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[54] **HYDRAULIC POWER CIRCUITS EMPLOYING REMOTELY CONTROLLED DIRECTIONAL CONTROL VALVES**
5 Claims, 1 Drawing Fig.

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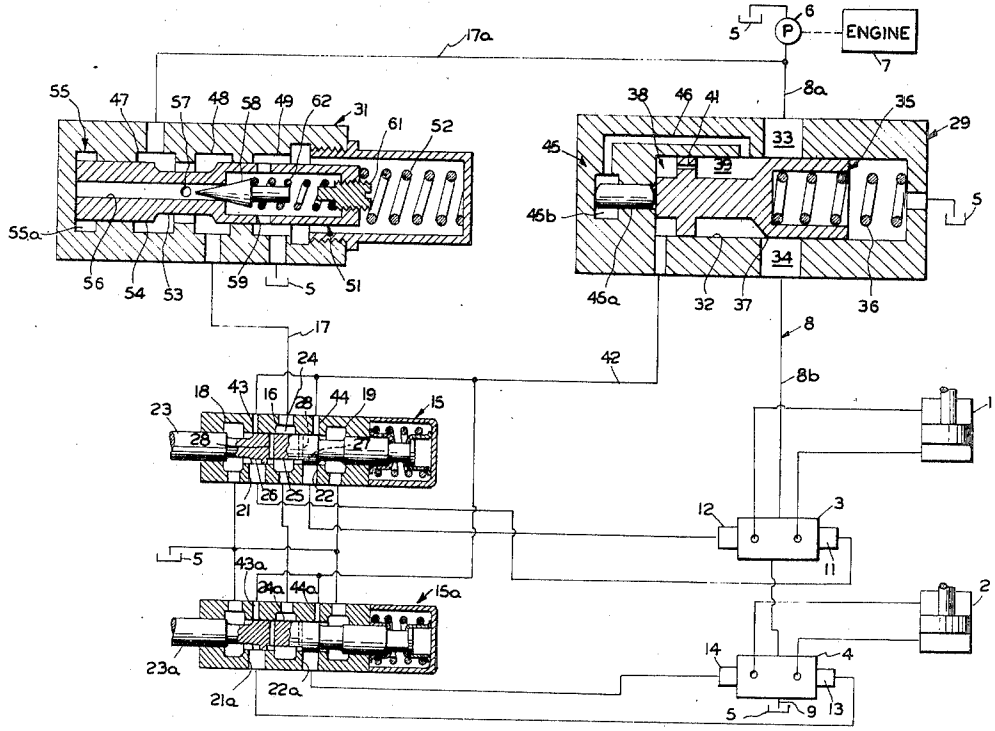
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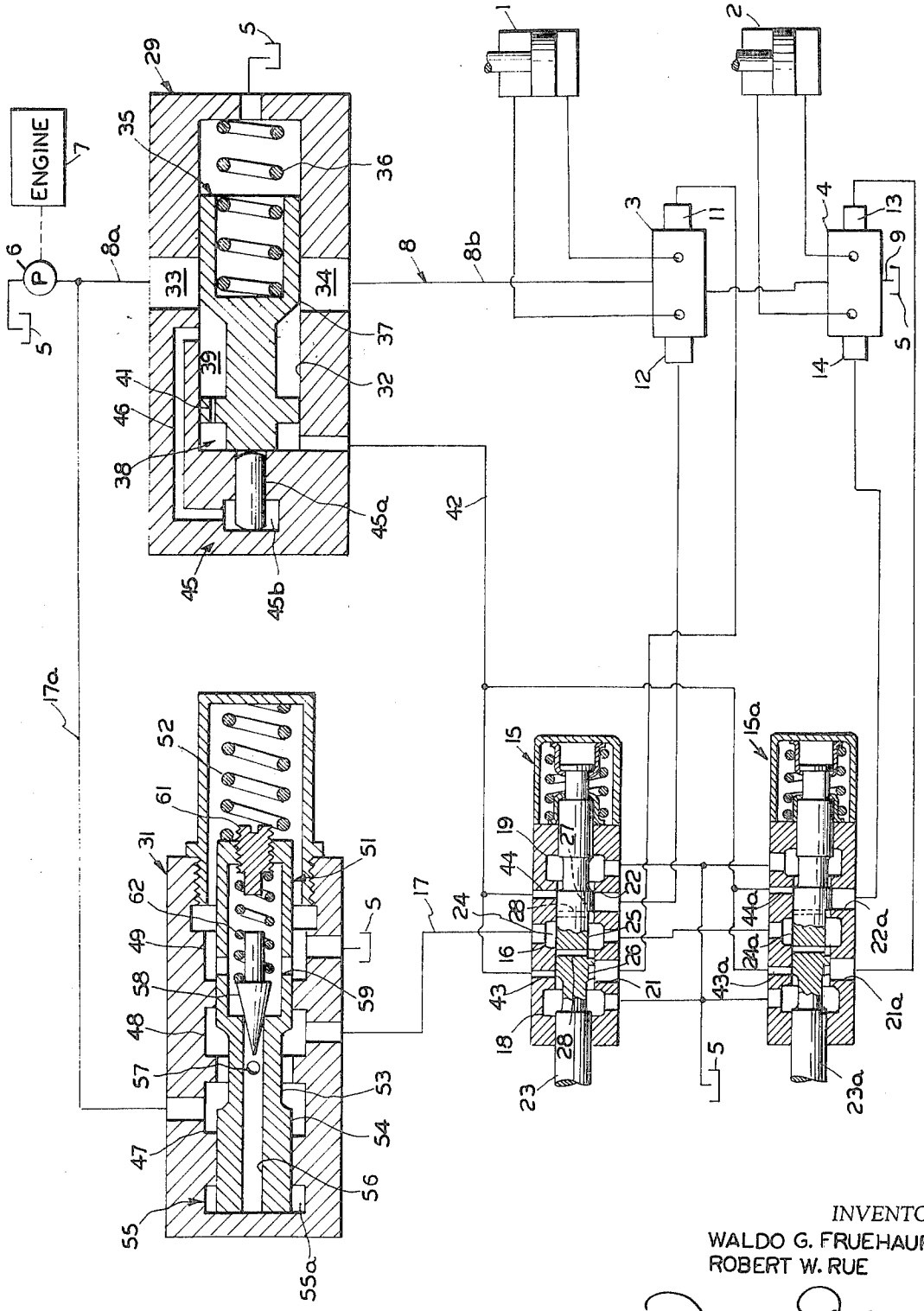
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ABSTRACT: Hydraulic power circuit in which each actuating cylinder is controlled by an open center, pilot-operated directional control valve which is operated by a low-pressure remote control system utilizing fluid taken from the same pump which supplies motive fluid for the cylinders. The fluid which is supplied to the pilot valves of the remote control system at a reduced pressure is drawn from the supply path for the directional control valves at a point upstream of a throttle valve. This valve is automatically controlled so that it assumes a low flow-restricting position when the pilot valves, and consequently the directional control valves, are in neutral position, and assumes a high flow-restricting position whenever one of the pilot valves is shifted away from neutral position. An override control, which responds to supply pressure, shifts the throttle valve to low flow-restricting position during actuation of a cylinder.





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HYDRAULIC POWER CIRCUITS EMPLOYING REMOTELY CONTROLLED DIRECTIONAL CONTROL VALVES

BACKGROUND AND SUMMARY OF THE INVENTION

Hydraulic power circuits employed on large construction and materials-handling vehicles, such as front end loaders, use high-capacity, open-center directional control valves which require large actuating forces. Therefore, from the standpoint of reducing operator fatigue, it is desirable that these valves be actuated through a low-pressure remote control system. The conventional remote control system includes a pair of piloted pressure motors for shifting each directional control valve, and a small, manually operated, pressure-graduating pilot valve for each set of motors which serves to furnish them with pilot fluid at a selectively variable pressure. The pilot valves are supplied with operating fluid at a substantially constant, low pressure by a source including a relief valve and a small pilot pump separate from the main pump which delivers motive fluid to the cylinders controlled by the directional control valves. While this type of remote control system performs satisfactorily, the additional pumping unit it requires is a source of inconvenience and expense to the vehicle manufacturer.

The object of this invention is to provide an improved power circuit of the kind just mentioned in which the operating fluid required by the remote control system is supplied by the main power pump. According to this invention, the supply conduit leading from the main pump to the directional control valves is equipped with a throttle valve, and the oil needed by the remote controls is taken from this path at a point upstream of that valve and is fed to the pilot valves through a pressure reducer. The throttle valve is controlled automatically by the pilot valves so that when all these valves are in neutral position, and consequently the open center path through the directional control valves is open, the throttle valve assumes a low flow-restricting position. As a result, the neutral pressure losses of the circuit are minimized. On the other hand, when any pilot valve is actuated, the throttle valve immediately assumes a high flow-restricting position and creates in the supply path a back pressure sufficient to permit proper operation of the remote controls.

Once the actuated directional control valve in the new circuit shifts away from its neutral position and begins to throttle the open center path and deliver fluid to a cylinder, the pump and the entire supply path will be subjected to the full load pressure imposed by the cylinder. Since this pressure is considerably higher than that required by the remote control system, the flow-restricting action of the throttle valve becomes superfluous and, in fact, constitutes a waste of energy. Accordingly, the invention also includes an override control with forces the throttle valve back to its low flow-restricting position as soon as the load pressure exceeds the requirement of the remote controls. This measure, of course, minimizes pressure losses during the cylinder actuation cycle.

BRIEF DESCRIPTION OF THE DRAWING

The preferred embodiment of the invention is described herein with reference to the accompanying drawing whose single FIGURE is a schematic diagram of the improved circuit.

DESCRIPTION OF PREFERRED EMBODIMENT

As shown in the drawing, the improved circuit is employed to actuate a pair of double-acting power cylinders 1 and 2 which may represent the tilt and lift cylinders, respectively, on a front end loader. The circuit includes a pair of directional control valves 3 and 4, a fluid reservoir or tank 5, and a fixed delivery pump 6 which is driven by the vehicle's engine 7 and is arranged to deliver oil to valves 3 and 4 through a supply conduit 8. Valves 3 and 4 are conventional open center, sliding plunger units and are connected in a tandem (i.e., series-parallel) flow circuit. Each valve has a neutral or hold position in which it hydraulically locks the associated cylinder and

completes an open center path leading from conduit 8 to return conduit 9, and a pair of actuating positions in each of which it closes the open center path and connects either the head or the rod end of the associated cylinder with supply conduit 8 while connecting the remaining end with tank 5. In addition, lift valve 4 has a float position in which it unloads pump 6 and opens a regeneration path between the opposite ends of cylinder 2. Although the valves 3 and 4 are shown as separate units, it will be understood that they normally are embodied in a common housing.

The plungers of directional control valves 3 and 4 are spring biased to their neutral positions and are shifted to their various actuating positions by a pair of opposed piloted pressure motors 11 and 12 or 13 and 14 which are controlled by a set of manually operated, pressure-graduating pilot valves 15 and 15*a*. Pilot valve 15 includes an inlet chamber 16 which is in constant communication with pilot supply conduit 17, a pair of exhaust chambers 18 and 19 which communicate with tank 5, and a pair of motor ports 21 and 22 which are connected with motors 11 and 12, respectively. The output pressures at ports 21 and 22 are controlled by a sliding valve spool 23 having a main land 24 containing a centrally located stopped flat 25 and a pair of axially aligned throttling notches 26 and 27. Spool 23 is keyed against rotation so that the flat 25 and the notches 26 and 27 are aligned with ports 21 and 22, and is equipped with a pair of transverse passages 28 which serve to balance the radially directed pressure forces. When spool 23 is in the illustrated neutral position, land 24 isolates inlet chamber 16 from the motor ports and other chambers, and notches 26 and 27 connect the motor ports 21 and 22 with exhaust chambers 18 and 19, respectively. Therefore, in this position both of the piloted motors 11 and 12 are vented, and directional control valve 3 assumes its neutral position. When spool 23 is shifted to the left from neutral position, port 22 remains vented, but communication between port 21 and chamber 18 through notch 26 is gradually restricted, and this port is connected with chamber 16 through a gradually opening support path across the peripheral spool edge at the left end of flat 25. As a result, the pressure at port 21 rises, and piloted motor 11 shifts valve 3 to one of its actuating positions. In a similar manner, rightward movement of spool 23 leaves port 21 vented, and gradually increases the pressure at port 22. This action, of course, causes piloted motor 12 to shift valve 3 to its other actuating position.

Pilot valve 15*a* is substantially identical to, and is connected in a parallel flow circuit with, valve 15. Thus, its spool 23*a* has a neutral position, in which ports 21*a* and 22*a* are vented and the associated directional control valve 4 assumes its neutral position, and is shifted in opposite directions from this position to raise the pressure at one of the motor ports and thereby cause piloted motor 13 or 14 to shift valve 4 to an actuating position. Since the magnitude of the pilot pressure depends upon the displacement of spool 23*a* from neutral position and determines how far valve 4 is moved from its neutral position, it follows that the operator can select either the power down or the float position for valve 4 by merely controlling the shift of spool 23*a* in one direction. In order to guard against inadvertent selection of float position, it is the usual practice to include in directional control valve 4 an extra biasing spring which is effective to supplement the centering spring as the valve plunger moves from power down to float position. Inclusion of the extra spring insures that valve 4 will not move to float position unless spool 23*a* is shifted a substantial distance beyond the position in which it causes valve 4 to assume the power down position.

The operating fluid required by the remote control system is tapped from a section 8*a* of main supply conduit 8 located upstream of a throttle valve 29 and is delivered to pilot supply conduit 17 through a branch conduit 17*a* and a pressure reducing valve 31. Throttle valve 29 includes a valve bore 32 which is intersected by two diametrically opposed ports 33 and 34 which are joined, respectively, to the sections 8*a* and 8*b* of main supply conduit 8, and which contains a reciproca-

ble metering spool 35. This spool is biased by compression spring 36 to the illustrated high flow-restricting position, wherein the peripheral edge 37 of the spool restricts to a substantial degree flow from port 33 to port 34 through bore 32, and is shifted by an actuating motor 38 to a low flow-restricting position in which ports 33 and 34 are in substantially free communication with one another. Actuating motor 38 is in constant communication with the space 39 within bore 32 via restricted passage 41, and thus always receives fluid at a pressure which varies with the position of spool 35, and is selectively connected to tank 5 via a common exhaust conduit 42 and any one of four vent ports 43, 44, 43*a*, 44*a* in the pilot valves 15 and 15*a*. Ports 43 and 44 cooperate with valve land 24 to define a vent valve which is closed when spool 23 is in neutral position and which opens as soon as the spool is shifted away from neutral position in either direction. Ports 43*a* and 44*a* cooperate with land 24*a* to define a similar vent valve which is opened and closed by movement of spool 23*a*. The restriction to flow through each of the ports 43, 44, 43*a*, and 44*a* is substantially less than the restriction afforded by passage 41; therefore, when either vent valve is open, the pressure in actuating motor 38 is dissipated and spring 36 is enabled to shift metering spool 35 to the high flow-restricting position. On the other hand, when both vent valves are closed, motor 38 will be subjected to the back pressure in space 39 and will shift spool 35 to the low flow-restricting position. The effective area of motor 38 (i.e., the cross-sectional area of bore 32) is so selected that the motor will be able to hold spool 35 in the low flow-restricting position at the pressure which prevails in space 39 when the spool is in this position and the directional control valves 3 and 4 are in their neutral positions.

Throttle valve 29 also includes a fluid pressure override motor 45 comprising a small piston 45*a* which bears against the left end of metering spool 35, and a working space 45*b* which is in continuous communication with bore space 39 via passage 46. As in the case of actuating motor 38, override motor 45 is arranged to shift metering spool 35 to the low flow-restricting position; however, since motor 45 has a much smaller effective area (i.e., the diameter of piston 45*a* is smaller than the diameter of bore 32), it requires a higher pressure in order to move the spool to that position. In particular, motor 45 will be ineffective to shift spool 35 to the low flow-restricting position until the pressure in space 39 rises above the level which prevails when spool 35 is in the high flow-restricting position and directional control valves 3 and 4 are in their neutral positions.

When one of the cylinders 1 and 2 is being actuated, the pressure in section 8*a* of supply conduit 8 will depend upon the load imposed on the cylinder as well as upon the position of metering spool 35. Since the load pressure varies and normally is much higher than the operating pressure of the remote control system, it is necessary to reduce the pressure of the fluid fed to pilot supply conduit 17. This function is performed by valve 31. This valve includes inlet, outlet and exhaust chambers 47, 48 and 49 which are connected, respectively, with conduits 17*a* and 17 and tank 5, and a reciprocable metering spool 51. Spool 51 is biased by spring 52 to the illustrated fully open position, in which its peripheral groove 53 interconnects chambers 47 and 48, and is shifted to the right to cause land 54 to progressively close communication between these chambers by a pressure motor 55. The working space 55*a* of this motor communicates with outlet chamber 48 via axial and radial passages 56 and 57 formed in spool 51; therefore under dynamic conditions, i.e., when oil is flowing through valve 31 to conduit 17, motor 55 will cause spool 51 to throttle this flow and limit the pressure in outlet chamber 48 to a regulated valve determined by the preload in spring 52. This throttling action of spool 51 obviously is not effective to limit the pressure in conduit 17 under static conditions, so, if the load pressure in conduit 17*a* is high at a time when the flow demand of the remote control system is zero, leakage across land 54 could raise the pressure in pilot supply conduit 17 to an excessive level. In order to guard against this, valve 31 is provided with a relief path leading from outlet chamber 48 to tank 5 and including radial and axial passages 57 and 56, respectively, relief valve 58, radial passages 59 and exhaust chamber 49. The cracking pressure of valve 58 depends upon the position of the seat 61 of its biasing spring 62 and is set considerably higher than the regulated output pressure level established by spool spring 52. Thus, inclusion of this valve within spool 51 will have no adverse effect upon the throttling action afforded by the spool.

In a representative application of the improved circuit, spring 52 of pressure reducer 31 is selected to provide the remote control system with a regulated pressure on the order of 200 p.s.i., relief valve 58 is set for a cracking pressure of about 500 p.s.i., and throttle valve 29 is designed so that the pressure drop between ports 33 and 34 is 50 p.s.i. or 200 p.s.i. depending upon whether spool 35 is in its low or high flow-restricting position. When the spools 23 and 23*a* of the two pilot valves 15 and 15*a* are in their neutral positions, piloted motors 11–14 will be vented and the vent valves 24, 43, 44 and 24*a*, 43*a*, 44*a* will be closed. Therefore, directional control valves 3 and 4 will assume their neutral positions, in which pump 6 is unloaded to tank 5, and actuating motor 38 in throttle valve 29 will be subjected to the pressure in bore space 39. As a result, motor 38 will shift metering spool 35 to the right to its low flow-restricting position and thereby minimize the neutral pressure losses of the circuit. Under this condition, the opposite ends of piston 45*a* are subjected to equal pressures, so override motor 45 has no effect upon the position of metering spool 35.

When the operator shifts pilot spool 23 or 23*a* away from its neutral position in order to actuate one of the cylinders 1 and 2, vent valve 24, 43, 44 or 24*a*, 43*a*, 44*a* immediately opens and the pressure in actuating motor 38 dissipates. Therefore, spring 36 now moves metering spool 35 to the illustrated high flow-restricting position and creates a back pressure in conduit section 8*a* which is 200 p.s.i. higher than the pressure in section 8*b*. Thus, regulator 31 will transmit to pilot supply conduit 17 fluid at the 200 p.s.i. regulated pressure needed for proper operation of the remote control system. After the actuated pilot spool 23 or 23*a* opens the associated vent valve, and thereby sets throttle valve 29 to its high flow-restricting position, it commences to raise the pressure at one or the other of its motor ports 21, 22 or 21*a*, 22*a*. As a result, the piloted motor 11, 12, 13 or 14 connected with this port begins to move the associated directional control valve 3 or 4 out of its neutral position. This action throttles the open center path and establishes supply and exhaust paths for the opposite ends of the associated cylinder 1 or 2. Inasmuch as the pressure of the pilot fluid supplied to valves 15 and 15*a* is raised to the required 200 p.s.i. level by the initial movement of the actuated spool 23 or 23*a*, and not as a result of the throttling action of the directional control valve, it will be apparent that the operator will have full control over the position of the directional control valve, and can cause it to meter flow to and from the active cylinder 1 or 2, throughout the actuating cycle. Thus, the improved circuit affords the same degree of control action as the conventional circuit which employs a separate pilot pump.

Although venting of actuating motor 38 inherently relieves the pressure acting on the right end of piston 45*a* and thereby renders override motor 45 effective to exert a shifting force on spool 35, this force will be less than that developed by spring 36 during the initial phase of an actuating cycle. Therefore, override motor 45 will not preclude spool 35 from moving to and remaining initially in the high flow-restricting position.

As the actuated directional control valve 3 or 4 shifts away from neutral position and begins to throttle the open center path, it produces an increasing back pressure in conduit 8. When the back pressure in conduit section 8*b* reaches 200 p.s.i., the pressure in bore space 39 and in conduit section 8*a* will be even higher; consequently, the throttling action of

valve 29 becomes superfluous as far as the pressure requirement of the remote control system is concerned. Therefore, at this time, override motor 45 will shift spool 35 to its low flow-restricting position. Since the pressure at which this action occurs normally is much lower than that required to move the cylinders 1 and 2, it follows that the energy losses attributable to valve 29 during actual movement of a cylinder are kept to a minimum.

When the operator returns the actuated pilot spool 23 or 23*a* to neutral position, it closes the vent valve 24, 43, 44 or 24*a*, 43*a*, 44*a* which had been open and causes the associated directional control valve 1 or 2 to return to its neutral position. Closure of the vent valve raises the pressure in motor 38 to the level prevailing in bore space 39, thereby disabling motor 45 and rendering motor 38 effective to hold metering spool 35 in its low flow-restricting position.

We claim:

1. In a hydraulic power circuit including an actuating motor (1); directional control valve means (3) connected with the actuating motor (1), a supply pump (6) and a fluid reservoir (5) and adapted to control flow from the pump to the motor and flow from the motor to the reservoir, the valve means having a neutral, open center condition and being actuated by fluid pressure motor means (11, 12); and a pilot valve (15) for supplying variable pilot pressures to said fluid pressure motor means and having a neutral position in which it causes that motor means to set the directional control valve means to neutral condition, the improvement which comprises

a. throttle valve means (35) interposed in the supply connection (8) between the pump (6) and the directional control valve means (3) and affording high and low flow-restricting conditions;

b. actuating means (36, 38, 41-44) responsive to movement of the pilot valve (15) for setting the throttle valve means (35) to the low flow-restricting condition when the pilot valve is in neutral position and for setting the throttle valve means to the high flow-restricting condition when the pilot valve is shifted away from neutral position;

c. override means (45, 46) responsive to the pressure in said supply connection (8) for overriding the actuating means and setting the throttle valve means (35) to the low flow-restricting position when the pressure exceeds a predetermined value; and

d. means (17, 17*a*), including a pressure reducer (31), for delivering fluid under pressure to the pilot valve (15) from a point (8*a*) in said supply connection (8) at which the pressure varies with the condition of the throttle valve means 35.

2. The improved hydraulic power circuit defined in claim 1 in which the actuating means comprises

a. means (36) biasing the throttle valve means (35) to the high flow-restricting condition;

b. a valve-actuating motor (38) adapted to shift the throttle valve means (35) to the low flow-restricting condition;

c. a restricted flow passage (41) connecting the valve-actuating motor (38) with a point (39) in the supply connection (8) at which the pressure varies with the condition of the throttle valve means (35);

d. an exhaust passage (42) leading from the valve-actuating motor (38) to the reservoir (5); and

e. a vent valve (24, 43, 44) operated in unison with the pilot valve (15) and serving to close the exhaust passage (42) when the pilot valve is in neutral position and to open that passage when the pilot valve is shifted away from neutral position.

3. The improved hydraulic circuit defined in claim 2 in which the override means comprises

a. a second valve-actuating motor (45) arranged to shift the throttle valve means (35) toward the low flow-restricting condition; and

b. means (46) connecting the second valve-actuating motor (45) with a point (39) in the supply connection (8) at which the pressure changes with the condition of the throttle valve means (35).

4. The improved hydraulic circuit defined in claim 3 in which

a. the valve-actuating motors (38, 45) are connected with the same point (39) in the supply connection (8); and

b. the effective area of the second valve-actuating motor (45) is smaller than the effective area of the first such motor (38).

5. The improved hydraulic circuit defined in claim 1 in which the pressure reducer (31) includes

a. a regulating valve having an inlet chamber (47) connected with said point (8*a*) in the supply connection (8), an outlet chamber (48) connected with the pilot valve (15), and means (51, 52, 55) responsive to the pressure in the outlet chamber (48) for throttling flow from the inlet chamber to the outlet chamber as required to limit that pressure to a predetermined level; and

b. a relief valve (58) interposed in a flow path (49, 56, 57, 59) interconnecting the outlet chamber (48) and the reservoir (5) and adapted to limit the pressure in the outlet chamber to a level higher than said predetermined level.

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