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Selected US specifications from IPC sub-class F16F

(54) Vibration damper with variable damping force

(57) A vibration damper includes a piston 7 provided with damping valves 18,11, and with a bypass 17 having an opening 15 of variable cross-section controlled by a slider 16. A spring-loaded damping valve 20 is arranged in the bypass to open on the compression stroke and is located upstream of the opening 15. The bypass 17 can also have effect during the extension stroke by virtue of pressure chambers 13 which act on valve 11. In addition to controlling the opening 15, the slider can be arranged to adjust the initial stress of the valve spring 19 (Figs. 7,8), and the valve body 20 may include a throttle opening (Fig. 4).

Fig. 2

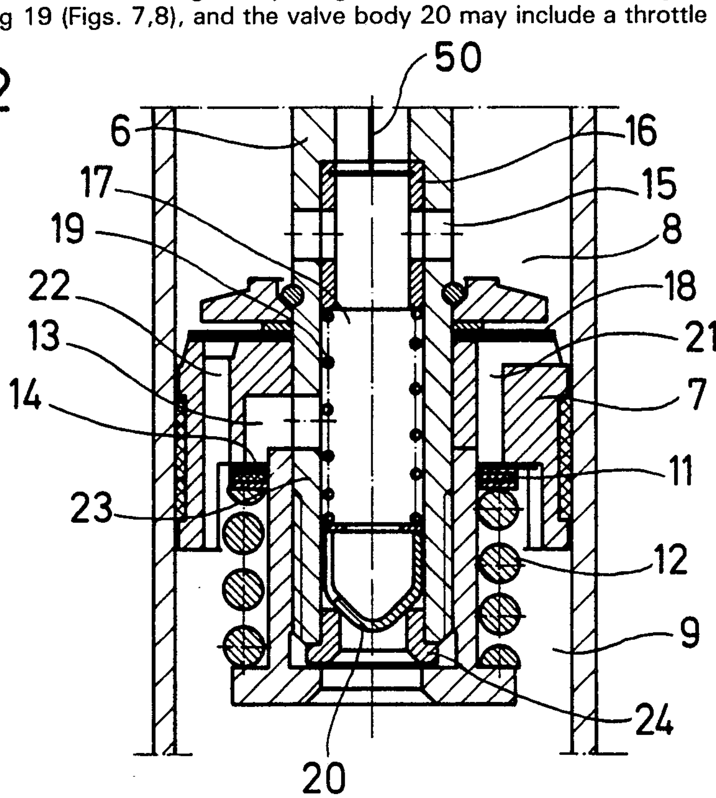

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Fig. 1

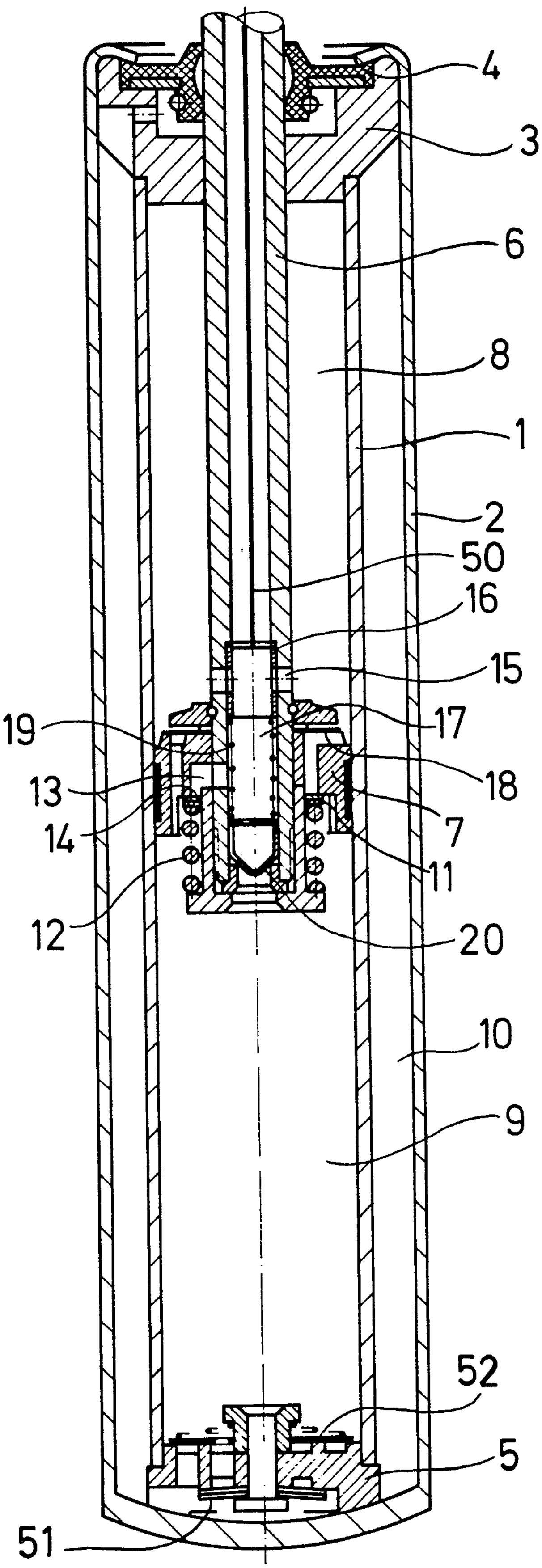

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Fig. 2

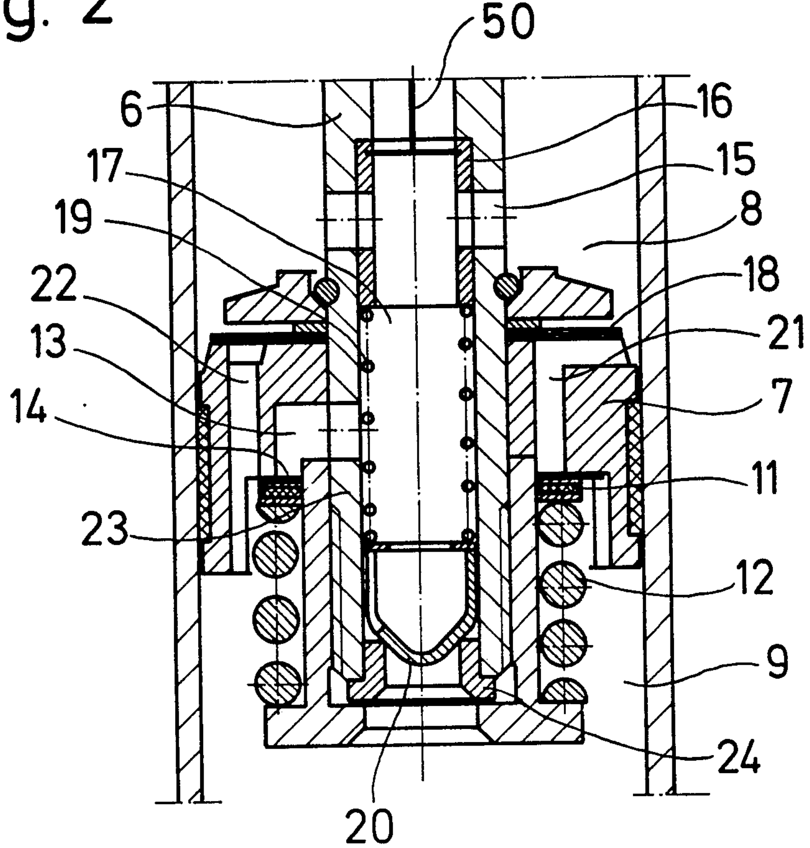


Fig. 3

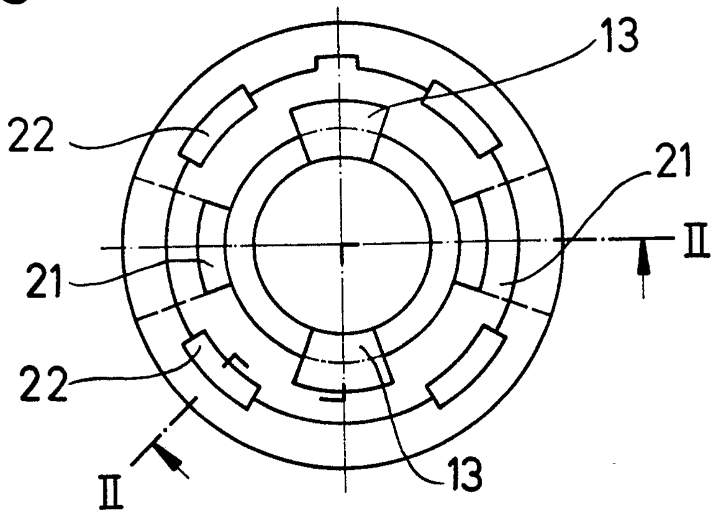


Fig. 4

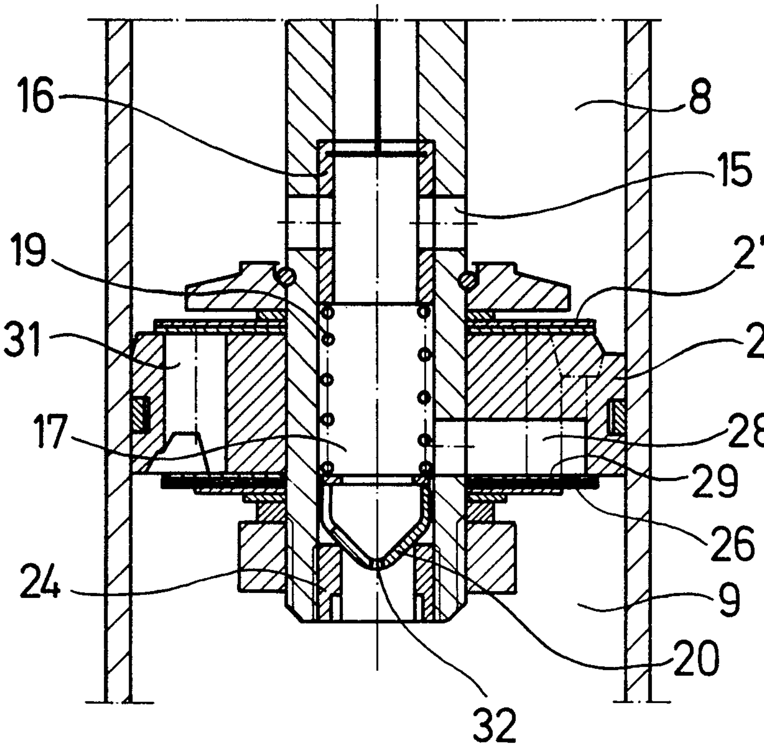


Fig. 5

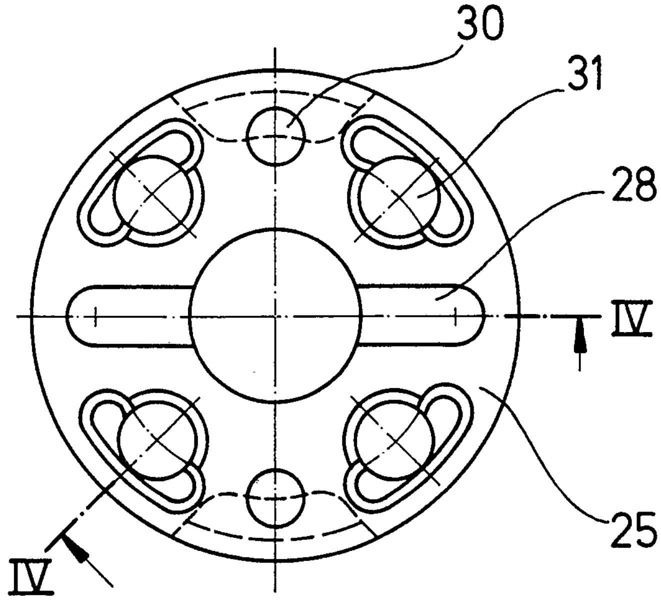


Fig. 6

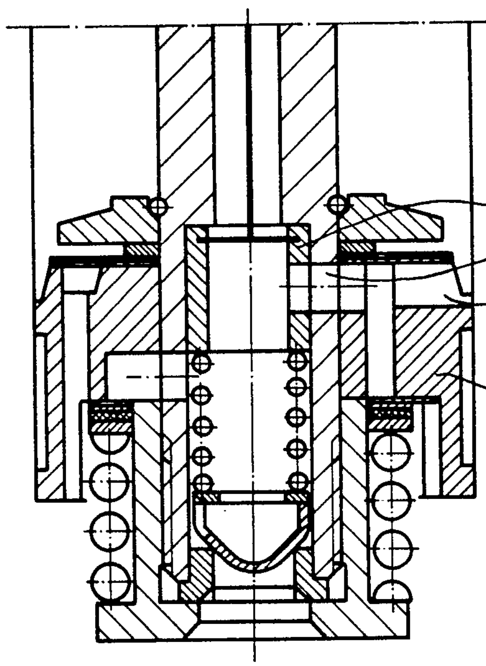


Fig. 7

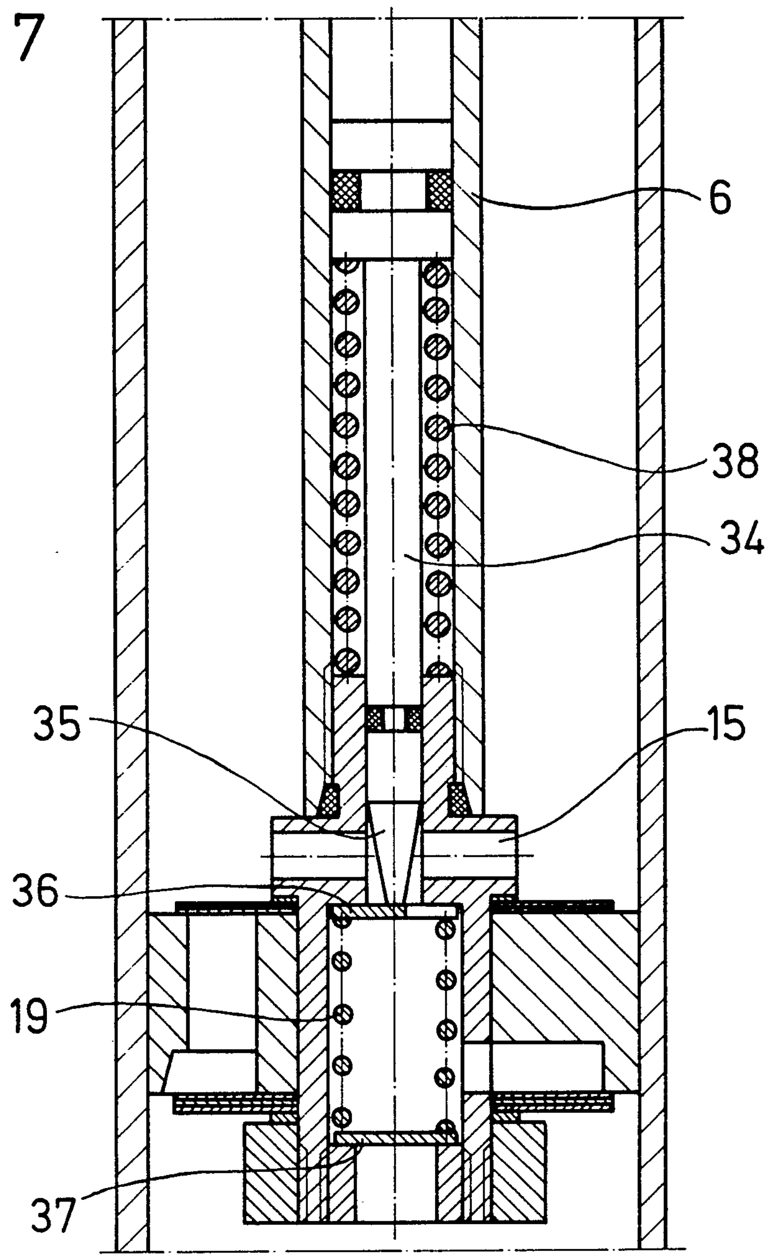
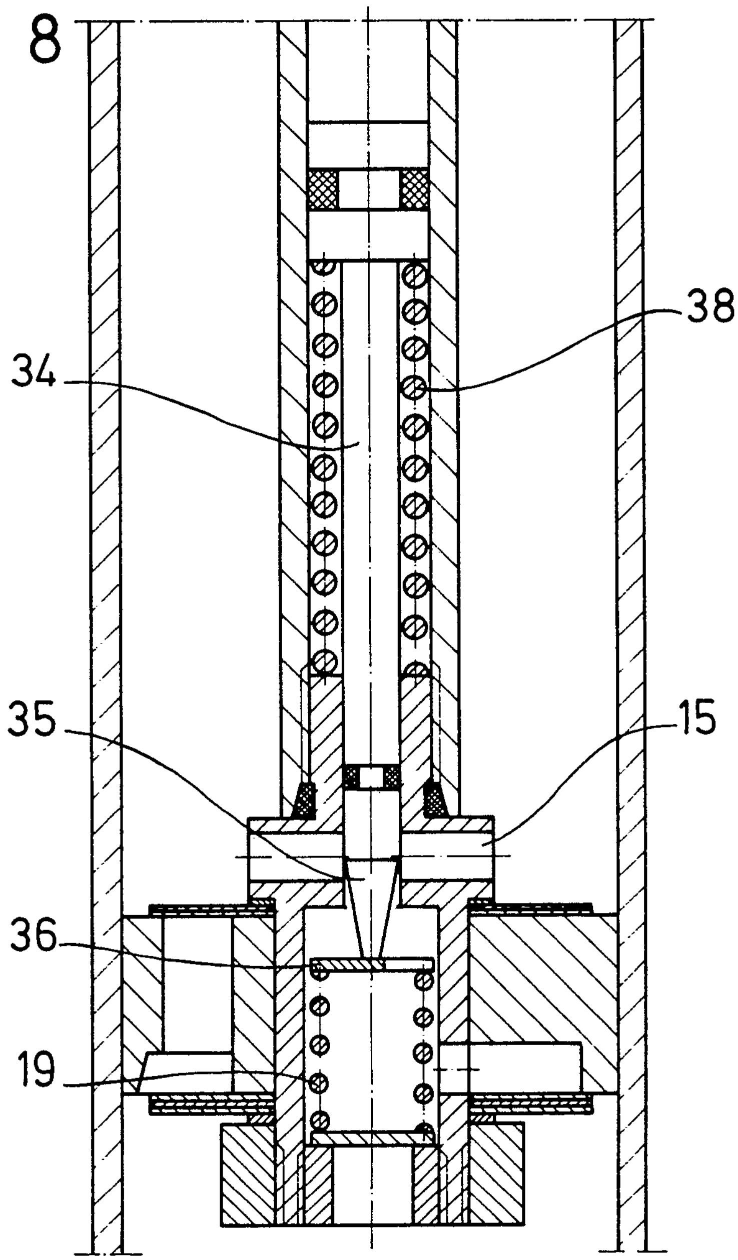


Fig. 8



SPECIFICATION

Vibration damper with variable damping force

The invention relates to a vibration damper with variable damping force for vehicles, in accordance with the opening statement of Claim 1.

Such a vibration damper is known from Fed. German P.S. No. 1,505,522, where the by-pass passage is formed by constricted openings in the piston rod which are closable more or less by a control slider situated in the central bore of the piston rod and thus the degree of damping is influenced in the same way for traction damping and compression damping. In this case the damping force characteristic is varied only in the lower speed range. It is not possible to realise an influence upon the damping characteristic curve which acts over the entire speed range; likewise differing setting of the characteristic variation for the traction and compression stages is not possible.

A different damping force variation for the traction and compression stages is shown in a vibration damper in accordance with Fed. German Publ. Spec. No. 3,438,467 where a by-pass of variable cross-section is combined with a non-return valve. This results in a different by-pass cross-section for the traction and compression directions, whereby a limitedly differing characteristic variation is rendered possible in the lower speed range of the piston movement, but not for the higher speed range. The construction expense of such a design is high and involves complicated assembly, by reason of the multiplicity of the components necessary for the damping force adjustment. Accordingly such a design becomes very expensive in production and also occupies a relatively large overall space in the axial direction, which has a stroke-shortening effect for the piston rod movement.

Furthermore a vibration damper is known from Fed. German Publ. Sp. No. 2,119,531 which comprises two series-arranged damping valve systems which are combined with a blockable passage whereby only one damping valve system can be bridged over. For the optional actuation of the one or the other valve system or their series arrangement it is proposed by Fed. German Pat. Appln. P 34 46 133.7 to effect this by a controllable by-pass. The construction expense of such a damping force adjustment is high and it requires considerable construction space.

It is the problem of the present invention to produce a vibration damper with adjustable damping force which renders possible with simple means a different level of the adjustment of the damping force over the whole speed range for traction and compression damping and with simple assembly possesses good operational reliability, occupies little construction space and guarantees easy fitting.

In accordance with the invention this problem is solved in that a spring-loaded by-pass damping valve allocated to compression damping is arranged parallel to the damping valve allocated to the compression direction, and is placed in the direction of flow before the passage of variable cross-section acting as after-constriction. The by-pass damping valve arranged parallel to the damping valve renders possible a compression damping which is adjustable over the entire speed range, while the adjusting mechanism is very simple in assembly and has high operational reliability and the by-pass damping valve itself consists merely of a valve body and a valve spring, guaranteeing simple fitting. According to the desired compression damping variation, by simple selection of the valve spring or by alteration of its initial stress it is possible to obtain the desired compression damping course over the whole speed range, while such a by-pass valve occupies a small construction space and renders easy fitting possible.

In further development of the invention the by-pass damping valve is arranged in an axial bore of the piston rod journal which receives the piston. This results in an especially favourable construction since no additional construction space at all is required in the axial direction of the unit by the by-pass damping valve. The valve body itself, according to one feature of the invention, is of cup-shaped formation and the cup edge forms the abutment face for the by-pass valve spring. Such a valve body is preferably made as a shaped sheet metal part. It is distinguished by good guidance, possesses low weight and is very stable of form.

According to one feature a very advantageous form of embodiment is obtained in that the by-pass damping valve allocated to compression damping is combined with a damping force adjustment acting during traction damping. Accordingly such a compression damping is readily combinable with known possible devices for adjusting the traction damping. It is especially advantageous to effect the combination with a traction damping system, in which case its adjustment is effected by an addable loaded surface acting upon the traction damping valve.

A very simple alteration of the initial stress of the by-pass valve spring is achieved in accordance with the invention in that this stress is variable by an axial slider and this axial slider is arranged for the variation of cross-section of the by-pass passages. Thus this double function of the axial slider makes it possible, by its longitudinal displacement, to stress the valve spring of the by-pass damping valve initially more strongly, whereby the share of the spring-loaded by-pass damping valve in the compression damping rises more strongly.

In further development of the invention the by-pass valve body comprises a pre-opening cross-section acting in the traction and compression directions. Thus a simple pre-opening variation is achieved by way of the controllable by-pass.

Especially in the case of single-tube vibration dampers where the radial bores arranged in the piston rod must not run over the piston rod seal, an extremely short overall height is obtained for the by-pass device according to the invention in that at least one radial bore of the by-pass passage co-operates with a radial passage arranged in the piston.

The invention will be explained in greater detail below by reference to the forms of embodiment represented in the drawing, wherein:—

Figure 1 shows a form of embodiment of the vibration damper according to the invention in longitudinal section;

Figure 2 shows the piston part according to Fig. 1 in enlarged representation;

Figure 3 shows the piston according to Fig. 2, seen from beneath;

Figure 4 shows a piston part in longitudinal section the piston being intended for a single-tube vibration damper;

Figure 5 shows the piston according to Fig. 4 in a view from beneath;

Figure 6 shows a form of embodiment in which the radial bore of controllable cross-section of the by-pass passage opens into a radial passage arranged in the piston;

Figure 7 shows a form of embodiment in which an axial slider serves for the variation of the initial stress of the by-pass valve spring;

Figure 8 shows the form of embodiment as shown in Fig. 7 with increased initial stress of the by-pass valve spring and reduced cross-section of the by-pass passage.

The vibration damper according to Fig. 1 possesses a cylinder 1 which is clamped concentrically in a container 2 by means of a piston rod guide 3 and a bottom valve 5. A hollow piston rod 6 is guided in the piston rod guide 3 and sealed to the exterior by a piston rod seal 4. The end of the piston rod 6 internal to the cylinder carries a piston 7 which separates a working chamber 8 above the piston from a working chamber 9 beneath the piston. Through the bottom valve 5 the working chamber 9 is in communication with a compensation chamber 10 which is arranged between cylinder 1 and container 2. The piston 7 is provided with passage openings allocated to the traction and compression stages and covered by corresponding damping valves. In this case the damping valve 11 loaded by a valve spring 12 is allocated to the traction stage, while the damping valve 18 consisting of spring washers serves for the compression stage damping. In the bottom valve 5 there are provided a further damping valve 51 allocated to the compression stage and a non-return valve 52 provided with perforations for the traction stage. These perforations are so arranged that in the compression stage they permit access of the pressure fluid in the working chamber 9 to the damping valve. In the piston rod 6 there is provided a by-pass consisting essentially of radial bores 15 and an axial bore 17, the radial bores 15 being variable in cross-section by a slider 16. Pressure chambers 13 which open into the axial bore 17 and form additional pressure-loaded surfaces for the damping valve 11 are allocated to the damping valve 11 responding in the traction stage. A by-pass damping valve consisting of a by-pass valve body 20 and a by-pass valve spring 19 is arranged parallel to the damping valve 18 of the piston 7 which responds in the inward movement of the piston rod 6.

Figs. 2 and 3 show the above-described piston arrangement and formation with the damping force adjusting arrangement in enlarged representation, the parts name hitherto being provided with the same references. In the Figures in each case the by-pass radial bore 15 is represented completely opened by the slider 16, so that in the outward movement of the piston rod 6 the damping fluid acts upon the damping valve 11 through the piston passages 21 allocated to the traction damping. At the same time damping fluid from the working chamber 8 passes through the radial bore 15 and the axial bore 17 into each pressure chamber 13, the surfaces 14 likewise being pressure-loaded. The pressure rise of the damping fluid in the working chamber 8, caused in the outward thrust movement of the piston rod 6, has the effect that the damping valve 11 is pressure-charged both through the piston passages 21 allocated to the traction damping and through the pressure chambers 13, and this pressure force acts against the force of the valve spring 12. This fluid pressure force acting upon the damping valve 11 is greater when the by-pass cross-section is opened, by reason of the pressure-loaded areas 14 formed by the pressure chambers 13, so that a slighter damping action is achieved. On closure of the radial bores 15 by the slider 16, which can be effected for example by rotation or axial displacement of the actuating rod 50, only the fluid pressure in the passages 21 acts upon the damping valve 11, which corresponds to a harder setting.

In the pressure stage, that is in the pushing of the piston rod 6 into the cylinder 1, when the radial bores 15 are closed the damping valve 18, allocated to compression damping, of the piston 7 acts, and damping fluid flows out of the working chamber 9 through the bores 22 and the opened valve cross-section into the working chamber 8. A further series-connected damping valve allocated to compression damping is provided in the bottom valve 5. When the radial bore 15 is opened the by-pass valve body 20 loaded by the by-pass valve spring 19 acts as by-pass damping valve arranged parallel to the damping valve 18, and the damping fluid flows from the working chamber 9 by way of this by-pass damping valve into the axial bore 17 of the piston rod journal 23 and thence through the radial bores 15 into the working chamber 8. Thus when the radial bores 15 are opened a softer piston pressure damping is achieved by the connected by-pass damping valve, and the spring-loaded by-pass damping valve can be made very variable by the strength and initial stress of the by-pass valve spring 19. The arrangement of such a by-pass damping valve on the pressure side renders it possible to adjust the course of the pressure characteristic of the damper, with regard to the driving comfort and driving behaviour of the vehicle, to the desired pressure damping values. According to the position of the slider 16 in relation to the radial bore 15, the spring-loaded by-pass valve body 20 takes over a corresponding proportion of the pressure power of the piston pressure damping. If the slider 16 closes the radial bores 15 completely, then only the damping valve 18 acts for the piston pressure damping, whereby the higher pressure damping is achieved.

In Figs. 4 and 5 there is represented a piston 25 of a single-tube vibration damper which comprises a damping valve 26 consisting of valve discs and allocated to traction damping and a damping valve 27 likewise formed of valve discs for the compression damping. This piston 25 is provided with piston passages 30 allocated to traction damping and with piston passages 31 for the compression damping. Moreover the radial bores 15 are in communication through the axial bore 17 with pressure chambers 28 which form the pressure-charged area 29 for the damping valve 26. The by-pass valve body 20 loaded by the by-pass valve spring 19 is provided with a pre-opening cross-section 32 which is effective in the traction and compression directions when the radial bores 15 are opened. Here again the soft damping force setting is achieved with the radial bores 15 fully opened, while in the traction direction the damping valve 26 is loaded by the piston passages 30 and the pressure-charged areas 29 formed by the pressure chambers 28. By rotation of the slider 16 the radial bores 15 experience a reduction of cross-section so that as a result of constriction the full damping pressure no longer acts upon the pressure-charged surfaces 29 and therefore as from a specific piston speed a damping pressure is established which lies between a maximum and a minimum predetermined characteristic. Thus when the radial bores are closed the maximum characteristic is reached, for only the piston passages 30 act upon the damping valve 26. Even the pre-opening cross-section 32 arranged in the by-pass valve body 20 is ineffective when the radial bores 15 are closed, while this cross-section acts in both directions of movement, that is for traction and compression damping, when the radial bores 15 are opened. The maximum compression damping is likewise achieved with the radial bores 15 closed, the piston passages 31 being effective upon the damping valve 27. When the radial bores 15 are opened the by-pass damping valve consisting of the by-pass valve spring 19, the by-pass valve body 20 and the pre-opening cross-section 32 acts parallel to the damping valves 27. In this case then the minimum compression damping characteristic is present. This compression damping characteristic can be varied by appropriate selection of the by-pass valve spring and its initial stress and by the size of the pre-opening cross-section 32.

The form of embodiment according to Fig. 6 differs from that according to Figs. 1 to 3 essentially in that each radial bore 15 closable by the slider 16 opens into a radial passage 33 of the piston 7. In manner of operation this form of embodiment likewise corresponds to that as described in Figs. 1 to 3.

In Figs. 7 and 8 there is shown a form of embodiment in which the by-pass damping valve allocated to compression damping comprises a valve plate 37 which is loaded by the by-pass valve spring 19, while on the other side this by-pass valve spring 19 bears upon a spring plate 36. In the hollow piston rod 6 there is a pressure-charged axial slider 34 which is axially displaceable, in dependence upon the pressure acting upon it, against the force of a spring 38 and acts with an extension piece 35 upon the spring plate 36, and this axial slider 34 serves at the same time for the variation of cross-section of the radial bores 15. In Fig. 7 these radial bores 15 are fully opened and the spring plate 36 lies with the upper end face on a corresponding stop face. This position corresponds to the minimum characteristic of the compression damping force. By pressure action upon the axial slider 34 the latter is displaced downwards against the force of the spring 38, so that the extension piece 35 entrains the spring plate 36 and thus initially stresses the by-pass valve spring 19. As from a specific distance of inward driving of the axial slider 34 moreover the cross-section of the by-pass radial bores 15 is reduced. In this way variable damping characteristics are achieved for the compression damping over the entire speed range. The maximum compression damping is reached when the axial slider 34 has closed the by-pass radial bores 15.

Since especially compression damping is of decisive importance for travelling comfort, for the above-described spring-loaded by-pass damping valve allocated to compression damping a very wide range for adjustability is achieved. The by-pass damping valve, which consists either of a by-pass valve body 20 or of a valve plate 37, can be varied extensively in damping power by replacement of the by-pass valve spring 19 or by variation of its initial stress, while without the action of the by-pass damping valve the compression-operating damping valve 18 or 27 allocated to the piston produces the maximum characteristic of the compression damping force. The slider 16 or the axial slider 34 renders possible a stepless variation of cross-section of the by-pass cross-section, so that not only are variable damping characteristics achieved over the entire speed range, but also a wide range of adjustabilities results and the adjusting device produced is itself very simple in assembly and fitting.

CLAIMS

1. Vibration damper with adjustable damping force for vehicles, in which a piston provided with damping valves and connected with a piston rod, in a cylinder, divides the damping-fluid-filled cylinder interior space into two working chambers, while for the adjustment of the damping force there is allocated to the damping valves a by-pass variable in dependence upon a control value and acting in the traction and compression directions of the piston rod, which by-pass comprises passages arranged in the piston rod and variable in cross-section by means of a slider, characterised in that a spring-loaded by-pass damping valve (by-pass valve spring 19; by-pass valve body 20, valve plate 37) allocated to compression damping is arranged parallel to the damping valve allocated to the compression direction (18, 27) and is placed in the flow direction before the passage (radial bore 15) of variable cross-section acting as after-constriction.

2. Vibration damper according to Claim 1, characterised in that the by-pass damping valve (by-pass valve spring 19, by-pass valve body 20, valve plate 37) is arranged in an axial bore (17) of the piston rod journal (23) receiving the piston (7, 25).

3. Vibration damper according to Claims 1 and 2, characterised in that the by-pass valve body (20) is of cup-shaped formation and the cup edge forms the abutment face for the by-pass valve spring (19).

4. Vibration damper according to Claims 1 to 3, characterised in that the by-pass damping valve (by-pass valve spring 19; by-pass valve body 20, valve plate 37) allocated to compression damping is combined with a damping force adjustment effective in traction damping.

5. Vibration damper according to Claims 1 to 4, characterised in that the initial stress of the by-pass valve spring (19) is variable by an axial slider (34) and this axial slider (34) is arranged for the variation of cross-section of the by-pass passage (radial bores 15).

6. Vibration damper according to Claims 1 to 5, characterised in that the by-pass valve body (20) possesses a pre-opening cross-section (32) acting in the traction and compression directions.

7. Vibration damper according to Claims 1 to 6, characterised in that to each radial bore (15) of the by-pass passage there is allocated a radial passage (33) arranged in the piston (7).

8. A vibration damper as claimed in Claim 1 substantially as described in the example shown in Figs. 1 to 3, Figs. 4 and 5, Fig. 6 or Figs. 7 and 8.

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