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(54) HYDRAULIC DAMPER

(71) We, TOKICO LTD., a Japanese body corporate of 6—3, Fujimi 1-Chome, Kawasaki-ku, Kawasaki-shi, Kanagawa-ken, Japan, do hereby declare the invention, for which we pray that a patent may be granted to us, and the method by which it is to be performed, to be particularly described in and by the following statement:—

The present invention relates to hydraulic dampers which are adapted to damp or alleviate vibrations of an automobile or the like.

One prior art hydraulic damper of a gas-enclosed type comprises a piston and piston rod assembly slidably fitted in a cylinder containing oil therein, a resisting force generating mechanism mounted on a piston portion of the piston and piston rod assembly to generate hydraulically a resisting force against reciprocating movement of the assembly in both directions, and a high pressure gas chamber for compensating for the change in volume caused by ingress or exit of piston rod relative to the cylinder. The high pressure gas chamber is formed in the cylinder and is partitioned by a free piston or the like, alternatively, the chamber is defined in the upper portion of an annular chamber defined between the outer diameter of the cylinder and an outer tube disposed to surround the cylinder, and the lower portion of the annular chamber is communicated with the interior of the cylinder.

Usually a gas of a very high pressure is enclosed in the cylinder and, therefore, gas enclosing equipment is expensive and, since the high pressure gas acts directly on a seal member disposed in the damper the seal member tends to wear which will deteriorate gas or liquid tightness of the damper. The durability of the damper is impaired and, it is very dangerous if the cylinder should fracture. The seal member is to withstand the high pressure and which increases manufacturing costs.

The present invention provides a hydraulic damper comprising a working chamber receiving hydraulic oil therein, a piston reciprocable in the working chamber and dividing the latter into inner and outer oil chambers, a piston rod with one end thereof secured to the piston and the other end thereof projecting through said outer oil chamber and through one end of the cylinder to the outside, a resisting force generating valve means mounted on the piston for resisting oil flow in both directions between opposite sides of the piston to generate a resisting force upon reciprocation of the piston in the working chamber, and a volume-compensating chamber disposed adjacent to the working chamber and receiving hydraulic oil and gas therein for compensating for the change in volume of the hydraulic oil in the working chamber which is caused by ingress or exit of the piston rod relative to the working chamber, wherein the hydraulic damper further comprises throttling passage means for communicating hydraulic oil in the volume compensating chamber with the inner oil chamber, said throttling passage means having a flow cross section such as to generate a pressure in the inner oil chamber sufficient to open the resisting force generating valve means when the piston executes a contraction stroke of the hydraulic damper.

The invention is further described, by way of example, with reference to the accompanying drawings, in which:—

Figure 1 is a longitudinal sectional view of a hydraulic damper according to the present invention;

Figure 2 is a longitudinal sectional view of a hydraulic damper illustrated as a second embodiment of the present invention;

Figure 3 is a longitudinal sectional view of a third embodiment of the present invention; and

Figures 4 and 5 are characteristic diagrams showing relationship between piston speed and resisting force.

Figure 1 shows a dual tube hydraulic damper having a construction generally similar to that of a prior art hydraulic damper used widely in a suspension system of an automobile or the like. The damper comprises a cylinder 1 filled with hydraulic oil therein to constitute a working chamber. An outer tube 2 surrounds the cylinder 1 to define therebetween an annular volume-compensating chamber A consisting of an oil chamber A_1 and a gas chamber A_2. A piston 3 is slidably fitted in the cylinder 1 and partitions the actuating chamber in the cylinder 1 into inner and outer oil chambers B and C. The piston 3 is secured to one end of a piston rod 4, and the other end of the piston rod 4 passes through the outer oil chamber C and through a rod guide 5 to project outside of the cylinder 1 where it carries a mounting ring 6. A washer 7 is fitted on a reduced diameter portion of the piston rod 4 and bears on a shoulder portion 4*a*. A retainer 8 is disposed overlapping with the washer 7. A valve disc 9 forming a resisting force generating valve mechanism is supported at one side of the outer periphery thereof on an annular projecting portion formed on the piston 3 and, at the other side (the upper side) of the inner periphery thereof on a seat part of the washer 7. The valve disc 9 deflects upward or downward to form an annular passage across the piston on the outer or the inner periphery of the disc 9. The lower end of the outer tube 2 is closed by a bottom cap 10, and a mounting ring 11 is secured to the bottom cap 10 by welding or the like. The lower end of the cylinder 1 is closed by a partitioning wall 12 having a small hole 12*a* therethrough. The inner oil chamber **B** is communicated permanently with the oil chamber A_1 through the small hole 12*a*, a space 13 between the wall 12 and the bottom cap 10, a cut-out portion 12*b* formed in the wall 12, and a passage 14 formed between the bottom cap 10 and the outer periphery of the cylinder 1.

In the embodment shown, the small hole 12*a* is formed in the wall 12, but, in the absence of a wall 12, the hole 12*a* could be formed in the lower end portion of the side wall of the cylinder 1 to interconnect the chambers A_1 and **B** directly.

The construction of the hydraulic damper shown in Figure 1 is generally similar to that of prior art hydraulic damper, but according to the present invention the diameter of the small hole 12*a* is reduced relative to the prior art damper and this enables the gas pressure enclosed in the gas chamber A_2 to be reduced, as will be described in detail hereinafter.

Upon the extension stroke of the damper, i.e., when the assembly consisting of piston 3 and the piston rod 4 moves upward in the drawing, oil in the outer oil chamber C defined on the upper side of the piston 3 and filled with oil will be pressurized and deflects the inner periphery of the valve disc 9 downward and flows into the inner oil chamber **B** whereby a resisting force is generated. An amount of oil corresponding to exit or the upward movement of the piston rod 4 is supplied into the oil chamber **B** from the oil chamber A_1 through the small hole 12*a* with a relatively small resisting force.

Upon the contraction stroke of the damper, i.e., when the assembly 3, 4 moves downward in the drawing, oil in the oil chamber **B** is pressurized to deflect the outer periphery of the valve disc 9 upwardly and will flow into the chamber C. An amount of oil corresponding to the ingress of the piston rod 4 into the cylinder 1 or the downward movement of the piston rod 4 will flow through the small hole 12*a* into the oil chamber A_1, whereby, a resisting force will be generated.

Assuming that a speed range of the piston is 0 to 2 m/s which is observed in usual hydraulic dampers used in vehicle suspension systems and, that the diameters of the piston and the piston rod are 25 mm and 12.5 mm respectively which are of standard sizes used widely in automobiles. Further, it is assumed that a desired resisting force F_N generated in the hydraulic damper upon the contraction or retraction stroke will conform to line "a" in Figure 4 and the maximum value F_N max is 100 kg when the piston speed is 2 m/s.

When the piston rod 4 ingresses into the cylinder 1, an oil of an amount corresponding to the volume of the rod will pass through the small hole 12*a* and a resisting force F_R (which will be referred hereinafter as resisting force of rod ingress) will be generated. The force F_R is denoted by the following equation.

$$F_R = CA^3/a^2 \times V \qquad (1)$$

wherein, C is a constant, A is a cross-sectional area of the piston rod 4, a is a cross-sectional area of the small hole 12*a*, and V is the piston speed. The maximum value F_R max is assumed as 10% of F_N max or 10 kg when the piston speed is 2.0 m/s, (then a value of 3.4 mm is obtained for the diameter of the hole 12*a*,) and which is depicted by line "b" in Figure 4.

Similarly, when the piston 3 moves downward in the cylinder 1 an amount of oil corresponding to the movement of the piston is assumed to pass through the small hole 12*a*, then, a resisting force F_P (which is referred hereinafter as resisting force of piston movement) and is depicted by line "c" in Figure 4, which is obtained from

entering the value 3.4 mm as the diameter of the hole 12*a* into an equation similar to the equation (1).

As shown clearly in Figure 4 the resisting force F_P to piston movement becomes larger than the required resisting force F_N when the piston speed exceeds a predetermined value X, in other words, when the piston speed exceeds the value X the valve disc 9 will deflect to generate the desired or normal resisting force F_N by a pressure generated in the oil chamber B.

By increasing pressure in the compensating chamber A it is possible to deflect the valve disc 9 when the piston speed is below the value X. (It will be noted that the pressure in the chamber A is assumed to be atmospheric pressure in obtaining the resisting force F_R and F_P.

The pressure P in the compensating chamber A required for overcoming the force of the valve disc 9 in a region where the piston speed is below the value X can be denoted by:

$$P=\text{maximum value of } (F_N-F_P)/AP \quad (2)$$

wherein A_P is a cross-sectional area of the piston. Thus, when the diameter of the hole 12*a* is 3.4 mm, the value of P is calculated to be about 2.4 kg/cm^2.

When the diameter of the small hole 12*a* is reduced, the resisting force of rod ingress F_R and the resisting force of piston movement F_P will change as shown in chain line b′, b″, b‴ and c′, c″, c‴ in Figure 4. Thus the pressure P can also be decreased.

In Figure 4, the desired resisting force F_N is shown to change along a line "d" when piston speed is below a predetermined value X_0. The line "d" is attained by a fixed orifice bypassing the valve disc 9 (not shown in Figure 1, but is shown as 3*b* in Figure 2 which will be described later), and defines a starting characteristic of the hydraulic damper.

By decreasing the diameter of the small hole 12*a* further, it is possible to obtain line C^4 or C^5 for the resisting force of piston movement F_P, which shows that a desired characteristics of the damper can be attained with atmospheric gas pressure enclosed in the volume-compensating chamber A by determining the cross-sectional area of the small hole 12*a* suitably.

The description has been shown with reference to a hydraulic damper having a specified dimension, but it will be noted that the diameter of the piston, the diameter of the rod, the maximum piston speed, and the maximum resisting force F max may be changed as desired.

The small hole 12*a* according to the present invention has not been provided in prior art hydraulic dampers, in which the gas pressure in the volume-compensating chamber acts directly on the lower side of the piston (inner oil chamber B) and, thus, when the gas pressure in the compensating chamber is low, a differential pressure sufficient to actuate the resisting force generating valve mechanism (to deflect the valve disc) will not be generated across the piston, whereby, desired characteristics of the hydraulic damper cannot be obtained.

Further description will be made with reference to a prior art hydraulic damper with principal dimensions similar to the first embodiment of the specified dimensions, i.e. the diameter of the piston is 25 mm, the diameter of the piston rod is 12.5 mm, the maximum piston speed is 2 m/s and the maximum resisting force F max is 100 kg. Then, gas pressure P which is required to overcome the resisting force of the valve disc 9 is:

$$P=Fmax/(A_p-A_R)=25 \text{ kg}$$

wherein, A_P is cross-sectional area of the piston, and A_R is cross-sectional area of the piston rod.

Accordingly, it will be understood that the gas pressure enclosed in the hydraulic damper of the present invention can be reduced to as low as one tenth of that in prior art hydraulic damper.

The embodiment illustrated in Figure 2 is a single tube type hydraulic damper adapted to use on a steering wheel and, parts corresponding to Figure 1 are denoted by similar numerals. In this embodiment, a volume-compensating chamber A is formed in the cylinder 1 and is partitioned by a partitioning wall 12 from the actuating chambers B, C. In the volume-compensating chamber A a free piston 15 is slidably fitted in the cylinder to divide the chamber A into an oil chamber A_1 and a gas chamber A_2. A small hole 12*a* is formed in the wall 12. In the drawing, a fixed orifice 3*b* is formed in the piston 3 for interconnecting the oil chambers B and C permanently to improve damping characteristics of the hydraulic damper when piston speed is a predetermined low speed region (starting characteristics).

Figure 5 is a characteristic diagram of the embodiment of Figure 2 and is similar to the diagram of Figure 4. The line a_1 (OA_1A_2) shows a desired resisting force F_N, and the line OA_1 is attained by the fixed orifice 3*b*, and the line A_1A_2 by the deflection of the valve disc 9. The resisting force of rod ingress F_R is shown by line B_1, and the resisting force of piston movement F_P is shown by line c_1.

The damper of the second embodiment is adapted particularly to use with a steering

wheel and is referred hereinafter as steering damper.

The working speed of the piston in a steering damper is usually 0 to 0.6 m/s, and the ratio in the diameter between the piston and the piston rod is usually about 3:1 while that in the hydraulic damper in usual suspension system is about 2:1.

Therefore, the difference between oil flow flowing through the fixed orifice 3*b* and that flowing through the small hole 12*a* is larger than that in the first embodiment, and by defining the diameter of the small hole 12*a* smaller than that of the fixed orifice 3*b*, the line c_1 of the resisting force of piston movement F_P has a gradient steeper than that of the line d of the fixed orifice. The resisting force of rod ingress F_R is, accordingly, has a gradient steeper than that of the first embodiment which is shown by full line b_1'. Since the line c_1 is steeper than the line d, the gas pressure in the gas chamber A_2 can be reduced to atmospheric pressure, and the damper can act satisfactorily in the working range of the piston speed of 0 to 0.6 m/s.

Upon the extension stroke of the damper, the oil chamber C is pressurized, and the pressurized oil can deflect the inner periphery of the valve disc 9 and flow into the oil chamber B. An amount of oil corresponding to exit of the piston rod 4 outward of the cylinder 1 flows into the oil chamber B with a small resisting force generated thereon. The gas in the gas chamber A_2 expands, and the free piston 15 moves accordingly toward the oil chamber C. The gas pressure in the chamber A has to overcome sliding resisting force of the free piston 15.

Figure 3 shows another embodiment of steering damper, and includes a volume-compensating chamber A disposed around the outer periphery of part of cylinder 1. Otherwise, the embodiment is generally similar to Figures 1 and 2, and corresponding numerals are applied to corresponding parts.

An outer tube 16 surrounds the lower portion of the outer periphery of cylinder 1 and a flexible tubular member 17 formed of, preferably, a resilient material such as rubber or the like is clamped at the opposite ends thereof between the outer tube 16 and the cylinder 1, thereby, an annular chamber A_1 for receiving oil therein is formed between the flexible member 17 and the cylinder 1 and an annular chamber A_2 for receiving gas therein is formed between the flexible member 17 and the outer tube 16. A small hole 12*a* interconnecting the oil chambers B and A_1 permanently is formed in the wall of the cylinder 1.

The construction and the operation of the embodiment of Figure 3 is otherwise the same as that of Figure 2, and the detailed description therefore will be omitted. In this embodiment, the flexible cylindrical member 17 will automatically follow the movement of the piston rod 3 in the extending direction when oil in the oil chamber A_1 of the volume-compensating chamber A flows into the oil chamber B through the small hole 12*a*, pressurized gas may not necessarily be enclosed in the gas chamber A_2.

As described heretofore in detail the hydraulic damper according to the present invention, comprises an actuating chamber receiving a piston slidably therein, a volume-compensating chamber disposed adjacent to the actuating chamber for compensating change in volume which is caused by ingress or exit of a piston rod relative to the actuating chamber, and a small hole connecting the actuating chamber to the volume compensating chamber and adapted to generate a resisting force larger than that of a resisting force generating valve mechanism disposed on the piston, whereby, the pressure of gas enclosed in the compensating chamber can be reduced considerably, thus, equipment for filling gas into the hydraulic damper can be simplified and the construction of the damper can be simplified and, further, the durability of the damper can be increased.

WHAT WE CLAIM IS:—

1. A hydraulic damper comprising a cylinder defining a working chamber receiving hydraulic oil therein, a piston reciprocable in the working chamber and dividing the latter into inner and outer oil chambers, a piston rod with one end thereof secured to the piston and the other end thereof projecting through said outer oil chamber and through one end of the cylinder to the outside, a resisting force generating valve means mounted on the piston for resisting oil flow in both directions between opposite sides of the piston to generate a resisting force upon reciprocation of the piston in the working chamber, and a volume-compensating chamber disposed adjacent to the working chamber and receiving hydraulic oil and gas therein for compensating for the change in volume of the hydraulic oil in the working chamber which is caused by ingress or exit of the piston rod relative to the working chamber, wherein the hydraulic damper further comprises throttling passage means for communicating hydraulic oil in the volume compensating chamber with the inner oil chamber, said throttling passage means having a flow cross-section such as to generate a pressure in the inner oil chamber

sufficient to open the resisting force generating valve means when the piston executes a contraction stroke of the hydraulic damper.

2. A hydraulic damper as claimed in claim 1 wherein said volume-compensating chamber comprises an oil chamber formed in the cylinder, and a free piston slidably disposed in the cylinder to form one end wall of the oil chamber and being biassed by the pressure of said gas.

3. A hydraulic damper as claimed in claim 1 wherein said volume-compensating chamber has a flexible wall separating an oil chamber from a gas chamber.

4. A hydraulic damper as claimed in claim 1 in which said volume-compensating chamber comprises an annular chamber disposed about said working chamber.

5. A hydraulic damper as claimed in any preceding claim wherein the pressure of the gas is lower than 3 kg/cm^2 when the damper is in its maximum extended position.

6. A hydraulic damper as claimed in any of claims 1 to 4, wherein the pressure of the gaseous fluid is atmospheric pressure when the damper is in its maximum extended position.

7. A hydraulic damper as claimed in any preceding claim in which said passage means comprises at least one restrictive orifice.

8. A hydraulic damper as claimed in any preceding claim in which said resisting force generating valve means comprises a single two-way pressure valve which opens in either direction responsively to the pressure drop there-across.

9. A hydraulic damper constructed and adapted to operate substantially as herein described with reference to and as illustrated in Fig. 1 of the accompanying drawings.

10. A hydraulic damper constructed and adapted to operate substantially as herein described with reference to and as illustrated in Fig. 2 of the accompanying drawings.

11. A hydraulic damper constructed and adapted to operate substantially as herein described with reference to and as illustrated in Fig. 3 of the accompanying drawings.

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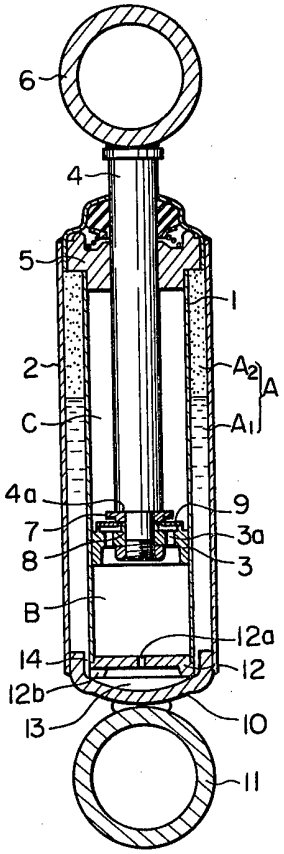
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Sheet 1

Fig. 1



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Fig. 2

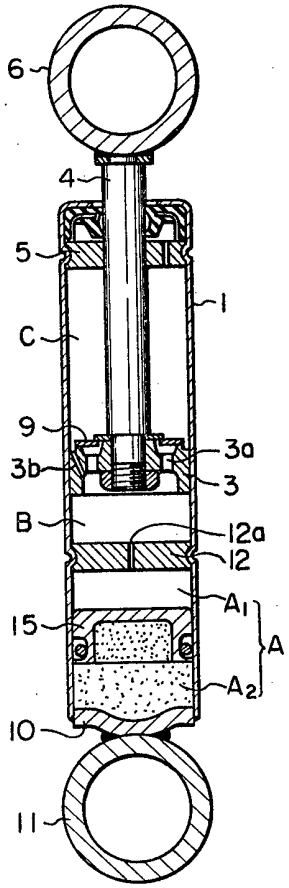
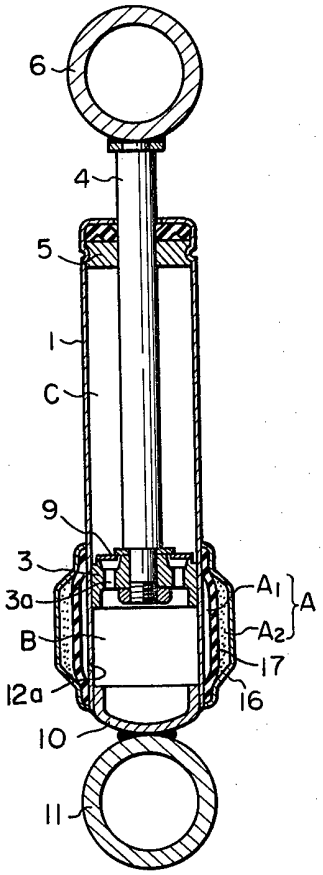
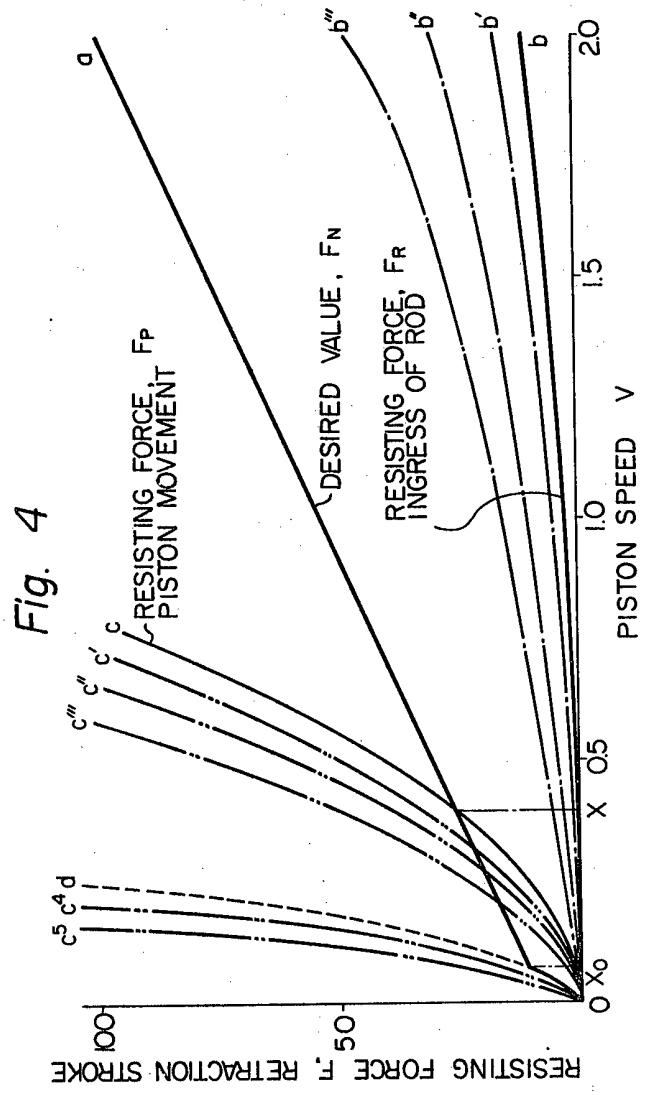


Fig. 3



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Fig. 4



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Fig. 5

