Sept. 11, 1962 J. T. GONDEK 3,053,186

TWO-STAGE HYDRAULIC PUMPS

Filed June 24, 1959 3 Sheets-Sheet 1

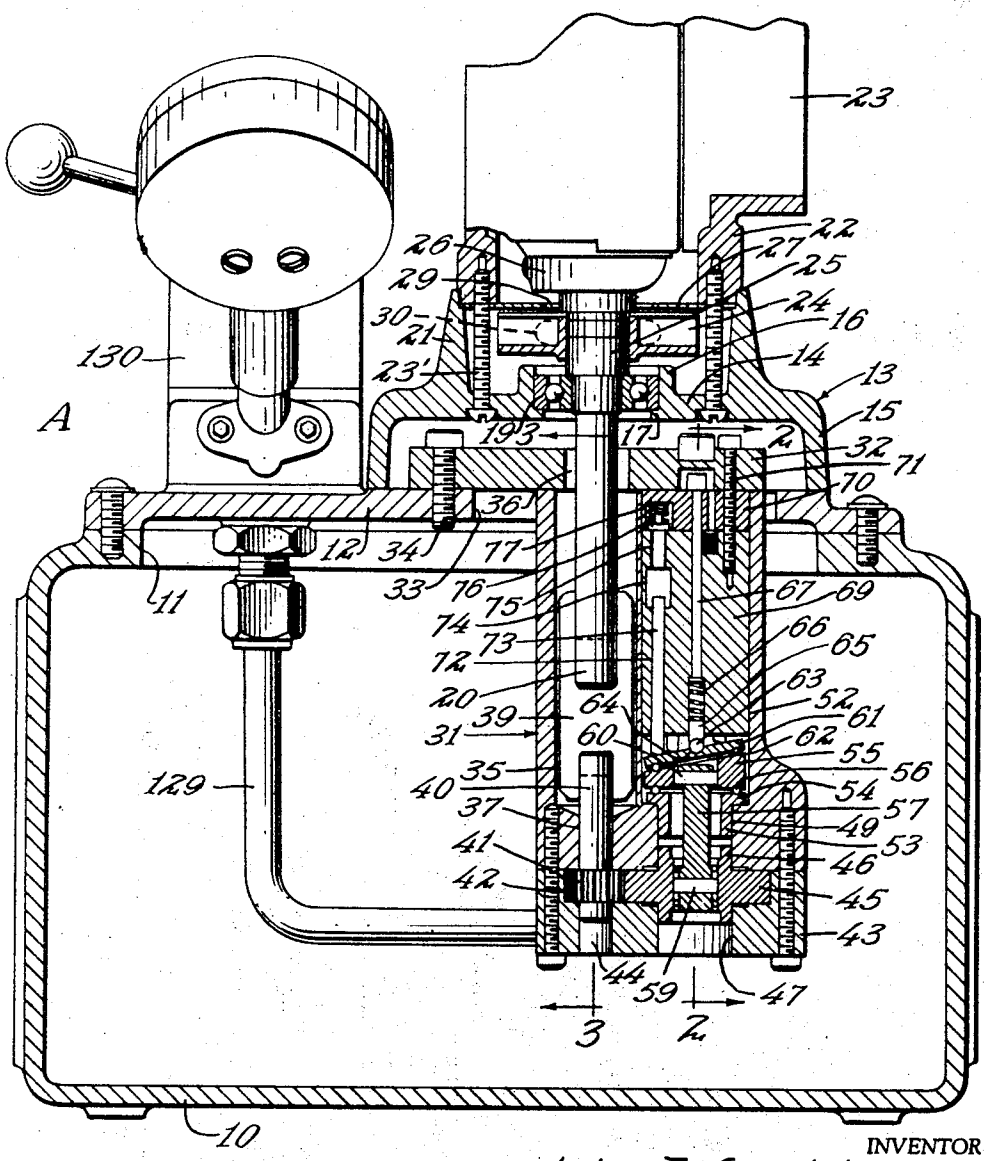


FIG. 1

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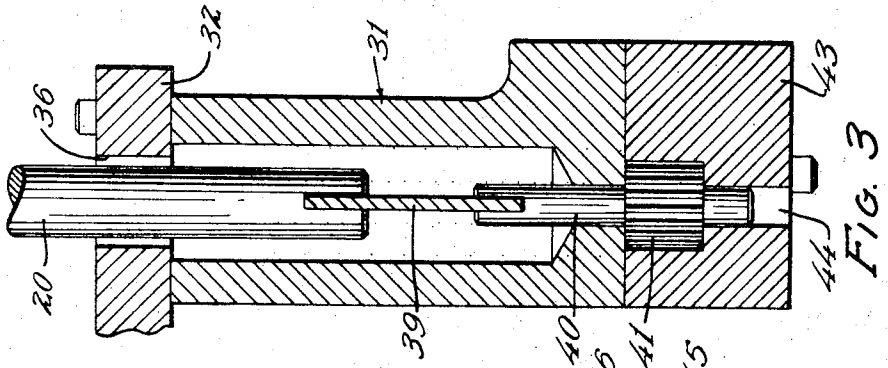


FIG. 3

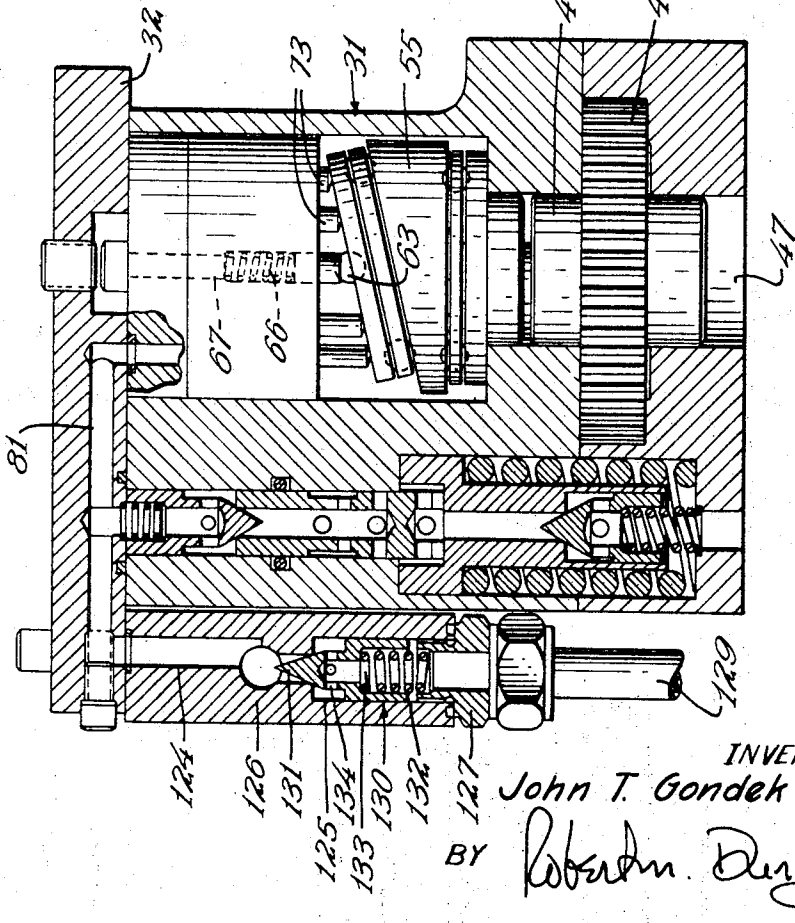


FIG. 2

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TWO-STAGE HYDRAULIC PUMPS

Filed June 24, 1959 3 Sheets-Sheet 3

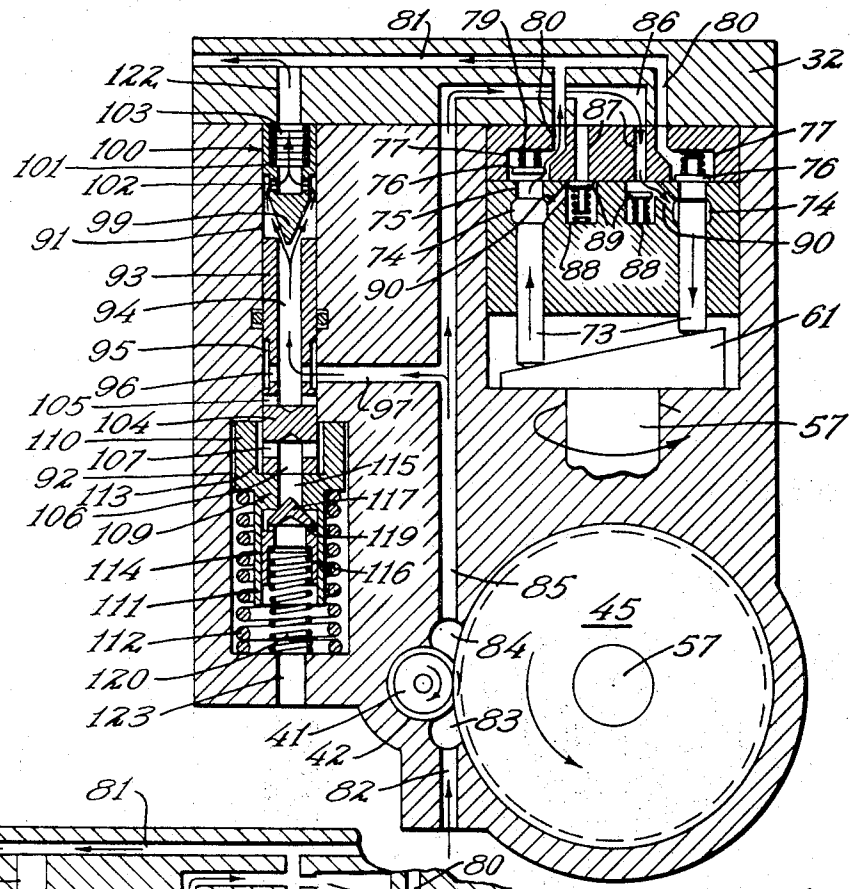


FIG. 4

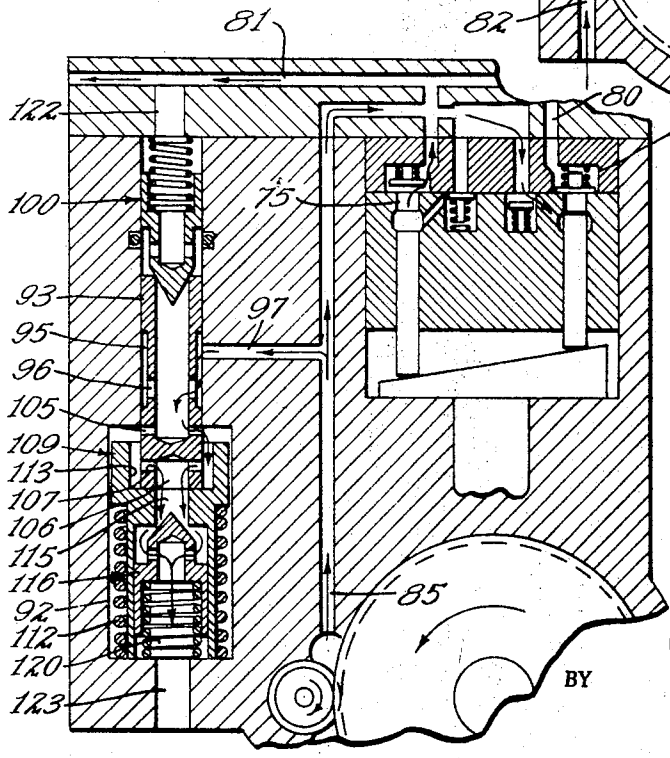


FIG. 5

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TWO-STAGE HYDRAULIC PUMPS

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6 Claims. (Cl. 103—5)

This invention relates to an improvement in two stage hydraulic pumps designed to provide a source of supply of fluid under pressure.

An object of the present invention lies in the provision of a two stage hydraulic pump capable of delivering a relatively large volume of fluid at a relatively low pressure or relatively small volume of fluid at a high pressure. As the pump slows in speed as the requirement for added pressure increases, the volume of hydraulic fluid delivered automatically decreases in a manner to compensate for varying conditions and varying pressure requirements.

A feature of the present invention resides in the provision of a pump which is preferably driven by a high speed motor of the series wound type which slows down as the load increases to about one-half its normal rate of speed. The motor drives one of a pair of meshing gears to cause rotation of the gears in unison. The meshing gears provide a gear pump capable of delivering a relatively large volume of hydraulic fluid at a relatively low pressure. As an example, the gear pump driven by a motor operating at 12,000 revolutions per minute a volume of 800 cubic inches of fluid per minute may be pumped by the gear pump at zero pressure. When the motor speed is cut in two, to rotate at 6,000 revolutions per minute, the gear pump may deliver 400 cubic inches per minute at a pressure of 600 pounds per square inch.

A feature of the present invention resides in the provision of a piston pump connected to the discharge of the gear pump. This piston pump is preferably an axial piston type capable of delivering fluid under high pressure, such as, for example, ten thousand pounds per square inch. Accordingly, the intake of the piston pump is supercharged by the gear pump so that the hydraulic fluid enters the piston pump under pressure. This arrangement has several advantages. By placing the intake of this pump under pressure, smaller passages may be used to convey the hydraulic fluid and smaller spring loaded intake check valves may be employed. Furthermore, during the portion of the piston stroke when the piston is drawing hydraulic fluid into the cylinder, the force of the hydraulic fluid acts against the pistons to maintain them in contact with the wabble plate. This eliminates the requirement for springs or other resilient means for retracting the pistons in their cylinders, thus greatly simplifying the pump construction and greatly lowering the cost thereof.

A further feature of the present invention resides in the provision of a pump incorporating a valve which may permit the by-passage of hydraulic fluid not required to supply the piston pump. As the pressure of the oil in the pump discharge increases, the speed of the operating motor is reduced and the volume of hydraulic fluid pumped decreases. Means are provided to by-pass the excess fluid from the gear pump back to the supply reservoir so that the excess of fluid being pumped by the gear pump may escape and return to the reservoir. The gear pump supplies fluid to the piston pump until a predetermined pressure is developed. During that time the speed of the motor reduces. When the valve changes position it permits the excess fluid from the gear pump not needed to supercharge the piston pump to return to the reservoir at low pressure, providing just sufficient pressure to perform its supercharging function. This permits the speed of the motor to again increase until

sufficient back pressure is built up to again reduce the motor speed.

These and other objects and novel features of the present invention will be more clearly and fully set forth in the following specification and claims.

In the drawings forming a part of the specification:

FIGURE 1 is a sectional view through the pump showing in general the arrangement of parts therein.

FIGURE 2 is a sectional view through the piston pump chamber, the position of the section being indicated by the line 2—2 of FIGURE 1.

FIGURE 3 is a sectional view through a portion of the pump, the position of the section being indicated by the line 3—3 of FIGURE 1.

FIGURE 4 is a diagrammatic view showing the manner in which the pump operates.

FIGURE 5 is a view similar to FIGURE 4, showing certain of the elements of the pump in a different relationship.

With reference first to FIGURE 1 of the drawings, it will be noted that the pump A includes a hollow reservoir **10** having an opened upper end **11** which is normally closed by a closure plate **12**. A motor supporting shroud **13** is secured to, and supported by, the closure plate **12** by bolts or similar means not illustrated in the drawings. The shroud **13** includes a partition wall **14** intermediate its upper and lower ends and includes a downturned peripheral flange **15** which may be provided with outwardly projecting mounting ears through which the shroud anchoring bolts may extend. The partition **14** includes an axial sleeve **16** having an inwardly projecting flange **17** at its lower end. The sleeve **16** supports a bearing **19** which in turn supports a motor shaft extension **20**.

The shroud **13** also includes an upwardly extending hollow flange **21** which supports the hollow cylindrical end **22** of the motor housing **23**. The motor housing is held in place upon the shroud **13** by bolts **23'** which extend upwardly through the partition wall **14** and are threaded into suitable threaded sockets in the end of the housing sleeve **22**.

A fan or blower **24** is mounted upon the motor shaft **25** below the lower end of the motor **26** and above the bearing **19**. A baffle **27** is interposed between the upper end of the shroud **21** and the lower end of the motor housing **23**, this baffle having an axial opening **29** extending therethrough through which air may be drawn. The shroud **13** is provided with one or more air outlet openings **30** through which the air may be forced. The air is drawn downwardly through the motor housing to maintain the motor at the proper temperature.

A pump body **31** is bolted or otherwise secured to the undersurface of a supporting plate **32** which overlies an opening **33** in the closure plate **12** and is secured to the closure plate by bolts such as **34**. The pump body **31** is provided with a vertical bore **35** which is aligned with an aperture **36** in the supporting plate **32**. The lower end of the bore **35** terminates in a relatively small diameter bearing portion **37** which is coaxial with the bore **35**. The lower end of the motor shaft extension extends downwardly into the bore **35** and is connected with a flexible coupling **39** to the upper end of a shaft **40** extending downwardly through the bearing **37**. The shaft **37** supports a drive gear **41** which is mounted in a cavity **42** in the upper surface of the pump end plate **43**. The end plate **43** is provided with an aperture **44** in alignment with the aperture **37** and into which the lower end of the shaft **40** extends for support.

The cavity **42** is also designed to accommodate the driven gear **45** which is provided with a hollow cylindrical hub **46** projecting above and below the gear. The

downwardly projecting end of the hub 46 is supported in an aperture 47 in the pump end plate 43. The upper end of the hub is rotatably supported within a coaxial bore 49 in the pump body 31. The bore 49 is in alignment with, and communicates with, the lower end of a larger diameter bore 52 extending to the upper end of the pump body 31.

A hollow cylindrical bearing support 53 is supported within the upper end of the bore 49 and includes a peripheral flange 54 at its upper end which overlies the shoulder between the bores 49 and 52. An angle plate 55 is rotatably supported upon a bearing 56 resting upon the flange 54. The angle plate 55 is connected for rotation in unison with the driven gear 45 by a drive connector 57 supported upon drive pins 59 and 60 mounted within the gear hub 46 and angle plate 55.

A swash plate 61 overlies the inclined upper surface of the angle plate in parallel relation thereto. A bearing 62 rides in opposed grooves on the upper surface of the angle plate and the undersurface of the swash plate so that one plate may rotate to some extent relative to the other. The swash plate is held against the lower bearing by a bearing ball 63 which is engaged in a recess 64 in the upper surface of the swash plate and which is held in position by a thrust pin 65 having a concave lower surface and a spring 66 which engages the thrust pin. The spring is held in place by a bearing screw 67 which extends through the center of the pump barrel 69. The pump barrel is mounted within the large diameter bore 52 above the swash plate 61 and a valve head 70 is also located in the bore 52 above the pump barrel. The supporting plate 32, the valve head 70 and the pump barrel 69 are held in place by suitable attaching screws 71.

The pump barrel 69 is provided with a series of angularly spaced cylinders 72 which are at equal radius from the axis of rotation. Pump pistons 73 are slidably supported in the cylinders 72 and the lower ends of the pistons are designed to bear against the upper surface of the swash plate 61. Fluid pressure chambers 74 are provided in the pump barrel 69 into which the upper ends of the pistons 73 may extend. Discharge passages 75 communicate with each pressure chamber 74 and a check valve 76 is mounted in a suitable socket 77 in the undersurface of the valve head 70, the check valve 76 being normally urged into a position to close the discharge passage by a spring 79.

In view of the fact that the various intake and outlet ports cannot readily be followed in actual sectional views of the pump, FIGURES 4 and 5 have been added which are diagrammatic views showing the relative position of the parts and showing the manner in which the hydraulic fluid may flow. As indicated in these figures, the sockets or chambers 77 communicate with a discharge connection 80 which leads to a common discharge passage 81. This view also shows the gears 41 and 45 forming the low pressure gear pump at right angles to their normal position so that the flow of hydraulic fluid through the system can be understood.

An intake passage 82 leads from the reservoir 10 to the intake 83 of the gear pump chamber 42. Fluid is pumped from the intake chamber 83 to the discharge chamber 84 which is connected by a passage 85 to the intake chamber 86 of the piston pump. This intake chamber 86 is form as a part of the supporting plate 32. The intake chamber 86 is connected by a passage 87 to a check valve chamber 88 including a check valve 89. Passages 90 connect to the check valve chambers 88 with the pressure chambers 74 at the upper end of the individual pistons 73. Thus it will be seen that the discharge from the gear pump 41, 45 is transmitted to the intake chamber of the piston pump and the pressure of the hydraulic fluid acts to open the check valve 89 during the intake stroke of each piston 73. This is of importance as the fluid under pressure exerts a force against the ends of the piston 73 forcing these pistons into contact with the swash plate which is diagrammatically illustrated at 61. In FIGURE 4 of the drawings, the left hand piston 73 has just completed its intake stroke and is just starting its pressure or power stroke. In this position, the upward movement of the piston 73 develops a greater pressure within the pressure chamber 74 than in inlet chamber 83 thus closing the check valves 89 and opening the discharge check valve 76, permitting fluid under higher pressure to pass through the passages 80 and 81 to the pump discharge.

Also in FIGURE 4, it will be noted that the right hand piston 73 has just completed its pressure stroke and is starting its intake stroke. The high pressure in the passage 80 acts to close the discharge check valve 76 and to open the intake check valve 89, the pressure within the intake chamber 86 exceeding that in the piston cylinder or pressure chamber 74.

The pump also includes valve means which controls the flow of hydraulic fluid from the gear pump. The pump housing includes a vertical bore which serves as a cylinder 91 which communicates with a relatively large diameter cylinder 92 at its lower end. A valve spool 93 is slidably supported in the cylinder 91 and includes an axial passage 94 through which the oil may flow. A peripheral groove 95 in the valve spool 93 is connected by apertures 96 to the interior of the spool. The peripheral groove 95 may register with a passage 97 communicating with the passage 85 leading from the discharge 84 of the gear pump 41, 45. The upper end of the valve passage 94 is open and is normally closed by the pointed end 99 of the check valve indicated in general by the numeral 100. The check valve 100 is provided by a hollow cylindrical end 101 which is slidably supported in the cylinder 93. Passages 102 form a communication between the hollow interior of the valve 100 and the area of the cylinder 93 about the pointed end 99 of the check valve. A spring 103 interposed between the interior of the hollow check valve and the mounting plate 32 normally urges the check valve 100 into closed position against the upper end of the valve 93.

The lower end of the valve spool 93 is closed as indicated at 104 and transverse passages 105 extend through the walls of the valve spool at the lower end of the hollow portion of the spool. The lower end of the valve spool includes an axial passage 106 which communicates with transverse passages 107 leading through the wall of the closed lower end of the valve spool.

A piston 109 is slidably supported in the large diameter bore 92. The piston 109 has a large diameter upper end 110 which is loosely supported in the bore 92 and includes a smaller diameter portion 111 which projects downwardly. A spring 112 is interposed between the bottom of the cylinder 92 and the shoulder formed in the piston by the portions of different diameter. The spring 112 urges the large diameter end of the piston against the upper end of the cylinder 92. An axial socket 113 in the upper end of the piston 109 is designed to accommodate the lower extremity of the valve spool 93. The lower end 111 of the piston 109 is also provided with an axial socket 114. A passage 115 connects the sockets 113 and 114.

A check valve 116 is slidably supported within the socket 114. The check valve 116 is in the form of a hollow piston having a pointed upper end 117 which is engageable with the wall of the passage 115 through the piston 109. Passages 119 extend through the wall of the piston adjoining the pointed end 117 so that fluid may flow from the passage 115 to the apertures 119 and the interior of the check valve when the check valve is in open position. The check valve 116 is normally urged to closed position closing the lower end of the piston passage 115 by a spring 120.

In its normal position of operation, the lower end of the valve spool 93 is held against the base of the socket 113 in the upper end 110 of the piston 109 by the spring

103 which urges the check valve 100 against the upper end of the valve spool. As the pump starts into operation, the discharge from the gear pump flows through the passage 85 to the intake chamber 86 of the piston pump. However, as the capacity of the gear pump 41, 45 is substantially greater than the piston pumps will accommodate, a portion of the fluid is bypassed through the passage 97, the groove 95, the opening 96, and the interior of the valve spool 93, acting against the pointed lower end 99 of the check valve 100 to open this valve. The fluid can flow about the pointed end of the check valve, through the openings 102 and through the hollow interior of the check valve. A passage 122 through the supporting block 32 connects with the discharge passage 81 and permits the discharge of the gear pump which is not used to supercharge the piston pump to be used. If the relatively large volume flow of fluid is not greatly impeded, the discharge from both of the pumps passes through the passage 81. However, when higher pressures are required in the discharge, a different action takes place and the valve mechanism moves from the position illustrated in FIGURE 4 to the position shown in FIGURE 5.

When the pressure in the discharge conduit 81 and passage 122 reaches a predetermined amount, as for example 600 pounds per square inch, the pressure against the top of the relief valve 100 forces the valve spool 93 downwardly compressing the spring 112. The check valve 100 is held in closed position by the pump discharge pressure. Fluid from the gear pump 41, 45 may now flow from the discharge connection 85 and passage 97 to the groove 95 encircling the valve spool and through the apertures 96 to the hollow interior of the valve. Downward movement of the valve spool 93 moves the portion of the lower end of the piston having the radial passages 105 into the larger cylinder 92, allowing the fluid to flow into the upper end of the cylinder 92. This fluid may flow into the socket 113 encircling the lower end of the valve spool and through the radial passages 107 to the axial passage 106. This axial passage 106 communicates with the axial passage 115 through the piston 109. The fluid under pressure acts against the upper end of the check valve 116 and acts to compress the spring 120. The spring 120 urges the check valve upwardly at such a tension that when a force of perhaps 160 pounds per square inch is experienced, the check valve will open. In the particular arrangement illustrated, this is just about enough to feed the high pressure pump without cavitation and to assure that the pistons follow the cam plate. A passage 123 is provided at the bottom of the cylinder 92 leading into the reservoir 10 so that the fluid not required to feed the piston pump may bypass to the reservoir.

With a reference now to FIGURE 2 of the drawings, it will be noted that the discharge pressure conduit 81 communicates with a vertical passage 124 which leads to a check valve chamber 125 in a member 126 adjacent the pump housing. A fitting 127 extends into the lower end of the chamber 125 to connect this chamber to the outlet connection 129.

A check valve 130 is slidably supported in the chamber 125 and includes a pointed end 131 which seats against the end of the passage 124. The opposite end of the check valve is provided with an axial socket 132 which contains a spring 133 designed to normally urge the check valve into closed position. Passages 134 extend between the exterior and interior of the check valve near the pointed end thereof. Accordingly, when sufficient pressure is exerted against the check valve to compress the spring 133, fluid may flow around the pointed end of the check valve, through the openings or passages 134 to the hollow interior of the check valve, and through the fitting 127 into the outlet pipe 129.

A suitable pressure relief valve may be provided which permits the escape of fluid under pressure when the pressure in the system exceeds a desired amount. This pressure relief valve is indicated in general by the numeral 130 and the detail of construction is not illustrated as it is unimportant to the invention.

The operation of the pump has been described in conjunction with the construction of the apparatus. The operation may best be noted from an observation of the diagrammatic views 4 and 5. Hydraulic fluid is drawn through the passage 82 into the gear pump 41, 45 and discharged through the passage 85 to the intake chamber 86 of the piston pump. While the pump is operating at low pressure, the excess of fluid pumped by the gear pump is directed through the passage 97 to the interior of the valve spool 93. The fluid acts to open the check valve 100 and flow through the discharge passage 81.

The pressure of the fluid from the gear pump permits the use of smaller spring loaded intake valves in smaller oil passages. Furthermore, this oil acts against the pistons of the piston pump to force the pistons against the swash plate 61, eliminating the requirement for springs. When the pressure in the discharge reaches a certain amount, such as 600 pounds per square inch as an example, the pressure in the discharge acts to close the check valve 100 and to force the spool valve 93 downwardly. This action causes the piston 109 to move downwardly and compress the spring 112. As a result, the excess oil from the gear pump can no longer flow directly into the discharge line.

When this action takes place, the excess fluid from the gear pump flows through the spool valve 93 and into the upper end of the larger cylinder 92. The pressure then opens the check valve 116 and returns to the reservoir 10.

Due to the fact that the driving motor may reduce in speed and provide more power when increased power is required at the discharge, both the gear pump and piston pump will be simultaneously reduced in speed. This action causes the corresponding reduction in the fluid pumped both by the gear pump and by the piston pump but the outlet from the gear pump is always in excess of the demands of the piston pump and thus always acts to supercharge the inlet of this second stage pump.

In accordance with the patent statutes, I have described the principles of construction and operation of my improvement in two stage hydraulic pumps, and while I have endeavored to set forth the best embodiment thereof, I desire to have it understood that changes may be made within the scope of the following claims without departing from the spirit of my invention.

I claim:

1. A hydraulic pump including a relatively high volume, relatively low pressure first stage pump, a relatively lower volume, relatively higher pressure second stage pump, means for driving said pumps, each of said pumps having an intake and a discharge, a direct, continuously open means connecting the discharge of the first stage pump to the intake of the second stage pump, a reservoir connected to the inlet of said first stage pump, a valve cylinder connected to the discharge of said first stage pump, a passage from one end of said cylinder to said discharge of said second stage pump, a passage connecting the other end of said cylinder to said reservoir, a valve piston slidable in said valve cylinder having a first hollow bore communicating with one end of said valve piston and with discharge pressure from said first stage pump, a first pressure regulator valve slidable in said cylinder independently of said piston and adjacent said one end of said piston, means urging said first pressure regulator valve against said piston to close said bore and operable upon a predetermined discharge pressure from said first stage pump to open said first named passage, said first pressure regulator valve being subject to second stage discharge pressure through said first named passage, said valve piston including a second hollow bore in its other end nor-

mally closed from communication with said first named bore, a by-pass in said cylinder and valve piston closed from communication between said bores in one position of said valve piston in said cylinder, and in communication between said bores in another extreme position, means normally urging said valve piston toward said one position and toward said first pressure regulator valve, second stage discharge pressure acting to close said first pressure regulator valve and to move said first pressure regulator valve and said piston to said other position when a predetermined pressure is attained, and a second pressure regulator closing said second hollow bore and operable upon a predetermined first stage discharge pressure to open said second passage.

2. The structure of claim 1 and in which said valve piston includes an axial sleeve projecting beyond said second bore and slidably supporting said second relief valve.

3. The structure of claim 1 and in which said pressure regulator valves include members seating against the ends of said bores.

4. The structure of claim 1 and in which said valve piston includes a small diameter end and a separate large diameter end, and in which said valve cylinder includes portions of different diameter to accommodate said piston ends.

5. A hydraulic pump including a relatively high volume, relatively low pressure first stage pump, a relatively lower volume, relatively higher pressure second stage pump, means for driving said pumps, each of said pumps having an intake and a discharge, a direct, continuously open means connecting the discharge of the first stage pump to the intake of the second stage pump, a reservoir connected to the inlet of said first stage pump, a valve cylinder connected to the discharge of said first stage pump, a passage from one end of said cylinder to said discharge of said second stage pump, a passage connecting the other end of said cylinder to said reservoir, a valve piston slidable in said valve cylinder having a first hollow bore communicating with one end of said valve piston and with discharge pressure from said first stage pump, a first pressure regulator valve slidable in said cylinder independently of said piston and adjacent said one end of said piston, resilient means urging said first pressure regulator valve against said piston to close said bore, said resilient means being supplemented by discharge pressure from said second stage pump through said first named passage, said pressure regulator valve being operable upon a predetermined discharge pressure from said first stage pump to open said first named passage, said valve piston including a second hollow bore in its other end normally closed from communication with said first named bore, a by-pass in said cylinder and said valve piston closed from forming communication between said bores in one extreme position of said piston in said cylinder and forming communication between said bores in another extreme position thereof, means normally urging said valve piston toward said one position, second stage discharge pressure acting to urge said first regulator valve and valve piston toward said other position when a predetermined pressure is attained.

6. The structure of claim 5 and including a second pressure regulator valve normally closing said second bore and operable, upon a predetermined increase of pressure in said second bore to open said second named passage.

References Cited in the file of this patent

UNITED STATES PATENTS

2,549,897	Evrell	Apr. 24, 1951
2,643,613	Westbury et al.	June 30, 1953
2,715,875	Towler et al.	Aug. 23, 1955
2,812,771	McFarland et al.	Nov. 12, 1957

FOREIGN PATENTS

305,328	Great Britain	Feb. 7, 1929
703,569	France	Feb. 9, 1931
781,204	Great Britain	Aug. 14, 1957
954,077	France	June 6, 1949