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(54) FLUID VALVE

(71) I, GORAN HENRIKSSON, a Swedish National, of Backsippestigen 4, S-575 00 Eksjo, Sweden, do hereby declare the invention for which I pray that a patent may be granted to me, and the method by which it is to be performed, to be particularly described in and by the following statement:

The present invention relates to devices for use in fluid systems, especially hydraulic systems, comprising a source (pump) for pressurized fluid and a consumer (cylinder) which can be connected thereto by means of a main conduit.

For many uses of such systems a device is required, viz. a so-called load-sensing valve which senses the pressure generated by the load in the pressurized liquid when the valve is closed and controls the pressure coming from the pump to a valve equal to or somewhat exceeding the valve generated by the load in the pressurized liquid. In this manner it is avoided that the load sinks if the pressure is too low when the load is coupled to the pump and also that the load starts with a jump when the pressure is too high at the coupling, viz. when the pressurized liquid from the pump is connected with the working space of the hydraulic cylinder. Hereinbelow any consumer which transforms the power of a pressurized liquid into work is designated cylinder (hydraulic motor). Likewise, hereinbelow the so-called load pressure is the liquid pressure below a cylinder piston which is in an intermediary position and is bearing a load. The load tends to force the cylinder piston backwards but the liquid is stopped by a closed valve.

The fluid pumps which can be used in the fluid system mentioned above can be conventional pressure-compensated pumps of the so-called Toma type which keep a constant pressure independently of the amount of liquid. It is also possible to use so-called pressure-flow-compensated pumps, viz. pumps which in addition to a normal output main conduit (pressure conduit) also have a so-called pilot conduit. The pressure conduit and the pilot conduit cooperate in such a manner that the pressure conduit always has a constant amount of pressure in excess of the pressure in the pilot conduit, normally about 13 bars.

The function of such a pump can take place between two extreme operation conditions. In the first extreme condition the pilot conduit is closed and the pump delivers 13 bars in the pressure conduit. This is achieved by providing the pressure conduit of the pump with a small leakage from the pressure side to the sump of the pump housing. The capacity is controlled automatically to near zero since the cylinder stroke of the pump pistons is small. In other words, when the pilot conduit is closed by a valve and has the pressure zero the pressure in the pressure conduit is 13 bars (as an example). In the other extreme condition the pilot conduit is connected to the output pressure conduit of the pump. Since the pressure conduit has a pressure of 13 bars the pilot conduit will also have a pressure of 13 bars and, according to the above, the pressure of the pressure conduit is increased to 13 + 13 = 26 bars. In the next phase the pressure conduit has a pressure of 26 + 13 = 39 bars, the pilot pipe has 26 bars, etc. When the pressures are maximal for the pump the capacity is controlled down to the zero valve.

The specific properties of the pressure-flow-compensated pumps have enabled control of *a single* consumer in an excellent manner from the point of view of efficiency and control accuracy (cf.Figure 1). It has not been possible to simultaneously control more than one cylinder in this manner owing to the complicated nature of the control system.

To enable a judgment of the present

invention it seems suitable also to describe a conventional system for pumps having constant capacity. In such systems the pump delivers a constant flow or, in other words, a flow which is proportional to the speed of rotation. When no pressurized liquid is fed to a consumer the oil from the pump is led back to the tank since the control slide is provided with a shunt function. In such a case the pump pressure is zero. When the pressurized liquid is to be fed to a consumer two functions shall occur in the valve: the shunt function shall be closed and the pump shall be connected to the consumer. In order to achieve a jerk-free start the pump pressure has to be built up (by closing the shunt function) to the consumer's load pressure before the pump is connected to the consumer. Since the *load pressure* in the same apparatus can be at any valve between zero and the maximum value it is impossible with this system always to achieve a coupling in which it is ensured that the *pump pressure* (obtained by restriction of the shunt pipe) *closely coincides with the load pressure.*

According to the present invention, in a fluid valve for controlling the supply of liquid under pressure from a pump to the load, a valve member is movable to put a valve inlet port into communication with a valve outlet port and to put the outlet port into communication with a regulating port which is connected to a regulator arranged to regulate the pressure of the liquid from the pump

Thus, as the valve is opened, the load pressure is connected to the pump pressure regulator, so that the pump output pressure is controlled to be a little above the load pressure.

The invention may be carried into practice in various ways, and certain embodiments will now be described by way of example with reference to the enclosed schematical drawings, in which:

Figure 1 shows the basic principle of a pressure-flow-compensated pump;

Figure 2 shows a circuit including a pump and control valve;

Figure 3 is a detail of the control valve according to Figure 2.

Figure 3a is a section on the line I-I of Figure 3; and

Figure 4 shows the use of the invention in a hydraulic system comprising a fluid pump with a constant flow.

The valve 1 of Figure 3 is connected to the delivery of the pump P1, through the inlet opening 8. The pump is of the pressure-flow-compensated type. The return conduit or pipe to the tank T is connected to openings 6 and 6*a*. A slide or valve member 2 in the valve has a middle land 12 and two end lands 13 and 13*a*, the slide diameter between the lands being reduced to provide valve passages. A lever 15 sets the position of the valve member 2.

The consumer, e.g. a hydraulic cylinder is connected to one of two outlet connections 7 it if is of the single action type; if the cylinder is of the double action type the opposite ends of the cylinder are connected to respective openings 7.

A servo line 16 (102) is connected to the pump P.1 from the downstream side of two non-return valves 5 each having its inlet connected to one of a pair of ports 4 leading from the valve housing at either end of the land 12 in its central position. There is also a leakage line from the line 16 to the tank via a restriction 10.

The ends of the lands 12, 13, 13*a* adjacent each passage 7 have shallow recesses 3 or 9 which diverge to be of increasing cross-sectional area towards the passage 7.

Figure 3 shows that the dimensons A and B are equal so that as the valve member 2 moves to the left, the inlet 8 comes into communication with the passage 7 by way of the recess 3 at the same instant that the passage 7 comes into communication with the opening to the port 4 leading to the servo line 16. The left passage 7 communicates with the return at 6 by way of the recess 9 at the same instant.

If, for instance, the servo line were to open first, the consumer will be connected to the servo line when the connection between pump and consumer is closed. The result will be that the load of the *consumer* (for instance a cylinder under load) will sink since liquid will then pass through the servo line 4 via the non-return valves 5 and the restriction 10 to the tank. The servo line must have a leakage possibility to the tank since otherwise the capacity regulator of the pump cannot return to nearly zero capacity.

Several valve sections can be coupled together and arranged as desired to serve one, or several single or double-action cylinders.

By connection of the pressure output and servo lines of several sections the above desired purposes are achieved.

In the case of single-action cylinders one of the servo conduits 4 would be closed since no pressurised oil is required for lowering of the load.

If several sections shall be manoeuvred simultaneously the pump pressure must be sufficiently high for acting on the receiver. Accordingly, the servo pressure shall be increased above the said pressure, e.g. 13 bars. This is made as indicated above (the servo line is connected with the consumer). To ensure that the servo oil does not pass from the valve section having the highest pressure into a section with a lower pressure each servo connection to the line 16 is

provided with a non-return valve 5.

The valve according to the invention can be also used in combination with fixed displacement pumps. Such pump types have a flow rate which is theoretically proportional to the speed of rotation. It has been explained above that the pumps in the case of conventional valves are unloaded by passing the oil through a shunt function in the valves. A load-sensing function is not possible to achieve without a serious complication of the valve.

The present invention makes it possible to use the same simple valve and to simultaneously achieve a load-sensing function by an attachment (control section). The control section is known *per se* as a control for pressure and flow but has not previously been connected with a valve of the type according to the present invention for achieving a load sensing function. The connection of the units is elucidated by Figure 4.

The pump P.2 delivers oil, partly to the control section 35, partly to the valve 1. In the mode of operation in which the pump is intended to work in an unloaded state the slide of the valve is in the neutral position. The servo line 16 is closed by the slide and drained by the restriction 10 to the tank. If the restriction opening 10 to the tank is larger than a restricted opening 37 in the end of a cup slide 36 of the control section, the pressure difference across the slide 36 will lift the slide 36 from its seat and the pumped oil will return to the tank T at 51.

A spring 38 is designed to exert a closing pressure of e.g. 13 bars on the slide 36 which has to be overcome before the slide rises to open the by-pass 51.

When the valve 1 is opened the pump output is connected to the line 16 and the upper side of the slide 36 simultaneously with its connection to the consumer, so that the by-pass line 51 is partially or completely closed.

The mode of action is thus analogous with the one for the use of the valve for controlling pressure-flow-compensated pumps, viz, the pressure is controlled so that in both cases the pump pressure is equal to the load pressure + 13 bars, viz 13 bars are always available to manoeuvre the capacity regulator of the pump and the control section respectively. In the last-mentioned case the springs 38 are dimensioned so as to give a shunt pressure of for instance 13 bars. Adjustment to maximum pressure is made by the set screw 41 by means of which it is possible to adjust the pressure of the spring 40 which keeps the valve head 39 against its seating. By lifting the head 39 the pressure in the servo line can be decreased and the pressure prevented to increase.

Further, the valve functions in both combinations as a pressure compensated flow control. The difference between a conventional valve and a pressure compensated valve can be elucidated by the following example. It is assumed that the pump delivers Q litres per minute at a primary pump pressure of 100 bars and at a restriction which results in a pressure drop of 50 bars, for example the consumer is fed at a pressure of 50 bars. Now, if this pressure must be changed, viz. the consumer meets an increased resistance, to for instance 75 bars, and Q is to be maintained, it will be necessary to change the restriction area so that the pressure decrease in the restriction will be 25 bars. This is what happens with devices which are not pressure compensated. In the both systems according to the invention the control is made with a constant pressure drop in the restriction, viz. for instance 13 bars, and this pressure drop is only dependent on Q and independent of the pressure requirements (resistance) of the receiver. This means that Q does not change with variations in pressure. The pressure drop or the control pressure 13 bars can, of course, be changed within very wide limits.

The valve device according to the invention can be used in and adapted to different control systems, e.g. for manual, pneumatic, hydraulic or electric action.

The sensitivity and rapidity of the valve can be changed by using different cross-section areas in ducts present and the hole 37 according to Figure 4 can in certain cases be excluded.

Between the pump and the opening 8 in the valve device a non-return valve can be inserted as a security against unintentional liquid losses from a loaded cylinder in certain cases.

Figures 5 and 6 illustrate an alternative embodiment of the valve device according to the invention. The broken line shown to the left in Figure 5 illustrates the central cross section plane of a valve which is similar to the embodiment according to Figure 3 except with respect to slide and the tapering recesses or notches. Thus in the device shown in Figures 5 and 6 the central land 12' and the outer lands, one of which, viz. 13a' being shown in Figure 5, are not provided with any recesses or notches but have a continuous, smooth cylindrical outer surface. Instead the recesses 3' and 9' are provided in the cylindrical wall of the valve housing which is preferably of cast iron. Also the recesses 3' and 9' converge as shown so that, as with Figure 3, the smaller area is uncovered first as the control land 12' moves to the left. The cross section of the recess taken in a plane perpendicularly of the direction of valve member movement, varies continuously in the direction of length

of the slide and the area of the recess at its small end is considerably less than, usually less than ¼ preferably less than ⅛ and, for instance about 1/15, the area at the large end. All recesses shown in the drawings are of similar design. The right end of the notch 3' is situated in the same or at least approximately the same plane, perpendicular to the direction of length of the slide, as the right end of the recess 120 in the wall of the slide bore, said recess 120 being connected to the servo line 4 leading to the pressure regulating device of the pump. This means that the right circumferential end edge of the valve section 12' will open the connection from the consumer conduit 7' through the recesses 120 to the line 4 at substantially the same time as the connection between the consumer conduit 7' and the control pressure port 8 of the valve which port is connected to the main outlet of the pump P. As the recesses 3' and 120 are provided in the wall of the bore in the slide housing at the casting of the housing their mutual positions will be very accurately defined, and since the fluid passage through both said recesses are controlled by one and the same end edge of the central section of the valve slide the position of the slide at the moment of opening said passages is very exactly defined and does not depend on various unavoidable tolerances in the dimensions of the slide and the housing.

The dimension of the notches 120 along the periphery of the valve slide section 12' is considerable as compared with the corresponding dimension of the right (small) end of the notches 3', whereby the pressure in the consumer line 7 can rapidly be transferred to the pump pressure regulating device before substantial flow through notch 3' takes place.

The continuous, linear or non-linear variation of the cross section (flow section) of the notches 3' is provided for the purpose of obtaining a *flow regulating function* of the valve. From a position of the valve operating lever 15 in which the connection between the consumer conduit 7 to the pressure port 8 through the small end of the notches 3 or 3' has just been established the flow can be increased in a controlled fashion by the lever 15 the deflection or reading of which being a direct measure of the fluid flow from the pump to the consumer and being preferably proportional to said flow.

It will be apparent from the foregoing that a device according to the invention is capable of providing a rapid *pump pressure regulating function* as required by the existing load pressure as well as a convenient fluid flow controlling function enabling the operator to move various loads at desired well-defined velocities without any risk that the load will sink when the load moving operation is started.

For practical purposes the notches 3 and 3' at the large end have a through-flow area permitting at least about 70%, for instance about 80-90%, of the maximum flow through the fully opened valve device to pass therethrough. Said maximum flow is usually defined by the through-flow areas of conduits and channels outside the valve device proper.

WHAT I CLAIM IS:

1. A fluid valve for controlling the supply of liquid under pressure from a pump to the load in which a valve member is movable to put a valve inlet port into communication with a valve outlet port and to put the outlet port into communication with a regulating port which is connected to a regulator arranged to regulate the pressure of the liquid from the pump.

2. A valve as claimed in Claim 1 in which the outlet port is arranged to be put into communication with both the inlet port and the regulating port at the same instant.

3. A valve as claimed in Claim 1 or Claim 2 comprising a land on the valve member slidable in relation to a part of the valve housing separating the inlet and outlet ports.

4. A valve as claimed in any of the preceding claims including a first and a second passage, both defined between the outer surface of the valve member, and a housing in which the valve member slides, the first passage extending substantially in the direction of valve member, and a housing in which the valve member slides the first passage extending substantially in the direction of valve member movement, and acting when the valve is open to put the inlet port into communication with the outlet port, and the second passage providing in one position of the valve member, fluid communication between the outlet port and the pump pressure regulator, the first and second passages being located near one end of the valve member spaced apart around the valve member.

5. A valve as claimed in Claim 4 in which the first and second passages are consituted as respective recesses in the wall of the valve housing.

6. A valve as claimed in any of the preced ing claims including a non-return valve disposed to be in the line of communication between the regulating port and the regulator.

7. A valve as claimed in any of the preceding claims including a non-return valve on the inlet side of the inlet port.

8. A valve as claimed in Claim 4 and any of Claims 5-7 in which the first passage is of increasing cross-sectional area along its length, the part of smallest area being uncovered first as the valve is opened.

9. A valve as claimed in any preceding claim in which the valve member is cylindrical.

10. A valve as claimed in any preceding claim in which its housing is formed of cast iron.

11. A valve as claimed in Claim 8, in which the smallest cross section area is less than ¼ of the largest cross section area at the other end of the passage.

12. A valve as claimed in Claim 8 in which the smallest cross section area is less than ⅛ of the largest cross section area at the other end of the passage.

13. A valve as claimed in Claim 8 or any of the preceding claims appendant to Claim 8 including an operating member for moving the valve member, the displacement of which is a measure of the flow permitted in dependence of the cross sectional area of the narrowest uncovered part of the passage.

14. A fluid valve constructed and aranged substantially as herein specifically described with reference to Figures 3 and 3*a* or Figures 5 and 6 of the accompanying drawings.

15. A valve as claimed in any of the preceding claims in combination with a pump whose outlet pressure can be controlled by the pump pressure regulator.

16. A combination as claimed in Claim 15 in which the pump is a positive displacement pump which is capable of pumping its outlet direct to exhaust by way of a valve responsive to pump pressure.

17. A combination as claimed in Claim 16 including a valve in the by-pass line which is normally closed when pump pressure appears at the regulating port.

18. A combination as claimed in Claim 17 including means for setting a bias against opening of the by-pass valve to establish a pressure drop between the pump pressure necessary to open the by-pass valve and the pump pressure normally required by the consumer.

19. A combination as claimed in Claim 18 in which the bias is adjustable.

20. A combination of a pump and a valve constructed and arranged substantially as herein specifically described with reference to Figure 4 of the accompanying drawings.

KILBURN & STRODE,
Chartered Patent Agents,
Agents for the Applicants.

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1572036 COMPLETE SPECIFICATION

3 SHEETS *This drawing is a reproduction of the Original on a reduced scale*

Sheet 1

Fig.1

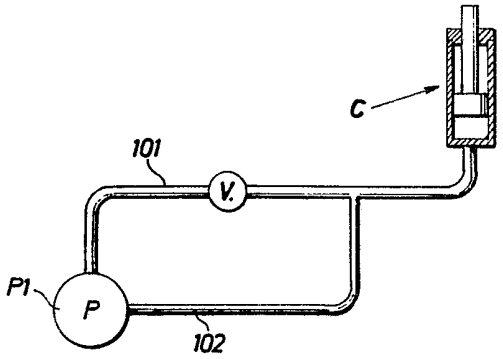


Fig.2

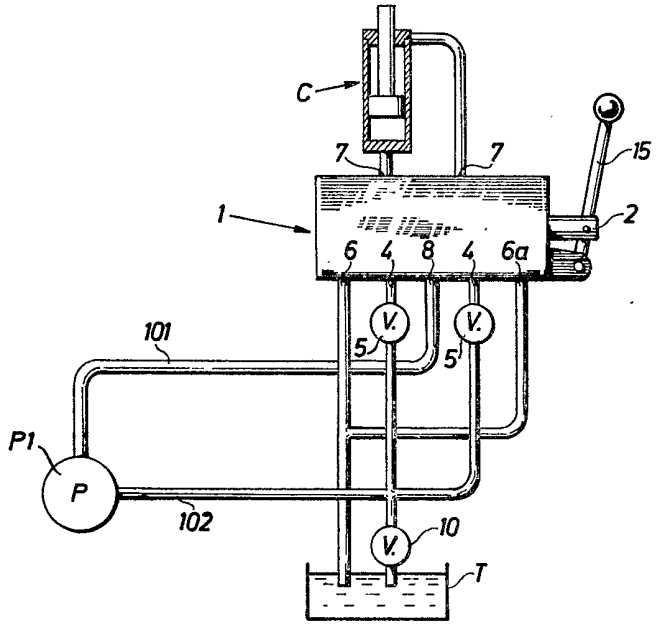


Fig.3

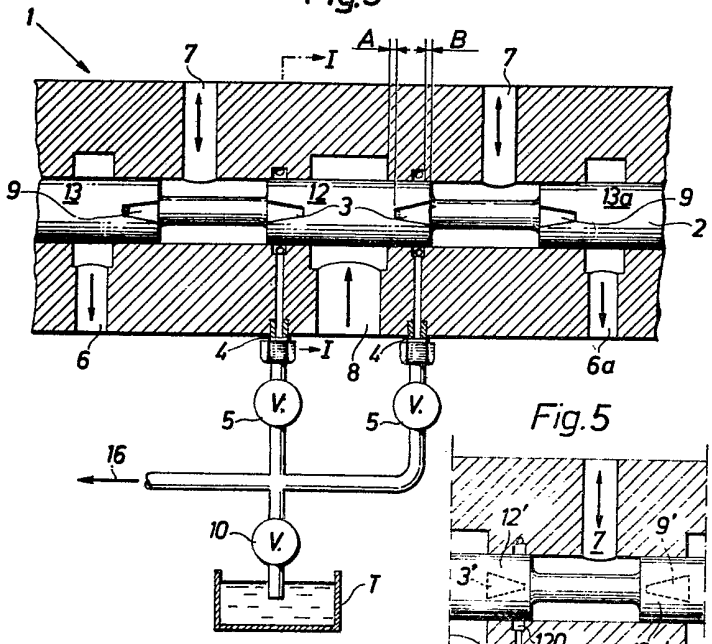


Fig.5

Fig.3a

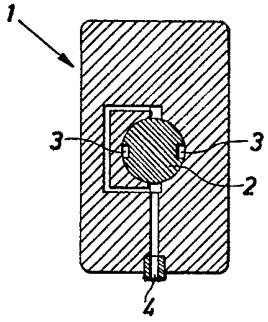
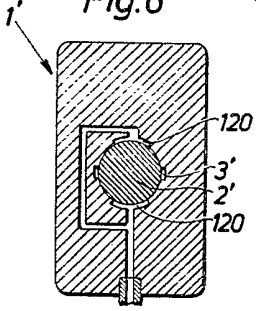


Fig.6



1572036 COMPLETE SPECIFICATION
3 SHEETS *This drawing is a reproduction of the Original on a reduced scale*
Sheet 3

Fig.4

