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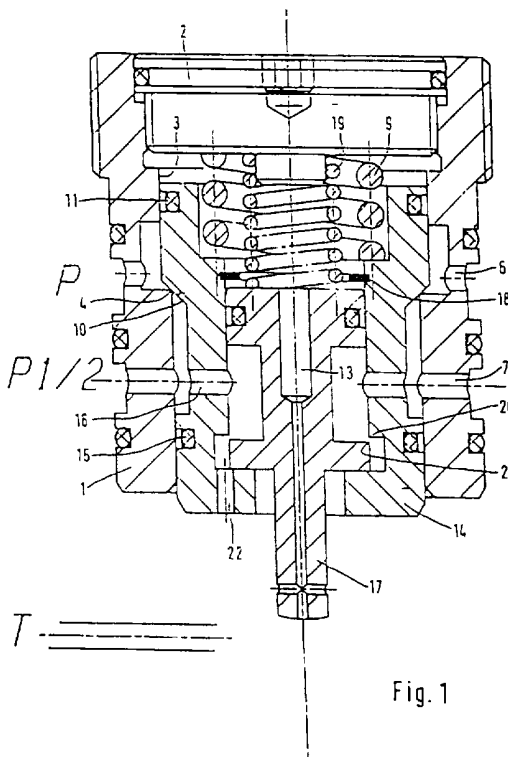
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(54) Servo valve arrangement

(57) A servo valve arrangement has a high-pressure P, low-pressure T and motor connection P1/2 and with a pressure control valve 4, 10; arranged between the high-pressure connection P and motor connection P and a return control valve 20, 21 arranged between the motor connection and low-pressure connection, the pressure control valve normally being closed and the return control valve normally being open. A common actuating member 17 closes the return control valve and opens the pressure control valve against the force of springs 9, 19. The two valve closure members are movable axially relative to each other such that movement of the actuating member closes the return valve before the pressure control valve is opened. The valve arrangement is suitable, in particular, for hydraulic power steering systems of motor vehicles with a so-called closed centre.



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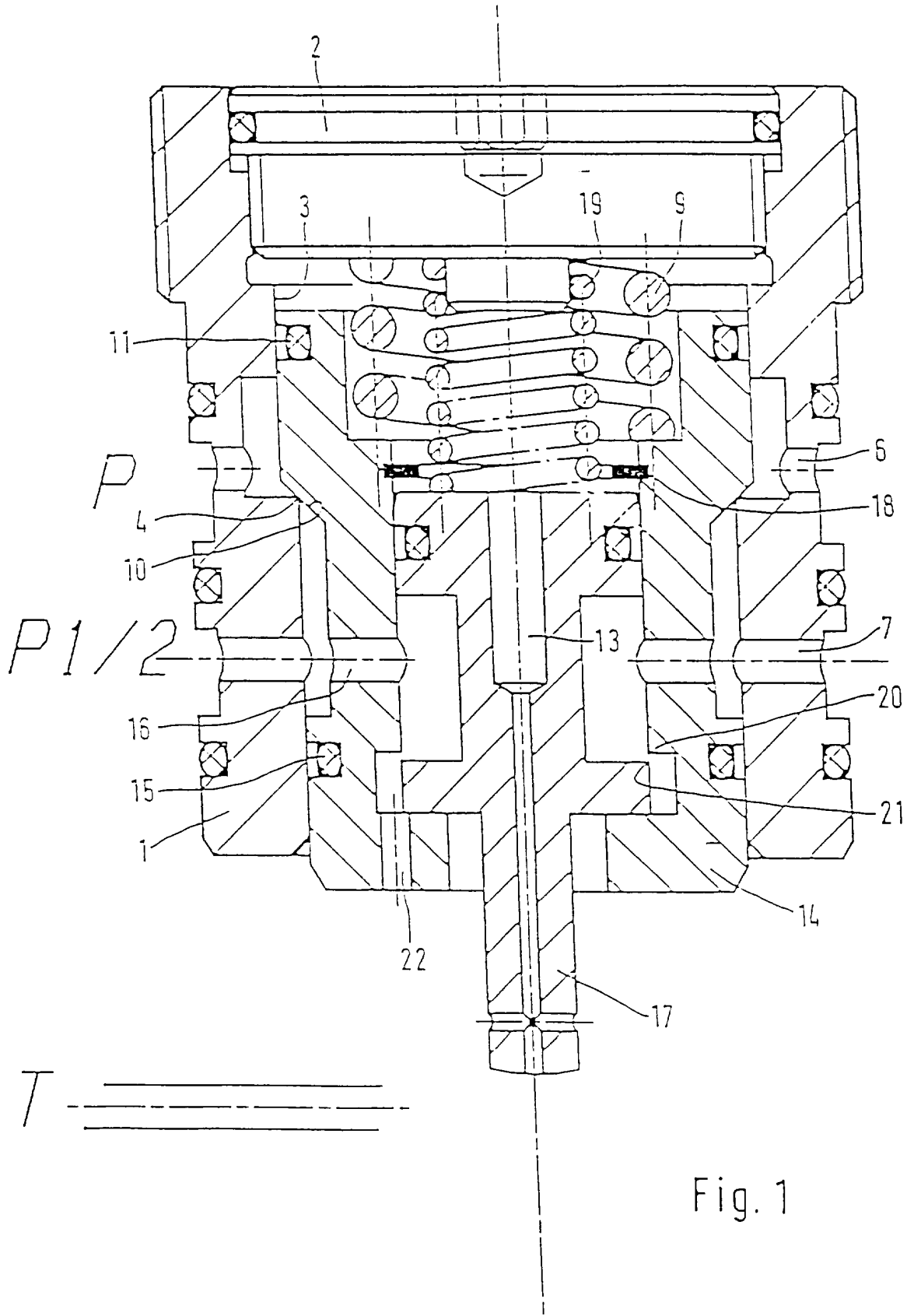


Fig. 1

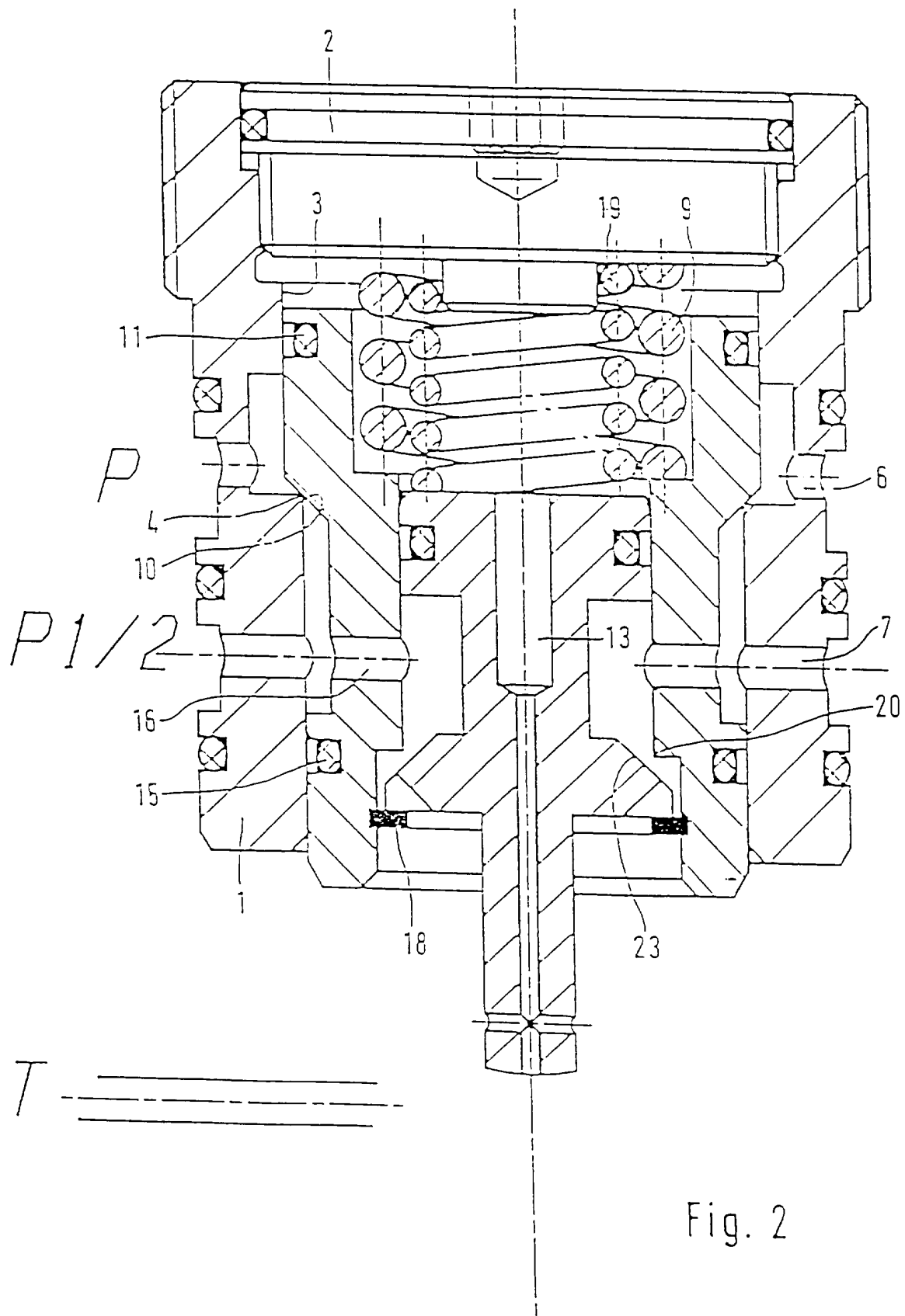


Fig. 2

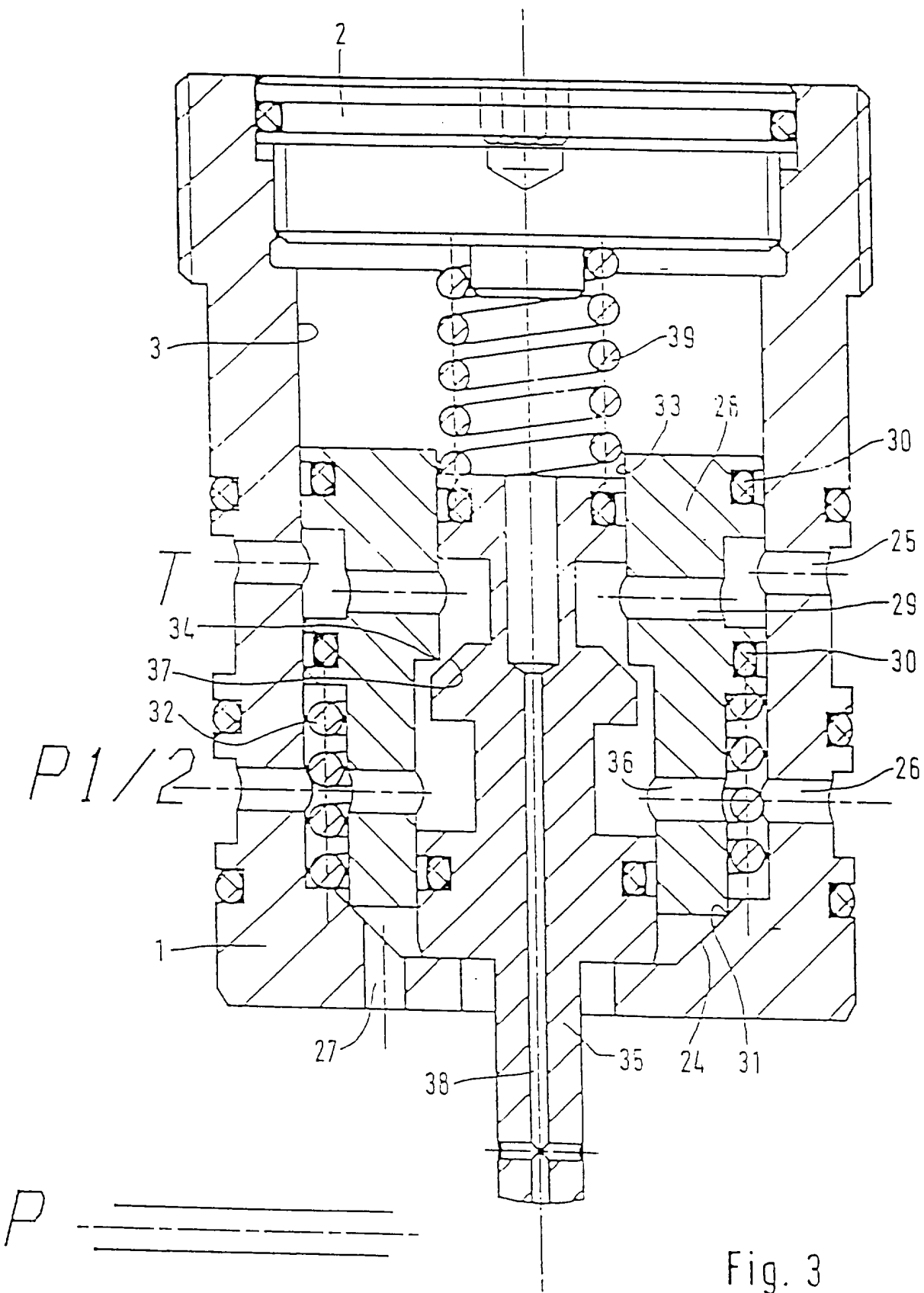


Fig. 3

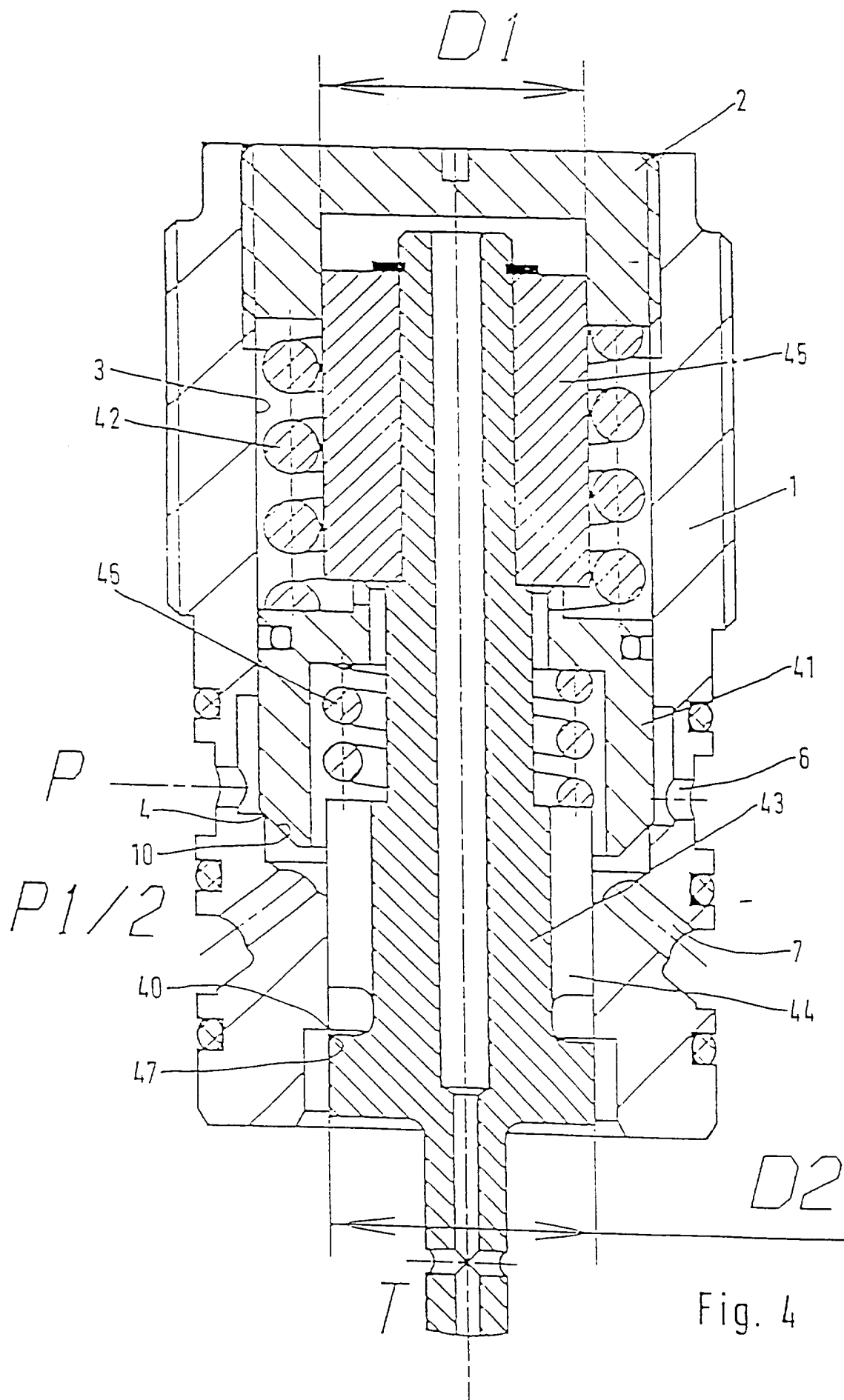


Fig. 4

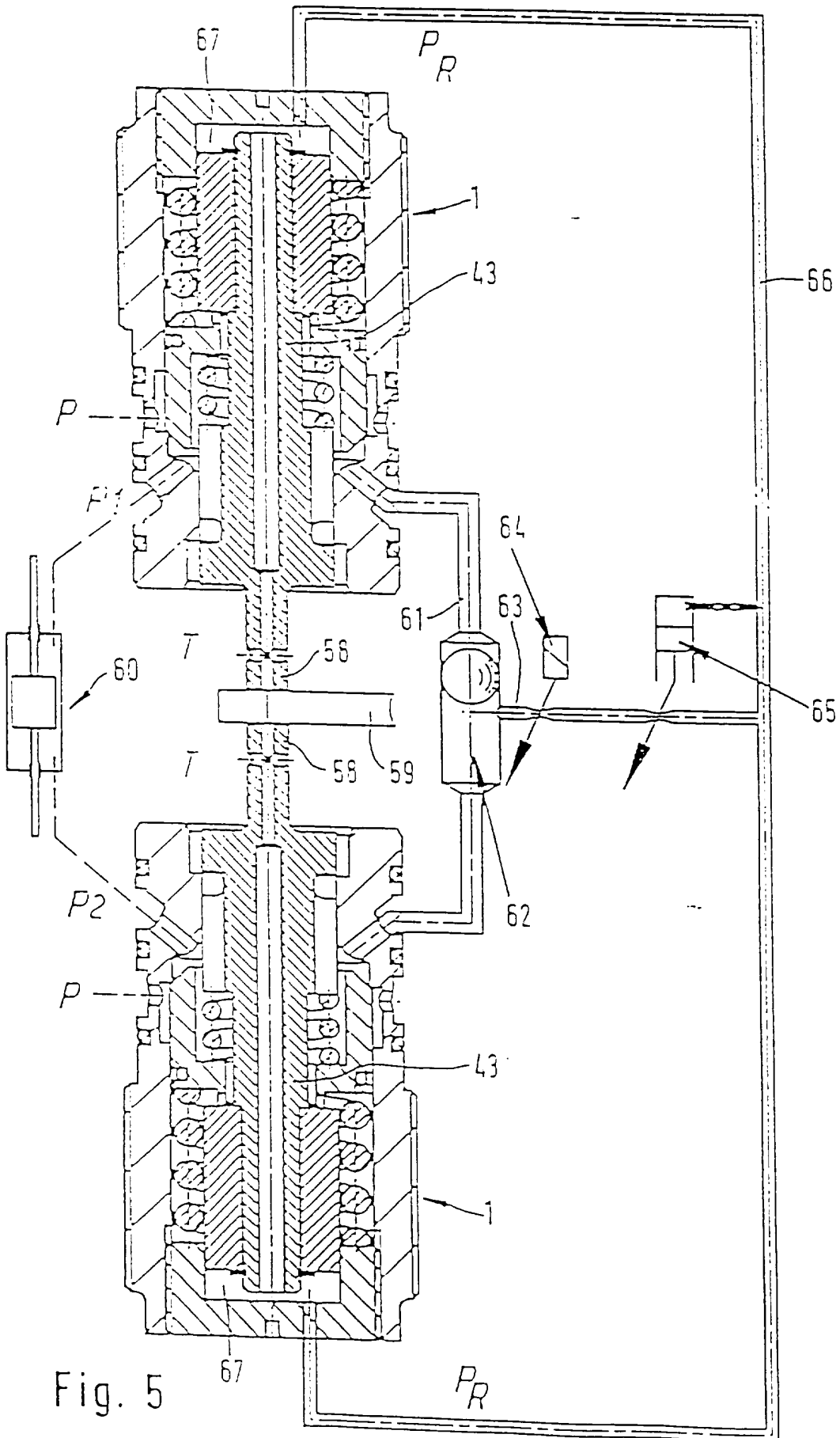


Fig. 5

Servo valve arrangement

The invention relates to a servo valve arrangement with high-pressure, low-pressure and motor connections and with a pressure control valve arranged between the high-pressure connection and motor connection and a return control valve arranged between the motor connection and low-pressure connection, the pressure control valve normally being closed and the return control valve normally being open, and both valves having axially displaceable control parts which are arranged equiaxially with one another and can be actuated by means of a common actuating member, whilst closing the return control valve and opening the pressure control valve.

In the case of hydraulic power steering systems which are currently customary in motor vehicles, hydraulic medium flows virtually constantly through the associated servo valve arrangement, specifically even when no servo force is required at all - such as, for example, in normal straight-ahead travel. The servo valve arrangement consists in principle of two parallel throttle routes which are arranged between a pressure connection and a low-pressure connection and each comprise two throttles which are controlled in opposite directions, are arranged in series, and between which a motor connection branches off in each case. Depending on the direction and magnitude of an actuating moment to be applied at the steering wheel, the input-side throttle of the one throttle route is increasingly opened and, at the same time, the output throttle of this throttle route is increasingly throttled, while the throttles of the other throttle route are controlled in the opposite direction in each case. In this way, a pressure difference which can be controlled according to direction and magnitude is generated between the motor connections of the two throttle routes, such that an appropriate servomotor generates a controllable servo force in the one direction or the other. In operating states in

which no servo force is required, all the throttles assume a central position so that the medium flows evenly through both throttle routes and there is pressure equilibrium between the motor connections.

Since, in these known power steering systems, a constant hydraulic flow has to be maintained, power is required constantly.

In terms of a reduction in the power requirement of motor vehicles, it is basically known to provide power steering systems with a so-called closed centre, i.e. as long as no servo force is required, the servomotor has no connection at all to the pressure source; in contrast, in this state the servomotor is only connected to the low-pressure connection or hydraulic reservoir in order to allow passive mobility of the servomotor.

In this context, attempts have already been made to implement power steering systems using the servo valve arrangements specified at the beginning. Essentially, two servo valve arrangements of the type specified at the beginning then have to be arranged parallel to one another between a pressure source or a pressure accumulator and a hydraulic reservoir. Depending on the actuating moment to be applied at the steering wheel, the pressure control valve of the one servo valve arrangement or the other is then opened to a greater or lesser extent so that in each case the motor connection of the one servo valve arrangement or the other is connected with controllable throttle resistance to the pressure source or the pressure accumulator and, at the same time, is blocked off relative to the low-pressure connection. The servomotor connected between the motor connections of the two servo valve arrangements thus generates a servo force in the one direction or the other.

The present invention seeks to ensure particularly good control characteristics in a servo valve arrangement of the type specified at the beginning.

According to the present invention there is provided a servo valve arrangement with high-pressure, low-

pressure and motor connections and with a pressure control valve arranged between the high-pressure connection and motor connection and a return control valve arranged between the motor connection and low-pressure connection, the pressure control valve normally being closed and the return control valve normally being open, and both valves having axially displaceable control parts which are arranged equiaxially with one another and can be actuated by means of a common actuating member, whilst closing the return control valve and opening the pressure control valve, wherein the pressure control valve comprises a seat valve and the return control valve is normally throttled.

According to a particularly preferred alternative embodiment of the invention, provision is made for the control part of the return control valve to be displaceable to a limited extent relative to the control part of the pressure control valve and to be coupled thereto in terms of drive in such a way that the pressure control valve only opens after the return control valve has closed. In this way, it can be guaranteed for virtually all operating states that the pressure connection and low-pressure connection remain separated from one another and virtually no hydraulic power losses whatsoever occur.

Advantageous embodiments of the invention will now be described by way of example with reference to the drawings, in which:-

Figures 1 to 4 show diagrams in axial section of different embodiments of the servo valve arrangement according to the invention;
Figure 5 shows an arrangement with a hydraulic reaction control;

In the embodiment illustrated in Figure 1, a valve sleeve 14 is guided so as to be displaceable inside the axial bore 3 of the valve cartridge 1, which valve sleeve is seated in the normal position with the conical section 10 on the annular edge 4 of the axial bore 3 and is preferably

provided with sealing rings 11 and 15 axially above the radial bores 6 and axially below the radial bores 7 of the valve cartridge 1, such that the annular spaces remaining between the wall of the axial bore 3 and the valve sleeve 14 in the region of the radial bores 6 and 7 are sealed off at the top and bottom. The valve sleeve 14 is normally pressed by the helical compression spring 9 with the conical section 10 against the annular edge 4 of the valve cartridge 1 so that the radial bores 6 and 7 are separated from one another.

Arranged in the valve sleeve 14 are radial bores 16 which constantly communicate with the radial bores 7 and connect the interior of the valve sleeve 14 to the radial bores 7.

A valve slide piston 17 is arranged so as to be axially displaceable to a limited extent inside the valve sleeve 14, the possible displacement path being bounded by a stop ring 18 arranged in the valve sleeve 14 and a lower annular rim on the valve sleeve 14.

The valve slide piston 17 is tensioned in the downward direction by means of a helical compression spring 19 whose upper end is supported on the base part 2 of the valve cartridge 1, such that the valve slide piston 17 attempts to assume the lower end position illustrated in the valve sleeve 14.

The valve slide piston 17 has a central section of small diameter, such that an annular space is formed axially between the ends of the valve slide piston 17 inside the valve sleeve 14, which annular space is connected to the radial bores 16 and at whose lower end in the valve sleeve 14 an annular edge 20 is formed which, in the lower end position of the valve slide piston 17 illustrated, is separated from an annular edge 21, formed on the valve slide piston 17, by a narrow annular gap which merges in a downward direction into an annular space, connected to the low-pressure connection T, inside the valve sleeve 14. An axial bore 22 is arranged in the valve sleeve 14 to connect

this annular space to the low-pressure connection T.

In the normal position illustrated, the radial bores 7 of the valve cartridge 1, connected to the motor connection P1 or P2, are connected to the low-pressure connection T via the radial bores 16, the annular gap between the annular edges 20 and 21 and the axial bore 22. In contrast, the radial bores 7 are separated from the radial bores 6 which are connected to the pressure connection P because the conical section 10 of the valve sleeve 14 is seated on the annular edge 4 of the valve cartridge 1. If the valve slide piston 17 is now raised at its ram-like lower end counter to the force of the helical compression spring 19, the annular gap between the annular edges 20 and 21 is closed, with the consequence that the motor connection P1 or P2 is shut off relative to the low-pressure connection T. If the valve slide piston 17 is raised further, it strikes against the stop ring 18, such that, during a further upward stroke of the valve slide piston 17, the valve sleeve 14 is carried along with it, the resistance of the helical compression spring 9 having to be overcome additionally. As soon as the conical section 10 is then lifted off the annular edge 4, the motor connection P1 or P2 is connected to the pressure connection P.

Since an axial bore 13 with a throttle effect is again arranged in the valve slide piston 17, the exchange of hydraulic medium again takes place between the space receiving the helical compression springs 9 and 19 and the low-pressure connection T counter to a specific throttle resistance which has a damping effect on the stroke movements of the valve sleeve 14 and the valve slide piston 17.

The embodiment illustrated in Figure 2 differs from the embodiment according to Figure 1 above all in the fact that a cone 23 is arranged on the valve piston 17 instead of the annular edge 21, which cone interacts with the annular edge 20 of the valve sleeve 14 and, together with the stop ring 18 arranged at the lower end of the valve

sleeve 14 in Figure 3, bounds the axial mobility of the valve piston 17 in the valve sleeve 14.

The functioning corresponds to a very great extent to the functioning of the embodiment according to Figure 1. If the valve piston 17 is raised, the annular gap between the cone 23 and the annular edge 20 is closed, such that the motor connection P1 or P2 is separated from the low-pressure connection T. With a further upward movement of the valve piston 17, the valve sleeve 14 is carried along with it, thus causing the pressure connection P to be connected to the motor connection P1 or P2.

In the embodiment illustrated in Figure 3, the axial bore 3 of the valve cartridge is designed as a cylindrical bore which is continuous to a very great extent, whose one end is closed by the base part 2, and whose other end is adjoined by an inner cone 24.

With axial spacing from one another, radial bores 25 which lead to a low-pressure connection T and radial bores 26 are arranged on the valve cartridge 1, which communicate with one of the motor connections P1 or P2. An axial opening 27 which communicates with the pressure connection P is disposed at the radially inner edge of the inner cone 24.

Accommodated with sliding displaceability inside the axial bore 3 is a valve sleeve 28 which has, in the region of the radial bores 25, a wide circumferential groove which communicates on the one hand with the radial bores 25 of the valve cartridge 1 and on the other hand with radial bores 29 of the valve sleeve 28. On both sides of the said circumferential groove, there are sealing rings 30 on the valve sleeve 28, which seal off the gap between the valve sleeve 28 and the axial bore 3 of the valve cartridge 1.

In the region of the radial bores 26 of the valve cartridge 1, the valve sleeve 28 has a reduced diameter in such a way that an annular space is formed between the valve sleeve 28 and the wall of the axial bore 3 of the valve cartridge 1, which annular space extends up to the inner

cone 24 of the valve cartridge 1, on which inner cone the valve cartridge 1 is seated with an annular edge 31 in the lower end position illustrated in Figure 3, such that the axial opening 27 is shut off relative to the radial bores 26. Arranged in the abovementioned annular space is a helical compression spring 32 which attempts to push the valve sleeve 28 upwards in Figure 3. Radial bores 36 are otherwise arranged in the region of this annular space in the valve sleeve 28.

The axial inner bore 33 of the valve sleeve 28 has an upper section of smaller diameter and a lower section of greater diameter. An annular step 34 is arranged between these sections.

A valve piston 35 is guided so as to be displaceable inside the inner bore 33, a piston-like upper end being sealed off with a seal relative to the wall of the inner bore 33 in the upper section of the inner bore 33 above the radial bores 29. In a corresponding manner, a lower, piston-like end of the valve piston 35 is sealed off by means of a sealing ring in the lower section of the inner bore 33 below the radial bores 36 of the valve sleeve 28.

Between its piston-like ends, the valve piston 35 is designed with a small diameter in the region of the radial bores 29 and 36 of the valve sleeve 28 in such a way that annular spaces are formed in the region of the radial bores 29 and 36 inside the valve sleeve 28 on both sides of a valve cone part 37 formed on the valve piston 35. The valve cone part 37 interacts with the annular step 34 of the valve sleeve 28 between the radial bores 29 and 36. In this case, the valve cone part 37 is arranged on the valve piston 35 in such a way that a gap remains between the valve cone part 37 and the annular step 34 as long as both the valve piston 35 and the valve sleeve 28 are located in the lower end positions illustrated.

The valve piston 35 is penetrated by an axial channel 38 which is throttled if appropriate, such that the space above the valve sleeve 28 and the valve piston 35

inside the valve cartridge 1 communicates with the pressure connection P.

The valve piston 35 is otherwise tensioned in the downward direction by a helical compression spring 39.

The pressure starting from the pressure connection P above the valve sleeve 28 causes the latter to try to sit with its annular edge 31 on the inner cone 24 counter to the force of the helical compression spring 32.

As long as the valve piston 35 then assumes its lower end position illustrated, the motor connection P1 or P2 and the low-pressure connection T are connected to one another via the radial bores 25 and 26 of the valve cartridge 1 and the radial bores 29 and 36 of the valve sleeve 28 and the annular gap between the valve cone part 37 and the annular step 34.

If the valve piston 35 is now raised counter to the force of its helical compression spring 39, the aforementioned annular gap is closed, i.e. the low-pressure connection T is separated from the motor connection P1 or P2. If the valve piston 35 is now raised further, the annular edge 31 of the valve sleeve 28 is lifted off the inner cone 24 of the valve cartridge 1, and the motor connection P1 or P2 is connected to the pressure connection P.

Since the pressure of the pressure connection P propagates via the radial bores 36 of the valve sleeve 28 right into the annular space inside the valve sleeve 28 below the valve cone part 37, the hydraulic compression forces acting on the valve sleeve 28 in the axial direction thereof are compensated with the consequence that the helical compression spring 32 can raise the valve sleeve 28 further. As a result, the annular gap between the valve cone part 37 and the annular step 34 is again opened, such that the motor connection P1 or P2 is also temporarily connected to the low-pressure connection T until the valve sleeve 28 is again forced in the downward direction against the raised valve piston 35 owing to the pressure drop in the region of

the annular step 34 due to the hydraulic pressure in the space of the valve cartridge 1 receiving the helical spring 39, the annular gap between the valve cone part 37 and the annular step 34 again being closed.

In this way, a pressure lying between the pressure of the pressure connection P and the (diminishing) pressure of the low-pressure connection T can be set at the motor connection P1 or P2 depending on the stroke of the valve piston 35 by means of subsequent strokes of the valve cartridge 1.

As soon as the valve piston 35 has again been brought into the lower end position illustrated, the valve sleeve 28 is also displaced into the lower end position.

In the embodiment according to Figure 4, the base part 2 is arranged at the upper end of the axial bore 3 of the valve cartridge 1 so as to be axially adjustable by screwing in the manner of a setting screw. The axial bore 3 communicates with a pressure connection P via radial bores 6 of the valve cartridge 1 and with the motor connection P1 or P2 via the further bores 7 arranged radially obliquely. The lower open end of the radial bore 3 communicates with the low-pressure connection T.

Formed in the region of the radial bores 6 on the wall of the axial bore 3 is a wide inner circumferential groove which is bounded at the bottom by the annular edge 4 whose diameter is smaller than the diameter of the axial bore 3 above the abovementioned annular groove. The annular edge 4 is located axially above the obliquely radial bores 7.

Axially below the bores 7, the axial bore 3 widens in the manner of a step at an annular edge 40.

Arranged with sliding displaceability in the axial bore 3 in the region axially above the annular edge 4 is a valve sleeve 41 which is forced in the downward direction by a helical compression spring 42, supported on the base part 2, in such a way that the conical section 10 arranged at the lower end face of the valve sleeve 41 is seated on the

annular edge 4 of the axial bore 3.

Arranged inside the axial bore 3 or the valve sleeve 41 is a valve piston 43 which is guided by means of axial webs 44 arranged on it with sliding displaceability in the section of the axial bore 3 below the obliquely radial bores 7 and is provided, on its axial end piece at the top in Figure 5, with a sleeve 45 which is firmly connected to it and interacts with a cup-like recess in the base part 2 for the axial guiding of the valve piston 43. The sleeve 45 additionally serves as a stop which interacts with the valve sleeve 41 and bounds the lower axial end position of the valve piston 43. This lower end position is assumed when, on the one hand, the sleeve 45 bears against the valve sleeve 41 and the latter is seated with the cone section 10 on the annular edge 4.

A passage or free space always remains between the valve sleeve 41 and the sleeve 45 and the section of the valve piston 43 adjoining it at the bottom, such that the space receiving the helical compression spring 42 constantly communicates with the space inside the valve sleeve 41.

Arranged inside the valve sleeve 41 is a further helical compression spring 46 which is supported, on the one hand, on the axial webs 44 of the valve piston 43 and, on the other hand, on a collar on the inner circumference of the valve sleeve 41 and attempts to push the valve sleeve 41 upwards relative to the valve piston 43 in Figure 5.

A collar with an annular edge 47 is arranged axially below the webs 44 on the valve piston 43, such that this collar is able to interact with the annular edge 40 of the axial bore 3 in the manner of a slide valve. In the lower end position of the valve piston 43 illustrated, a narrow annular gap remains between the annular edges 40 and 47, such that the low-pressure connection T can communicate with the obliquely radial bores 7 and correspondingly with the motor connection P1 or P2.

Normally, the spring 42 holds the valve sleeve 41 and the spring 46 holds the valve piston 43 in the

respectively illustrated lower end position. The motor connection P1 or P2 is thus shut off relative to the pressure connection P, but connected to the low-pressure connection T. In this state, the pressure at the pressure connection P brings about only a virtually vanishing force which attempts to lift the valve sleeve 41 off the annular edge 4. This is based on the fact that the outside diameter of the valve sleeve 41 is only slightly larger than the diameter of the annular edge 4. The helical compression spring 42 can thus keep the valve sleeve 41 reliably in the closed position.

If the valve piston 43 is now moved in the upward direction, the annular edges 40 and 47 running one above the other block the connection between the motor connection P1 or P2 and the low-pressure connection T. As soon as the helical compression spring 46, whose tensioning force in the positions illustrated in Figure 5 of the valve piston 43 and valve sleeve 41 is less than the tensioning force of the helical compression spring 42, is tensioned sufficiently in the course of the upward movement of the valve piston 43, the valve sleeve 41 is carried along with it in the upward direction, thus causing the pressure connection P to be connected to the motor connection P1 or P2. The pressure of the pressure connection P otherwise propagates right into the space of the helical compression spring 42, such that the valve sleeve 41 is subjected to hardly any resulting hydraulic force in the axial direction.

In the example illustrated, it is assumed that the diameters D1 and D2 of the sleeve 45 and the annular edge 40 are of equal size, such that the hydraulic forces acting axially on the valve piston 43 are mutually compensated. The force/path characteristic of the valve piston 43 is correspondingly determined solely by the springs 42 and 46. However, it is basically also possible for the said diameters D1 and D2 to be of different dimensions and, in particular, for D2 to be larger than D1. A hydraulic restoring force must thus also be overcome during a further

upward stroke of the valve piston 43 when the valve sleeve 41 has been lifted off the annular edge 4.

The stroke path to be travelled by the valve piston 43, which is necessary in order to lift the valve sleeve 41 off the annular edge 4, is determined by the axial setting of the base part 2.

As soon as the valve piston 43 is again moved in the downward direction and the annular gap between the annular edges 40 and 47 is again opened, the valve sleeve 41 also again reaches the lower end position illustrated.

Figure 5 firstly shows by way of example how the valve cartridges 1 according to the invention can be arranged in a power steering system of a motor vehicle. Arranged between the ram-like actuating members 58 of two valve cartridges 1 arranged equiaxially with one another is a finger 59 which can be displaced in the axial direction of the valve cartridges 1 and is mechanically coupled to the vehicle steering system in such a way that it executes a displacement in the one direction or the other depending on the direction and magnitude of the actuating moment to be applied at the steering wheel. As long as no actuating moment is required, i.e. the vehicle steering system remains in the respective position without effort, the finger 59 assumes a central position in which both actuating members 58 of the two valve cartridges 1 have their end position which is advanced in the direction of the finger 59. Correspondingly, the motor connections P1 and P2 are hydraulically connected to the low-pressure connection T.

As soon as the finger 59 is displaced out of the central position illustrated, one valve cartridge 1 is actuated in each case, with the consequence that the associated motor connection P1 or P2 is caused to be hydraulically connected to the respective pressure connection P. Correspondingly, one side of the servomotor 60 is then acted upon with increased pressure; the servomotor 60 then generates a servo force which assists the respective steering manoeuvre by the driver.

The arrangement explained here is possible for all the valve cartridges illustrated.

Furthermore, Figure 5 shows a possibility as to how a hydraulic reaction control can be implemented - in particular together with the valve cartridges 1 illustrated in Figure 5.

Arranged between the motor connections P1 and P2 is a line 61 with a shuttle valve 62 which prevents a direct connection between the motor connections P1 and P2, but connects the motor connection P1 or P2 with the respectively higher pressure to a connection 63. This connection 63 can be connected via a proportional valve 64 and a cutoff valve 65 arranged in series therewith to a pressure line 66 which, in turn, connects the chambers 67 remaining between the base parts 2 and the facing end faces of the valve pistons 43 of the valve cartridges 1, which chambers, in turn, are connected to the low-pressure connection T via throttled axial bores in the valve pistons 43.

The proportional valve 64 can be controlled, for example, in dependence on the travelling speed. The cutoff valve 65 closes at a predetermined pressure in the pressure line 66.

In this way, a control pressure P_R can be generated in the pressure line 66, which control pressure, on the one hand, is dependent on the respectively higher pressure at the motor connections P1 and P2 and, on the other hand, can additionally be influenced by a predeterminable parameter, such as, for example, the travelling speed. This control pressure P_R generates, at the valve cartridge 1 actuated in each case by the finger 59, a force which counteracts the actuation, such that an actuation resistance occurs in the vehicle steering system, which actuation resistance varies analogously to the servo force generated by the servomotor 60.

The servo valves described are not only suitable for power steering systems of motor vehicles.

It is also possible, for example, to control the

pressure in a (single-action) displacer or piston/cylinder assembly using one of the servo valves described, such that a (comparatively high) setting force is generated analogously to the actuating force of the displacer or piston/cylinder assembly acting on the servo valve.

In this case, the servo valve and the displacer or piston/cylinder assembly can be combined to form a so-called linear booster.

The braking force of a vehicle brake can otherwise also be controlled using the servo valves according to the invention if said brake is designed as a pressure accumulator brake, in which the pressure of a pressure accumulator is fed under control to an actuating assembly of the brake designed as a displacer assembly.

Claims

1. A servo valve arrangement with high-pressure, low-pressure and motor connections and with a pressure control valve arranged between the high-pressure connection and motor connection and a return control valve arranged between the motor connection and low-pressure connection, the pressure control valve normally being closed and the return control valve normally being open, and both valves having axially displaceable control parts which are arranged equiaxially with one another and can be actuated by means of a common actuating member, whilst closing the return control valve and opening the pressure control valve, wherein the pressure control valve comprises a seat valve and the return control valve is normally throttled.

2. A servo valve arrangement according to Claim 1, wherein the control part of the return control valve is displaceable to a limited extent relative to the control part of the pressure control valve and is coupled thereto in terms of drive in such a way that the pressure control valve only opens after the return control valve has closed.

3. A servo valve arrangement according to Claim 1 or Claim 2, wherein the control part of the pressure control valve is coupled resiliently to the control part of the return control valve.

4. A servo valve arrangement according to any one of Claims 1 to 3, wherein the control part of the pressure control valve in its closed position is hydraulically held in its closed position against the force of an opening spring

5. A servo valve arrangement according to any one of Claims 1 to 3, wherein the control part of the pressure control valve in the open state is hydraulically substantially in equilibrium.

6. A servo valve arrangement substantially as described herein with reference to and as illustrated in the accompanying drawings.



Application No: GB 9614404.3
Claims searched: 1-6

Examiner: Steve Waller
Date of search: 30 September 1996

Patents Act 1977
Search Report under Section 17

Databases searched:

UK Patent Office collections, including GB, EP, WO & US patent specifications, in:
UK Cl (Ed.O): F2V VA10, VA11; B7H HHQ
Int Cl (Ed.6): B62D 5/08
Other: ONLINE: WPI

Documents considered to be relevant:

| Category | Identity of document and relevant passage | Relevant to claims |
|----------|---|--------------------|
| X | GB 1,413,111 (DOWTY) See page 2 lines 85 to page 3 line 6 | 1,2,3,5 |
| X | GB 1,251,315 (LUCIFER) See figure | 1,2,3,5 |
| X | GB 1,249,264 (ALFRED TEVES) See figures | 1,2,3,5 |
| X | GB 1,155,146 (FORD) See page 2 lines 12 to 20 | 1,2,3 |
| X | WO 93/02904 A1 (ZF FRIED-RICHSHAFEN) See abstract | 1,2,3 |
| X | US 4,699,231 (ZF FRIED-RICHSHAFEN) See col 3 line 44 to col 4 line 18 | 1,2,3 |

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