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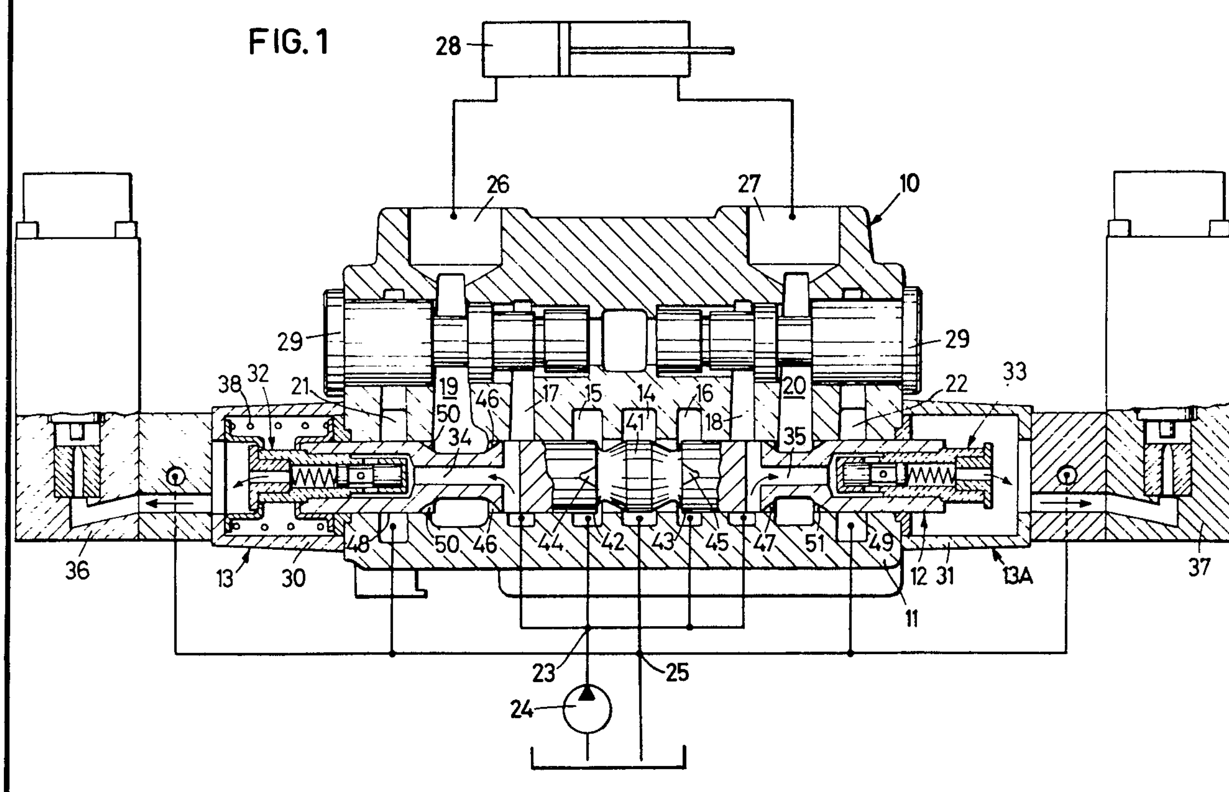
(54) Hydraulic directional control valve

(57) In a hydraulically operated directional control valve of open-centre type the spool (12) is remotely operated electro-hydraulically by valves (36,37) controlling flow of pressurized liquid from a pump connection port (23) to drain. The pressurized liquid acting in chambers (13,13A) on the ends of the spool. In order that an adequate operating pressure may always be available, the spool (12) is adapted upon movement away from the neutral position to restrict the flow path between the pump connection port (23) and the tank connection port (25) rapidly enough to ensure that the pressure produced by the pump increases to the value required for a firmly controlled displacement of the spool (12) throughout the displacement range before the opening of the flow path between the pump connection port (23) and the load connection port (26, 27) begins.

FIG. 1



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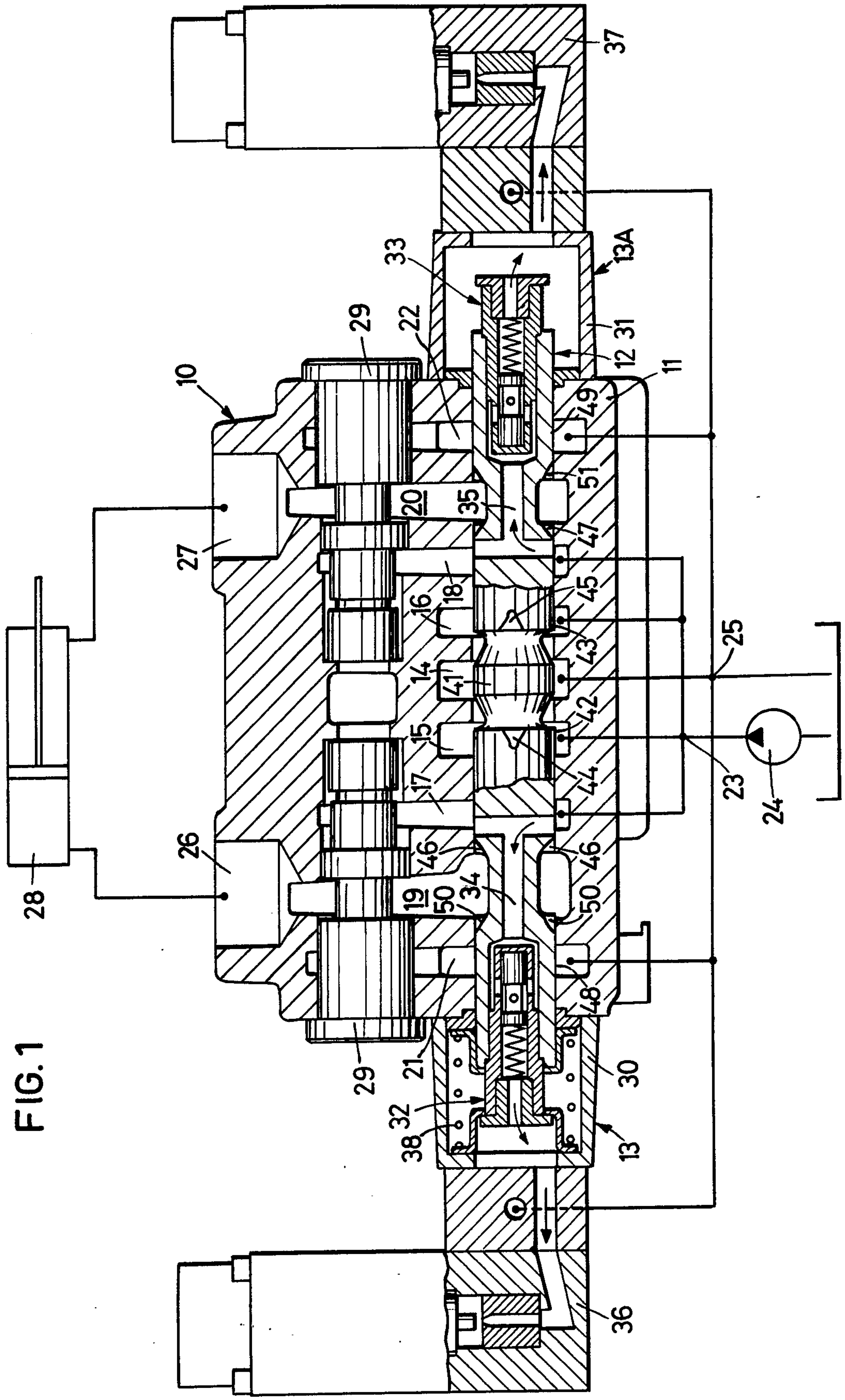
10
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13A
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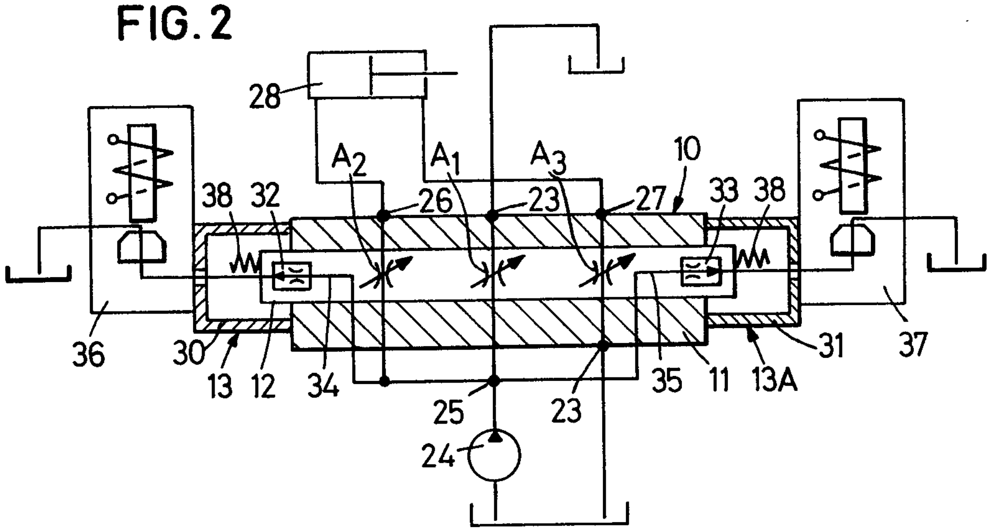
FIG.1

FIG. 2


28
A2
A1
A3
10
38
32
26
23
27
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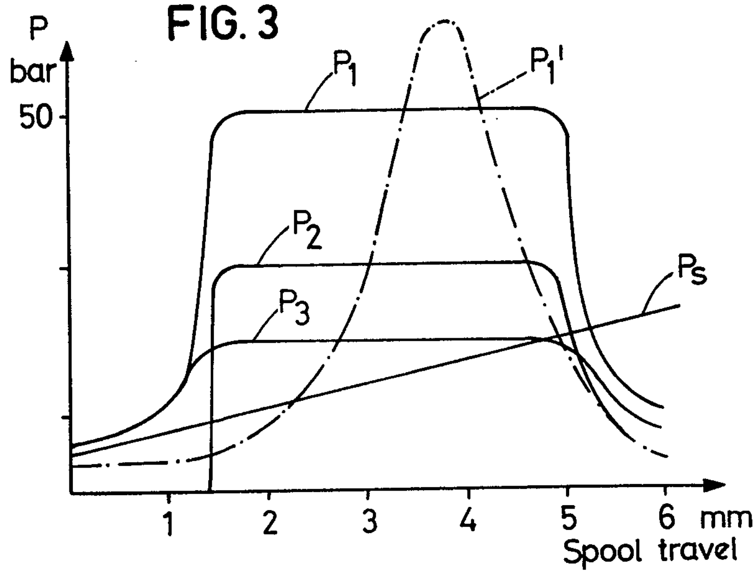
38
36
30
13
12
34
25
23
35
11
13A
31
37
24

FIG. 3


P
bar
50
P1
P1'
P2
P3
Ps
1
2
3
4
5
6
mm
Spool travel

SPECIFICATION

Hydraulic directional control valve

This invention relates to a hydraulic directional control valve of the kind comprising a valve housing having a pump connection port, a load connection port and a tank connection port, and a valve spool provided in the valve housing and operable by a hydrostatic actuator for controlling the fluid flow paths between the connection ports, the valve spool being displaceable between a neutral position in which the pump connection port is in substantially unrestricted fluid flow communication with the tank connection port and a working position in which the pump connection port is in fluid flow communication with the load connection port.

In hydraulic systems in which a directional control valve of this kind, that is, a so-called open-center valve, is used to control the flow of hydraulic liquid between the pump and the load, the pump supplies the hydraulic liquid at a very low pressure, e.g. 5 bar (1 bar = 10^5 Pa), when the system is idling. This low pressure is insufficient to enable the actuator to displace the valve spool from the neutral or centre position to the working position in a reliable manner; only after the valve spool has travelled a certain distance and adequately restricted the flow path for the liquid—the flow path is substantially unrestricted in the neutral position—has the pressure produced by the pump, and thus the pressure available to the actuator, increased sufficiently to provide for a reliable displacement of the spool throughout the spool travel range.

It has been common practice, therefore, to provide the system with a separate pump (pilot pump) in order to ensure that the valve actuator pressure is adequate, regardless of the position of the spool. A separate valve actuator pump unavoidably increases the cost, however, and also leads to certain other disadvantages. In accordance with the present invention there is provided, in a hydraulic control valve of the kind described above, a pressure-compensated flow control valve, the inlet of which is connected with the pump connection port of the valve housing and the outlet of which is connected to the actuator, the spool being adapted during its travel from the neutral position towards the working position to restrict the flow path between the pump connection port and the tank connection port, before it opens the flow path between the pump connection port and the load connection port, to such an extent that the pressure drop between the pump connection port and the tank connection port increases to a value at least equal to the hydrostatic pressure that is required in the actuator for displacing the valve spool throughout the spool travel range.

The hydraulic liquid required for the spool actuator is thus taken by way of a pressure-compensated flow control valve from the pump feeding the load. A flow corresponding to the setting of the pressure-compensated flow control valve therefore is continuously branched off from the flow supplied by the pump, but this branched-off flow need not be more than a small fraction of the total flow supplied by the pump. Under idling conditions, the branched-off flow of course is also at a very low pressure, e.g. 5 bar.

Even if the spool is carefully balanced, this low pressure is not adequate to enable the actuator to displace the spool throughout the spool travel range; it only suffices to displace the spool a very small distance from the neutral position.

In conventional directional control valves, this distance is too short for the pressure drop over the valve to increase sufficiently before the resistance to displacement of the spool prevails over the pressure produced by the pump.

In the directional control valve according to the invention, however, the spool is adapted during its movement from the neutral position towards the working position to restrict the flow path between the pump connection port and the tank connection port sufficiently rapidly to cause the pressure drop across the valve resulting from the displacement of the spool to rise to the value required for a reliable displacement of the spool before the spool commences opening the flow path between the pump connection port and the load connection port. In other words, the directional control valve according to the invention is constructed such that the pressure drop across it increases much more steeply as the spool travels than in the conventional directional control valves, namely, over the first portion of the travel range of the spool.

In order that the energy losses resulting from the pressure drop across the directional control valve may be kept at a minimum, the pressure drop, once the pressure required for a reliable displacement of the spool has been reached, should remain substantially constant upon continued travel of the spool away from the neutral position.

Once it has been established how the pressure drop curve, that is, the line constituting a graphical representation of the relationship of the pressure drop across the directional control valve to the travel of the spool from the neutral position, should run in order that the requirements set forth above may be satisfied, the realization of the pressure drop curve causes no fundamental problems, although it is to be expected that the detailed final design of the parts of the spool and the valve housing influencing the pressure drop curve has to be determined at least partially by triai and error.

If the valve is of the sliding spool type, a particularly simple and appropriate design may be achieved if the pressure-compensated flow control device is provided in the spool proper adjacent one end thereof.

A more detailed description of the invention follows hereinafter, reference being had to the accompanying drawings showing an exemplary embodiment of the valve according to the invention, in which:

Figure 1 is a longitudinal sectional view of a directional control valve constructed in accordance with the invention, the valve being for a load which is here assumed to be a double-acting hydraulic cylinder;

Figure 2 is a diagram serving to illustrate the operation of the directional control valve shown in Fig. 1;

Figure 3 is a graphical representation of the relationship of, on the one hand, the pressure drops across various parts of the directional control valve and the pressure required to displace the spool and, on the other hand, the travel of the spool from the neutral position.

The directional control valve 10 shown in Fig. 1 is a four-way spring-centered proportional sliding-spool valve the spool movements of which are controlled electrohydraulically in both directions. For the sake of simplicity, it is assumed here that the valve has only a single spool.

The cast valve housing 11 thus contains an axially displaceable valve spool 12 the ends of which are connected to or forms the pistons of a pair of single-acting hydraulic actuators operating in opposite directions. These actuators are generally designated 13 and 13A, respectively, and may be of a design known per se.

Within the bore containing the valve spool 12 there are nine recesses formed by annular grooves 14, 15, 16, 17, 18 19, 20, 21 and 22. The grooves 15–22 are arranged in two groups and disposed substantially symmetrically on either side of the central groove 14. Thus, for each groove on one side of the groove 14 there is a corresponding groove on the other side.

The central groove 14 is in constant communication with the liquid reservoir or tank of the hydraulic system and accordingly is always virtually unpressurised. The same applies to the two outermost grooves 21 and 22. The connection port of the housing 11 through which the grooves 14, 21, 22 communicate with the tank is symbolically represented by a dot 23. The grooves 15, 16, 17, 18 are in constant and direct communication with the symbolically illustrated pump 24 by way of a pump connection port 25 provided in the housing and symbolically represented by a dot, and these grooves thus are always at the pressure produced by the pump. The two remaining grooves 19 and 20 communicate by way of load connection ports 26 and 27 with respective sides of the double-acting cylinder 28 constituting the load in this case.

A bore extending through the valve housing 11 in parallel with the bore for the valve spool 12 houses valve cartridges 29. These have no immediate bearing on the invention and therefore will not be described in greater detail. It should be noted, however, that they leave an unobstructed flow passage for the hydraulic liquid between the grooves 19, 20 and the load connection ports 27, 27.

The two single-acting hydraulic actuators 13 and 13A are similar to one another and mounted on opposite sides of the valve housing 11 so as to coact with respective ends of the spool 12. Each actuator 13, 13A comprises a cap 30, 31 enclosing the adjacent end of the spool 12 projecting from the valve housing 11. This end forms the piston of the actuator, and the pressure prevailing within the cap thus tends to displace the spool 12 away from the actuator.

Hydraulic liquid is continuously fed to each cap 30, 31 through the adjacent spool end, namely, through a pressure-compensated flow control valve (a regulator maintaining a constant volumetric flow rate) 32, 33 the inlet of which is in open communication with the valve housing groove 17, 18 through a spool passage 34, 35 when the spool is in the neutral position. The hydraulic liquid continuously flows from the cap 30, 31 and through a solenoid-operated pressure control valve 36, 37 of a type well known per se. By means of the pressure control valve a continuously variable pressure within the cap 30, 31, and thus a continuously variable force acting on the valve spool 12, may be produced. The valve spool is constantly biassed toward the illustrated neutral position by a spring 38, and to the extent that the hydraulic force acting on the end of the spool prevails over the force of the spring and any other forces coacting with the spring, it results in displacement of the spool. The solenoid-operated pressure control valve 36, 37 may of course be replaced by a manually operated restriction or by some other device by means of which a variable hydraulic pressure within the cap 30, 31 may be produced.

As long as the system is idling, that is as long as the spool is in the illustrated neutral position and the pressure control valves 36 and 37 allow the hydraulic liquid to pass to the tank with an insignificant pressure drop, the pump 24 operates at a very low pressure, e.g. 5 bar. The main portion of the stream of hydraulic liquid produced by the pump then returns directly to the tank through the central groove 14. Two small branch streams, the volumetric flow rate of which is determined by the setting of the pressure-control valves 32 and 33 and may be on the order of one litre per minute for each valve, pass with an insignificant pressure drop through the spool passages 34 and 45, the pressure-compensated flow control valves 32 and 33 and the pressure control valves 36 and 37 back to the tank.

If it is desired to displace the spool 12 to the right, for example, towards the working position corresponding to a displacement to the right of the piston of the cylinder 28, the pressure control valve 36 is operated such that the pressure within the cap 30 is increased. Initially, of course, the maximum pressure that can be brought about within the cap 30 cannot exceed the idling pressure of the pump, and in practice is somewhat lower than the idling pressure. It therefore does not suffice for effecting a firmly controlled displacement of the spool throughout the travel range of the spool. The initial portion (1–2 mm) of the spool travel does not require a substantial force, however, and the pressure therefore is sufficient to cause the spool to travel that distance.

In accordance with the invention, the spool 12 during its travel to the right over the initial portion will restrict the flow path between the pump connection groove 15 and the tank connection groove 14 to such an extent that the pressure produced by the pump will increase to 50 bar, for example, before the opening of the flow path between the groove 17 and the groove 19 commences. This pressure will of course be available in the cap 30 (the small pressure drop between the groove 17 and the cap 30 is disregarded here), and ensures that the spool 12 will be displaceable to the right in a firmly controllable manner throughout its travel range.

Under the action of the increased pressure the spool thus can travel further so that a flow path is established between the groove 17 and the groove 19 and between the groove 20 and the groove 22. At the same time, the flow path between the groove 18 and the spool passage 35 is restricted.

In accordance with a preffered feature of the invention, the spool 12 is designed such that a substantially constant pressure drop of 50 bar, for example, is maintained between the pump connection port 25 and the groove 14 over the further spool travel. The pressure drop inevitably causes an undesirable loss of energy and it is desirable therefore that the pressure drop does not increase with further travel of the spool, or at least does not increase more than is necessary to ensure that an adequate spool displacement force is available throughout the spool travel range.

Fig. 2 is a diagrammatic representation of the system shown in Fig. 1. In the diagram the variable restriction A_1 represents the flow path between the pump connection port 25 and the tank connection port 23 that is, the flow path between the groove 15 and the groove 14. The variable restriction A_2 represents the flow path established between the pump connection port 25 and the load connection port 26 (the flow path between the groove 17 and the groove 19) upon spool displacement to the right, and the variable restriction A_3 represents, for the same direction of spool displacement, the flow path between the load connection port 27 and the tank connection port 23 (the flow path between the groove 20 and the groove 22).

In the neutral position, the restriction A_1 is fully open while the restrictions A_2 and A_3 are fully closed. During the initial portion of the spool travel the restriction A_1 is partially closed, and the restrictions A_2 and A_3 remain closed. After the closing of the restriction A_1 has progressed far enough to cause the pressure produced by the pump to reach the predetermined value of 50 bar, for example, further spool travel will cause the restrictions A_2 and A_3 to open gradually in proportion to the further closing of the restriction A_1. In order that the pressure drop may then remain approximately constant over the remaining portion of the spool travel range, the restrictions are varied such that the pressure drop across the restriction A_1 is approximately equal to the sum of the pressure drops across the restrictions A_2 and A_3.

The graphic representation in Fig. 3 shows the variation of the pressure drops (ordinate axis) across the three restrictions with the travel of the spool from the neutral position (abscissa axis). The curve P_1 represents the pressure drop across the restriction A_1, the curve P_2 represents the pressure drop across the restriction A_2 and the curve P_3 represents the pressure drop across the restriction A_3. The straight line P_s represents the pressure required to displace the spool. The dash-dot curve P_1' finally, shows the variation of the pressure drop in a typical known directional control valve.

As shown in Fig. 3, the pressure drop P_1 in the directional control valve according to the invention is well above the spool displacing pressure P_s throughout the portion of the spool travel range in which the flow path between the pump and the load is open. In the known directional control valves, on the other hand, the pressure drop P_1' is only slightly higher, or even lower, than the pressure required for displacing the spool; the line P_s may be regarded as applicable both to the valve according to the invention and the known valves.

The pressure drop curves P_1, P_1', P_2 and P_3 in Fig. 3 are valid in the case where the volumetric flow rate is the rated volumetric flow rate of the valve. However, because the curve P_1 of the valve according to the invention is well above the line P_s, a pressure that is adequate for a firmly controlled spool displacement is obtained even if the actual volumetric flow rate should be substantially lower than the rated volumetric flow rate.

The pressure drop characteristic described above and illustrated in Fig. 3 may be obtained by suitable design and sizing of the spool and, more particularly, of the lands thereof which cooperate with the valve housing elements defining the grooves 14–22 in order to block and open the flow paths upon movement of the spool.

To this end, the width of the spool land 41 which registers with the central groove 14 in the neutral positon of the spool is sufficiently large to ensure that a spool travel of 1–2 mm is sufficient to bring about the required restriction of the flow path. On the other hand, the width of the land may not be so large as to cause an unduly large pressure drop in the neutral position of the spool. The same considerations apply to the inner edge of the two spool lands 42 and 43 which are located on either side of the central land 41.

In order that the restriction of the flow path between the pump connection port 25 and the tank connection port 23 may not take place too abruptly, the spool lands 42 and 43 are provided with tapering notches 44 and 45, respectively, at the end of the lands adjacent the central land 41 (only one notch is shown on each land but in practice it is normally preferable to provide each land with at least two notches. Similar notches 46 and 47 are provided at the opposite end of the spool lands 42 and 43, respectively, and also at the inner ends of the two outermost spool lands 48 and 49 similar notches 50 and 51, respectively, are provided. As will be evident from the foregoing, the notches 44 and 45 each form parts of the restriction A_1, while the notches 46 and 47 form parts of the restriction A_2 and the notches 50 and 51 form parts of the restriction A_3.

The pressure drop characteristic being given, the design of the spool 12 presents no fundamental difficulties; in principle the design may be accomplished analytically. However, in practice it is to be expected that the exact dimensions and the exact shape of the notches 44, 45, 46, 47, 50, 51 at least to some extent have to be determined for each valve type and valve size by trial and error.

In the case where the pump 24 is driven by a power source, such as an internal combustion engine of a vehicle, having a large speed range, the idling speed may be inadequate to produce the pressure of 5 bar, for example, that is required to effect the initial travel of the spool. A back-pressure valve connected in series with the directional control valve may be used to ensure that an adequate idling pressure of 5 bar, for example, is available, regardless of the speed of the power source. However, at high speeds, such a valve would cause an undesirable additional pressure drop and, consequently, an increased loss of energy.

In a modification (not shown) of the system illustrated in the drawings, the just-mentioned problem of always providing an adequate idling pressure without unduly increasing the energy loss, the pump 24 feeds the directional control valve through a flow dividing valve. This valve divides the stream produced by the pump into a primary stream which is fed to the pump connection port 25 and a secondary stream which is fed to the tank through a back-pressure valve. The back-pressure valve maintains a predetermined minimum pressure of 5 bar, for example, on the inlet side thereof and thus maintains such pressure on the outlet side of the pump. The pressure-compensated flow control valves 32 and 33 are fed from the pump upstream of the flow dividing valve. Thus, regardless of the speed of the power source, a pressure equal to the predetermined minimum pressure can always be obtained within the caps 30 and 31.

Although the valve shown and described by way of example is a four-way valve for a double-acting load, the invention is applicable to other types of valves, such as a three-way valve the spool of which is hydraulically displaceable from the neutral position in one direction only.

CLAIMS

1. A hydraulic directional control valve comprising a valve housing having a pump connection port, a load connection port and a tank connection port, and a valve spool provided in the valve housing and operable by a hydrostatic actuator for controlling the fluid flow paths between the connection ports, the valve spool being displaceable between a neutral position in which a pump connection port is in substantially unrestricted fluid flow communication with the tank connection port and a working position in which the pump connection port is in fluid flow connection with the load connection port, and a pressure-compensated flow control valve, the inlet of which is connected with the pump connection port and the outlet of which is connected to the actuator, the spool being adapted during its travel from the neutral position towards the working position to restrict the flow path between the pump connection port and the tank connection port, before it opens the flow path between the pump connection port and the load connection port, to such an extent that the pressure drop (P_1) between the pump connection port and the tank connection port increases to a value at least equal to the hydrostatic pressure that is required in the actuator for displacing the valve spool throughout the spool travel range.

2. A control valve according to claim 1, wherein the spool is adapted such that after the said pressure drop (P_1) between the pump connection port and the tank connection port has reached said value it remains approxi-

mately constant with continued travel of the spool away from the neutral position.

3. A control valve according to claim 1 or claim 2, wherein the spool is axially displaceable and the pressure-compensated flow rate control valve is disposed in the spool adjacent one end thereof.

4. A control valve according to any preceding claim, wherein there is a said hydrostatic actuator and a said pressure-compensated flow control valve associated with travel of said spool valve in each of the two directions of movement thereof.

5. A control valve according to any preceding claim, wherein said actuator is operable by way of a continuously variable fluid pressure control element.

6. A hydraulic directional control valve as hereinbefore described with reference to the accompanying drawings.

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