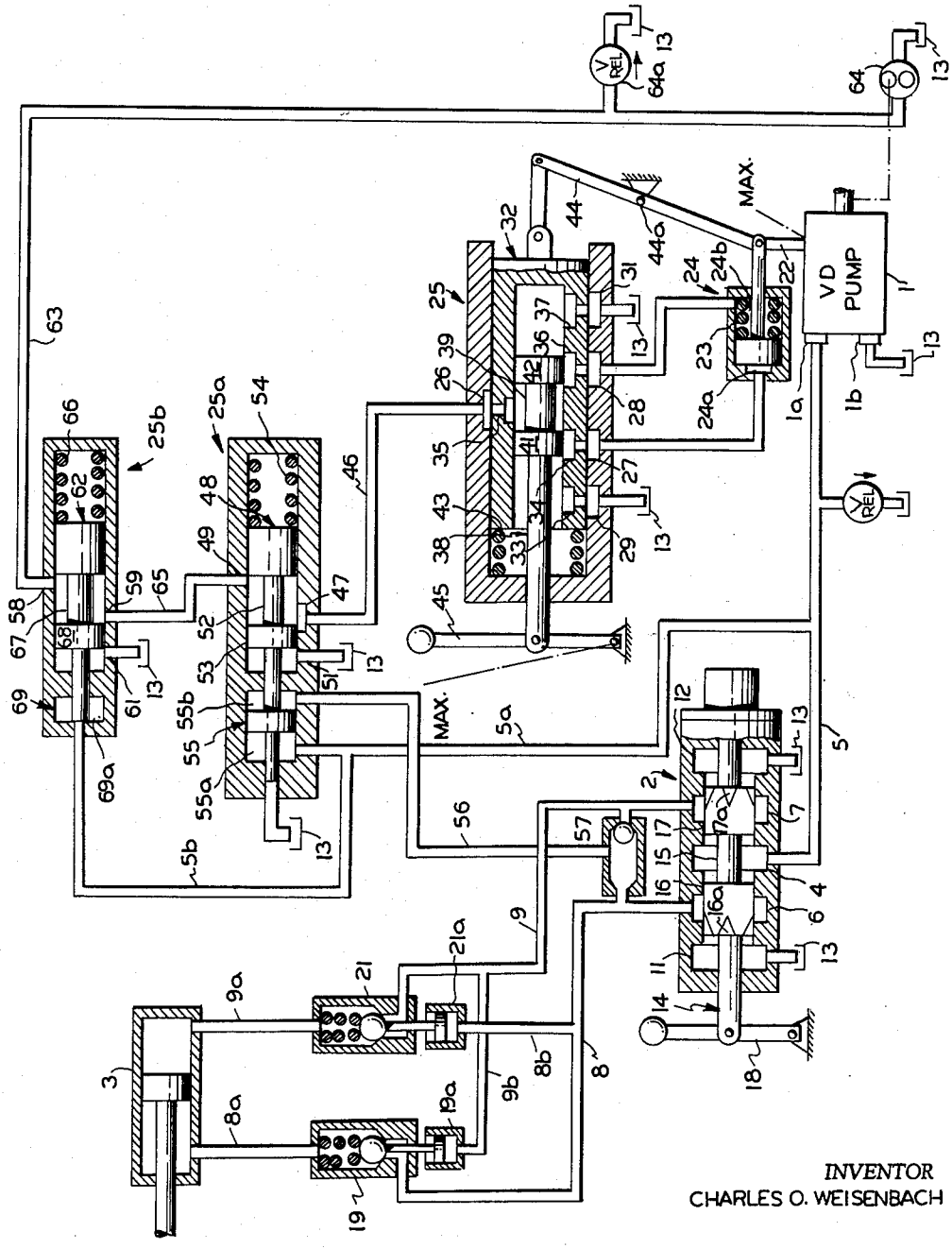
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HYDRAULIC SYSTEM

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HYDRAULIC SYSTEM

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This invention relates to hydraulic power systems employing a variable delivery, dscharge pressure compensated pump and a closed center distributing valve. As used herein, the term variable delivery, discharge pressure compensated pump means a pump whose delivery rate is automatically varied inversely with discharge pressure when that pressure is above a predetermined value.

In systems of this kind, the pressure upstream of the closed center distributing valve, i.e., the discharge pressure of the pump, is maintained substantially constant at a relative high value, for example, 3000 p.s.i., regardless of the flow demand imposed on the pump. When the distributing valve is closed and there is no demand for high pressure oil, the delivery rate of the pump is automatically reduced to a low value equal to the rate of leakage from the system and energy is conserved. However, since leakage occurs at a high pressure level, some energy is lost and the oil is heated. On the other hand, when the distributing valve is open and is being use to meter or throttle the flow of oil to a load requiring an operating pressure well below the setting of the pump compensator, there is a rather large pressure drop across the distributing valve. This too is wasteful of energy and causes heating of the oil.

The object of this invention is to provide an improved system of the type under discussion which limits maximum system pressure by means of the pump compensator, and which includes additional controls for maintaining a constant, relatively small pressure differential across the distributing valve at times when the valve is open and oil is being metered to the load, and for establishing a constant discharge pressure for the pump which is substantially lower than the compensator setting during periods when the distributing valve is closed. The addition of these novel controls reduces energy losses and heating of the oil under idle and active conditions of the system and thus overcomes the disadvantage of the prior system mentioned earlier.

The preferred embodiment of the invention is described herein in detail with reference to the accompanying drawing whose single figure is a schematic diagram of the improved system.

As shown in the drawing, the system includes a variable displacement pump 1 and a closed center directional control valve 2 and is employed to operate a double-acting piston motor 3. Motor 3 may be used to actuate a variety of loads, such as the boom or bucket of a loader or the blade of a bulldozer. Directional control valve 2 comprises an inlet chamber 4 which is connected with the discharge port 1*a* of pump 1 via supply conduit 5, a pair of motor chambers 6 and 7 which are connected with the rod and head ends of motor 3 by conduits 8 and 8*a* and 9 and 9*a* respectively, a pair of exhaust chambers 11 and 12 which are in continuous communication with reservoir 13, and a sliding valve plunger 14. The valve plunger 14 is formed with a central annular groove 15 and a pair of lands 16 and 17 in which are cut a plurality of V-shaped throttling notches 16*a* and 17*a*, respectively. In the illustrated neutral position of plunger 14, the land 16 and 17 issolate the inlet chamber 4 from the other chambers and the throttling notches 16*a* and 17*a* connect the motor chambers 6 and 7, respectively, with the exhaust chambers. The valve plunger 14 is shiftable in opposite directions from the neutral position by actuating lever 18 to connect a selected one of the two motor chambers 6 and 7 with the inlet chamber 4 and disconnect it from the adjacent exhaust chamber, and to reduce the restriction to flow from the remaining motor chamber to its adjacent exhaust chamber.

The conduits 8, 8*a* and 9, 9*a* are provided with conventional pilot operated check valves 19 and 21, respectively, which prevent drifting of motor 3 in either direction under the action of the load. The check valves 19 and 21 are equipped with piloting motors 19*a* and 21*a*, respectively, which are connected with the conduits 9 and 8 by branch conduits 9*b* and 8*b* respectively. Thus, the supply pressure in the conduit 8 or 9 to which directional control valve 2 is delivering fluid acts directly to open the check valve in that conduit and indirectly, through one of the piloting motors, to open the check valve in the remaining conduit.

Pump 1 may take various forms, but preferably is a rotary cylinder barrel, longitudinally reciprocating piston unit such as the ones marketed by my assignee under the trademark Dynapower. The pump is driven by an engine (not shown) and is supplied by the reservoir 13 with which its inlet port 1*b* is connected. The displacement of pump 1 is controlled by a lever 22 which is biased toward the illustrated zero displacement position by the thrust forces of the pump pistons and by a coil compression spring 23, and is positioned by a double-acting motor 24 which is under the control of a manually operated servo valve 25, a differential control valve 25*a* and a compensator pilot valve 25*b*. The servo valve 25 includes a housing containing a control chamber 26, a pair of motor chambers 27 and 28 which are connected with the opposed working chambers 24*a* and 24*b*, respectively, of motor 24, and a pair of exhaust chambers 29 and 31 which are connected with reservoir 13, Slidable within the housing of valve 25 is a valve sleeve 32 formed with five spaced, radial passages 33–37 which register, respectively with chambers 29, 27, 26, 28 and 31, and which contains a reciprocable valve plunger 38. The valve plunger 38 is formed with an annular groove 39 and a pair of lands 41 and 42, the latter being so dimensioned that when the plunger 38 is in the illustrated null position with respect to valve sleeve 32 each of the passages 34 and 36 communicates with both the passage 35 and one of the passages 33 and 37. Valve sleeve 32 is biased to the right by a coil compression spring 43 and is shiftable in the opposite direction by a follow-up lever 44 pivoted at 44*a*. Valve plunger 38, on the other hand, is shiftable to the left from the illustrated position by a manually operated actuator 45.

The control chamber 26 of servo valve 25 is connected by conduit 46 with motor port 47 of the differential control valve 25*a* which includes a valve plunger 48 that serves to control communication between that port and control port 49 and exhaust port 51. Valve plunger 48 is biased to the left to a supply position in which annular groove 52 interconnects ports 47 and 49, and land 53 blocks port 51, by a coil compression spring 54, and is shiftable in the opposite direction to a vent position, in which this connection is interrupted and port 47 communicates with port 51, by a double-acting motor 55. It will be noticed that motor port 47 is wider than land 53 and that therefore, as valve plunger 48 moves between its supply and exhaust positions it passes through a series of positions, termed the control range, in which motor port 47 communicates with both of the ports 49 and 51. The left hand working chamber 55*a* of motor 55 is in continuous communication with the supply conduit 5 through branch conduit 5*a*, and the right hand chamber 55*b* is selectively connected with whichever of conduits 8 and 9 is the higher pressure conduit through conduit 56 and a pressure responsive shuttle valve 57. From the description

of operation appearing below, it will be apparent that the valve 25*a* controls the displacement of pump 1 in such manner that the pressure differential between conduit 5 and either conduit 8 or conduit 9 normally is maintained substantially constant whenever directional control valve 2 is metering oil to motor 3.

The control port 49 of differential control valve 25*a* is vented and pressurized in accordance with the pressure in supply conduit 5 by the compensator pilot valve 25*b*. The pilot valve 25*b* includes inlet, motor and exhaust ports 58, 59 and 61, respectively, and a sliding valve plunger 62 that selectively connects motor port 59 with the other two ports. Inlet port 58 receives oil through conduit 63 from a fixed displacement control pump 64 that is driven in unison with pump 1 and whose discharge pressure is limited by a relief valve 64*a*, motor port 59 is connected by a conduit 65 with the control port 49 of valve 25*a*, and exhaust port 61 is in continuous communication with reservoir 13 A coil compression spring 66 biases valve plunger 62 to the illustrated supply position in which annular groove 67 interconnects ports 58 and 59, and land 68 blocks port 61, and a piston motor 69 is provided for shifting the plunger to the vent position in which land 68 blocks port 58 and ports 59 and 61 communicate with each other. Valve plunger 62 also has a lap position, intermediate the other two positions, in which land 68 isolates motor porat 59 from both of the ports 58 and 61. The working chamber 69*a* of motor 69 is in constant communication with supply conduit 5 through branch conduits 5*a* and 5*b*. As will be seen from the following description of operation, the valve 25*b* limits the pressure in conduit 5 to a maximum value determined by the setting of spring 66, by varying the displacement of pump 1.

Operation

With the actuators 18 and 45 in their illustrated neutral positions, valves 25, 25*a* and 25*b* assume their illustrated positions and the displacement of pump 1 is zero. Oil discharged by pump 64 is delivered to the control chamber 26 of servo valve 25 through conduit 63, inlet port 58, plunger groove 67, motor port 59, conduit 65, control port 49, plunger groove 52, motor port 47 and conduit 46, and from there it returns to reservoir 13 along two parallel paths, one including radial passage 35, plunger groove 39, radial passages 34 and 33, and exhaust chamber 29, and the other path including radial passage 35, plunger groove 39, radial passages 36 and 37, and exhaust chamber 31. Lands 41 and 42 restrict the flow of oil through these paths so that motor chambers 27 and 28 are subjected to backpressures which keep the working chambers 24*a* and 24*b* of control motor 24 liquid-filled. However, these backpressures are not high enough to cause control motor 24 to move the displacement control lever 22 away from its zero displacement position.

The operator increases the displacement of pump 1 by shifting valve plunger 38 of servo valve 25 to the left so that land 41 interrupts communication between radial passages 33 and 34 and land 42 interrupts communication between radial passages 35 and 36. The pressures in working chambers 24*a* and 24*b* now rise and fall, respectively, and control motor 24 commences to move lever 22 toward its maximum displacement position. As the lever 22 moves in the displacement-increasing direction, follow-up link 44 shifts valve sleeve 32 to the left, and, when the displacement control lever 22 reaches a position corresponding to the selected position of actuator lever 45, valve sleeve 32 will again be in a null position relative to valve plunger 38 and control motor 24 will come to rest. Since the control motor is not hydraulically locked, the biasing forces acting on lever 22 tend to return it to the zero displacement position. However, since movement of the lever 22 in this direction is accompanied by rightward movement of valve sleeve 32 relatively to valve plunger 38, working chamber 24*a* is pressurized as soon as lever 22 leaves the position established

by actuator 45 and the lever is returned to that position. It is thus seen that the servo control valve 25 is basically a position responsive control. However, it must be kept in mind that the servo valve 25 functions in this way only as long as oil under a pressure high enough to actuate motor 24 is being supplied to control chamber 26.

As displacement control lever 22 moves away from the zero displacement position, pump 1 commences to discharge oil to supply conduit 5, and, since directional control valve 2 is in its neutral position, the pressure in this conduit immediately rises. This pressure is transmitted to the working chamber 55*a* of motor 55 where it develops a force tending to shift valve plunger 48 to the right. Since, at this time, working chamber 55*b* is vented to reservoir 13 via conduit 56, shuttle valve 57, motor chamber 6, throttling notches 16*a* and exhaust chamber 11, motor 55 will move valve plunger 48 to a position in which land 53 is spanned by motor port 47 and the backpressure established in motor port 47 is reduced to a level just sufficient to enable control motor 24 to hold the displacement control lever 22 in its current position. If this reduction in the pressure of the oil being delivered to servo valve 25 occurs as control motor 24 is moving lever 22 to the position dictated by actuator 45, the lever 22 will come to rest short of that position and establish a discharge rate for pump 1 that just equals the rate of leakage from the system. In the illustrated installation, it is assumed that reservoir pressure is 50 p.s.i. and that spring 54 is so designed that a pressure differential of 150 p.s.i. between working chambers 55*a* and 55*b* is required in order to enable motor 55 to move valve plunger 48 to and hold it in a position in the center of its control range. Therefore, when directional control valve 2 is closed, servo control valve 25 is rendered incapable of effecting a further increase in pump displacement when the pressure in supply conduit 5 reaches about 200 p.s.i.

If the pressure in conduit 5 should rise above 200 p.s.i., as a result, for example, of an increase in the speed of the engine driving pump 1, motor 55 will shift valve plunger 48 to the right to either a new position within its control range in which the backpressure in port 47 is merely reduced or, if the pressure rise occurs suddenly, to the vent position. The biasing forces acting on displacement control lever 22 may now move it in the displacement-reducing direction with an accompanying transfer of oil from working chamber 24*a* to reservoir 13 along a path including motor chamber 27, radial passage 34, plunger groove 39, radial passage 35, control chamber 26, conduit 46, motor port 47, and exhaust port 51. When the change in displacement effected by this movement of lever 22 offsets the change in the rate of discharge attributable to the increase in driving speed, system pressure will decrease to the desired level of 200 p.s.i. and spring 54 will return valve plunger 48 to a position within its control range in which the backpressure in port 47 enables motor 24 to balance the biasing forces acting on displacement control lever 22. On the other hand, if the driving speed of pump 1 decreases and the pressure in conduit 5 drops below 200 p.s.i., spring 54 will shift valve plunger 48 toward its supply position and thus raise the pressure of the oil being delivered to motor 24. Under these conditions the control motor 24 will move lever 22 in the displacement-increasing direction toward the position called for by actuator 45. When system pressure is restored to the desired level, motor 55 will move valve plunger 48 back to a position within the control range in which motor 24 is again caused to balance the biasing forces acting on lever 22. From the foregoing discussion it will be apparent that as long as directional control valve 2 is closed, differential control valve 25*a* will maintain the pressure in conduit 5 constant at a relatively low value and, unless the operator has selected a very small displacement for pump 1, valve sleeve 32 will assume a position to the right of a null position relative to valve plunger 38.

In order to cause motor 3 to drive the load to the right, the operator shifts valve plunger 14 of the directional control valve 2 to the left to thereby cause plunger groove 15 to interconnect chambers 4 and 6 and to decrease the restriction to flow through notches 17*a*. Oil under pressure in conduit 5 may now flow to the rod end of motor 3 through inlet chamber 4, plunger groove 15, motor chamber 6, conduit 8, check valve 19 and conduit 8*a*. The supply pressure in conduit 8, acting through piloting motor 21*a*, opens check valve 21 and allows oil in the head end of motor 3 to return to reservoir 13 along a path comprising conduit 9*a*, check valve 21, conduit 9, motor chamber 7, throttling notches 17*a*, and exhaust chamber 12. If shuttle valve 57 is not already in its illustrated position, it will shift to that position as soon as valve 2 is opened so that motor 55 will respond to the pressure differential between conduit 5 and 8.

As directional control valve 2 is opened, the pressure rises in conduit 8 and motor 55 and spring 54 immediately shift valve plunger 48 of the differential control valve 25*a* to its supply position. Control motor 24 now begins to move displacement control lever 22 in the displacement-increasing direction toward the position selected by actuator 45. When a pressure differential of 150 p.s.i. between conduits 5 and 8 has been re-established, motor 55 will move valve plunger 48 of valve 25*a* back to a position within its control range, thereby reducing the backpressure in port 47 and preventing motor 24 from further increasing the displacement of pump 1. It is assumed that valve 2 is metering oil to motor 3, i.e., that valve 2 is limiting the flow to motor 3 to a rate less than the selected displacement of pump 1, and therefore lever 22 comes to rest in a position short of that called for by actuator 45. The valve 25*a* and the lever 22 will remain at rest until the operator changes the position of the directional control valve 2. If he closes this valve slightly to retard motion of motor 3, the pressure in conduit 5 will rise and motor 55 will shift valve plunger 48 to the right to reduce the pressure in working chamber 24*a*. This allows the biasing forces acting on lever 22 to move it in the displacement-reducing direction. As a result, the pressure in conduit 5 will decrease. When the desired pressure differential of 150 p.s.i. has been re-established, spring 54 will return valve plunger 48 to a position within its control range in which the opposing forces exerted on lever 22 are again balanced. On the other hand, if the operator opens valve 2 to increase the speed of motor 3, differential control valve 25*a* will automatically effect a corresponding increase in the displacement of pump 1. Thus, throughout the metering range of directional control valve 2, valve 25*a* will vary the displacement of pump 1 in inverse relation to the degree of throttling at the directional control valve to thereby maintain the pressure differential across this valve substantially constant at 150 p.s.i. Since the absolute pressure in the system depends upon the magnitude of the load being moved by motor 3, and changes in this load produce equal changes in the pressures in working chambers 55*a* and 55*b*, it will be apparent that the control action of valve 25*a* is independent of the magnitude of the load.

When valve 2 is moved to a position outside its metering range, i.e., when it permits a rate of flow to motor 3 greater than that called for by actuator 45, the pressures in conduits 5 and 8 become substantially equal and spring 54 moves valve plunger 48 of the differential control valve 25*a* to its illustrated supply position. This enables the servo control valve 25 to move displacement control lever 22 to the position selected by the operator. When correspondence between the positions of lever 22 and actuator 45 has been established, valve sleeve 32 will be in a null position relative to valve plunger 38.

When motor 3 has moved the load to the desired position, the operator returns directional control valve 2 to its illustrated neutral position, thereby venting working chamber 55*b* of motor 55 and permitting differential control valve 25*a* to reduce the displacement of pump 1 and again establish a pressure of about 200 p.s.i. in supply conduit 5.

Movement of motor 3 to the left is effected by shifting valve plunger 14 to the right. This causes plunger groove 15 to interconnect chambers 4 and 7 and reduces the restriction to flow through notches 16*a*. Now fluid under pressure is delivered to the head end of motor 3 along a path including conduit 5, inlet chamber 4, plunger groove 15, motor chamber 7, conduit 9, check valve 21 and conduit 9*a*, and the rod end of the motor is vented to reservoir 13 along a path including conduit 8*a*, check valve 19, conduit 8, motor chamber 6, throttling notches 16*a*, and exhaust chamber 11. During this mode of operation, the valves 25 and 25*a* function in exactly the same way as in the case of rightward movement of motor 3, except that now, since shuttle valve 57 assumes its leftmost position, valve 25*a* responds to the pressure differential between conduits 5 and 9 rather than conduits 5 and 8.

If, at any time during operation, the pressure in supply conduit 5 rises to the maximum for which the system is designed (assumed to be 3000 p.s.i. in the illustrated embodiment), motor 69 will shift valve plunger 62 of the compensator valve 25*b* to its lap position, in which land 68 isolates port 59 from both of the ports 58 and 61, and thus cuts off the supply of oil to the valves 25*a* and 25. If this condition arises at a time when pump 1 is operating below maximum displacement, this interruption of the supply of oil prevents the operator from further increasing the displacement of the pump by shifting actuator 45. If the pressure should continue to rise, motor 69 will shift valve plunger 62 to its vent position and open an exhaust path from working chamber 24*a* to reservoir 13 comprising motor chamber 27, radial passage 34, plunger groove 39, radial passage 35, control chamber 26, conduit 46, motor port 47, plunger groove 52, control port 49, conduit 65, motor port 59 and exhaust port 61. Opening of this path decreases the pressure in working chamber 24*a* and allows the biasing forces acting on lever 22 to move it in the displacement-reducing direction. When the displacement of pump 1 has been reduced sufficiently to restore system pressure to the desired maximum of 3000 p.s.i., spring 66 will shift valve plunger 62 back to its lap position and close the exhaust path just mentioned. Since follow-up link 44 will have moved valve sleeve 32 to the right from a null position with respect to valve plunger 38, land 41 will now isolate radial passage 34 from radial passage 33. Thus, if valve 25*a* is in its supply position when valve 25*b* returns to its lap position, motor 24 will be hydraulically locked and will hold lever 22 in the reduced displacement position. On the other hand, if valve plunger 48 of valve 25*a* is in a position within its control range at the time valve 25*b* shifts back to its lap position, working chamber 24*a* will communicate with reservoir 13 through the ports 47 and 51 of this valve and the control motor will not be hydraulically locked. In this case, as soon as the pilot valve 25*b* shifts to the lap position the pressure in working chamber 24*a* will tend to decrease and the biasing forces acting on lever 22 will tend to move it in the displacement-reducing direction. However, as soon as the lever 22 moves away from the displacement position in which system pressure is stabilized at 3000 p.s.i., the pressure in conduit 5 will decrease and spring 66 will shift valve plunger 62 to its supply position. This action will raise the pressure in working chamber 24*a* and cause the motor 24 to return lever 22 to the position from which it tended to move.

The spring 66 of compensator pilot valve 25*b* is so designed that a system pressure of 3000 p.s.i. is required to hold valve plunger 62 in its lap position. Therefore, as long as the pressure in conduit 5 is at this level, valve 25*b* will exert an overriding control effect and position lever 22 in a smaller displacement position than that called for by either of the valves 25*a* and 25. Under

severe loading conditions, valve 25*b* may even cause displacement control lever 22 to move to its zero displacement position. When the load on the supply system decreases to a level at which the pressure in conduit 5 is below 3000 p.s.i. even when the pump 1 is operating at the displacement selected by the operator through valve 25, the valve plunger 62 of compensator pilot valve 25*b* returns to its illustrated supply position and valves 25 and 25*a* regulate pump displacement in the normal manner previously described.

When the system is in operation and valve 2 is closed for relatively short periods of time, the operator leaves actuator 45 in its selected position so that pump 1 continues to deliver fluid to supply conduit 5. Under these conditions, valve 25*a*, as described above, maintains discharge pressure at a low value and thus minimizes energy losses. During periods of prolonged inactivity, or at times when conduits 5, 5*a* or 5*b* must be opened, the operator moves actuator 45 back to its zero displacement position. When he does this, valve plunger 38 is shifted to the right from a null position with respect to valve sleeve 32 thereby causing land 41 to isolate radial passage 34 from radial passage 35 and causing land 42 to isolate radial passage 36 from radial passage 37. The pressures in working chambers 24*b* and 24*a* of motor 24 now increase and decrease, respectively, so that the motor 24 acts in aid of the biasing forces being exerted on lever 22 and moves the lever 22 to the zero displacement position. When the lever reaches this position, valve plunger 38 will be in the neutral position relative to the housing of valve 25 and follow-up link 44 will have returned valve sleeve 32 to a null position relative to the valve plunger.

It should be realized that the servo control 25 of the illustrated embodiment is an optional feature of the invention provided to enable the operator to limit pump displacement to a valve less than its design maximum and also to enable him to reduce pump displacement to zero during long periods of system inactivity. In installations where these features are not required, valve 25 can be eliminated and working chamber 24*a* can be connected directly to conduit 46. In this case, working chamber 24*b* is continuously vented to reservoir 13 so that motor 24, in effect, becomes a single-acting motor.

As stated previously, the drawing and description relate only to the preferred embodiment of the invention. Since many changes can be made in the structure of this embodiment without departing from the inventive concept, the following claims should provide the sole measure of the scope of the invention.

What I claim is:

1. In combination
(a) a variable delivery hydraulic pump;
(b) a discharge pressure compensator for varying pump delivery rate in inverse relation to discharge pressure to thereby limit that pressure to a predetermined value;
(c) a fluid pressure motor;
(d) distributing means, including a closed center distributing valve capable of throttling flow, for selectively delivering fluid discharged by the pump to the motor;
(e) means effective when the distributing valve is throttling flow to the motor and discharge pressure is below said predetermined value to vary pump delivery rate in inverse relation to the pressure differential across the distributing valve to thereby maintain said differential constant at a preselected value; and
(f) means effective when the distributing valve is closed for varying pump delivery rate in inverse relation to discharge pressure to thereby maintain said pressure constant at a value materially lower than said predetermined value.

2. In combination
(a) a variable delivery hydraulic pump;
(b) a discharge pressure compensator for varying pump delivery rate in inverse relation to discharge pressure to thereby limit that pressure to a predetermined value;
(c) a fluid pressure motor;
(d) distributing means, including a closed center distributing valve capable of throttling flow, for selectively delivering fluid discharged by the pump to the motor, the distributing means including passage means which is vented when the distributing valve is closed and which is subject to the pressure in the fluid passing from the distributing valve to the motor when that valve is open; and
(e) means responsive to the differential between pump discharge pressure and the pressure in said passage means and effective when pump discharge pressure is below said predetermined value and the distributing valve is throttling flow to the motor for varying pump delivery rate to thereby maintain said differential constant.

3. In combination
(a) a variable delivery pump having a delivery control element biased toward a minimum delivery position;
(b) a fluid pressure control motor for shifting the delivery control element toward a maximum delivery position;
(c) a source of control fluid under pressure and a reservoir;
(d) a control conduit connected with the control motor;
(e) a pilot valve responsive to the discharge pressure of the pump and arranged to connect the control conduit with the source and the reservoir when the pressure is below and above, respectively, a predetermined value;
(f) a fluid pressure work motor;
(g) distributing means, including a closed center distributing valve capable of throttling flow, for selectively delivering fluid discharged by the pump to the work motor, the distributing means including passage means which is vented to the reservoir when the distributing valve is closed and which is subject to the pressure in the fluid passing from the distributing valve to the work motor when that valve is open;
(h) a differential control valve interposed in the control conduit between the control motor and the pilot valve and shiftable in opposite directions from a position in which it connects the control motor with both the pilot valve and the reservoir, movement of the valve in a first direction serving to restrict communication between the control motor and the reservoir and movement in the second direction serving to restrict communication between the control motor and the pilot valve;
(i) means biasing the differential control valve in the first direction;
(j) means responsive to the pressure in said passage means urging the differential control valve in the first direction; and
(k) means responsive to the discharge pressure of the pump for urging the differential control valve in the second direction.

4. The combination defined in claim 3 including
(a) a servo control valve interposed in the control conduit between the differential control valve and the control motor, the servo control having an input member and a follow-up member movable relatively to each other from a null position in which the control motor is connected with both the differential control valve and the reservoir, relative movement of the valve members in oposite directions from the null position disconnecting the control motor from the differential control valve and the reservoir, respectively; and
(b) a follow-up connection between the control motor

and the follow-up member of the servo control valve.

5. In combination

(a) a variable delivery pump having a delivery control element biased toward a minimum delivery position;

(b) a fluid pressure control motor for shifting the delivery control element toward a maximum delivery position;

(c) a source of control fluid under pressure and a reservoir;

(d) a control conduit connected with the control motor;

(e) a pilot valve responsive to the discharge pressure of the pump and arranged to connect the control conduit with the source and the reservoir when the pressure is below and above, respectively, a predetermined value;

(f) a double-acting work motor having opposed working chambers;

(g) distributing means, including a closed center directional control valve capable of throttling flow for selectively connecting the opposed working chambers with the pump and reservoir, the distributing means including a pair of passage means which are vented to reservoir when the directional control valve is closed and one or the other of which is subject to the pressure in the fluid passing from the distributing valve to the work motor when that valve is open;

(h) a differential control valve interposed in the control conduit between the control motor and the pilot valve and shiftable in opposite directions from a position in which it connects the control motor with both the pilot valve and the reservoir, movement of

the valve in a first direction serving to restrict communication between the control motor and the reservoir and movement in the second direction serving to restrict communication between the control motor and the pilot valve;

(i) means biasing the differential control valve in the first direction;

(j) first and second opposed fluid pressure responsive means acting on the differential control valve, the first means being responsive to the discharge pressure of the pump and urging the differential control valve in the second direction; and

(k) a shuttle valve for connecting the second pressure responsive means with whichever of the passage means is at the higher pressure.

6. The combination defined in claim 5 including

(a) a servo control valve interposed in the control conduit between the differential control valve and the control motor, the servo control having an input member and a follow-up member movable relatively to each other from a null position in which the control motor is connected with both the differential control valve and the reservoir, relative movement of the valve members in opposite directions from the null position disconnecting the control motor from the differential control valve and the reservoir, respectively; and

(b) a follow-up connection between the control motor and the follow-up member of the servo control valve.

No references cited.

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