A Study on the Performance Characteristics of Variable Valve for Reverse Continuous Damper

Se Kyung Oh, Young Hwan Yoon, and Ary Bachtiar Krishna

***Abstract*—**Nowadays, a passenger car suspension must has high performance criteria with light weight, low cost, and low energy consumption. Pilot controlled proportional valve is designed and analyzed to get small pressure change rate after blow-off, and to get a fast response of the damper, a reverse damping mechanism is adapted. The reverse continuous variable damper is designed as a HS-SH damper which offers good body control with reduced transferred input force from the tire, compared with any other type of suspension system. The damper structure is designed, so that rebound and compression damping forces can be tuned independently, of which the variable valve is placed externally. The rate of pressure change with respect to the flow rate after blow-off becomes smooth when the fixed orifice size increases, which means that the blow-off slope is controllable using the fixed orifice size. Damping forces are measured with the change of the solenoid current at the different piston velocities to confirm the maximum hysteresis of 20 N, linearity, and variance of damping force. The damping force variance is wide and continuous, and is controlled by the spool opening, of which scheme is usually adapted in proportional valves. The reverse continuous variable damper developed in this study is expected to be utilized in the semi-active suspension systems in passenger cars after its performance and simplicity of the design is confirmed through a real car test.

***Keywords*—**Blow-off, damping force, pilot controlled proportional valve, reverse continuous damper.

I. INTRODUCTION

TO enhance the performance of the suspension system of vehicles, electronically controlled suspension systems have been studied and developed based upon the sky-hook damping algorithm presented by Karnopp [1] in 1974. In the 1990's hydraulic active suspension systems emerged in Japan [2]-[3].

The systems have disadvantages with the increase of weight, energy consumption, and price since it needs separately installed hydraulic semi-active suspension systems [4]-[5]. In 1994, a vehicle with a continuous variable damper appeared [6]. In this system, the orifice in the piston valve was regulated by a step motor of 9 to 140 steps, of which switching impact is reduced, but the reaction speed becomes slow. During the extension stroke, the damping force is changed over a wide range, while the damping force various becomes limited during the compression stroke.

In this study, a continuous variable damper was developed for semi-active suspension systems of passenger cars. It has a wide range of damping force in both compression and extension strokes**.**

NOMENCLATURE

A	[m^2]	Area
F	[kg]	Force
P	[kg/m^2]	Pressure
Q	[m^3/s]	Flow rate
Special characters		
C_d	[-]	Coefficient of discharge
ρ	[kg/m^3]	Density
Subscripts		
1		Inlet pressure (high pressure)
2		Orifice pressure(medium pressure)
3.		Outlet pressure (low pressure)
c		Cross sectional area of fixed orifice
o		Open
s		Shut
sp		Spring
v		Cross sectional area of variable orifice

TABLE I
SKY-HOOK DAMPING ALGORITHM

		Vertical motion of body (less than 2Hz)	
		Up	Down
Relative motion between body and wheels	Compression	Soft	Hard
	Rebound	Hard	Soft

II. NECESSITY OF REVERSE TYPE DAMPER

In sky-hook damping algorithm, the damping force is regulated according to the vertical motion of the body and the relative motion between the body and the wheels that shows four cases as in Table I. In this system, the damper should response within 10 m/s, since the vehicle wheels oscillate faster than 10Hz and additional valve mechanism may be installed to control damping force during compression and extension strokes, respectively. In the reverse damping mechanism, the damping force is regulated mechanically by the inside fluid state with no sensor to detect the relative motion between the wheels and body due to the vertical vibration of body.

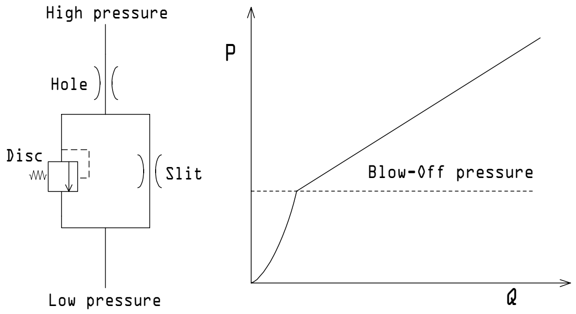
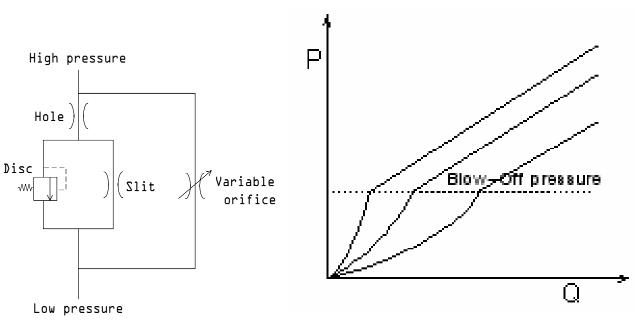


Fig. 1 Passive damper

III. Analysis of Damping

Fig. 2 shows a three stage variable damper, there is an additional oil passage compared with the passive damper as shown in Fig. 1. This shows the hydraulic circuit and pressure flow rate relation. Damping force is controlled by the orifice sizes. Variable orifice size is regulated in three stages, and also damping force is changed in three stages.

The change rates of damping force are different with the flow velocities, because there is only one blow-off pressure. Fig.3 shows the hydraulic circuit and pressure-flow rate relation of the variable damper designed to reduce the difference of the damping force change rates in high and low flow velocities. Three blow-off pressures result as the shutter rotates. The hardest damping force takes place when first and second orifices shut simultaneously. As the shutter rotates, the first variable orifice opens, later on the second variable orifice opens, accordingly pressure-flow rate relation varies. When only the first variable orifice opens, the *P-Q* curves have abrupt changes at blow-off pressures of first sub disc and main disc. When both of first and second variable orifices open, the damping force reduces suddenly at blow-off pressure of two sub discs and main disc.



(a) Hydraulic circuit (b) P-Q curves

Fig. 2 Three stage variable damper

In this study, to overcome such weakness, pilot controlled proportional valve mechanism is utilized as shown in Fig. 4. In reverse continuous variable damper, damping force can be standardized but it is too hard to reduce the slope of the *P-Q* curve after blow-off, since the orifice size cannot be increased due to limited space.

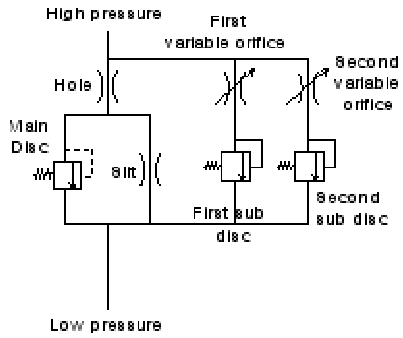
$$Q = C_d \cdot A_c \cdot \sqrt{\frac{2\cdot(P_1 - P_2)}{\rho}} = C_d \cdot A_v \cdot \sqrt{\frac{2\cdot(P_2 - P_3)}{\rho}} \quad (1)$$

$$P_1 - P_2 = \frac{\rho}{2}\cdot\frac{1}{A_c^{\,2}}\cdot\left(\frac{Q}{C_d}\right)^2 \quad (2)$$

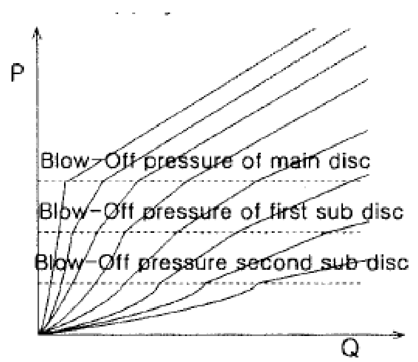
$$P_2 - P_3 = \frac{\rho}{2}\cdot\frac{1}{A_v^{\,2}}\cdot\left(\frac{Q}{C_d}\right)^2 \quad (3)$$

$$F_o = (P_1 - P_2)\times A_o = \frac{\rho}{2}\cdot\frac{1}{A_c^2}\cdot\left(\frac{Q}{C_d}\right)^2\cdot A_o \quad (4)$$

$$\frac{\rho}{2}\cdot\frac{1}{A_c^2}\cdot\left(\frac{Q}{C_d}\right)^2\cdot A_o + \frac{\rho}{2}\cdot\frac{1}{A_v^2}\cdot\left(\frac{Q}{C_d}\right)^2\cdot(A_o - A_s) = F_{spring} \quad (5)$$



(a) Hydraulic circuit



(b) *P-Q* curves

Fig. 3 Variable damper of orifice control

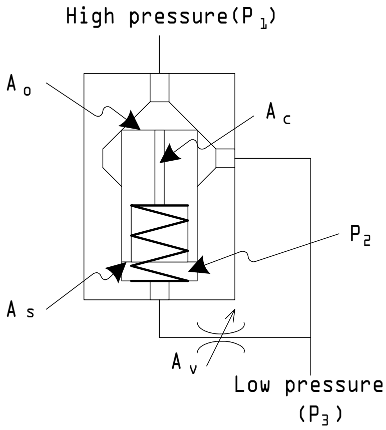


Fig. 4 Schematic of pilot controlled proportional valve

Fig. 4, A_c and A_v are cross sectional areas of fixed and variable orifices, respectively. These two orifices determine control pressure P_2 before blow-off, oil flows through A_c and A_v . Where F_o, F_s , F_{sp} ,A_o and A_s are valve opening and shutting forces, initial load on the valve, two end cross sectional areas of the valve, respectively. Flow rate of blow-off is obtained as a function of A_v

$$Q = C_d \cdot \sqrt{\frac{2 \cdot F_{spring}}{\rho \cdot (\frac{A_o}{A_c^2} + \frac{A_o - A_s}{A_v^2})}} \tag{6}$$

F_{sp} is constant, thus

$A_o < A_s$; Q increases at blow-off as A_v decreases..

$A_o > A_s$; Q decreases at blow-off as . A_v decreases.

From the pressure-flow rate reaction, P-Q curve, at blow-off', P_1-P_2 can be calculated.

$$P_1 - P_3 = \frac{\rho}{2} \cdot (\frac{Q}{C_d})^2 \cdot (\frac{1}{A_c^2} + \frac{1}{A_v^2}) \tag{7}$$

$$\frac{\rho}{2} \cdot (\frac{Q}{C_d})^2 \cdot \frac{1}{A_v^2} = \frac{F_{spring} - \frac{\rho}{2} \cdot (\frac{Q}{C_d})^2 \cdot \frac{1}{A_c^2} \cdot A_o}{(A_o - A_s)} \tag{8}$$

$$P_1 - P_3 = \frac{F_{spring}}{A_o - A_s} - \frac{\rho}{2} \cdot (\frac{Q}{C_d})^2 \cdot \frac{1}{A_c^2} \cdot \frac{A_s}{A_o - A_s} \tag{9}$$

Fig. 5 and Fig. 6 show forces subject to the valve and pressure with respect to flow rate.

In the case of $A_o < A_s$, for the small variation of A_v, blow-off pressure change a lot, for the constant increase of A_v, blow-off pressure reduces much in the beginning. But the reducing rate lessens later. It is relatively hard to get high pressure with the same flow rates.

In case of $A_o >$ A_s, as in Fig. 5, since the angle between F_s curve F_o curve is larger than the case of $A_o < A_s$ blow-off pressure is less sensitive with respect to the A_v variation. When A_v increases slightly, blow-off pressure decreases slightly. High pressure can be obtained with a relatively small blow amount, and all boundary of pressure can be decided easily. Consequently, the case of $A_o > A_s$ is desirable when the pilot controlled proportional damper is utilized on suspension systems.

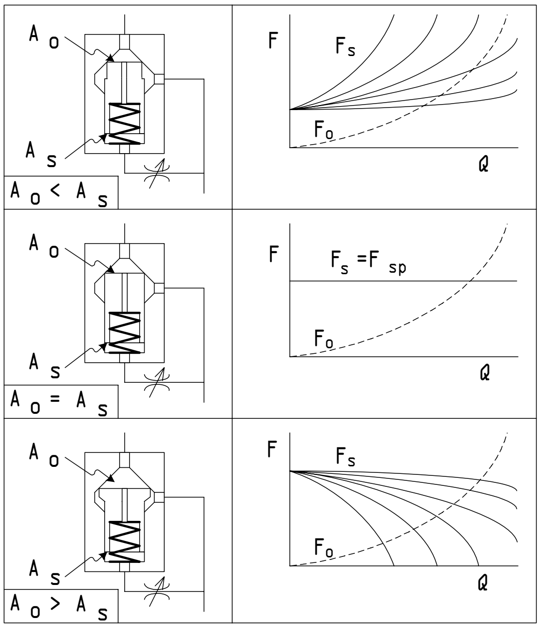


Fig. 5 F-Q diagrams for
$A_0 < A_s$, $A_0 = A_s$, $A_0 > A_s$

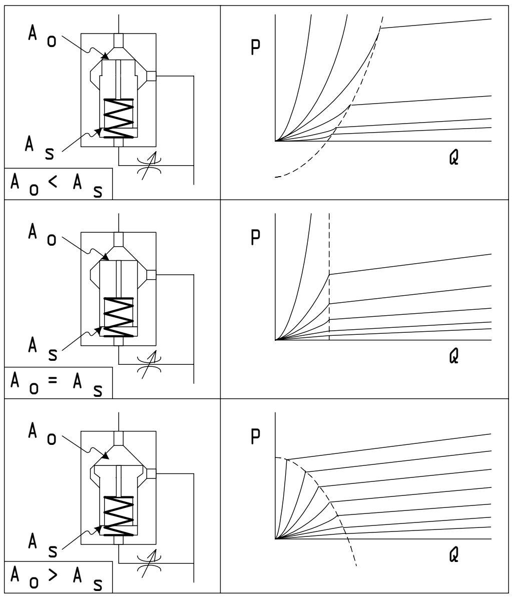


Fig. 6 P-Q diagrams for
$A_0 < A_s$, $A_0 = A_s$, $A_0 > A_s$

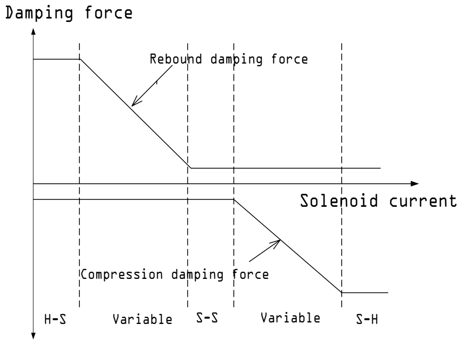


Fig. 7 Damping characteristic of reverse damper (HS-SH damper)

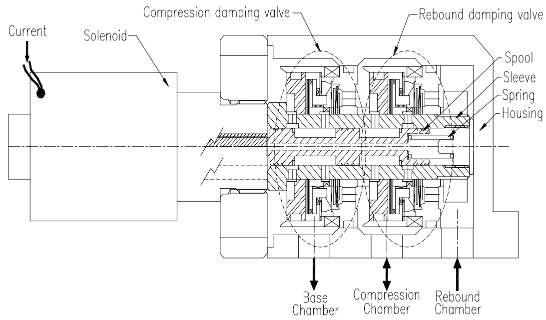


Fig. 8 Sectional drawing of continuous variable valve

IV. Variable Valve Design

Damping force is controlled by the orifice size of the piston valve in most dampers, where the large of compression damping force should be small enough to prevent cavitations in extension chamber.

In this study, to assure enough variable range of damping force in compression, external variable valve is installed, and to get fast response at small pressure change rate after blow-off, a reverse pilot controlled proportional valve is designed. The reverse damper, which is a HS-SH damper as shown in Fig. 7, offers good body control with reduced transferred input force from time, compared with other types of dampers.

The variable valve in designed as reverse type, as in Fig. 7, which has H-S range, where extension damping force is hard and compression damping force is soft, and S-H range with soft extension damping and hard compression damping.

Fig. 8 shows continuous variable valve which was designed in this study, where extension at compression variable valves werehao assembled in series, for oil to flow to compression chamber from extension chamber through extension variable valve in extension stroke, in compression stroke, oil flows to extension chamber from compression chamber through compression valve

The damping force is controlled by regulating disc pressure in pilot valve by the change of spool opening according to the solenoid input currency.

V. Damper Design

Fig. 9 shows variable damper in this study, which is an external variable valve type damper. In piston, then is a hole to assure blow only to extension chamber, and on the extension chamber side of the side thin plate is located to roll as a check valve. In body valve, there is a hole in the same way, to assure oil flows only to compression chamber. To prevent oil leakage from the chambers O-rings are installed in the middle cylinder.

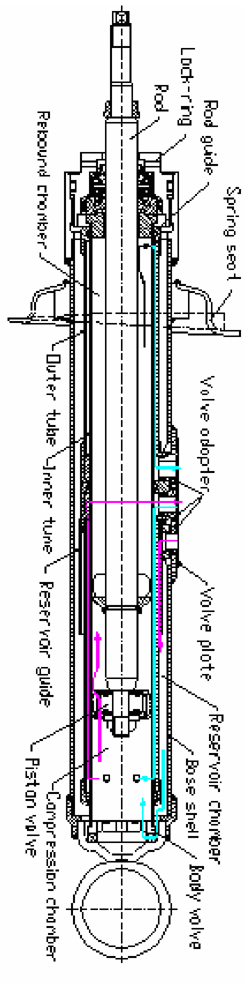


Fig. 9 Sectional drawing of variable damper

During extension stroke, oil flows as follows:

1) Extension chamber → inner cylinder upper hole → extension variable valve →inner cylinder lower hole → compression chamber.

2) Reservoir chamber → body valve → compression chamber.

During compression stroke, oil flows as follows:

3) Compression chamber → piston valve → extension chamber.

Fig. 10 Photograph of reverse continuous variable damper

4) Compression chamber → inner cylinder lower hole → compression variable valve → guide in the reservoir chamber → reservoir chamber.

Flow rate is calculated as the product of the cross sectional area of the piston rod and flow velocity. There is no relation between extension damping forces and compression damping force, which are to be controlled independently using extension variable valve and compression variable valve, respectively

Fig. 10 shows the reverse continuous variable damper developed in this study.

VI. Variable Valve

It takes too much time and endeavors to tune the assembled continuous variable damper using damping force tester. To save time and endeavor, variable valve and damper are tuned separately. Fig. 11 shows schematic diagram of the experiment circuit to test the variable valve during extension and compression stroke. To prevent the influence of test valve opening, compensation flow control valve is used. Fig. 12 and Fig. 13 shows pressure vs. flow rate curve.

As solenoid current becomes large, that is, the fixed orifice size increases the blow-off slope decreases. This result coincides with the results of Fig. 6. Thus, the blow-off slope can be controlled by regulating the fixed orifice size

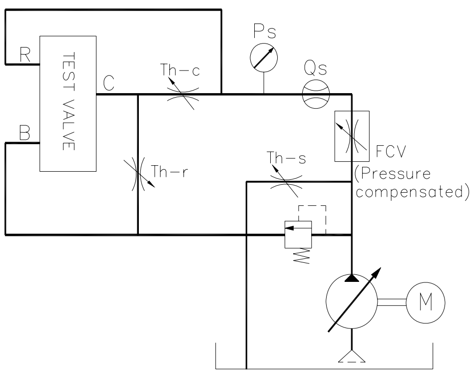


Fig. 11 Schematic diagram of the experimental circuit

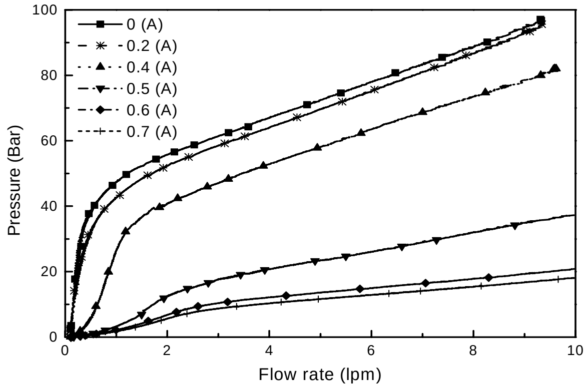


Fig. 12 *P-Q* characteristic at hole orifice 1.0x2

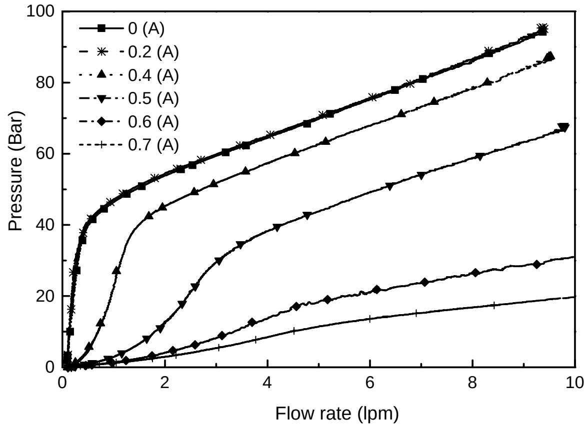


Fig. 13 *P-Q* characteristic at hole orifice 1.0x4

Fig. 14 shows pressure change with respect to solenoid current in certain flow amount, which shows similar change pattern to the damping force change in reverse damper.

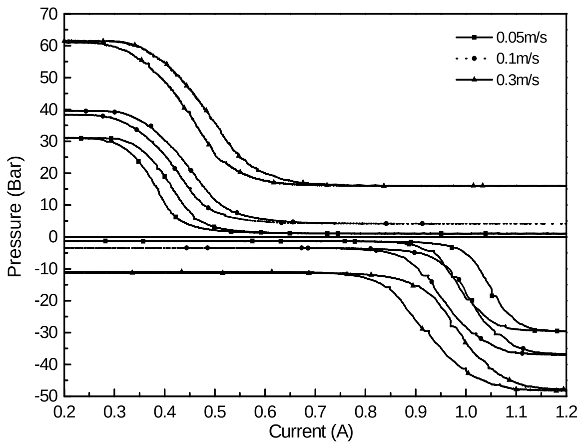


Fig. 14 *P-I* characteristic of variable valve

VII. CONTINUOUS VARIABLE DAMPER

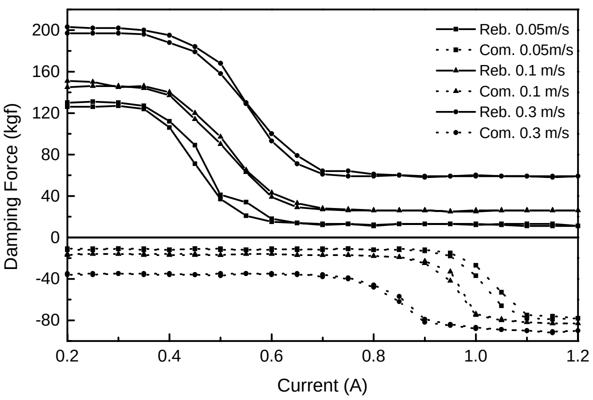


Fig. 15 *F-I* characteristics of reverse continuous variable damper

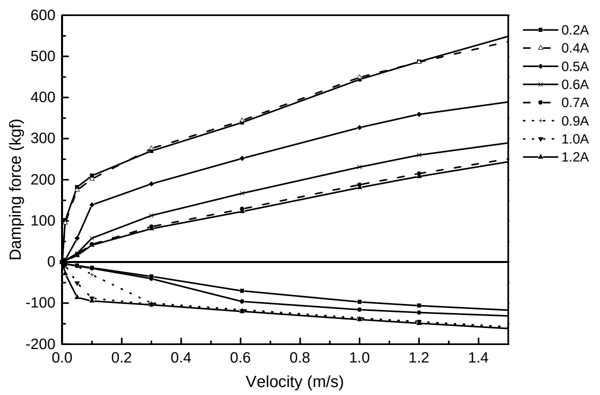


Fig. 16 *F-V* characteristics of reverse continuous variable damper

In Fig. 16, damping force change with respect to solenoid current is shown, when piston velocities are 0.5, 0.1, and 0.3 m/sec. From the figure, it is known that the linearity of damping force change is good, maximum hysteresis is below 20kgf, and reverse mechanism, which is of H-S, S-S, S-H range, is operated. Fig.16 shows damping force with respect to excitation velocity at the different solenoid currents. Excitation displacements are 40 mm, velocities are from 0.01 m/sec to 1.5 m/sec. When solenoid current is 0.2 A, H-S appears, but S-H appears with 1.2 A solenoid current.Damping force variation ranges are 200 kgf in extension at 90 kgf in compression at the velocity of 0.3 m/sec

VIII. CONCLUSION

To develop continuous variable damper for semi-active suspension of a passenger cars, variable damping mechanisms were analyzed and a damper was designed and developed. The performance was confirmed through experiments.

The variable valve was developed with pilot proportional valve, which has wide continuous variance of damping force, and reverse damping mechanism to get prompt response, by the spool opening.

The reverse continuous variable damper was developed, in which extension and compression damping force was tuned independently by extension and compression valves, respectively, and variable valve was placed externally.

Semi-active suspensions are to be in the mainstream of the future passenger car suspensions, because of high performance with low cost and low energy consumption also market share is expected to be extended.

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