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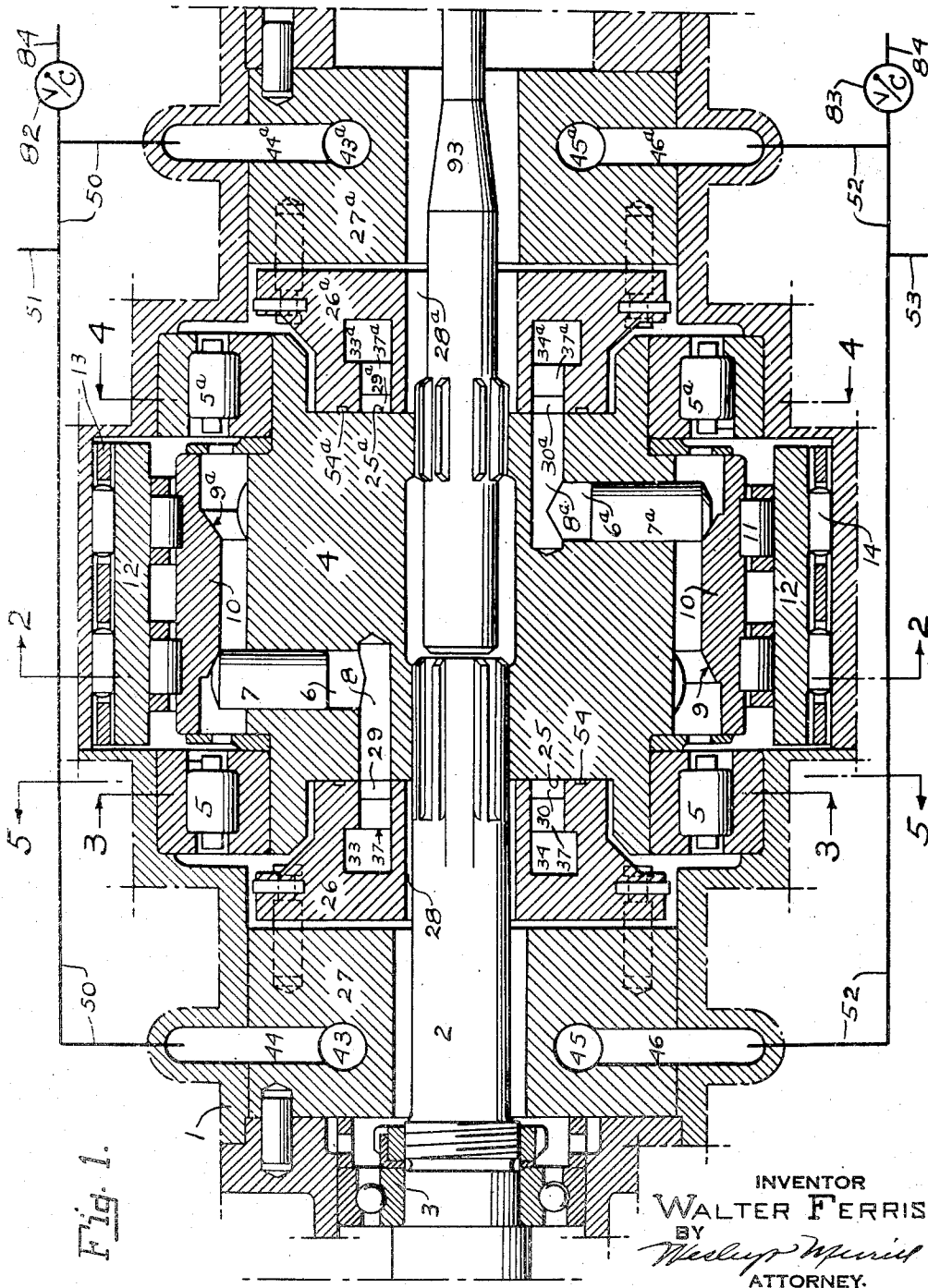
W. FERRIS

2,484,337

HYDRODYNAMIC MACHINE

Filed May 1, 1943

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2,484,337

HYDRODYNAMIC MACHINE

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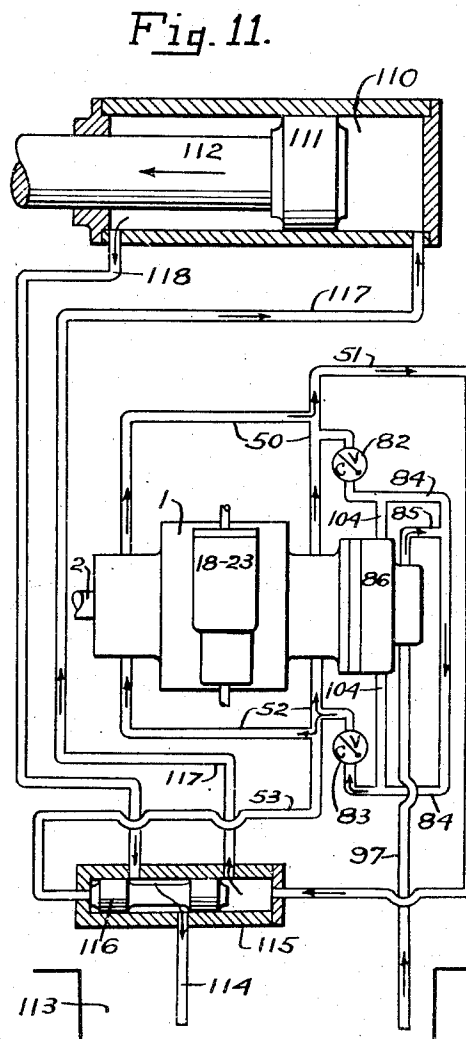
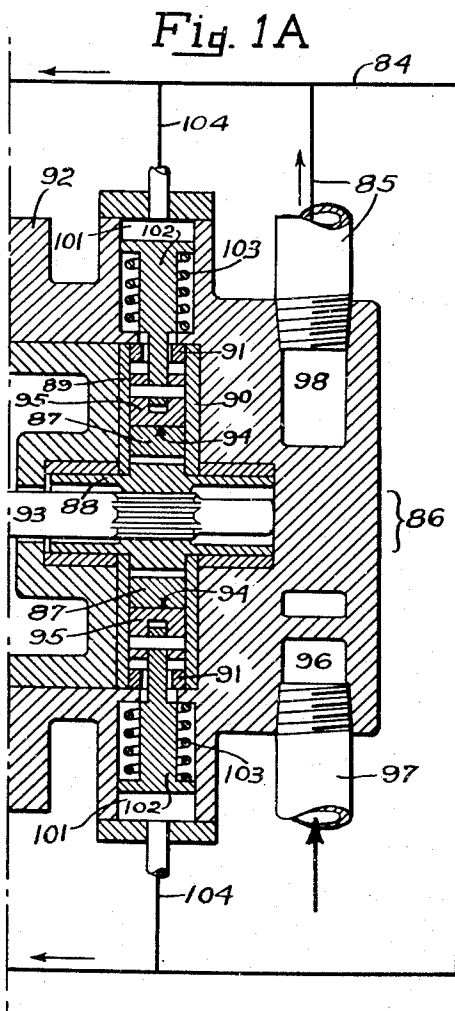
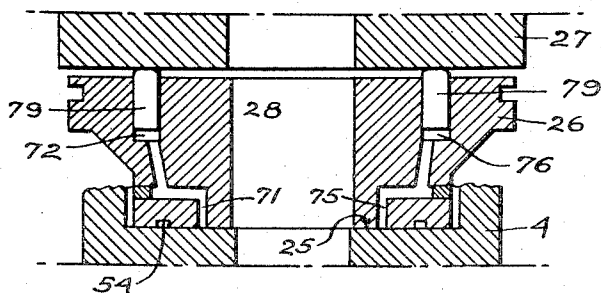


Fig. 6.



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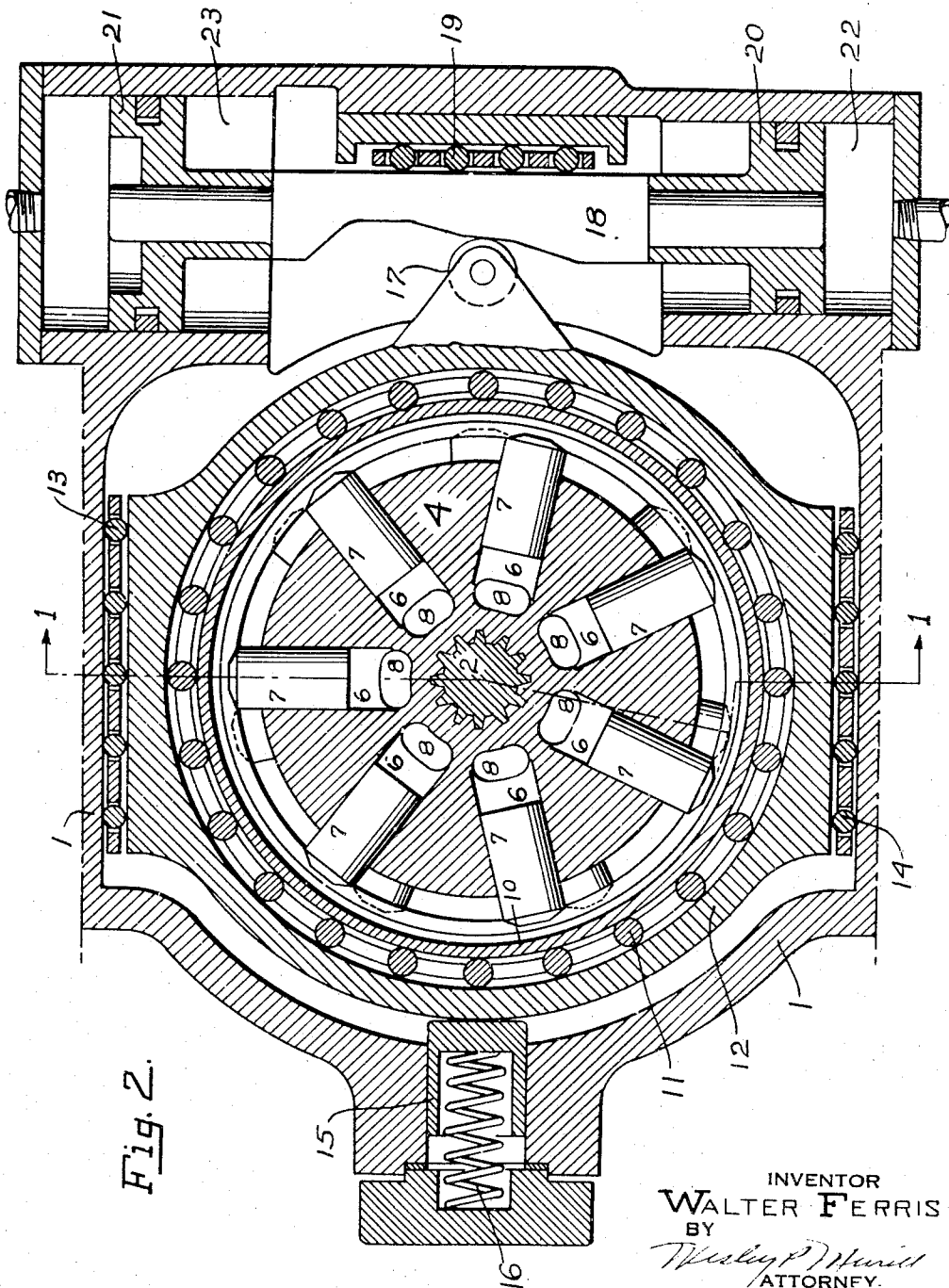
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Fig. 3.

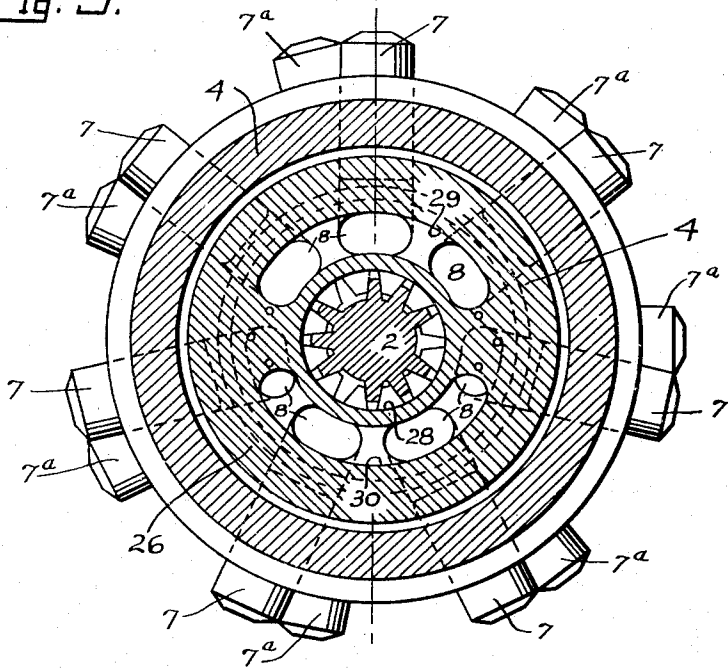
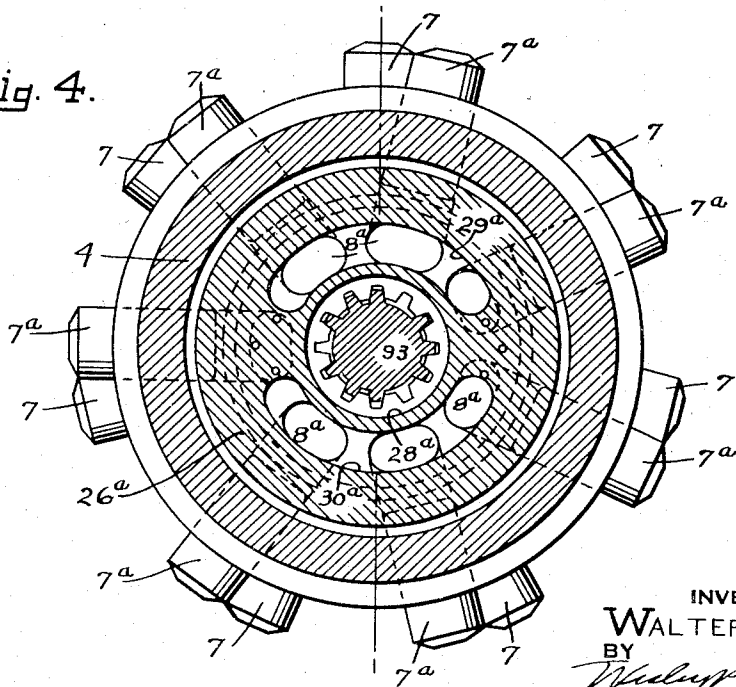


Fig. 4.



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Fig. 7.

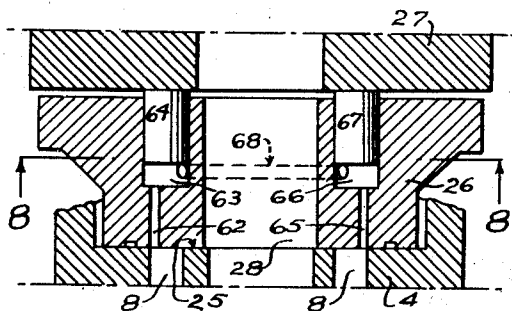


Fig. 5.

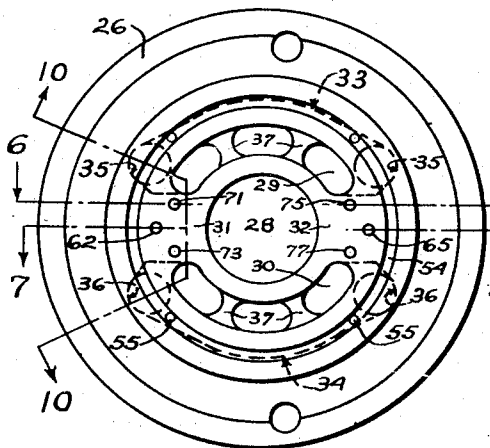


Fig. 9.

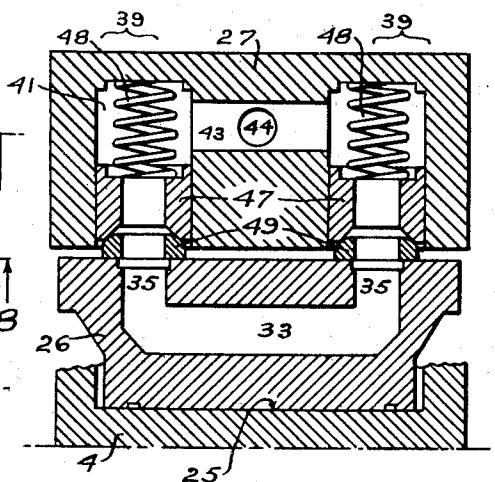


Fig. 8.

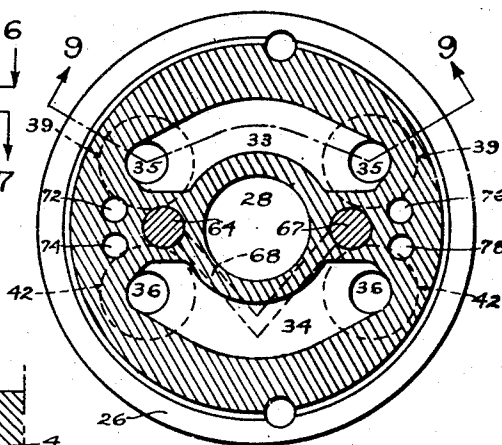
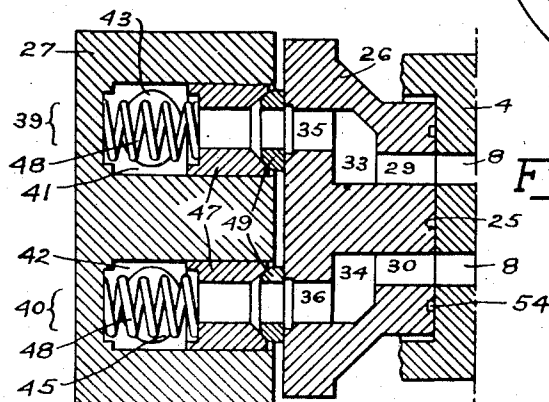


Fig. 10.



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Fig. 12.

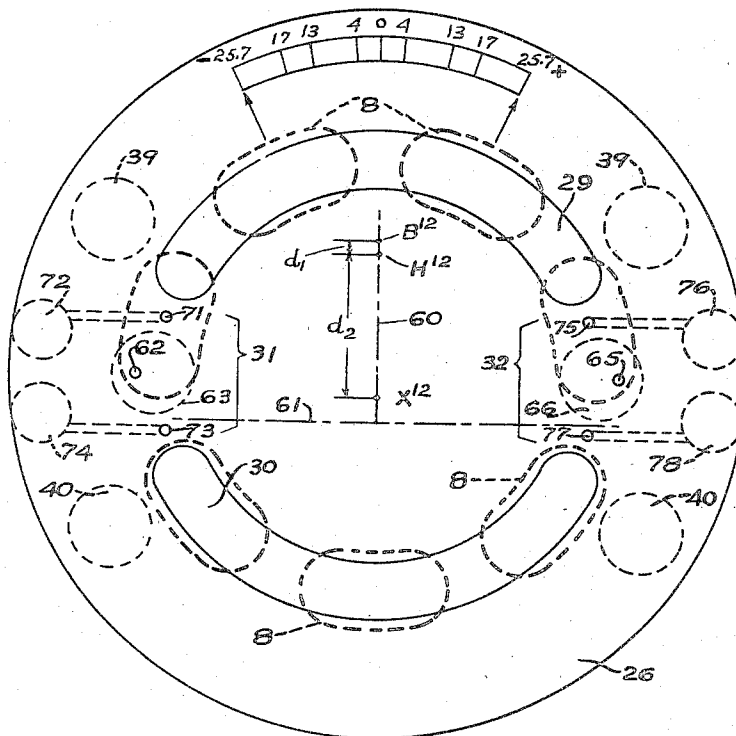
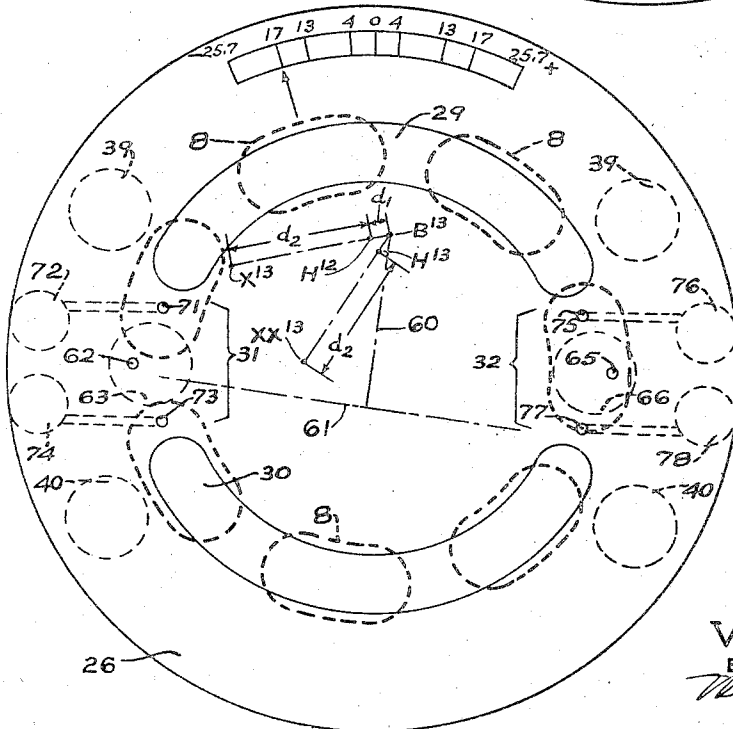


Fig. 13.



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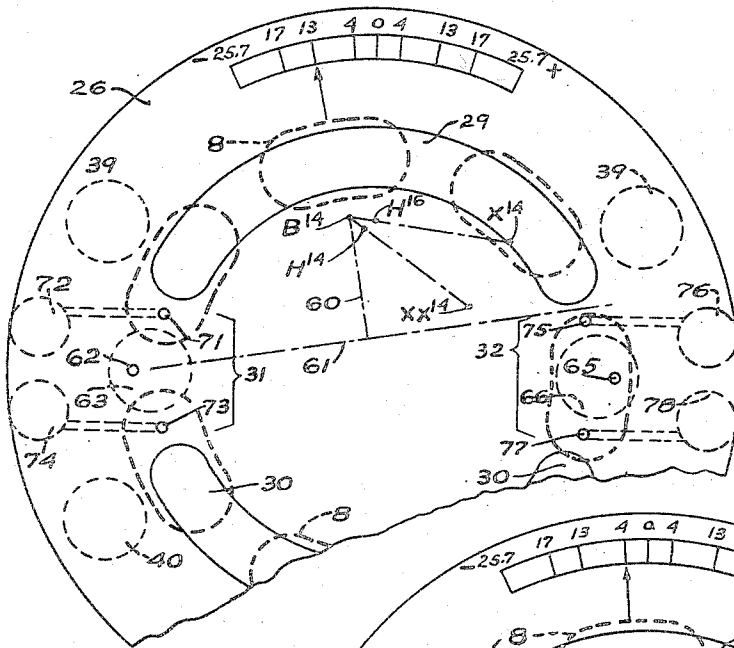


Fig. 14.

Fig. 15.

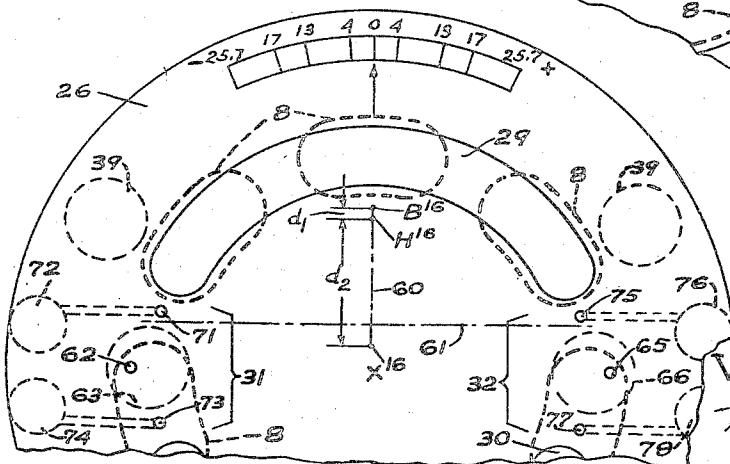
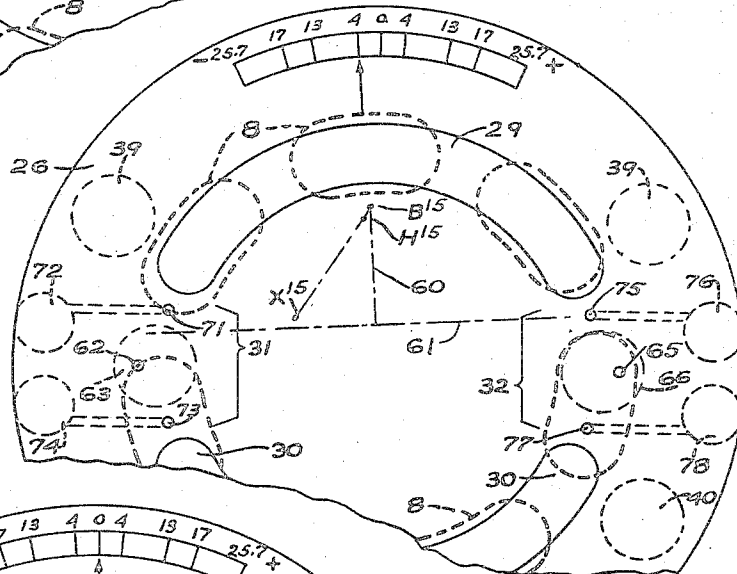


Fig. 16.

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2,484,337

HYDRODYNAMIC MACHINE

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Application May 1, 1943, Serial No. 485,279

29 Claims. (Cl. 103—161)

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This invention relates to rotary hydrodynamic machines of the type in which pistons and cylinders are arranged in a cylinder barrel and the pistons reciprocate in the cylinders when the cylinder barrel rotates. Such a machine will function as a pump when the cylinder barrel is rotated mechanically and it will function as a motor when motive liquid is supplied to the cylinders.

For the purpose of illustration, the invention has been shown embodied in a hydrodynamic machine which is of the radial type and has floating flat valves arranged upon opposite ends of its cylinder barrel to control the flow of liquid to and from the cylinders but the invention is not limited to a machine of the radial type nor to a machine of the flat valve type.

In a machine of the flat valve type, the cylinders communicate with passages which extend through the end of the cylinder barrel with which the flat valve is in contact. The outer ends of the passages constitute cylinder ports each of which, during rotation of the cylinder barrel, registers alternately with a high pressure port and a low pressure port which are formed in the valve diametrically opposite each other and are adapted to be connected to opposite sides of an external circuit.

The pressure in the ports and in the lubricating film between the face of the valve and the end of the cylinder barrel tends to separate the valve from the cylinder barrel so that, in order to maintain a liquid seal therebetween, it is necessary to urge the valve toward the cylinder barrel with a force which varies in accordance with variations in the pressure of the motive liquid and to provide means which prevent the cylinder barrel from being moved axially by the force exerted by the liquid on the end thereof but which increases the cylinder barrel bearing load with a resultant decrease in the efficiency of the machine.

The liquid in the cylinder barrel passages is compressed when the passages communicate with the high pressure valve port and it expands when the passages communicate with the low pressure valve port. This alternate compression and expansion of the liquid in the passages also reduces the efficiency of the machine.

The present invention has as an object to provide a radial hydrodynamic machine having flat valves upon both ends of the cylinder barrel so that the axial force exerted by the liquid upon one end of the cylinder barrel is counterbalanced by the force exerted by the liquid upon the opposite end of the cylinder barrel.

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An advantageous result of arranging valves upon both ends of the cylinder barrel is that the total volume of the cylinder barrel passages is greatly reduced with a resultant reduction in the quantity of liquid that is alternately compressed and expanded.

Another object is to provide a hydrodynamic machine with a flat valve having struts extending across its ports to stiffen the valve and thereby prevent any material distortion of the valve due to variations in pressure and temperature.

Another object is to provide means for supercharging the pump at a pressure high enough to permit its intake passages to be reduced in diameter to a size no larger than the necessary size of the discharge passages to thereby materially reduce the area of the flat valve and the overall size of the pump.

Another object is to provide a large capacity pump with an auxiliary pump for supplying liquid thereto in sufficient volume and at a high enough pressure to keep the large capacity pump fully supercharged.

Another object is to provide a variable delivery pump with a supercharging pump which delivers liquid to the variable delivery pump at a predetermined pressure and varies its rate of delivery in accordance with variations in the rate of delivery of the variable delivery pump.

Another object is to provide a hydrodynamic machine with flat valves on both ends of the cylinder barrel and to so arrange the cylinder barrel ports and the valve ports at one end of the cylinder barrel relative to the corresponding cylinder ports and valve ports at the other end of the cylinder barrel as to split the phase of the machine.

Another object is to provide a hydrodynamic machine having a valve arranged upon the end of its cylinder barrel and a drive shaft extending through the valve and connected to the cylinder barrel to drive the same.

Another object is to provide a hydrodynamic machine having an annular valve arranged upon the end of its cylinder barrel and provided with opposed high and low pressure ports to register with cylinder ports formed in the end of the cylinder barrel, a drive shaft extending through the valve and connected to the cylinder barrel to rotate it, hold-up motors arranged at opposite sides of the shaft and between the valve ports for urging the valve toward the cylinder barrel, and means responsive to a cylinder port passing to or from the high pressure valve port for ener-

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gizing hold-up motors upon opposite sides of the drive shaft.

Other objects and advantages will appear from the description hereinafter given of a hydrodynamic machine in which the invention is embodied.

The invention is exemplified by the hydrodynamic machine shown in part in detail and in part schematically in the accompanying drawings in which the views are as follows:

Fig. 1 is a longitudinal sectional view showing a hydrodynamic machine having flat valves arranged upon both ends of its cylinder barrel, the view through cylinder barrel only being taken on the irregular line 1—1 of Fig. 2.

Fig. 1A is a longitudinal sectional view of an auxiliary pump which supercharges the machine shown in Fig. 1 when that machine is employed as a pump, the view being a continuation of Fig. 1.

Fig. 2 is a transverse section taken on the line 2—2 of Fig. 1.

Figs. 3 and 4 are transverse sectional views illustrating the method of splitting the phase of the pump, the views being taken in the planes indicated, respectively, by the lines 3—3 and 4—4 of Fig. 1.

Fig. 5 is a face view of one of the flat valves, the view being taken in the plane indicated by the line 5—5 of Fig. 1.

Fig. 6 is a longitudinal section through one of the flat valves and the adjacent portions of the distributing block and cylinder barrel, the view being taken on the line 6—6 of Fig. 5.

Fig. 7 is a view similar to Fig. 6 but taken on the line 7—7 of Fig. 5.

Fig. 8 is a transverse section through one of the flat valves as indicated by the line 8—8 of Fig. 7.

Fig. 9 is a longitudinal section through one of the flat valves and the adjacent portions of the cylinder barrel and the distributing block, the view being taken on the irregular line 9—9 of Fig. 8.

Fig. 10 is a view similar to Fig. 9 but taken on the irregular line 10—10 of Fig. 5.

Fig. 11 is a diagram of a hydraulic circuit in which the machine shown in Fig. 1 is employed as a pump and is supplied with liquid solely by the auxiliary pump shown in Fig. 1A.

Figs. 12 to 16 are diagrams illustrating the relation between the cylinder ports and the valve ports during certain parts of a revolution of the cylinder barrel.

The machine chosen for illustration has its mechanism arranged within and carried by a casing 1 which consists of a plurality of parts and in practice has a plurality of passages formed therein but, in order to simplify the drawings, only fragments of the casing have been shown and certain of the passages have been represented by external channels.

Power is transmitted to or from the machine through a shaft 2 which is journaled in a bearing 3 carried by casing 1. Shaft 2 has its inner end splined or otherwise suitably connected to a cylinder barrel 4 which is rotatably supported within casing 1 by bearings 5 and 5^a carried thereby. Cylinder barrel 4 has a plurality of cylinders formed therein and shown arranged in two circular rows, each cylinder in one row being indicated by the reference numeral 6 and each cylinder in the other row being indicated by the reference numeral 6^a. Each cylinder 6 has a piston 7 fitted therein and it has its inner end con-

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nected to a passage 8 which is formed in cylinder barrel 4 and extends through the left end thereof. Each cylinder 6^a has a piston 7^a fitted therein and it has its inner end connected to a passage 8^a which is formed in cylinder barrel 4 and extends through the right end thereof.

The machine shown will function as a motor when motive liquid is supplied to cylinders 6 and 6^a and it will function as a pump when shaft 2 is driven from an external source of power. In order to simplify the description, the machine will be referred to as a pump but it is to be understood that the term "pump" as used herein is descriptive only and in no way limits the invention to a pump.

The cylinders have been shown as being radial and the pistons have been shown as having the outer ends or heads thereof in contact with conical reaction surfaces 9 and 9^a arranged upon the inside of an annular thrust member 10 which extends around cylinder barrel 4, the outer end of each piston being so shaped that the reaction surface makes contact therewith upon a single spot which is offset far enough from the piston axis to cause the piston to rotate in its cylinder as it is reciprocated therein during rotation of cylinder barrel 4 and thrust member 10 as fully explained in Patent No. 2,074,068.

The volumetric capacity of the pump may be increased by increasing the number of rows of cylinders, or it may be increased by increasing the number of cylinders in each row and inclining alternate cylinders in each row in opposite directions and having the outer ends of the pistons in contact with a circular reaction surface as shown in Patent No. 2,406,138. If the pump is to operate at a high pressure, such as 2500# per sq. in., the pistons are reduced in diameter and provided on their outer ends with enlarged heads as shown in Patent No. 2,074,068.

Thrust member 10 is rotatably supported as by means of a roller bearing 11. In a constant displacement pump or motor, thrust member 10 is adapted to rotate upon a stationary axis but it has been shown supported by bearing 11 within a displacement varying member or slide block 12 which is slidably supported within casing 1 as by means of two roller bearings 13 and 14 so that when functioning as a pump it will deliver liquid in a direction and at a rate dependent upon the direction and distance the axis of slide block 12 is offset from the axis of cylinder barrel 4.

Slide block 12 may be shifted in response to operation of a suitable control such as the control illustrated and described in Patent No. 2,406,138. As shown in Fig. 2, slide block 12 is constantly urged toward the right by a servo-motor 15 which engages the left side thereof and includes a spring 16. A roller 17 carried by slide block 12 at the right side thereof is urged by servo-motor 15 against the face of a cam 18 which is guided by a roller bearing 19 and has its ends connected to pistons 20 and 21 fitted, respectively, in cylinders 22 and 23 carried by casing 1, roller bearing 19 preventing cam 18 from being deflected by the thrust of servo-motor 15.

Cylinder 22 is continuously supplied with liquid at a substantially constant pressure and cylinder 23 is larger than cylinder 22 so that, when it is supplied with liquid at the same pressure, piston 21 will move cam 18 downward and permit servo-motor 15 to move slide block 12 toward the right and, when cylinder 23 is opened to drain, piston 20 will move cam 18 upward and cause it to

move slide block 12 toward the left as fully explained in Patent No. 2,406,138.

In order to maintain the overall length of the pump at a minimum, cylinder barrel 4 has been shown as having its ends recessed and passages 8 and 8^a terminating within the recesses. The recessed end portions are made flat and smooth to form valve seats 25 and 25^a for two identical valves 26 and 26^a which cooperate with passages 8 and 8^a to control the flow of liquid to and from cylinders 6 and 6^a. Valves 26 and 26^a are arranged, respectively, between cylinder barrel 4 and two identical distributing blocks 27 and 27^a which are fixed in stationary positions within casing 1. Each valve has been shown as being formed from a single piece but in practice it is formed by separate pieces which are welded together.

Valve 26 (Figs. 1 and 5) is provided at its center with an opening 28 which is slightly larger than drive shaft 2 so that shaft 2 may pass freely therethrough and valve 26 may be properly adjusted to its seat 25. The inner end or face of valve 26 has formed therein two diametrically opposed substantially crescent shaped ports 29 and 30 with which each passage 8 communicates alternately as cylinder barrel 4 rotates, the outer ends of passages 8 constituting cylinder ports which cooperate with valve ports 29 and 30 to control the flow of liquid to and from cylinders 6.

The adjacent ends of ports 29 and 30 are separated from each other by portions of the valve face which constitute liquid seals or bridges 31 and 32, the angular length of each bridge being equal to or slightly greater than the angular length of a cylinder port 8 and all of the cylinder ports being of the same length.

Ports 29 and 30 communicate, respectively, with two chambers 33 and 34 (Figs. 8-10) which are formed within valve 26 and have passages 35 and 36 leading therefrom respectively through the outer end of the valve. In order to prevent the face of valve 26 from being distorted due to changes in pressure and temperature, a plurality of integral ties or struts 37 (Figs. 1 and 5) extend across each of ports 29 and 30 and are spaced from the face of the valve.

Distributing block 27 has been shown as having two hold-up motors 39 arranged therein above the horizontal centerline of valve 26 and two hold-up motors 40 arranged therein below the horizontal centerline of valve 26 but the number of hold-up motors may be varied.

Each passage 35 communicates with the cylinder 41 of a hold-up motor 39 and each passage 36 communicates with the cylinder 42 of a hold-up motor 40. The two cylinders 41 are connected by a passage 43 to each other and to a passage 44 which extends outward through the side of block 27 for connection to one side of an external circuit, and the two cylinders 42 are connected by a passage 45 to each other and to a passage 46 which extends outward through the side of block 27 for connection to the other side of an external circuit.

The four hold-up motors 39 and 40 are identical and each has a hollow piston 47 fitted in its cylinder and urged by a spring 48 against an annular sealing member 49 which is urged by spring 48 against the outer end of valve 26, the contacting surfaces of member 49 and valve 26 being ground flat and the contacting surfaces of piston 47 and member 49 being ground spherical to provide self-aligning liquidtight seals therebetween. The openings through pistons 47 and

member 49 provide communication between the passages in the valve and the inner ends of the hold-up cylinders so that liquid may flow freely through the valve and the distributing block to and from the external circuit.

The force exerted upon pistons 47 by springs 48 is transmitted to valve 26 and any pressure prevailing in the hold-up motor cylinders acts upon the ends of pistons 47 and is transmitted therethrough to valve 26. Consequently, valve 26 is urged against its seat 25 by the constant force supplied by the four springs 48, by a force which varies in accordance with variations in the pressure of the motive liquid and is supplied by the pressure in hold-up motors on the pressure side of the machine, and by any pressure that may prevail in the hold-up motors on the low pressure side of the machine.

Valve 26^a is identical to valve 26 and distributing block 27^a is identical to distributing block 27. Consequently, they have not been fully illustrated and a description thereof is deemed unnecessary, like parts and passages as far as shown being indicated by like reference numerals with the exponent "a" added to the reference numerals applied to valve 26^a and distributing block 27^a. Passage 44 in block 27 and passage 44^a in block 27^a are connected to each other by a channel 50 which is also connected to a channel 51 by means of which the machine may be connected to one side of an external circuit. Passage 46 in block 27 and passage 46^a in block 27^a are connected to each other by a channel 52 which is also connected to a channel 53 by means of which the machine may be connected to the other side of an external circuit. Channels 50 and 52 have been shown as external channels but in practice they are formed in the casing of the machine.

Passages 8 and 8^a are equally spaced in cylinder barrel 4 and valves 26 and 26^a are symmetrical about their horizontal centerlines so that, when cylinder barrel 4 is rotated, the machine will function as a pump and will deliver liquid in a direction and at a rate dependent upon the direction and rate of rotation of cylinder barrel 4 and upon the direction and distance slide block 12 is offset from its neutral position and, when motive liquid is supplied to the machine, it will function as a motor and rotate shaft 2 in a direction and at a rate dependent upon the direction and rate of flow of the motive liquid and upon the direction and distance slide block 12 is offset from its neutral position, it being understood that, when slide block 12 is in its neutral position, thrust member 10 is concentric with cylinder barrel 4 so that pistons 7 and 7^a will not be reciprocated either by rotation of cylinder barrel 4 or by motive liquid supplied to the machine.

Assuming that cylinder barrel 4 is being rotated in a clockwise direction in respect to Fig. 2 and that slide block 12 is offset to the left of its neutral position in respect to that figure, the pistons in the lower half of cylinder barrel 4 will move progressively outward and the pistons in the upper half of cylinder barrel 4 will be forced progressively inward by thrust member 10. The cylinders of the outward moving pistons will be filled with liquid which flows thereto through the passages 8 and 8^a connected thereto, the lower parts of valves 26 and 26^a and the lower parts of distributing blocks 27 and 27^a from channel 52 the liquid being supplied to channel 52 from the external circuit and/or from an auxiliary

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pump as will presently be explained. The inward moving pistons will eject liquid from their cylinders through the passages 8 and 8^a connected thereto, the upper parts of valves 26 and 26^a, the upper parts of distributing blocks 27 and 27^a, channel 50 and channel 51 to the external circuit and the resistance encountered by the liquid in the external circuit will cause pressure to rise.

If cylinder barrel 4 is rotated in the opposite direction or if slide block 12 is shifted to the other side of its neutral position, the machine will deliver liquid into channel 52 and ports 30 and 30^a will then be the high pressure ports and ports 29 and 29^a will then be the low pressure ports. If the motive liquid is supplied to cylinders 6 and 6^a through one port of each of valves 26 and 26^a, the machine will function as a motor and discharge liquid at a low or zero pressure through the other ports of each of valves 26 and 26^a.

It is well known that, in order to obtain quiet and smooth operation of a hydrodynamic machine, only one cylinder port should open to a valve port at any given instant, and, if practicable, the phase of the machine should be split as explained in Patent No. 1,998,984. This is accomplished by providing the cylinder barrel at each of its ends with an odd number of ports according to the usual practice and offsetting either the valve ports or the cylinder ports at one end of the cylinder barrel from the corresponding ports at the other end of the cylinder barrel.

For example, valves 26 and 26^a may be so arranged that the corresponding ports thereof are in axial alinement with each other, as shown in Fig. 1, and passages 8 and 8^a may be offset from each other as shown in Figs. 3 and 4, it being noted that a passage 8 is centered upon the vertical centerline of valve 26 in Fig. 3 and that the passage 8^a nearest thereto is offset approximately one-quarter of a cylinder phase from the vertical centerline of valve 26^a. The arrangement is such that only one cylinder port (passage 8 or 8^a) will open to a valve port at any given instant.

It has previously been stated that valves 26 and 26^a are identical and it will be apparent from the foregoing that they function in exactly the same manner. Also, both valves are subjected in exactly the same manner to forces which tend to move them away from their seats and to identical greater forces which hold them to their seats. It is therefore deemed sufficient to explain only the forces acting upon valve 26 as such an explanation is likewise applicable to valve 26^a.

When the machine is performing work either as a pump or as a motor, the liquid in one port of valve 26 is under a high pressure, the liquid in the other port of valve 26 is under a low, zero or negative pressure, and minute quantities of liquid will escape from the high pressure port and spread over valve seat 25 and the face of valve 26 to form a lubricating film therebetween. The pressure prevailing in the high pressure port will extend into the lubricating film, and the pressure prevailing in one or both ports and in the film will tend to move the valve away from its seat 25.

The area of film into which pressure may extend is limited by an annular groove 54 which is formed in the face of the valve and has a plurality of drain ducts 55 leading therefrom so that liquid entering groove 54 may drain into casing 1. The area of valve face outside of groove 54 simply acts as a bearing area as there is no pressure in the film between it and valve seat 25.

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Due to the pressure being very high in one valve port and very low or absent in the other valve port, the film pressure varies from a high pressure at the edge of the high pressure port to a low or zero pressure at the edge of the other port and to zero at the inner edge of the valve seat and at the edge of groove 54. Consequently, the center of pressure is not on the center of the valve but, if the pressure in the low pressure port is zero or negative, the center of pressure is near the center of a high pressure area which extends around the high pressure port or, if there is pressure in the low pressure port, there will be one center of pressure near the center of the high pressure area which extends around the high pressure port and another center of pressure near the center of a low pressure area which extends around the low pressure port.

However, the force exerted by the liquid in the low pressure area is ordinarily so small that it may be disregarded in the present explanation and only the high pressure area considered, it being understood that the center of high pressure is on one side of the cylinder barrel axis when one port is the high pressure port and on the other side of the cylinder barrel axis when the other port is the high pressure port.

Since an area of the valve equal to the area of the high pressure port or ports is under full working pressure and since a film of liquid leaks from the high pressure port to the nearest drained areas so that the film is under full working pressure at the edges of the high pressure port and has zero pressure at the edges of the nearest drained area, the pressure area may be considered as being equal to the area or areas of the high pressure port or ports plus one half the valve face area across which the liquid passes to drain. This pressure area will be referred to herein as the "blow-off area." The force which tends to move the valve away from its seat, hereinafter referred to as the "blow-off force," is determined by multiplying the blow-off area by the unit working pressure. To simplify the computation, all leakage across the valve face is assumed to be either radial or circumferential.

The blow-off area will vary greatly as cylinder barrel 4 rotates for the reason that the cylinder barrel is always provided with an odd number of cylinder ports so that an odd number of cylinder ports and an even number of cylinder ports communicate alternately with the high pressure valve port during rotation of the cylinder barrel. For example, if each end of cylinder barrel 4 is provided with seven cylinder ports 8 as shown, three cylinder ports 8 will communicate with the high pressure valve port during a part of a revolution of cylinder barrel 4 and four cylinder ports 8 will communicate with the same valve port during the preceding and succeeding parts of a revolution. Consequently, the blow-off area will vary as the cylinder ports move across bridges 31 and 32.

Various critical positions of the cylinder ports relative to the valve ports are shown in Figs. 12 to 16 which are purely schematic and are provided solely for the purpose of illustrating various positions to which the centers of the blow-off and hold-up forces move during rotation of the cylinder barrel. These figures are in effect views of the face only of valve 26 with the cylinder ports superimposed thereon but struts 34, drain grooves 54 and opening 28 have been omitted to simplify the views, the other openings in the valve face being shown in full lines, the cylinder ports being

represented by heavy dotted lines and the cylinders behind the valve being shown in light dotted lines. In order that the cylinder ports may be readily recognized in their various positions, the valve ports have been shown in these figures as being narrower than the cylinder ports but it is to be understood that the valve ports and the cylinder ports are of equal width as shown in Figs. 1 and 10.

The blow-off area is determined by its radial width, which is constant, and its angular length which varies as the cylinder barrel rotates. Since the valve ports and the cylinder ports are of the same width and arranged in radial alinement, the width of the blow-off area is computed as being equal to the port width plus one half the distance from one edge of the valve port to the edge of opening 28 (Fig. 5) plus one half the distance from the other edge of the valve port to the edge of drainage groove 54.

The angular length of the blow-off area is computed as being equal to the angular distance between the farthest apart ends of the high pressure valve or cylinder ports plus one half of the distances from such port ends to the low pressure valve or cylinder ports nearest those ends.

The blow-off force, which is determined by multiplying the blow-off area by the unit working pressure, has its center located a short distance inward from high pressure port 29 upon a line 60 passing through the valve center and normal to a line 61 which extends across the ends of the blow-off area. In each of Figs. 12 to 16, it is assumed that valve port 29 is the high pressure port and the center of the blow-off force has been indicated by the reference letter B having added thereto an exponent corresponding to the number of the figure.

When three cylinder ports 8 are in communication with port 29 and are symmetrical about the vertical centerline of the valve as shown in Fig. 16, the blow-off area will extend through an angular distance of about 155° from a point half way between the end of the left hand pressure port 8 and the adjacent end of the port 8 on bridge 31 to a point half way between the end of the right hand pressure port 8 and the adjacent end of the port 8 on bridge 32, and the line 61 extending across the ends of the blow-off area will be parallel to the horizontal centerline of the valve so that the line 60 upon which the center B¹⁶ of the blow-off force is located will coincide with the vertical centerline of the valve.

Hold-up motors 39 are of such size that the sum of the forces exerted thereby exceeds by a small amount the total blow-off force when three cylinder ports are under pressure and are symmetrically located as shown in Fig. 16. If the center of the blow-off force remained stationary, hold-up motors 39 could be so located that the center of the hold-up force exerted thereby would coincide with the center of the blow-off force and thus hold the valve uniformly in engagement with its seat.

Since the center of the blow-off force does not remain stationary as will presently be explained, hold-up motors 39 are equally spaced from the vertical centerline of the valve and located at such a distance from the horizontal centerline of the valve that the hold-up forces exerted thereby are centered upon the vertical centerline of the valve at a point spaced a very short distance d_1 inward from the center B¹⁶ of the blow-off force

when three cylinders are symmetrical as shown in Fig. 16.

Due to the centers of the hold-up and blow-off forces being offset from each other, the hold-up force exerted by motors 39 may be considered as being divided into a main hold-up force, which is slightly greater in magnitude than blow-off force B¹⁶ and has its center H¹⁶ located on the vertical centerline of the valve at a distance d_1 from the center of force B¹⁶ as indicated in Fig. 16, and a small excess force which is equal in magnitude to the difference between force H¹⁶ and the total force exerted by motors 39 and which has its center X¹⁶ located on the vertical centerline of the valve at such a distance d_2 from the center H¹⁶ of the hold-up force that $B^{16}d_1 = X^{16}d_2$ so that the reaction to force X¹⁶ establishes equilibrium between the blow-off and the hold-up forces.

In each of Figs. 12 to 16, the centers of the blow-off and main hold-up forces have been shown in only approximately their correct locations and the center of the excess hold-up force has simply been shown in a convenient location with no attempt to even approximate its correct location.

When four cylinder ports 8 are in communication with high pressure valve port 29 and are symmetrical about the vertical centerline of the valve as shown in Fig. 12, each end of the blow-off area will be at a point half way between the cylinder port 8 on the valve bridge and the adjacent end of low pressure valve port 30 so that the blow-off area extends through an angular distance of about 205°. Consequently, the blow-off area and the blow-off force are each ²⁰⁵/₁₅₅ or 1.32 times greater than when three cylinder ports are in communication with valve port 29 and are symmetrical about the vertical centerline of the valve. The blow-off force not only increases in magnitude but its center shifts downward from the position shown in Fig. 16 for the reason that the increase in blow-off area is downward. Since the line 61 extending across the ends of the blow-off area is parallel to the horizontal centerline of the valve, the center B¹² of the blow-off force will be located on the vertical centerline of the valve as shown in Fig. 12.

If the machine is to function solely as a pump, phase changes in the magnitudes and locations of blow-off forces are not important so long as the constant hold-up force (from motors 39) always exceeds the blow-off force and is located at a smaller radius. As the cylinder barrel of a pump is always rotating at high speed, there is not time for the supporting film to be squeezed out during phase changes of blow-off force, and hence considerable excess hold-up forces are easily carried by the film. Under these conditions some or all of the auxiliary hold-up motors to be later described may be omitted, at the same time correspondingly enlarging constant hold-up motors 39 and bringing them closer to the center of the valve. But in order that the machine may function as a motor, especially at very slow speeds, it is necessary to divide the total hold-up force into a smaller constant force from motors 39, plus intermittently acting supplementary forces from auxiliary hold-up and balancing motors which are energized and deenergized as the blow-off forces increase or decrease, and as the center of resultant blow-off force moves in or out, or right or left. The effect is to maintain a step-by-step increase and decrease of the resultant hold-up force with corresponding changes in its location, thereby maintaining the mag-

nitude of the hold-up force much less in excess of the blow-off force and keeping the center of hold-up forces closer to that of the blow-off force than can be done if only constantly acting motors 39 are used.

Additional hold-up force could be applied by an auxiliary motor arranged at the center of the valve if the valve had a solid center but, since valve 26 has opening 28 extending there-through, an additional hold-up force may be provided by two auxiliary hold-up motors arranged at opposite sides of opening 28.

As shown, a duct 62 (Fig. 5) leads from the face of bridge 31 to a cylinder 63 (Fig. 7) which extends inward from the rear end of valve 26 and has a piston 64 fitted therein to engage the face of distributing block 27, and a duct 65 leads inward from the face of bridge 32 to a cylinder 66 which extends inward from the rear end of valve 26 and has a piston 67 fitted therein to engage the face of distributing block 27. Cylinders 63 and 66 are arranged upon the horizontal centerline of the valve and are connected to each other by a duct 68 (Figs. 7 and 8) so that, when a cylinder port having pressure therein communicates with the outer end of either duct 62 or 65, both cylinders 63 and 66 will be energized and urge valve 26 toward valve seat 25 with a force which is proportional to the pressure and to the combined areas of the two cylinders.

The outer ends of ducts 62 and 65 are so located in the bridges and cylinder ports 8 are of such length that a cylinder port establishes communication between high pressure valve port 29 and one of ducts 62 and 65 the instant that a fourth cylinder port opens to valve port 29, and one or both of ducts 62 and 65 are open to pressure during the entire time that four cylinder ports are under pressure so that auxiliary motors 63-64 and 66-67 are energized and increase the total hold-up force to compensate for the increase in the blow-off force.

With certain numbers of cylinders and proper proportioning of cylinder ports and with properly selected diameters of ducts 62 and 65, it is possible to produce the result of energizing both of cylinders 63 and 66 at all times when the greatest number of cylinder ports are under pressure without providing duct 68 between these cylinders. In this case both cylinders are simultaneously energized and deenergized by the action of the cut-off edges of oblong ports 8 on the ends of ducts 62 and 65. As this method of cross-connecting is more difficult to explain and produces no different result than duct 68, it has not been shown.

The main and auxiliary hold-up motors are so proportioned that the total hold-up force exerted thereby is divided into a main hold-up force, which has its center H^{12} located on the vertical centerline of the valve at a distance d_1 toward the valve center from the center B^{12} of the blow-off force, and an excess force which has its center X^{12} located upon the vertical centerline of the valve at such a distance d_2 from the center of the hold-up force H^{12} that $B^{12}d_1 = X^{12}d_2$ so that the reaction to force X^{12} establishes equilibrium between the hold-up and blow-off forces.

If the cylinder barrel has seven equally spaced ports as shown, the cylinder ports are spaced 51.4° apart and, when occupying the position shown in Fig. 12, the cylinder ports will be offset one half of a cylinder phase or 25.7° from the position shown in Fig. 16. Assuming that the

cylinder ports are in zero position when located as shown in Fig. 16 and starting with the cylinder ports located as shown in Fig. 12, the number of cylinder ports in communication with high pressure valve port 29 will change from four to three during rotation of the cylinder barrel from -25.7° to 0° and will change from three to four during rotation of the cylinder barrel from 0° to $+25.7^\circ$. The number of cylinder ports in communication with the high pressure valve port will thus change from three to four and from four to three during each cylinder phase.

As the cylinder barrel rotates, the blow-off area will decrease on one bridge and increase on the other bridge and thus cause the center of the blow-off force to shift away from the centerline of the valve. For example, when the cylinder ports have moved to the position shown in Fig. 13, the blow-off area will extend over almost all of bridge 32 and over only about one half of bridge 31 so that a line 61 passing across the ends of the blow-off area will be inclined to the horizontal centerline of the valve. Since the center of the blow-off force is always located upon a line 60 which passes through the valve center and is normal to line 61 as previously explained, the center B^{13} will be offset toward the right from the vertical centerline of the valve as shown in Fig. 13.

Since duct 65 is exposed to pressure, auxiliary hold-up motors 63-64 and 66-67 continue to be energized so that the center H^{12} of the main hold-up force exerted by the main and auxiliary hold-up motors does not shift from the position indicated in Fig. 12, the hold-up force center H^{12} being also indicated in Fig. 13.

It has previously been explained that the centers B and H of the blow-off and main hold-up forces are spaced apart a distance d_1 , that the excess hold-up force has its center X located upon a line passing through the force centers B and H, and that the center X of the excess force is spaced such a distance d_2 from force center H that $Bd_1 = Xd_2$. With the cylinder ports located as shown in Fig. 13, a line passing through force centers B^{13} and H^{12} lies at an angle to the horizontal centerline of the valve and the center X^{13} of the excess force is located upon that line at a considerable distance from the vertical centerline of the valve.

In machines having a sufficiently large number of cylinder ports, the center of the excess force does not shift far enough from the central part of the valve to appreciably effect the operation of the machine in which case the auxiliary hold-up motors are sufficient to compensate for variations in the blow-off force.

In most machines, however, the center of the excess force will shift to positions some of which are so close to the edge of the valve seat that there would be an insufficient area of oil film under pressure to support it which would lead to the danger of the oil film breaking down and permitting metal to metal contact which would cause abrasion of the valve and/or its seat and would also impose a brake load upon the cylinder barrel. Also, the opposite edge of the valve might be lifted away from the valve seat. To avoid such dangers, the machine may be provided with a plurality of balancing motors which are successively energized as the cylinder barrel rotates.

As shown, four balancing cylinders 72, 74, 76 and 78 (Fig. 8) are formed in valve 26 and each has a piston 79 (Fig. 6) fitted therein to engage the face of distributing block 27 so that when

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pressure is admitted to a cylinder it will act upon the end of that cylinder and urge valve 26 toward its seat 25 with a force proportional to the pressure and to the area of the cylinder. Cylinders 72 and 74 are arranged behind bridge 31 upon opposite sides of the valve centerline and communicate, respectively, with ducts 71 and 73 which terminate on the face of bridge 31 as shown in Figs. 5 and 6. Cylinders 76 and 78 are arranged behind bridge 32 upon opposite sides of the valve centerline and communicate, respectively, with ducts 75 and 77 which terminate on the face of bridge 32.

The ends of ducts 71, 73, 75 and 77 are so located in bridges 31 and 32 that, when the center B of the blow-off force has moved far enough from the vertical centerline of the valve to cause the center X of the excess hold-up force exerted by the main and auxiliary hold-up motors to shift too far from the central area of the valve, pressure is supplied to one or more of the balancing motors which add to the total hold-up force and cause the center H of the main hold-up force to move radially inward and also laterally in the same direction that the center B of the blow-off force has moved. Since the center of the excess force is located upon a line passing through the centers B and H of the blow-off and main hold-up forces, shifting the center of the main hold-up force both laterally and radially has the effect of swinging this line toward the vertical centerline of the valve so that the center of the excess force is moved inward away from the edge of the valve seat.

For example, when the cylinder ports have moved to the positions shown in Fig. 13, the center H^{13} of the main hold-up force will still be in the same position as when four cylinder ports are symmetrical about the vertical centerline of the valve but the center B^{13} of the blow-off force will have moved to the right of the vertical centerline of the valve. Since the excess hold-up force is always upon a line passing through the centers of the blow-off and main hold-up forces, the excess hold-up force X^{13} exerted by the main and auxiliary hold-up motors will have shifted a considerable distance toward the left from the vertical centerline of the valve as indicated in that figure and as previously explained.

At this time however, ducts 71, 75 and 77 are open to cylinder ports containing liquid under pressure so that cylinders 72, 76 and 78 are energized and exert additional hold-up forces upon the valve. The addition of the hold-up forces exerted by the balancing motors to the hold-up forces exerted by the main and auxiliary hold-up motors not only increases the hold-up force but also causes the center H^{13} of the total hold-up force to move inward and toward the right from the center H^{12} of the hold-up force exerted by the main and auxiliary hold-up motors as indicated in Fig. 13 and thereby cause the line passing through the centers of the blow-off and main hold-up motors to swing toward the vertical centerline of the valve so that the center XX^{13} of the excess hold-up force is located within the central area of the valve.

When four cylinder ports are in communication with high pressure valve port 29 and are symmetrical about the vertical centerline of the valve as shown in Fig. 12, ducts 71 and 75 are open to high pressure cylinder ports so that balancing cylinders 72 and 76 are energized and exert hold-up forces upon the valve. Since cylinders 72 and 76 are equally spaced from the vertical centerline

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of the valve, the resultant of the forces exerted thereby will fall upon the vertical centerline of the valve as does the resultant of the forces exerted by the main and auxiliary hold-up motors.

Adding the forces exerted by cylinders 72 and 76 to the forces exerted by the main and auxiliary motors causes the center of the total hold-up force to move radially inward but the distance moved is so small that additional illustration is deemed unnecessary and, for the purpose of the following explanation, centers H^{12} and X^{12} may be considered to be the centers of the main and auxiliary hold-up forces when cylinders 72 and 76 are energized.

During rotation of the cylinder ports through a limited angular distance from the positions shown in Fig. 12, one cylinder port gradually moves off from bridge 31 and another cylinder port gradually moves farther upon bridge 32 so that the blow-off area remains nearly constant but the center of the blow-off force moves toward the right as indicated at B^{13} in Fig. 13 and the center of the excess hold-up force gradually swings outward as indicated at X^{13} in Fig. 13.

The shifting of the center of the blow-off force and the excess hold-up force is gradual until the cylinder ports have rotated to a position of about -17° at which time the cylinder port on bridge 32 registers with duct 77 to energize cylinder 78 which causes the center of the main hold-up force to suddenly shift toward the right and the center of the excess hold-up force to shift into the central area of the valve as indicated at H^{13} and XX^{13} , respectively, in Fig. 13 and as previously explained.

As the cylinder ports continue to rotate clockwise from the positions shown in Fig. 13, the area of bridge 32 under pressure will gradually increase and the area of bridge 31 under pressure will gradually decrease which will cause the center B^{13} of the blow-off force and the center XX^{13} of the excess hold-up force to gradually move farther from the vertical centerline of the valve. However, the force centers B^{13} and XX^{13} will have moved but slightly from the positions shown in Fig. 13 before the cylinder ports will have rotated to a position of about -12.85° at which time the cylinder port on bridge 32 will open to low pressure valve port 30, as shown in Fig. 14, thereby suddenly reducing the number of cylinder ports under pressure from four to three reducing the blow-off area by an amount substantially equal to the entire area of bridge 32.

The sudden reduction in the number of cylinder ports under pressure with the resultant reduction in the blow-off area causes the blow-off force to be suddenly reduced in magnitude and its center B^{14} to suddenly shift radially outward. The center B^{14} of the blow-off force will also shift to the left of the vertical centerline of the valve, as indicated in Fig. 14, for the reason that the blow-off area still extends over a part of bridge 31.

This change in the magnitude and center of the blow-off force is compensated for by a corresponding change in the magnitude and center of the hold-up force for the reason that, when the cylinder port on bridge 32 opened to low pressure valve port 30 to reduce the blow-off area, it simultaneously opened ducts 65, 75 and 77 to port 30 so that auxiliary hold-up motors 63-64 and 66-67 and balancing cylinders 76 and 78 were de-energized and not only reduce the magnitude of the hold-up force but caused its center to shift radially outward and toward the left.

If balancing cylinder 72 were not energized, the

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total hold-up force would be exerted by hold-up motors 39 and its center H^{16} would be located upon the vertical centerline of the valve, as shown in Fig. 16, so that a line passing through centers B^{14} and H^{16} (Fig. 14) would extend toward the edge of the valve and the excess hold-up force would have its center X^{14} located upon that line at a point so close to the edge of the valve that there might not be sufficient film area under pressure to support it.

However, duct 71 is still open to pressure so that cylinder 72 is energized and exerts sufficient additional hold-up force to cause the center H^{14} of the main hold-up force to be located inward and toward the vertical centerline of the valve from H^{16} so that the center XX^{14} of the excess force is located upon that line and within the central area of the valve as indicated in Fig. 14.

As the cylinder ports continue to rotate in a clockwise direction from the positions shown in Fig. 14, the area of bridge 31 under pressure will gradually decrease and the center of the blow-off force will gradually move toward the vertical centerline of the valve but the hold-up force will remain constant and its center H^{14} will remain stationary until the cylinder ports have rotated to a position of about 4° at which time the center of the blow-off force will have moved quite close to the vertical centerline of the valve, as indicated at B^{15} in Fig. 15, and the center X^{15} of the excess hold-up force would be located in the central area of the valve.

As the cylinder ports rotate clockwise from the position shown in Fig. 15, the center of the blow-off force will continue to move toward the vertical centerline of the valve, the cylinder port overlapping bridge 31 will move out of registry with duct 71 so that the pressure in cylinder 72 will gradually decrease and cause the center of the hold-up force to move toward the right until the cylinder ports are in 0° position at which time the centers of the blow-off and hold-up forces are located on the vertical centerline of the valve as shown in Fig. 16.

In the foregoing description of the action of the auxiliary hold-up and balancing motors, it has been assumed that each would be instantly energized or de-energized in response to a high pressure valve port moving into or out of registry with the duct connected thereto. This is not strictly correct however, as there will be some leakage through the film to a motor whenever the blow-off area extends over the duct leading to that motor.

Such film leakage into a motor will raise pressure therein whenever a high pressure port is so near to the duct leading to that motor that the leakage into the motor is in excess of the leakage therefrom through the piston fit. The pressure raised in a hold-up or balancing motor by film leakage will create a slight additional hold-up force which will cause a slight change in the magnitude and center of the hold-up force but such changes are so minor that they have been disregarded in order to simplify the explanation of the action of the auxiliary hold-up and balancing motors in compensating for the changes in the magnitude and center of the blow-off force.

Rotation of the cylinder barrel from -25.7° as shown in Fig. 12 to zero as shown in Fig. 16 constitutes one half of a cylinder phase and rotation of cylinder barrel from zero to $+25.7^\circ$ constitutes the second half of a phase which is opposite to the first half. It is therefore deemed unnecessary to illustrate the various positions

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through which the cylinder ports move during the second half of the phase and only a brief description thereof will be given.

As the cylinder ports move clockwise from the positions shown in Fig. 16, a cylinder port will advance across bridge 32 and cause the blow-off area to increase and the center of the blow-off force to move to the right of the vertical centerline. When the cylinder ports have reached a position of about $+4^\circ$, duct 75 will be uncovered and cylinder 76 will be energized which will cause the center of the hold-up force to move toward the right so that the centers of the blow-off, main hold-up and excess hold-up forces will be located in positions similar to but on the opposite side of the vertical centerline from the positions indicated at B^{15} , H^{15} and X^{15} on Fig. 15.

As the cylinder port continues to move across bridge 32, the blow-off area will increase and cause the center of the blow-off force to move farther toward the right but the center of the excess hold-up force will not move too close to the edge of the valve for the reason that cylinder 76 is energized.

When the cylinder barrel has rotated to a position of about $+12.85^\circ$, the cylinder on bridge 31 will open to high pressure port 29, thereby suddenly changing the number of cylinder ports under pressure from three to four, extending the blow-off area over bridge 31 and energizing auxiliary hold-up motors 63—64 and 66—67 and balancing cylinders 72 and 74. Increasing the blow-off area and the number of cylinders under pressure will cause the blow-off force to increase in magnitude and its center to shift radially inward and toward the left but energizing auxiliary motors 63—64 and 66—67 and balancing cylinders 72 and 74 causes the hold-up force to increase in magnitude and its main center to shift radially inward and toward the left to compensate for the increase and shift of the blow-off force and to maintain the center of the excess hold-up force within the central area of the valve, the centers of the blow-off force, the main hold-up force and the excess hold-up force being located in positions somewhat similar to but opposite to the positions indicated at B^{13} , H^{13} and XX^{13} on Fig. 13.

During continued rotation of the cylinder barrel, the blow-off area will gradually decrease on bridge 31 and gradually increase on bridge 32 which will cause the center of the blow-off force to move gradually toward the vertical centerline. As the cylinder ports pass a position of about $+17^\circ$, the cylinder port on bridge 31 passes out of registry with duct 73 so that cylinder 74 is deenergized and the center of the hold-up force shifts toward the vertical centerline. When the cylinder barrel has rotated to $+25.7^\circ$, the blow-off areas on the bridges will be equal and the centers of the blow-off and hold-up forces will be located on the vertical centerline as shown in Fig. 12, thus completing one cylinder phase and a similar cylinder phase will be repeated each time the cylinder barrel rotates through an angular distance of 51.4° .

The hold-up force on each of valves 26 and 26^a is thus so varied as to its magnitude and as to the location of its center that it compensates for variations in the magnitude and the center of the blow-off force in all critical positions of the cylinder barrel and maintains each valve in the proper relation to its seat at all times, thereby enabling the machine to operate efficiently and, when

functioning as a motor, to operate at slow speed and to start under full load.

It has previously been stated that, when the machine is to function as a pump, it is supercharged by an auxiliary pump which is capable of supplying at a predetermined pressure the entire volume of liquid required by the machine in order that the intake ports in the valves and the intake passages need be no larger, respectively, than the discharge ports in the valves and the discharge passages. By keeping the valve ports as small as possible, the area of each valve may be maintained at a minimum so that the blow-off area and consequently the blow-off force may each be maintained at a minimum. By keeping the passages as small as possible, the overall diameter of the machine may be kept at a minimum.

As shown, channels 50 and 52 (Fig. 1) are connected, respectively, through check valves 82 and 83 to opposite ends of a channel 84 which, as shown in Fig. 1A, is connected intermediate its ends by a channel 85 to the outlet of an auxiliary pump 86, check valves 82 and 83 permitting liquid to flow freely from channel 84 into either channel 50 or 52 but preventing flow of liquid in the opposite direction.

In order that the machine shown in Fig. 1 may be supercharged with liquid at a predetermined pressure when it is at its maximum displacement, auxiliary pump 86 should have a volumetric capacity in excess of the maximum capacity of the machine shown in Fig. 1 particularly as some of the liquid may be required for control purposes such as for operating the displacement control shown in Fig. 2.

While pump 86 may be of any suitable type, it preferably is a variable displacement constant pressure pump and it has been shown as being of the vane type but it has not been fully illustrated and will be only briefly described for the reason that it is similar in all essential respects to the pump fully disclosed in Patent No. 2,238,062 to which reference may be had for details of construction.

As shown, pump 86 has a plurality of slidable vanes 87 arranged substantially radially in a circular rotor 88 which is fitted between two cheek plates 89 and 90 and inside of an annular spacing ring 91. The rotor, cheek plates and spacer ring are fitted within a suitable circular recess formed in the pump casing 92 which may be attached to the casing 1 of the machine shown in Fig. 1. Rotor 88 in the present instance is splined upon a drive shaft 93 which extends outward through casing 92, through distributing block 27^a (Fig. 1) and valve 26^a and into cylinder barrel 4 to which it is splined so that rotor 88 is driven in unison with cylinder barrel 4.

The outer ends of vanes 87 engage a normally elliptical vane track 94 which extends around rotor 88 inside spacer ring 91 and has its long diameter arranged in the plane of Fig. 1A and greater than the diameter of rotor 88 to provide two pumping chambers upon opposite sides of rotor 88. At the ends of its long diameter, vane track 94 is fixed to two movable bridges 95 which are closely fitted between cheek plates 89 and 90 and with which the ends of the vanes cooperate to divide each pumping chamber into an intake chamber and a discharge chamber. The intake chambers (not shown) are connected to an intake port 96 (Fig. 1A) which is connected by a channel 97 to a tank or other suitable source of liquid. The outlet chambers (not shown) are

connected to an outlet port 98 to which channel 85 is connected.

At the ends of its short diameter, vane track 94 is arranged very close to the periphery of rotor 88 and is fixed to spaced ring 91 to provide two stationary bridges (not shown) which are spaced 90° from bridges 95 and with which the ends of the vanes cooperate to provide seals between the two pumping chambers.

The arrangement is such that, when rotor 88 is rotated, the vanes in passing from the stationary bridges through the intake chambers to the movable bridges will draw liquid through channel 97 and port 96 into the intake chambers and, in moving across the movable bridges, the liquid trapped between adjacent vanes will be discharged into the discharge chambers and thence through outlet port 98, the rate at which liquid is discharged being substantially proportional to the sum of the distances between movable bridges 95 and the periphery of rotor 88 as fully explained in Patent No. 2,238,062.

In order to automatically regulate pump displacement, two cylinders 101 are carried by or formed in pump casing 92 diametrically opposite each other and two pistons 102 are fitted, respectively, in cylinders 101 and connected to the two movable bridges 95, respectively, by suitable stems or piston rods which extend through suitable openings in spacer ring 91, each piston being urged outward by a spring 103 and by pump pressure acting upon the face of the bridge 95 connected thereto as fully disclosed in Patent No. 2,238,062.

Cylinders 101 are connected, respectively, by two channels 104 to channel 84 so that pistons 102 are urged inward by pump pressure acting upon the outer faces thereof. The tension of each spring 103 and the area of each piston 102 relative to the pressure area on the bridge 95 connected thereto are so proportioned that pistons 102 will be held in their outermost positions until pump pressure reaches a predetermined maximum, which is high enough to force liquid at the required velocity through the passages and ports in the machine shown in Fig. 1, and then the pressure acting upon pistons 102 will cause them to move inward and reduce the displacement of pump 86 until the rate of pump delivery is just sufficient to maintain the maximum pressure constant.

For the purpose of illustration, the machine shown in Fig. 1 has been shown in Fig. 11 as being supercharged by auxiliary pump 86 and as being employed as a power pump to supply motive liquid to a cylinder 110 having a piston 111 fitted therein and provided with a rod 112 which extends through one end only of cylinder 110, the power pump being indicated in its entirety on Fig. 11 by reference numeral 1 and its control mechanism being indicated by the reference numerals 18-23 which correspond, respectively, to the reference numerals applied in Figs. 1 and 2 to the casing of the machine and to the parts of the control mechanism.

As shown, auxiliary pump 86 is supplied with liquid from a reservoir 113 into which the liquid discharged from cylinder 110 flows through a discharge channel 114 which is connected intermediate the ends of a valve casing 115 having a valve 116 fitted therein. Valve casing 115 has channels 51 and 53 connected to opposite ends thereof and it is connected at opposite sides of discharge channel 114 to opposite ends of cylinder 110 by channels 117 and 118 respectively.

Assuming that the parts are in the positions shown, that power pump 1 is at zero stroke, that the apparatus is idle and that there is no pressure in the circuit so that the displacement of pump 86 is maximum, the apparatus will operate as follows:

When shaft 2 is driven, main pump 1 and auxiliary pump 86 will both be driven for the reason that cylinder barrel 4 (Fig. 1) is connected to shaft 2 and rotor 88 (Fig. 1A) is connected to cylinder barrel 4 by shaft 93. Auxiliary pump 86 will draw liquid from reservoir 113 and discharge it into channel 84 at maximum rate until the pressure in channel 84 reaches the predetermined maximum and then the pressure will cause pistons 102 to move movable bridges 95 inward until pump 86 is discharging at a very limited rate which is just enough to maintain that maximum pressure constant. The maximum pressure will extend through check valves 82 and 83 to both sides of pump 1.

If control 18-23 is then operated to cause power pump 1 to discharge into channel 50, liquid can then flow from channel 52 into pump 1 which will cause the pressure in channel 84 to drop momentarily and permit springs 103 to move pistons 102 and bridges 95 outward and increase the displacement of pump 86 until it is supplying the entire requirements of pump 1 and has increased the pressure to maximum and then the displacement of pump 86 becomes substantially constant. The liquid discharged by auxiliary pump 86 will then flow through check valve 83 and channel 52 to pump 1 to supercharge it and the liquid discharged by pump 1 will flow through channels 50 and 51, valve casing 115 and channel 117 to cylinder 110 and cause piston 111 to move toward the left and expel liquid from cylinder 110 through channel 118, valve casing 115 and channel 114 into reservoir 113 as indicated by the arrows on Fig. 11.

When pump 1 is reversed, as by operating control 18-23, it will discharge into channel 52 and the liquid discharged by auxiliary pump 86 will flow from channel 84 through check valve 82 and channel 50 into pump 1 to supercharge it at the predetermined maximum pressure and the liquid discharged by pump 1 will flow through channels 52 and 53 to valve casing 115, shift valve 116 toward the right and then flow through valve casing 115 and channel 118 to cylinder 110 and cause piston 111 to move toward the right and expel liquid from cylinder 110 through channel 117, valve casing 115 and channel 114 into reservoir 113.

The hydrodynamic machine described herein is susceptible of various 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 and passages leading outward from said cylinders, certain of said passages being arranged to terminate at one end of said cylinder barrel and the other passages being arranged to terminate at the other end of said cylinder barrel, two valves arranged at opposite ends of said cylinder barrel and cooperating with said passages to control the flow of liquid to and from said cylinders, and means for maintaining said valves in operative relation with the ends of said cylinder barrel.

2. In a hydrodynamic machine, the combination of a rotatable cylinder barrel having pistons and cylinders arranged therein and passages lead-

ing from a part of said cylinders through one end of said cylinder barrel and other passages leading from the other cylinders through the other end of said cylinder barrel, two valves arranged at opposite ends of said cylinder barrel and cooperating with said passages to control the flow of liquid to and from said cylinders, and means for urging said valves toward said cylinder barrel by applying equal forces to the outer ends of said valves so that the force exerted upon said cylinder barrel by one valve is counterbalanced by the force exerted thereon by the other valve.

3. In a hydrodynamic machine, the combination of a rotatable cylinder barrel having pistons and cylinders arranged therein and passages leading from a part of said cylinders through one end of said cylinder barrel and other passages leading from the other cylinders through the other end of said cylinder barrel, two valves arranged at opposite ends of said cylinder barrel and cooperating with said passages to control the flow of liquid to and from said cylinders, and means for urging said valves toward said cylinder barrel by applying to the outer ends thereof forces which vary in accordance with variations in the liquid pressures in said valves and which are equal and opposite in direction so that the force exerted upon said cylinder barrel by one valve is counterbalanced by the force exerted thereon by the other valve.

4. In a hydrodynamic machine, the combination of a rotatable cylinder barrel having pistons and cylinders arranged therein and passages leading from a part of said cylinders through one end of said cylinder barrel and other passages leading from the other cylinders through the other end of said cylinder barrel, two valves arranged at opposite ends of said cylinder barrel and cooperating with said passages to control the flow of liquid to and from said cylinders, means for maintaining said valves in operative relation with the ends of said cylinder barrel, and a shaft connected to said cylinder barrel and extending outward through one of said valves for driving or to be driven by said cylinder barrel.

5. In a hydrodynamic machine, the combination of a rotatable cylinder barrel having a valve seat arranged upon an end thereof and pistons and cylinders arranged therein and passages leading from said seat to said cylinders, a valve arranged with its face substantially in engagement with said seat to control the flow of liquid to said cylinders and having formed in its face two diametrically opposed substantially crescent shaped ports with which each of said passages registers alternately as said cylinder barrel rotates, a drive shaft extending through said valve between said ports and connected to said cylinder barrel to drive the same or to be driven thereby, and hydraulic means arranged alongside said shaft for urging said valve against said seat.

6. In a hydrodynamic machine, the combination of a rotatable cylinder barrel having pistons and cylinders arranged therein and passages leading from a part of said cylinders through one end of said cylinder barrel and other passages leading from the other cylinders through the other end of said cylinder barrel, two valves each having a high pressure port and a low pressure port formed therein arranged at opposite ends of said cylinder barrel and cooperating with said passages to control the flow of liquid to and from said cylinders, said liquid spreading over the face of each valve and the adjacent end of said cylinder barrel to form a lubricating film there-

between and the pressure in said high pressure ports extending into said films and tending to move said valves away from said cylinder barrel, means for holding said valves substantially in engagement with the ends of said cylinder barrel, and channels connecting said high pressure ports together, so that the pressures prevailing in the high pressure port and in the film at one end of said cylinder barrel are identical to the pressures prevailing in the high pressure port and the film at the other end of said cylinder barrel, to thereby hydrostatically balance said cylinder barrel.

7. In a hydrodynamic machine, the combination of a rotatable cylinder barrel having pistons and cylinders arranged therein and passages leading from a part of said cylinders through one end of said cylinder barrel and other passages leading from the other cylinders through the other end of said cylinder barrel, two valves each having a high pressure port and a low pressure port formed therein arranged at opposite ends of said cylinder barrel and cooperating with said passages to control the flow of liquid to and from said cylinders, said liquid spreading over the face of each valve and the adjacent end of said cylinder barrel to form a lubricating film therebetween and the pressure in said high pressure ports extending into said films and tending to move said valves away from said cylinder barrel, a shaft connected to said cylinder barrel and extending outward through one of said valves for transmitting motion to or from said cylinder barrel, means for holding said valves substantially in engagement with the ends of said cylinder barrel, and channels connecting said high pressure ports together, so that the pressures prevailing in the high pressure port and in the film at one end of said cylinder barrel are identical to the pressures prevailing in the high pressure port and the film at the other end of said cylinder barrel, to thereby hydrostatically balance said cylinder barrel.

8. In a hydrodynamic machine, the combination of a rotatable cylinder barrel having pistons and cylinders arranged therein and cylinder ports arranged at opposite ends thereof and communicating with said cylinders, two valves arranged at opposite ends of said cylinder barrel and provided with high pressure ports and low pressure ports which cooperate with said cylinder ports to control the flow of liquid to and from said cylinders, and means for maintaining said valves in operative relation with the ends of said cylinder barrel, said cylinder ports and valve ports being so arranged relative to each other that during rotation of said cylinder barrel only one cylinder port will open to a valve port at any one time.

9. In a hydrodynamic machine, the combination of a rotatable cylinder barrel having pistons and cylinders arranged therein and cylinder ports arranged at opposite ends thereof and communicating with said cylinders, two valves arranged at opposite ends of said cylinder barrel and provided with high pressure ports and low pressure ports which cooperate with said cylinder ports to control the flow of liquid to and from said cylinders, and means for maintaining said valves in operative relation with the ends of said cylinder barrel, said cylinder ports and valve ports being so arranged that a plurality of cylinder ports will open successively to high pressure valve ports and a plurality of cylinder ports will open successively to low pressure valve ports but

only one cylinder port will open to a valve port at any given instant.

10. In a hydrodynamic machine, the combination of a rotatable cylinder barrel having pistons and cylinders arranged therein and cylinder ports arranged at opposite ends thereof and communicating with said cylinders, two valves arranged at opposite ends of said cylinder barrel and provided with high pressure ports and low pressure ports which cooperate with said cylinder ports to control the flow of liquid to and from said cylinders, and means for maintaining said valves in operative relation with the ends of said cylinder barrel, said valves being so arranged that the ports in one valve are in axial alignment with the corresponding ports in the other valve and said cylinder barrel having the ports in one end thereof offset from the ports in the other end thereof so that during rotation of said cylinder barrel only one cylinder port will open to a valve port at any one time.

11. In a hydrodynamic machine, the combination of a rotatable cylinder barrel having pistons and cylinders arranged therein and cylinder ports arranged at opposite ends thereof and communicating with said cylinders, two valves arranged at opposite ends of said cylinder barrel and provided with high pressure ports and low pressure ports which cooperate with said cylinder ports to control the flow of liquid to and from said cylinders, and means for maintaining said valves in operative relation with the ends of said cylinder barrel, said valves being so arranged that the ports in one valve are in axial alignment with the corresponding ports in the other valve and said cylinder barrel having the ports in one end thereof offset such a distance from the ports in the other end thereof that during rotation of said cylinder barrel ports in opposite ends of said cylinder barrel will open successively to high pressure valve ports and ports in opposite ends of said cylinder barrel will open successively to low pressure valve ports but only one cylinder port will open to a valve port at any one time.

12. In a hydrodynamic machine provided with a rotatable cylinder barrel having a plurality of pistons and cylinders arranged therein, a valve seat arranged upon the end thereof and a plurality of cylinder ports formed in said seat and communicating with said cylinders, the combination of a valve engaging said seat and provided with inlet and discharge ports with which each of said cylinder ports registers alternately as said cylinder barrel rotates, hold-up motors communicating with said valve ports and energized by the pressure therein for exerting a hold-up force upon said valve to urge it toward said seat, and a plurality of auxiliary motors adapted when energized to exert additional hold-up force upon said valve, said auxiliary motors being connected to each other and each having a duct extending therefrom through the face of said valve into the path of said cylinder ports so that said auxiliary motors are energized in response to a cylinder port having pressure therein registering with one of said ducts.

13. In a hydrodynamic machine provided with a rotatable cylinder barrel having a plurality of pistons and cylinders arranged therein, a valve seat arranged upon the end thereof and a plurality of cylinder ports formed in said seat and communicating with said cylinders, the combination of a valve engaging said seat and provided with inlet and discharge ports with which each

of said cylinder ports registers alternately as said cylinder barrel rotates, hold-up motors communicating with said valve ports and energized by the pressure therein for exerting a hold-up force upon said valve to urge it toward said seat, and a plurality of auxiliary motors spaced from said hold-up motors and adapted when energized to exert additional hold-up force upon said valve, said auxiliary motors being connected to each other and each having a duct extending therefrom through the face of said valve into the path of said cylinder ports so that when a cylinder port having pressure therein registers with one of said ducts said auxiliary motors will be energized and cause said hold-up force to be increased and its center to be shifted.

14. In a hydrodynamic machine provided with a rotatable cylinder barrel having a plurality of pistons and cylinders arranged therein, a valve seat arranged upon the end thereof and a plurality of cylinder ports formed in said seat and communicating with said cylinders, the combination of an annular valve engaging said seat and provided with inlet and discharge ports with which each of said cylinder ports registers alternately as said cylinder barrel rotates, a shaft extending through said valve and connected to said cylinder barrel to rotate the same, hold-up motors communicating with said valve ports and energized by the pressure therein for exerting a hold-up force upon said valve to urge it toward said seat, and two auxiliary motors arranged at opposite sides of said shaft upon the centerline of said valve and adapted when energized to exert additional hold-up force upon said valve, said auxiliary motors being connected to each other and each having a duct extending therefrom through the face of said valve into the path of said cylinder ports so that both of said auxiliary motors are energized in response to a cylinder port having pressure therein registering with either of said ducts.

15. In a hydrodynamic machine provided with a rotatable cylinder barrel having a plurality of pistons and cylinders arranged therein, a valve seat arranged upon the end thereof and a plurality of cylinder ports formed in said seat and communicating with said cylinders, the combination of a valve engaging said seat and provided with inlet and discharge ports with which each of said cylinder ports registers alternately as said cylinder barrel rotates, hold-up motors communicating with said valve ports and energized by the pressure therein for exerting a hold-up force upon said valve to urge it toward said seat, a plurality of auxiliary motors adapted when energized to exert additional hold-up force upon said valve, said auxiliary motors being connected to each other and each having a duct extending therefrom through the face of said valve into the path of said cylinder ports so said auxiliary motors are energized in response to a cylinder port having pressure therein registering with one of said ducts, and a plurality of balancing motors adapted when supplied with pressure to exert additional hold-up forces upon said valve and each having a duct leading therefrom through the face of said valve into the path of said cylinder ports so that pressure is supplied to each balancing motor in response to a cylinder port containing pressure registering with the duct connected thereto.

16. In a hydrodynamic machine, the combination with a cylinder unit having a valve seat arranged upon an end thereof and pistons and cyl-

inders arranged therein and passages leading from said seat to said cylinders, of a valve arranged with its face substantially in engagement with said seat to control the flow of liquid to said cylinders and having formed in its face an elongated pressure port with which each of said passages registers during operation of said machine, and at least one strut extending across said port to stiffen the face of said valve and thereby prevent any material distortion thereof due to changes in temperature or pressure.

17. In a hydrodynamic machine, the combination with a cylinder unit having a valve seat arranged upon an end thereof and pistons and cylinders arranged therein and passages leading from said seat to said cylinders, of a valve arranged with its face substantially in engagement with said seat to control the flow of liquid to said cylinders and having formed in its face two diametrically opposed substantially elongated ports with which each of said passages registers alternately during operation of said machine, and a plurality of struts spaced from said face and extending across said ports to stiffen said face and thereby prevent any material distortion thereof due to changes in temperature and pressure.

18. A pumping unit, comprising a main pump having a large volumetric capacity, means for varying the displacement of said pump, an auxiliary pump connected to said main pump to supply all of the liquid required thereby, and means operable substantially simultaneously with the operation of said displacement varying means during operation of said main pump for correspondingly varying the displacement of said auxiliary pump.

19. A pumping unit comprising a variable displacement main pump having a large volumetric capacity, a variable displacement auxiliary pump mechanically connected to said main pump to be driven in unison therewith and hydraulically connected to said main pump for delivering thereto sufficient liquid under pressure to supply the entire requirements of said main pump, and means responsive to the pressure created by said auxiliary pump for varying the displacement of said auxiliary pump to thereby enable said auxiliary pump to maintain its output equal to but not in excess of requirements.

20. In a hydrodynamic machine provided with a rotatable cylinder barrel having a plurality of pistons and cylinders arranged therein, a valve seat arranged upon the end thereof and a plurality of cylinder ports formed in said seat and communicating with said cylinders, the combination of a valve engaging said seat and provided with arcuate diametrically opposed high and low pressure ports with each of which said cylinder ports register successively as said cylinder barrel rotates so that the number of cylinder ports containing high pressure varies between a minority and a majority and the pressure in said cylinder ports causes alternate minor and major blow-off forces to be exerted upon said valve, and means for exerting upon said valve hold-up forces in opposition to said blow-off forces and including main hold-up means continuously energized by pressure from said high pressure valve port and so located and proportioned as to prevent said minor blow-off force from lifting said valve from its seat, and auxiliary hold-up means intermittently energized in response to the number of cylinder ports containing high pressure changing from a minority to a majority, said auxiliary hold-up means being so proportioned and located

that the total force exerted by said main and auxiliary hold-up means prevents said major blow-off force from lifting said valve from its seat.

21. In a hydrodynamic machine provided with a rotatably cylinder barrel having a plurality of pistons and cylinders arranged therein, a valve seat arranged upon the end thereof and a plurality of cylinder ports formed in said seat and communicating with said cylinders, the combination of a valve engaging said seat and provided with high pressure and low pressure ports with which each of said cylinder ports registers alternately and with each of which said cylinder ports register successively as said cylinder barrel rotates so that the number of cylinder ports containing high pressure varies between a minority and a majority and the pressure in said cylinder ports causes alternate minor and major blow-off forces to be exerted upon said valve, and means for exerting upon said valve hold-up forces in opposition to said blow-off forces and including main hold-up means continuously exerting upon said valve a hold-up force which is proportional to the pressure in said high pressure port and is somewhat greater than said minor blow-off force and has its center spaced radially inward from the center of said minor blow-off force, and auxiliary hold-up means adapted to exert additional hold-up force upon said valve in response to the number of cylinder ports containing high pressure changing from a minority to a majority, said auxiliary hold-up means being so proportioned and located that the auxiliary and main hold-up means together exert a total hold-up force which is somewhat greater than the major blow-off force and has its center spaced inward from the center of the major blow-off force.

22. In a hydrodynamic machine, the combination with a cylinder unit having pistons and cylinders arranged therein and a valve seat arranged upon an end thereof and passages leading from said seat to said cylinders, of a valve for controlling the flow of liquid to and from said cylinders and comprising an integral structure having a front wall arranged substantially in engagement with said seat, a rear wall connected to said front wall by peripheral wall and an internal wall whereby said walls enclose two chambers one of which is adapted to contain liquid under high pressure, said rear wall being pierced by openings for the ingress and egress of liquid which openings are not large enough to materially affect the rigidity of said rear wall, said front wall functioning as a precise planar valve face and being pierced and thereby weakened by two long arcuate ports which communicate with said two chambers respectively and have their outer boundaries close to or in alignment with the inner face of said peripheral wall so that pressure in said higher pressure chamber could distort said peripheral web, and at least one strut extending across the high pressure port to prevent radial distortions of said peripheral wall which distortions if permitted in the plane of the weakened front wall and prevented by the rigid rear wall would necessarily distort the face of the front wall from its precise planar form.

23. A pumping unit, comprising a reversible variable delivery main pump having a large volumetric capacity and ports which function interchangeably as intake and discharge ports, an auxiliary pump driven in unison with said main pump, and means for connecting said aux-

iliary pump to said ports and including means for directing the entire output of said auxiliary pump to the port which at that time is functioning as an intake port, said auxiliary pump having displacement varying means normally tending to adjust it to maximum displacement and adapted to decrease its displacement in response to auxiliary pump pressure exceeding a predetermined maximum to thereby enable said auxiliary pump to supercharge said main pump at a predetermined uniform pressure.

24. A pumping unit comprising a variable delivery main pump having a large volumetric capacity, a large capacity auxiliary pump connected to said main pump and tending to deliver liquid thereto in excess of the requirements of said main pump, and means solely responsive to auxiliary pump pressure reaching a predetermined maximum for reducing the displacement of said auxiliary pump to thereby enable said auxiliary pump to supercharge said main pump at said maximum pressure.

25. A pumping unit comprising a casing, a variable delivery main pump arranged within said casing, a variable delivery auxiliary pump arranged within said casing and driven in unison with said main pump, said auxiliary pump being connected to said main pump and tending to deliver liquid thereto in excess of the requirements of said main pump, and means solely responsive to auxiliary pump pressure reaching a predetermined maximum for reducing the displacement of said auxiliary pump to thereby enable said auxiliary pump to supercharge said main pump at said maximum pressure.

26. A hydraulic pump or motor unit comprising a casing having a bearing at one end and inlet and outlet ports at its opposite end, a rotor having a shaft at one end journaled in the bearing in the casing and a radial valve-face at its opposite end, said rotor embodying a series of cylinders with ports opening through the valve-face, pistons in the cylinders, a valve-plate adapted to seat against the valve-face on the end of the rotor and provided with ports for registering with the ports therein, and a plurality of hollow pistons projecting from the side of the valve-plate opposite to that seated against the valve-face on the rotor with their bores communicating with the ports in the valve-plate, said hollow pistons projecting into the inlet and outlet ports in the casing to adapt the motive fluid acting against their annular ends to maintain the valve-plate pressed against the valve-face on the rotor with a uniformly-distributed force, and the hollow pistons on the pressure side of said unit being located with their axes radially outward from the axis of the valve-plate at such a distance that the resultant force representing the combination of the thrust on the hollow pistons and the blow-off force shall lie within one-half of the outside radius of the valve-plate bearing area on the rotor.

27. A hydraulic pump or motor unit comprising a casing having inlet and outlet ports for the motive fluid, a rotor embodying a plurality of cylinders rotatable within the casing and having a valve-face at one end, ports in the valve-face communicating with the cylinders, pistons in the cylinders, a valve-plate having one side engaging the valve-face at the end of the rotor and formed with ports communicating with the ports therein, and a plurality of hollow pistons on the opposite side of the valve-plate projecting

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therefrom into the ports in the end of the casing, said hollow pistons adapted to receive the pressure of the motive fluid against their annular ends and the hollow pistons on the pressure side of said unit being so arranged and located at a distance radially outward from the axis of the valve-plate that the resultant force representing the combination of the thrust on said hollow pistons and the blow-off force shall lie within one-half of the outside radius of the valve-plate bearing area on the rotor.

28. A pumping unit comprising a variable delivery main pump adapted to deliver liquid to an external circuit, a variable delivery auxiliary pump connected to said main pump for supercharging the same and tending to deliver liquid to said main pump at a rate in excess of the requirements of said main pump, and means solely responsive to auxiliary pump pressure reaching a predetermined maximum for reducing the displacement of said auxiliary pump until said auxiliary pump is delivering just enough liquid to said main pump to maintain said maximum pressure constant regardless of the pressure created in said external circuit by said main pump.

29. A pumping unit comprising a main pump adapted to deliver liquid to an external circuit, a vane pump connected to said main pump to supercharge the same and having displacement

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varying means urged toward maximum displacement position and causing said vane pump to tend to deliver liquid to said main pump in excess of the requirements thereof, and means operated by liquid from said vane pump and responsive to vane pump pressure reaching a given maximum for operating said displacement means to adjust said vane pump until it is delivering just enough liquid to maintain said maximum pressure constant.

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