Feb. 27, 1951 A. GABRIEL 2,543,624

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FIG. 1.

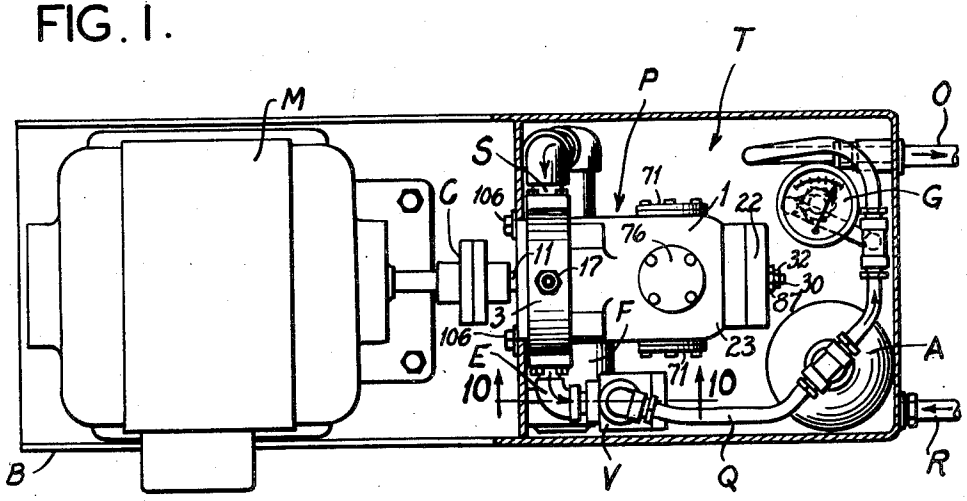
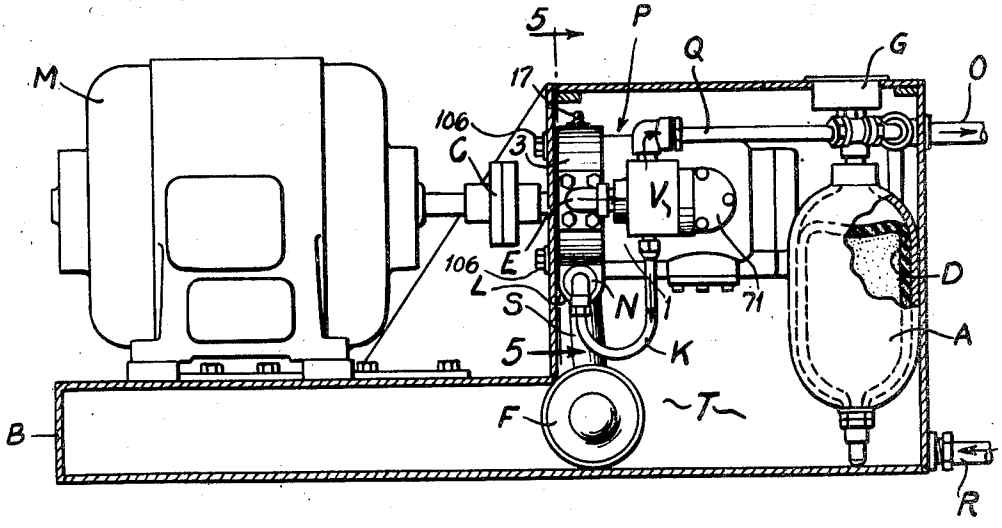


FIG. 2.



Adam Gabriel,
Inventor.
Haynes and Koenig,
Attorneys.

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FIG. 3.

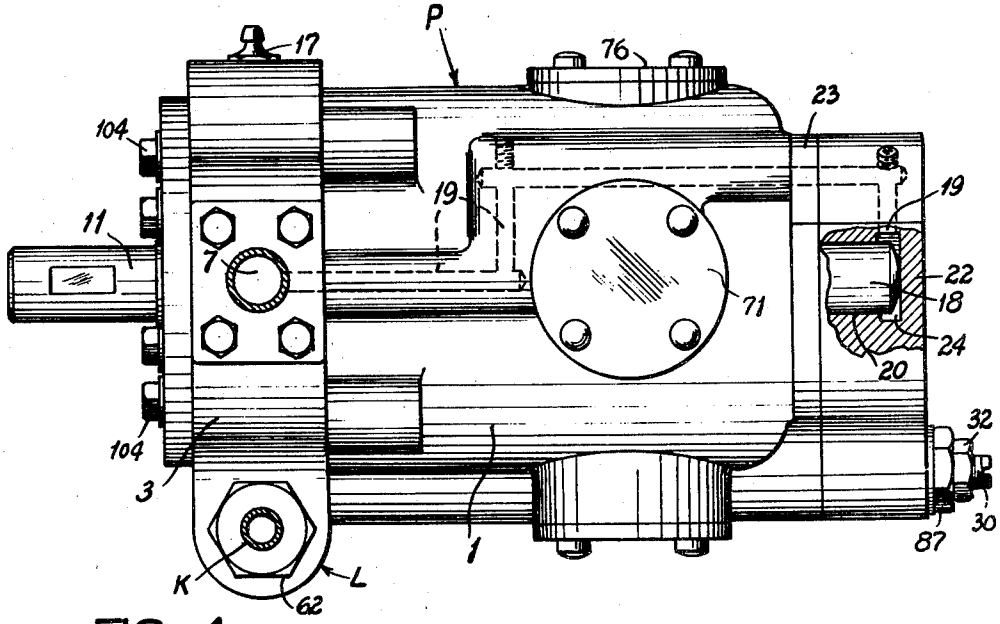
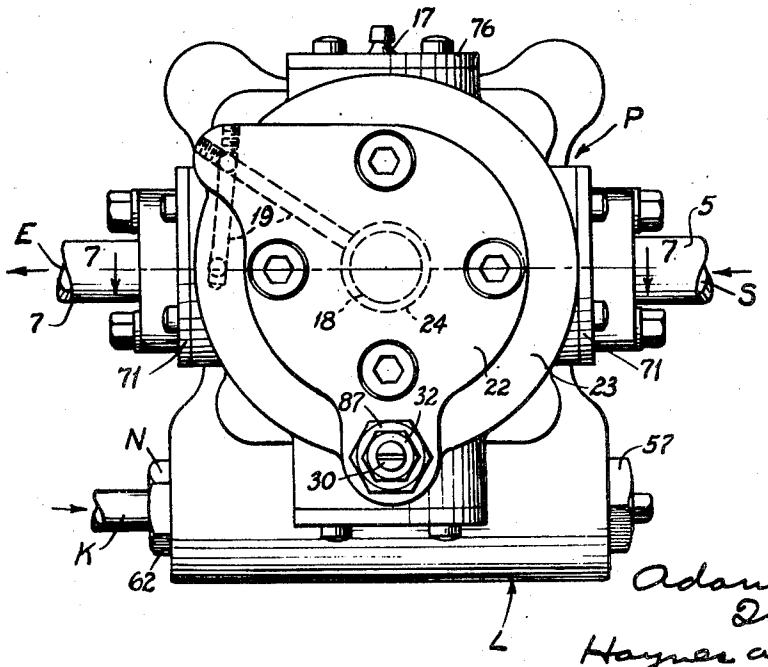


FIG. 4.



Adam Gabriel,
Inventor.
Haynes and Koenig
Attorneys.

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FIG. 5.

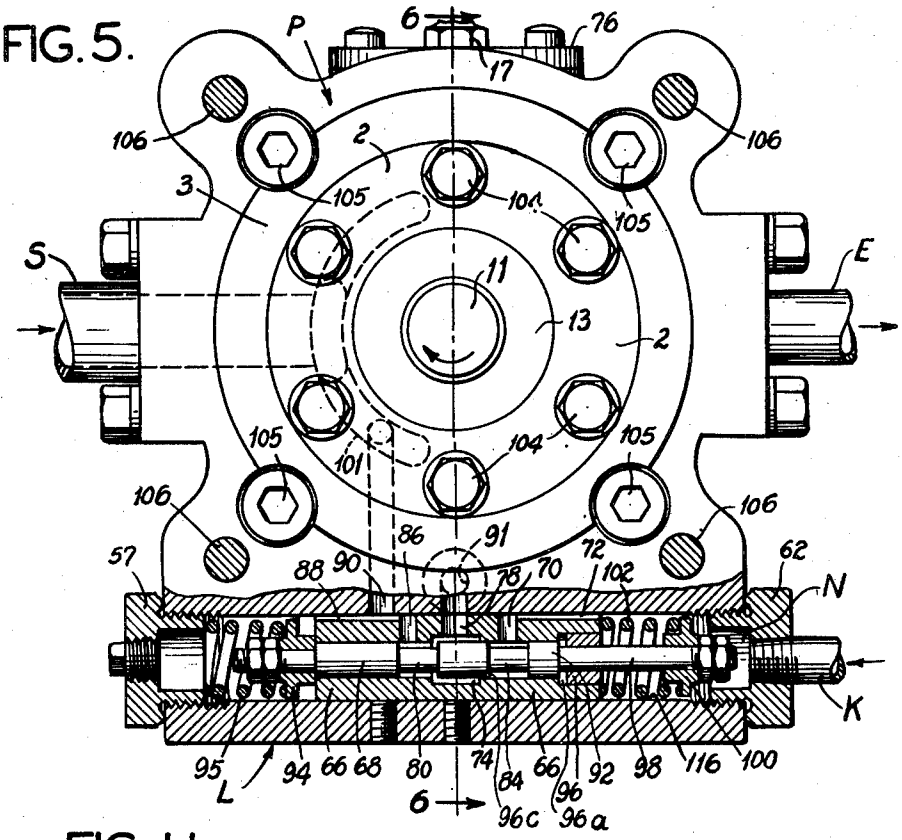
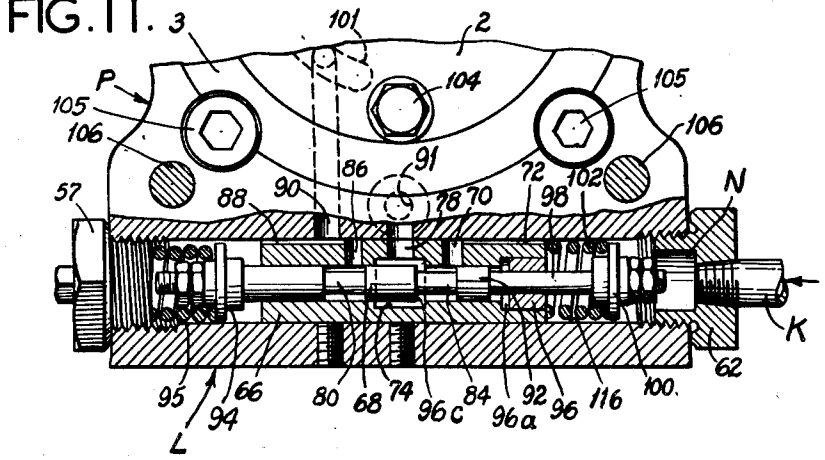


FIG. 11.



Adam Gabriel,
Inventor.
Haynes and Koenig,
Attorneys.

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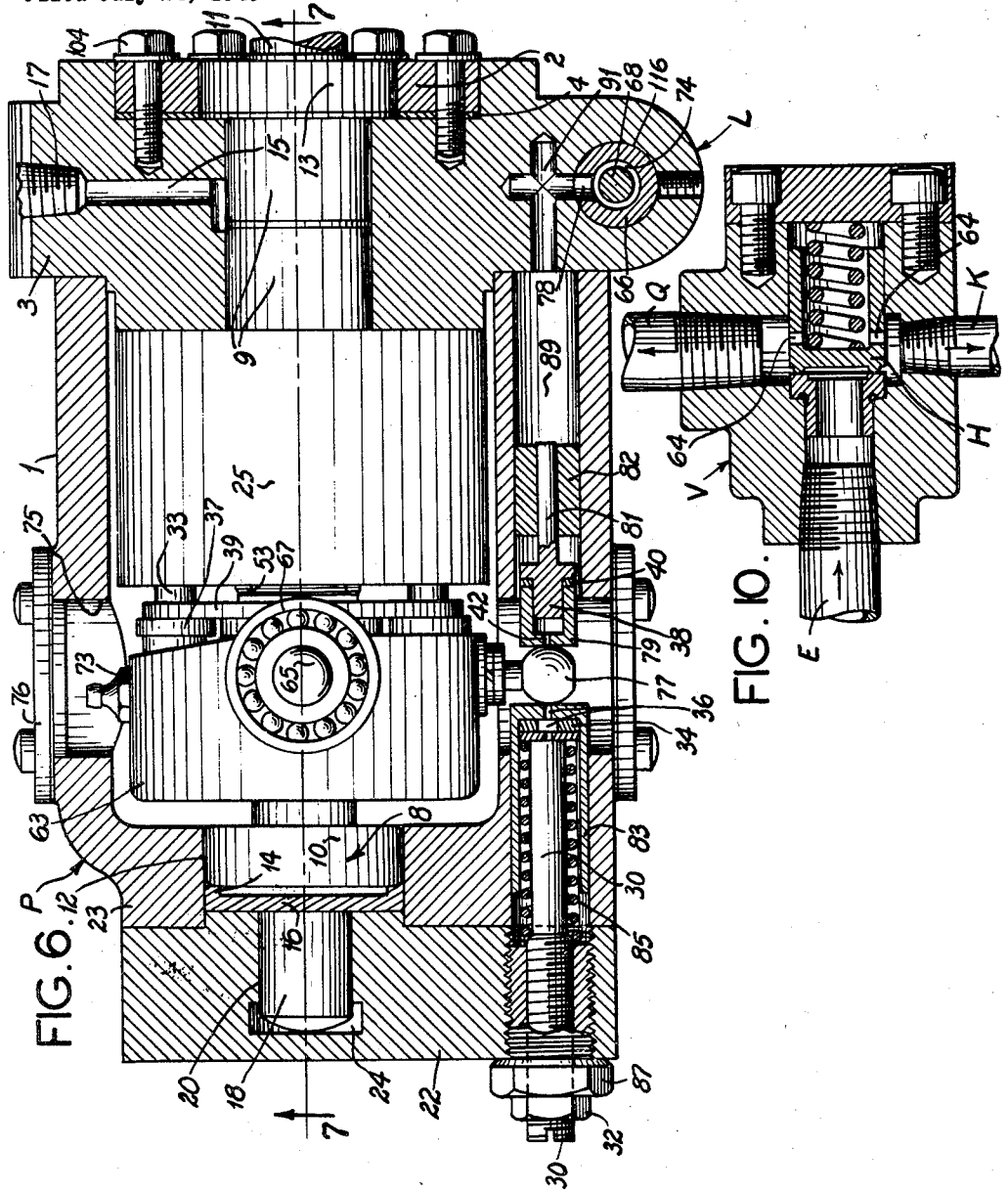


FIG. 6.

FIG. 10.

Adam Gabriel,
Inventor.
Haynes and Koenig,
Attorneys.

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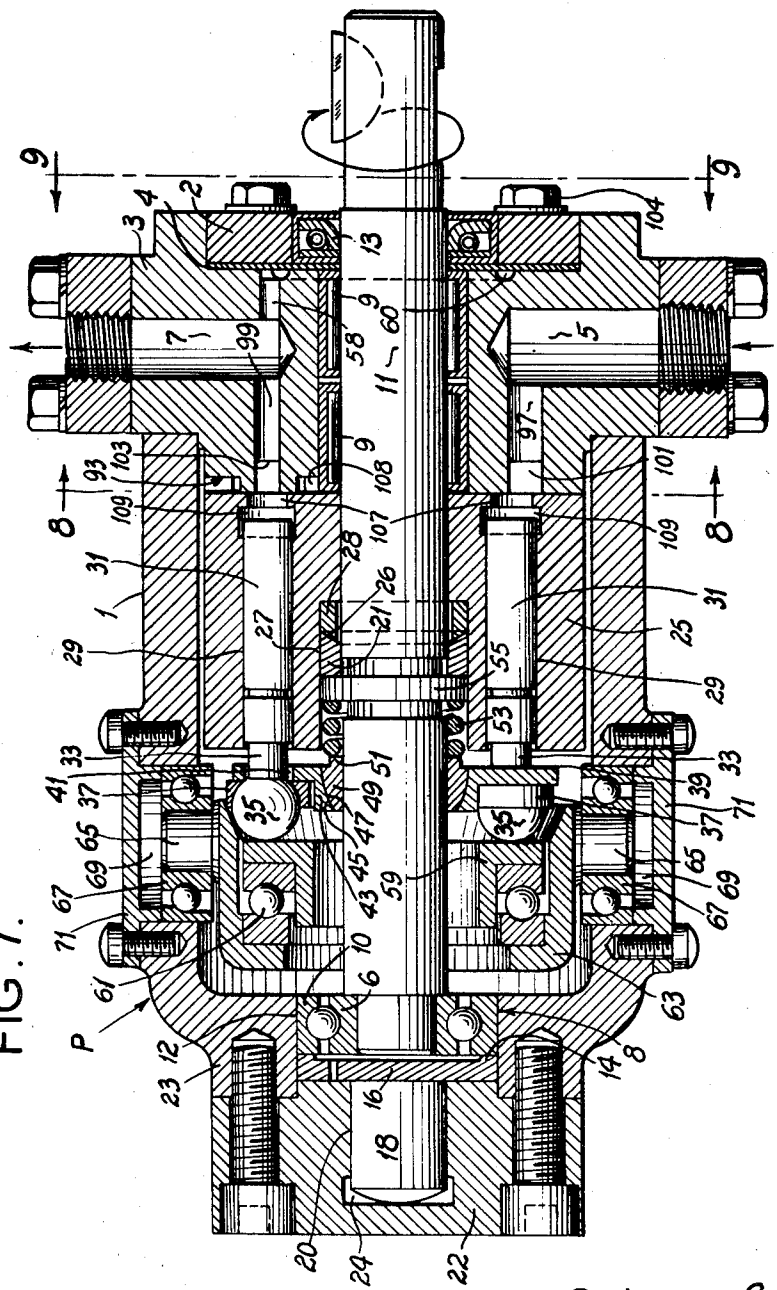


FIG. 7.

Adam Gabriel,
Inventor.
Haynes and Koenig,
Attorneys.

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FIG. 8.

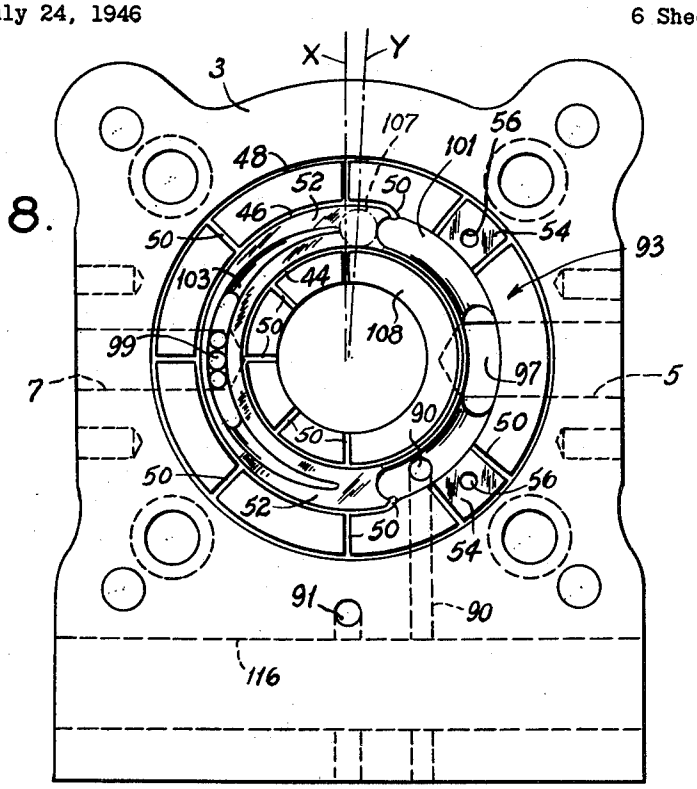
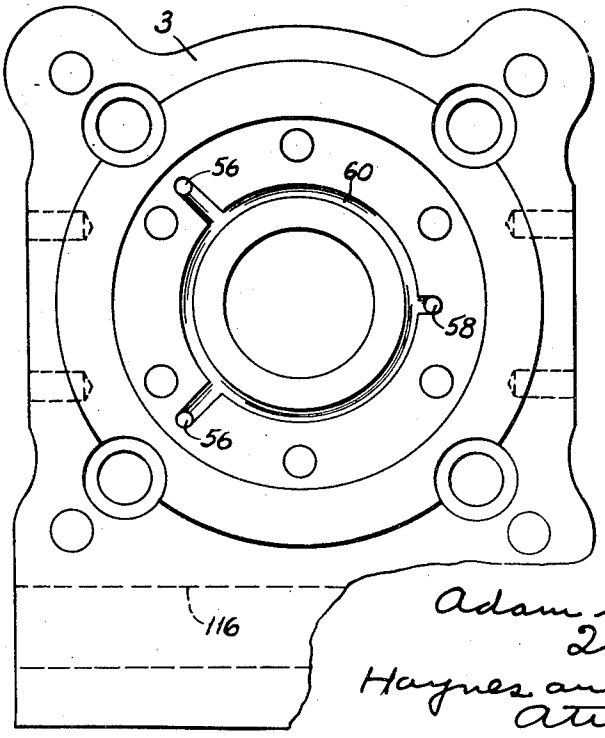


FIG. 9



Adam Gabriel,
Inventor.
Haynes and Koenig,
Attorneys.

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UNITED STATES PATENT OFFICE

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Adam Gabriel, River Forest, Ill., assignor, by mesne assignments, to Acme Industrial Hydraulics, Inc., Chicago, Ill., a corporation of Illinois

Application July 24, 1946, Serial No. 685,861

7 Claims. (Cl. 103—162)

1

This invention relates to pumps, and with regard to certain more specific features, to positive-displacement hydraulic pumps.

The invention is an improvement upon that described in my United States patent application Serial No. 587,151, filed April 7, 1945, for Pump, which matured into Patent No. 2,430,764 on November 11, 1947.

Among the several objects of the invention may be noted the provision of means for obtaining in a simplified structure an improved operation at the rotary sealing surfaces of the apparatus; the provision of a balanced construction in apparatus of the class described which reduces wear; the provision of apparatus of the class described having means for eliminating flutter in its controls and reduction of cavitation in the flow therethrough; and the provision in apparatus of this class of an improved pressure responsive means for control purposes. Other objects will be in part obvious and in part pointed out hereinafter.

The invention accordingly comprises the elements and combinations of elements, features of construction, and arrangements of parts which will be exemplified in the structures hereinafter described, and the scope of the application of which will be indicated in the following claims.

In the accompanying drawings, in which one of various possible embodiments of the invention is illustrated,

Fig. 1 is a top plan view of apparatus for carrying out the invention;

Fig. 2 is a front elevation, parts being shown in section;

Fig. 3 is a front elevation of a pump per se, being viewed from the left-hand side of Fig. 4;

Fig. 4 is an end elevation of the pump viewed from its right-hand end, as shown in Figs 2 and 3;

Fig. 5 is a vertical section taken on line 5—5 of Fig. 2, parts being broken away;

Fig. 6 is a vertical section taken on line 6—6 of Fig. 5;

Fig. 7 is a horizontal section taken on line 7—7 of Figs. 4 and 6;

Fig. 8 is a left-end view of a pump head, being viewed from line 8—8 of Fig. 7;

Fig. 9 is a right-end view of said pump head with a packing gland and shaft removed, the view being taken on line 9—9 of Fig. 7;

Fig. 10 is an enlarged detail section taken on line 10—10 of Fig. 1; and,

Fig. 11 is a fragmentary view similar to the lower broken-away part of Fig. 5 but showing an alternative position of valve parts.

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Similar reference characters indicate corresponding parts throughout the several views of the drawings.

Referring now more particularly to Figs. 1 and 2, letter B shows a box-like, liquid-tight base. On a horizontal part of the base is mounted a driving motor M. On the vertical wall thereof is mounted a pump P. The motor M and the pump P are operatively joined by means of a coupling C. The pump P is located in a sump tank or compartment T formed by walls of the base B. This tank T has a return line connection R from a hydraulic work circuit and a pressure outlet connection O to said work circuit. The pump P has a suction inlet pipe S leading from a filter F which is located in the sump tank T. The pump P has a pressure exit E which leads to a check valve V (see also Fig. 10). The check valve V is connected to the outlet O by means of a pipe Q. The check valve V has a connection K with a control port N of the pump P.

Coupled to the pipe Q is an accumulator A, which is in the form of a metal chamber containing a resilient (for example artificial rubber) bag D. This bag contains a gas under pressure. The accumulator A serves as a reservoir for liquid under pressure and is utilized when the pump is operating under conditions to be described. Attached also to the line Q is a gage G, the face of which is exposed to the outside of the sump T.

In general, when the motor M operates, the pump P operates to withdraw hydraulic fluid from the sump tank T through the filter F and suction line S. The pump P engenders pressure which is delivered to the exit E and thence through the lines Q and O, the gage G showing the pressure and the accumulator A tending to compensate surges. The valve V includes a spring-pressed check head H which closes whenever the pump P reduces its delivery to line Q. This closing cuts off line Q from the exit E. It also cuts off the control connection K from the exit E, but by reason of openings 64 in check head H does not cut off pipe Q from pipe K. The advantages of this check valve arrangement will be described below.

Referring now more particularly to Figs. 3–7, numeral 1 indicates a hollow cylindric case to which is bolted head 3. This head 3 has an inlet 5 and an outlet 7. The inlet 5 is connected to the suction line S (Fig. 1) and the outlet 7 is connected to the exit E. Centrally, the head 3 carries needle bearings 9 for a power shaft 11 which is driven by the motor M. A lubricant retainer is shown at 13, being held in place by a

retainer ring 2. The retainer ring 2 also holds in place a packing gasket 4, the purpose of which will be made clear. The bearing 9 is greased through the grease passage 15 which leads from a lubricant supply fitting 17 (Figs. 4 and 6).

At its other end the shaft 11 is supported in the inner race 6 of a thrust bearing 8. The outer race 10 of the thrust bearing 8 slides axially in an end bore 12 of the case 1. An outer flange 14 of a pressure plate 16 bears on the outer race 10. The plate 16 receives thrust from a piston 18 which slides in a cylinder 20, bored into an end cap 22 which is bolted to an extension 23 of the case 1. The end of the piston 18 is located in an opening 24 which is in communication with the outlet 7 via a passage 19 (see Figs. 3 and 4). Thus any pressure engendered by these pumps is applied axially to the piston 18 and hence axially to the shaft 11 through the thrust bearing 8.

Keyed to the shaft 11 is a rotary cylinder block 25. Exteriorly the block 25 is of cylindric shape, being spaced within the case 1, as indicated in Figs. 6 and 7. The normal rotation of the block 25 is as indicated by the arrows in Fig. 5. At spaced intervals around the axis of rotation of the block 25 are pump cylinders 29. These are located parallel to said axis and carry reciprocating pistons 31. Any suitable number of cylinders and pistons may be used, depending upon the diameter of the apparatus. In the present example there are nine of each of them. At its left end, each piston 31 (Fig. 7) has a reduced neck 33, beyond which is a spherical ball end or head 35. Under each ball end 35 is a circular washer forming a rocker seat 37 having a hollow internal spherical shape which forms a seat for a corresponding head 35. Since the piston 31 and heads 35 are preferably made of steel, the seats are made of some material forming a suitable bearing for the steel, such as bronze. The seats rest upon a rocker disc 39, having suitable openings 41 for freely clearing the necks 33. These openings 41 allow for relative rocking between the rocker disc and the pistons.

The rocker disc 39 has an inside boss 43 in which is an internal spherical surface 45 engaging an external spherical surface 47 of a rocker seat 49. Thus the rocker disc 39 may rock relatively to the rocker seat 49.

The rocker seat 49 is slidable on the shaft 11 and its right-hand end is made as a boss 51 which engages the end of a coil spring 53. The opposite end of this spring 53 acts upon a shoulder 55 of the shaft 11. This shoulder 55 abuts a ring 21 which slides in a recess 27 of the cylinder barrel 25. A spherical surface 26 of the ring abuts a corresponding spherical surface of a second ring 28 which rests against the bottom of said recess 27. Thus the shoulder 55, which is under thrust from the piston 18, exerts thrust to the right upon the cylinder barrel 25 through the rings 21 and 28. The shoulder 55 provides a reaction point for the spring which in turn presses the rocker seat 49 to the left. This results in the ball ends 35 being pressed toward the left.

All of the piston heads 35 are biased to engage a circular, rotary, stroke-varying race 59 which is carried upon a thrust bearing assembly 61, all annularly surrounding the shaft 11. Bearing assembly 61 is carried in an annular cradle 63 which has lateral coaxial gudgeons 65. The gudgeons 65 are supported in bearings 67, said bearings being supported within openings 69 of the case 1. The common axis of bearings 67 and gudgeons 65 is perpendicular to, and intersects the axis of the shaft 11. The outer races of the bearings 67 are retained by cover plates 71.

On its top the cradle 63 is provided with a lubricant fitting 73 (Fig. 6) which is accommodated in an opening 75 having an upper cover plate 76. This plate may be opened for lubrication of the fitting 73.

At its bottom the cradle 63 carries a ball control stud 77 (Fig. 6). This stud 77 is caught between a head 79 and a cup 83. Within the cup 83 is a spring 85 which reacts from an adjustable nut 87. Threaded through the nut 87 is a post 30, which may be held in adjusted position by a lock nut 32. Between the end of the post 30 and the cup 83, are annular perforated shims 34. The arrangement is such that the maximum leftward position of the ball stud 77, as illustrated in Fig. 6, is one in which the plane of the race 59 is perpendicular to the axis of the shaft 11. Openings 36 in the end of the cup 83 and in the shims 34 act as ports between the inside and the outside of the cup 83, the cup exerting some damping action as it moves.

The piston 81 slides in a fixed bushing 82 and responds to liquid pressure in a cylinder 89 formed in the case 1. The head 79 is carried on the left end portion 38 of the piston 81, a shim 40 fixing its location. The head 79, being hollow, is provided with a port 42 which provides damping action.

The cylinder 89 communicates with a passage 91 in the head 3. This passage 91 communicates with the outlet port 7 (Fig. 7) by way of a valve construction shown in general at L (Figs. 5, 6 and 11). The valve L will be described below, but from the above it will be clear that admission of pressure to the cylinder 89 will result in the piston 81 moving to the left and moving ball stud 77 to the position as determined by stop post 30 wherein cradle 63 is vertical (Fig. 6). With the cradle 63 in this position, the stroke-varying race 59 is in such position that the pistons 31 do not move axially in cylinders 29 and the pump idles. When pressure in cylinder 89 is released, spring 85 biases stud 77 and piston 81 to the right (Fig. 6). This rocks cradle 63 and stroke-varying race 59 to an angular position wherein the pistons 31 have full strokes in their cylinders 29.

Fig. 8 shows the character of the inner face of the head 3 against which presses the end of the rotary cylinder 25. Fig. 9 shows the outer face of the head 3.

The inner face 93 of the head 3 includes an inlet port 97 which is in communication with the inlet 5 (Fig. 8). It also includes an outlet port 99 which is in communication with the outlet 7. On the face 93 is a semi-circular groove 101 which is in communication with the inlet port 97. At 103 is shown a crescent-shaped groove which is in communication with the outlet port 99. Grooves 101 and 103 place the cylinders 29 in communication with the inlet when the pistons 31 therein are axially retracting, and place them in communication with the outlet 7 when the pistons are axially advancing. The arrangement between the upper ends of the grooves 101 and 103 is such that when the end 107 of a cylinder 29 moves from the top of the inlet port 101 to the top of the outlet port 103 (Fig. 8) said cylinder end, as indicated by the dotted lines at 107 in this figure, will be in communication with the outlet groove 103 before the respective cylinder reaches dead center (compare the center lines X and Y). This has the effect of minimizing suction cavitation.

On the inner face 93 are circular grooves 44, 46 and 48. With these communicate radial grooves 50. These grooves 44, 46, 48 and 50 form a network of communication channels with the semicircular inlet port 101. They also serve to isolate around the crescent-shaped outlet port 103 a surface which has been lightly shaded and indexed 52. They also serve to isolate two surfaces 54, which have also been lightly shaded. These surfaces 54 are ported, as shown at 56, the ports 56 being in communication with the port 58 through a circular groove 60 on the outside of the head 3 (Fig. 9). The port 58 connects with the outlet port 7 and outlet groove 103, and thus communicates outlet pressure to the surfaces 54 via the groove 60 and ports 56. The purpose of thus isolating surfaces 52 and 54 is to prevent the pressure which tends to leak out from the crescent-shaped groove 103 and from the ports 56 from spreading any more than over the areas of the surfaces 52 and 54. This spread is indicated by the light surface shading in Fig. 8. In other words, the grooves around these surfaces are arranged to release pressure on all portions of surfaces 93 which are outside of the shaded surfaces. Hence, only a predetermined and desirable axial pressure is exerted to the left on the cylinder 25 (Fig. 7) to act as a counterbalance for the axial pressure to the right from the piston 18. This counterbalancing is not in the nature of an equality, since some preponderance of pressure is required from the piston 18 toward the right in order to seat the cylinder 25 on the surface 93. The purpose of the ports 56 is to gain application of direct axial pressure to the cylinder 25 on a side of the vertical axis X which is opposite the direct pressure applied from the crescent-shaped outlet 103. This approximately balances the moments of axial pressure from the head 3 toward the cylinder 25, there being a three-point, peripherally distributed application of the reactive pressure to the cylinder 25. This avoids canting and binding thereof. The grooves 44, 46, 48 and 50 prevent the undue spread of any leakage pressure from the ports 103 and 56. This prevents exertion of axial pressure over such an area that the cylinder 25 might be lifted from its seat on the surface 93.

It will also be noted that the innermost radial grooves 50 communicate with the bearings 9 at a ledge 108 of reduced elevation relative to surface 93. Thus any high-pressure leakage tendency into these bearings is communicated to the suction port connection 101. Also, the outermost radial groove 50 connects the space around the cylinder 25 with the suction passage 101. Hence there is no high-pressure leakage from the pump except to the suction line.

In order to reduce the reaction pressure from the cylinders 29, their ends are restricted as indicated at 107 (Fig. 7). Grooves 109 serve to prevent the piston ends from cutting shoulders at the ends of the piston strokes.

The valve L shown in Figs. 5, 6 and 11 will now be described. It consists in a cross bore 116 in the bottom of the head 3. This bore is enclosed by screw caps 57 and 62. Threaded into the cap 62 is the line K, which, as will be recalled, is in communication with the pressure lines Q and O, via the valve V. It should be noted in connection with valve V that the communication between line K and line Q is permanent because of cross passages 64 in the valve head H.

In the bore 116 is a fixed bushing 66 which is formed to receive a valve stem 68. The bushing 66 includes an inlet 70 which through a groove 72 is always in communication with the line K. From an open space 74 in the bushing 66 is a connection 78 with the passage 91 communicating with control cylinder 89. The valve stem 68 is formed with two grooves 80 and 84. When the valve stem is at the right (Fig. 5) the inlet passage 70 is cut off from the connection 78. When the valve stem is to the left (Fig. 11), these passages are in communication so that pressure may be delivered from the line K via groove 72, port 70, groove 84, space 74 and passages 78 and 91 to the control cylinder 89.

The bushing 66 also includes a relief port 86 which, through a groove 88, is in communication with a passage 90 which leads to the semicircular inlet port 101. Thus, when the valve stem 68 is at the right (Fig. 5), the control cylinder 89 is in communication with the inlet port 101 via passages 91, 78, 74, groove 80, port 86, groove 88 and passage 90. When the valve stem is to the left (Fig. 11), this connection is cut off.

At the left end of the valve stem is a head 94 which provides a reaction point for a spring 95, the latter pressing against said cap 57. By contact with the bushing 66 (Fig. 5) the head 94 also forms a limiting stop to right-hand movement of the valve stem.

The valve stem 68 has an extension 98 which fits with just sufficient clearance in a bushing 96 to provide for some passage of liquid between the extension 98 and bushing 96. This flows into and out of the variable space 96*a* between the bushing 96 and the right-hand end of stem 68, thus forming a dash-pot damping arrangement. On the extension 98 is carried a head 100 which is biased to the right by a spring 102 reacting from the bushing 66. This bias is equal to and in the same direction as that of spring 95. When the pressure in line K exceeds the bias pressure of springs 95 and 102, it forces the valve stem 68 to the left. As the piston portion 92 on the valve stem moves to the left, the volume of the chamber 96*a* is increased and liquid flows into this chamber between extension 98 and bushing 96. This has a dash-pot or damping effect on the valve stem 68. When the valve stem 68 is moved by the pressure to the position shown in Fig. 11, the pressure is delivered through groove 72, port 70, groove 84, space 74 and passages 78 and 91 to the control cylinder 89. This pressure forces piston 81 (Fig. 6) and stud 77 to the left to the limiting position determined by stop 30. When the stud is in this limiting position, the cradle 63 is in vertical or neutral position.

When the valve stem 68 is moved to the right by springs 95 and 102 upon decrease of pressure in line K, it assumes the Fig. 5 position. Thereupon, spring 85 (Fig. 6) forces stud 77 and piston 81 to the right and forces liquid through 91, 78, 74, 80, 88, 90 and into inlet port 101. When valve stem 68 moves from left to right or from right to left, liquid must either escape from or enter chamber 96*a*. It does so through the clearance between bushing 96 and extension 98. This clearance is kept at a minimum to prevent chattering or fluttering. This arrangement provides for a dash-pot or damping effect on the valve stem 68 in either direction of its movement.

Springs 95 and 102 are adjusted to such initial compression that the edge 96*c* of the intermediate enlarged portion of the valve stem 68 will enter chamber 74 when the required maximum pressure is reached.

This valve-controlled application of pressure to the control cylinder 89 is an improvement over the direct application of pressure to the corresponding control cylinder in my said application since it provides for unloading of the pump when used in conjunction with the accumulator A. The control cylinder 89, due to the action of valve K, is either under full pressure overcoming the entire bias of spring 85 or under only the pressure of the suction line. When it is under the latter pressure, full pumping occurs. When it is under full pressure, no pumping occurs, the pump being unloaded. The valve stem 68 and its piston and extension portions 92 and 98 are preferably of integral construction. Spring 95 and 102 are adjusted to exert equal bias on the valve stem. Two springs are used to make the spring action symmetrical.

It will be noted from Fig. 5 that the ring 2 is held to the head 3 by means of bolts 104. The head 3 is held to the body 1 by means of bolts 105. The entire pump is supported in the base B (Figs. 1 and 2) by bolts 106.

Operation is as follows:

Assume that the shaft 11 is power-operated by the motor M in the direction shown by the arrows in the various figures. Since at the start of rotation there is presumably no pressure in the outlet 7, there will be no pressure in the control cylinder 89. The spring 85 then angles the stud 77 to the right of the position shown in Fig. 6. This angles the cradle 63 and the stroke-varying race 59 (see Figs. 5 and 7). Then as the cylinder block 25 rotates, the heads 35 of the pistons will move in an angular plane, moving to the right under compression as they traverse the outlet groove 103; and moving to the left with suction as they traverse the inlet groove 101. Thus compression and suction strokes are performed. It will be noted from Fig. 8 that each connection with the outlet port 103 is made ahead of dead center which prevents suction pressure from reaching a cavitating value. The above transfers fluid from the inlet 5 to the outlet 7 and builds up pressure in the latter and supplies the volume required by the consuming appliance attached to the outlet line O.

As pressure builds up in outlet 7, it also builds up in the accumulator A and in line K. Pressure in line K is delivered to the bore 116 (Figs. 5 and 11). When this pressure exceeds the bias of springs 95 and 102, valve stem 68 moves to the left from the Fig. 5 position. When the required maximum pressure is reached, the edge 96c enters chamber 74. Liquid then flows into control cylinder 89, in the manner previously described. The pressure of the liquid in cylinder 89 forces piston 81 and stud 77 to the left to the position shown in Fig. 6, wherein cradle 63 is in neutral position and no liquid is pumped. In this position, the stud 77 has moved cup 83 to the left limit determined by stop 30 and spring 85 is compressed. When the pressure in line K decreases, springs 95 and 102 bias valve stem 68 to the right to the Fig. 5 position. The control cylinder 89 is then vented to port 101 in the manner previously described and spring 85 (Fig. 6) biases stud 77 and piston 81 to the right. This moves cradle 63 to an angular position wherein the pump operates to pump liquid at full capacity.

With this arrangement, there is no intermediate pumping stage. The pump is either operating at full capacity or is unloaded. By "unloaded" is meant that the pump is idling and consuming no more power than is necessary to overcome mechanical friction of rotation. The accumulator A is provided to store liquid under pressure to overcome the initial compression of springs 95 and 102. If the accumulator were omitted, the pressure in line K and on valve stem 68 would build up immediately upon starting of pumping and would force valve stem 68 to the left. This would admit pressure to control cylinder 89 and move cradle 63 to neutral position. As soon as cradle 63 moved to neutral position, no liquid would be pumped and the pressure in line K would rapidly drop to zero, whereupon the cradle 63 would be biased to the right by spring 85 and pumping would be resumed. The result would be too much on-and-off motion of the cradle. With the accumulator A, however, when the cradle moves to neutral position the pressure in line K drops only gradually, due to the volume of liquid stored in the accumulator under pressure. The accumulator functions as a reservoir for maintaining a substantial volume of liquid under pressure and a certain amount of this liquid is supplied through line K to chamber 116 to maintain valve stem 68 in the position wherein it admits pressure to control cylinder 89. This pressure holds cradle 63 in neutral position until the pressure in line K drops to such value that valve stem 68 moves to the right, vents cylinder 89, and permits cradle 63 to rock to position to start pumping to build up the reserve that was removed from the accumulator.

As stated previously, springs 95 and 102 are adjusted to exert equal bias on valve stem 68. If they are set, for example, so that 900 lbs. pressure is required to overcome their combined bias, any pressure above 900 lbs. in line K wil force valve stem 68 to the left from its Fig. 5 position. When the pressure reaches 1000 lbs., for example, the edge 96c enters chamber 74 and the full 1000 lbs. pressure is delivered to control cylinder 89. This full pressure overcomes the bias of spring 85 (Fig. 6) and forces stud 77 to its left limit as determined by stop 30, thus unloading the pump. Spring 85 has an initial compression sufficient to force liquid out of cylinder 89 and rock cradle 63 into pumping position. Thus, the control is such that the pump operates either at full capacity or idles, and constant on-and-off operation is eliminated.

Advantages of the construction, particularly as related to the construction shown in my said application above mentioned, are the use of outlet pressure for applying an axial thrust to the shaft 11 for seating the end of the cylinder 25 on the ported surface 93 of the head 3.

Another advantage is that any leakage which may occur between the end of the cylinder 25 and the face 93 is not allowed to extend in its effect until the total pressure engendered thereby may lift the cylinder 25 from the surface 93. This is because the effect of leakage under pressure is limited to the shaded surfaces shown in Fig. 8. Besides, the reactive pressure on the cylinder 25 due to outlet pressure is distributed peripherally so as to prevent canting action on the cylinder.

An additional advantage of the invention is the use of the control valve L for applying outlet pressure to the control cylinder 89, which upon release allows said control cylinder to bleed to the inlet suction port 101, instead of to the outlet pressure port as heretofore. An additional advantage in connection with said valve L is the dash-pot damping arrangement which avoids

flutter of the valve and which hence avoids fluttering control.

Another advantage is in the arrangement in the hydraulic circuit of the relief check valve of Fig. 10. This arrangement of this check valve prevents back flow of pressure from the pressure line to the pump when the pump reduces its flow. At the same time, it communicates the back pressure to the line K and hence to the valve L and the control cylinder 89, so that said back pressure may be used for control purposes, thus saving many pump starts such as occur when a line such as K is connected permanently to line E, instead of permanently to line Q. That is to say, line K loses pressure less quickly when connected to the large volume in lines Q and O than when connected to the relatively small volume in line E (assuming valve V to be shut).

The construction of the valve V shown in Fig. 10 has certain advantages. These accrue due to having the pipe K connected to the pipe Q through the openings 64 in the valve head H. By having the connection thus arranged, there is eliminated the necessity for coupling the line K into the line Q by special connections. Furthermore, in all modifications of the apparatus there is absolute certainty that line K will not inadvertently be coupled into the line E, which ordinarily would appear to be as good a connection as to the line Q but which is not, as explained above.

Another advantage of the invention is the simplicity of the one-piece head 3, involving no floating sealing disc or plate.

In view of the above, it will be seen that the several objects of the invention are achieved and other advantageous results attained.

As many changes could be made in the above constructions without departing from the scope of the invention, it is intended that all matter contained in the above description or shown in the accompanying drawings shall be interpreted as illustrative and not in a limiting sense.

I claim:

1. A pump comprising a casing having a head in which are an axially presented inlet and an axially presented outlet, a rotary block in the casing having axially disposed cylinders communicating successively with the inlet and the outlet, hydraulic means biasing said block axially in the casing for contact and against said head, said hydraulic means having a pressure communication with said outlet, and grooves in the contact surface of the head surrounding the outlet and communicating with the inlet to limit leakage pressure which is exerted upon the block against said bias.

2. A pump comprising a cylindric casing having a head at one end in which are interiorly axially presented on opposite sides of its center line an inlet and an outlet, a rotary block in the casing having axially disposed cylinders communicating successively with the inlet and the outlet, hydraulic piston means at the end of said casing opposite said head and biasing said block axially in the casing for sealing contact against said head, said hydraulic piston means having a pressure communication with said outlet, and means communicating between the outlet side of the head and the opposite side for applying pressure to the block in a direction against the bias from said hydraulic piston means.

3. A pump comprising a cylindric casing having a head at one end in which are interiorly axially presented on opposite sides of its center line an inlet and an outlet, a rotary block in the casing having axially disposed cylinders communicating successively with the inlet and the outlet, hydraulic piston means at the end of said casing opposite said head and biasing said block axially in the casing for sealing contact against said head, said hydraulic piston means having a pressure communication with said outlet, means communicating between the outlet side of the head and the opposite side for applying pressure to the block in a direction against the bias from said hydraulic piston means, and grooved means extending from the inlet and so related to all of the pressure connections on the face of the head as to limit additional leakage pressure tending to bias the block from the head.

4. A pump comprising a casing having a head at one end upon the inner surface of which are an inlet and an outlet axially directed, a rotary block within the casing having axially disposed cylinders which are open ended toward the inner surface of said head and communicating successively with the inlet and the outlet, a shaft rotary with and carrying said block, hydraulic means in the casing opposite said head adapted to bias said shaft and the supported block axially toward said inlet and said outlet, a pressure connection between said hydraulic means and said outlet, and means for communicating pressure from said outlet to limited areas of said inner face of the head for application of pressure to certain contacting surfaces between the block and the head.

5. A pump comprising a casing having a head at one end upon the inner surface of which are an inlet and an outlet axially directed, a rotary block within the casing having axially disposed cylinders which are open ended toward the inner surface of said head and communicating successively with the inlet and the outlet, a shaft rotary with and carrying said block, hydraulic means in the casing opposite said head adapted to bias said shaft and the supported block axially toward said inlet and said outlet, a pressure connection between said hydraulic means and said outlet; and means for communicating pressure from said outlet to limited areas of said inner face of the head for application of pressure to certain contacting surfaces between the block and the head, said surfaces being distributed peripherally so as to minimize canting forces on the block.

6. In a rotary pump of the class described, a head having an axially presented sealing surface in which are partially circular inlet and outlet ports on opposite sides of a reference line, at least one pressure opening presented on said surface on the inlet side of said line and communicating with the pressure outlet, and groove means located around said pressure outlet and said pressure openings adapted to limit the area to which leaakge pressure may distribute itself over said surface, said groove means communicating with said inlet.

7. A pump comprising a hollow casing including a head having an inwardly directed sealing surface in which is a suction inlet and a pressure outlet, said head having a central opening, a shaft passing through said opening, a rotary block on the shaft located in the casing and having axially disposed cylinders communicating successively with the inlet and the outlet, said cylinders opening from an end of the block which has a face contacting the inner surface of said head, means in the opposite end of the casing

biasing the block into engagement with said head, ports in said surface of the head communicating with said outlet, grooved means in said surface of the head surrounding said outlet and said ports and communicating with the inlet, whereby only limited areas between the rotary block and the inner surface of said head are subject to pressure due to leakage, the positions of said outlet and said ports being such as to minimize canting of the block in the casing.

ADAM GABRIEL.

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