A Study on the Improvement of the Load Pressure Feedback Mechanism of the Proportional Pressure Control Valve

Inho Oh*, Jiseong Jang, Illyeong Lee***,**
Daijong Chung** and Sunghyun Cho******

(Received April 22, 1998)

The proportional pressure control valve having versatile functions and higher performance is an essential component in the open loop controlled rear wheel steering gear of the four wheel steering system on a passenger car. In this study, the authors suggest a new type of load pressure feedback mechanism which can make it easy to change the control range of load pressure without changing the capacity of solenoid. The concept of the suggested mechanism, composed of the pressure chamber with throttles in series, was described. The mathematical model was derived from the rear wheel steering gear system consisting of a valve and a cylinder for the purpose of analyzing the valve characteristics. And the programme for computing the characteristic of the valve was developed. Experiments were carried out to confirm the performance of the valve and computations were performed to ascertain the usefulness of the developed programme. The results from the computations fairly coincide with those from the experiments. The results from the experiments and computations show that the performance of new valve is as good as that of the already developed one and the new valve has an advantage in the easiness in varying the control range of load pressure.

Key Words: Four Wheel Steering System, Rear Wheel Steering Gear, Proportional Pressure Control Valve, Load Pressure Feedback Mechanism

1. Introduction

In recent years, studies on the four wheel steering(4WS) systems which can improve the stability and maneuverability of a passenger car have been reported (Kanazawa and Edahiro, 1990) and the applications of 4WS systems to the passenger cars are introduced (Eguchi et al., 1987; Hosokawa et al., 1988).

The rear wheel steering gear, which is composed of a proportional pressure control valve and a spring centered cylinder, controls the steering angle and phase of the rear wheels by controlling the pressure at the cylinder. It has a few advantages such as the compactness in size, the simplicity of the system composition and the feasibility of the open loop control.

The proportional pressure control valve is an essential component in the above mentioned rear wheel steering gear and it should have not only versatile functions such as the directional control, the proportional pressure control and the unloading but also high performance on the control range of load pressure, the linearity, the hysteresis characteristic and especially the response characteristic. Therefore, a direct operating 4-way pressure control valve would be an ideal choice and the capacity of power source of the valve should be as small as possible.

It is hard to find such a valve satisfying the above mentioned functions and performance criteria in the usual industrial devices (Korea UCD Co., 1991; Parker Fluidpower, 1989). There is a direct operating proportional pressure control

* Korea Maritime University, Pusan, Korea
** Graduate School, Tokyo Institute of Technology, Japan
*** Pukyong National University, Pusan, Korea
**** R&D Centre, Mando Machinery Co., Ltd., Seoul, Korea

valve for the 4WS system using reaction pin mechanism for the pressure feedback method to get a wide control range of load pressure while the capacity of proportional solenoid is as small as possible (Komatsu and Akaiwa, 1988). But this type has a limitation in designing the valve. For example, if higher load pressure is required, the capacity of solenoid should be increased and/or the sectional area of reaction pin should be decreased.

Thus, in this study, the authors suggest a new type of pressure feedback mechanism for the proportional pressure control valve. The proportional pressure control valve with the mechanism suggested in this study can obtain the higher load pressure with the solenoid of relatively small capacity. The performance of the valve was investigated through the experiments and simulations and the results from the simulations were compared with those of the experiments.

2. Suggestion of a New Type of Load Pressure Feedback Mechanism

The new type of load pressure feedback mechanism is composed of a feedback pressure chamber and the throttles in the inlet and outlet oil passages connected to the feedback pressure chamber in series. Figure 1 shows the concept of the load pressure feedback mechanism and the operating principle of the valve with the new mechanism is shown in Fig. 2.

In Fig. 1, if the volume between the throttle R_1 and throttle R_2 is very small, the influence due to the compressibility of oil can be neglected. Thus, the flow rate through the inlet and outlet oil passages can be represented as follows.

$$Q=C_1 \cdot A_1\sqrt{\frac{2}{\rho}(P_l-P_f)} = C_2 \cdot A_2\sqrt{\frac{2}{\rho}(P_f-P_t)} \quad (1)$$

where P_l is the load pressure at the load line, P_f is the pressure at the feedback pressure chamber, P_t is the pressure at the return line, C_1 and C_2 are the discharge coefficients of the throttles R_1 and R_2, respectively. A_1 and A_2 are the sectional areas of the throttles R_1 and R_2, respectively.

From Eq. (1), the relation among P_l, P_f and P_t can be obtained as follows:

$$P_l=\left(1+\frac{C_2^2 \cdot A_2^2}{C_1^2 \cdot A_1^2}\right) \cdot P_f-\frac{C_2^2 \cdot A_2^2}{C_1^2 \cdot A_1^2} \cdot P_t. \quad (2)$$

Supposing that the value of $C_2^2 \cdot A_2^2/C_1^2 \cdot A_1^2$ in the parenthesis of the first term of the right hand side of Eq. (2) is much larger than 1, the Eq. (2) can be simplified as

$$p_l=\frac{C_2^2 \cdot A_2^2}{C_1^2 \cdot A_1^2} \cdot (P_f-P_t). \quad (3)$$

If C_2/C_1 is constant in Eq. (3), (P_f-P_t) is proportional to P_l. Discharge coefficients C_1 and C_2 can be described as the functions of Reynolds number (Takenaka and Urata, 1970). They are anticipated to vary synchronously as the throttles R_1 and R_2 are connected in series. Therefore, the ratio of C_2 to C_1 might become constant unless the range of Reynolds number is in the vicinity of zero value. The relation between P_l and P_f can be proportional if the value of P_t is near zero. Thus, this mechanism can enable P_f to vary proportionally to P_l if the sizes of throttles R_1 and R_2 are chosen appropriately.

When A_2 is much larger than A_1, the value

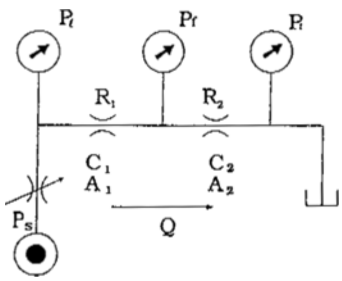

Fig. 1 Conceptional diagram of a new type of load pressure feedback mechanism.

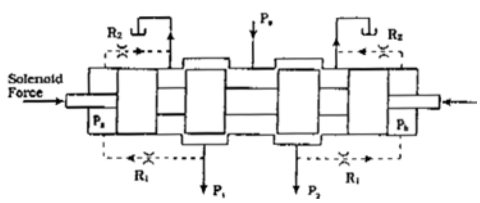

Fig. 2 Schematic of the proportional pressure control valve with the new mechanism.

of P_f is much smaller than that of P_l. In such case, we can make a direct operating proportional pressure control valve that can be actuated by the solenoid of relatively small capacity and easily change the control range of the load pressure by changing the combination of the throttles.

3. Mathematical Model of the New Valve

To investigate the characteristics of the proportional pressure control valve with the new mechanism, the mathematical model of the valve shown in Fig. 3 is derived.

The equations which represent the flow rates [m^3/s] at throttles in the valve are as follows.

$$Q_1 = C_{d1} \cdot w \cdot (u+x)\sqrt{\frac{2}{\rho}(P_s - P_1)} \quad (4)$$

$$Q_2 = C_{d2} \cdot w \cdot (u-x)\sqrt{\frac{2}{\rho}(P_s - P_2)} \quad (5)$$

$$Q_3 = C_{d3} \cdot w \cdot (u+x)\sqrt{\frac{2}{\rho} \cdot P_2} \quad (6)$$

$$Q_4 = C_{d4} \cdot w \cdot (u-x)\sqrt{\frac{2}{\rho} \cdot P_1} \quad (7)$$

where C_{d1}, C_{d2}, C_{d3} and C_{d4} are the discharge coefficients of the throttles, w is the area gradient of opening[m^2/m] ($w = \pi \cdot d_s$, d_s is the diameter of the spool section[m]), u is the underlap of spool[m], x is the displacement of spool[m], P_s is the supply pressure[Pa] and P_1 and P_2 are the pressures at lines[Pa].

The followings are the equations representing the flow rates at throttles in the inlet and outlet oil passages[m^3] connected to the feedback pressure chambers.

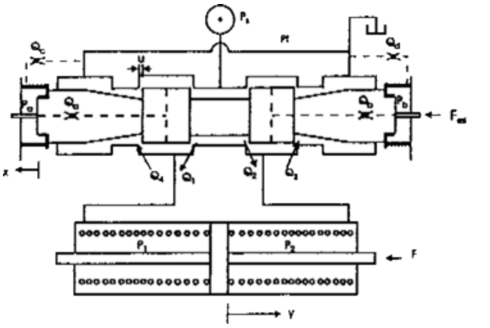

Fig. 3 Schematic of the rear wheel steering gear.

$$Q_a = C_{da} \cdot A_a\sqrt{\frac{2}{\rho}(P_1 - P_a)} \quad (8)$$

$$Q_b = C_{db} \cdot A_b\sqrt{\frac{2}{\rho}(P_2 - P_b)} \quad (9)$$

$$Q_c = C_{dc} \cdot A_c\sqrt{\frac{2}{\rho} \cdot P_a} \quad (10)$$

$$Q_d = C_{dd} \cdot A_d\sqrt{\frac{2}{\rho} \cdot P_b} \quad (11)$$

where C_{da}, C_{db}, C_{dc} and C_{dd} are the discharge coefficients of the throttles, A_a, A_b, A_c and A_d are the sectional areas of each throttle[m^2], and P_a and P_b are the pressures at feedback pressure chambers[Pa].

Applying the continuity equation to each oil passage and chamber yields the followings.

$$Q_1 = Q_4 + Q_a + A_p\frac{dy}{dt} + \frac{V_1}{\beta}\frac{dP_1}{dt} \quad (12)$$

$$Q_3 = Q_2 - Q_b + A_p\frac{dy}{dt} - \frac{V_2}{\beta}\frac{dP_2}{dt} \quad (13)$$

$$Q_a = Q_c - A_s\frac{dx}{dt} + \frac{V_a}{\beta}\frac{dP_a}{dt} \quad (14)$$

$$Q_b = Q_d + A_s\frac{dx}{dt} + \frac{V_b}{\beta}\frac{dP_b}{dt} \quad (15)$$

where A_p is the sectional area of piston[m^2], A_s is the sectional area of spool[m^2], V_1 and V_2 are the volumes of each chamber in cylinder[m^3] including the volumes of oil pipes, V_a and V_b are the volumes of each feedback pressure chamber [m^3] and β is the bulk modulus of elasticity of oil [Pa].

The equations which represent the motions of the spool and piston can be obtained by applying Newton's second law.

$$K_{amp} \cdot K_{sol} \cdot v_i - A_s(P_a - P_b) = m_s\frac{d^2x}{dt^2} + b_s\frac{dx}{dt} + k_s x \quad (16)$$

$$A_p(P_1 - P_2) - F = m_p\frac{d^2y}{dt^2} + b_p\frac{dy}{dt} + k_p y \quad (17)$$

where K_{amp} is the gain of amplifier for the proportional solenoid[A/V], K_{sol} is the gain of proportional solenoid[N/A], v_i is the voltage applied to the amplifier as the reference input signal[V], m_s is the mass of spool[kg], m_p is the mass of piston[kg], b_s is the viscous damping coefficient of spool[N · s/m], b_p is the viscous damping coefficient of piston[N·s/m], k_s is the stiffness of spool retaining spring[N/m] and k_p is the stiffness of cylinder spring[N/m]. F is the steering load[N] and the value of steering load is known to be 350N~1500N. As the steering load is smaller than the spring load of cylinder, the steering load is not considered in this study.

4. Experiment and Simulation

4.1 The experiment

Figure 4 shows a test bench for the experiments. The test bench consists of a power unit, a rear wheel steering gear composed of a proportional pressure valve and a cylinder, variety of sensors and amplifiers for measuring the physical quantities and a personal computer equipped with an interface card for recording data and sending the reference input signal.

The instruments and equipments of the test bench are described in Table 1. The pump used in the power unit is a vane pump of which the displacement is 10cc/rev and the nominal delivery pressure is 14MPa. The delivery flow rate of the pump can be regulated by controlling the revolution rate of drive motor by means of the inverter.

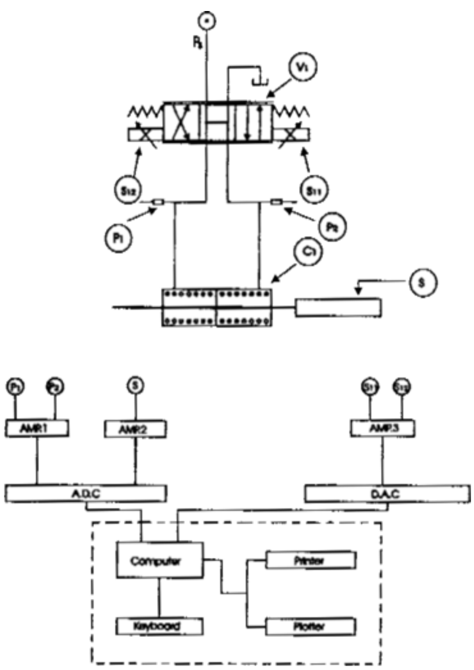

Fig. 4 Schematic of the test bench for experiments

The PWM(pulse width modulation) amplifier is used for the proportional valve control and its dither frequency is in the range of 0~200Hz. The interface card has an analog-digital converter and a digital-analog converter with the 12bit resolution.

In order to ascertain the linearity and the hysteresis characteristic of the valve, the pressure at each load line was measured while the reference input signal was increased then decreased step by step. The transient response characteristic of the valve was inspected by putting the step signal as the reference input. The maximum supply pressure of 7MPa, delivery flow rate of 7*l*/min and the oil temperature of 40℃ were maintained while the experiments were carried out.

4.2 The simulation

A numerical simalation based on the equations described in Sec. 3 is carried out to analyze the characteristics of the valve. Runge-Kutta method with the increment time of 1.2ms was utilized as a computation algorithm for the programme.

The discharge coefficients of the throttles in the valve and the oil passages were obtained from the

Table 1 Instruments and equipments of the test bench.

Symbol	Instruments	Manufacturer
C_1	4WS cylinder	Kayaba
V_1	P. C. V	
S_{11}, S_{12}	Solenoid	Kayaba
P_1, P_2	Pressure Transducer	NEC Sanei
S	Leanear displacement transformer	Temposonics
A_1	Strain amplifier	Cheon-he
A_2	Amplifier for L. D. T	Temposonics
A_3	Amplifier for P. C. V	Uchida

preliminary experiments and the results from the experiments were curve fitted as the functions of Reynold's number by the least square method. Reynold's number was defined as follows.

$$Re = \frac{v \cdot 4D_h}{\nu} \tag{18}$$

Table 2 Discharge coefficients of the valve and throttles.

	Discharge coefficient
Main valve	$1.3\times10^{-7}\times Re^3-4.714\times10^{-5}\times Re^2+6.28959\times10^{-3}\times Re-1.74619\times10^{-2}$
Orifice $\phi0.2$	$1.6\times10^{-7}\times Re^2+1.426\times10^{-3}\times Re-1.425\times10^{-1}$
Orifice $\phi0.4$	$3\times10^{-8}\times Re^3-1.704\times10^{-5}\times Re^2+4.42529\times10^{-3}\times Re+1.80583\times10^{-2}$
Orifice $\phi0.6$	$1\times10^{-10}\times Re^3-6.3\times10^{-7}\times Re^2+8.3044\times10^{-4}\times Re-1.16944\times10^{-3}$
Orifice $\phi0.8$	$1\times10^{-8}\times Re^2+3.891\times10^{-4}\times Re-1.74619\times10^{-2}$
Orifice $\phi1.0$	$2.19\times10^{-6}\times Re^2+2.24\times10^{-4}\times Re+8.38\times10^{-2}$

Table 3 Physical parameters of the system.

Parameters	Value	Unit
Density of oil	869	kg/m^3
Bulk modulus of elasticity of oil	1.8×10^9	Pa
Kinetic viscosity of oil	3.2×10^{-5}	m^2/s
Area of piston	1.65×10^{-3}	m^2
Mass of piston	6.53×10^{-1}	kg
Mass of spool	7×10^{-2}	kg
Constant of cylinder spring	6.45×10^6	N/m
Constant of valve spring	4.4×10^4	N/m
Volume of cylinder and pipe	1.496×10^{-4}	m^3

where v is the average velocity at the throttle[m/s], D_h is the hydraulic diameter[m] ($D_h=A/l$, A is the sectional area[m^2] and l is the peripheral length of the opening[m]) and ν is the kinematic viscosity of oil[m^2/s]. Table 2 shows the discharge coefficients of throttles.

The other physical parameters of the system were obtained from the catalogues and/or by the measurements. The value of constants were indicated in Table 3.

5. Results and Discussions

5.1 The results from the experiments

The parameters of the valve used for the experiment were as follows. The diameters of the orifices in the inlet and outlet oil passages connected to the feedback pressure chambers were 0.4 mm and 1.0 mm, respectively. The underlap of spool was 0.27 mm and the stiffness of the spool retaining spring was 44 N/mm. The dither frequency of the proportional amplifier was set at 75 Hz.

A typical result from the experiment on the steady state characteristic of the valve was shown in Fig. 5. It shows that there is a small dead zone which seems to be caused by the spool retaining spring but the relation between the input voltage and the line pressure is fairly linear and the hysteresis which might be owing to the characteristic of the proportional solenoid is relatively small (Backe, 1983). The linearity and the hysteresis characteristic are as good as those of the

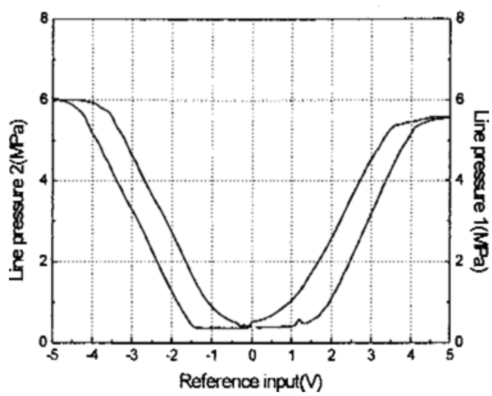

Fig. 5 Steady state characteristic of the valve with the new mechanism (experimental result, dither frequency 75Hz)

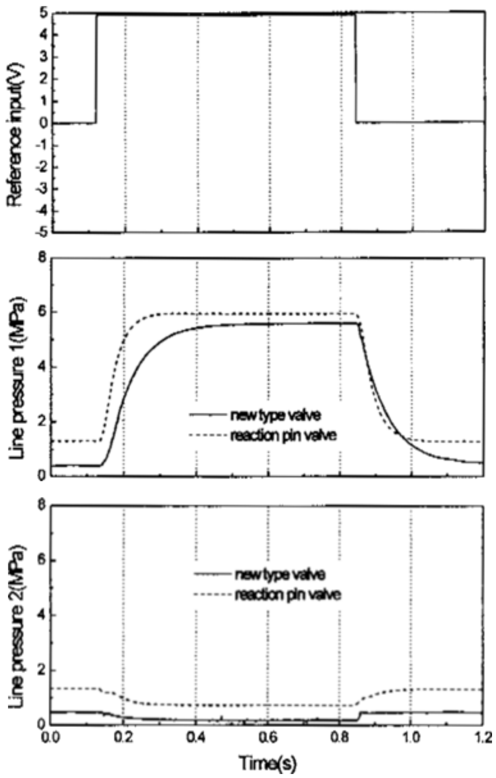

Fig. 6 Dynamic characteristic of the valve with the new mechanism (experimental result, dither frequency 75Hz, ref. input positive).

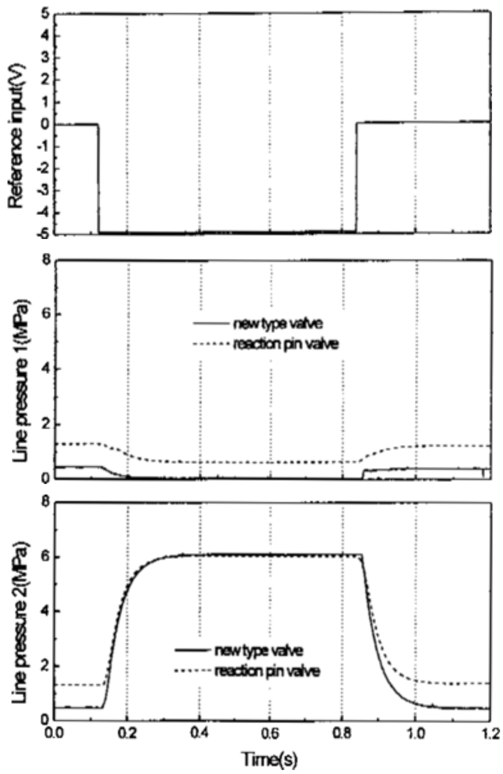

Fig. 7 Dynamic characteristic of the valve with the new mechanism (experimental result, dither frequency 75Hz, ref. input negative).

conventional valve(Oh, Jang and Lee, 1996). There appears the difference of maximum pressures between two load lines. It seems to be caused by the spool and sleeve of the valve not being geometrically symmetric and the stiffness of the spool retaining springs not being exactly the same.

The results from the experiments performed while the parameters of the valve are varing show that the range of dead zone is influenced by the stiffness of the spool retaining spring. The larger the ratio of the diameters of the throttles at the inlet and outlet oil passages connected to the pressure chamber, the wider the range of control pressure. Range of the dead zone becomes narrower and the linearity is improved as the underlap is getting smaller. However, the hysteresis dose not seem to be influenced by the underlap size.

Figures 6 and 7 show the typical results from the experiment on the characteristic of the step response of the valve. Figures 6 and 7 show the result when the reference input was positive(+) and negative(−), respectively. The solid line indicates the response curve of the newly proposed valve and the dashed line indicates the response curve of the conventional reaction pin type valve. The rise times are about 165ms in Fig. 6 and 103ms in Fig. 7. The difference on response characteristics between Figs. 6 and 7 seems to be caused by the fact that the spool and sleeve of the valve were not exactly symmetric and the size of throttles in the left side and right side oil passages in the valve are not identical. The response characteristic is fairly good compared with that of the already developed one, which has the reaction pin mechanism as the load pressure feedback method. The response become faster as the stiffness of the spool retaining spring is getting smaller. The size of underlap dose not affect the response characteristic of the newly proposed valve while it inflnence that of the valve having

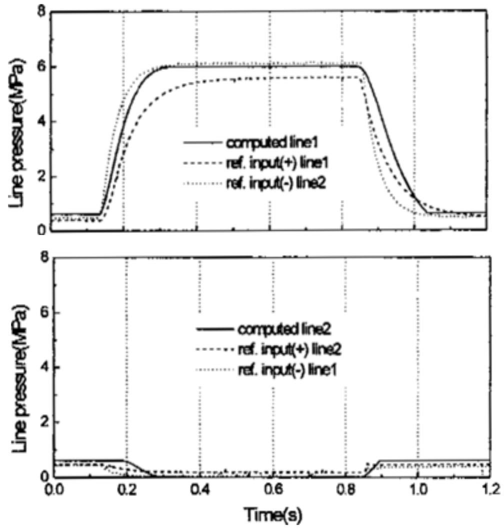

Fig. 8 Dynamic characteristic of the valve with the new mechanims (computed result).

reaction pin mechanism (Oh, Jang and Lee, 1996). This is one of the advantage of the newly proposed valve in this study over the conventional one having reaction pin mechanism as the pressure feedback method.

5.2 The results from the simulations

In order to analyze the theoretical performance of the valve and to confirm the adequacy of the theoretic analysis, the step responses were computed by the programme mentioned in Sec. 4. Figure 8 shows a typical result from the computation. The parameters of the valve were the same as those in the experiments. The figure indicates that the rise time is about 143ms and there exists the discrepancy in rise time between the results from the experiments and the computation. It also shows a little difference in the pressure rise tendency compared with Figs. 6 and 7. It seems that the discrepancy of the rise time and the pressure rise tendency is caused by the limitation in simulating the complex states of flows in the valve. The simulation shows that the stronger the stiffness of the valve retaining spring, the slower the response. The influence of the size of the spool underlap is the same as that in the experiments.

Figure 9 shows the comparison between the experimental and the computational results on the influences of the valve parameters shown in Table

Table 4 Parameters of valves in Fig. 9.

Valve	Parameters		
	Underlap (mm)	Throttles (ϕ-ϕ)	Spring constant (N/m)
1	0.27	0.4-1.0	4.4×10^4
2	0.22	0.4-1.0	4.4×10^4
3	0.17	0.4-1.0	4.4×10^4
4	0.27	0.4-1.0	2.8×10^4
5	0.22	0.4-0.8	4.4×10^4
6	0.22	0.2-0.6	4.4×10^4

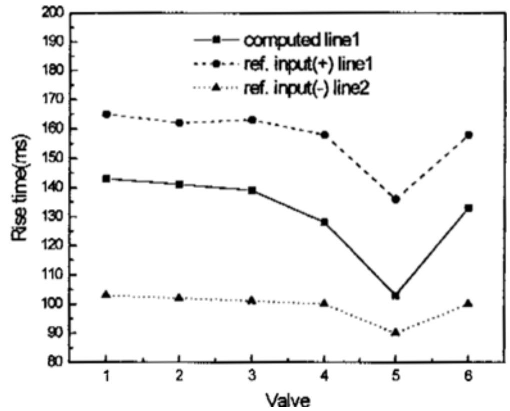

Fig. 9 Comparison of experimental results to simulated results.

4. It might be said that there is a bit of discrepancy in rise time but the differences are almost constant through the cases and the tendency of the rise time in the computational results fairly coincides with that in the experimental results. Therefore, it is possible to investigate the influences of the valve parameters to the response characteristic by using the simulation programme.

6. Conclusion

The result obtained in this study can be briefed as follows.

(1) A new type of pressure feedback mechanism suitable for the proportional pressure control valve for 4WS system on a passenger car was proposed.

(2) It was confirmed through the experiments

that the linearity and the hysteresis characteristic of the proportional pressure control valve with the newly proposed pressure feedback mechanism were fairly good and the control range of the load pressure can easily be changed by varying the combination of the throttles in the inlet and outlet oil passages connected to the pressure chamber.

(3) The response characteristic of the valve is relatively good and the response is not influenced by the size of the spool underlap. Therefore, it might be possible to reduce the power loss without deterioration of the valve performance when the spool is at a neutral position.

(4) A numerical simulation programme for the analysis of the valve characteristic was developed and the coincidence of the simulated results with the experimental results was confirmed.

The valve having the pressure feedback mechanism proposed in this study can be used for the electro-hydraulic rear wheel steering gear on a passenger car. The influences of the valve parameters to the response characteristic can be investigated by the simulation in the course of the valve design.

Acknowledgement

This study was supported by the Korea Science and Engineering Foundation (No. : 961-1001-002-1).

References

Backe, W., 1983, *Hydraulische und Pneumatische steuerung und Antriebe* (*Grundlage und Moderne Technologie*), Division of Mechanical Engineering of Korea Advanced Institute of Science & Technology, pp. 2-5～2-6.

Eguchi, T., Sakita, Y., Siraishi, K., Mori, K., Kawagoe, K., Ishikawa, Y. and Kaneko, S., 1987, "The Development of 'Super HICAS', a New Four Wheel Steering System with Phase-reversal Control," *NISSAN Technical Report*, NO. 25, pp. 19～29.

Hosokawa, K., Furutani, S., Ohmura, H. and Note, K., 1988, "Development of MAZDA Vehicle-Speed-Sensing Four-Wheel-Steering System," *MAZDA Technical Report*, No. 6, pp. 102～109.

Kanazawa, H. and Edahiro, T., 1990, "Trends of Four Wheel Steering System and Hydraulic Technic," *Journal of the Japan Hydraulics & Pneumatics Society*, Vol. 21, No. 2, pp. 127～133.

Komatsu, K. and Akaiwa, M., 1988, "Pressure Control Valve," *Japan Patent Bulletin*, Showa 63-236110, pp. 72～73.

Korea UCD Co., 1991, *Total Program* (Catlog), Korea UCD Co., Seoul, pp. 553～566.

Oh, I., Jang, J. and Lee, I., 1996, "Characteristics of the Proportional Pressure Control Valve for 4 Wheel Steering System on the Passenger Car," *Transactions of Korea Society of automotive Engineers*, Vol. 4, No. 4, pp. 87～96.

Parker Fluidpower, 1989, *HPI Proportional Valves System* (catalog HPC2400), HPI Operation Hydraulic Valve Division, Sturtevant, pp. 2～21.

Takenaka, T. and Urata, E., 1970, *Oil Hydraulics*, Yogendo, Tokyo, pp. 176～204.