

PATENT SPECIFICATION (11)

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(54) DUAL RANGE PRESSURE DEPENDENT VARIABLE FLOW FLUID DELIVERY SYSTEM

(71) We, CATERPILLAR TRACTOR CO., a Corporation organised and existing under the laws of the State of California, United States of America, of 100 N.E. Adams Street, Peoria, State of Illinois 61602, United States of America, do hereby declare the invention, for which we pray that a patent may be granted to us, and the method by which it is to be performed, to be particularly described in and by the following statement:—

This invention concerns a dual range pressure dependent variable flow fluid delivery system.

Variable displacement pumps are often utilized in current machinery since their flow output can be reduced to a minimum when there is flow requirement in the fluid delivery circuits which they supply while still maintaining a positive pressure in the circuit with minimum horsepower consumption. For example, U.S. Patent 2,835,228 illustrates a system in which such a pump is employed and the pressure level in the fluid delivery circuit is employed to control the displacement of the pump. By utilizing pump output pressure to control the displacement of the pump, system operating pressure is maintained when all the work circuits are inactive with near zero displacement of the pump, as it need only replenish the leakage in the delivery circuit to maintain the system pressure. Under such conditions the forces controlling pump displacement are counterbalanced and a reduction in pressure when a work circuit is operated will cause the pump to increase displacement to meet the increased flow demands. This system, as well as the system illustrated in U.S. Patent No. 3,350,881 is often referred to as a pressure compensated delivery system because the output flow is pressure dependent.

Also, since pump displacement can be varied, such systems are less dependent on engine r.p.m. and offer uniform response in work circuits over wide variation of engine speeds. In addition such systems provide faster response as they can be economically operated at a positive pressure at or near operating pressure.

However, in the above systems, operation in a pressure range close to the maximum pressure that any work circuit requires is necessary and unnecessary horsepower is consumed. This is also true of the pump systems disclosed in our Patent Application No. 27481/73 (Serial No. 1,433,024), and our Patent Application No. 27894/72 (1,333,540) which illustrate a cartridge controlled swash plate axial piston pump, as is generally described herein.

While two level pressure operation for fluid delivery circuits are known in open center type systems, for example, see United States Patent No. 3,416,561, such systems are not available in pressure compensated variable flow delivery systems having a positive pressure.

Thus, it is a general purpose of this invention to provide a pressure compensated variable flow fluid delivery system capable of operating at two separate and distinct pressure levels, with full pressure compensation at both levels.

A dual range pressure compensated flow delivery system according to the invention includes a driven variable displacement pump having its inlet connected to a source of hydraulic fluid and its outlet connected to a fluid delivery circuit with an actuator controlling the displacement of the pump and a pressure responsive control means communicating with the outlet of the pump operable to control the actuator to adjust pump displacement in proportion to pump output pressure over at least a portion of its displacement in a first preselected pressure range and pressure responsive valve means communicating with the fluid delivery circuit which is operable to adjust the control means so it will adjust pump displacement in proportion to pump output pressure over at least a portion of its displacement at a second preselected pressure range, the second preselected pressure range being higher than the first preselected pressure range.

The pressure responsive valve means may be incorporated downstream of closed center control valves in the fluid delivery circuit so

it will not be activated unless a work circuit is actuated which requires a greater pressure in the fluid delivery circuit than is available in the first pressure range where pressure compensation is provided.

A preferred embodiment of the invention will be described with reference to the accompanying drawings in which:—

Fig. 1 is a schematic of a new multi-stage pressure compensated fluid delivery system illustrating some of the components therein in section;

Fig. 2 is a graph illustrating the flow percentage of the pump versus pressure; and

Fig. 3 is a corresponding graph having a horsepower percentage on one ordinate with the same pressure scale on the other ordinate as the graph of Fig. 2.

Referring to Fig. 1, a variable displacement pump 10 is illustrated, which has the conventional components found in an axial-piston type positive displacement pump, along with a special cartridge actuator system 11 composed of a servo cartridge 12 and a slave or compensating cartridge 13. The conventional components of the pump include a drive shaft 14 connected to a prime mover (not shown) driving a cylinder 15 in which are mounted a plurality of longitudinally reciprocating pistons 16. The cylinder is mounted on a valve plate 17 having conventional kidney-shaped ports (one shown in phantom) and the pistons have their projecting ends journaled in separate slippers 18 with a ball and socket attachment. As the slippers rotate with the cylinder, they are supported on a pivoted swish plate 19. When the swash plate is oriented so it is in a radial plane with the cylinder rotating, the pistons will not reciprocate in the cylinder. However, as the swash plate is pivoted (tilted) as shown in Fig. 1, these pistons will reciprocate and draw hydraulic fluid from one kidney-shaped port and discharge it into the other.

From the above description it can be appreciated that the position (tilt) of the swash plate determines the fluid output volume (displacement) of the pump 10. In the embodiment described, the servo-cartridge 12 and the compensating cartridge 13 collectively form the actuator system 11 for the pump. However, the principles of the invention can be applied to actuators exterior of the pump housing and to other positive displacement pumps.

In the present invention it is the object to control the fluid output of the pump in two distinct pressure ranges, i.e., one range being from 0—2100 psi, and the other range from 2050 to approximately 3500 psi. Thus, the remainder of the description will be directed to the components employed to reach this objective, as well as those components which illustrate the environment of the invention.

In the latter respect the outlet 20 of the pump is connected by a conduit 21 to the valve bodies 22 of a series of closed center valves 23 and its inlet 24 is in communication with the reservoir 25, as illustrated by broken line 26. With all the valves closed a small amount of fluid will be lost due to leakage and the pump will have a small output flow to hold the pressure at approximately 2100 psi. A relief valve 27 is connected to conduit 21 and will vent at approximately 3650 psi to protect the fluid delivery system from rupturing due to over pressures.

Each closed center valve 23 is the same and therefore one is described and illustrated with its associated fluid components. Ports 30 of the valve 23 are connected to conduits 31 and 32 that respectively put them in communication with the ports 33 and 34 of a conventional lock valve 35, whose associated ports 37 and 38 respectively communicate with opposite sides of a fluid jack 39 through conduits 40 and 41. The valve prevents loads L and L' on the jack from applying fluid pressure through the referenced conduits to the valve 23 because of the ball check valves 42, which are individually displaced by pressure during valve actuation by a pressure actuated shuttle 43, which allows one side of the jack to exhaust when the opposite side is pressurized by its associated control valve.

With the aforescribed system, the cartridges 12 and 13 are employed to control the volumetric flow (displacement) of the pump proportional to pressure by controlling the swash plate position (tilt). In this regard the fluid output pressure from the pump is used with differentially sized actuating surfaces plug spring loadings in the several cartridges to control the swash plate position through communication with the pump outlet 20 via passages 44 and 45.

In this respect pressurized fluid from passage 45 acts directly on plunger 46 to decrease displacement while this same pressure acts directly on plunger 47 through passage 44 (and valving hereinafter described) which is assisted by springs 48 through spool 49 and pin 47a abutting on the outboard end plunger 47 to increase pump displacement. Spool 49 contains an axial passage therein to vent fluid through the spool to drain as shown in Fig. 1. Assuming a low pressure condition, i.e., below 1700 psi, the swash plate 19 will be in the position illustrated in Fig. 1. As the pressure builds due to full pump output, the force of plunger 46, augmented by the sum of the slipper forces, due to the reaction forces of the pistons 10, will commence to counterbalance spring augmented pressure on plunger 47.

At any pump pressure above atmospheric the two pistons 50 and 50A operate as a 130

pressure reducing valve to maintain a fixed fraction of pump discharge pressure in spring 51 cavity, passages 53 and 70 and chamber 54. Thus, the pump discharge pressure acts on piston 50 area and this force is just balanced by spring 51 force plus the force of reduced pressure in spring 51 cavity acting on end area of piston 50A.

The spring 51 force is relatively small and the fractional ratio of reduced pressure is substantially equal to the area ratios of the pistons 50 and 50A. In the actual circuit the reduced pressure is approximately $\frac{1}{7}$ of pump discharge pressure.

In chamber 54 the reduced pump pressure acts on the enlarged land portion 55 to move the servo valve spool 49 against the bias of the inner spring 48, the movement being proportional to reduced pump pressure after the initial bias of spring 48 is overcome. Thus, at 242 psi reduced pressure (1700 psi discharge pressure) the spool 49 starts to move, at 271 psi reduced pressure (1900 psi discharge) the spool 49 is half way toward zero displacement position, and at 300 psi reduced pressure (2100 psi discharge) the spool 49 is at the zero displacement position.

The operation of the servo spool and swash plate is such that first the servo spool moves in response to an increase (or decrease) in reduced pressure, and then the swashplate and follow-up sleeve move in the same direction a fraction of a second later to stabilize at the new servo position; this is explained more completely as follows: With valve 57 closed as shown in Fig. 1 and the swashplate at some angle the pump discharge pressure in conduits 20, 21, 44, 45 will rapidly rise and simultaneously the reduced pressure in 51, 53, 54 and 70 will also rise and move servo spool 49 against the bias of spring 48. This will close the communication of passage 44 with the chamber end of slug 47 and open a path 56 to reservoir 25 venting pressurized fluid, thereby allowing the pressure in 45 acting on piston 46 to move the swashplate 19 towards the zero displacement position. As swashplate slug 47, and attached piston 47a move towards zero the free end of piston 47A abuts plate 101, which is fixed into follow-up sleeve 49A, and all these parts move against bias of spring 48A in the direction the servo spool 49 moved. When the servo spool 49 stops at any position due to balance of forces (reduced pressure on enlargement 55 equals bias of spring 48) then within a fraction of a second the swashplate 19, slug 47, 47A, 101 and follow up sleeve 49A, catch up to the servo spool, so that: the communication area of passage 44 to slug 47 and the area of path 56 to reservoir will adjust so the pressure in chamber end of 47 is just sufficient to keep the swashplate 19, 47, 47A, 101 and

49A centered on the new servo spool 49 position. As a result, the pump displacement will eventually stabilize, near zero displacement, delivering only sufficient flow to the fluid delivery system to maintain 2100 psi therein when all the control valves 23 are in neutral.

If the illustrated control valve 23 is now actuated by moving its spool 57, the pressure in the fluid delivery conduit 21 would drop, at least initially, causing the pump to increase displacement to meet the additional flow demands of the circuit, since pressurized fluid in conduit 21 passing flow control spool 58, has opened a valve inlet 59 by the action of the fluid pressure on one end of this spool through passage 58a, thereby overcoming spring 69.

Assuming valve spool 57 is shifted to pressurize conduit 32 connected to one of the ports 30, lead L^1 resisting the movement the piston 61 in the jack 39 would be opposed, as shuttle 43 and ball 42 are displaced in opposite direction by the fluid pressure conduit. If the force of load L^1 resisting movement of the piston requires less than 1700 psi in the fluid delivery circuit, the swash plate 19 will adjust for full displacement. Thereafter a decrease in displacement will occur as the pressure increases in the fluid delivery circuits from 1700 psi to approximately 1800 psi due to an increased resisting force of load L^1 . As pump pressure reaches 2050 psi (about 1250 psi in the fluid delivery circuit), due to increased loading of the jack, the dual compensating feature of the invention activates.

This is accomplished by communicating pressure from the pressurized port 30 of valve 23 through passage 62 having a ball check 63 closing a corresponding path to the exhausting port 30, to the chamber 64 containing spring 60 and thence to a compensator valve 65 via a one-way check valve 66 and a conduit 67. All the valves 23 are commonly connected through conduit 67, as illustrated in Fig. 1 and would function in the same manner as now described.

The compensator valve includes a spool 68 biased against the pressure in conduit 67 by a spring 69, which spool controls the fluid communication between the passage 53 in the pump via conduit 70 and one end of a plunger 71 which is abutted on the spring end of spool 49. Thus if the pressure in conduit 67 overcomes the bias on the spool 68 reduced pressure from passage 53 is communicated to the outboard end of the plunger and increases the biasing force on spool 49 so the previously described pump displacement control will now operate in the same manner, but between a pressure range of 2050 and 3600 when either load L or L^1 requires such a pressure range in the fluid delivery circuit. The relief valve 27 opens at

3650 psi to prevent over pressures in the fluid delivery circuit.

By referring to Fig. 2 the operation of the dual compensation is readily apparent. Assuming the control valve 23 is in neutral, the pump operates at 2150 psi (point A) with nominal pump displacement to account for leakage in the circuits. Operating one of the spools 57 will usually result in an initial pressure drop, causing the displacement/pressure curve to move to line B, if the pressure drops to below 170 psi, as indicated by the broken lines C. Thereafter the displacement/pressure profile will move along line B with increasing pressure in the fluid delivery circuit to line D and thence downward toward point A. However, as point A is approached the spool 68 shifts at point F increasing the pump to full displacement at approximately 2050 psi pump pressure along broken line G due to the increased bias provided by plunger 71. Pressure thereafter can increase along broken line H with the pump at full displacement until broken line I is intercepted at which time displacement decreases as pressure increases from 2800 psi to 3600 psi. Thus, through the additional bias on spool 49 by plunger 71, the second range of pressure compensation is automatically achieved when needed in the work circuits connected to the fluid delivery circuit.

The advantage of the invention can be appreciated by reviewing the horsepower (H.P. percentage) pressure envelope of Fig. 3 designated with A¹, C¹, B¹, D¹ and F¹ with the corresponding displacement (flow percentage)/pressure envelope A, C, B, D and F of Fig. 2 along with a comparison of the F¹, G¹, H¹, I¹ envelope with the F, G, H and I envelope of Fig. 2. Obviously when the work requirements can be satisfied at the low pressure compensated range, an economical benefit is realized, with the dual pressure compensated arrangement, since it is not necessary to work at the high pressure compensated range requiring more horsepower. Obviously, the capacity of the pump must be adequate to meet the demands of the several circuits and this is assumed in the above description.

WHAT WE CLAIM IS:—

1. A dual range pressure compensated fluid delivery system giving pressure compensated output flow over two independent pressure ranges comprising:

a variable displacement pump driven by a prime mover and having an inlet connectable to a source of hydraulic fluid and an outlet for delivery of pressurized fluid;

a fluid delivery circuit connected to said outlet having at least one work circuit connected thereto;

an actuator system connected to said pump to control its displacement;

pressure responsive control means operably associated with said actuator system and having communication with said outlet of said pump whereby an output flow of said pump is proportional to its outlet pressure over a part of its displacement within a first preselected pressure range; and

pressure responsive valve means in communication with pressure in said fluid delivery circuit and associated with said pressure responsive control means, said pressure responsive valve means being operable above a preselected pressure to adjust said control means whereby said output flow of said pump is proportional to its outlet pressure over part of its displacement within a second preselected pressure range, said second preselected pressure range being higher than said first preselected pressure range.

2. A dual range pressure compensated fluid delivery system defined in claim 1 wherein the variable displacement pump is of the axial piston type.

3. A dual range pressure compensated fluid delivery system defined in claim 1 or 2 wherein the fluid delivery circuit includes at least one closed center valve between the fluid delivery circuit and at least one work circuit.

4. A fuel range pressure compensated fluid delivery system defined in claim 3 wherein the pressure responsive valve means is in communication with a portion of the fluid delivery circuit downstream of said at least one closed center valve.

5. A dual range pressure compensated fluid delivery system defined in claim 3 or 4 wherein a lock valve is disposed between the fluid delivery circuit and at least one work circuit whereby pressures in a hydraulic motor in said work circuit cannot activate the pressure responsive valve means unless its associated closed center control valve is actuated.

6. The dual range pressure compensated fluid delivery system defined in claim 3, 4, or 5 wherein a plurality of closed center control valve are disposed between the fluid delivery circuit and a corresponding number of work circuits, each control valve operable to actuate a separate work circuit, and common conduit means connecting each of the control valves with the pressure responsive valve means through one way check valves whereby any work circuit activated can operate said pressure responsive valve means.

7. A dual range pressure compensated fluid delivery system defined in claim 1



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3650 psi to prevent over pressures in the fluid delivery circuit.

By referring to Fig. 2 the operation of the dual compensation is readily apparent. Assuming the control valve 23 is in neutral, the pump operates at 2150 psi (point A) with nominal pump displacement to account for leakage in the circuits. Operating one of the spools 57 will usually result in an initial pressure drop, causing the displacement/pressure curve to move to line B, if the pressure drops to below 170 psi, as indicated by the broken lines C. Thereafter the displacement/pressure profile will move along line B with increasing pressure in the fluid delivery circuit to line D and thence downward toward point A. However, as point A is approached the spool 68 shifts at point F increasing the pump to full displacement at approximately 2050 psi pump pressure along broken line G due to the increased bias provided by plunger 71. Pressure thereafter can increase along broken line H with the pump at full displacement until broken line H with the pump at full displacement until broken line I is intercepted at which time displacement decreases as pressure increases from 2800 psi to 3600 psi. Thus, through the additional bias on spool 49 by plunger 71, the second range of pressure compensation is automatically achieved when needed in the work circuits connected to the fluid delivery circuit.

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a fluid delivery circuit connected to said outlet having at least one work circuit connected thereto;

an actuator system connected to said pump to control its displacement;

pressure responsive control means operably associated with said actuator system and having communication with said outlet of said pump whereby an output flow of said pump is proportional to its outlet pressure over a part of its displacement within a first preselected pressure range; and

pressure responsive valve means in communication with pressure in said fluid delivery circuit and associated with said pressure responsive control means, said pressure responsive valve means being operable above a preselected pressure to adjust said control means whereby said output flow of said pump is proportional to its outlet pressure over part of its displacement within a second preselected pressure range, said second preselected pressure range being higher than said first preselected pressure range.

2. A dual range pressure compensated fluid delivery system defined in claim 1 wherein the variable displacement pump is of the axial piston type.

3. A dual range pressure compensated fluid delivery system defined in claim 1 or 2 wherein the fluid delivery circuit includes at least one closed center valve between the fluid delivery circuit and at least one work circuit.

4. A fuel range pressure compensated fluid delivery system defined in claim 3 wherein the pressure responsive valve means is in communication with a portion of the fluid delivery circuit downstream of said at least one closed center valve.

5. A dual range pressure compensated fluid delivery system defined in claim 3 or 4 wherein a lock valve is disposed between the fluid delivery circuit and at least one work circuit whereby pressures in a hydraulic motor in said work circuit cannot activate the pressure responsive valve means unless its associated closed center control valve is actuated.

6. The dual range pressure compensated fluid delivery system defined in claim 3, 4, or 5 wherein a plurality of closed center control valve are disposed between the fluid delivery circuit and a corresponding number of work circuits, each control valve operable to actuate a separate work circuit, and common conduit means connecting each of the control valves with the pressure responsive valve means through one way check valves whereby any work circuit activated can operate said pressure responsive valve means.

7. A dual range pressure compensated fluid delivery system defined in claim 1



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5 wherein the actuator system includes two pressure responsive members, one of said pressure responsive members being in communication with pump output pressure and
10 having biasing means associated therewith and wherein the pressure responsive control means includes a servo-valve means associated with one of said pressure responsive members operable to control the actuator system as a function of said pump output pressure.

15 8. A dual range pressure compensated fluid delivery system defined in claim 7 wherein the pressure responsive valve means activates additional biasing means on the servo-valve means thereby causing the pump

to operate within the second preselected pressure range.

9. A dual range pressure compensated fluid system defined in claim 8 wherein the additional biasing means includes a reacting surface operated by pump pressure. 20

10. A dual range pressure compensated fluid delivery system substantially as described herein with reference to the accompanying drawings. 25

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