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CONTROL APPARATUS FOR HYDRAULIC SYSTEMS

Filed Dec. 11, 1947

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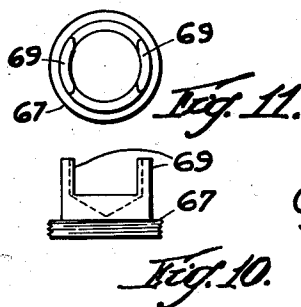
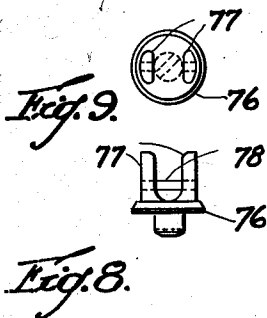
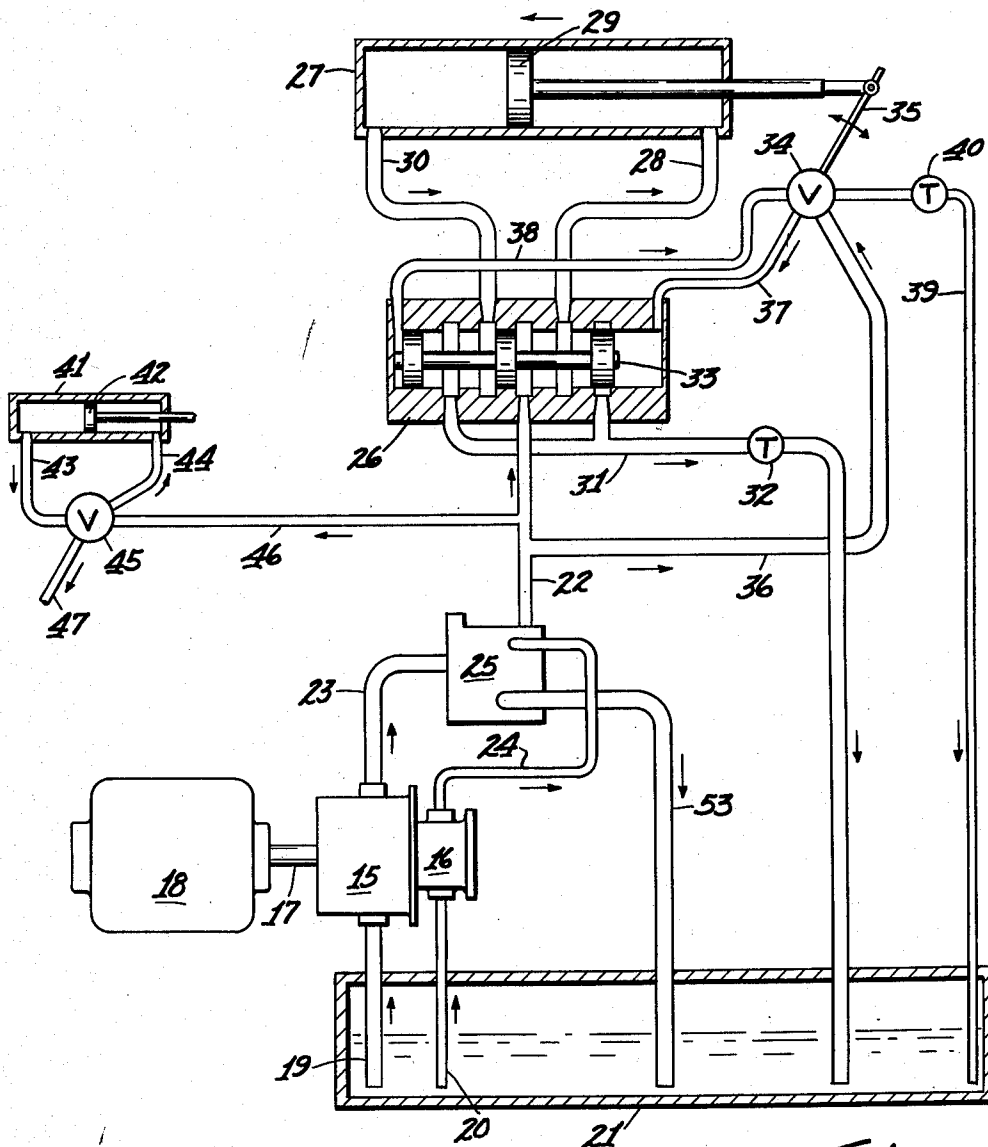


Fig. 1.

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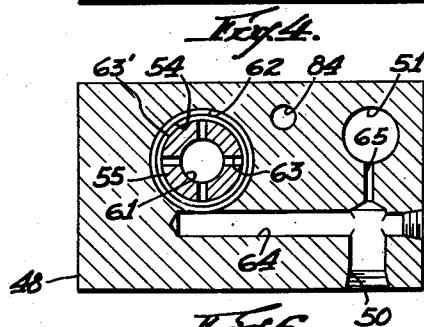
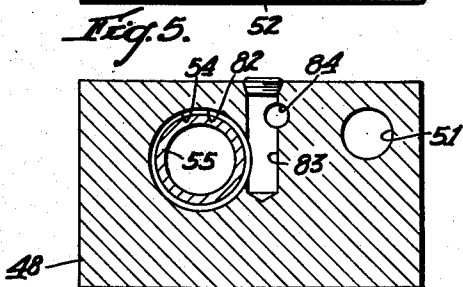
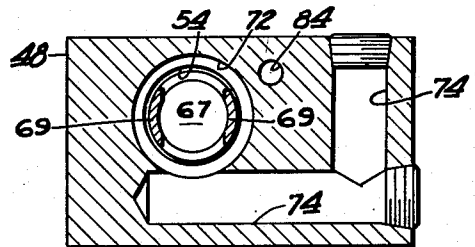
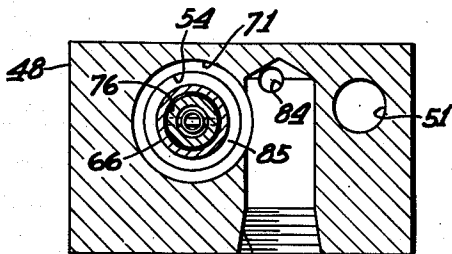
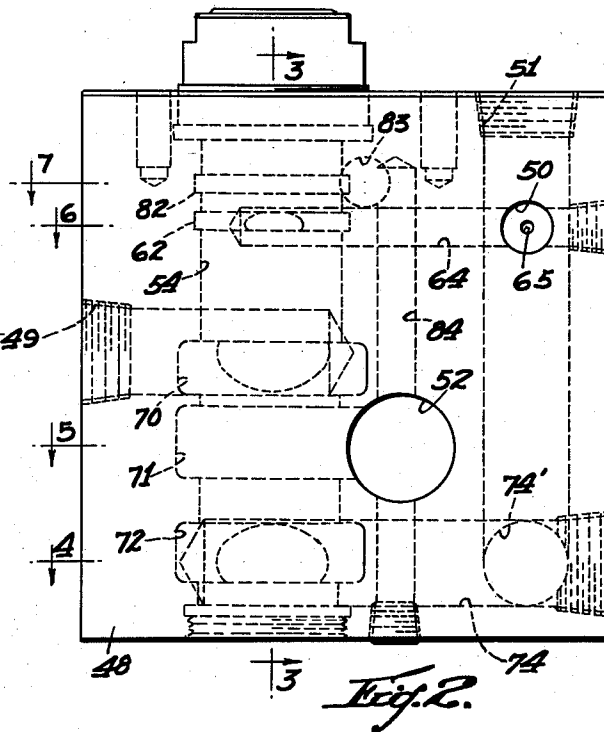
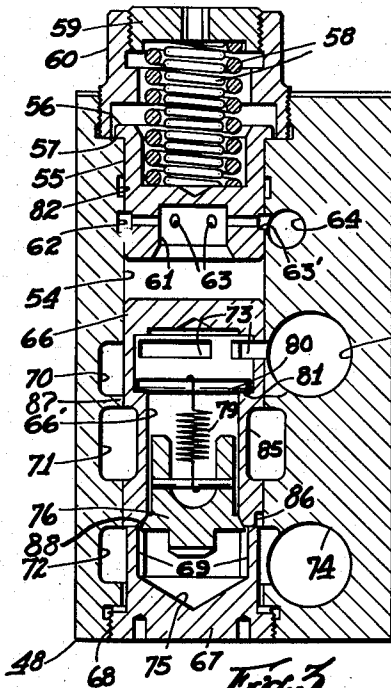
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2 SHEETS—SHEET 2



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CONTROL APPARATUS FOR HYDRAULIC SYSTEMS

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This invention relates to a combined rapid and slow traverse hydraulic system, and more particularly to a hydraulic system requiring both the rapid and slow traverse cycles thereof to be performed under a constant machine oil pressure at all times.

A primary object of the present invention is to provide a variable capacity hydraulic system with dump and relief valve mechanism which will be effective to maintain the machine oil pressure substantially constant, regardless of the oil flow demand.

A further object of the present invention is to provide a dump and relief valve mechanism which is so constructed that it will maintain a constant machine oil pressure when connected in a hydraulic system of the type having a pair of constantly running pumps, which are connected to a traversing mechanism for the purpose of actuating the latter at different rates of speed.

Another object of the present invention is to provide a dump and relief valve mechanism which is adapted to maintain a constant machine oil pressure in a rapid and slow traverse hydraulic system, regardless of whether or not other auxiliary hydraulic units, such as work-clamping means, are actuated from the main hydraulic line during the traversing cycles.

A still further object of the present invention is to provide a dump and relief valve mechanism which is adapted to be used in conjunction with a dual pump hydraulic traversing system of either open-feed or locked feed type, for the purpose of maintaining within said system a constant machine oil pressure and which will operate without producing squealing and chatter and will be effective to maintain the oil temperature at a low figure at all times.

With the above and other objects in view, as will hereinafter appear, the invention comprises the devices, combinations and arrangements of parts hereinafter set forth and illustrated in the accompanying drawings of a preferred embodiment of the invention from which the several features and the advantages attained thereby will be readily understood by those skilled in the art.

In the accompanying drawings:

Fig. 1 represents schematically a two-pump hydraulic system of the type to which the dump and relief valve mechanism of my invention is applicable.

Fig. 2 represents a side elevation of the present dump and relief valve mechanism.

Fig. 3 represents a sectional view taken substantially along the line 3-3 of Fig. 2.

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Fig. 4 represents, on a reduced scale, a sectional view substantially in the plane of the line 4 of Fig. 2.

Fig. 5 represents, on a reduced scale, a sectional view substantially in the plane of the line 5 of Fig. 2.

Fig. 6 represents, on a reduced scale, a sectional view substantially in the plane of the line 6 of Fig. 2.

Fig. 7 represents, on a reduced scale, a sectional view substantially in the plane of the line 7 of Fig. 2.

Fig. 8 represents a side elevation view of a check valve used in the present dump and relief valve mechanism.

Fig. 9 represents a top plan view of the check valve illustrated in Fig. 8.

Fig. 10 represents a side elevation view of a screw plug used in the present dump and relief valve mechanism.

Fig. 11 represents a top plan view of the screw plug illustrated in Fig. 10.

In the hydraulic system shown schematically by Fig. 1, there is shown a relatively large capacity pump 15 and a relatively small capacity pump 16 connected in parallel, and operated on a single shaft 17 from a common motor 18. Suitable conduits 19 and 20 respectively connect the pumps 15 and 16 to a fluid supply tank 21.

In said system, the pumps 15 and 16 are adapted to operate continuously for the delivery of fluid from the tank 21 through conduits 23 and 24 respectively that are connected to the dump and relief valve mechanism of my invention, designated generally by the numeral 25. From the latter, the pumped fluid is adapted, as herein-after described, to pass in the demanded amount to the fluid-operated elements of the system by way of a single delivery connection 22; and is adapted to have its undemanded portion returned to the tank 21 by way of a return connection 53. In the hydraulic system of Fig. 1, the delivery conduit 22 is connected, as shown, to the interior of a conventional valve 26 which is adapted for controlling the operation of a usual type of power cylinder or liquid-operated motor 27. For example, as illustrated in Fig. 1, the line 22 is connected directly with the mid-portion of the control valve 26 whose movable piston element 33 is shown in a position to direct the oil from power line 22 into the right-hand end portion of the power cylinder 27 by way of a line 28. Thus, the power cylinder's piston 29 will advance toward the left-hand end of the cylinder, thereby discharging exhaust fluid through a line 30 which is connected to an intermediate portion of the

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control valve 26. It is also to be noted that each end portion of the valve 26 is connected in a conventional manner to a line 31 which discharges into the supply tank 21 by way of a throttle 32 which is adapted to be manually or automatically operated for the purpose of varying the traversing speed of the piston 29. It is obvious that when the control valve piston 33 is moved to the right, it will be effective to connect the power line 22 with the line 30 of the power cylinder, and also to connect the power cylinder line 28 with the discharge line 31, all to the end that the piston 29 will be forced toward the right-hand end of the power cylinder.

For actuating the control valve piston 33, a conventional type of four-way valve 34 is connected in the system. More specifically, the valve 34 is adapted to be actuated by means of an appropriate mechanism 35 connected with the power piston 29. Also, the valve 34 is connected directly with the power line 22 by way of a line 36. Therefore, when the valve 34 is actuated in one direction, it will be effective to supply fluid from the power line 22 directly to the right-hand end of the control valve 26 by way of a line 37. Such a flow of fluid will, of course, force the piston 33 into the position illustrated in Fig. 1. During this portion of the control valve cycle, oil from the left-hand end of the control valve 26 may escape through a line 38 which is connected to a discharge line 39 by way of the valve 34. If desired, a dwell throttle 40 may be provided in the line 39 for controlling the dwell of the traversing mechanism. The free end portion of the line 39 is connected with the tank 21. Obviously, when the four-way valve mechanism 34 is shifted in the other direction, it will be effective to reverse the functions of the lines 37 and 38 to the end that the control valve piston 33 will be shifted toward the right-hand end portion of the control valve cylinder, thereby reversing the direction of movement of the power piston 29.

If desired, various auxiliary hydraulic mechanisms may be operated from the main power line 22. For example, in Fig. 1 there is illustrated a second hydraulic cylinder 41 which may be used for actuating work-clamping mechanisms. Such a cylinder 41 may be provided with the usual piston 42 which is actuated by fluid pressure delivered to the cylinder, in the conventional manner, by means of lines 43 and 44 which are connected to a four-way manually or automatically operated valve 45 which in turn is connected directly to the power line 22 by way of a line 46. Also, the valve 45 may be provided with a discharge line 47 for delivering the exhaust fluid directly to the tank 21.

Figs. 2 to 11, inclusive, particularly illustrate the present dump and relief valve mechanism which is designated by the numeral 25 of Fig. 1. In a preferred form of this dump and relief valve mechanism, it is constructed as a single unit, contained in a suitable housing 48. Referring particularly to Fig. 2, the housing 48 is provided with an inlet conduit 49 having the line 23 from the large-capacity pump 15 connected directly thereto. Also, the small capacity pump line 24 is adapted to be connected directly to an inlet conduit 50 provided in the housing 48. A vertically disposed outlet passage 51, provided in the housing 48, is adapted to have the main power line 22 connected directed therewith. Still referring to Fig. 2, the housing 48 is provided with a return conduit 52, to which the return line 53

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is connected for the delivery back to tank 21 of all of the pumped fluid in excess of that which is being passed into the system by way of power line 22.

Referring particularly to Figs. 2 and 3, the housing 48 is provided with a vertically disposed valve-receiving bore or chamber 54, in the top portion of which is slidably received a valve 55 which is provided with a flange 56 at its upper end. In opposition to said flange 56, the bore 54 has a shoulder 57 which serves to limit downward movement of valve 55 under the force of a pair of coil springs 58, 58, the lower portions of which engage the valve 55. The upper portions of the springs 58, 58 engage an adjusting screw 59 which is threadably received within a plug 60 screwed within the upper portion of the bore 54. The screw 59 is adapted to vary the pressure of the springs 58, 58 as the same is adjusted within the plug 60.

The valve 55 is provided with a central hollow downwardly opening portion 61 which, in the position shown in Fig. 3, communicates directly with an annular bore groove 62 by way of radial conduits or ports 63, 63 and a valve groove 63' whose width is that of the land or ungrooved bore portion between bore groove 62 and an upper bore groove 62. This groove 62 is provided about the bore 54 and communicates directly with a horizontal passage 64 which, in turn, is connected directly with the inlet passage 50 to which is connected the oil line 24 from the low capacity pump 16 (see Fig. 6). From this, it is to be understood that fluid from the low capacity pump is adapted to be delivered into the valve bore space below the valve 55 by way of the passages 50 and 64, grooves 62 and 63' and the radial conduits 63.

Referring particularly to Figs. 2 and 6, it is to be noted that the oil inlet passages 50 and 64 from the small capacity pump 16 are connected by way of a relatively restricted orifice 65 to the outlet passage 51 that serves the power line 22. From this, it will be understood that some of the fluid under pressure coming from the low capacity pump 16 will pass directly into the outlet passage 51 and power line 22, but at a pressure normally somewhat lower than that prevailing within the passage 50, because of the restricted character of the orifice 65.

Referring again to Fig. 3, there is slidably disposed within the lower portion of the valve bore 54, a second or control valve 66 in the form of a hollow piston, whose skirt portion extends downwardly within the bore 54 into abutting relation, in the valve position of Fig. 3, with spaced legs 69, 69 that are provided by a plug 67 which closes the lower end of bore 54. Figs. 10 and 11 illustrate the closure plug 67 which is adapted to be threaded as at 68 directly into the lower portion of the bore 54. Provided upon this screw plug 67 are the spaced leg portions 69, 69 which are adapted for engagement with the valve 66 in the manner as illustrated in Fig. 3. Surrounding the valve bore 54 and adapted to communicate therewith in the zone occupied by the valve 66 are three relatively large vertically spaced annular grooves 70, 71 and 72. The upper groove 70 communicates directly with the inlet passage 49, which as hereinabove noted, is connected with the large capacity pump 15 by way of the line 23. Also, the inner hollow portion 66' of the valve 66 communicates, in the position shown in Fig. 3, directly with the annular groove 70 by way of a plurality of ports 73, 73. Thus,

fluid from the large capacity pump 15 has direct access to the interior portion 66 of the valve 66 by way of the annular groove 70 and the valve ports 73, 73.

Referring particularly to Figs. 2, 3 and 5, it is to be noted that the intermediate annular groove 71 communicates directly with the horizontally disposed passage 52 which in turn is connected to the tank 21 by way of the return line 53. In the position of valve 66, shown by Fig. 3, it presents a reduced diameter portion 85 of its skirt in opposed relation to the bore groove 71. The latter, as shown in Fig. 3, is separated from the upper bore groove 70 by a relatively narrow land 87. Thus, although the valve 66 is positioned in Fig. 3 to cut off any access of the pumped oil to the return groove 71, it will be evident that a slight upward movement of valve 66 from its illustrated lowermost position will place the return groove 71 in full communication with the groove 70 and thereby cause the fluid coming from the large pump 15 to be returned to the tank 21.

Referring now to Figs. 2, 3 and 4, it is to be understood that the lower annular groove 72 communicates directly with a horizontally disposed passage 74, one end portion of which is connected directly with the vertically disposed discharge conduit 51 by way of a short cross passage 74. Also, it is to be further understood that this annular groove 72 communicates directly with the inner portion 75 of the closure plug 67 by virtue of the fact that the plug legs 69, 69 are spaced apart from each other.

Referring particularly to Figs. 3, 8 and 9, there is provided within the lower hollow portion 66 of the valve 66, a check valve 76 which is provided with a pair of spaced legs 77, 77 between which is mounted a cross pin 78. Associated with the check valve 76 is a coil spring 79, one end of which is fastened to the cross pin 78 and the other end of which is fastened to a second cross pin 80. Thus the spring 79 serves to hold valve 76 yieldingly against its annular seat 88 which is provided by the lower end of the valve 66. This second cross pin 80 is mounted upon a shoulder portion 81 of the valve 66. It is obvious that when the check valve 76 is in the closed position shown by Fig. 3, it will prevent any flow of fluid from the lower portion of the bore 54 from entering the hollow portion of the valve 66.

Referring particularly to Figs. 2, 3, 5 and 7, it is to be noted that the upper portion of the bore 54 is provided with the annular groove 82 which communicates directly with a transverse passage 83 which in turn is connected with a vertically disposed passage 84. Referring particularly to Figs. 2 and 5, it will be noted that this vertical passage 84 communicates with the return passage 52, so that fluid from the small pump 16 that reaches the groove 82 and passage 83 will ultimately find its way back into the tank 21. However, as shown in Fig. 3, the groove 82 is normally blocked off by valve 55 from any access of fluid thereto.

In the operation of the present system, let it be assumed that the large pump 15 has a capacity of eleven and one-half gallons per minute, while the low capacity pump 16 has a capacity of one and one-half gallons per minute, thus giving a combined pump capacity of thirteen gallons per minute.

Assuming that the various elements of the dump and relief valve 25 are disposed in the positions shown in Fig. 3 and that the throttle 32 is

wide open when the system is initially placed in operation, it is to be understood that oil from the small capacity pump 16 will be delivered into the bore 54 between valves 55 and 66, by way of the passages 50 and 64, and the radial parts 63, 63. This oil will be effective to build up a static oil pressure between the valves 55 and 66, which pressure will be effective to urge these two valves in opposite directions, thereby forcing the valve 55 against the springs 58, 58, and the valve 66 against the legs 69, 69 of the screw plug 67. Due to the existence of the orifice 65 between the passages 50 and 51, the oil pressure in the space between the valves 55 and 66 will be initially greater than that existing within the outlet passage 51 and its connecting power line 22.

Under these open throttle conditions, the oil from the large capacity pump 15 is adapted to reach the interior space 66 of valve 66 by way of the passage 49, the annular groove 70, and the valve ports 73, 73. As thus delivered into the hollow portion 66 of the valve 66, the oil from the large capacity pump will immediately force the check valve 76 open and flow into the passage 74 by way of the interconnecting annular groove 72. Thus, the oil from both pumps 15 and 16 is fully available, when throttle 32 is opened for rapid traverse, for flow into outlet passage 51 to supply the demands of power line 22. Although the oil from the large capacity pump will have a tendency to lift the valve 66 within the bore 54, it will be understood that whenever the oil demand, as determined by the throttle 32, is large, the back pressure within the passage 51 and the line 22 will be lowered and thus, by virtue of the orifice 65, the oil pressure effective against the upper portion of the valve 66 will be greater than that which tends to move said valve upwardly.

However, with throttle 32 moved, for slow traverse purposes to a position where the oil demand becomes less than thirteen gallons per minute (the assumed combined capacity of the two pumps), it will be obvious that the back pressure within the line 22 and the passage 51 will tend to increase, thus increasing the oil pressure within the upper portion of the bore or chamber 54 and within the hollow portion 66 of the valve 66. As this back pressure increases, it will be effective to shift the valve 55 upwardly within the bore 54 and against the force of the springs 58, 58 and thereby tend to cut off communication between the valve groove 63 and the oil supply groove 62. Of course, the springs 58, 58 are adjusted to a predetermined or desired setting by means of the adjusting screw 59 as hereinabove noted.

When communication has been cut off between the valve groove 63 and the groove 62, no further oil can gain entrance into the upper portion of the bore or chamber 54 and thereby increase the pressure within this portion of the bore. However, as the back pressure increases within the main line 22 and the passage 51, it will soon be great enough to move the valve 66 upwardly within the bore 54 and, of course, this upward movement of the valve 66 cannot occur until the predetermined force of the springs 58, 58 is overcome, for the liquid between the valves 55 and 66 is substantially incompressible. This initial upward movement of the valve 66 will cause valve 55 to raise in synchronism therewith, thus placing the valve groove 63 of valve 55 in partial communication with the annular groove 82 so that the oil pressure between valves 55 and 66 is slightly relieved by way of the conduits 83, 84 and 52. Also, this upward movement

of valve 66 will be effective almost immediately, to put the large pump delivery passage 49 in partial communication with groove 71, return passage 52, and dump line 53, to the end that less than the total capacity of the large pump 15 is delivered to the main line 22, thus maintaining a constant predetermined oil pressure in the system. It should be understood that as soon as the oil between the valves 55 and 66 is so relieved, the springs 58, 58 will force the valve 55 down again thereby maintaining a constant pressure upon the valve 66.

If the above slight upward movements of the valves 55 and 66 are not sufficient to maintain a constant liquid pressure within the system, the valves 55 and 66 will continue their upward movements until all of the flow from large pump 15 is diverted to the return line 52, 53 and a greater portion of the oil between the two valves is relieved by way of valve groove 63' and bore groove 82. At this juncture, the check valve 76 will close under the biasing pressure of its spring 79 and the force of the fluid within the lower portion of the valve bore 54. Again, as soon as the oil is relieved between the two valves by way of groove 82, the valve 55 will tend to shift downwardly under the force of its springs. It should be realized that when the entire capacity of the large pump 15 is so dumped, only one and one-half gallons of the oil per minute, or the assumed capacity of pump 16, is delivered to the main line 22, thus maintaining the oil pressure constant within the traversing system.

If, however, the above relieving and dumping action is not sufficient to maintain a constant oil pressure, the valve 66 will be further elevated under the pressure of the oil in the lower portion of the bore 54, thereby again raising valve 55 to its flow-diverting position where some of the oil from the small pump 16 will enter the groove 82 and pass by way of conduits 83 and 84 to the lower end of valve bore 54. In the ensuing further rise of valve 66, the excess of oil thus diverted to the lower end of bore 54 escapes into return groove 71 by way of an exterior groove 86 provided at the lower end of valve 66. Thus, there will be now building up of oil pressure within the system, even when the latter's demands for oil are well below the capacity of the small pump 16.

From the above, it should be understood that the dump and relief valve 25 maintains a constant machine oil pressure within the line 22 at all times regardless of the oil demand and the amount of this pressure is determined by the setting of the springs 58, 58. It should be further understood that the pressure of the oil trapped between the two valves 55 and 66 is, during the operation of the system under load, kept substantially constant, as determined by the setting of springs 58, 58, by virtue of the fact that this oil is constantly relieved through the bore groove 82 and valve groove 63' as the valve 66 raises in response to positive fluctuations in the back pressure of the oil in the line 22.

In other words, once the valve 55 has been raised, under the force of oil from the small capacity pump, so that the valve groove 63' no longer communicates with the chamber or bore groove 62, any slight positive or negative fluctuation in the liquid back pressure above or below the predetermined value will be effective to shift the valve 55 slightly upwardly or downwardly thereby to maintain the liquid pressure within the liquid-operated motor system substantially

at the predetermined value at all times within the range of desired normal flow requirements of the system. The presence of the restricted orifice 65 is effective to maintain the valve 66 in its illustrated or nonflow-diverting position until the back pressure within the system reaches the above noted predetermined value.

From its raised position, whenever the back pressure in the system tends to decrease, due to a greater oil demand, the valve 66 will shift downwardly, thus permitting a greater amount of oil to be delivered from the pumps 15 and 16 to the line 22, but always at the same constant predetermined pressure.

In the event that the auxiliary or clamping cylinder 41 is operated during the operation of the power cylinder 27, it will be clear that the machine oil pressure will be substantially unaffected for the reason that the two valves 55 and 66 will automatically adjust themselves to produce a greater flow of oil into the line 22 in immediate response to any tendency of the pressure to drop within line 22.

As hereinabove described, the present control apparatus has two pumps 15 and 16 associated therewith, but it is to be understood that such an arrangement of valves will also function efficiently if only one pump is connected in circuit therewith. For example, a large capacity pump may be provided with one output line which may be connected to the valve housing inlet 50, and another to the housing inlet 49. By this arrangement, a single pump having, for example, an output of thirteen gallons per minute could be effective to operate the system hereinabove described in substantially the same manner as effected by the two pumps 15 and 16. In this connection, it should be understood that various settings of the springs 58, 58 may be made and the orifice 65 may have its diameter altered in order properly to accommodate the valve for use with different system requirements.

I claim:

1. Control apparatus of the class described for delivering continuously pumped fluid under a predetermined substantially constant pressure to a variable demand hydraulic system, with diversion of the undemanded pumped fluid to the pump supply, said apparatus including a valve casing having two intakes for the pumped fluid, a single outlet by which the pumped fluid is delivered to the system, a return connection through which undemanded pumped fluid is diverted to the pump supply, and a restricted orifice connecting one of said intakes to said outlet, a valve bore in said casing, a piston valve movable in said bore and adapted in one position to pass fluid from the other intake to said outlet and in another position to divert said fluid to said return connection, a second piston valve disposed in said bore in spaced relation to the first valve and maintained normally by spring pressure in a position to connect the space between said two valves with said first mentioned intake, for holding said first valve in non-diverting position, until the system's back pressure prevailing in said space increases to a predetermined value, whereupon said second valve is moved to disconnect said space from said first intake, thereby relieving said space sufficiently for the system's back pressure effective on said first valve's other end, to move same into fluid diverting position.

2. Control apparatus as claimed in claim 1, having one fluid intake of higher capacity than

the other, said higher capacity intake being associated with the first-mentioned fluid diverting valve.

3. Control apparatus as claimed in claim 1, wherein the fluid trapped in the space between the two valves communicates the movement of the first valve to the second valve.

4. Control apparatus as claimed in claim 1, wherein continued movement of the second valve vents the fluid trapped in said space to said other end of the first valve.

5. Control apparatus as claimed in claim 1, wherein continued movement of said first valve vents the fluid effective on its said other end into said return connection.

6. A valve mechanism of the class described comprising a valve casing having two intakes for pumped fluid, a single outlet port for delivery of demanded portions of the pumped fluid, a return connection through which undemanded portions of the pumped fluid are diverted to the pump supply, and a restricted orifice connecting one of said intakes to said outlet, a valve bore in said casing, a piston valve movable in said bore and adapted in one position to pass fluid from the other of said intakes to said outlet and in another position to divert said fluid to said return connection, a second piston valve arranged in said bore in spaced relation to the first valve and maintained normally by spring pressure in a position to connect the space between said two valves with said first mentioned intake, thereby to hold said first valve in non-diverting position until the back pressure of the demanded pumped fluid prevailing in said space increases to a predetermined value, whereupon said second valve moves to disconnect said space

from said first intake, thereby relieving said space sufficiently so that the demanded fluid's back pressure effective on said first valve's other end moves the same into fluid diverting position.

7. A valve mechanism as claimed in claim 6, having one fluid intake of higher capacity than the other, said higher capacity intake being associated with the first mentioned fluid diverting valve.

8. A valve mechanism as claimed in claim 6, wherein the fluid trapped in the space between the two valves communicates the movement of the first valve to the second valve.

9. A valve mechanism as claimed in claim 6, wherein continued movement of the second valve vents the fluid trapped in said space to said other end of the first valve.

10. A valve mechanism as claimed in claim 6, wherein continued movement of the first valve vents the fluid effective on its other end into said return connection.

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