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[56] **References Cited**
UNITED STATES PATENTS
 2,980,135 4/1961 Tennis..... 137/596.12
 3,154,100 10/1964 Winston..... 137/625.6X
 3,464,443 9/1969 Tennis..... 137/596.12
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[54] **FLUID FLOW CONTROL INSTRUMENTALITY**
13 Claims, 3 Drawing Figs.
 [52] U.S. Cl..... **137/596.12**
 [51] Int. Cl..... **F16k 11/10**
 [50] Field of Search..... **137/596.12,**
596.13, 596.2, 596.18, 625.6, 625.63, 625.66

ABSTRACT: A relief valve obstructs flow of exhaust fluid from the contracting end of a cylinder to an outlet so that exhaust fluid under pressure will be available for flow to a supply passage leading to the expanding end of the cylinder whenever pressure in the supply passage drops to a predetermined value. The flow obstructing effect of the relief valve is governed by control means which causes the relief valve to least obstruct exhaust flow to the outlet at times when high pressure obtains in the supply passage.

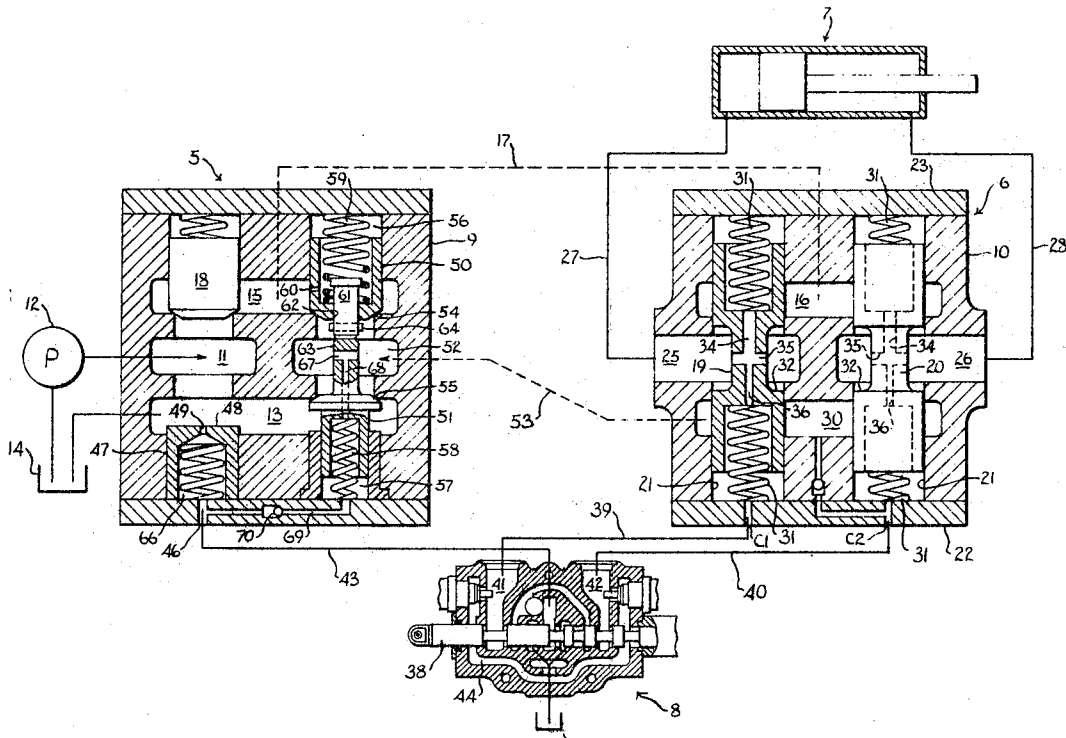
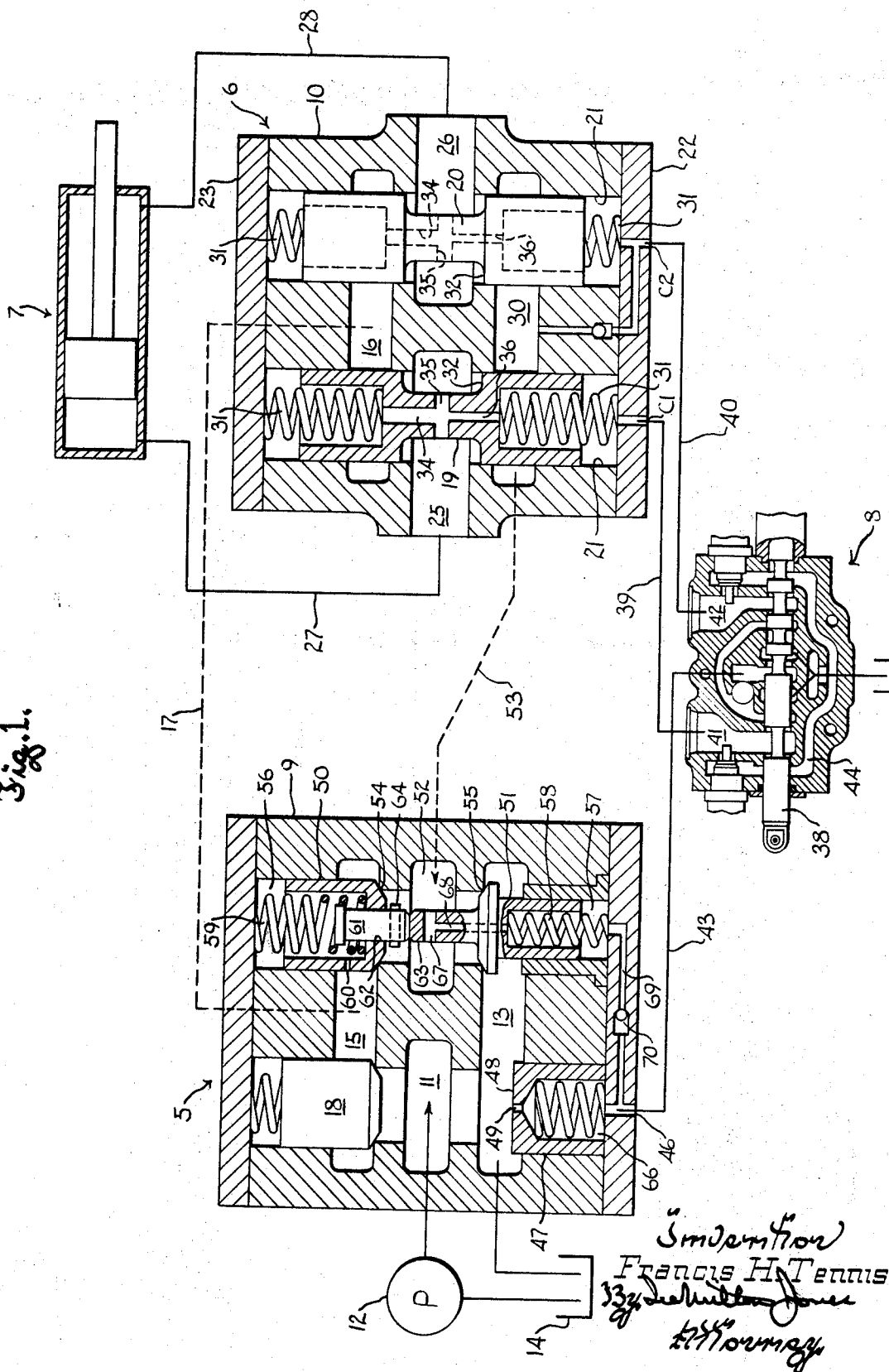


Fig. 1.



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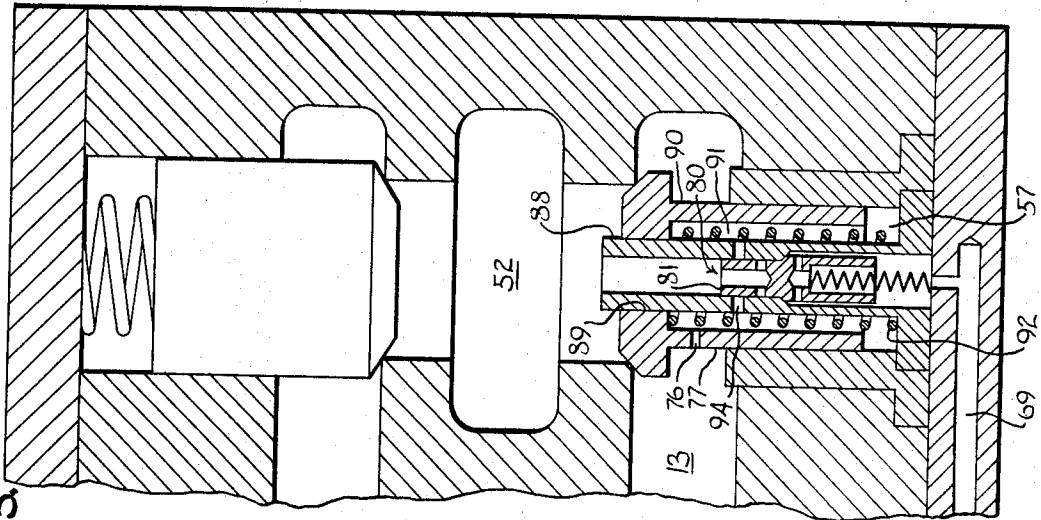


Fig. 3

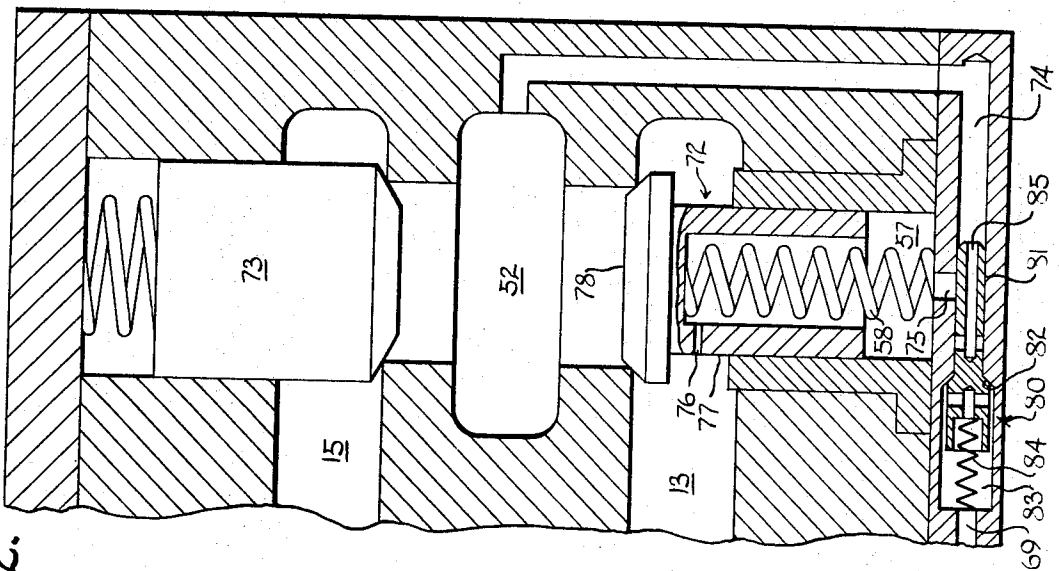


Fig. 2

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FLUID FLOW CONTROL INSTRUMENTALITY

This invention relates to fluid flow control instrumentalities, and has more particular reference to improvements in void control mechanisms.

As is well known, double acting hydraulic cylinders such as are used on certain mobile equipment are most often operatively connected to loads that frequently act to expel pressure fluid from one end of the cylinder faster than pressure fluid can be supplied to its other end by the control valve governing the cylinder. The load drives the cylinder at such times, and an objectionable void is created in the expanding end of the cylinder as a result.

Efforts have been made to relieve voids formed in cylinders in this manner through incorporation in their control valves of a so-called anticavitation check valve located in a feedback passage leading to the supply passage from the exhaust passage. This check valve was arranged to be kept closed by pressure of fluid in the supply passage, and to open whenever there was insufficient pressure in the supply passage to keep it closed.

It can be readily seen that while this expedient may have provided for some relief of voids, it could not prevent void formation. This results from the fact that as long as the reservoir receiving the exhaust fluid was maintained at atmospheric pressure, the return fluid in the exhaust passage was also at substantially the same pressure, and incapable of opening the anticavitation valve until a subatmospheric pressure obtained in the supply passage. Hence, a void had to be created before the anticavitation check valve could open to relieve the void.

My U.S. Pat. No. 2,980,135 issued Apr. 18, 1961 discloses how the above described shortcomings of anticavitation valves can be overcome by use of what is referred to as a loader circuit which assures delivery of load pressurized exhaust fluid to the supply passage for flow to the expanding end of a cylinder, together with pump fluid, before a void is formed in the cylinder. This loader circuit operates by reason of a restriction device in the exhaust passage, to make it possible for the anticavitation check valve to open in response to pressure well above atmospheric, and thus not only prevent formation of voids but to also prevent serious loss of working pressure in the cylinder at times when it tends to be driven by its load.

Exhaust pressures of this magnitude, however, tend to overload the pump, especially when a heavy load is encountered by a cylinder supplied with pressure fluid from the pump; at which time the pump must not only work to overcome the load but to also overcome the restriction to exhaust flow.

With this objection in mind, it is a purpose of this invention to provide, in a fluid flow control instrumentality having a void control check valve and supply and exhaust passages the latter of which leads to an outlet, a movable flow restriction means that obstructs exhaust flow to the outlet, and control means therefor by which the flow obstructing effectiveness of the flow restriction means is minimized at times when high fluid pressures obtain in the supply passage.

More specifically, it is a purpose of this invention to provide a fluid flow control device such as described in the preceding paragraph, wherein the control means is activated in response to high pressure in the supply passage and in turn effects actuation of the movable flow restricting means to a non-obstructing position.

Still another object of this invention resides in the provision of a fluid flow control device with a void control check valve and a low pressure relief valve to regulate exhaust flow to an outlet, wherein control means for the low pressure relief valve causes the latter to resist opening to a greater extent when high pressure obtains in the exhaust passage of the instrumentality than when low pressure obtains in the exhaust passage.

With these observations and objects in mind, the manner in which the invention achieves its purpose will be appreciated from the following description and the accompanying drawings. This disclosure is intended merely to exemplify the invention. The invention is not limited to the particular structure or method disclosed, and changes can be made therein which lie within the scope of the appended claims without departing from the invention.

The drawings illustrate several complete examples of the physical embodiment of the invention constructed according to the best modes so far devised for the practical application of the principles thereof, and in which:

FIG. 1 is a more or less diagrammatic component sectional view illustrating how this invention can be incorporated in a control valve mechanism for a double acting hydraulic cylinder; and

FIGS. 2 and 3 illustrate modifications of the invention.

Referring now to the accompanying drawing, the numerals 5 and 6 respectively designate the inlet section and one control section of a pilot operated divided spool control valve mechanism for a reversible fluid motor such as the double acting hydraulic cylinder indicated at 7. A pilot valve 8 governs operation of the valve mechanism in the manner described in my copending application Ser. No. 676,547, filed Oct. 19, 1967, now U.S. Pat. No. 3,464,443 to which reference may be had for a more complete disclosure of the divided spool valve mechanism.

The inlet section 5 has a body 9 which is adapted to be flatwise superimposed upon and secured to the body 10 of the control section 6, to serve that control section and others like it that may be secured in a stacked relation thereto. The inlet section 5 has an inlet 11 to receive pressure fluid from a source such as the pump 12, and it has an outlet passage 13 through which exhaust fluid can be returned to a reservoir 14. It also has an inlet passage 15 which registers with a supply passage 16 in the control section 6. The dashed line 17 indicates that passages 15 and 16 are communicated at all times.

Pressure fluid entering the inlet 11 must unseat a check valve 18 in order to gain access to the inlet passage 15 for flow to the supply passage 16 in the control section 6.

The control section contains a pair of control spools 19 and 20, slidably mounted in parallel bores 21 in the body 10, one in each bore. Covers 22 and 23 close the bores 21 at their opposite ends and cooperate therewith to define pressure chambers into which the opposite ends of the spools project.

Service passages 25 and 26, one for each spool, intersect the bores 21 substantially medially of their ends. As shown, the service passage 25 is associated with the spool 19 and is connectable with the head end of the cylinder 7 as by a pressure fluid line 27. The other service passage 26 is associated with the spool 20 and is connectable with the rod end of the cylinder as by a pressure line 28.

The service passages 25-26 open outwardly to opposite sides of the body 10, from locations midway between the supply passage 16 and an exhaust passage 30. The supply and exhaust passages 16 and 30 respectively, are elongated and parallel, and each communicates with both of the bores 21 in which the valve spools operate.

In their neutral positions, to which the spools 19-20 are biased by springs 31 at each end thereof, a medial circumferential groove 32 in each spool registers with the service passage associated with that spool, while the cuplike end portions of the spools extend across the supply and exhaust passages 16 and 30, respectively, to block communication thereof with the service passages.

From the description thus far, it will be apparent that, depending upon which side of neutral the spool 19 is shifted to, it will control fluid flow to and from the head port of cylinder 7; while spool 20 will control fluid flow to and from the rod port of the cylinder. This follows by reason of the fact that each spool is operable to selectively communicate its service port with either the supply passage 16 or the exhaust passage 30.

The spools are actuated to their feed and vent positions described by flow of fluid under pressure into one or the other of the pressure chambers at their opposite ends, and the response of spools depends upon whether the upper or the lower of the two pressure chambers associated with each spool is at a predominating pressure. For the above purposes, each spool is provided with an axial passage 34 which communicates with its upper pressure chamber and also opens through radial ports 35 to its circumferential groove 32. The passage 34 also opens to the lower pressure chamber of the as-

sociated spool through a restricted axial passage 36. In addition pilot control ports C1 and C2 in the body open to the pressure chambers defined by the lower end portions of the bores 21.

Consequently, if pressurized control fluid is directed into the control port for either spool, the restricted passage 36 therein will cause a pressure to be created in the adjacent pressure chamber greater than obtains in the chamber at the opposite end of the spool, and the spool will be shifted upwardly thereby from neutral to its feed position at which groove 32 communicates its service passage 25 (or 26) with the supply passage 16.

On the other hand, if the lower pressure chamber of either spool is vented, it is possible for pressure fluid from its service passage to flow into the passage 34 and because of the restricted passage 36 create a pressure in the upper pressure chamber of a value exceeding that in the vented lower chamber to cause the spool to be shifted to its vent position communicating its service passage with the exhaust passage 30.

The pilot valve 8 is operable to selectively vent either control port C1 and C2 and to concurrently deliver pressurized control fluid to the other control port. Hence, either spool is actuatable to its cylinder venting position when the other spool is actuated to its cylinder feeding position.

Pressure lines 39 and 40 communicate the pilot service passages 41 and 42 with control ports C1 and C2, respectively, to enable the pilot valve to control the spools as described. When the pilot spool 38 is in its neutral position shown, however, it closes off the control ports C1 and C2, and allows pump fluid flowing into its inlet from an auxiliary supply line 43, to flow through an open center groove in the spool to the reservoir. As in most conventional spool valves, the pilot spool is shiftable out of neutral to each of a pair of operating positions to selectively communicate either of its service passages with the pilot inlet and the other service passage with an outlet passage 44.

The pilot supply line 43 connects with a port 46 that is located in the inlet section 5 and opens to a pressure chamber containing a pistonlike pump unloading valve 47. The face 48 of the unloading valve is exposed to pressure of pump output fluid in the inlet 11, and an orifice 49 therein opening through its face provides for flow of pump output fluid into the space in the chamber behind the unloading valve, and to the pilot valve through its supply line 43.

As long as the pilot spool 38 is in its neutral position shown, it vents the unloading valve chamber and pump output fluid in the inlet 11 acts to open the unloading valve to cause the entire output of the pump to be dumped to the outlet passage 13. Whenever the pilot spool is moved to either of its operating positions, however, it closes off the port 46 from the pilot outlet so that pressure of inlet fluid can build up in the space behind the unloading valve and cooperate with its biasing spring to hold the valve in a normal operating position closing off communication between the inlet 11 and the outlet passage 13.

According to this invention, the inlet section 5 is provided with an antivoid check valve 50, and a low pressure relief or regenerative pressure valve 51. These valves face one another from opposite sides of an exhaust chamber 52 which is communicated with the exhaust passage 30 in the control section 6 by a passageway not shown, but indicated by the dashed line 53.

Branch passages lead in opposite directions from the exhaust chamber 52. One of them opens to the inlet passage through a valve seat 54 normally engaged by the antivoid check valve 50, and the other opens to the outlet passage 13 through a valve seat 55 normally engaged by the low pressure relief valve 51. The seats 54 and 55 are coaxial and in register with chambers 56 and 57 in which their respective pistonlike valve members are slidably mounted.

Exhaust fluid from the passage 30 in the control section (or sections) will ordinarily flow to the exhaust chamber 52, but it

cannot pass from there to the outlet passage 13 until its pressure is great enough to unseat the relief valve 51. The low pressure relief valve thus provides a flow restricting device that obstructs exhaust flow to the outlet passage 13, and allows such flow only after the pressure of exhaust fluid has been elevated in chamber 52 to a value high enough to overcome the biasing force of the spring 58 tending to hold the low pressure relief valve closed.

While the antivoid valve is yieldingly biased toward its closed position by a spring 59, pressure fluid from the inlet passage 15 can flow into its chamber 56 through an orifice 60 to also exert closing force on the valve. Normally, all exhaust fluid entering the exhaust chamber 52 flows past the low pressure relief valve to the outlet passage, without disturbing the antivoid valve 50.

In operation, whenever the load on the cylinder 7 causes pressure fluid to be expelled from one end thereof at an abnormally high rate, the pressure invariably drops in the other end of the cylinder because of the inability of the pump to supply enough pressure fluid thereto. This drop in pressure on the supply side is accompanied by a proportionate rise in pressure on the exhaust side, because of the flow obstructing effect of the low pressure relief valve. As soon as the pressure of exhaust fluid in chamber 52 is at a high enough value, it unseats the void control valve 50 before pressure in the supply passage 15 drops to atmospheric, so that load pressurized exhaust fluid can flow into the inlet passage 15 for delivery to the starving end of the cylinder, together with pump output fluid.

Voids most often tend to form when pressure fluid is being directed into the head end of cylinder 7 and fluid is being expelled from its other at an abnormally high rate by a heavy load pulling on the piston rod to extend it. Such expelled fluid is directed into the exhaust passage 30, from whence it flows to the exhaust chamber 52 and encounters the low pressure relief valve 51. The latter causes the pressure of the exhaust fluid to build up rapidly to the value necessary to unseat the antivoid valve 50, so that exhaust fluid can then flow back to the expanding larger end of the cylinder together with pump output fluid to prevent a void from forming in the head end of the cylinder.

There are times when operating conditions require the low pressure relief valve to be biased closed by a substantially strong spring so as to maintain a substantially high pressure in the exhaust passage 30 and chamber 52 during operation of the cylinder. The pump must not only then overcome the load driven by the cylinder but it must also overcome the force of the relief valve biasing spring. It can be readily seen that this can cause severe overloading of the pump at times when the load on the cylinder is high to begin with.

According to this invention, the relief valve 51 is rendered ineffective at times when a heavy load is being driven by the cylinder, and there is no possibility of drawing a void in its expanding end. This can be accomplished as a consequence of the high pressure that then obtains in the inlet passage 15, as by a plunger 61 that responds to said pressure and physically moves the low pressure relief valve to a wide open position.

The plunger 61 is slidably mounted in a hole 62 in the antivoid valve, coaxial therewith, so as to have one end inside its chamber 56 to be subjected to the pressure of inlet fluid therein. The other end of the plunger projects toward and into endwise abutting engagement with a stem 63 on the low pressure relief valve.

It will thus be seen that whenever the pressure of fluid in the inlet passage 15 rises to a predetermined high value in consequence of a heavy load being driven by the cylinder 7, the plunger 61 will be actuated downwardly in response to the force that inlet pressure fluid in chamber 56 exerts upon its upper end, to push against the stem 63 on the low pressure relief valve 51 and open it. As soon as the load on the cylinder eases, the biasing spring for the relief valve will move the same part way toward its closed position, thus moving the plunger 61 along with it to a retracted position.

A cross pin 64 extending through the lower end portion of the plunger 61 limits the extent to which it can move upwardly into the antivoid valve 50 at times when exhaust pressure exceeds inlet pressure.

Because of the disabling plunger 61 for the relief valve 51, therefore, it is possible to keep the pressure of exhaust fluid in the exhaust passage of the valve mechanism at a desired substantially high value, as for example up to 500 lbs. p.s.i., without danger of overloading the pump at times when a heavy load is being driven by the cylinder 7.

The inlet section 5 may be designed to make use of exhaust pressure fluid for stabilizing the pressure responsive valve mechanisms at times when the load tends to drive the cylinder. At such times, pressure in the inlet 11 drops to a value at which it is no longer able to hold one of the spool valves in its feed position or to hold the unloading valve against fluttering; while exhaust pressure rises.

For this purpose, the stem 63 on the low pressure relief valve is provided with a cross bore 67 opening to the exhaust chamber 52 and communicating with an axial passage 68 leading through the relief valve to the chamber 57 in which it operates. A passageway 69 in the body 9 communicates the chamber 57 with the port 46 and hence with the space in the unloading valve chamber 66 behind the unloading valve. A check valve 70 in passageway 64 allows high pressure exhaust fluid in chamber 57 to flow to the chamber 66 and firmly hold the unloading valve 47 therein in its closed position despite loss of pressure in the inlet 11 ordinarily relied upon to hold the unloading valve closed; and such load pressurized exhaust fluid also flows to the pressure chamber of that spool which is in its feed position to assure against return of said spool to its neutral position.

Hence, a portion of passageway 69 which extends from the check valve 70 to the port 46 is always in communication with the inlet 11 through the hollow interior of the unloading valve and the orifice 49 in its face. This feature is quite important to the FIG. 2 embodiment of the invention now about to be described.

The FIG. 2 version of the invention is like that previously described in that the low pressure relief valve 72 also least obstructs flow of exhaust fluid to the outlet passage 13 at times when the cylinder is driving a heavy load and high pressure obtains in the inlet 11, and obstructs exhaust flow to the outlet to a greater extent when the load drives the cylinder and low pressure obtains in the inlet 11 and high pressure obtains in the exhaust chamber 52.

The low pressure relief valve 72 is again located opposite an antivoid check valve 73, and the exhaust chamber 52 is similarly situated between the two valves. Hence, the antivoid valve controls communication between the exhaust chamber 52 and the inlet passage as before; while the relief valve 72 controls communication between the exhaust chamber 52 and the outlet passage 13.

In this case, however, the exhaust chamber 52 is communicable with the relief valve chamber 57 through a passageway 74 that opens to chamber 57 through a control port 75 therein. The chamber 57 is also communicable with the outlet passage 13 through an orifice 76 in the cylindrical side wall 77 of the relief valve. The orifice 76 is much smaller in diameter than the control port 75 so that exhaust fluid under pressure from the chamber 52 can flow into the chamber 57 at a rate faster than it can be exhausted therefrom through the orifice 76. Such exhaust fluid entering chamber 57 can then exert force on internal surfaces of the relief valve having an area somewhat smaller than that of its face 78, so that the relief valve will then open only when the force which exhaust fluid at a substantially high pressure exerts upon the face 78 of the valve exceeds the combined forces of the biasing spring 58 and that which exhaust fluid at the same high pressure exerts upon the internal surfaces of the relief valve.

Communication of the exhaust chamber 52 with the relief and unloading valve chambers is governed by a special check valve 80 which takes the place of the check valve 70 described

hereinbefore. The valve 80 has a stem 81 that slidably fits the passageway 74 and closes it off from the control port 75 when the check valve is in a closed position engaging a seat 82. The seat 82 opens to a chamber 83 formed as an enlargement of the passage 69 which, as described earlier, communicates with the inlet 11. A spring 84 yieldingly holds the check valve on its seat.

It will thus be seen that the check valve 80 will be held on its seat not only by the force of its spring 84, but also by a force which high pressure in the inlet 11 exerts thereon at times when the controlled cylinder encounters heavy loads. It will then close off the control port 75 for the relief valve chamber to assure that the orifice 76 will vent the interior of chamber 57 to the reservoir through outlet passage 13. Consequently, relief valve 72 will then offer least resistance to exhaust flow to the outlet passage 13, to prevent overloading of the pump at times when all of its effort is needed to drive the load on the cylinder.

When the opposite condition is encountered, namely at times when the load drives the cylinder, exhaust fluid is returned to chamber 52 at an abnormally high rate, and pressure decreases proportionately in the supply passages, the check valve 80 is caused to open under force exerted thereon by load pressurized exhaust fluid to let such exhaust fluid flow into the relief valve chamber 57. The force which such load pressurized exhaust fluid in chamber 57 exerts upon the internal surfaces of the relief valve opposes the opening force which exhaust fluid at the same pressure exerts upon the larger area face 78 of the relief valve, so that the valve in effect serves to obstruct exhaust flow to the outlet to a considerably greater extent than before, when high pressure obtained in the inlet and supply passages. This, of course, serves to raise the pressure in the exhaust chamber 52 to the desired high value for recirculation back to the inlet passage 15 for void prevention.

It is to be observed that the stem 81 on the check valve 80 is formed with a passage 85 which provides for flow of exhaust fluid at high pressure to the passage 69 leading to the unloading valve chamber and to the pilot valve, whenever the check valve 80 is opened by such high pressure exhaust fluid.

In the embodiment of the invention seen in FIG. 3, the inlet connected passage 69 is the same as in the FIG. 1 embodiment, and the special check valve 80 is mounted within a tube 88 fixed with respect to the body and projecting coaxially through the chamber 57 and through a hole 89 in the face of the relief valve 90. Hence, it can be said that the relief valve 90 is in the nature of a piston that operates within an annular cylinder 91 containing the valve biasing spring 92.

The cylinder 91 is vented to the outlet passage 13 through the orifice 76 in the cylindrical side wall 77 of the relief valve as before, but exhaust fluid from chamber 52 flows to the cylinder 91 through the open upper end of the tube 88 and radial holes 94 in its wall that are normally closed by the stem 81 on the check valve.

As shown, the check valve seat is located within the tube, and when the check valve opens in response to return of exhaust fluid to chamber 52 at an abnormally high rate, some of the exhaust fluid flows past the check valve, as before, to the chamber containing the unloading valve and to the pilot valve. The remainder of such exhaust fluid flows into the cylinder 91 to exert greater closing bias thereon.

From the foregoing description, together with the accompanying drawings, it is believed that it will be apparent to those skilled in the art that this invention provides a fluid flow control instrumentality having an antivoid valve and featuring a return flow restricting mechanism which acts to minimize obstruction to exhaust flow when high pressure obtains on the supply side of the instrumentality and to obstruct exhaust flow to a greater extent when low pressure obtains on the supply side and exhaust fluid is returned to the instrumentality at an abnormally high rate.

I claim:

1. A fluid flow control instrumentality having an exhaust passage to receive fluid expelled from the contracting side of a fluid motor, a member to obstruct exhaust flow from said passage to an outlet so as to cause increase in the pressure of exhaust fluid returning to the exhaust passage at an abnormally high rate, and a void control check valve member that opens in response to said increased pressure of exhaust fluid to pass such fluid back to a pressure fluid supply passage from which the expanding side of the motor is fed, characterized by:
 - A. said flow obstructing member being movable toward a nonobstructing position, against yielding bias, in response to pressure of fluid in the exhaust passage; and
 - B. fluid pressure responsive control means sensitive to the pressure of fluid in the supply passage and cooperating with said flow obstructing member to so govern its effectiveness as to cause it to least obstruct exhaust flow to the outlet as a consequence of the effect upon the control means of high fluid pressure in the supply passage, when fluid pressure in the exhaust passage is at a low value, and to cause the flow obstructing member to obstruct exhaust flow to the outlet to a greater extent when said last named pressure values are reversed.
2. The fluid flow control instrumentality of claim 1, wherein said flow obstructing member comprises a low pressure relief valve biased toward a seated position closing off exhaust flow to the outlet.
3. The fluid flow control instrumentality of claim 2, further characterized by:
 - A. an exhaust chamber to which the exhaust passage opens, from which exhaust fluid can flow to the outlet when said relief valve is open, and from which exhaust fluid can flow to the supply passage when the void control valve is open; and
 - B. means mounting the void control and relief valve members in opposing relation to one another across said exhaust chamber.
4. The fluid flow control instrumentality of claim 3, further characterized by:
 - A. a pressure chamber for the void control valve member which is communicated with the supply passage so that fluid from the latter can fill said pressure chamber and hold the void control valve member closed when high pressure obtains in the supply passage; and
 - B. said control means comprising a plunger axially slidably mounted in the void control valve member with one end portion in said pressure chamber and its other end portion projecting toward the relief valve member to engage the same and actuate it toward a nonobstructing position in consequence of high pressure in the supply passage.
5. The fluid flow control instrumentality of claim 1, further characterized by:
 - A. said movable flow obstructing member comprising a piston;
 - B. a cylinder in which the piston operates, and into which the piston is slidable toward its nonobstructing position;
 - C. passage means to communicate said cylinder with the exhaust passage;
 - D. orifice means by which the interior of the cylinder can be maintained at outlet pressure except when high pressure fluid from the exhaust passage can flow thereinto through said passage means; and
 - E. said control means comprising a valve plunger that is movable to a position blocking said passage means in response to high pressure in the supply passage, said plunger being sensitive to pressure of fluid in the exhaust passage to be movable in response to high exhaust fluid pressure therein to a position communicating the exhaust passage with the interior of the cylinder through said passage means.
6. The flow control instrumentality of claim 5, further characterized by:
 - A. said passage means opening to the cylinder through an outer portion of the piston; and
 - B. said control plunger being mounted in the piston.

7. A fluid flow control instrumentality having a body with an inlet passage and an outlet passage, characterized by the following:
 - A. an exhaust passage having a pair of branches,
 1. one of said branches opening to the inlet passage through a first valve seat that faces away from the exhaust passage,
 2. and the other branch opening to the outlet passage through a second valve seat that faces away from the first seat;
 - B. a void control valve member yieldingly biased toward a closed position engaging said first seat but capable of actuation to an open position in response to a predetermined pressure differential that is produced when inlet fluid pressure drops to a value below that of return fluid in the exhaust passage;
 - C. a relief valve yieldingly biased toward a closed position engaging said second seat, for obstructing flow of exhaust fluid to the outlet passage to thereby cause return fluid pressure in the exhaust passage to more quickly effect opening of the void control valve at times when pressure of fluid in the inlet passage is diminishing and exhaust fluid is returned to the exhaust passage at an abnormally high rate; and
 - D. control means cooperating with said relief valve for effecting movement thereof to a nonobstructing position in response to high fluid pressure in the inlet passage.
8. A fluid flow control instrumentality having a body with an inlet passage and an outlet passage, characterized by:
 - A. an exhaust passage having a first branch leading back to the inlet passage and a second branch leading to the outlet passage;
 - B. corresponding first and second valve seats in said branches, facing away from the exhaust passage;
 - C. corresponding first and second fluid pressure responsive valve members cooperable with said seats and spring biased toward closed positions engaging the same, the second valve member providing an obstruction to flow of exhaust fluid to the outlet passage; and
 - D. and control means operable in response to high pressure in the exhaust passage for effecting increase in the closing bias on said second valve member.
9. The flow control instrumentality of claim 8, wherein said last named means is operable to effect translation of high pressure exhaust fluid into closing force upon the second valve member.
10. The fluid flow control instrumentality of claim 9, further characterized by:
 - A. said second valve member comprising a piston having a cylindrical sleeve with an orifice opening to the outlet passage;
 - B. a cylinder in which the piston operates and normally vented to the outlet passage through said orifice;
 - C. a passageway communicating the cylinder with the exhaust passage;
 - D. said control means comprising a fluid pressure responsive plunger normally occupying a position closing off said passageway but movable out of said position in response to force exerted thereon by high fluid pressure in the exhaust passage.
11. The fluid flow control instrumentality of claim 10, wherein said plunger is subject to pressure of fluid in the inlet passage and maintained thereby in said passageway closing position except when pressure of exhaust fluid rises to a value exceeding that of inlet fluid.
12. The fluid flow control instrumentality of claim 10, further characterized by:
 - A. said passageway being provided by a tube in the cylinder having a forward end portion projecting coaxially through the piston and opening to said second branch of the exhaust passage, said tube having its rearward end fixed to and closed by the body and having a port in its wall opening to the cylinder;

- B. said plunger being biased toward a position closing said port and having a first portion upon which return fluid at high pressure in the exhaust passage can act to move the plunger out of its normal position and having a second portion upon which pressure fluid in the rearward portion of the tube can act tending to hold the plunger seated; and
- C. passage means communicating the inlet passage with the space inside the rearward portion of the tube, so that inlet fluid at normally high pressure values holds the plunger in its normal port closing position.
13. A fluid flow control instrumentality having a body characterized by:
- A. laterally opposite inlet and exhaust chambers each having branches extending outwardly in opposite directions therefrom;
- B. an inlet passage at one side of said chambers joining with one branch of each chamber;
- C. an outlet passage at the opposite side of said chambers joining with the other branches thereof;
- D. a first pair of valve members, one for each chamber and controlling fluid flow therefrom to the inlet passage, said valve members being adapted to open against yielding bias in response to pressure in their respective chambers of a value exceeding the pressure of fluid in the inlet passage;

- E. a second pair of valve members, one for each chamber and controlling fluid flow therefrom to the outlet passage, said last designated valve members being adapted to open against yielding bias in response to pressure in their respective chambers of a value exceeding the pressure in the outlet passage;
- F. a pressure chamber in which each of said last designated valve members operates;
- G. one of said outlet controlling valve members comprising an unloading valve having an orifice to communicate its pressure chamber with the inlet chamber so that inlet fluid under pressure can hold the valve closed except at times when its pressure chamber is vented;
- H. the other of said outlet controlling valve members comprising a regenerative pressure valve which obstructs flow of fluid from the exhaust chamber to the outlet passage and functions whenever exhaust fluid is returned to said exhaust chamber at an abnormally high rate to cause exhaust fluid pressure to force open the valve member governing flow from the exhaust chamber to the inlet passage; and
- I. means responsive to high pressure of fluid at the inlet chamber for effecting reduction in the degree to which flow from the exhaust chamber to the outlet passage is obstructed by said regenerative pressure valve.