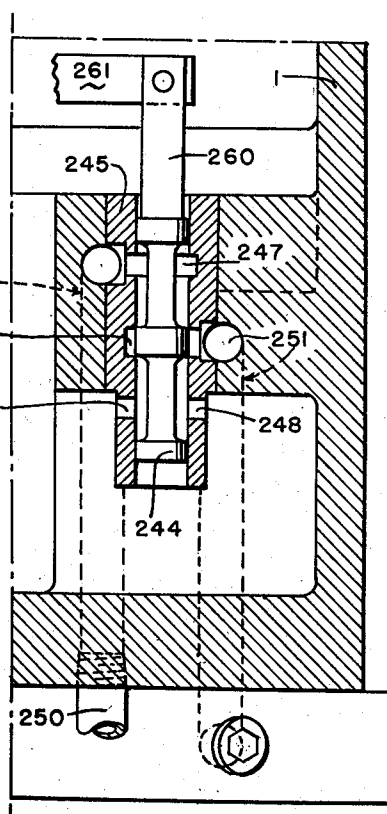
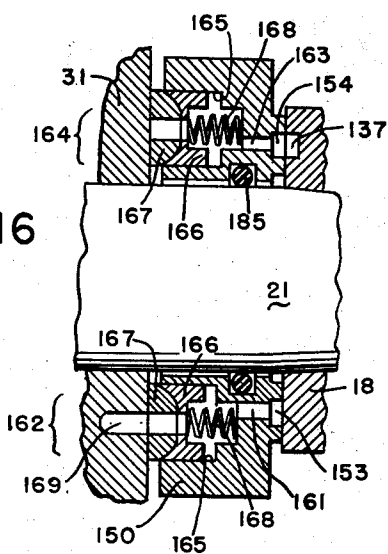


FIG. 16



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FIG. 19

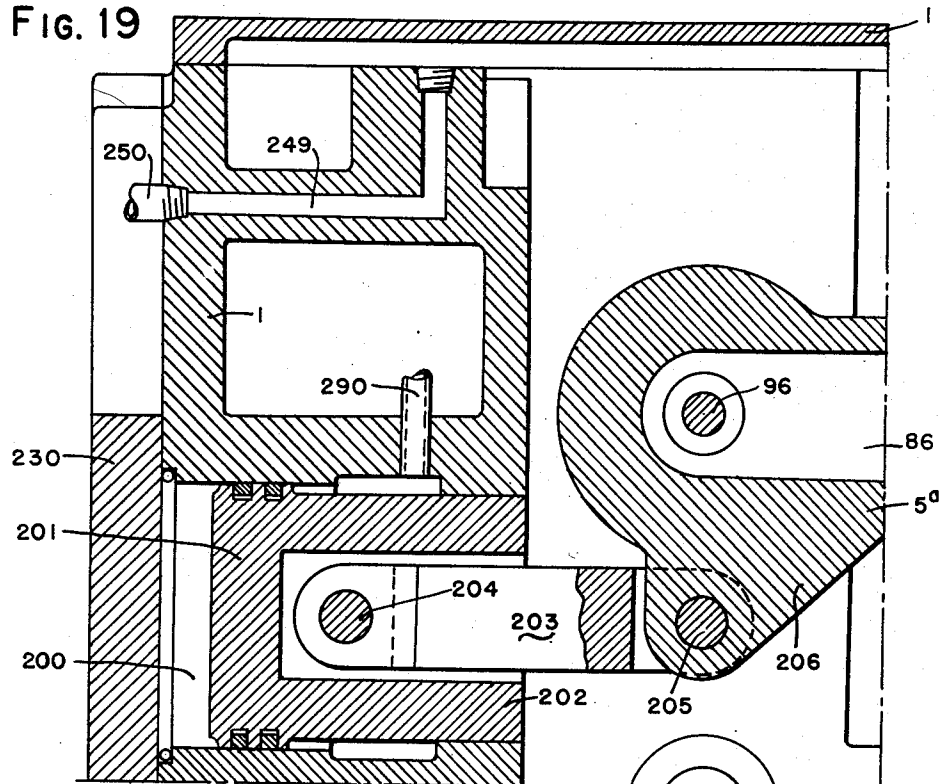
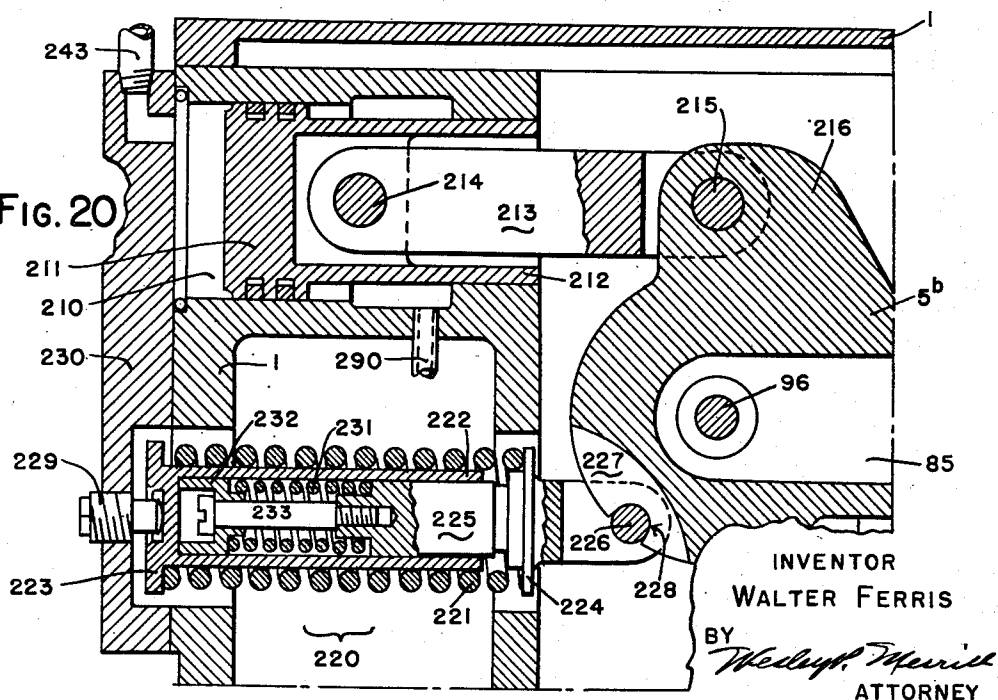


FIG. 20



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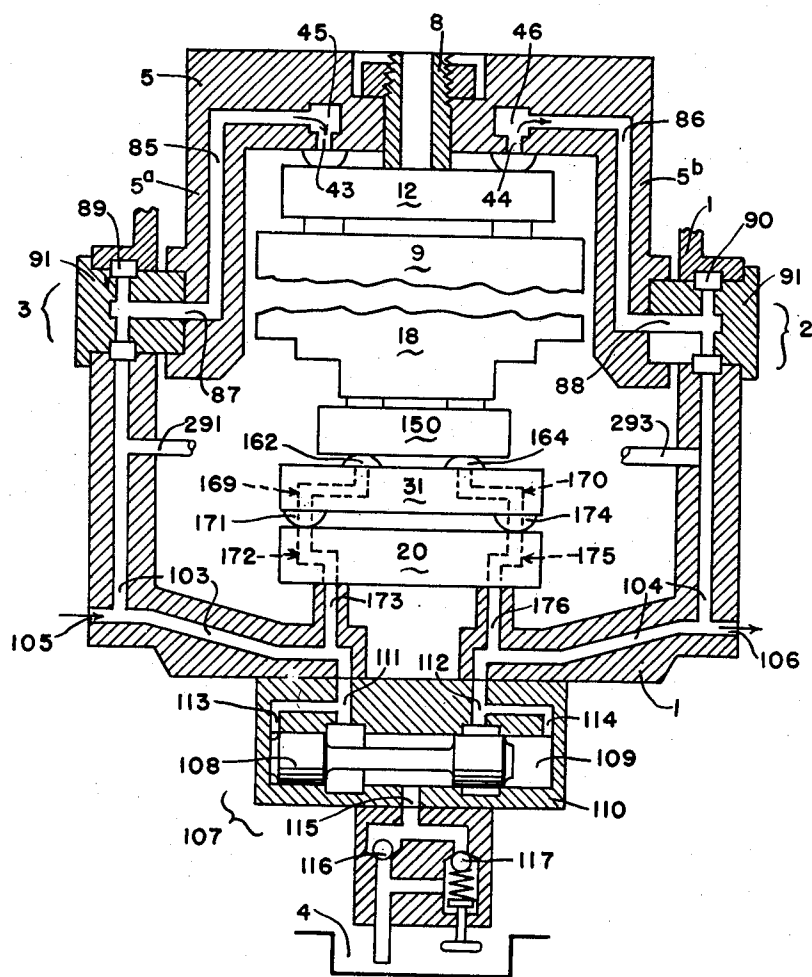
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FIG. 23



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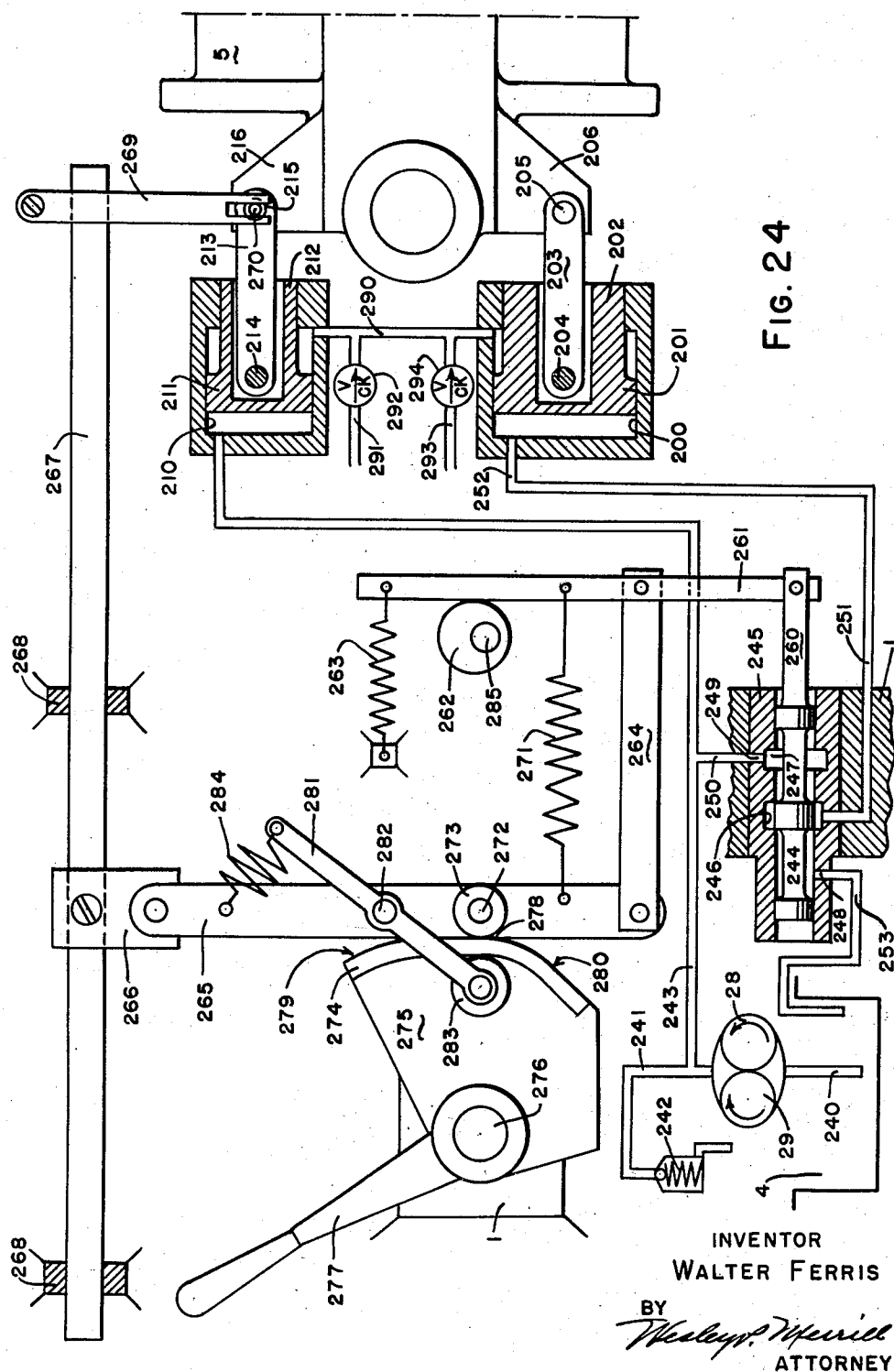
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AXIAL TYPE HYDRODYNAMIC MACHINE

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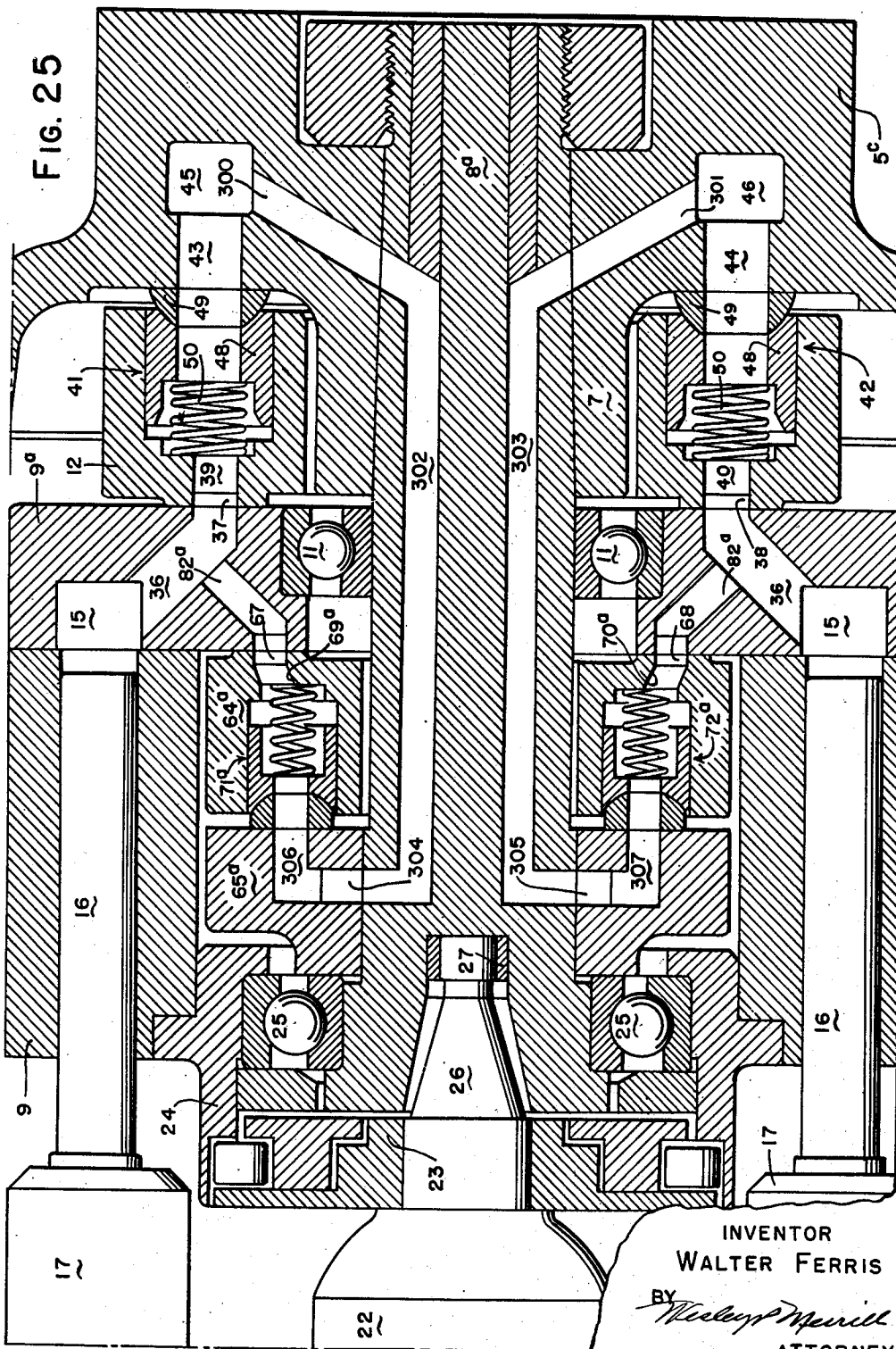
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AXIAL TYPE HYDRODYNAMIC MACHINE

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UNITED STATES PATENT OFFICE

2,661,701

AXIAL TYPE HYDRODYNAMIC MACHINE

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Application October 3, 1947, Serial No. 777,794

26 Claims. (Cl. 103—162)

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This invention relates to rotary hydrodynamic machines of the type having a rotatable cylinder barrel, a plurality of pistons and cylinders arranged in the cylinder barrel around and parallel to its axis of rotation and a rotatable thrust member which cooperates with the pistons and is inclined to the cylinder barrel axis when the machine is performing useful work.

Such a machine will function as a pump when driven mechanically and it will function as a motor when supplied with motive liquid but, since the functions of a pump and the functions of a motor are substantially opposite to each other, the machine will be referred to herein as a pump in order to simplify the description of a machine in which the invention is embodied but it is to be understood that the term "pump" as used herein means a hydrodynamic machine regardless of whether the machine functions as a pump or as a motor. For the purpose of illustration, the invention has been shown embodied in an angle type axial pump but certain features of the invention are applicable to other types of axial pumps.

The present invention has as an object to provide an axial type pump which when operating at high pressures will have a higher overall efficiency than the prior axial type pumps.

Another object is to provide an axial type pump capable of delivering liquid at very high pressures and at very large volumetric rates relative to the size of the pump.

Another object is to provide hydraulic means for limiting the axial thrust upon the bearings which support the rotatable parts.

The flow of liquid to and from the pumping cylinders of an axial type pump is ordinarily controlled by a flat valve which engages the end of the cylinder barrel and is provided with two ports with which the cylinder ports register alternately during operation of the pump. When the pump creates pressure, the liquid in the ports and in the lubricating film between the valve and the cylinder barrel exerts a blow-off force which tends to move the cylinder barrel away from the valve.

Another object of the present invention is to provide hydraulic means in addition to the pumping pistons for exerting a hold-up force upon the cylinder barrel in opposition to the blow-off force.

Another object is to provide an axial type pump with a main flat valve and an auxiliary flat valve which together control the flow of liquid to and from the pumping cylinders and the auxiliary valve also applies a hold-up force to the cylinder barrel in opposition to the blow-off force exerted

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by the liquid between the main valve and the end of the cylinder barrel.

Another object is to provide an angle type pump having novel means for varying its displacement.

Another object is to provide an angle type pump with hydraulic means for relieving the thrust member bearing from the greater part of the axial thrust imposed thereon when the pump is creating pressure.

Another object is to provide an axial type pump in which the pumping pistons are connected to the thrust member through universal joints which are lubricated in a novel manner.

Another object is to provide an axial type pump in which the cylinder barrel is driven through a universal joint and novel means are provided for lubricating that joint.

Other objects and advantages will be apparent from the following description of the embodiments of the invention shown in the accompanying drawings in which the views are as follows:

Fig. 1 is a central vertical longitudinal section through a pump in which the invention is embodied, the view being taken on the line 1—1 of Fig. 2 and showing the pump adjusted to deliver liquid in one direction.

Fig. 2 is a central sectional plan view taken on the lines 2—2 of Figs. 3 and 4 but showing the pump adjusted to zero displacement, certain parts being broken away and omitted in order that the view may be on the same scale as Fig. 1.

Fig. 3 is a transverse vertical section taken on the irregular line 3—3 of Fig. 2.

Fig. 4 is a transverse vertical section taken on the line 4—4 of Fig. 1.

Fig. 5 is a face view of the valve for controlling the flow of liquid to and from the pumping cylinders, the view being taken on the line 5—5 of Fig. 1 but drawn to a larger scale.

Figs. 6 and 7 are sectional views taken through the valve and portions of the adjacent parts, the planes of the views being indicated, respectively, by the lines 6—6 and 7—7 of Fig. 5.

Fig. 8 is a transverse section showing the arrangement of the cylinders in the cylinder barrel and the face of a valve-like hold-up element which assists in holding the ends of the cylinder barrel against the face of the valve shown in Fig. 5, the view being taken in the plane indicated by the line 8—8 of Fig. 1 but drawn to a larger scale.

Figs. 9 and 10 are sections through the valve-like hold-up element and portions of the adjacent parts, the views being taken on the lines 9—9 and 10—10 of Fig. 8.

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Fig. 11 is a transverse section taken on the line 11—11 of Fig. 1 but drawn to a larger scale.

Fig. 12 is an enlarged sectional view of one of universal joints for connecting the pumping pistons to the thrust member which causes the pistons to reciprocate when the pump is driven.

Fig. 13 is a section taken on the line 13—13 of Fig. 12 but drawn to a still larger scale.

Fig. 14 is an end view of the thrust member which causes the pistons to reciprocate when the pump is driven, the view being taken in the plane indicated by the line 14—14 of Fig. 1 but drawn to a larger scale.

Fig. 15 is a transverse sectional view taken on the irregular line 15—15 of Fig. 1 but drawn to a larger scale and showing a valve-like hold-up element for exerting upon the thrust member forces counter to the pumping forces.

Fig. 16 is a section through the valve-like hold-up element shown in Fig. 15 and portions of the adjacent parts, the view being taken on the line 16—16 of Fig. 15.

Fig. 17 is a sectional view showing a connection through which liquid is supplied to the valve-like hold-up element shown in Fig. 15, the view being taken on the line 17—17 of Fig. 1.

Fig. 18 is an end view of the pump looking toward the left hand end of Fig. 1.

Fig. 19 is a longitudinal section taken through a portion of the pump on the irregular line 19—19 of Figs. 3 and 18.

Fig. 20 is a longitudinal section taken through a portion of the pump on the irregular lines 20—20 of Figs. 3 and 18.

Fig. 21 is a section taken on the irregular line 21—21 of Fig. 18.

Fig. 22 is a section taken on the line 22—22 of Fig. 18 but drawn to a larger scale.

Fig. 23 is a schematic drawing illustrating the hydraulic circuit of the pump.

Fig. 24 is a schematic drawing illustrating the mechanism for controlling the pump.

Fig. 25 is a longitudinal section through a portion of a pump in which the valve-like hold-up element shown in Figs. 1, 2 and 8 is replaced by an auxiliary valve which urges the cylinder barrel against the main valve and also provides ports and passages for the flow of liquid to and from the pump cylinders so that the ports and passages in the main valve may be correspondingly reduced in size, the plane of the view being indicated by the irregular line 25—25 of Fig. 8.

General arrangement

The pump chosen for illustration has its mechanism enclosed by a housing 1 having webs or ribs arranged therein and two trunnions 2 and 3 (Fig. 3) fixed in its side walls in axial alinement with each other. Housing 1 is adapted to be supported upon a tank or reservoir 4 from which the pump is supplied with liquid. Reservoir 4 has not been illustrated in detail but is shown diagrammatically in Figs. 23 and 24.

Trunnions 2 and 3 pivotally support a hollow cradle or yoke 5 having two integral arms 5^a and 5^b formed upon opposite sides thereof and an end wall or head 5^c rigidly secured to its rear or right hand end and forming a fluid tight joint therewith. Arms 5^a and 5^b are pivoted upon trunnions 2 and 3 respectively, and a bearing 6 is arranged between each arm and the trunnion upon which it is pivoted.

Yoke head 5^c is of substantial thickness and it has an integral hub 7 extending from its inner or forward face. A horn 8 is rigidly secured in hub

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7 and head 5^c upon the axis of yoke 5 to rotatably support an annular cylinder barrel 9 having an annular head 9^a arranged upon its rear end, a bearing 11 being arranged between cylinder head 9^a and horn 8.

Cylinder barrel 9 and its head 9^a are ordinarily made as separate parts for convenience in manufacturing and the two parts are rigidly secured to each other to form a fluid tight joint therebetween. The rear face of end head 9^a engages an annular flat valve 12 which extends around hub 7 between cylinder head 9^a and yoke head 5^c to control communication between the cylinders in cylinder barrel 9 and opposite sides of a hydraulic circuit as will presently be explained.

Cylinder barrel 9 has a plurality of cylinders 15 arranged therein around and parallel to its axis of rotation and a piston 16 is fitted in each cylinder. Each piston 16 is connected by a ball and socket joint 17 to a thrust member 18 which is supported by a bearing 19 from a web 20 of housing 1. Thrust member 18 is fixed to the inner end of a drive shaft 21 which extends through the front or left hand wall of housing 1 for connection to a source of power.

Rotation is imparted from shaft 21 to cylinder barrel 9 through a universal joint 22 having the front or driven part thereof fixed for rotation with shaft 21 and the rear or driving part thereof fixed to the inner part 23 of an Oldham coupling the outer part 24 of which is fixed to the outer end of cylinder barrel 9. A bearing 25 is arranged between horn 8 and the outer part 24 of the Oldham coupling so that cylinder barrel 9 is supported by bearings 11 and 25 for rotation upon the axis of horn 8. The rear part of universal joint 22 is provided with a tail shaft 26 which has its rear end journaled in a bearing 27 arranged within horn 8 and which holds the axis of the rear part of universal joint 22 upon the axis of cylinder barrel 9.

The arrangement is such that when shaft 21 is rotated, cylinder barrel 9 will be rotated from shaft 21 through universal joint 22 and Oldham coupling 23—24. The rear face of thrust member 18 rotates in a plane normal to the axis of shaft 21 and, if the axis of cylinder barrel 9 is inclined to the axis of thrust member 18 by tilting yoke 5 upward as shown in Fig. 1 or by tilting yoke 5 downward, thrust member 18 will reciprocate pistons 16 in their cylinders during rotation of thrust member 18 but, if the axis of cylinder barrel 9 is coincident with the axis of thrust member 18, no reciprocation of pistons 16 will occur.

Yoke 5 is adapted to be tilted upon trunnions 2 and 3 and held in adjusted positions by hydraulically actuated pump displacement varying means to be presently described. Liquid for operating the displacement varying means may be supplied from any suitable source such as a gear pump having its two gears 28 and 29 arranged in a suitable chamber formed in a gear pump housing 30 which is fixed within main housing 1 and has a cover plate 31 to enclose the gears within the chamber as shown in Fig. 1. Gear 28 is fixed upon shaft 21 and gear 29 is mounted upon a shaft 32 which is carried by housing 30 and its cover plate 31. Gear pump housing 30 carries a bearing 33 which with bearing 19 rotatably supports shaft 21.

Cylinder barrel, valve and hold-up means

As previously explained, cylinder barrel 9 contains a plurality of cylinders 15, eleven being shown in Fig. 8. A passage 36 (Figs. 1 and 11)

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extends from the inner or rear end of each cylinder 15 through cylinder head 9^a the rear face of which constitutes a seat for valve 12. Liquid flows through passages 36 to and from cylinders 15 and the pressure in that liquid creates a blow-off force which tends to separate the cylinder barrel and the valve as will presently be explained. Since the blow-off force is substantially proportional to the area of the valve in contact with the cylinder barrel, each passage 36 is extended inward and rearward at an angle to the cylinder axis and then extended rearward in order that the contact face of valve 12 may be as small as practicable. The rear ends of passages 36 are elongated in a circumferential direction to form cylinder ports as shown in Fig. 11.

During rotation of cylinder barrel 9, each passage 36 will register successively with two arcuate ports 37 and 38 (Figs. 5 and 7) which are formed in valve 12 and extend inward from the contact face thereof. A plurality of holes 39 extend rearward from the bottom of port 37 and an equal number of holes 40 extend rearward from the bottom of port 38. Each hole 39 communicates with a hold-up motor 41 which is arranged within valve 12 and each hole 40 communicates with a hold-up motor 42 which is arranged within valve 12. Each hold-up motor 41 communicates with a hole 43 which is formed in yoke head 5^c and each hold-up motor 42 communicates with a hole 44 which is formed in yoke head 5^c. All of the holes 43 communicate with a chamber 45 (Figs. 4 and 2) which is formed in yoke head 5^c upon one side of the pump centerline, and all of the holes 44 communicate with a chamber 46 which is formed in yoke head 5^c upon the other side of the pump centerline.

Hold-up motors 41 and 42 are identical and each includes a cylinder 47 (Fig. 7) which is formed in valve 12 in alignment with a hole 39 or 40, a hollow piston 48 which is fitted in cylinder 47, an annular sealing member 49 which is arranged between piston 48 and yoke head 5^c and is concentric with piston 48 and with a hole 43 or 44, and a spring 50 which is arranged between piston 48 and the end wall of cylinder 47 to urge valve 12 against end head 9^a and to urge piston 48 against sealing member 49 and sealing member 49 against yoke head 5^c. The rear end of sealing member 49 and the front face of yoke head 5^c adjacent hole 43 or 44 are made flat and smooth and the front end of sealing member 49 and the rear end of piston 48 are made spherical and smooth to form substantially liquid tight joints therebetween.

The adjacent ends of ports 37 and 38 are spaced apart an angular distance equal to or slightly greater than the angular length of a passage 36 to provide two sealing arcs or bridges 51 and 52 which constitute seals between the two ports.

When cylinder barrel 9 is rotated, each piston 16 will be drawn outward during one-half of each revolution of cylinder barrel 9 and it will be forced inward during the other half of each revolution of cylinder barrel 9 as will presently be explained. If yoke 5 is tilted upward as shown in Fig. 1, each outward moving piston 16 will draw liquid into its cylinder 15 through the passage 36 connected thereto, port 37, holes 39, hold-up motors 41 and holes 43 from chamber 45 during the time the passage 36 is in communication with port 37, then the liquid will momentarily be trapped in the cylinder as the passage 36 passes across bridge 51, and then the piston as it moves outward will eject liquid from its cylin-

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der through the passage 36, port 38, holes 40, hold-up motors 42 and holes 44 into chamber 46. If yoke 5 is tilted downward or if cylinder barrel 9 is rotated in the opposite direction, the flow of liquid will be reversed.

Liquid will escape at a minute rate from the high pressure port 37 or 38 and form a lubricating film between the contact face of valve 12 and end head 9^a. The pressure in the lubricating film will be equal to pump pressure at the edges of the high pressure port 37 or 38, it will gradually decrease to zero at the inner and outer edges of the valve face adjacent the high pressure port and the pressure will be substantially zero in the film adjacent the low pressure port.

When the pump is creating pressure, the liquid in the lubricating film adjacent the high pressure port and the liquid in the high pressure port will exert a blow-off force which tends to move valve 12 and cylinder head 9^a away from each other and which is proportional to pump pressure at any given position of cylinder barrel 9. However, the same pressure extends into the hold-up motors 41 or 42 on the high pressure side of the pump and causes them to urge valve 12 against cylinder head 9^a with a force proportional to pump pressure.

The pump is ordinarily provided with an odd number of cylinders. Therefore, the number of passages 36 containing liquid under pressure will vary by one each time the outer end of a passage 36 passes from one to the other of the two valve ports 37 and 38. This variation in the number of passages 36 containing liquid under pressure causes the blow-off force to vary in magnitude and the center of the blow-off force to shift.

In order to compensate for the variations in the magnitude and center of the blow-off force, the hold-up motors are so located and are provided in such numbers and sizes that the sum of the hold-up forces exerted by the hold-up motors on the pressure side of the pump is slightly in excess of the total blow-off force and the resultant of the hold-up forces is approximately coincident with the center of the blow-off force when the minimum number of passages 36 contain liquid under pressure, and the pump is provided with balancing motors at least one of which is momentarily energized and exerts a hold-up force each time the number of passages 36 containing liquid under pressure changes from minimum to maximum and the balancing motors are so located and of such size and number that the sum of the hold-up forces exerted by the energized hold-up and balancing motors is slightly in excess of the total blow-off force and the resultant of the hold-up forces is approximately coincident with the center of the blow-off force when the maximum number of passages 36 contain liquid under pressure.

For the purpose of illustration, the pump has been shown in Figs. 5 and 6 as being provided with two balancing motors but the number may be varied according to requirements. One of the balancing motors includes a cylinder 53 which is formed in valve 12 behind bridge 51 and a piston 54 which is fitted in cylinder 53 and engages yoke head 5^c. The other balancing motor includes a cylinder 55 which is formed in valve 12 behind bridge 52 and a piston 56 which is fitted in cylinder 55 and engages yoke head 5^c. Ducts 57 and 58 extend, respectively, from cylinders 53 and 55 through the faces of bridges 51 and 52 into the path of passages 36 so that each balancing motor is energized each time a passage 36

containing pressure registers with the duct 57 or 58 leading to that motor.

The arrangement is such that when the pump is operating, the hold-up motors 41 or 42 on the high pressure side of the pump are continuously energized and exert the entire hold-up force when the minimum number of passages 36 contain liquid under pressure at which time the blow-off force is minimum. Each time a passage 36 passes from the high pressure port 37 or 38 onto a bridge 51 or 52, the pressure in that passage increases the magnitude and shifts the center of the blow-off force but, as soon as that passage opens slightly to the duct 57 or 58 in that bridge, the balancing motor 53—54 or 55—56 behind that bridge is energized and thereby increases the magnitude and shifts the center of the hold-up force. As soon as that passage opens slightly to the low pressure port, the blow-off force is correspondingly decreased and the balancing motor is deenergized.

Likewise, each time a passage 36 is passing from the low pressure port across a bridge 51 or 52 opens slightly to the high pressure port, the liquid in that passage is suddenly compressed and increases the magnitude and shifts the center of the blow-off force but that passage is at that time in registry with the duct 57 or 58 in that bridge so that the balancing motor 53—54 or 55—56 behind that bridge is energized and thereby increases the magnitude and shifts the center of the hold-up force until the following passage 36 registers with the same duct and thereby de-energizes the balancing motor. The hold-up force is thus increased when the blow-off force increases and is decreased when the blow-off force decreases.

When the pump is discharging liquid and is creating pressure, the blow-off force urges cylinder barrel 9 toward the left in respect to Fig. 1, and the liquid in the cylinders 16 containing pressure acts upon the inner ends of those cylinders and exerts thereon forces which may be designated as "pumping forces" and which urge cylinder barrel 9 toward the right in opposition to the blow-off force.

If the pump were supercharged, that is, if liquid were supplied to the intake of the pump at a substantial pressure, the ports and passages in valve 12 would need to be only large enough to permit the required volume of liquid to flow therethrough at a high velocity. Therefore, the valve face subjected to pressure could be kept at a minimum so that the blow-off force would be minimum and the difference between the blow-off force and the sum of the pumping forces would not be great enough to impose an excessive load upon the cylinder barrel bearings.

However, it is not practical to supercharge a pump having a substantial volumetric capacity and, in order that the pump may draw enough liquid through the valve to completely fill its cylinders, the ports and passages in the valve must be large enough to permit that liquid to flow therethrough at a low velocity. Ports and passages of such large size require that the contact face of the valve be considerably larger than would be necessary if the pump were supercharged. With the face of the valve having such a large area subjected to pressure, the blow-off force is greatly in excess of the pumping forces.

If no other means were provided, the blow-off force in excess of the sum of the pumping forces would impose so large an axial thrust upon the cylinder barrel bearings that it would be neces-

sary to employ bearings so large that the efficiency of the pump would be greatly reduced. In order to keep the efficiency as high as possible and to avoid the wear and eventual destruction resulting from imposing large axial thrusts upon mechanical bearings, the pump is provided with hydraulic means which exerts a force in opposition to the blow-off force. The hydraulic means also functions as a liquid bearing which is far more efficient than a mechanical bearing.

As shown in Figs. 1 and 2, the annular inward portion of cylinder head 9^a is engaged by an annular valve-like hold-up element 64 which extends around horn 8 and is arranged within the annular inner wall of the body portion of cylinder barrel 9. The pistons of the hold-up and balancing motors, which are arranged within element 64 as will presently be described, react against an annular abutment 65 which is fixed upon horn 8 within the annular inner wall of cylinder barrel 9 and is prevented from moving toward the left in any suitable manner such as by engagement with a shoulder formed upon horn 8. Two diametrically opposed pins 66 are fixed in abutment 65 and extend into element 64 to restrain it from radial movement while permitting the rear face of element 64 to adjust itself to the front face of cylinder head 9^a.

Element 64 is similar to but is smaller than valve 12. If it were to function also as a valve as illustrated in Fig. 25, it could be the same as valve 12 except that it would be smaller. As shown in Figs. 8 and 9, element 64 has two arcuate ports 67 and 68 formed in the rear face thereof, four holes 69 and four holes 70 extending inward from the bottoms of ports 67 and 68 respectively into communication with four hold-up motors 71 and 72 respectively. Port 67 and hold-up motors 71 in element 64 are opposed to port 37 and hold-up motors 41 in valve 12, and port 68 and hold-up motors 72 in element 64 are opposed to port 38 and hold-up motors 42 in valve 12.

Each of hold-up motors 71 and 72 includes a cylinder 73 (Fig. 9) which is formed in element 64, a piston 74 which is fitted in cylinder 73 and engages abutment 65, and a spring 75 which is arranged between piston 74 and the rear wall of cylinder 73. Springs 75 urge pistons 74 against abutment 65 and urge element 64 against cylinder head 9^a and cylinder head 9^a against valve 12.

Element 64 is also provided with balancing motors which correspond to the balancing motors 53—54 and 55—56 in valve 12. As shown in Figs. 8 and 10, one balancing motor has its piston 76 fitted in a cylinder 77 which is formed in element 64 and has a duct 78 extending therefrom through the rear face of element 64 at a point between the upper ends of ports 67 and 68, and the other balancing motor has its piston 79 fitted in a cylinder 80 which is formed in element 64 and has a duct 81 extending therefrom through the rear face of element 64 at a point between the lower ends of ports 67 and 68.

Liquid for energizing hold-up motors 71 and 72 and balancing motors 76—77 and 79—80 is supplied thereto from passages 36 through an equal number of passages 82 each of which, as shown in Figs. 9, 10 and 11, extends from a passage 36 through the front face of cylinder head 9^a. The front ends of passages 82 are arranged in a circle having the same radius as the circle upon which ports 67 and 68 (Fig. 8) are ar-

ranged, and the front end of each passage 82 is elongated to a length equal to or slightly less than the angular distance between the adjacent ends of ports 67 and 68.

The arrangement is such that, when the cylinder barrel is rotated, passages 82 will register successively with each of ports 67 and 68 and each of ducts 78 and 81, and each passage 82 will register with one port 67 or 68, then with one duct 78 or 81, then with the other port 67 or 68 and then with the other duct 78 or 81 during each revolution of the cylinder barrel.

When the pump discharges through port 38 (Fig. 5), hold-up motors 42 will be energized and urge valve 12 against cylinder head 9^a. Pressure will extend from the passages 36 on the pressure side of the pump through the passages 82 connected thereto, port 68 (Fig. 8) and holes 70 into hold-up motors 72 and cause them to urge element 64 against cylinder head 9^a in opposition to the forces exerted by hold-up motors 42. Each time a passage 36 containing pressure opens to a duct 57 or 53 (Fig. 5) a balancing motor 53-54 or 55-56 will be energized and cause valve 12 to exert additional hold-up force upon cylinder head 9^a but at the same time that passage 36 opens to duct 57 or 53, the passage 82 connected to that passage 36 opens to the duct 78 or 81 on the same side of the pump and causes the hold-up motor 77-78 or 80-81 to be energized and exert upon cylinder head 9^a a force in opposition to the force exerted thereon by balancing motor 53-54 or 55-56. When the pump discharges through port 37 (Fig. 5), the operation will be the same except that the hold-up motors on the other side of the pump will be energized.

The cylinder barrel is thus urged toward the left by the forces exerted thereon through valve 12 and it is urged toward the right by the pumping forces and the forces exerted thereon through hold-up element 64. The parts may be so proportioned that the forces which tend to move the cylinder barrel in one direction at any given instant are substantially counterbalanced by the forces which tend to move the cylinder barrel in the opposite direction at the same instant so that there will be little if any axial thrust upon cylinder barrel bearings 11 and 25. Preferably, the pump is so constructed that the sum of the forces which tend to move the cylinder barrel toward the left is slightly in excess of the sum of the forces which tend to move the cylinder barrel toward the right and the difference between the two forces imposes upon bearing 25 an axial thrust which is so slight that it does not increase the friction of the bearing to any appreciable extent.

To minimize the loads on bearings 25 and 11 upon which cylinder barrel 9 is mounted it is necessary to take account of the magnitudes and radial locations of the forces imposed on cylinder head 9^a by the pumping pistons 16, the main valve 12 and the hold-up element 64. In general, the sum of the pumping piston forces and the hold-up force, acting toward the right in respect to Fig. 1, should approximately equal the main valve force acting toward the left. Any excess is taken as an axial thrust on bearing 25 or bearing 11. Also, when the pump is discharging in one direction, the pumping piston force and the hold-up force tend to tilt the cylinder barrel clockwise, putting a downward radial load on bearing 11 and an equal upward radial load on bearing 25, and the main valve force tends

to tilt the cylinder barrel counterclockwise, loading the bearings in the opposite directions. To minimize the loads on both bearings, these opposing tilting moments should be approximately equal. The magnitude of the valve forces is conditioned by the necessary port areas.

During operation of the pump, liquid escapes in minute quantities from the high pressure ports and passages and forms a lubricating film between the contiguous faces of cylinder head 9^a and hold-up element 64. The liquid in ports 67 and 68, and passages 82 and in the lubricating film keeps cylinder head 9^a and hold-up element 64 out of metal-to-metal contact with each other and forms therebetween a liquid bearing.

It has previously been pointed out that, in a pump having a large volumetric capacity, it is not practical to restrain the cylinder barrel from axial movement solely by mechanical bearings for the reason that the blow-off force in excess of the sum of the pumping forces is so great that it would impose an excessive thrust upon the mechanical bearings. Therefore, the pump may be made in larger volumetric capacities than would be possible if the cylinder barrel were restrained from axial movement solely by mechanical bearings.

Also, since the liquid bearing is far more efficient than a mechanical bearing, a smaller pump provided with a liquid bearing would require less power to drive it than would be necessary if the cylinder barrel were restrained from axial movement solely by mechanical bearings.

Rotation of the cylinder barrel tends to cause valve 12 and hold-up element 64 to rotate. Rotation of valve 12 is prevented by two diametrically opposed pins 83 (Fig. 2) which extend into it and into the yoke head 5^c. Pins 83 support valve 12 in the proper location and they are fitted therein and in yoke head 5^c with sufficient clearance to permit valve 12 to adjust itself to cylinder head 9^a. Element 64 cannot rotate as it is supported by pins 66 from abutment 65 which is fixed upon horn 8 as by being keyed thereto.

Hydraulic circuit and trunnions

As shown in Figs. 2, 3, 4 and 23, the chambers 45 and 46 in yoke head 5^c communicate, respectively, with two passages 85 and 86 which extend longitudinally through arms 5^b and 5^a of yoke 5 and communicate, respectively, with two transverse passages 87 and 88 which are formed in the front end portions of arms 5^b and 5^a and are in axial alinement with trunnions 3 and 2 respectively. Passages 87 and 88 communicate through trunnions 3 and 2, respectively, with two annular passages 89 and 90 which are formed in opposite side walls of housing 1 around trunnions 3 and 2.

Each of trunnions 2 and 3 includes a tubular bearing member 91 which is rigidly secured in a side wall of housing 1 and has an arm of yoke 5 journaled thereon by means of bearing 6. Member 91 has two tubular pistons or bushings 92 and 93 closely fitted therein to form substantially fluid tight joints therewith and a plurality of holes 94 formed in its side wall between the adjacent ends of bushings 92 and 93 to provide communication between its interior and the annular passage 89 or 90 which extends around it.

The inner end of bushing 92 engages an annular sealing member 95 which engages the adjacent arm of yoke 5 and extends around the passage 87 or 88 therein. The abutting faces of

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member 95 and yoke 5 are made flat and smooth and the abutting faces of member 95 and bushing 92 are made spherical and smooth to form substantially liquid tight joints.

The outer end of bushing 93 is concave and in engagement with the spherical head of a bolt 96 which extends through the adjacent arm of yoke 5 and has a sealing member 97 adjustably secured to its inner end. The outer face of sealing member 97 is convex and in contact with the concave inner face of an annular seat 98 which is rigidly secured in the inner wall of the adjacent arm of yoke 5 and forms a fluid tight joint therewith. The abutting surfaces of members 97 and 98 are made spherical and smooth to form a fluid tight joint.

Bushings 92 and 93 are urged apart by a helical spring 99 which is arranged around bolt 96 and bears against the adjacent ends of bushings 92 and 93. Spring 99 urges bushing 92 against sealing member 95 and sealing member 95 against yoke 5, and it urges bushing 93 against the head of bolt 96. Since inward movement of bushing 92 and sealing member 95 is limited by yoke 5, spring 99 urges bolt 96 outward and thereby urges sealing member 97 against seat 98.

When pressure is created by the pump, it acts upon the adjacent ends of bushings 92 and 93 and augments the force exerted thereon by spring 99. The pressure acts upon the exposed area of sealing member 97 and tends to move it away from its seat 98 but the pressure also acts upon the inner end of bushing 93 and upon the exposed area of the head of bolt 96 and tends to move bolt 96 outward. The area subjected to the outward acting pressure is as great as or is somewhat greater than the area subjected to the inward acting pressure so that the forces exerted by the liquid upon the trunnions are equalized or nearly equalized. Preferably, the parts are so proportioned that the outward acting force is somewhat in excess of the inward acting force so that sealing member 97 is urged against seat 98 by spring 99 and by the difference between the two forces.

The arrangement is such as to provide in effect non-rigid extensions of yoke arms 5^a and 5^b into the respective bearing members 91. Each extension conducts liquid between a yoke arm passage 85 or 86 and an annular passage 89 or 90 and both extensions rotate within bearing members 91 when yoke 5 is swung upward or downward to vary pump displacement. The spherical members 95, 96 and 97 permit deflections of the yoke to occur without affecting the liquid seals of the hydraulic circuit or causing strain or wear on the sealing members.

Annular passages 89 and 90 communicate, respectively, with two passages 103 and 104 which are formed in the walls of housing 1 and communicate, respectively, with two ports 105 and 106 by means of which the pump may be connected to an external circuit. Passages 103 and 104 also extend through the bottom wall of housing 1 for connection to a supply of liquid such as reservoir 4.

As shown in Fig. 23, communication between reservoir 4 and passages 103 and 104 may be controlled by an automatic suction valve 107 having a valve plunger 108 fitted in the bore 109 of a valve casing 110 which may be attached to the bottom wall of housing 1. Valve 107 has two ports 111 and 112 which communicate with passages 103 and 104, respectively, and which also communicate, respectively, through two passages 113 and 114 with opposite ends of bore 109. Valve

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107 also has a port 115 arranged between ports 111 and 112 to communicate with a check valve 116 and a resistance valve 117 which open in opposite directions. Check valve 116 permits liquid to be drawn freely into bore 109 from reservoir 104 but it prevents liquid from being discharged into reservoir 104 except through resistance valve 117.

If yoke 5 is tilted upward as shown in Fig. 1, the pump when started will discharge liquid through valve 12, holes 44, chamber 46, passage 86, trunnion 2, passages 90 and 104 and port 106 to the external circuit. If valve plunger 108 is in its right hand position, the resistance in the circuit will cause pump pressure to rise and liquid to flow through passage 114 to the right end of bore 109 and shift valve plunger 108 to the position shown so that the pump cannot discharge through valve 107 into reservoir 4 when pump pressure becomes high enough to open resistance valve 117.

The liquid returned from the circuit flows through port 105, passages 103 and 89, trunnion 3, passage 85, chamber 45, holes 43 and valve 12 to cylinder barrel 9. If the liquid returned from the circuit is less than pump requirements, enough liquid to make up the deficiency will be drawn from reservoir 4 through check valve 116 and suction valve 107 into passage 103 where it joins the liquid returned from the circuit. If the liquid returned from the circuit is in excess of pump requirements, the excess liquid will be discharged through passage 103, suction valve 107 and resistance valve 117 into reservoir 4. If yoke 5 is tilted downward, the flow will be reversed.

The piston assemblies

The piston assemblies chosen for illustration are of the general form shown in Fig. 6 of Patent No. 2,638,850 but they may be one of the other forms shown in that application to which reference may be had for details of construction and an explanation of the advantages of this type of piston assembly.

As shown in Fig. 12, each piston 16 has a bore 121 extending inward from its front end and an annular flange 122 extending around its front end. A connecting rod 123 extends into bore 121 and engages the rear wall thereof. Preferably, the end of rod 123 is spherical and the end wall of bore 121 is flat so that rod 123 can rock upon and make a spot contact with the end wall of bore 121 as the angle between the axis of piston 16 and the face of thrust member 18 changes during rotation of cylinder barrel 9 and thrust member 18.

As shown in Figs. 12 and 13, each ball and socket joint 17 includes a ball 124, which has its shank 125 fixed in thrust member 18, and a spherical socket 126 which is fitted to ball 124 and is formed upon connecting rod 123. Ball 124 and socket 126 are arranged within two annular sockets 127 and 128 having projections 129 and 130 formed upon the respective ends thereof so that projections 129 on socket 127 will fit between the projections 130 on socket 129, projections 130 on socket 128 will fit between the projections 129 on socket 127, and the sockets may be fastened together by a snap ring 131 fitting in an annular groove 132 formed in projections 129 and 130.

The end wall of socket 127 is provided with a spherical surface 133 complementary to and in contact with ball 124, and the end wall of

socket 128 is provided with a spherical surface 134 to engage a complementary spherical surface formed upon a collar 135 which is slidable upon piston 16. Suitable springs means, such as a stack of Belleville washers 136, is arranged between collar 135 and the flange 122 on connecting rod 123 to yieldingly hold the parts together and enable the ball and socket joint to pull piston 16 outward on a suction stroke.

The arrangement is such that, during one-half of each revolution of cylinder barrel 9 and thrust member 18, piston 16 is forced into its cylinder by force transmitted thereto through rod 123, socket 126, ball 124 and its shank 125 from thrust member 18, and during the other half of each revolution of cylinder barrel 9 and thrust member 18, piston 16 is retracted on a suction stroke by force transmitted thereto through flange 122, spring means 136, collar 135, sockets 127 and 128, ball 124 and its shank 125 from thrust member 18.

There is substantially no rubbing movement between the end of connecting rod 123 and the end wall of bore 121 so that it is not necessary to maintain a pressure lubricating film therebetween when piston 16 is creating pressure. Consequently, a very large force equal to the bearing value of the metal may be transmitted through rod 123 to piston 16 so that a pump provided with piston assemblies of the type shown is capable of creating pressures greatly in excess of the pressures which the prior pumps of the same general type could create. However, lubrication of these surfaces may be accomplished as explained in the patent referred to above.

Socket 126 makes a rubbing contact with ball 124 so that a lubricating film of liquid must be maintained therebetween and pressure must be maintained in the film during the time that piston 16 is creating pressure in its cylinder but, since the ball and socket joint is located outside the cylinder, ball 124 and socket 126 may be made of such size that the area of the lubricating film between them is ample to transmit sufficient force to create the desired high pressure in the cylinder.

Liquid for lubricating each ball and socket joint 17 is supplied thereto through a port 137 (Fig. 14) which is formed in the front face of thrust member 18 substantially in radial alignment with that joint and which communicates through a duct 138 with an annular groove 139 (Fig. 12) formed in the shank 125 of that joint. Groove 139 communicates with a duct 140 which extends therefrom through ball 124 into communication with a recess 141 formed in socket 126 upon the axis of rod 123.

In order that each ball and socket joint may be supplied with liquid under pressure only during the time that the piston 16 connected thereto is creating pressure in its cylinder, each port 137 is connected to the high pressure side of the hydraulic circuit during the discharge stroke of the piston and it is connected to the low pressure side of the circuit during the suction stroke of the piston as will presently be explained.

The arrangement is such that, when a piston 16 is creating pressure in its cylinder, the port 137 in radial alignment with that piston is open to the high pressure side of the circuit and liquid at the same pressure will flow from the port 137 through duct 138, groove 139 and duct 140 into recess 141 and form a lubricating film between head 124 and socket 126. The pressure will ex-

tend into recess 141 and into the lubricating film when the piston is creating pressure in its cylinder but there will be no pressure therein when there is no pressure in the cylinder.

The liquid in this lubricating film and in recess 141 when under pressure will exert a "blow-off force" which tends to move socket 126 away from ball 124. In order that this blow-off force may not become so great that it will force the mating surfaces of ball 124 and socket 126 apart and thus permit substantial leakage loss of the working liquid, an annular pressure limit groove 142 having a diameter greater than piston 16 is cut around socket 126 in a plane normal to the axis of connecting rod 123 and a drain duct 143 is extended from groove 142 outward through socket 126 to permit liquid to escape freely from groove 142. Pressure in the film is thus limited to the area between recess 141 and groove 142 and the film in this area may be designated as a "pressure lubricating film" but there is no pressure in the lubricating film between ball 124 and the outer part of socket 126 nor in the lubricating film between ball 124 and socket 127. The ball and socket joint is thus adequately lubricated and it is capable of transmitting very high pumping forces as explained in the application referred to above.

The main bearing thrust limiting mechanism

It has previously been explained that, when yoke 5 is tilted either upward or downward, pistons 16 are reciprocated in their cylinders by thrust member 18. The forces required to enable pistons 16 to create pressure in their cylinders react upon thrust member 18 and tend to move it and drive shaft 21 forward or toward the left in respect to Fig. 1. If no other means were provided, the thrust imposed upon bearings 19 and 33 would necessitate the use of larger bearings and result in reduced efficiency of the pump.

However, the present invention provides a hydraulic hold-up element which counterbalances the greater part of the pumping forces and forms a liquid thrust bearing which is far more efficient than a mechanical thrust bearing. The hold-up element is similar to and functions in the same manner as the previously described hold-up element 64 and it has the additional function of connecting the ports 137 in thrust member 18 to opposite sides of the hydraulic circuit alternately as thrust member 18 rotates.

As shown in Figs. 1 and 2, the front face of thrust member 18 engages the rear face of an annular hold-up element 150 which extends around drive shaft 21 and is restrained from rotary and radial movement by two pins 151 inserted into it and into gear pump cover 31. Pins 151 are fitted with sufficient clearance to permit element 150 to adjust its rear face to the front face of thrust member 18.

Each port 137 (Fig. 14) in thrust member 18 is adapted to register alternately with two diametrically opposed arcuate ports 153 and 154 (Figs. 15 and 16) which are formed in the rear face of hold-up element 150 and have the adjacent ends thereof spaced apart a distance equal to or slightly greater than the length of a port 137 to provide a seal between ports 153 and 154 when a port 137 passes from one to the other of ports 153 and 154 during rotation of thrust member 18.

As ports 137 pass from one to the other of ports 153 and 154, they communicate with two holes 155 and 156 each of which is formed in

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element 150 at a point mid-way between the adjacent ends of ports 153 and 154. Hole 155 extends forward and radially outward into communication with a cylinder 157 (Fig. 1) having a piston 158 fitted therein to engage gear pump cover plate 31. Each hole 156 extends forward and radially outward into communication with a cylinder 159 having a piston 160 fitted therein to engage cover plate 31. Cylinders 157 and 159 and pistons 158 and 160 constitute balancing motors which correspond to the balancing motors 53—54 and 55—56 in value 12 (Fig. 6).

Port 153 communicates through four holes 161 (Fig. 15) with four hold-up motors 162 and port 154 communicates through four holes 163 with four hold-up motors 164. Hold-up motors 162 and 164 are alike and, as shown in Fig. 16, each includes a cylinder 165 which extends into hold-up element 150 from the front face thereof, a tubular piston 166 which is fitted in cylinder 165, an annular sealing member 167 which is arranged between piston 166 and the rear face of gear pump cover plate 31, and a spring 168 to hold element 150 against thrust member 18 and to hold piston 166 against sealing member 167 and sealing member 167 against cover plate 31. The abutting faces of sealing member 167 and cover plate 31 are made flat and smooth and the abutting faces of piston 166 and sealing member 167 are made spherical and smooth to form fluid tight joints.

As shown in Figs. 1, 15, 16 and 23 liquid under pressure is supplied to ports 153 and 154 selectively through two U-shaped passages 169 and 170 which are formed in cover plate 31 with the upper and lower ends thereof extending through the rear face of cover plate 31. The upper end of passage 169 communicates with port 153 through one of the hold-up motors 162 and a hole 161. The upper end of passage 170 communicates in a similar manner with port 154 through one of the hold-up motors 164 and a hole 163.

The lower end of passage 169 communicates through a connection 171 with a passage 172 which is formed in web 20 and is connected to passage 103 by a passage 173 as shown in Fig. 1. The lower end of passage 170 communicates through a connection 174 with a passage 175 which is formed in web 20 and is connected to passage 104 by a passage 176 as indicated in Figs. 15 and 23.

The two connections 171 and 174 are alike and, as shown in Fig. 17, each includes a cylinder 177 which is formed in web 20 and communicates with a passage 172 or 175, a piston 178 which is fitted in cylinder 177, a sealing member 179 which is arranged between piston 178 and the face of cover plate 31, and a spring 180 to urge piston 178 against sealing member 179 and sealing member 179 against cover plate 31. The abutting faces of sealing member 179 and plate 31 are made flat and smooth and the abutting faces of sealing member 179 and piston 178 are made spherical and smooth to provide a substantially fluid tight joint between plate 31 and web 20.

The arrangement is such that, when pressure is created in passage 103, the pressure extends therefrom through passages 173 and 172, connection 171, passage 169, one of the hold-up motors 162 and a hole 161 into port 153 and then the pressure extends through the other three holes 161 into the other three motors 162 so

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that all four hold-up motors 162 are energized. Likewise, when pressure is created in passage 104, the pressure extends therefrom through passages 176 and 175, connection 174, passage 170, one of the hold-up motors 164 and a hole 163 into port 154 and then the pressure extends through the other three holes 163 into the other three motors 164 so that all four of the motors 164 are energized.

When yoke 5 is tilted upward as shown in Fig. 1 and drive shaft 21 is rotated in a counter-clockwise direction in respect to Fig. 3, pistons 16 are forced progressively inward as they move around the right hand side of the cylinder barrel axis. The pistons in the right hand side of the cylinder barrel, that is the inward moving pistons, eject liquid from their cylinders through valve 12, holes 44, chamber 46, passage 38, trunnion 2 and passages 90 and 104 to the external circuit.

Any resistance to the flow of liquid in the circuit causes the pistons 16 in the right hand side of the cylinder barrel to create pressure, and the pumping forces necessary to enable those pistons to create that pressure impose an axial thrust upon the right hand side of thrust member 18 but the same pressure energizes hold-up motors 164 and causes them to urge hold-up element 150 against thrust member 18.

Also, the same pressure extends into the ports 137 in the right hand side of thrust member 18 so that the ball and socket joints connecting thrust member 18 to the pistons in the right hand side of cylinder barrel 5 are provided with pressure lubrication as explained above.

It has previously been explained that the number of cylinders 16 containing liquid under pressure changes by one each time a passage 36 passes from one to the other of the two ports 37 and 38 in valve 12 (Fig. 5). Therefore, the total pumping force changes in magnitude and its center shifts each time there is a change in the number of cylinders 16 containing liquid under pressure.

Hold-up motors 164 (Figs. 15 and 16) are so proportioned and located that the total hold-up force exerted thereby is in opposition to but preferably slightly less than the total pumping force. Each time that the number of cylinders 15 containing liquid under pressure changes from minimum to maximum, a port 137 containing liquid under pressure opens to a hole 155 or 156 and effects energization of a balancing motor 157—158 or 159—160 which urges hold-up element 150 against thrust member 18 and thereby causes the total hold-up force to increase in magnitude and its center to shift in the same direction that the center of the total pumping force shifts. When yoke 5 is tilted downward, the flow of liquid will be reversed, the hold-up motors on the left hand side of the pump will be energized, and the ball and socket joints on the left hand side of the pump will be provided with pressure lubrication.

The liquid supplied to hold-up element 150 escapes at a minute rate from the high pressure port 153 or 154 and forms a lubricating film between it and thrust member 18. The liquid in this film and in ports 153, 154 and 137 constitutes a liquid thrust bearing which is far more efficient than a mechanical thrust bearing.

The arrangement is such that element 150 exerts upon thrust member 18 a hold-up force which is at all times nearly as great as the thrust imposed thereon by the pumping forces so that only a small axial thrust is imposed upon bearings 19 and 33 which thus need be only large enough to

rotatably support thrust member 18 and shaft 21 and to withstand the small axial thrust. Therefore, the bearing losses are much less and the efficiency of the pump is considerably greater than if the entire axial thrust were imposed upon the mechanical bearings as is the case in the prior pumps.

Universal joint lubrication

It has previously been explained that liquid escapes at a minute rate from the high pressure port 153 or 154 (Fig. 15) in hold-up element 150 and forms a lubricating film between thrust member 18 and hold-up element 150. The liquid which escapes radially inward from the high pressure port 153 or 154 may be employed to lubricate universal joint 22.

As shown, escape of this liquid along shaft 21 is prevented by thrust member 18 being closely fitted thereon and by an "O-ring" packing 185 (Fig. 16) which encircles shaft 21 and extends into a suitable groove formed in hold-up element 150. A groove 166 (Fig. 1) formed in shaft 21 communicates through a hole 187 with a bore 188 which extends inward from the right end of shaft 21 and has the hub 189 of universal joint 22 closely fitted therein. A hole 190 extends through hub 189 to the interior of universal joint 22.

The inward leaking liquid flows through groove 186, hole 187, bore 188 and hole 190 into the interior of joint 22 to lubricate it. Liquid leaking into bore 188 in excess of requirements escapes therefrom through a low pressure relief valve 191, which is fixed in bore 188 and is spaced from hub 189, and a hole 192 which extends outward from bore 188 at a point at the left of packing 185.

The displacement varying mechanism

It has previously been explained that pump displacement is zero when the axis of yoke 5 is coincident with the axis of shaft 21 and that the pump will discharge liquid in a direction and at a rate dependent upon the direction and the angular distance yoke 5 is swung upward or downward upon trunnions 2 and 3 from its neutral position.

Yoke 5 is adapted to be swung upward or downward upon trunnions 2 and 3 and to be held in adjusted positions by two hydraulic servo-motors the larger one of which includes a cylinder 200 (Fig. 19) which is formed in housing 1, a piston 201 which is fitted in cylinder 200 and has a hollow shank 202 slidable through the inner end of cylinder 200, and a connecting rod 203 which has one end arranged within shank 202 and connected thereto by a pin 204 and its other end bifurcated and connected by a pin 205 (Figs. 3 and 19) to a lug 206 formed upon the front end of yoke arm 5^a.

The smaller servo-motor includes a cylinder 210 (Fig. 20) which is formed in housing 1, a piston 211 which is fitted in cylinder 210 and has a hollow shank 212 slidable through the inner end of cylinder 210, and a connecting rod 213 which has one end arranged within shank 212 and connected thereto by a pin 214 and its other end bifurcated and connected by a pin 215 (Figs. 3 and 20) to a lug 216 formed upon the front end of yoke arm 5^b.

The overhanging weight of yoke 5 and the parts carried thereby is counterbalanced in part by a spring assembly 220 (Fig. 20) which is so constructed that it will support yoke 5 in its neutral position when servo-motors 200-201 and 210-211 are de-energized.

As shown, a helical compression spring 221 extends around a cylinder 222 with one of its ends bearing upon a flange 223 on the outer end of

cylinder 222 and its other end bearing upon a flange 224 formed upon a push rod 225 which has one end portion made cylindrical and fitted in cylinder 222 and its other end portion bifurcated and provided with a pin 226. The left hand end of yoke arm 5^b has a portion 227 which is reduced in thickness to fit into the bifurcated end of push rod 225, and a semi-circular recess 228 is formed in portion 227 at a point below the trunnion axis. Pin 226 bears in recess 228 and the left hand end of cylinder 222 bears upon an adjusting screw 229 which is threaded through the front plate 230 of housing 1 so that spring 221 tends to swing yoke 5 upward but it is not strong enough to do so. The left end of push rod 225 is engaged by one end of a helical compression spring 231 having the other end thereof bearing upon an abutment 232 which is connected to push rod 225 by a bolt 233 and bears against the end wall of cylinder 222 when yoke is at or below central position.

Springs 221 and 231 are so adjusted by means of screw 229 and bolt 233 that the combined resistances of the two springs is just sufficient to hold yoke 5 in its horizontal or neutral position when servo-motors 200-201 and 210-211 are de-energized. When the head end of cylinder 200 is supplied with motive liquid, piston 201 will swing yoke 5 upward upon trunnions 2 and 3, spring 221 will assist piston 201 in swinging yoke 5 upward and in holding it in adjusted positions, and push rod 225 will move toward the right in cylinder 222 and thereby move abutment 232 away from the end wall of cylinder 222.

When the head end of cylinder 200 is opened to exhaust, yoke 5 will be swung downward by gravity and by servo-motor 210-211 which has been shown as being continuously energized when the pump is in operation. Just as or just before yoke 5 reaches its horizontal or neutral position, abutment 232 engages the end wall of cylinder 222 so that spring assembly 220 will support yoke 5 in its neutral position if or when both servo-motors are de-energized. Thereafter, if servo-motor 200-201 is de-energized and servo-motor 210-211 is energized, yoke 5 will be swung downward below its horizontal position by gravity and by servo-motor 210-211 against the resistance of spring assembly 220.

Liquid for energizing servo-motors 200-201 and 210-211 may be supplied from any suitable source such as gear pump 28-29 which has been shown in Fig. 24 as drawing liquid from reservoir 4 through a channel 240 and discharging it into a channel 241. Liquid discharged by gear pump 29-30 in excess of requirements is exhausted through a low pressure relief valve 242 into reservoir 4. For the purpose of illustration, servo-motor 210-211 has been shown as being continuously supplied with motive liquid and servo-motor 200-201 has been shown as being supplied with liquid under the control of a pilot valve but the flow of liquid to and from both servo-motors may be controlled if desired.

As shown, channel 241 is connected to cylinder 210 by a channel 243 so that servo-motor 210-211 is continuously energized when the pump is operating. The flow of liquid to and from cylinder 200 is controlled by a pilot valve 244 (Fig. 22) fitted in a sleeve 245 which is fixed in housing 1. For the purpose of illustration, sleeve 245 has been shown as having an annular groove or control port 246 formed in its inner wall approximately mid-way between its ends, an annular groove or pressure port 247 formed in its inner wall at one side of port 246 and a plurality

of drain holes 248 extending therethrough at the other side of port 246 but in practice sleeve 245 is of somewhat different construction.

Port 247 communicates with a channel 243 which is formed in housing 1 and which, as shown in Fig. 24, is connected by a channel 250 to channel 243 so that port 247 is continuously supplied with motive liquid by gear pump 28—29 when the pump is operating. Port 246 communicates with a channel 251 (Figs. 21 and 22) which is formed in housing 1 and is connected to cylinder 200 by a channel 252 which is formed in front plate 230. Holes 248 open directly to the interior of housing 1 whence liquid is drained into reservoir 4 but for the purpose of illustration Fig. 24 shows one of the holes 248 connected to a channel 253 which discharges into reservoir 4.

The central head or piston of valve 244 is substantially the same width as port 246 and blocks it when valve 244 is in its neutral position. Since port 247 is continuously supplied with motive liquid as explained above, the arrangement is such that, when pilot valve 244 is shifted in one direction, liquid may flow from port 247 through sleeve 245, port 246 and channels 251 and 252 to cylinder 200 and cause piston 201 to swing yoke 5 upward. Servo-motor 210—211 will resist the upward swing of yoke 5 but servo-motor 200—201 is enough larger than servo-motor 210—211 to overcome the resistance. When pilot valve 244 is shifted in the opposite direction, liquid may escape from cylinder 200 through channels 252 and 251, port 246, sleeve 245 and holes 248, thereby permitting yoke 5 to be swung downward by gravity and by servo-motor 210—211 which is continuously energized as explained above.

Pilot valve 244 may be controlled by the mechanism shown schematically in Fig. 24. As shown, pilot valve 244 has its stem 260 pivoted to one end of a lever 261 which may have its other end pivoted upon a stationary pin but which has been shown as being held against a fulcrum 262 by a spring 263. Lever 261 is pivoted at a point intermediate stem 260 and fulcrum 262 to one end of a link 264 having the other end thereof pivoted to one end of floating lever 265. The other end of lever 265 is pivoted to a block 266 adjustably fixed upon a rod 267 which is slidably mounted in a plurality of guides 268 carried by housing 1.

Rod 267 has a second block 269 adjustably fixed thereon adjacent one end thereof. Block 269 is pivotally connected to lug 216 on yoke 5 by means of a pin 270 which has been shown in Fig. 3 as being secured to or formed integral with pin 215. All of the slack in the joints between levers 261 and 265, link 264, rod 267 and yoke 5 is taken up by a spring 271 which urges levers 261 and 265 toward each other.

Lever 265 has a pin 272 fixed therein intermediate the ends thereof and a roller 273 is journaled upon pin 272 to engage a cam 274 fixed to or formed integral with a cam plate 275 which is fixed for rotation with a shaft 276 journaled in a portion of housing 1. The control mechanism is preferably arranged within housing 1 and shaft 276 extends through the wall thereof and has a hand lever 277 fixed upon its outer end.

The outer or convex face of cam 274 in contact with roller 273 has a short central or neutral portion 278 which is concentric with shaft 276, a portion 279 which extends from one end of portion 278 and has a gradually increasing radius and a portion 280 which extends from the other

end of portion 278 and has a gradually decreasing radius.

Roller 273 is held in contact with cam 274 by a lever 281 which is pivoted intermediate its ends upon a pin 282 carried by lever 265. One end of lever 281 carries a roller 283 to engage the concave inner face of cam 274 and the other end of lever 281 is urged toward lever 265 by a spring 284 which thus urges roller 273 against cam 274.

The arrangement is such that, when hand lever 277 is swung in a counterclockwise direction from its neutral position, cam 274 will swing the lower end of lever 265 toward the left in respect to Fig. 24, link 264 will swing the lower end of lever 261 toward the left and lever 261 will shift pilot valve 244 toward the left to connect port 246 to port 247. Then liquid from gear pump 28—29 will flow through channels 241, 243, 250 and 249, port 247, sleeve 245, port 246 and channels 251 and 252 to cylinder 200 and cause piston 201 to swing yoke 5 upward, thereby causing the pump to discharge liquid in one direction as previously explained.

As yoke 5 swings upward, pin 270 will shift rod 267 toward the left and block 266 on rod 267 will move the upper end of lever 265 toward the left as fast as yoke 5 swings and thereby tend to close port 246 as fast as lever 277 opens it. Therefore, movement of yoke 5 ceases substantially as soon as movement of lever 277 ceases and the displacement of the pump is determined by the position of lever 277.

When lever 277 is swung in a clockwise direction, cam 274 will swing the lower end of lever 265 toward the right in respect to Fig. 24, link 264 will swing the lower end of lever 261 toward the right and lever 261 will shift pilot valve 244 toward the right to connect port 246 to holes 248. Then liquid can escape from cylinder 200 through channels 252 and 251, port 246, sleeve 245 and holes 248 to exhaust, thereby permitting servo-motor 210—211 to swing yoke 5 downward. If yoke 5 has previously been swung upward as explained above, downward movement thereof will reduce pump displacement. If lever 277 is swung far enough to cause roller 273 to ride along cam face portion 279, yoke 5 will be swung downward below its horizontal position and the pump will discharge liquid in the opposite direction.

As yoke 5 swings downward, pin 270 will shift rod 267 toward the right and block 266 on rod 267 will move the upper end of lever 265 toward the right as fast as yoke 5 swings downward and thereby tend to close port 246 as fast as lever 277 opens it. Therefore, movement of yoke 5 ceases as soon as movement of lever 277 ceases and the displacement of the pump is determined by the position of lever 277.

Blocks 266 and 269 are initially adjusted as closely as possible in such positions that roller 273 contacts cam 274 at the center of cam face portion 278 when yoke 5 is in its horizontal or neutral position. Since all of the control mechanism except lever 277 is arranged with housing 1, adjustment can not readily be made after the pump is fully assembled. Therefore, abutment 262 is preferably a cam fixed upon a shaft 285 which extends through the wall of housing 1 so that the cam may be turned from the outside of housing 1 to effect a final accurate adjustment of the control mechanism.

When the pump is discharging liquid under pressure, the forces transmitted from thrust member 18 through the piston assemblies to the

liquid in the cylinders on the high pressure side of the pump tend to move cylinder barrel 9 and yoke 5 toward the right in respect to Fig. 1, thereby imposing upon trunnions 2 and 3 a thrust equal to the sum of the pumping forces.

If the pump were not too large and were to operate at moderate pressures, the right hand ends of cylinders 200 and 210 could be open to exhaust or be connected to each other as the pumping forces would not be great enough to overload the trunnions or to cause deflection of the parts.

However, the pump illustrated in the drawings has a very large volumetric capacity and it is capable of creating very high pressures at which pressures the pumping forces are very great. In order to prevent the pumping forces from imposing excessive thrusts upon the trunnions and to avoid the possibility of deflection of parts due to the high pumping forces, the present invention provides hydraulic means for opposing the pumping forces and thereby reducing the thrusts upon the trunnions.

This may be accomplished by exposing the right hand faces of pistons 201 and 211 to pump pressure. As shown schematically in Fig. 24, the right hand ends of cylinders 200 and 210 are connected to each other by a channel 290 so that liquid may flow from one to the other when pistons 201 and 211 are shifted in opposite directions to swing yoke 5 upward or downward as explained above. Channel 290 is connected to channel 103 (Fig. 23) by a channel 291 having a check valve 292 arranged therein, and channel 290 is connected to channel 104 by a channel 293 having a check valve 294 arranged therein. Check valves 292 and 294 permit liquid to flow into channel 290 from the channel 103 or 104 which at that time is the discharge channel but they prevent liquid from flowing from channel 290 into the channel 103 or 104 which at that time is the intake channel.

The arrangement is such that, when the pump is creating pressure, the right hand faces of pistons 201 and 211 are subjected to pump pressure which causes servo-motors 200-201 and 210-211 to exert forces upon yoke 5 in opposition to the pumping forces. The right hand faces of pistons 201 and 211 have the same area so that the forces exerted by the two servo-motors are equal.

Since servo-motors 200-201 and 210-211 are necessarily connected to the arms 5^a and 5^b of yoke 5, the forces exerted thereby are offset laterally inward from yoke bearings 6 and are offset laterally outward from the resultant of the pumping forces which is offset laterally outward from the yoke axis a little more than one-half of the radius of the piston circle. There is a slight lateral movement of the resultant of the pumping forces each time there is a change in the number of cylinders containing pressure but such movement may be disregarded.

In the pump shown, the distance between the resultant of the pumping forces and the center of the adjacent yoke bearing 6 is about three-eighths of the distance between the centers of yoke bearings 6. Therefore, the thrust exerted upon the adjacent bearing 6 is equal to about five-eighths of the sum of the pumping forces and the thrust exerted upon the far bearing 6 is equal to about three-eighths of the sum of the pumping forces. If servo-motors 200-201 and 210-211 are so proportioned that each exerts a force equal to one-half of the sum of the pumping forces, one servo-motor will reduce the thrust upon the bearing 6

on the discharge side of the pump to an amount equal to about one-eighth of the sum of the pumping forces and the other servo-motor will over-balance the thrust upon the bearing 6 on the intake side of the pump by an amount equal to about one-eighth of the sum of the pumping forces.

FIG. 25

This figure shows a portion of a pump which is the same as the pump described above except that it is provided with an auxiliary flat valve, which together with the main valve controls the flow of liquid to and from the cylinders, and with additional passages for the flow of liquid to and from the auxiliary valve.

More specifically, an auxiliary flat valve 64^a is substituted for the valve-like hold-up element 64 shown in Figs. 1, 2 and 8, and yoke end head 5^c, horn 8, cylinder head 9^a and abutment 65 are replaced, respectively, with a yoke and head 5^c, a horn 8^a, a cylinder head 9^a and an abutment 65^a having passages therein to provide with valve 64^a additional paths for the flow of liquid to and from cylinders 15. The other parts shown are the same as the corresponding parts of the previously described pump and they have been indicated by the same reference numerals so that further description thereof is unnecessary.

Yoke end head 5^c and horn 8^a are the same, respectively, as yoke end head 5 and horn 8 except that yoke end head 5^c has two passages 390 and 391 formed therein and extending through its inner wall from chambers 45 and 46, respectively, and horn 8^a has two passages 302 and 303 formed therein with the rear ends thereof in registry with passages 300 and 301 respectively. The front ends of passages 302 and 303 extend outward through the periphery of horn 8^a and communicate, respectively, with two chambers 304 and 305 formed in abutment 65^a which is tightly pressed upon horn 8^a and corresponds to the abutment 65 shown in the preceding figures.

Also, cylinder barrel 9 is provided with an end head 9^a which is the same as end head 9 except that its passages 82^a are larger in diameter than passages 82 between their elongated front ends and passages 36. In order that the elongated front ends of passages 82^a may properly cooperate with the ports in valve 64^a, they are the same size as shown in Fig. 11.

Valve 64^a has formed in the face thereof two arcuate grooves or ports 67 and 68 which are the same as the ports 67 and 68 shown in Fig. 8. Port 67 communicates through a plurality of holes 69^a with an equal number of hold-up motors 71^a, and port 68 communicates through an equal number of holes 70^a with an equal number of hold-up motors 72^a. Since hold-up motors 71^a and 72^a are identical in structure and function to hold-up motors 41 and 42 (Fig. 7) except that they have been shown as being smaller, a description thereof is deemed unnecessary. Each hold-up motor 71^a communicates with a passage 306 which is formed in abutment 65^a and communicates with chamber 304, and each hold-up motor 72^a communicates with a passage 307 which is formed in abutment 65^a and communicates with chamber 305. Valve 64^a is otherwise the same as hold-up element 64 and it performs the same functions in addition to providing additional paths for the flow of liquid to and from cylinder 15.

The pump functions in the same manner as the previously described pump except that liquid

flows to and from cylinders 15 in part through main valve 12 and in part through auxiliary valve 64^a.

That is, if the pump is adjusted to deliver liquid in one direction, a part of the liquid discharged from the cylinders 15 on one side of the pump will flow through passages 36 and main valve 12 to chamber 46 as previously explained and another part thereof will flow through passages 36 and 82^a, port 68, holes 73^a, hold-up motors 72^a, passages 307, chamber 305 and passages 303 and 301 to chamber 46. At the same time, the cylinders 15 on the other side of the pump will be supplied with liquid from chamber 45 in part through main valve 12 as previously explained and in part through passages 309 and 302, chamber 304, passages 306, hold-up motors 71^a, holes 69^a, port 67 and passages 82^a and 36. If the pump is adjusted to deliver liquid in the opposite direction, the liquid will flow through the same channels but in the opposite direction.

By diverting a part of the flow through auxiliary valve 64^a, the ports and passages in main valve 12 may be made smaller and permit the contact face of valve 12 to be made smaller, thereby reducing the forces exerted upon the cylinder barrel.

The invention herein set forth is susceptible of various other modifications and adaptations without departing from the scope of the invention which is hereby claimed as follows:

1. In a hydrodynamic machine, the combination of a rotatable cylinder barrel having pistons and cylinders arranged therein, means for rotating said cylinder barrel, means for reciprocating said pistons in response to rotation of said cylinder barrel to thereby cause liquid to flow into and out of said cylinders, a flat valve engaging the end of said cylinder barrel to control the flow of liquid to and from said cylinders, the liquid between said cylinder barrel and said valve tending to move said cylinder barrel away from said valve, a hold-up element arranged between the ends of said cylinder barrel and engaging said cylinder barrel in opposition to said valve, hold-up motors arranged upon opposite sides of the centerline of said cylinder barrel for urging said element against said cylinder barrel, and means for supplying liquid from the cylinders on either side of said centerline to the hold-up motors on the same side of said centerline to energize the same.

2. In a hydrodynamic machine, the combination of a rotatable cylinder barrel having pistons and cylinders arranged therein, means for rotating said cylinder barrel, means for reciprocating said pistons in response to rotation of said cylinder barrel to thereby cause liquid to flow into and out of said cylinders, a flat valve engaging the end of said cylinder barrel to control the flow of liquid to and from said cylinders, the liquid between said cylinder barrel and said valve tending to move said cylinder barrel away from said valve, a hold-up element arranged between the ends of said cylinder barrel and engaging said cylinder barrel in opposition to said valve, hold-up motors arranged upon opposite sides of the centerline of said cylinder barrel for urging said element against said cylinder barrel, means for supplying liquid from the cylinders on either side of said centerline to the hold-up motors on the same side of said centerline to energize the same and also to form between said hold-up element and said cylinder barrel a lubri-

cating film of liquid to thereby provide a liquid thrust bearing for said cylinder barrel.

3. In a hydrodynamic machine, the combination of a rotatable cylinder barrel having pistons and cylinders arranged therein, means for rotating said cylinder barrel, means for reciprocating said pistons in response to rotation of said cylinder barrel to thereby cause liquid to flow into and out of said cylinders, a flat valve engaging the end of said cylinder barrel to control the flow of liquid to and from said cylinders, an abutment, and hydraulic hold-up motors arranged within said valve and energized by the liquid flowing therethrough, said motors when energized reacting against said abutment and urging said valve against said cylinder barrel.

4. In a hydrodynamic machine, the combination of a rotatable cylinder barrel having pistons and cylinders arranged therein, means for rotating said cylinder barrel, means for reciprocating said pistons in response to rotation of said cylinder barrel to thereby cause liquid to flow into and out of said cylinders, a main flat valve engaging the end of said cylinder barrel and having two ports with which each of said cylinders communicates alternately, and an auxiliary flat valve arranged between the ends of said cylinder barrel and engaging said cylinder barrel in opposition to said main valve and having two ports with which each of said cylinders communicates alternately, said valves together controlling the flow of liquid to and from said cylinders.

5. In a hydrodynamic machine, the combination of an abutment arranged adjacent the rear end of said machine and restrained from axial movement, an annular cylinder barrel arranged in front of said abutment and having an end head which extends inward beyond the inner wall thereof, pistons and cylinders arranged within said cylinder barrel, means for rotating said cylinder barrel, means for reciprocating said pistons in response to rotation of said cylinder barrel to thereby cause liquid to flow into and out of said cylinders, a flat valve arranged between said abutment and said end head in contact with the rear face of said end head to control the flow of liquid to and from said cylinders, a hold-up element engaging the front face of said end head, hold-up motors arranged upon opposite sides of the centerline of said cylinder barrel for urging said element against said end head, and means for supplying liquid from the cylinders on either side of said centerline to the hold-up motors on the same side of said centerline to energize the same.

6. In a hydrodynamic machine, the combination of an abutment arranged adjacent the rear end of said machine and restrained from axial movement, an annular cylinder barrel arranged in front of said abutment and having an end head which extends inward beyond the inner wall thereof, pistons and cylinders arranged within said cylinder barrel, means for rotating said cylinder barrel, means for reciprocating said pistons in response to rotation of said cylinder barrel to thereby cause liquid to flow into and out of said cylinders, a main flat valve engaging the rear face of said end head and having two ports with which each of said cylinders communicates alternately, an auxiliary flat valve engaging the front face of said end head and having two ports with which each of said cylinders communicates alternately, said valves together controlling the flow of liquid to and from said cylinders, hold-up motors ar-

ranged upon opposite sides of the centerline of said cylinder barrel for urging said auxiliary valve against said end head, and means for supplying liquid from the cylinders on either side of said centerline to the hold-up motors on the same side of said centerline to energize the same.

7. In a hydrodynamic machine, the combination of an abutment arranged adjacent the rear end of said machine and restrained from axial movement, a horn rigidly secured in said abutment and extending forward therefrom, an annular cylinder barrel extending around and rotatably supported upon said horn and having an annular end head which extends inward beyond the inner wall thereof, pistons and cylinders arranged within said cylinder barrel, means for rotating said cylinder barrel, means for reciprocating said pistons in response to rotation of said cylinder barrel to thereby cause liquid to flow into and out of said cylinders, a flat valve arranged between said abutment and said end head in contact with the rear face of said end head to control the flow of liquid to and from said cylinders, a hold-up element engaging the front face of said end head, hold-up motors arranged upon opposite sides of the centerline of said cylinder barrel for urging said element against said end head, and means for supplying liquid from the cylinders on either side of said centerline to the hold-up motors on the same side of said centerline to energize the same.

8. In a hydrodynamic machine, the combination of an abutment arranged adjacent the rear end of said machine and restrained from axial movement, a horn rigidly secured in said abutment and extending forward therefrom, an annular cylinder barrel extending around and rotatably supported upon said horn and having an annular end head which extends inward beyond the inner wall thereof, pistons and cylinders arranged within said cylinder barrel, means for rotating said cylinder barrel, means for reciprocating said pistons in response to rotation of said cylinder barrel to thereby cause liquid to flow into and out of said cylinders, a flat valve arranged between said abutment and said end head in contact with the rear face of said end head to control the flow of liquid to and from said cylinders, a hold-up element engaging the front face of said end head, hold-up motors arranged within said element upon opposite sides of the centerline of said cylinder barrel for urging said element against said end head, and means for supplying liquid from the cylinders on either side of said centerline to the hold-up motors on the same side of said centerline to energize the same.

9. In an angle type hydrodynamic machine, the combination of a housing, two trunnions carried by said housing upon opposite sides thereof, a yoke journaled upon said trunnions and having a rear wall, an annular cylinder barrel rotatably supported within said yoke and having an end head which extends radially inward beyond the inner wall thereof, means for rotating said cylinder barrel, pistons and cylinders arranged within said cylinder barrel parallel to its axis of rotation, means for reciprocating said pistons in response to rotation of said cylinder barrel to thereby cause liquid to flow into and out of said cylinders, a flat valve arranged between said rear wall and said end head in contact with the rear face of said end head to control the flow of liquid to and from said cylinders, a hold-up element engaging the front face of said end head, hold-up motors arranged upon opposite sides of the centerline of

said cylinder barrel for urging said element against said end head, and means for supplying liquid from the cylinders on either side of said centerline to the hold-up motors on the same side of said centerline to energize the same.

10. In an angle type hydrodynamic machine, the combination of a housing, two trunnions carried by said housing upon opposite sides thereof, a yoke journaled upon said trunnions and having a rear wall, an annular cylinder barrel rotatably supported within said yoke and having an end head which extends radially inward beyond the inner wall thereof, means for rotating said cylinder barrel, pistons and cylinders arranged within said cylinder barrel parallel to its axis of rotation, means for reciprocating said pistons in response to rotation of said cylinder barrel to thereby cause liquid to flow into and out of said cylinders, a main flat valve engaging the rear face of said end head and having two ports with which each of said cylinders communicates alternately, an auxiliary flat valve engaging the front face of said end head and having two ports with which each of said cylinders communicates alternately, said valves together controlling the flow of liquid to and from said cylinders, hold-up motors arranged upon opposite sides of the centerline of said cylinder barrel for urging said auxiliary valve against said end head, and means for supplying liquid from the cylinders on either side of said centerline to the hold-up motors on the same side of said centerline to energize the same.

11. In an axial type hydrodynamic machine, the combination of a rotatably supported cylinder barrel having an odd number of cylinders arranged therein parallel to its axis of rotation, a valve engaging the rear end of said cylinder barrel to control the flow of liquid to and from said cylinders, a thrust member arranged forward of said cylinder barrel and having formed in its front end as many recesses as there are cylinders with said recesses arranged in a circular row and each recess in the same radial plane as one of said cylinders, the axes of said cylinder barrel and said thrust member being inclined to each other when the machine is performing useful work, means for rotating said cylinder barrel and said thrust member in unison, a piston fitted in each cylinder and connected to said thrust member to be reciprocated thereby during rotation of said cylinder barrel and said thrust member, a non-rotatable hold-up element engaging the front end of said thrust member and having two arcuate grooves formed in its rear face upon opposite sides of the centerline of said machine to supply liquid to said recesses successively during rotation of said thrust member, means for supplying liquid from the cylinders on each side of said machine to the groove in the same side of said hold-up element, and a stationary abutment arranged forward of said hold-up element, said hold-up element having an equal number of hold-up motors arranged therein in communication with each of said grooves and in contact with said abutment, a hole formed in each of the spaces between the adjacent ends of said grooves, and a balancing motor arranged therein in communication with each of said holes and in contact with said abutment.

12. The combination set forth in claim 11 in which liquid is supplied to each of the grooves in said hold-up element through one of the hold-up motors in communication with that groove.

13. In an axial type hydrodynamic machine,

the combination of a rotatable cylinder barrel having a plurality of cylinders arranged therein parallel to its axis of rotation, a valve engaging the rear end of said cylinder barrel to control the flow of liquid to and from said cylinders, a thrust member arranged forward of said cylinder barrel, means for rotatably supporting said thrust member, the axes of said cylinder barrel and said thrust member being inclined to each other when the machine is performing useful work, means for rotating said cylinder barrel and said thrust member, a piston fitted in each cylinder, a ball and socket joint connecting each piston to said thrust member and arranged outside of said cylinder barrel to effect reciprocation of said pistons during rotation of said cylinder barrel and said thrust member, a hydraulic hold-up element engaging the front end of said thrust member to exert thereon forces in opposition to the forces transmitted by said pistons to said thrust member, means for supplying liquid from the cylinders on each side of the centerline of said machine to the same side of said hold-up element, and means for supplying liquid from said hold-up element to said joints to lubricate the same and to hydraulically balance the piston thrust.

14. In an axial type hydrodynamic machine, the combination of a rotatable cylinder barrel having a plurality of cylinders arranged therein parallel to its axis of rotation, a valve engaging the rear end of said cylinder barrel to control the flow of liquid to and from said cylinders, a thrust member arranged forward of said cylinder barrel, means for rotatably supporting said thrust member, the axes of said cylinder barrel and said thrust member being inclined to each other when the machine is performing useful work, means for rotating said cylinder barrel and said thrust member, a piston fitted in each cylinder, a ball socket joint connecting each piston to said thrust member and arranged outside of said cylinder barrel to effect reciprocation of said pistons during rotation of said cylinder barrel and said thrust member, a valve-like hold-up element engaging the front end of said thrust member and having ports in the rear face thereof, means for supplying liquid from the cylinders on each side of the centerline of said machine to the same side of said hold-up element, and channels extending through said thrust member into said joints and having an end of each adapted to register with said ports successively to conduct liquid therefrom to said joints to lubricate the same.

15. In an axial type hydrodynamic machine, the combination of a rotatable cylinder barrel having a plurality of cylinders arranged therein parallel to its axis of rotation, a valve engaging the rear end of said cylinder barrel to control the flow of liquid to and from said cylinders, a thrust member arranged forward of said cylinder barrel, the axes of said cylinder barrel and said thrust member being inclined to each other when the machine is performing useful work, a piston fitted in each cylinder and connected to said thrust member to be reciprocated thereby during rotation of said cylinder barrel and said thrust member, a shaft for rotating said thrust member, means including a universal joint for connecting said cylinder barrel to said shaft to cause it to rotate in unison with said thrust member, said thrust member including a flange portion for reciprocating said pistons and a bearing portion smaller in diameter than said flange portion, a bearing arranged around said bearing portion

to rotatably support said thrust member, a valve-like hold-up element engaging the front end of said bearing portion, means for supplying liquid from the cylinders on each side of the centerline of said machine to the same side of said hold-up element, means including passages formed in said shaft for directing liquid from said hold-up element to said universal joint to lubricate the same.

16. In an angle type hydrodynamic machine, the combination of a housing, two trunnions carried by said housing upon opposite sides thereof, a yoke journaled upon said trunnions, a cylinder barrel rotatably supported by said yoke and having cylinders arranged therein parallel to its axis of rotation, a thrust member rotatably supported by said housing, the axes of said cylinder barrel and said thrust member being inclined to each other when the machine is performing useful work, means for rotating said cylinder barrel and said thrust member, a piston fitted in each cylinder and connected to said thrust member to be reciprocated thereby during rotation of said cylinder barrel and said thrust member, servo-motor means for swinging said yoke upon said trunnions, means for supplying motive liquid to said servo-motor means to energize the same, a pilot valve for controlling the flow of liquid to and from said servo-motor means and movable to and from a neutral position in which it prevents flow of liquid to or from said servo-motor means, a control element for shifting said pilot valve, and a follow-up mechanism connecting said pilot valve to said element and to said yoke for returning said pilot valve to its neutral position in response to said yoke reaching a position corresponding to the position of said control element.

17. In an angle type hydrodynamic machine, the combination of a housing, two trunnions carried by said housing upon opposite sides thereof, a yoke journaled upon said trunnions for rotation thereon in a vertical plane, a cylinder barrel rotatably supported by said yoke and having cylinders arranged therein parallel to its axis of rotation, a thrust member rotatably supported by said housing, means for rotating said cylinder barrel and said thrust member, a piston fitted in each cylinder and connected to said thrust member to be reciprocated thereby during rotation of said cylinder barrel and said thrust member, a large servo-motor adapted when energized to swing said yoke upward, a small servo-motor adapted to swing said yoke downward when it is energized and said large servo-motor is de-energized, spring means adapted when both of said servo-motors are de-energized to hold said yoke in a neutral position, means for supplying motive liquid to said servo-motors to energize the same, and means for controlling the flow of liquid to and from said large servo-motor.

18. In an angle type hydrodynamic machine, the combination of a housing, two trunnions carried by said housing upon opposite sides thereof, a yoke journaled upon said trunnions for rotation thereon in a vertical plane, a cylinder barrel rotatably supported by said yoke and having cylinders arranged therein parallel to its axis of rotation, a thrust member rotatably supported by said housing, means for rotating said cylinder barrel and said thrust member, a piston fitted in each cylinder and connected to said thrust member to be reciprocated thereby during rotation of said cylinder barrel and said thrust member, servo-motor means adapted when active to swing

said yoke upon said trunnions, means for rendering said servo-motor means active or inactive, and spring means adapted when said servo-motor means is inactive to hold said yoke in a neutral position in which the axis of said cylinder barrel is coincident with the axis of said thrust member.

19. In an angle type hydrodynamic machine, the combination of a housing having passages arranged therein for connection to an external circuit, two hollow trunnions arranged in opposite sides of said housing and each provided with a connection to communicate with one of said passages, a cylinder barrel having pistons and cylinders arranged therein, a yoke for carrying said cylinder barrel journaled upon said trunnions and having two passages through which liquid flows to and from said cylinders and which communicate with the interiors of said trunnions respectively through the inner ends thereof, and universally jointed sealing means cooperating with each trunnion and said yoke to provide a substantially fluid tight joint therebetween and to compensate for slight misalignment of parts.

20. In an angle type hydrodynamic machine, the combination of a housing having passages arranged therein for connection to an external circuit, two hollow trunnions arranged in opposite sides of said housing and each provided with a connection to communicate with one of said passages, a cylinder barrel having pistons and cylinders arranged therein, a yoke for carrying said cylinder barrel journaled upon said trunnions and having two passages through which liquid flows to and from said cylinders and which communicate with the interiors of said trunnions respectively through the inner ends thereof, means for providing substantially fluid tight joints between said yoke and each of said trunnions, and two opposed pistons arranged in each trunnion and subjected to the pressure therein, one of said pistons urging said yoke in one direction and the other of said pistons urging said yoke in the opposite direction with a substantially equal force to thereby prevent said pressure from causing deflection of said yoke.

21. In an angle type hydrodynamic machine, the combination of a housing, two trunnions carried by said housing upon opposite sides thereof, a yoke journaled upon said trunnions, a cylinder barrel rotatably supported by said yoke and having cylinders arranged therein parallel to its axis of rotation, a thrust member rotatably supported by said housing, the axes of said cylinder barrel and said thrust member being inclined to each other when the machine is performing useful work, means for rotating said cylinder barrel and said thrust member, a piston fitted in each cylinder and connected to said thrust member to be reciprocated thereby during rotation of said cylinder barrel and said thrust member, any pressure in said cylinders causing said yoke to exert thrusts proportional to said pressure upon said trunnions, and means for swinging said yoke upon said trunnions and hydraulic means for exerting forces proportional to said pressure upon said yoke in opposition to said thrusts.

22. In an angle type hydrodynamic machine, the combination of a housing, two trunnions carried by said housing upon opposite sides thereof, a yoke journaled upon said trunnions, a cylinder barrel rotatably supported by said yoke and having cylinders arranged therein parallel to its axis of rotation, a thrust member rotatably supported by said housing, the axes of said cylinder barrel and said thrust member being inclined to each

other when the machine is performing useful work, means for rotating said cylinder barrel and said thrust member, a piston fitted in each cylinder and connected to said thrust member to be reciprocated thereby during rotation of said cylinder barrel and said thrust member, any pressure in said cylinders causing said yoke to exert thrusts proportional to said pressure upon said trunnions, servo-motor means connected to said yoke, means for supplying to one end of said servo-motor means liquid from the cylinders containing pressure to thereby enable said servo-motor means to exert forces proportional to said pressure upon said yoke in opposition to said thrusts, and means for supplying liquid to said servo-motor means at the other end thereof to enable said servo-motor means to swing said yoke upon said trunnions.

23. In an axial type hydrodynamic machine which includes a rotatably supported cylinder barrel having a plurality of cylinders and pistons arranged therein parallel to its axis of rotation, a valve engaging the rear end of said cylinder barrel to control the flow of liquid to and from said cylinders and provided with a high pressure port and a low pressure port which are arranged upon opposite sides of said machine and with which each cylinder communicates alternately as said cylinder barrel rotates, a rotatably supported thrust member arranged forward of said cylinder barrel, the axes of said cylinder barrel and said thrust member being inclined to each other when the machine is performing useful work, and means for rotating said thrust member and said cylinder barrel in unison, the combination of ball and socket joints arranged outside said cylinders and connecting said pistons to said thrust member to effect reciprocation of said pistons during rotation of said thrust member and said cylinder barrel when the axes thereof are inclined to each other, and means for supplying liquid from said high pressure port to the ball and socket joints on the same side of said machine as said high pressure port to thereby provide a lubricating film of liquid between the mating surfaces of each ball and socket joint, said ball and socket joints being too large to be arranged within said cylinders and being of sufficient size to enable said thrust member to transmit very high pumping forces to said pistons without rupturing the lubricating films between the mating surfaces of said joints, each of said pistons having a recess formed therein and a substantially flat bearing surface at the end of said recess, and each of said joints having a push-rod fixed thereto and engaging the bearing surface on the piston associated with that joint.

24. In an axial type hydrodynamic machine which includes a rotatably supported cylinder barrel having a plurality of cylinders and pistons arranged therein parallel to its axis of rotation, a valve engaging the rear end of said cylinder barrel to control the flow of liquid to and from said cylinders and provided with a high pressure port and a low pressure port which are arranged upon opposite sides of said machine and with which each cylinder communicates alternately as said cylinder barrel rotates, a rotatably supported thrust member arranged forward of said cylinder barrel, the axes of said cylinder barrel and said thrust member being inclined to each other when the machine is performing useful work, and means for rotating said thrust member and said cylinder barrel in unison, the combination of a hydraulic hold-up element engaging

the front end of said thrust member to exert thereon forces in opposition to the forces transmitted by said pistons to said thrust member, means for supplying liquid from the valve port on one side of said machine to the same side of said hold-up element, ball and socket joints arranged outside said cylinders and connecting said pistons to said thrust member to effect reciprocation of said pistons during rotation of said thrust member and said cylinder barrel 10 when the axes thereof are inclined to each other, and means for supplying liquid from said hold-up element to said ball and socket joints to thereby provide a lubricating film of liquid between the mating surfaces of each joint, said ball and socket joints being too large to be arranged within said cylinders and being of sufficient size to enable said thrust member to transmit very high pumping forces to said pistons without rupturing the lubricating films between the mating surfaces of said joints.

25. In a hydrodynamic machine which includes a rotatable cylinder barrel having a plurality of cylinders arranged therein parallel to its axis of rotation, a valve engaging the rear end of said cylinder barrel to control the flow of liquid to and from said cylinders, a thrust member arranged forward of said cylinder barrel, the axes of said cylinder barrel and said thrust member being inclined to each other when the machine is performing useful work, and a piston fitted in each cylinder and connected to said thrust member to be reciprocated thereby during rotation of said cylinder barrel and said thrust member, the combination of a shaft for rotating said thrust member, means including a universal joint for connecting said cylinder barrel to said shaft to cause it to rotate in unison with said thrust member, said thrust member including a flange portion for reciprocating said pistons and a bearing portion smaller in diameter than said flange portion, a bearing arranged around said bearing portion to rotatably support said thrust member, a hold-up element engaging the front end of said bearing portion, means for supplying liquid from the cylinders on each side of the center-line of said machine to the same side of said hold-up element to cause said element to exert upon said thrust member forces in opposition to the forces transmitted by said pistons to said thrust member and to provide between said element and said thrust member a lubricating film of liquid which also functions as a liquid thrust bearing for said thrust member, and means including passages formed in said shaft for directing liquid from 55

said film to said universal joint to lubricate the same.

26. In a hydrodynamic machine which includes a rotatable cylinder barrel having a plurality of cylinders arranged therein parallel to its axis of rotation, a valve engaging the rear end of said cylinder barrel to control the flow of liquid to and from said cylinders, a thrust member arranged forward of said cylinder barrel, means for rotatably supporting said thrust member, the axes of said cylinder barrel and said thrust member being inclined to each other when the machine is performing useful work, and a piston fitted in each cylinder and connected to said thrust member to be reciprocated thereby during rotation of said cylinder barrel and said thrust member, the combination of a shaft for rotating said thrust member, means including a universal joint for connecting said cylinder barrel to said shaft to cause it to rotate in unison with said thrust member, a hold-up element engaging the front end of said thrust member, means for supplying liquid from the cylinders on each side of the center-line of said machine to the same side of said hold-up element to cause said element to exert upon said thrust member forces in opposition to the forces transmitted to said pistons by said thrust member and to provide between said element and said thrust member a lubricating film of liquid, channel means in said shaft for conducting to said universal joint liquid which leaks from said film, and means including a resistance valve for permitting liquid to escape from said channel means.

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References Cited in the file of this patent UNITED STATES PATENTS

Number	Name	Date
Re. 21,758	Foisy	Apr. 1, 1941
2,114,076	Golz	Apr. 12, 1938
2,141,935	Rose	Dec. 27, 1938
2,284,169	Vickers	May 26, 1942
2,313,407	Vickers et al.	Mar. 9, 1943
2,397,314	Grosser	Mar. 26, 1946
2,406,138	Ferris	Aug. 20, 1946
2,421,846	Neuland	June 10, 1947
2,455,330	Denison	Nov. 30, 1948
2,458,935	Ferris et al.	Jan. 11, 1949
2,525,979	Vickers	Oct. 17, 1950

FOREIGN PATENTS

Number	Country	Date
527,486	Great Britain	1940