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(54) Improvements in hydraulic pressure converters

(57) An hydraulic pressure converter incorporates a differential intensifier piston (3) which is reciprocable in a bore (2) in a housing (1) to cause a pump-rod (4) defined by the portion of the piston (3) which is of smaller area to draw fluid into a pump chamber (9) and discharge fluid from the chamber (9). Reciprocation of the piston (3) is determined by a supply of hydraulic-operating fluid under pressure for generating a force to urge the piston (3) in a first direction, and means (35) are provided for urging the piston (3) in a second opposite direction when the force is reduced by a control valve (16) responsive to displacement of the piston (3) in the said first direction. The piston (3) is provided at the end remote from the pump-rod (4) with a tail-rod (5) which works in a third portion (12) of the bore (2) to balance the pump-rod (4).

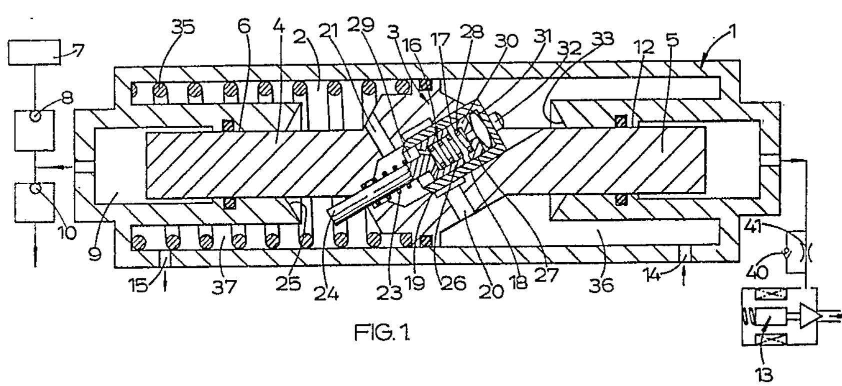

FIG. 1

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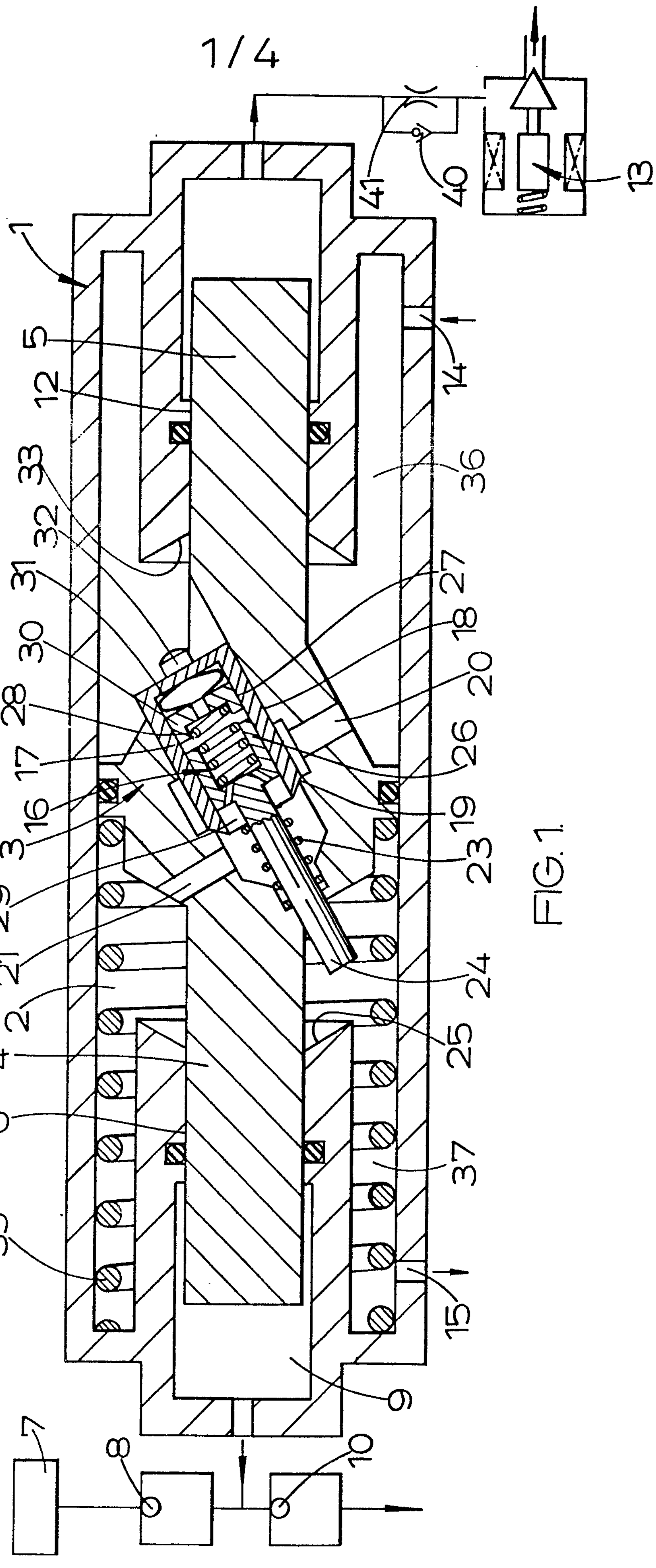

1
5
12
33
32
31
30
28
17
16
3
29
21
2
4
6
35
36
27
18
20
26
19
23
24
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37
15
9
14
41
40
13
7
8
10

FIG.1

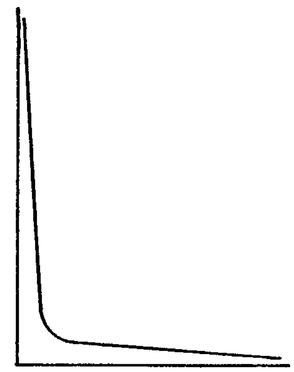

FIG. 2.

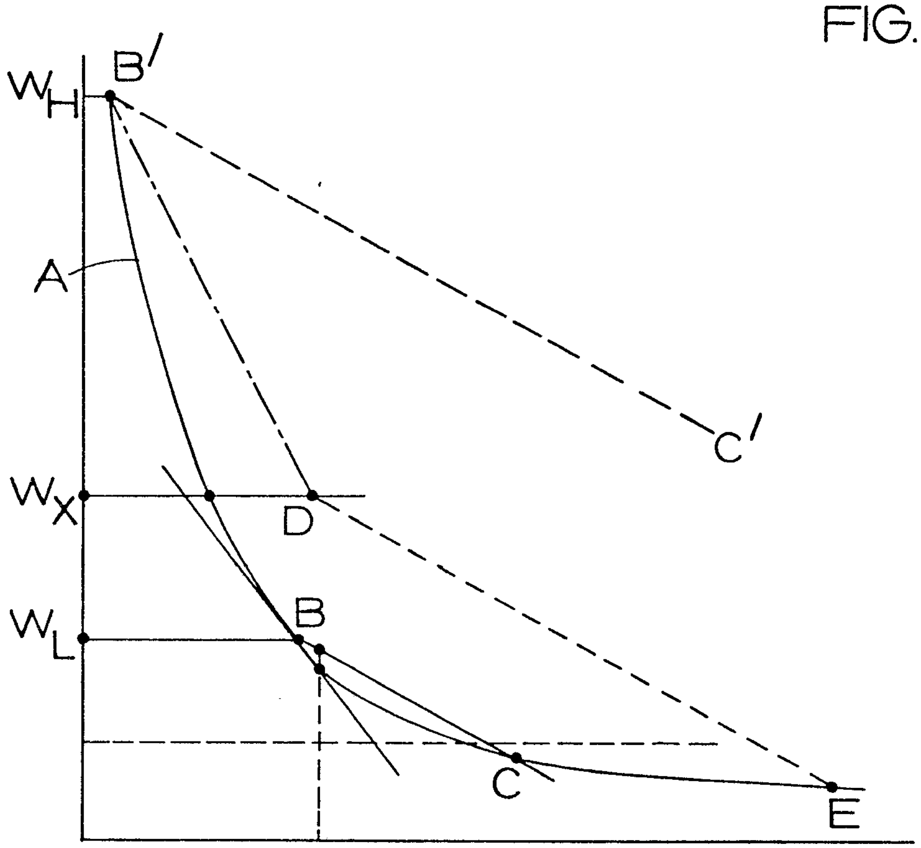

FIG. 3.

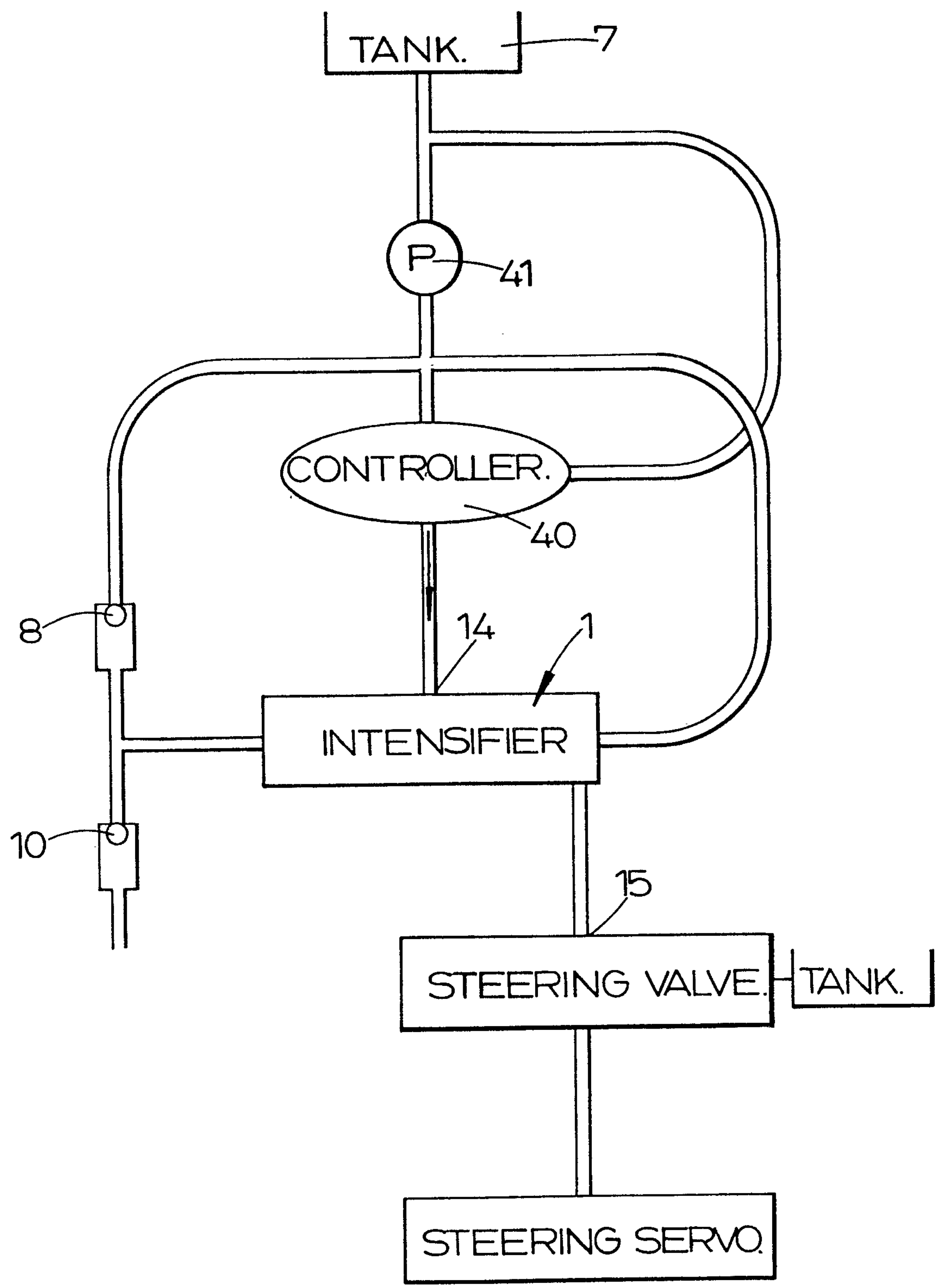
TANK.
7
P
41
CONTROLLER.
40
8
14
1
INTENSIFIER
10
15
STEERING VALVE.
TANK.
STEERING SERVO.

FIG.4.

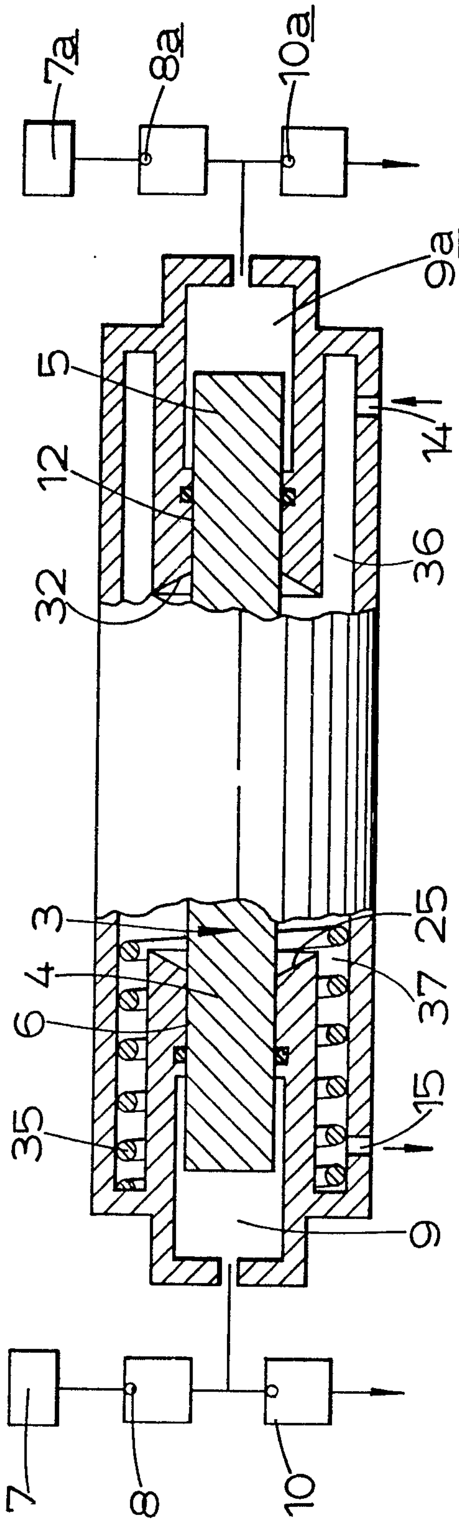

FIG. 5.

SPECIFICATION

Improvements in hydraulic pressure converters

This invention relates to hydraulic pressure converters of the kind for generating an intensified or increased hydraulic pressure in a pressure space in response to a lower hydraulic pressure supply.

In vehicles it is generally considered too expensive to provide a high pressure pump for use in a braking system alone. Thus it is desirable to be able to utilise a pump already available in a vehicle, for example a power steering, or lubrication pump, or a pump provided for other ancilliary equipment, such as a self-levelling suspension mechanism. This, in itself, creates difficulties since a pump already installed in a vehicle may not be entirely suitable for powering a braking system.

It is known to provide, in such systems, hydraulic pressure converters which incorporate a piston adapted to generate an intensified or increased hydraulic pressure for use in the braking system, in response to the lower pressure of the supply from the existing steering pump or the like.

In the hydraulic pressure converter described in our GB Patent Specification No. 2,016,587 the piston comprises a differential piston which works in a bore of corresponding stepped outline to define a pressure generating chamber at the end of the piston which is of smaller area, and a pair of actuating chambers on opposite sides of the portion of the piston which is of greater area, the actuating chambers being connected to a pump and to a steering control valve respectively and being interconnected by a by-pass passage. The piston is caused to reciprocate in the bore by the action of a control valve which is operated at the opposite extremes of piston travel alternately to open and close the by-pass passage such that, when the valve is closed the piston is driven in one direction, against the action of a biassing force, by the pressure of the fluid from the pump acting in one of the actuating chambers, thus displacing fluid simultaneously from the other actuating chamber to the steering control valve, whereas, when the control valve is open, the biassing force is effective to drive the piston in the other direction whilst fluid from the pump flows directly to the steering control valve through the by-pass passage.

Since the intensified or increased pressure depends upon pressure generated in an existing pump, any fluctuations in the output from the pump will result in magnified pulsations being produced by the pressure converter.

According to our invention in an hydraulic pressure converter of the kind set forth comprising a differential intensifier piston working in a stepped bore in a housing and a pressure space defined in the end of the housing which is of smaller area between the bore and a pump-rod defined by the portion of the piston which is of smaller area, the piston being adapted to be reciprocated in the bore whereby the pump-rod is adapted to draw hydraulic fluid from an inlet into the pressure space through a first one-way valve and to pump hydraulic fluid from the pressure space to an outlet through a second one-way, and means for reciprocating the piston in the bore, the means comprising a supply of hydraulic-operating fluid under pressure for generating a force to urge the piston in a first direction away from a datum position, valve means responsive to the displacement of the piston in the first direction for reducing the magnitude of the force when the displacement attains a predetermined value, and means for urging the piston in a second direction opposite to the first when the force is reduced to the smaller value by the valve means, the piston is provided at the end remote from the pump-rod with a tail-rod which works in a third portion of the bore to balance the pump rod.

Preferably the tail-rod has an area substantially equal to that of the pump-rod.

Our invention has the advantage that a chamber defined between the tail-rod and the third portion of the bore may be adapted to act to damp or oppose the return movement of the piston, thereby reducing the return stroke velocity of the piston to a value below which neither the flow of fluid through the open valve means, nor any consequent back pressure, can cause premature re-closure of the valve means.

One embodiment of our invention and a modification are illustrated in the accompanying drawings in which:—

Figure 1 is a section through an hydraulic pressure converter;

Figure 2 is a graph of valve load plotted against valve opening travel;

Figure 3 is the graph of Figure 2 on an enlarged scale and condensed for clarity;

Figure 4 is a layout of a modified system; and

Figure 5 is a layout of a double-acting version.

The pressure converter illustrated in Figure 1 comprises a housing 1 having a bore 2 in which works an intensifier piston 3. The piston 3 is provided with axially extending oppositely directed pump and tail rods 4 and 5 respectively of equal areas.

The pump-rod 4 works in a bore portion 6 to draw fluid from a tank 7 and through a first one-way valve 8 and into a pump chamber 9 on inward movement of the rod 4, and discharge the fluid from the pump chamber 9 and to a brake circuit through a second one-way valve 10, upon movement of the rod 4 in the opposite direction.

The tail-rod 5 works in a similar bore portion 12 at the opposite end of the housing 1 of which the outer end is connected to tank through a normally closed solenoid operated valve 13.

An inlet port 14 on the side of the piston adjacent to the tail-rod 5 is connected to a steering pump of a vehicle, and an outlet port 15 on the opposite side of the piston 3 is connected to the steering valve.

Communication between chambers 36, 37 on opposite sides of the piston 3 is controlled by a control valve 16. The control valve 16 comprises a valve member 17 which works in an inclined bore 18 traversing the piston 3 and engageable at its inner end with a seating 19 to control communication between complementary passages 20 and 21. The valve member 17 is normally urged away from the seating 19 by a compression return spring 23.

A stem 24 projecting from the valve member 17 in a direction towards a cam surface 25 surrounding the inner end of the bore 6 carries a head 26 which works in the bore 27 of the valve member 17 and a compression spring 28 normally urges the head 26 against a stop member 29 in the adjacent end of the bore 27 and which also forms an abutment for the return spring 23.

A floating piston 30 also slides in the bore of the valve member 17, and is interposed between the spring 28 and a Belleville or washer like spring 31.

A second stem 32 projects from the end of the valve member 17 remote from the stem 24 in a direction towards a second cam surface 33 which surrounds the inner end of the bore 5.

In an inoperative position the piston 3 is held in a retracted position by a large compression return spring 35 and, in this position, the valve 16 is held in a closed position by the engagement of the stem 32 with the cam surface 33.

In operation, fluid from the power steering pump is supplied to the inlet 14, and acts in the chamber 36 to move the piston 3 away from the cam surface 33. Movement of the piston 3 is opposed by the spring 35, the friction of piston seals, the pressure drop across the supply valve 10, and any pumping forces in chamber 9, and these opposing forces generate a pressure in chamber 36 which acts to move the piston 3, and also acts on the end of the valve member 17 remote from the stem 24 to hold the valve 16 closed against the spring 23.

As the piston 3 moves to the left the stem 24 engages with the cam surface 25, and the pressure in chamber 36 acting on the valve member 17 produces a pressure force which acts to compress springs 28 and 31. The spring 28 is compressed first, until the head 26 engages the piston 39, and thereafter the spring 31 is compressed. The force in the springs 28 and 31 is also an opposing force, which acts to increase the pressure in the chamber 36. The increased pressure in chamber 36 in turn increases the pressure force acting on the valve member 17 to compress the springs 28 and 31 further. However the increase in the pressure force from chamber 36 acting on the valve member 17 is less than the increase in the spring force acting on the valve member 17, because of the relative areas of the valve member 17 and the piston 3. Thus, at some point the force in the springs 28 and 31 is greater than the pressure force acting on the valve member 17, from the chamber 36, and the valve 16 opens to allow fluid flow from the chamber 36 to the chamber 37. Details of the valve opening are shown in the graph of Figure 3.

Figure 2 shows a curve A of "valve rate" that is valve load, i.e. the hydraulic force tending to close the valve 16 due to the leftward power fluid flow—plotted against the travel of the valve member 17 in opening the valve 16. For clarity the graph of Figure 2 is condensed into Figure 3. As shown in Figure 3, the slope of a tangent drawn to this curve at any point represents the "valve rate" at that point. Sloping lines on the graph can also represent spring rates—the steeper the slope the higher the rate of the spring represented. The lower broken horizontal line between W_L and the horizontal axis represents the load of the spring 23.

If there is no output resistance for the pump 4 then the pressure in chamber 36 during a power stroke will be low, and let the pressure in chamber 36 when the valve 16 starts to open correspond to a valve load W_L on the graph. The horizontal line W_LB represents the valve opening at constant load, so that as the valve 16 opens the spring 28 is not released but the piston 3 slows down as more inlet flow is taken by the valve. At point B (on the rate curve) the whole inlet flow is taken by the valve 16 and the piston 3 stops as the valve 16 stalls. With the piston 3 stationary, the pressure in chamber 36 is dependent only on the valve opening, so that further operation is determined by the valve rate curve A. The line BC on the graph, represents the rate of the spring 28 which is lower than the valve rate at B. Any slight variation in the flow or the load from the pump chamber will render the valve 16 unstable, and it will open to C, drawing energy from the spring 28, as at any point between B and C the spring load is more than the valve load.

Now consider a similar sequence at high pressure—when there is output resistance for the pump 4. The valve load as the valve 16 starts to open will now start at W_H and the point B' corresponding to B will be higher. The line B'C' indicates a very large stroke (more than is available) would be required to compress the spring 28 to a load corresponding to W_H. This cannot be overcome by pre-loading the spring 28 because such pre-loading would provide a spring characteristic which would follow line B'C' down to the pre-load, and would then drop vertically to provide an infinite spring rate below this load. Since the spring rate must be less than the valve rate in order to open the valve, the valve would not open at loads below the pre-load. The additional high rate Belleville springs 31 can provide a solution. Their rate is represented by the line B'D on the graph and this will successfully open the valve at all loads down to W_X where the valve rate is just higher than the spring 31 rate. At lower loads the lower rate spring 28 begins to expand and causes the overall spring rate to follow the curve B'DE—successfully opening the valve with a stroke which is short in comparison with that which would have resulted from the use of a single spring having a relatively low rate.

When the intensifier is operating at very low pressures, the main resistance to leftward piston movement will be provided by the return spring 35. The pressure differential across the piston 3 caused by movement against this resistance must be large enough to hold the valve 16 closed against the opposing action of spring 23. Movement of the piston 3 to the right during the return stroke will also tend to generate a pressure differential in the same direction as that which obtained during the leftward stroke. Since the valve 16 is open during the return stroke, one might expect this pressure differential to be very small. However, in the embodiment shown in Figure 1, the return movement could be very rapid since, apart from the pressure differential, only friction of the seals oppose movement of the piston 3. Thus there is a danger that the piston speed may reach the point where the resulting pressure drop through the open valve 16 exceeds the load in spring 23, with the result that the valve 16 snaps shut and forshortens the return stroke. Means for damping this return movement must be provided. Such damping means may take any convenient form, for example damping means may be constituted by increased seal friction. It is preferred however to provide a one-way valve 40 and a restrictor 41 in parallel, with the restrictor provided in the line between the tail-rod 5 and the valve 13, as shown in Figure 1.

In the construction described above energy stored in the springs 28 and 31 is operative to open the valve 16 past the stall point during any temporary reduction in the fluid flow rate or other opposing load which the fluid is supporting.

When the solenoid-operated valve 13 is de-energised it acts as a one-way valve to trap fluid in the outer end of the bore 12. This forms an hydraulic lock to hold the intensifier in the off position.

In a modified construction as shown in the layout of Figure 4 the pump suction inlet draws from a point upstream of a flow controller 40 on the steering pump 41. This has the advantage that sufficient pressure is available to return the intensifier piston 3 to its retracted position. Thus the spring 35 can be omitted.

The advantage of providing an augmented return force is achieved only because steering control valves generally provide an intentional degree of restriction, even in a "straight-ahead" position, in order to improve the response characteristics of the valve. If this were not so, there would normally be negligible pressure in the circuit, and the modification of Figure 3 would have no benefit.

Alternatively the tail-rod 5 can be exposed to this pressure. The pump 41 would take less force to drive it, and the flow into the tail-rod would not deplete the steering valve flow since the tail-rod 5 is connected between the power steering pump 41 and the flow controller 40.

In the double-acting construction illustrated in Figure 5 the piston 3 is arranged such that the tail-rod 5 also acts as a pump rod to draw fluid from a tank *7a* and through a first one-way valve *8a* and into a pump chamber *9a* upon inward movement of the rod 5 with the piston 3 moving against the load in the spring 35, and discharge the fluid from the chamber *9a* and to a brake circuit through a second one-way valve *10a*, upon movement of the rod 5 in the opposite direction.

The construction and operation of the hydraulic pressure converter illustrated in Figure 5 is otherwise the same as that of Figure 1 and corresponding reference numerals have been applied to corresponding parts.

Claims

1. An hydraulic pressure converter of the kind set forth comprising a differential intensifier piston working in a stepped bore in a housing and a pressure space defined in the end of the housing which is of smaller area between the bore and a pump-rod defined by the portion of the piston which is of smaller area, the piston being adapted to be reciprocated in the bore whereby the pump-rod is adapted to draw hydraulic fluid from an inlet into the pressure space through a first one-way valve and to pump hydraulic fluid from the pressure space to an outlet through a second one-way, and means for reciprocating the piston in the bore, the means comprising a supply of hydraulic-operating fluid under pressure for generating a force to urge the piston in a first direction away from a datum position, valve means responsive to the displacement of the piston in the first direction for reducing the magnitude of the force when the displacement attains a predetermined value, and means for urging the piston in a second direction opposite to the first when the force is reduced to the smaller value by the valve means, in which the piston is provided at the end remote from the pump-rod with a tail-rod which works in a third portion of the bore to balance the pump rod.

2. A converter as claimed in Claim 1, in which the tail-rod has an area substantially equal to that of the pump-rod.

3. A converter as claimed in Claim 1 or Claim 2, in which a chamber defined between the tail-rod and the third portion of the bore acts to damp or oppose the return movement of the piston.

4. A converter as claimed in any of Claims 1 or 3, in which the tail-rod is exposed to pressure from a steering circuit.

5. An hydraulic pressure converter substantially as described herein with reference to and as illustrated in Figures 1 to 4 of the accompanying drawings.

6. An hydraulic pressure converter substantially as described herein with reference to and as illustrated in Figure 5 of the accompanying drawings.

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