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(58) Field of search
F2S

(54) Adjustable hydraulic dampers and damping valves therefor

(57) A damping valve (7) for incorporation within the bypass passage (25) of an adjustable hydraulic damper includes at least one pressure-dependent spring-loaded valve (10a) in series with an axially movable valve member (9) which is arranged to open or close a port (8) of the damping valve (7) under the control of an electromagnet (11). The damping valve can be a single-acting valve which has a single spring-loaded valve (10a) and which is suitable for use in a two-tube damper in which the port (8) leads into a compensating chamber (21). Alternatively, the damping valve can be a double-acting valve which has two spring-loaded valves in series and which is suitable for use in a single tube damper in which the port leads directly into the lower working chamber of the damper.

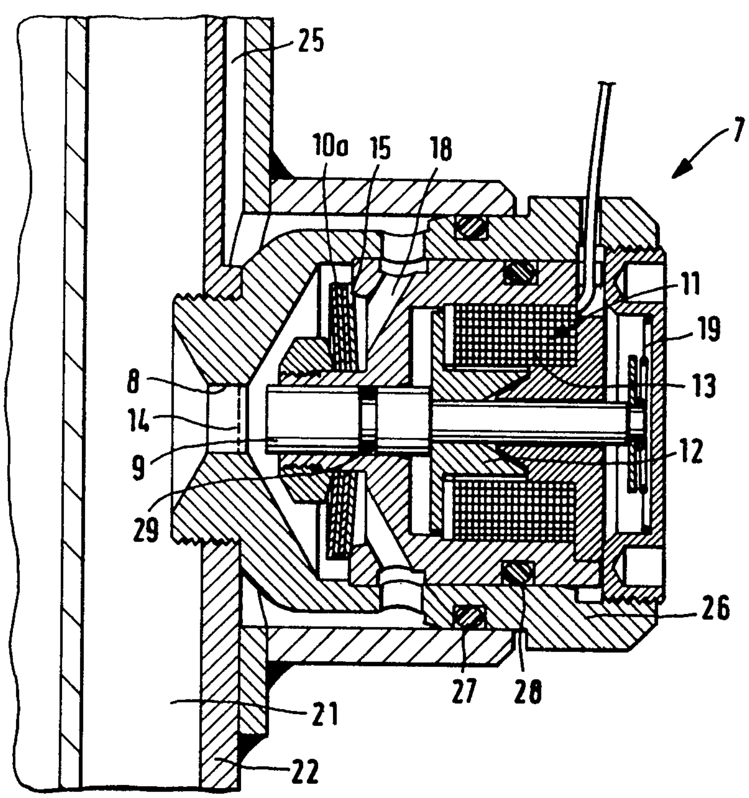

FIG. 3

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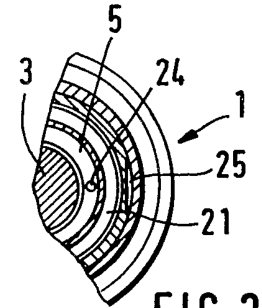

FIG.2

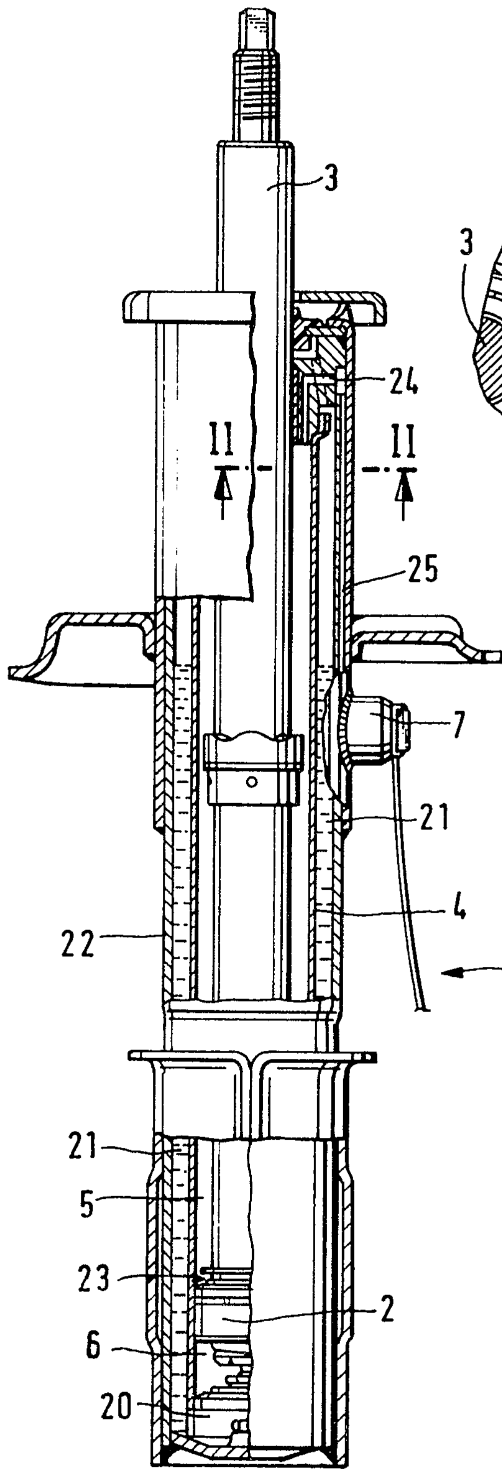

FIG.1

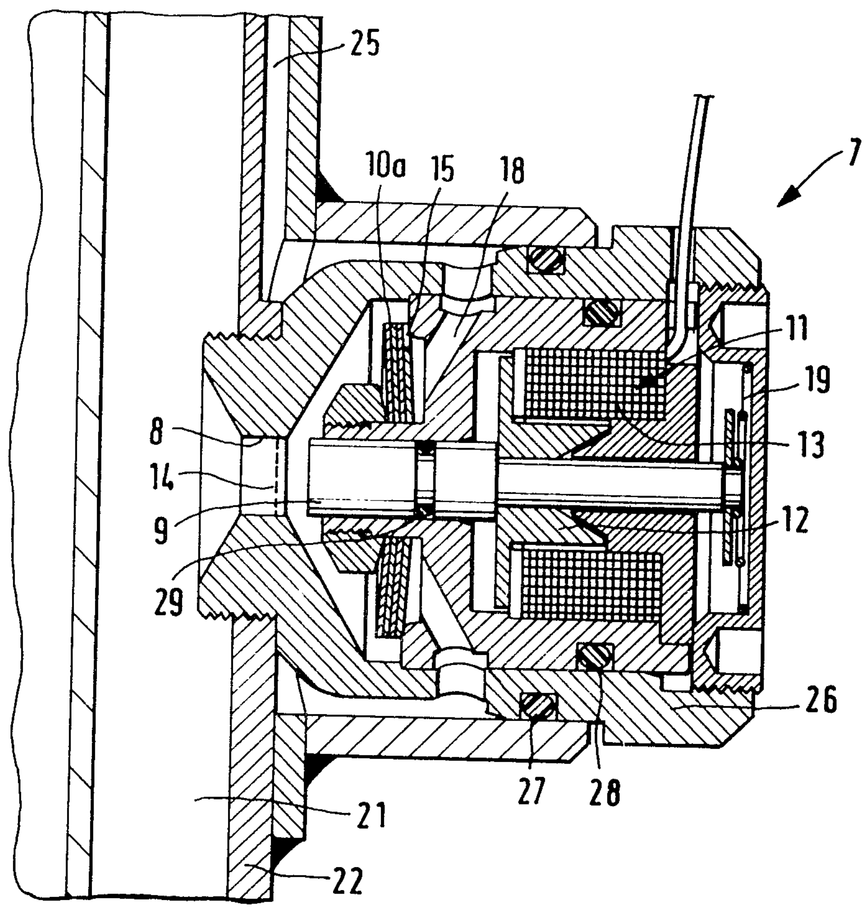

FIG. 3

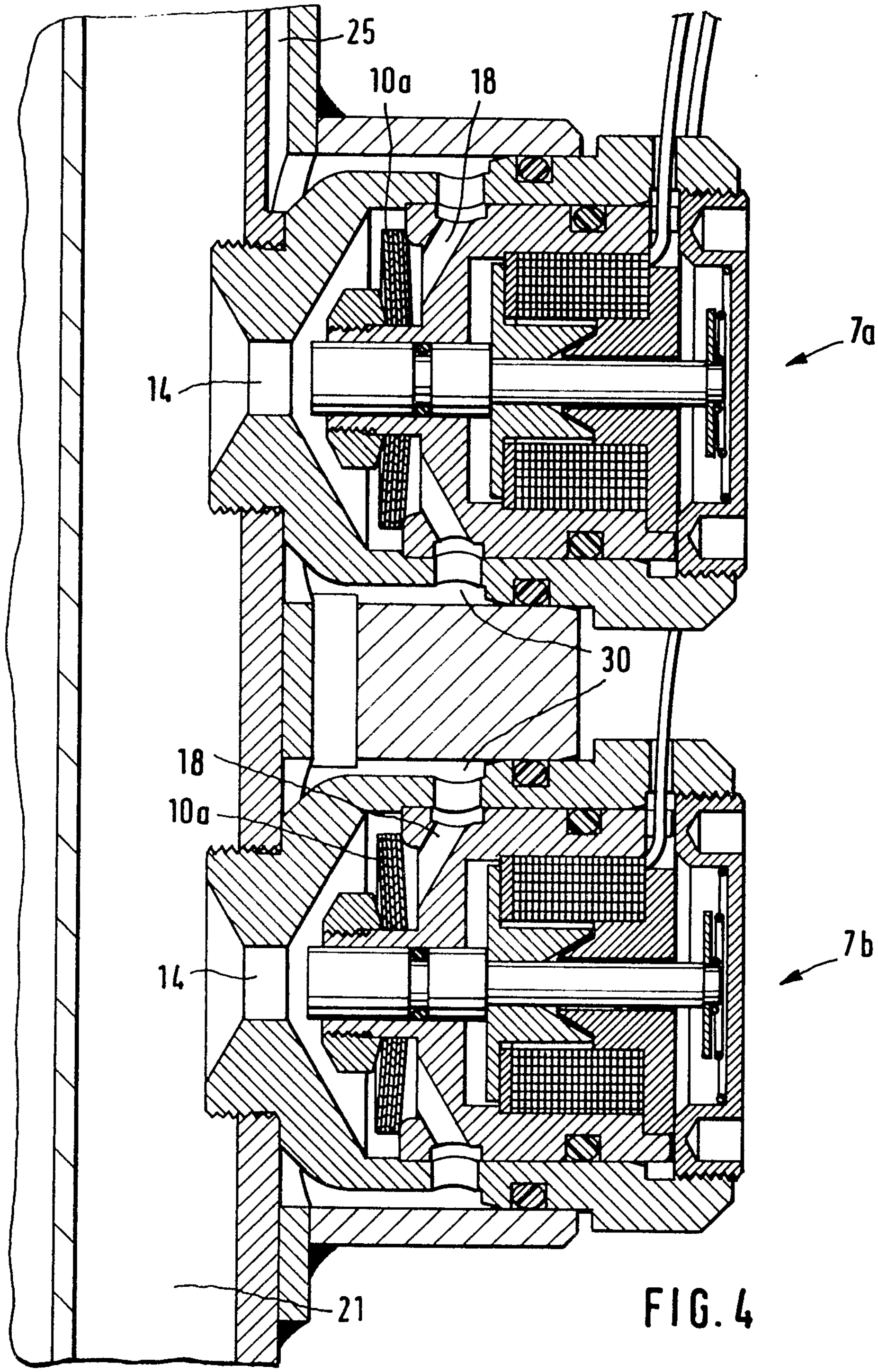

FIG. 4

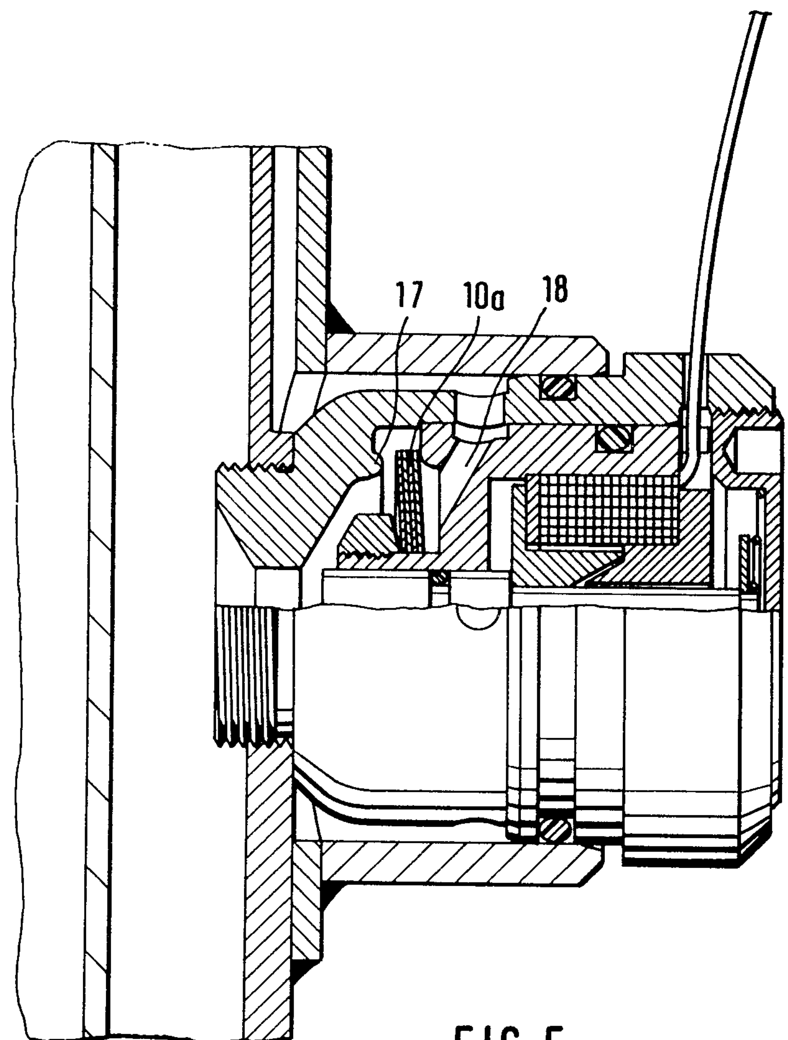

FIG. 5

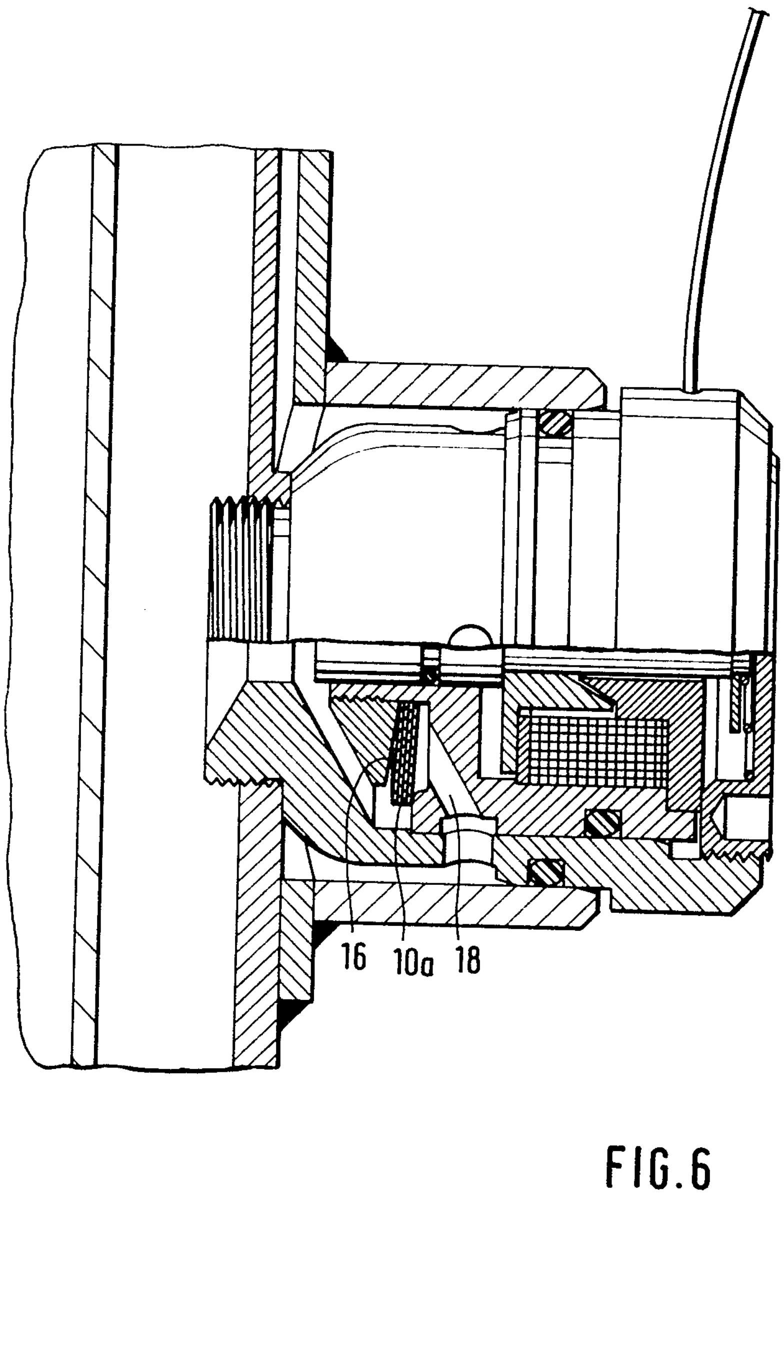

FIG. 6

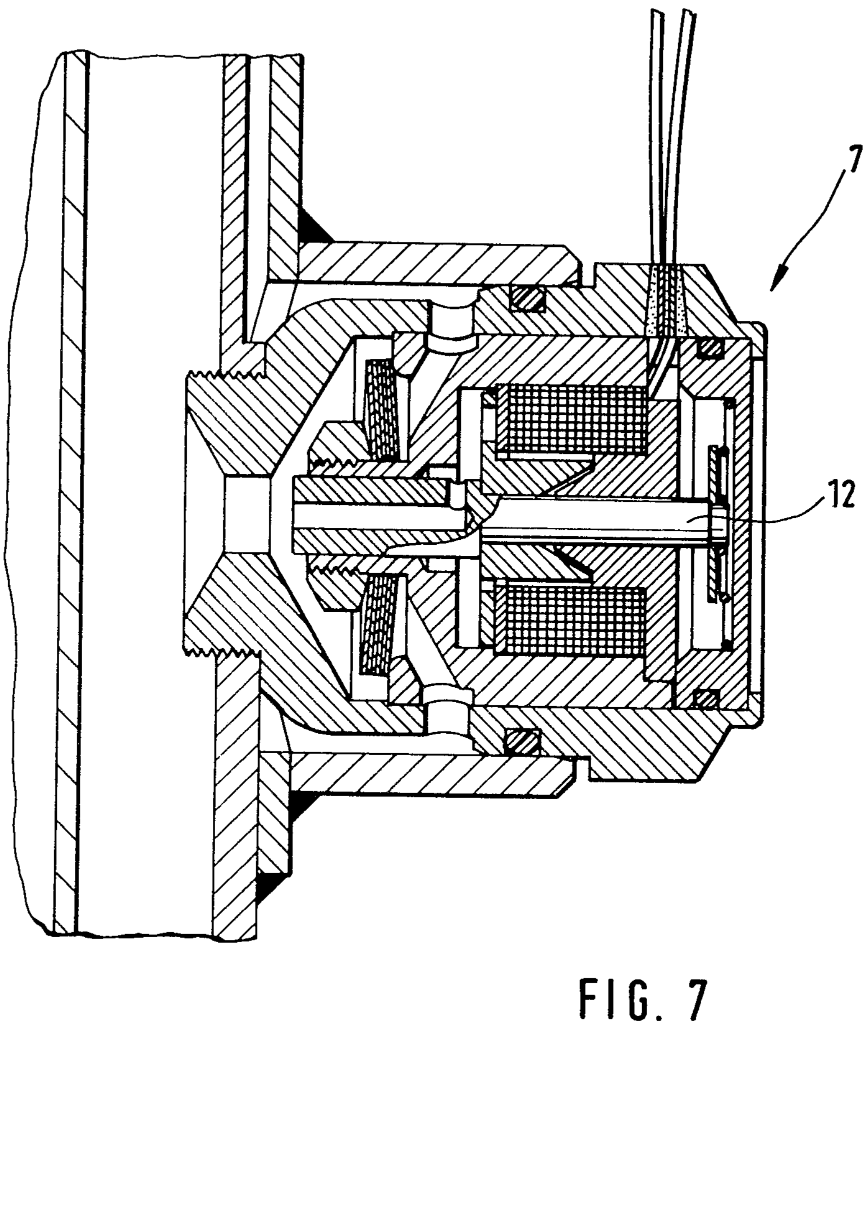

FIG. 7

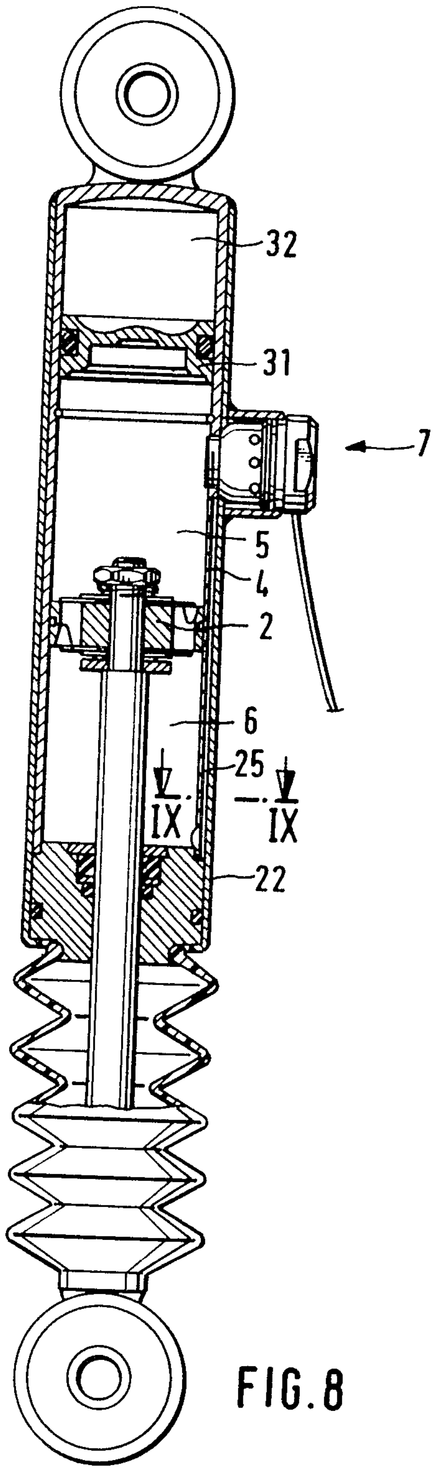

FIG.8

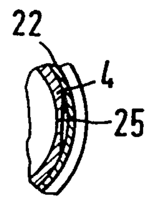

FIG.9

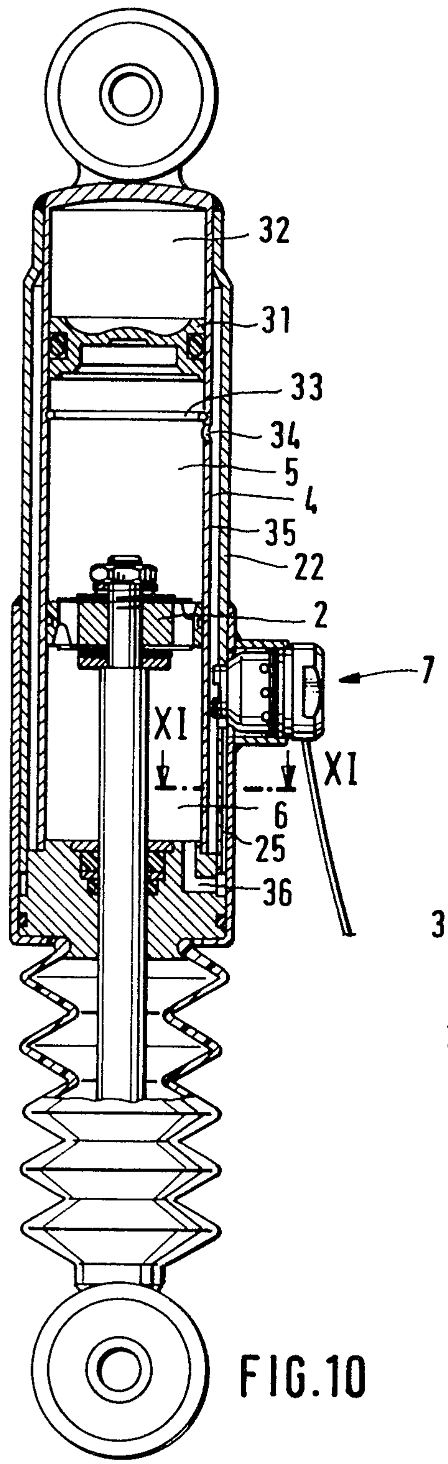

FIG.10

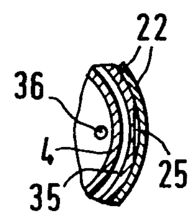

FIG.11

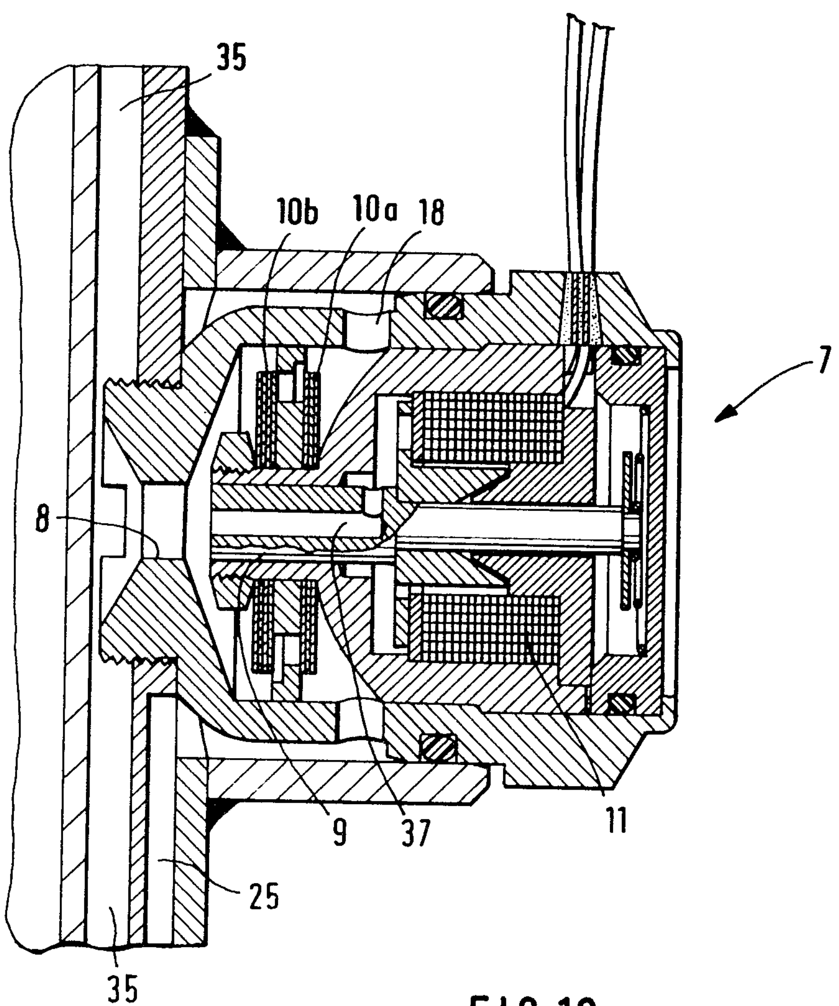

FIG.12

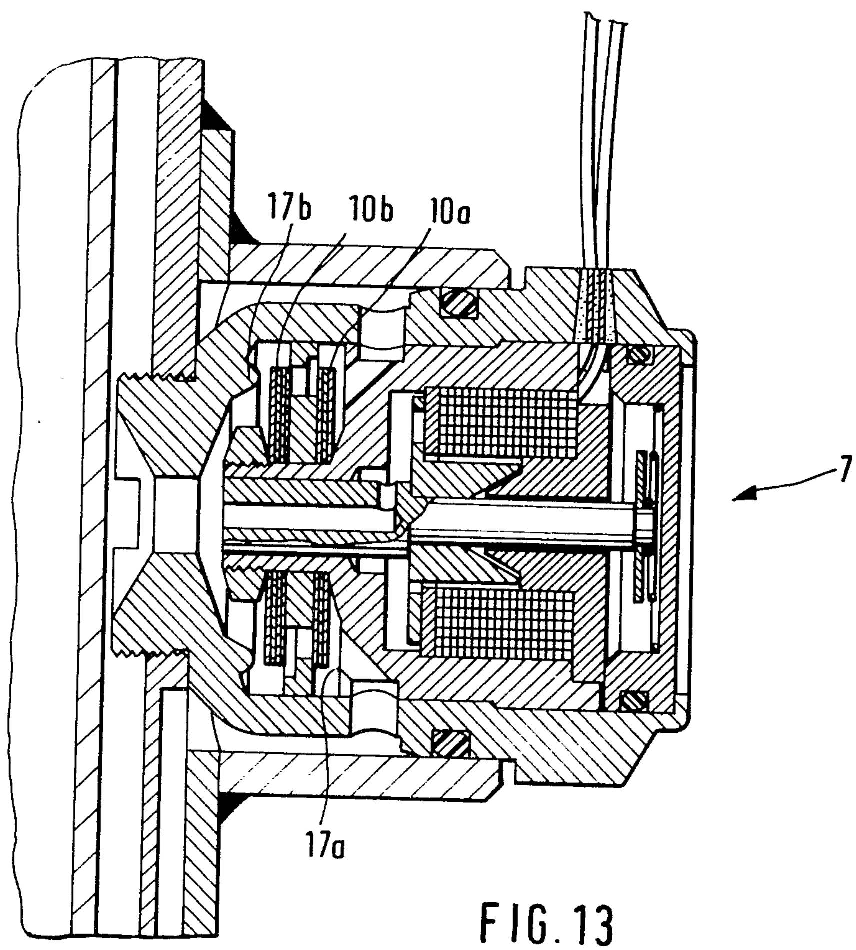

FIG. 13

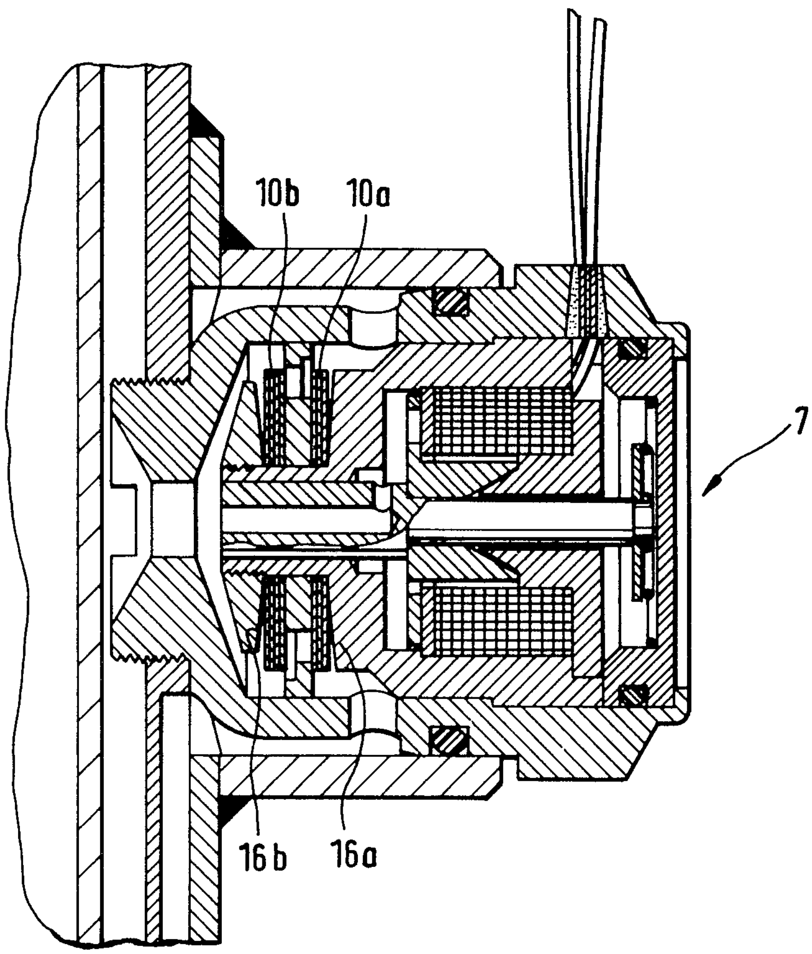

FIG. 14

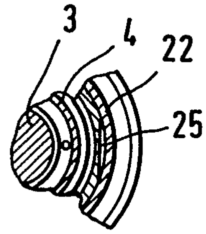

FIG.16

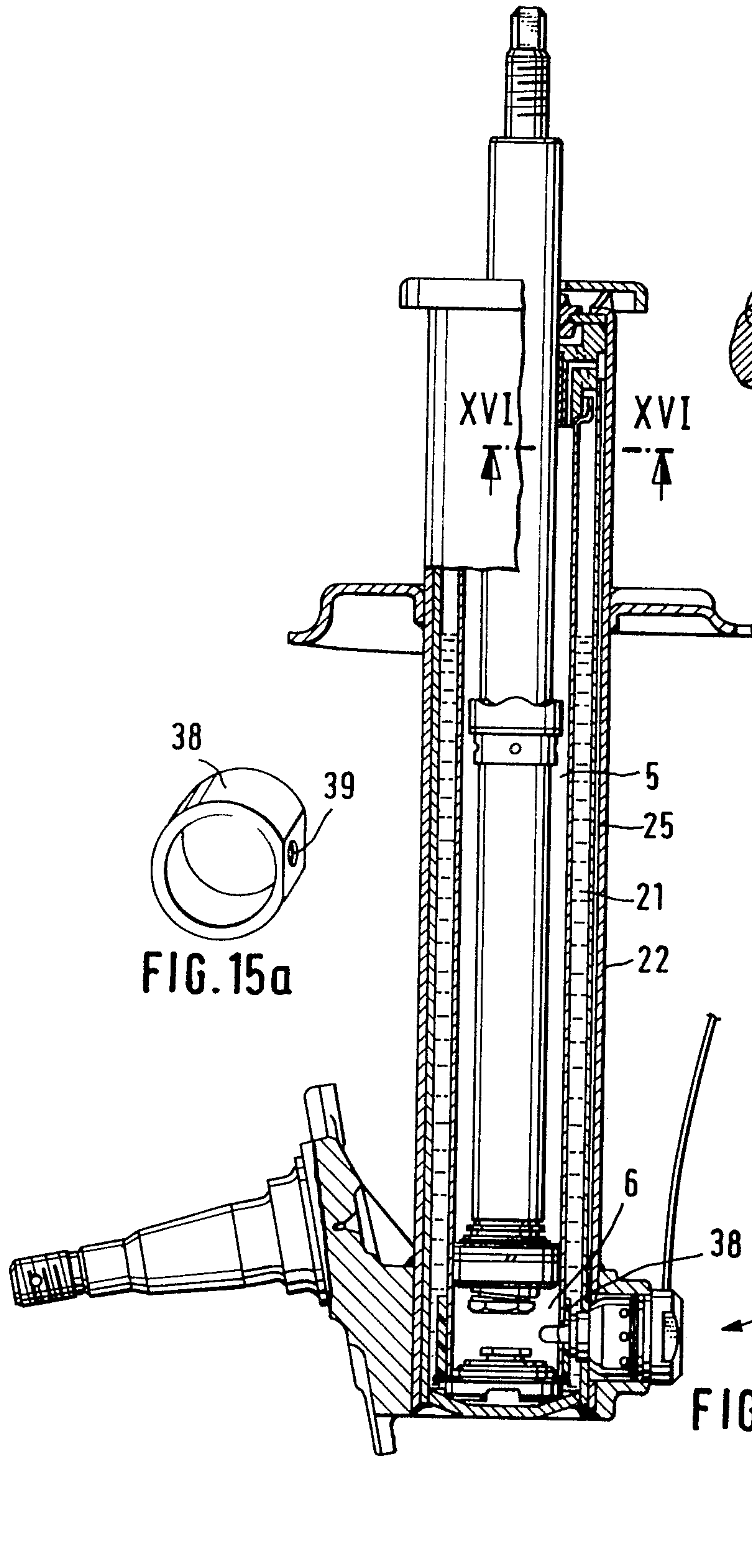

FIG.15a

FIG.15

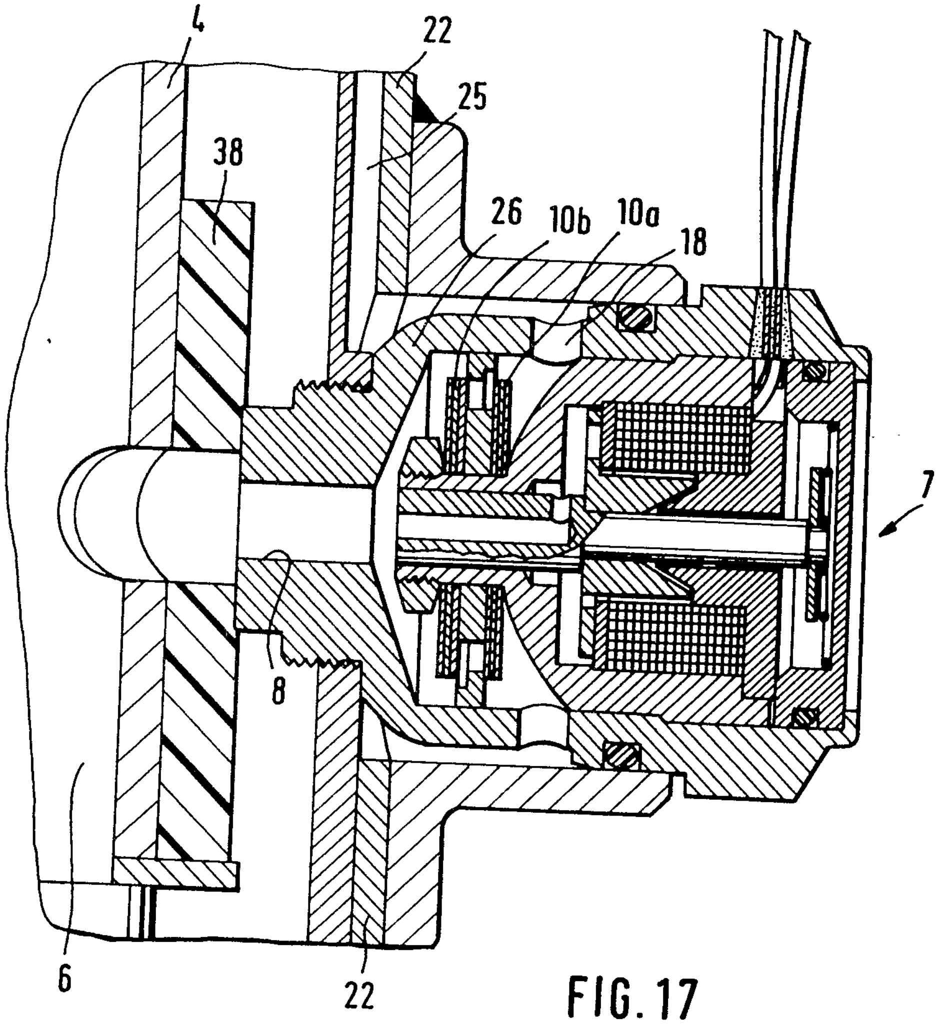

FIG. 17

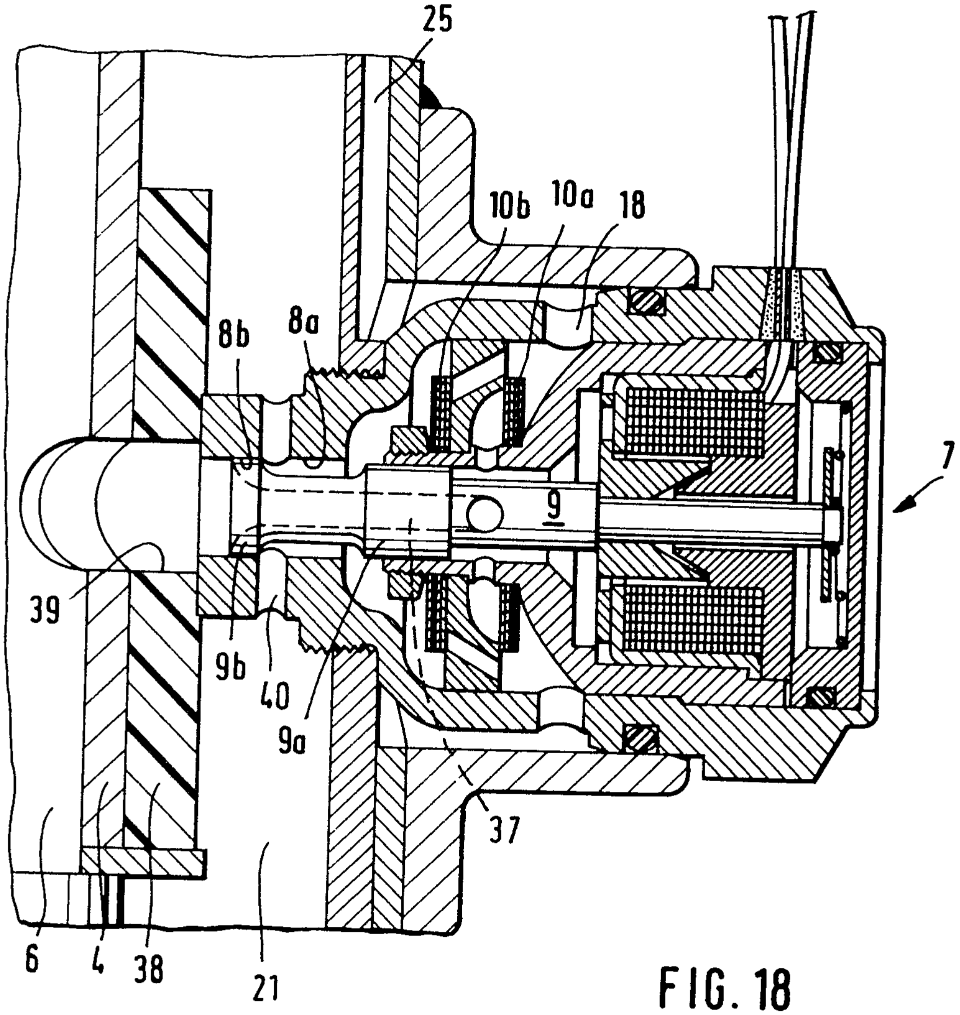

FIG. 18

SPECIFICATION

Adjustable hydraulic dampers and damping valves therefor

This invention relates to adjustable hydraulic dampers of the kind which comprise a working cylinder divided by a working piston secured on a piston rod into two working chambers filled with damping fluid, and which have a bypass passage in parallel with the working cylinder. Dampers of this kind will hereinafter be referred to as of the kind set forth. This invention also relates to damping valves for incorporation in the bypass passage of dampers of the kind set forth.

Telescopic hydraulic dampers are known (e.g. DE-PS 26 55 705 and DE-PS 27 44 301), in which a working cylinder is divided by a damping piston into two working chambers, the damping piston having valves of a mechanical construction for producing the damping force. Such a conventional vibration damper has the drawback that the associated damping characteristic is fixed in accordance with the type of vehicle and consequently is a compromise for different ride conditions. The use of such telescopic dampers for the vibration damping of a vehicle is restricted by the limited parameters in the operation, and also different states of loading can exert an influence on the system.

One known method of active vibration damping in which the damping force can be adjusted is disclosed in DE-OS 27 38 455. However, this arrangement suffers from the drawback that the damping losses must be restored by an externally provided pump. Such a system involving the input of external energy requires an additional externally fed hydraulic system and is consequently very expensive. Not all types of vehicle have or indeed permit the provision of such additional hydraulic equipment. Moreover, in the event of failure of the external pump the whole system fails and the ride safety is no longer guaranteed.

Other hydraulic vibration dampers are known which are electrically controlled, (e.g. DE-AS 14 05 781), and in which the damping characteristic can be adjusted by means of an electromagnet. In such an arrangement a valve member alters the cross-sectional area of a throttle point. However, such throttles only allow the setting of a progressive throttling which acts in parallel with the normal damping valves to set a markedly restricted force/speed characteristic of the damping piston. It has turned out in practice that such throttles are scarcely suited to achieve satisfactorily the required adjustment of the damping force characteristic.

It is the aim of the invention to provide a vehicle damping system with a variable degree of damping so that any desired adjustment of the damping in the extension and contraction phases can be achieved by means of a variably controlled damping valve.

According to a first aspect of the present invention there is provided a damping valve for incorporation in the bypass passage of an adjustable hydraulic damper of the kind set forth, the damping valve comprising at least one pressure-dependent spring-loaded valve in series with an axially movable, controllable valve member which co-operates with a port.

A damping valve in accordance with this aspect of the invention has the advantage that it can be incorporated in different variations of vehicle vibration damping systems. For instance, the damping valve may be arranged in parallel with the conventional throttle valves in the damping piston and/or in the base of the cylinder, or the damping valve may be mounted in the bypass passage with non-return valves arranged in the damping piston or in the base of the cylinder. Another possibility is that the damping valve may be arranged in a lockable vibration damper without further throttle valves being present in the damping piston or in the base of the cylinder. A further advantage is given by the series arrangement of the pressure-dependent spring-loaded valve, in that the port and the axially movable controllable valve member can have such a large cross-sectional area that there is scarcely any throttling of the damping medium. In this way, in conjunction with the spring-loaded valve, it is possible to achieve the required large degree of variability in the damping characteristics, with the possible inclusion of parallel-connected conventional damping valves in the working piston and/or in the base of the cylinder.

In one preferred form the damping valve is a single-acting damping valve which has a single pressure-dependent spring loaded valve which is arranged to be located upstream of the valve member for the normal direction of fluid flow through the valve. This form of valve may be conveniently incorporated in the bypass passage of a two-tube damper of the kind set forth in which the bypass passage connects the upper working chamber with a compensating chamber which is in communication with the lower working chamber, and in which the damping fluid is arranged to flow through the bypass passage and the single-actinq damping valve in the same direction in the extension and contraction phases.

In another preferred form the damping valve is a double-acting damping valve which has two spring-loaded valves in series with the valve member, one of the spring-loaded valves being arranged to control the flow of fluid in one direction through the damping valve and the other spring-loaded valve being arranged to control the flow of fluid in the opposite direction through the damping valve. This form of valve may be conveniently incorporated in the bypass passage of a single-handed damper of the kind set forth in which the bypass passage forms a direct connection between the upper working chamber and the lower working chamber, and in which the damping fluid flows through the bypass passage and the double-acting damping valve in opposite directions in the extension and contraction phases.

In a further embodiment the damping valve has at least two pressure-dependent spring-loaded valves in series with the axially movable valve member and has at least two separate inlets, the port forming an outlet and the arrangement being such that only one spring-loaded valve controls the flow of fluid through the damping valve from a first inlet to the port, and that two spring-loaded valves in series control the flow of fluid from the second inlet to the port. This embodiment may be conveniently incorporated in a two-tube damper of the kind set forth with the port of the damping valve leading into a compensating chamber, the first inlet leading from the bypass passage which is connected to the upper working chamber and the second inlet leading from the lower working chamber. In this arrangement only one spring-loaded valve controls the flow of fluid in the contraction phase and fluid in the extension phase.

Preferably, in order to achieve a simple and economical control of the valve, the axially movable valve member is controlled by an electromagnet. In this case, the construction of the valve is conveniently such that damping fluid can flow around the electromagnet. Furthermore, the armature of the electromagnet may conveniently constitute the axially movable valve member.

Alternatively, the axially movable valve member may be exposed to and controlled by an external fluid pressure.

The damping valve can thus be controlled either electrically, hydraulically or pneumatically, so that the damping characteristics can be adjusted to suit the prevailing road and vehicle conditions. Preferably, the valve member is urged by a spring towards the port so that in its rest condition the valve member closes off the port. This helps to ensure that the ride safety is maintained in the event of failure of the control of the damping valve when conventional damping valves operate in parallel with the bypass.

The port and the valve member may together constitute a throttle in order to provide a further degree of variability of the damping characteristic.

In a preferred embodiment the or each spring-loaded valve in in the form of a spring-loaded disc valve which may comprise a valve member in the form of at least one spring disc which is pre-loaded so as to be engageable with a valve seating. Alternatively, the spring-loaded disc valve may comprise a valve member in the form of a disc which is urged into engagement with the valve seating by at least one coil spring.

Preferably, the or each spring-loaded valve has a permanently open constant throttle between its valve member and valve seating which influences the damping force in the lower range of piston speed.

The or each spring-loaded valve may be provided with a stop which limits the stroke of its valve member. As a further possible variation the valve member of the or each spring-loaded valve may be engageable with a second valve seating so as to close the flow connection when that valve member is exposed to a fluid pressure in excess of a predetermined pressure.

The damping valve may be provided with plurality of feed bores, each of which is of limited cross-sectional area so as to constitute a throttle at the inlet of the damping valve.

According to a second aspect the present invention consists in an adjustable hydraulic damper of the kind set forth which incorporates, in its bypass passage, at least one damping valve in accordance with the first aspect of the invention.

A damper in accordance with the second aspect of the invention may incorporate a plurality of damping valves which are arranged in parallel. Preferably the same bypass passage incorporates the plurality of valves.

Where a plurality of damping valves are provided in the bypass passage of a single tube damper the arrangement is preferably such that at least one damping valve controls the flow of fluid in the contraction phase and at least one further valve controls the flow of fluid in the extension phase.

The damping valve or valves may be controlled by an electronic circuit in response to signals produced by a sensor or sensors. In such an arrangement, with the aid of the appropriate sensors for determining the instantaneous values of various parameters, the damping characteristic can be adjusted to suit the prevailing condition of loading, the driving condition of the vehicle and the speed of travel. A convenient arrangement of a vibration damping system may be achieved when the data at the front axle is used as signals for the rear axle. Moreover a multi-parameter control system can be provided by the use of several sensors.

Some preferred embodiments of both aspects of the present invention will now be described by way of example only, with reference to the accompanying drawings in which:

Figure 1 is a diagrammatic section through a two-tube adjustable hydraulic damper in accordance with the second aspect of the invention with valves in its damping piston and in its base and with a single-acting damping valve in a bypass passage extending from the upper working chamber into a compensating chamber;

Figure 2 is a transverse cross-section on the line II-II of Figure 1;

Figure 3 is a section through a single-acting damping valve in accordance with the first aspect of the invention;

Figure 4 is a section through an arrangement of two single-acting damping valves in parallel;

Figure 5 is a partial section through another single-acting damping valve in accordance with the first aspect of the invention;

Figure 6 is a partial section through a further single-acting damping valve which has a spring-loaded disc valve of restricted stroke;

Figure 7 is a section through yet another single-acting damping valve in accordance with the first aspect of the invention in which an electromagnet controlling the valve is immersed in the damping fluid;

Figure 8 is a diagrammatic section through a single tube adjustable hydraulic damper in accordance with the second aspect of the invention which is provided with valves in the piston and with a double-acting damping valve in a bypass passage extending from the upper working chamber into the lower working chamber;

Figure 9 is a transverse cross-section on the line IX-IX of Figure 8;

Figure 10 is a section through a damper similar to that of Figure 8 but in which the bypass extends from the upper working chamber through an annular space into the lower working chamber;

Figure 11 is a transverse cross-section on the line XI-XI of Figure 10;

Figure 12 is a section through a double-acting valve in accordance with the first aspect of the invention and in which an electromagnet controlling the valve is immersed in the damping fluid;

Figure 13 is a section through another double-acting damping valve in accordance with the first aspect of the invention;

Figure 14 is a section through a further double-acting damping valve in accordance with the first aspect of the invention and which has two spring-loaded disc valves of limited stroke;

Figure 15 is a diagrammatic section through a two-tube adjustable hydraulic damper in accordance with the second aspect of the invention and which is provided with valves in the working piston and in its base and with a damping valve in a bypass passage extending from the upper working chamber into either the lower working chamber or the compensating chamber;

Figure 16 is a transverse cross-section on the line XVI-XVI of Figure 15;

Figure 17 is a section through a double-acting damping valve in accordance with the first aspect of the invention and which is adapted to be used in the damper of Figure 15 and 16 in a bypass passage from the upper working chamber to the lower working chamber;

Figure 18 is a section through a triple-acting damping valve in accordance with the first aspect of the invention and which is adapted to be used in the damper of Figures 15 and 16 in a bypass passage from the upper working chamber to the compensating chamber.

The two-tube adjustable hydraulic damper illustrated in Figure 1 is a spring suspension strut for use in a vehicle vibration damping system. The damper comprises a working cylinder 4 and a damping piston 2 secured on a piston rod 3. The piston 2 divides the working chamber 4 into an upper working chamber 5 and a lower working chamber 6. The damping piston 2 is furthermore fitted with damping valves. Arranged in the base 20 of the cylinder 4 are further valves, through which the volume of damping fluid displaced by the piston rod is forced into a compensating chamber 21 which is formed between the wall of the cylinder 4 and the inner wall of an outer tube 22.

The upper working chamber 5 is connected to the compensating chamber 21 through a bypass comprising a passage 24, a bypass passage 25 and a damping valve provided in the bypass passage 25. The damping valve 7 regulates the flow of damping fluid through the bypass from the upper working chamber 5 into the compensating chamber 21.

Figure 2 is a transverse section through the damper 1 with the piston 3 being arranged in its centre and the upper working chamber 5 being connected through the passage 24 and the bypass passage 25 to the compensating chamber 21. The bypass passage 25 is formed by the ground-away portion of a surrounding tubular component.

Figure 3 shows a single-acting damping valve 7 which is formed as a unit, and which is incorporated within the bypass passage 25 of a two-tube adjustable damper such as illustrated in Figures 1 and 2. In use damping fluid flows from the bypass passage 25, through feed bores 18 of the valve 7, past a spring-loaded valve 10, through the cross-sectional area 14 of a port 8 and into the compensating chamber 21. The spring-loaded valve 10 is formed as a spring-loaded disc valve and has a constant throttling cross-sectional area 15 which allows flow through the spring-loaded valve 10 in the low piston velocity range. The port 8 can be closed by an axially movable valve member 9 which is controlled by an electromagnet 11 comprising a coil 13 and an armature 12. A restoring spring 19 exerts a permanent force on the valve member so that when the electromagnet 11 is de-energised the valve is closed. The housing 26 of the damping valve 7 is screwed into the outer tube 22 of the damper and the electromagnet 11 is sealed from the bypass passage 25 by means of seals 27, 28, 29. As the valve seating of the port 8 is arranged downstream of the spring-loaded valve 10, in the direction of the flow from the upper working chamber 5 to the compensating chamber 21, the valve member 9 is free from any effects of damping back-pressure, and so a low power electromagnet can be provided. Therefore, on energisation of the electromagnet, it is only necessary to overcome the force of the restoring spring 19. Since the valve housing 26 is formed as a sealed unit which can be screwed into the outer-tube 22 of a damper it is a simple matter to replace the damping valve 7. In this arrangement the flow through the valve in the extension and compression phases takes place in the same direction, namely from the bypass passage 25, and through the feed bores 18 in a direction towards the spring-loaded valve 10.

Figure 4 illustrates an arrangement in which two damping valves 7a, 7b are connected in parallel in a bypass passage 25. The feed bores 18 of both valves 7a, 7b are connected together through an annular passage 30. The provision of two or more damping valves arranged in parallel can permit a further degree of adjustment if different forms of spring-loaded valves 10 are used in different damping valves. The plurality of damping valves may also be connected so that different damping characteristics in the extension and contraction phases can be obtained. An appropriate electronic circuit may, in such an embodiment, be provided for controlling the various possible ways of switching the plurality of damping valves.

Figure 5 shows another single-acting damping valve similar to that of Figure 3 but in which a second valve seating 17 is provided for the spring-loaded valve 10a. The spring-loaded valve 10a can perform an axial stroke when there is appropriate damping pressure from the feed bores 18, and when a predetermined pressure is exceeded the spring-loaded valve 10a engages the second valve seating 17 and thereby closes off the flow of damping fluid through the valve.

Figure 6 shows a further single-acting damping valve similar to that of Figure 3 but in which the valve 10a is restricted in its stoke by a stop 16. When a predetermined pressure is reached in the bypass passage the valve 10a engages the stop 16 and a large cross-sectional area is opened through the spring-loaded valve 10a. With a further rise in pressure in the feed bores 18 the cross-sectional area of the opening through the spring-loaded valve is kept constant.

Figure 7 shows yet another single-acting damping valve similar to that of Figure 3 but in which the armature 12 of the electromagnet is not sealed off so that damping fluid can flow around it. An advantage of this modification is that the sealing for the armature 12 can be omitted and because the friction is reduced the power requirements of the electromagnet 11 can be reduced. This reduction lies in the fact that the armature 12 does not have to operate against the pressure of the damping fluid. Moreover the cooling and the noise level of the valve 7 are improved.

Figure 8 shows a single tube adjustable hydraulic damper for use in a vehicle vibration damping system in which the upper working chamber 5 of the damper is directly connected to the lower working chamber 6 through the bypass passage 25. In this case the damper is a single tube gas pressure damper in which the upper working chamber 5 has an additional dividing piston 31 which separates a gas space 32 from the main working chamber 5. The damping valve 7 is connected in the bypass passage 25 in such a manner as to project directly into the upper working chamber 5. The piston 2 is arranged to be axially movable in the working cylinder 4 with its stroke limited so that it does not reach the region of the damping valve 7.

The damping valve 7 of the damper of Figure 8 has damping fluid flowing through it in opposite directions in the extension and contraction phases. In the extension phase the fluid passes from the lower working chamber 6 through the bypass passage 25, past the damping valve 7 and into the upper working chamber 5, whereas in the contraction phase the direction of flow is reversed.

Figure 9 shows a section through the damper of Figure 8 with the bypass 25 being formed between a locally flattened portion of the working cylinder 4 and an outer tube 22.

Figure 10 also shows a single-tube gas pressure damper in which the working cylinder 4 is divided by the damping piston 2 into an upper working chamber 5 and a lower working chamber 6. A gas space 32 is separated from the upper chamber 5 by a dividing piston 31. A spring ring 33 is provided for limiting the axial movement of the dividing piston 31.

In the damper shown in Figure 10 the damping valve 7 is arranged in an outer tube 22 and the upper working chamber is permanently in communication with an annular space 35 through the bore 34 provided in the working cylinder 4. This annular space 35 thereby forms a part of the upper working chamber 5. The flow through the damping valves 7 in the contraction phase is from the upper working chamber through the annular space 35 to the damping valve 7 and from there through the bypass passage 25 and the bore 36 into the lower working chamber 6. In the extension phase the direction of flow is reversed. As in the embodiment of Figures 8 and 9 the flow of fluid through the damping valve 7 is in opposite directions in the extension and contraction phases.

Figure 11 is a section through the damper of Figure 10 showing the working cylinder 4, the annular chamber 35, and the outer tube 22 which is made of double-wall construction. The inner wall of the outer tube 22 has a locally flattened portion which forms the by-pass passage 25 which is in communication with the lower working chamber 6 through a bore 36.

Figure 12 shows a double-acting damping valve 7 which is suitable for incorporation in the embodiments of Figure 8 and 10 in which damping fluid flows through the bypass passage 25 in opposite directions in the extension and contraction phases. Figure 12 in particular shows the damping valve 7 disposed between a bypass passage 25 and an annular space 35 such as shown in Figure 10. The damping valve 7 once again has a port 8 which can be closed off by an axially movable valve member 9 and two spring-loaded valves in the form of spring discs 10a and 10b which control the flow of the fluid through the valve in different directions. The valve member 9 is provided with a central bore 37 by means of which the electromagnet 11 is immersed in damping medium. When the valve member 9 opens the port 9 fluid flows through the damping valve 7 in the contraction phase from the annular space 35, through the port 8, past the spring-loaded valve 10a and through the bores 18 which serve as discharge bores into the bypass passage 25 and from there into the lower working chamber 6.

In the extension phase, the fluid flows from the bypass passage 25, through bores 18 which now serve as feed bores, past the spring-loaded valve 10b and through the port 8 into the annular space 35 and thence into the upper working chamber 5.

Figures 13 and 14 show damping valves 7 which are modifications of the double-acting valve illustrated in Figure 12.

In Figure 13 the spring-loaded valves 10a and 10b are each provided with a second valve seating, 17a and 17b respectively. In this arrangement the second valve seatings 17a and 17b act in a similar manner to the valve seating 17 of Figure 5, that is when a predetermined pressure is exceeded, the damping valve 7 is blocked off by one of the spring-loaded valves 10a or 10b. In the contraction phase, when the predetermined pressure is reached, the spring-loaded valve 10a engages the second valve seating 17a. In the extension phase, when the predetermined pressure is reached, the spring-loaded valve 10b engages the second valve seating 17b and thereby prevents further flow through the damping valve 7. The construction of the spring-loaded valves 10a and 10b may be varied to set a different predetermined pressure.

In Figure 14 the spring-loaded valves 10a and 10b are each provided with a stop, 16a and 16b respectively. By means of the stops the stroke of each of the spring-loaded valves 10a and 10b is restricted so that when a predetermined pressure is exceeded the cross-sectional area for the flow does not increase any further. The valve 10a is constructed so that when the predetermined pressure is exceeded in the contraction phase the stop 16a is engaged. In the extension phase and with the flow in the opposite direction the valve 10b is displaced until it engages the stop 16b.

Figure 15 shows a two-tube vibration damper in which the bypass passage 25 connects the upper working chamber 5 either directly to the lower chamber 6 or to the compensating chamber 21 and the lower working chamber 6. In these arrangements the damping valve 7 is mounted by means of a flange on the outer tube 22 and is connected to the lower working chamber 6 through an intermediate ring 38.

Figure 15a shows in detail the intermediate ring 38 which is provided with an opening 39.

Figure 16 is a section through the damper of Figure 15 and shows the working cylinder 4, the piston rod 3 and an outer tube 22, again of double-wall construction. The bypass passage 25 is again formed by a locally flattened portion on the inner wall of the outer tube.

Figure 17 shows a double-acting damping valve incorporated in the damper illustrated in Figure 15. The working cylinder 4 of the damper is surrounded by the intermediate ring 38 and the valve housing 26 of the damping valve is screwed to the outer tube 22 of the damper. In this embodiment damping fluid again flows through the damping valve in opposite directions in the extension and contraction phases. In the contraction phase the damping fluid flows from the lower working chamber 6, through the port 8, past the spring-loaded valve 10a and through the bores 18 into the bypass passage 25. In the extension phase the damping fluid flows in the opposite direction from the upper working chamber 5 through the bypass passage 25 and the feed bores 18, past the spring-loaded valve 10b and through the port 8 into the lower working chamber 6.

Figure 18 shows a further damping valve which is suitable for use in the embodiment of Figure 15. Here the damping valve 7 is provided with a valve member 9 which is of stepped shape having cylindrical portions 9a and 9b of larger diameter and an intermediate portion of smaller diameter. The cylindrical portion 9a can open or close off a port 8a leading in to the compensating chamber 21 through at least one opening 40, but the lower working chamber 6 is permanently cut off from the compensating chamber 12 by the cylindrical portion 9b of the valve member 9. The valve member 9 also has a central bore 37 which is in communication with the lower working chamber 6.

When the bypass passage is opened by the damping valve, i.e., when the port 8a is opened by the cylindrical portion 9a, the damping fluid flows in the contraction phase, from the lower working chamber 6 through the opening 39 in the intermediate ring 38, through the central bore 37, past the spring-loaded valve 10a and then both through the bores 18 into the bypass passage 15 and also through the spring-loaded valve 10b past the port 8a and the openings 40 and into the compensating chamber 21. This means that in the contraction phase the damping fluid passes through both spring-loaded valves 10a and 10b and is thereby distributed both to the upper working chamber 5 and also into the compensating chamber 21.

In the extension phase, the damping fluid flows from the lower working chamber passage 25, past the feed bores 18 to the spring-loaded valve 10b, through the port 8a and the openings 40 and into the compensating chamber 21. This means that with the valve member in the same position, in the contraction phase the flow is through both spring-loaded valves 10a and 10b in series, whereas in the extension phase the fluid only flows through valve 10b.

CLAIMS

1. A damping valve for incorporation in the bypass passage of an adjustable hydraulic damper of the kind set forth, the damping valve comprising at least one pressure-dependent spring-loaded valve in series with an axially movable, controllable valve member which co-operates with a port.

2. A single-acting damping valve according to claim 1, which has a single pressure-dependent spring-loaded valve which is arranged to be located upstream of the axially movable valve member for the normal direction of fluid flow through the valve.

3. A damping valve according to claim 1, which has at least two pressure-dependent spring-loaded valves in series with the axially movable valve member.

4. A double-acting damping valve according to claim 3, in which one of the spring-loaded valves is arranged to control the flow of fluid through the damping valve in one direction, and another spring-loaded valve is arranged to control the flow of fluid through the damping valve in the opposite direction.

5. A damping valve according to claim 3, which has at least two inlets and in which the port forms an outlet, the arrangement being such that only one spring-loaded valve controls the flow of fluid from a first inlet to the port and two spring-loaded valves in series control the flow of fluid from a second inlet to the port.

6. A damping valve according to claim 5, in which the axially movable valve member is provided with an axial bore which extends to one end of the member to form one of the inlets.

7. A damping valve according to any of the preceding claims, in which the axially movable valve member is controlled by an electromagnet.

8. A damping valve according to claim 7, in which the construction of the valve is such that damping fluid can flow around the electromagnet.

9. A damping valve according to claim 7 or claim 8, in which the armature of the electromagnet constitutes the axially movable valve member.

10. A damping valve according to any of claims 1 to 6,in which the axially movable member is exposed to and is controlled by an external fluid pressure.

11. A damping valve according to any of the preceding claims, in which the port and the valve member together constitute a throttle.

12. A damping valve according to any of the preceding claims, in which the valve member is spring-urged towards the port so that in its rest condition the valve member closes off the port.

13. A damping valve according to any of the preceding claims, in which the or each spring-loaded valve is in the form of a spring-loaded disc valve.

14. A damping valve according to claim 13, in which the or each spring-loaded disc valve comprises a valve member in the form of at least one spring disc which is pre-loaded so as to be engageable with a valve seating of the disc valve.

15. A damping valve according to claim 13, in which the or each spring-loaded disc valve comprises a valve member in the form of a disc which is urged into engagement with a valve seating of the disc valve by at least one coil spring.

16. A damping valve according to any of the preceding claims, in which the or each spring-loaded valve has a permanently open throttle between its valve member and its valve seating.

17. A damping valve according to any of the preceding claims, in which the or each spring-loaded valve has a stop which limits the stroke of its valve member.

18. A damping valve according to any of claims 1 to 16, in which the valve member of the or each spring-loaded valve is engageable with a second valve seating so as to close off communication through the valve when the valve member is exposed to a fluid pressure in excess of a predetermined pressure.

19. A damping valve according to any of the preceding claims, which is provided with a plurality of feed bores, each feed bore being of limited cross-sectional area so as to constitute a throttle.

20. A single-acting damping valve according

to claim 2 and substantially as described herein with reference to any one of Figures 3, 5, 6 or 7 of the accompanying drawings.

21. A double acting damping valve according to claim 4 and substantially as described herein with reference to any one of Figures 12, 13, 14 or 17 of the accompanying drawings.

22. A damping valve according to claim 5 and substantially as described herein with reference to Figure 18 of the accompanying drawings.

23. An adjustable hydraulic damper of the kind set forth which incorporates, in its bypass passage, at least one damping valve in accordance with any of the preceding claims.

24. An adjustable hydraulic damper of the kind set forth in which the bypass passage is arranged to connect the upper working chamber to a compensating chamber which is in communication with the lower working chamber, the bypass passage incorporating at least one single-acting damping valve in accordance with claim 2, or any of claims 7 to 20 as dependent from claim 2, the arrangement being such that damping fluid flows through the bypass passage and the damping valve in the same direction in the extension and contraction phases of the damper.

25. An adjustable hydraulic damper of the kind set forth in which the bypass passage forms a direct connection between the upper working chamber and the lower working chamber, thc bypass passage incorporating at least one double-acting damping valve in accordance with claim 4, or any one of claims 7 to 19 or 21 as dependent from claim 4, the arrangement being such that damping fluid flows through the bypass passage and the damping valve in opposite directions in the extension and contraction phases of the damper.

26. An adjustable hydraulic damper of the kind set forth in which the bypass passage forms a direct connection between the upper working chamber and the lower working chamber, the bypass passage incorporating a plurality of single-acting damping valves in accordance with claim 2, or any one of claims 7 to 20 as dependent from claim 2, at least one of the damping valves being arranged to control the flow of damping fluid through the bypass passage in one direction in the extension phase of the damper, and at least one other of the valves being arranged to control the flow of damping fluid through the bypass passage in the opposite direction in the contraction phase of the damper.

27. An adjustable hydraulic damper of the kind set forth in which the bypass passage is arranged to connect the upper working chamber to a compensating chamber which is in communication with the lower working chamber, the bypass passage incorporating at least one damping valve in accordance with claim 5, or any one of claims 6 to 19 or 22 as dependent from claim 5, the arrangement being such that the port of the damping valve leads into the compensating chamber and the first inlet is in communication with the lower working chamber so that only one spring-loaded valve controls the flow of fluid in the extension phase and two spring-loaded valves in series control the flow of fluid in the contraction phase.

28. An adjustable hydraulic damper according to any one of claims 23 to 25 or 27 which incorporates a plurality of damping valves arranged in parallel in the bypass passage.

29. An adjustable hydraulic damper according to any one of claims 23 to 28 in which the bypass passage is formed by a flattened or ground-away portion of a tubular component of the damper.

30. An adjustable hydraulic damper according to any one of claims 23 to 29 in which an electronic circuit is provided to control the damping valve or valves in response to signals produced by a sensor or sensors.

31. An adjustable hydraulic damper substantially as described herein with reference to Figures 1 and 2 of the accompanying drawings and which incorporates one or more single-acting damping valves substantially as described herein with reference to any one of Figures 3 to 7 of the-accompanying drawings.

32. An adjustable hydraulic damper substantially as described herein with reference to Figures 8 and 9 of the accompanying drawings and which incorporates a double-acting damping valve substantially as described herein with reference to any one of Figures 12 to 14 of the accompanying drawings.

33. An adjustable hydraulic damper substantially as described herein with reference to Figures 10 and 11 of the accompanying drawings and which incorporates a double-acting damping valve substantially as described herein with reference to any one of Figures 12 to 14 of the accompanying drawings.

34. An adjustable hydraulic damper substantially as described herein with reference to Figures 15 and 16 of the accompanying drawings and which incorporates a damping valve substantially as described herein with reference to Figure 17 of the accompanying drawings.

35. An adjustable hydraulic damper substantially as described herein with reference to Figures 15 and 16 of the accompanying drawings and which incorporates a damping valve substantially as described herein with reference to Figure 18 of the accompanying drawings.

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