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DEVELOPMENT OF A REVERSE CONTINUOUS VARIABLE DAMPER FOR SEMI-ACTIVE SUSPENSION

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ABSTRACT−Semi-active suspension systems are greatly expected to be in the mainstream of future controlled suspensions for passenger cars. In this study, a continuous variable damper for a passenger car suspension is developed. It is controlled actively and exhibits high performance with light weight, low cost, and low energy consumption. To get fast response of the damper, reverse damping mechanism is adapted, and to get small pressure change rate after blow-off, a pilot controlled proportional valve is designed and analyzed. The reverse continuous variable damper is designed as a HS-SH damper which offers good body control with reduced transferred input force from tire, compared with any other type of suspension system. The damper structure is designed, so that rebound and compression damping force can be tuned independently, of which 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. 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 real car test.

KEY WORDS : Semi-active suspension, Continuous variable damper, Reverse damper, Damping force

1. INTRODUCTION

The shock absorber has the task to quickly dampen the body motion of a vehicle and to keep dynamic wheel force changes small. For this purpose conventional shock absorbers with a fixed force curve are mostly used. With the goal to further improve the handling and driving comfort, shock absorber manufacturers and car manufacturers are mutually developing adaptive shock absorber systems. In the first development stage, shock absorbers were created having two to four selectable force curves.

To enhance the performance of vibration control of vehicles, electronically controlled suspension systems have been studied and developed based upon sky-hook damping algorithm presented by Karnopp (1974).

In 1990s hydraulic active suspension system emerged in Japan (Buma *et al.*, 1990, Buma *et al.*, 1991). But the system has disadvantages with the increase of weight, energy consumption, and price since it needs separately installed hydraulic source to control actively. To overcome these disadvantages, semi-active suspension systems have been developed (Knutson, 1991, Wobner and Causeman, 1992). Semi-active suspension systems are greatly expected to be in the mainstream of future controlled suspensions for passenger cars. In 1994, a vehicle with continuous variable damper appeared (Emura *et al.*, 1994). In this system, the orifice in the piston valve is regulated by step motor which has 9 to 140 steps to reduce the switching impact, but the reaction speed becomes slow. During extension stroke, damping force is changed in a wide range, while the damping force varying range becomes narrow during compression stroke to limit the performance tuning.

In this study, a reverse continuous variable damper was developed to control the vibration of vehicle body with reduced transfer force from tire. It is controlled actively and exhibits high performance with lightweight, low cost and energy consumption. It has a wide range of damping force variance in both of compression and extension strokes.

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Table 1. Sky-hook damping algorithm.

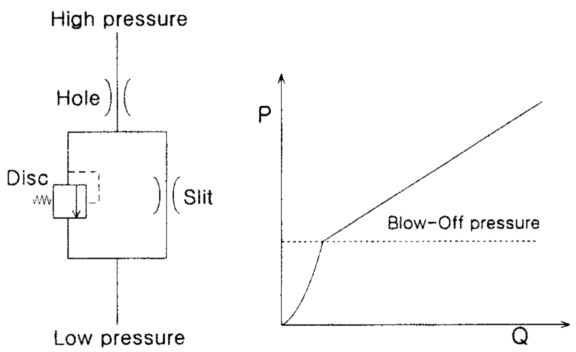
		Vertical motion of body (less than 2 Hz)	
		Up	Down
Relative motion between body and wheels	Compress-ion	Soft	Hard
	Rebound	Hard	Soft

2. DAMPING MECHANISMS OF VEHICLE SUSPENSION

In sky-hook damping algorithm, damping force is regulated according to the vertical motion of body and the relative motion between body and wheels, as shown in Table 1. In this system, the damper should respond within 10 milliseconds, since the vehicle wheels oscillate faster than 10 Hz and additional valve mechanism needs to be installed to control damping force during compression and extension strokes, respectively. From the viewpoint of economy, it is indispensable to develop a damper of which damping force is regulated mechanically by the inside fluid state with no sensor to detect the relative motion between wheels and body.

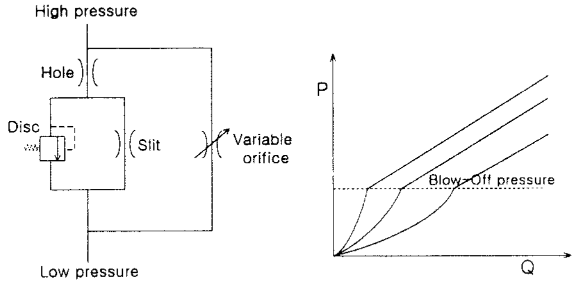
Figure 1 shows hydraulic circuit and pressure-flow rate relation that is *P–Q* curve of a passive damper. In this damper, there is a hole in the piston on which a disk with slots is placed to close the hole. In three stage variable damper shown in Figure 2, there is additional oil passage with variable orifice compared with the passive damper. Damping force is controlled by orifice size. The variable orifice size is regulated in three stages, and also damping force is changed in three stages. There comes blow-off once and the damping force change rate is much reduced after blow-off. In variable damper with two variable orifices designed to reduce the difference of the damping force change rates before and after blow-off, there come three blow-offs as the shutter rotates as shown in Figure 3. 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, and accordingly damping force varies.

When only the first variable orifice opens, the damping force has abrupt changes at blow-offs of the first sub disc and main disc. When both of the first and the second variable orifices open, the damping force reduces suddenly again at blow-offs of two sub discs and main disc. In this



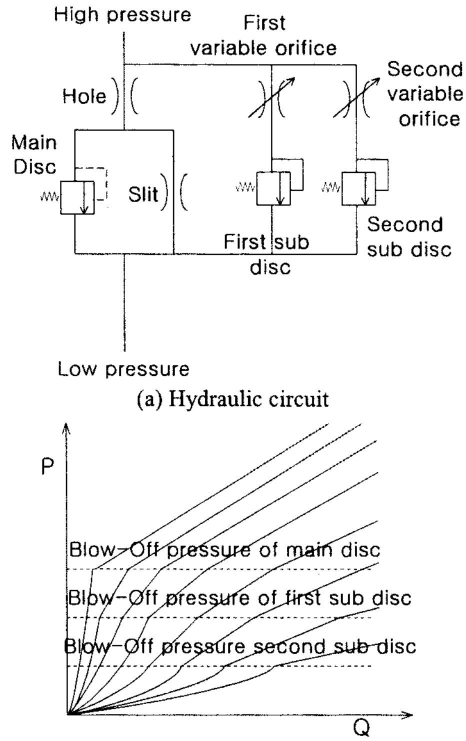
(a) Hydraulic circuit (b) $P-Q$ curve

Figure 1. Passive damper.



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

Figure 2. Three stage variable damper.



(b) $P-Q$ curves

Figure 3. Variable damper of orifice control.

variable damper, the damping force can be standardized but it is too hard to reduce the damping force change rates enough after blow-off, since the orifice size cannot be increased sufficiently due to the limited space.

In this study, to overcome such weaknesses, pilot controlled proportional valve mechanism is utilized as shown in Figure 4. In Figure 4, A_c and A_v are cross sectional areas of fixed and variable orifices. These two orifices determine control pressure P_2 before blow-off, oil flows through A_c and A_v. Flow rate Q becomes as the below, with flow coefficient C_d and fluid density ρ.

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

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

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

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

F_o, F_s, F_{sp}, A_o and A_s are valve opening and shutting

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

forces, initial spring load on the valve, two end cross sectional areas of the valve, respectively. When F_o and F_s are the same, blow-off takes place, and F_{sp} becomes Flow rate of blow-off is obtained as a function of A_v

$$Q = C_d \cdot \sqrt{\frac{2 \cdot F_{sp}}{\rho \cdot \left(\frac{A_o}{A_c^2} + \frac{A_o - A_s}{A_v^2}\right)}} \quad (6)$$

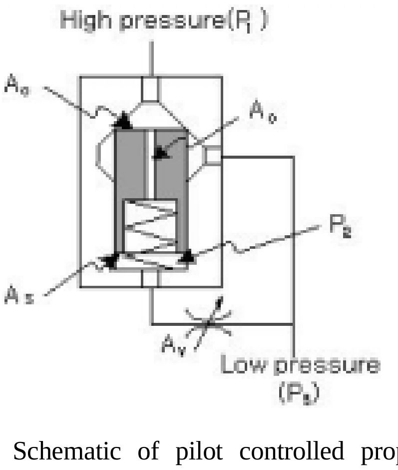


Figure 4. Schematic of pilot controlled proportional valve.

F_{sp} is constant, thus

$A_o < A_s$; Q is increases at blow-off as A_v decreases
$A_o = A_s$; Q is constant at blow-off as A_v decreases
$A_o > A_s$; Q is decreases at blow-off as A_v decreases

From the pressure-flow rate reaction that is P–Q curve, P_1–P_2 can be calculated at blow-off

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

$$\frac{\rho}{2} \cdot \left(\frac{Q}{C_d}\right)^2 \cdot \frac{1}{A_c^2} = \frac{F_{sp} - \frac{\rho}{2} \cdot \left(\frac{Q}{C_d}\right)^2 \cdot \frac{1}{A_c^2} \cdot A_o}{(A_o - A_s)} \quad (8)$$

$$P_1 - P_3 = \frac{F_{sp}}{A_o - A_s} - \frac{\rho}{2} \cdot \left(\frac{Q}{C_d}\right)^2 \cdot \frac{1}{A_c^2} \cdot \frac{A_s}{A_o - A_s} \quad (9)$$

Figure 5 shows hydraulic pressure with respect to the varying flow rate in the pilot controlled proportional valves with different cross sectional areas of two ends. In case that A_o is smaller than A_s, for the small variation of A_v, blow-off pressure changes a lot, for the constant increase of A_v, blow-off pressure decrease fast in the beginning, the decreasing rate lessons later. It is relatively hard to get high pressure with a certain flow rate. This case is not recommended to use in damper design, because it is hard to get required damping force at low velocities. In case that A_o is lager than A_s, it is said to be

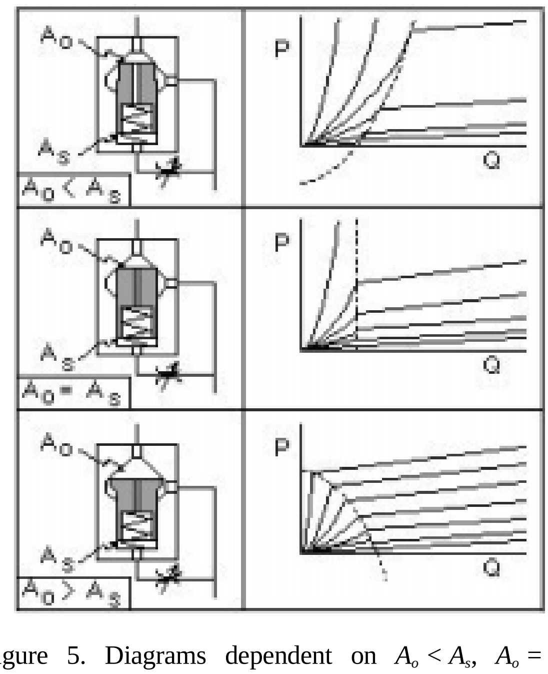


Figure 5. Diagrams dependent on $A_o < A_s$, $A_o = A_s$, $A_o > A_s$.

that blow-off pressure is less sensitive with respect to A_v variation than the former case. When A_v increases simply, blow-off pressure decreases simply. High pressure can be obtained at a relatively small flow rate 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 in suspension systems.

3. DESIGN OF REVERSE CONTINUOUS VARIABLE DAMPER

Damping force is controlled by the orifice size of the piston valve in most dampers. The variance range of compression damping force should be small enough to prevent cavitation in extension chamber. In this study, to assure enough variance range of damping force in compression stroke, external variable valve is installed, and to get fast response to small pressure change rate after blow-off, a pilot controlled proportional valve is designed, where reverse mechanism is adapted.

Reverse damper, which is a HS-SH damper as in Figure 6, offers good body control with reduced transfer force from tire, compared with other types of dampers. In H-S range, extensive damping force is hard and compressive damping force is soft, and S-H range represents soft extension damping and hard compression damping. Figure 7 shows continuous variable valve designed in this study, where extension and compression variable valves are 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 current.

Figure 8 shows variable damper designed in this study, which is an external variable valve type damper. In piston, there is a hole, where the thin plate is located on the extension chamber side to assure oil flow only to extension chamber like a check valve. In body valve, there is a similar hole to assure oil flow only to compression chamber. To prevent oil leakage from the chambers, O-rings are installed in the middle cylinder. During extension stroke, oil flows as the below.

(1) Extension chamber → inner cylinder upper hole → extension variable valve → inner cylinder lower hole → compression chamber. Flow rate is calculated as the product of the difference between cross sectional areas of piston and piston rod and piston velocity.

(2) Reservoir chamber → body valve → compression

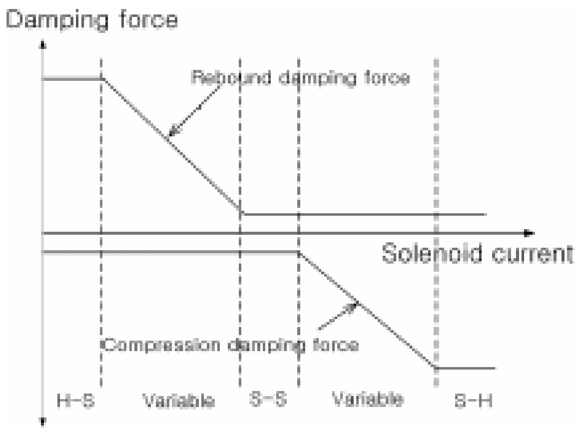


Figure 6. Damping characteristic of reverse damper (HS-SH damper).

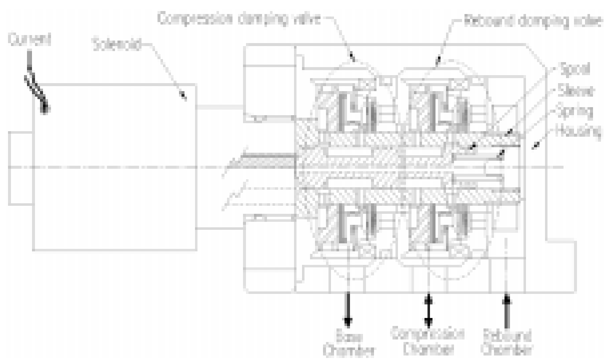


Figure 7. Sectional drawing of continuous variable valve.

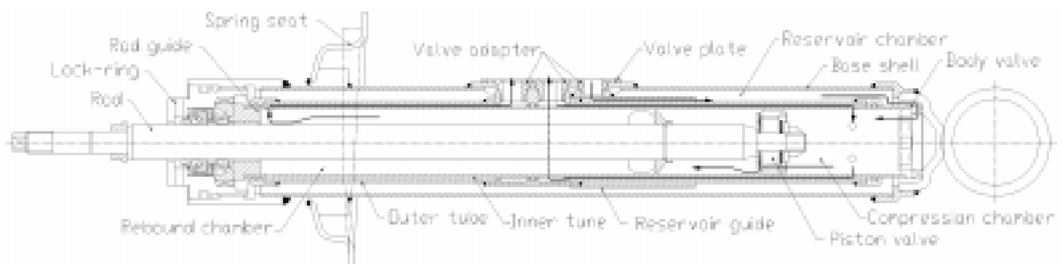


Figure 8. Sectional drawing of variable damper.

chamber. Flow rate is calculated as the product of the cross sectional areas of the piston rod and flow velocity. During compression stroke, oil flows as the below.

(1) Compression chamber → piston valve → extension chamber. Flow rate is calculated as the product of the difference between the cross sectional areas of piston and rod and flow velocity.

(2) 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 and compression damping forces, which are controlled independently using extension and compression variable valves, respectively.

4. RESULTS AND DISCUSSION

It takes too much time and endeavor for tuning the assembled continuous variable damper. To save time and endeavor, variable valve and damper are tuned separately. Figure 9 shows schematic diagram of the experiment circuit to test the variable valve during extension and compression strokes. To prevent the influence of test valve opening, compensation flow control valve is used. Figure 10 shows pressure change with respect to flow rate. As solenoid current becomes large, the fixed orifice size increases and the blow-off slope decreases. Thus, the blow-off slope is said to be controllable by regulating the fixed orifice size. In Figure 11, damping force change with respect to solenoid current is shown when piston velocities are 0.5, 0.1, and 0.3 m/sec, from which it is found to be the same pattern as shown in Figure 6. It is also known that the linearity of damping force change can be achieved, maximum hysteresis is below 20 N, and reverse mechanism is operated, composed of H-S, S-S, and S-H range. In Figure 12, damping forces with respect to excitation velocities are shown at different solenoid

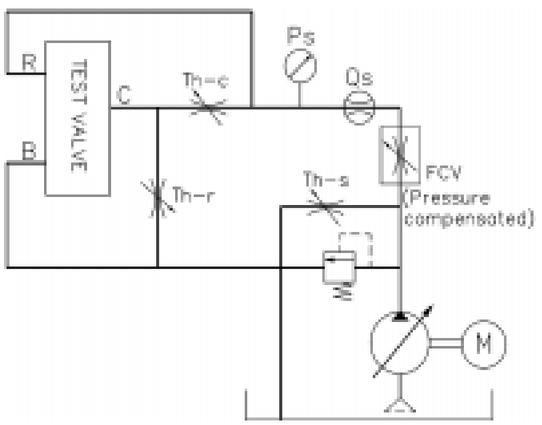


Figure 9. Schematic diagram of the experimental circuit.

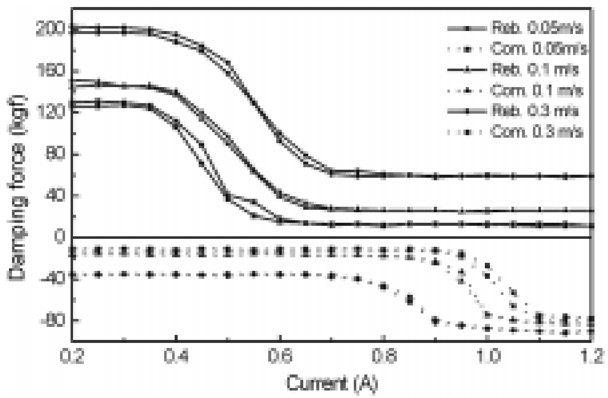


Figure 11. F–I characteristics of reverse continuous variable damper.

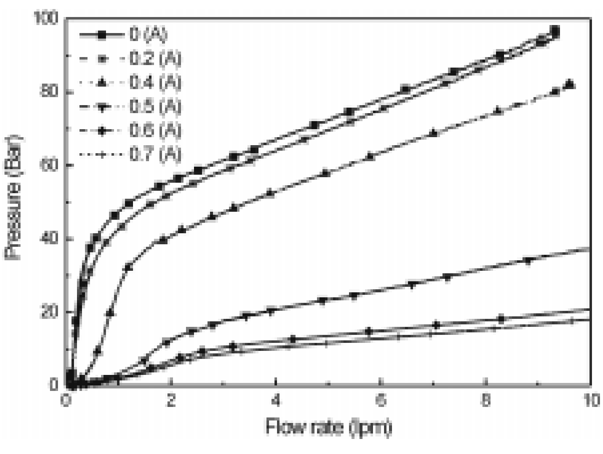


Figure 10. P–Q characteristics at hole orifice $\phi 1.0 \times 2$.

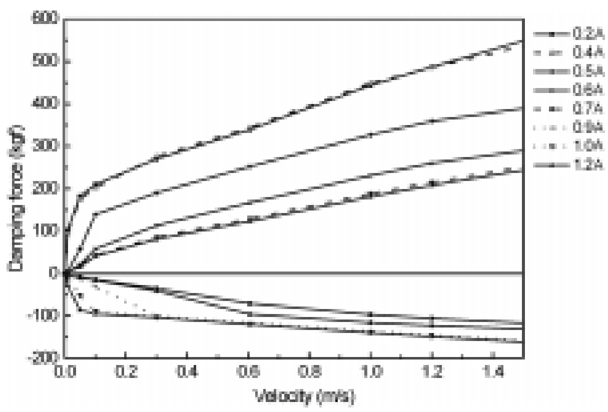


Figure 12. F–V characteristics of reverse continuous variable damper.

currents. Tolerance of 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 at 1.2 A solenoid current. Damping force variation ranges are 200 N in extension and 90 N in compression at the velocity of 0.3 m/sec.

5. CONCLUSION

For the vibration control of vehicle body with light-weight, low cost and energy consumption, damping mechanisms are investigated and a damper for semi-active suspensions was developed. The performance was confirmed through experiments. The results are summarized as the below.

(1) Reverse damping mechanism is adapted to reduce the transferred force from tire.

(2) Pilot controlled proportional valve is used to get small pressure change rate after blow-off and wide continuous variance of damping force.

(3) The damper structure is designed so that rebound and compression damping force can be tuned independently using variable valve placed externally.

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 real car test.

REFERENCES

Buma, S., Kondo, H., Sato, K. and Aburaya, T. (1990). Development of the Toyota active control suspension for the 1989 Celica. *Toyota Technical Paper.* **40**, **2**.

Buma, S., Satou, H., Yonekawa, T., Ohnuma, Y., Hattori, K. and Shiguhara, M. (1991). Synthesis and development of the active control suspension. *Journal of Japanese Society of Mechanical Engineers(C)* **57**, **534**.

Emura, J., Kakizaki, S., Yamaoka, F., Nakamura, M. (1994). Development of the semi-active suspension system based on the sky-hook damper theory. *SAE Paper No. 940863*.

Karnopp, D. C., Grosby, M. J. and Harward, R. A. (1974). Vibration control using semi-active force generator, American Society of Mechanical Engineers, *Journal of Engineering for Industry* **98**.

Knutson, D. (1991). Selection and design of electro-hydraulic valves for electrically-controlled automotive suspension system. *SAE Paper No. 912500*.

Wobner, F. and Causeman, P. (1992). Continuously adjustable shock absorber, *Proceedings of International Symposium on Advanced Vehicle Control 923055*.