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DEVELOPMENT OF A WATER HYDRAULIC PRESSURE-COMPENSATED FLOW CONTROL VALVE

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Abstract

This paper submits a new design of a pressure-compensated flow control valve for water hydraulics. The operating pressure difference range and flowrate range of the developed valve are 1 - 14 MPa and 2.7 - 17 l/min, respectively. A pressure-compensator valve and a metering valve are connected in series, placing the former at the upstream side. The major features of the designed valve are that the pressure-compensator valve has two throttles to prevent cavitation, a ring for flow force compensation, and a viscous damper to stabilise the motion of the valve. Valve dimensions were determined based on dynamic and static analysis. An experimental study was carried out for a produced valve. No cavitation noise was observed for operating pressures up to 14 MPa. Mounting the flow force-compensating ring reduced flowrate variation from 8 % to 4 % of reference flowrate, while it increased hysteresis from 0.5 % to 2.5 % of reference flowrate.

Keywords: water hydraulics, flow control valve, pressure compensation, double throttle, flow force compensation, cavitation, hysteresis

1 Introduction

Flow control valves include simple throttle valves such as needle valves and gate valves. Advanced, and commonly used, flow control valves have a pressure-compensation and/or temperature-compensation function. A pressure-compensated flow control valve is composed of a metering valve and a pressure compensator valve (hereafter a PC valve). The metering valve determines flowrate; the PC valve works to keep the pressure difference across the metering valve constant. This paper considers a pressure-compensated flow control valve to be used in a tap water hydraulic system. The change of viscosity of water due to temperature change is more moderate than other hydraulic fluids. Therefore, temperature-compensation is not dealt with in this paper.

There are two-way flow control valves and three-way flow control valves; to keep load flowrate constant, the former adjust pressure loss while the latter deliver excess flowrate (Cundiff, 2001, Pandharikar et al., 2002). The valve designed in this paper belongs to two-way flow control valves.

In a two-way flow control valve, a metering valve and a PC valve are connected in series; one type of flow control valve places a PC valve on the upstream side (Fig. 1(a)) and the other type of flow control valve places PC valve on the downstream side (Fig. 1(b)). The function of both types is the same while the leakage characteristic is different. For water hydraulics, type (a) is more suitable because type (b) has a leakage path that directly connects upstream and downstream of the flow control valve (Trostmann, 1996; Wu et al., 2007; Suzuki and Urata, 2007). Therefore, this paper studies the type (a) valve.

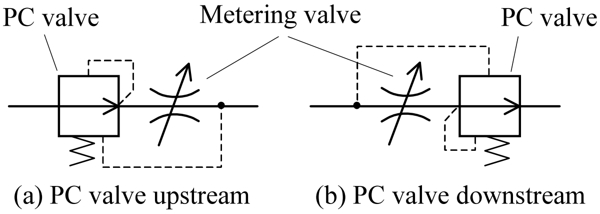

Fig. 1: *Two types of pressure-compensated flow-control valve*

When pressure difference across the flow control valve increases, the pressure drop across the PC valve increases because the pressure drop across the metering orifice is almost constant. The increased pressure difference increases the flow velocity through the throttles

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on the PC valve. The higher the flow velocity, the greater flow force closing the valve. The motion of the valve in the closing direction results in a decrease of flowrate. Consequently, an increase of pressure difference across a pressure-compensated flow control valve results in a slight decrease of flowrate (Andersson, 1984; Handroos and Vilenius, 1991; Trostmann, 1996; Wu et al., 2007). The rate of flowrate decrease accompanying the increased pressure difference becomes more distinct at higher flowrates.

Pressure drop across the flow-metering valve is normally small; most of the inlet-outlet pressure difference is consumed at the PC valve. While a single throttle for a high-pressure difference easily induces cavitation, conventional PC valves have a single throttle. In a pressure-compensated flow control valve for water hydraulics, therefore, a counter-measure against cavitation is desirable (Pandharikar, 2002). The valve also requires countermeasures against problems originating from the low viscosity of water: friction and wear of the sliding parts, high leakage, and low damping.

The basic structure of the valve designed in this paper was planned to respond to the above-listed problems in water hydraulics. The dimensions of the valve are determined based on numerical simulations of the static and dynamic characteristics. Experiments for the static characteristic at a pressure of up to 14 MPa were conducted and the flow control valve performance, namely, pressure-flow characteristics, flowrate variation and hysteresis, was investigated.

2 Structure of the Designed Valve

Figure 2 shows a cross-section of the developed valve. The mounting surface dimensions follow ISO code: 6263-03-03-0-97. The valve body and most of the inner parts are made of copper alloy and stainless steel, respectively. Water enters through the inlet port, passes through the PC valve and the flow control needle throttle (= metering orifice), and then exits through the outlet port. Adjusting the displacement of the metering orifice enables determination of the reference flowrate.

Figure 2 shows the no-flow state in that the bias-spring forces the PC valve to its left end position. Downstream of the metering orifice, which is the outlet of the valve, is hydraulically connected to the right end of the PC valve. Similarly, the upstream of the metering orifice, which is the valve chamber of the PC valve, is hydraulically connected to the left end of the PC valve via the central hole of the PC valve.

The working principle of the PC valve is the same as in conventional pressure-compensated flow control valves. The explanation in the next paragraph is only to remind readers.

In the flow condition, the pressure upstream of the metering orifice rises and the resultant force, which is equal to the product of the resultant pressure difference across the metering orifice and the cross-sectional area of the PC valve, pushes the PC valve to the right. The PC valve then moves to a position where the force exerted by the fluid pressures and the bias-spring achieve equilibrium. Thus, the pressure difference across the metering orifice is determined by pre-compression of the bias-spring. Any change in pressure difference across the metering orifice moves the PC valve to a new position so as to return the pressure difference across the metering orifice to its value before the change.

Definitions of the functions and dimensions of the valve are given in Fig. 3. The reference dimensions of the developed valve are shown in Table 1. These dimensions were determined based on the design analysis described in Section 3.

Table 1: Reference dimensions of the developed valve

PC valve					
D_1	10.01 mm	d_1	9.985 mm	h_{01}	1 mm
D_2	10.01 mm	d_2	9.985 mm	h_{02}	1 mm
d_t	13.95 mm	l_t	8 mm	δ_t	30 μm
d_p	19.95 mm	l_p	12 mm	δ_p	30 μm
k	17.4 N/mm	x_0	4.7 mm	D_0	10.01 mm
V_m	1 cm^3	V_{p0}	0.4 cm^3	m	41 g
Flow metering valve					
d_c	5.5 mm	θ_c	20 deg	V_{c0}	4 cm^3

To prevent cavitation, multi-stage pressure reduction is effective (Berger, 1983; Liu et al., 2002; Suzuki and Urata, 2005; Nie et al., 2006). The design in this paper adopts the two serial throttles in the PC valve, as in the former designs of the present author (Suzuki and Urata, 2007). The dimensions of the two throttles are made equal, i.e. $d_1 = d_2$, $D_1 = D_2$ and $h_{01} = h_{02}$, so that the pressure drops across each throttle are equal.

The intermediate pressure in the chamber between the two throttles is determined by the ratio of each opening of the throttles. Therefore, when the opening is small, the influence of machining error on the intermediate pressure becomes relatively large. To accurately determine the axial distance between the two throttles, the valve sleeve is made up of three pieces. The axial length of the central piece of the sleeve, which provides the stationary edge of the first throttle, is made equal to the distance between the two metering edges of the PC valve.

A steady state flow force pushes the PC valve in the closing direction. To compensate this force, a ring is fitted into the annular clearance between the sleeve of diameter D_0 that is slightly larger than the first throttle diameter d_1. Pressure p_u being exerted on the annular clearance between D_0 and d_1 generates a force to open the PC valve.

Coulomb friction acting on the force-compensating ring might cause hysteresis of the valve characteristics. In this study, two kinds of seal ring were tried: a commercially obtained O-ring with PTFE slipper ring, a self-made ultra-high-molecular-weight polyethylene (UHMW-PE) ring. While UHMW-PE has low sliding resistance with low rate of water absorption, its dimensional change with temperature is rather large. Therefore, the width of the ring was made a little smaller than the width of the fitting groove on the PC valve.

While the low viscosity of water cannot give the PC valve sufficient damping, the PC valve has a negative damping length, which may cause vibration of the valve. To obtain sufficient damping force, a damping mechanism is installed, which is composed of a damping chamber and an annular clearance. A wear-ring, which protects the valve from wear and prevents off-centring of the valve, is mounted on the right-side land of PC valve.

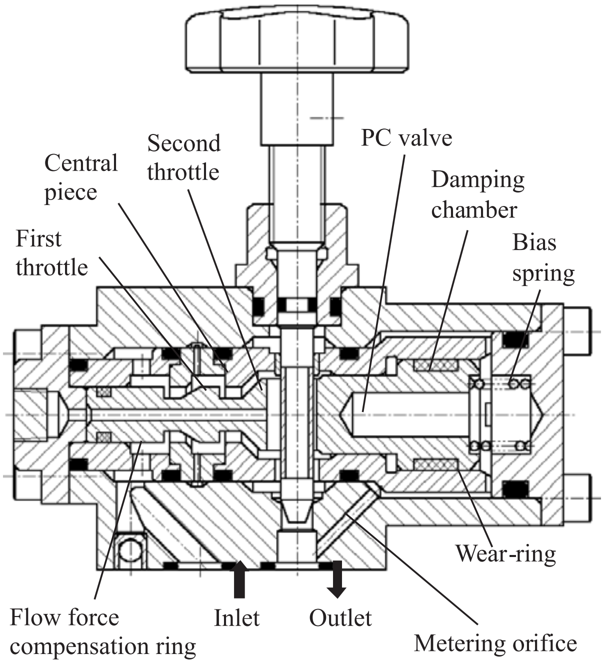


Fig. 2: *Cross-section of the developed valve*

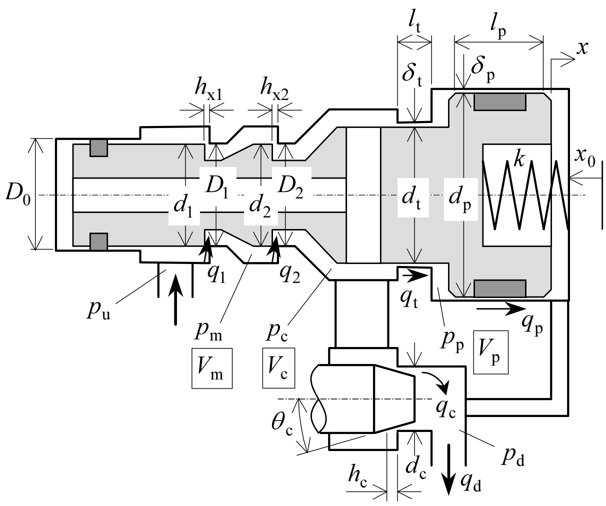


Fig. 3: *Valve functions and dimensions*

A major feature of the designed valve is the crossing of the PC valve and the metering valve by a lateral hole on the PC valve (Fig. 4). This design greatly simplified manufacturing of the valve body compared with that of conventional pressure-compensated flow control valves. The valve spindle of the metering orifice passes through an ellipse-like hole made on the PC valve. A Nylon tube covers the valve spindle to protect it from metallic contact with other parts.

Fig. 4: *The PC valve and the metering valve spindle*

3 Analysis for Design

To determine the dimensions of the valve, a design analysis was carried out. The following assumptions were made to simplify the basic equations:

- Flow through a narrow annular clearance is regarded as laminar flow between parallel planes.
- The viscous drag force acting on the valve due to leakage flow is negligibly small compared to forces exerted by bias-spring and hydraulic pressures.
- Cavitation does not occur at the valve throttles.
- Jet angles of flow through the PC valve throttles are constant (69 degrees).
- Elastic deformation of valve parts is small and can be ignored.

3.1 Flow through Restrictors

This sub-section is concerned with basic equations. The flowrates through the restrictors are as follows.

$$q_1 = c_{d1} d_1 \pi \sqrt{2(h_{x1}^2+\delta_1^2)(p_u-p_m)/\rho} \quad \text{for } h_{x1} \ge 0 \tag{1}$$

$$q_2 = c_{d2} d_2 \pi \sqrt{2(h_{x2}^2+\delta_2^2)(p_m-p_c)/\rho} \quad \text{for } h_{x2} \ge 0 \tag{2}$$

$$q_c = c_{dc} \pi h_c \sin\theta_c \left(d_c - \frac{h_c}{2}\sin 2\theta_c\right)\sqrt{2(p_c-p_d)/\rho} \tag{3}$$

$$q_t = \frac{\pi d_t \delta_t^3}{12\mu l_t}(p_c-p_p) \tag{4}$$

$$q_p = \frac{\pi d_p \delta_p^3}{12\mu l_p}(p_p-p_d) \tag{5}$$

where c_{d1}, c_{d2} and c_{dc} are the discharge coefficients of the first and second throttles of the PC valve and the flow control orifice, respectively.

The axial- and radial clearances of the first and second throttles of the main valve are

$$\left.\begin{aligned} h_{x1} &= h_{01} - x, \quad \delta_1 = (D_1 - d_1)/2 \\ h_{x2} &= h_{02} - x, \quad \delta_2 = (D_2 - d_2)/2 \end{aligned}\right\} \tag{6}$$

respectively, because the valve is assumed to be a rigid body.

3.2 Dynamic Equations

To determine the dimensions of the damping mechanism, dynamic analysis is necessary even though the dynamic characteristics of the valve are not the research object of this study. This sub-section derives the equation of motion of the valve.

The equation of motion of the PC valve is

$$\begin{aligned}(m+m_{\mathrm{f}})\frac{d^2x}{dt^2}+\mathrm{sgn}\left(\frac{dx}{dt}\right)F_{\mathrm{c}}+k(x+x_0)\\ = A_{\mathrm{c}}p_{\mathrm{c}}-A_{\mathrm{u}}p_{\mathrm{u}}+A_{\mathrm{m}}p_{\mathrm{m}}+A_{\mathrm{p}}p_{\mathrm{p}}-A_{\mathrm{d}}p_{\mathrm{d}}+F_{\mathrm{s}}+F_{\mathrm{d}}\end{aligned} \tag{7}$$

where

$$\left.\begin{aligned}&A_{\mathrm{c}}=\tfrac{\pi}{4}\left(D_0^2+d_{\mathrm{t}}^2-d_2^2\right),\ A_{\mathrm{u}}=\tfrac{\pi}{4}\left(D_0^2-d_1^2\right),\\ &A_{\mathrm{m}}=\tfrac{\pi}{4}\left(d_2^2-d_1^2\right),\ A_{\mathrm{p}}=\tfrac{\pi}{4}\left(d_{\mathrm{p}}^2-d_{\mathrm{t}}^2\right),\ A_{\mathrm{d}}=\tfrac{\pi}{4}d_{\mathrm{p}}^2\end{aligned}\right\} \tag{8}$$

$$\begin{aligned}F_{\mathrm{s}}=2\pi\cos\theta_{\mathrm{f}}\Big\{&c_{\mathrm{d1}}d_1\sqrt{h_{\mathrm{x1}}^2+\delta_1^2}\,(p_{\mathrm{u}}-p_{\mathrm{m}})+\\ &c_{\mathrm{d2}}d_2\sqrt{h_{\mathrm{x2}}^2+\delta_2^2}\,(p_{\mathrm{m}}-p_{\mathrm{d}})\Big\}\end{aligned} \tag{9}$$

$$\begin{aligned}F_{\mathrm{d}}=\frac{dx}{dt}\pi\sqrt{2\rho}\Bigg\{&c_{\mathrm{d1}}d_1 l_1 h_{\mathrm{x1}}\sqrt{\frac{p_{\mathrm{u}}-p_{\mathrm{m}}}{h_{\mathrm{x1}}^2+\delta_1^2}}+\\ &c_{\mathrm{d2}}d_2 l_2 h_{\mathrm{x2}}\sqrt{\frac{p_{\mathrm{m}}-p_{\mathrm{c}}}{h_{\mathrm{x2}}^2+\delta_2^2}}\Bigg\}\end{aligned} \tag{10}$$

Because $d_1 = d_2$ in this design, $A_m = 0$. Then, the term of p_m in Eq. 7 vanishes, although p_m implicitly influences the flow force. Moreover, Eq. 9 indicates that p_m has no influence on the steady-state flow force F_s if the two throttles on the PC valve have equal dimensions and discharge coefficients. Therefore, an unexpected change of the intermediate pressure due to some dimensional inaccuracy of valve parts will have little influence on the pressure flow characteristics of the flow control valve.

Finally, the equations of continuity are as follows.

$$\frac{dp_{\mathrm{m}}}{dt}=\frac{\beta}{V_{\mathrm{m}}}(q_1-q_2) \tag{11}$$

$$\frac{dp_{\mathrm{c}}}{dt}=\frac{\beta}{V_{\mathrm{c0}}+A_{\mathrm{c}}x}\left(q_2-q_{\mathrm{c}}-q_{\mathrm{t}}-A_{\mathrm{c}}\frac{dx}{dt}\right) \tag{12}$$

$$\frac{dp_{\mathrm{p}}}{dt}=\frac{\beta}{V_{\mathrm{p0}}+A_{\mathrm{p}}x}\left(q_{\mathrm{t}}-q_{\mathrm{p}}-A_{\mathrm{p}}\frac{dx}{dt}\right) \tag{13}$$

$$q_{\mathrm{d}}=q_{\mathrm{c}}+q_{\mathrm{p}}+A_{\mathrm{p}}\frac{dx}{dt} \tag{14}$$

where V_{c0} and V_{p0} are the initial values of V_c and V_p at $x = 0$, respectively.

3.3 Simulation

The pressure-flow characteristic, namely, the relationship between the inlet-outlet pressure difference and the flowrate through the valve (= discharge flowrate), is obtained as the numerical solution of Eq. 1 to 14, eliminating the time-derivative terms. Figure 5 shows the calculation results of the flowrate, where the parameter is the opening of the metering orifice, h_c. Figure 5(a) shows the result without flow force compensation; Fig. 5(b) shows the result with flow force compensation. The case without flow force compensation was calculated as $D_0 = d_1$.

Figure 5(a) reveals that if any flow force compensation is not undertaken, the flow force decreases the flowrate with increase of the reference flowrate. In addition, the flowrate decreases with increase of the pressure difference across the flow control valve. Figure 5(b) shows that the flow force compensation reduces the rate of decrease of the flowrate. A trial calculation showed that a lager value of D_0 improves the pressure-flow characteristic in the higher flowrate range. However, it increased the flowrates by more than the reference values in the lower flowrate range. The dimensions in Table 1 were selected considering these results.

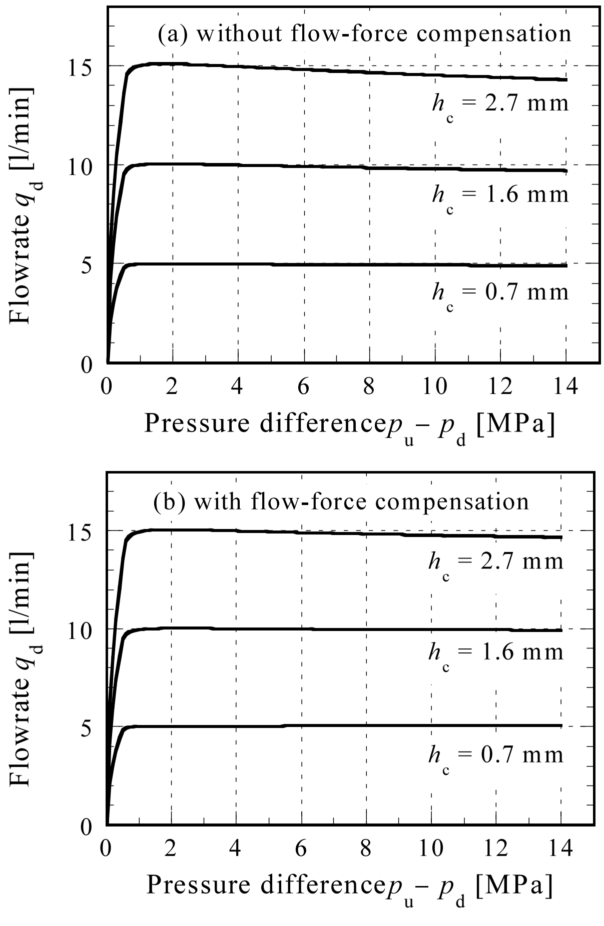


Fig. 5: *Simulation results of static characteristics*

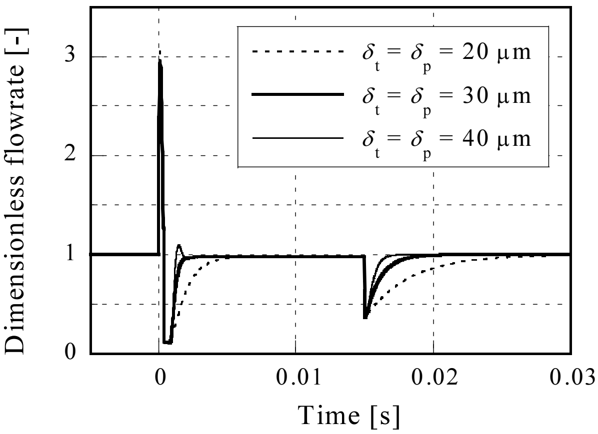
To determine the size of the damper, the response of the flowrate to a stepwise change of the upstream pressure was calculated. The MATLAB/Simulink® was used for the modelling of Eq. 1 to 14. The Runge-Kutta method of fourth-order was used as the solver. The time step for calculation was a fixed step of 1 μs.

The downstream pressure was set to zero and a rectangular wave of the upstream pressure was used as the input signal. The upstream pressure, which is 2 MPa initially, changes to 14 MPa at 0 ms, and then returns to 2 MPa again at 15 ms. Corresponding to the pressure change, the PC valve moves to the closing position in 0 to 15 ms, and then moves back to its initial position in 15 to 30 ms.

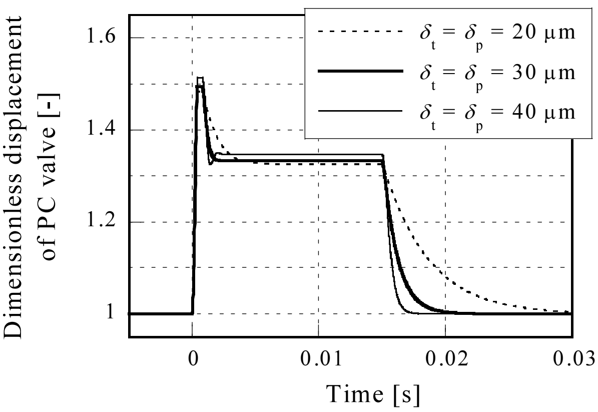
The response of the discharge flowrate was calculated for the initial value of 15 l/min, changing dimensions of the damper of the PC valve. To detect the influence of the damper clearance, the Coulomb friction was put to zero in this calculation.

Figure 6 (a) and (b) shows the responses of the discharge flowrate q_d and the PC valve displacement x, respectively, in which the clearance of the damper δ_t and δ_p varied ± 10 μm to the reference value, 30 μm.

A flowrate spike appears at the instant of the upstream pressure rise; a fast closing of the PC valve corresponding to the pressure rise generates undershoot of the flowrate. The flowrate then gradually returns to the reference value, although a small overshoot appears again for a larger clearance. When the upstream pressure drops (not shown in Fig. 6), the PC valve opens and a similar transient response in the reverse direction occurs. The flow through the damper clearance makes the response speed a little slower for pressure drops than for pressure rises.



(a) Flowrate



(b) Displacement of PC valve

Fig. 6: *Simulation results of dynamic response*

4 Experiment

4.1 Experimental Rig

Figure 7 illustrates the experimental rig. A three-throw piston pump with an accumulator was used as the water pressure source. The rated pressure is 21 MPa and the rated flowrate is 20 l/min. The inlet pressure of the test flow control valve was adjusted by controlling a bypass flow with a manually operated needle valve connected parallel to the test valve. Since there is no pressure loading at the outlet, the outlet pressure remained lower than 30 kPa. To keep the accuracy of the flow measurement within about 2 %, two flowmeters of different ranges were selectively used (see Fig. 7). The resolution of the pressure measurement system was about 10 kPa. The water temperature was kept at 25 - 30 °C by a cooler in the return-line to the reservoir.

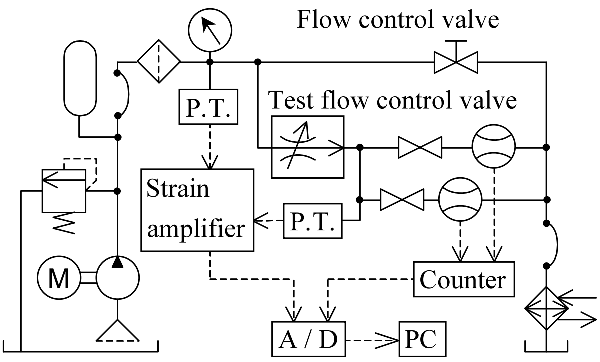


Fig. 7: *Experimental rig*

4.2 Experimental Results

Figure 8 shows the pressure-flow characteristic using a PC valve without a flow force compensation ring. The range of inlet pressure is 0-14 MPa, and the reference flowrates are 5, 10 and 15 l/min. The white symbols express data points for increasing of the inlet pressure; the black symbols express data points for decreasing of the inlet pressure. Bold curves are re-plots of the simulation shown in Fig. 5. The simulation agrees well with the experiment except a small deviation for the reference flowrate of 15 l/min. The maximum flowrate deviation from the reference flowrate is 8 % that occurred at 15 l/min. The hysteresis, namely, the difference of flowrates due to increasing and decreasing of the inlet pressure, is about 0.5 %. This suggests that the mounted wear-ring generates very low Coulomb friction.

Figure 9 shows the pressure-flow characteristic using a PC valve with an O-ring covered by a slipper ring made of PTFE. The measured flowrates for increasing inlet pressures agree well with those of the simulation. However, the results measured for decreasing pressure deviate from the simulation results. Larger deviations are observed for higher reference flowrates; these reach about 2.5 % for the reference flowrate of 15 l/min. The cause of the hysteresis is supposed to be the increased Coulomb friction between the PTFE slipper and the valve sleeve.

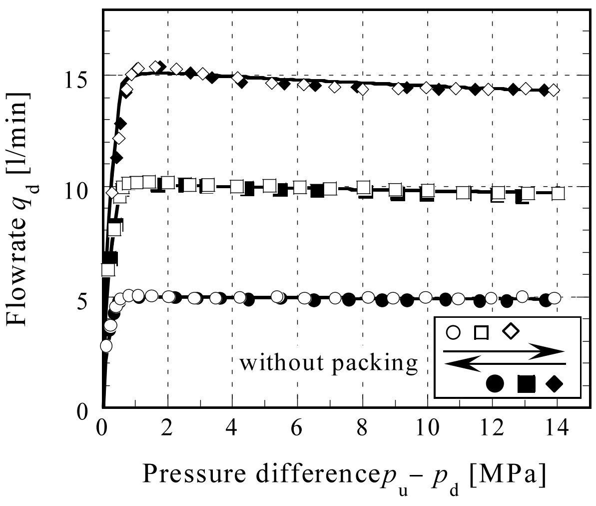


Fig. 8: *Pressure-flow characteristic without flow force compensation*

To reduce the frictional force, the O-ring with slipper was changed to a ring made of UHMW-PE. Figure 10 shows the measured pressure-flow characteristic with the UHMW-PE ring. The flowrate decreases with increase of inlet pressure up to 4 MPa. This tendency is similar to the pressure-flow characteristics shown in Fig. 8; this fact suggests that a small gap is remaining between the ring and the sleeve. The flowrate increases for the pressure difference $p_u - p_d$ greater than 8 MPa. Consequently, flow deviation from the reference value becomes similar to the case of the O-ring with slipper. The magnitude of the hysteresis is reduced to 1.5 % of the reference flowrate of 15 l/min, which is about a half of the hysteresis for the O-ring with slipper.

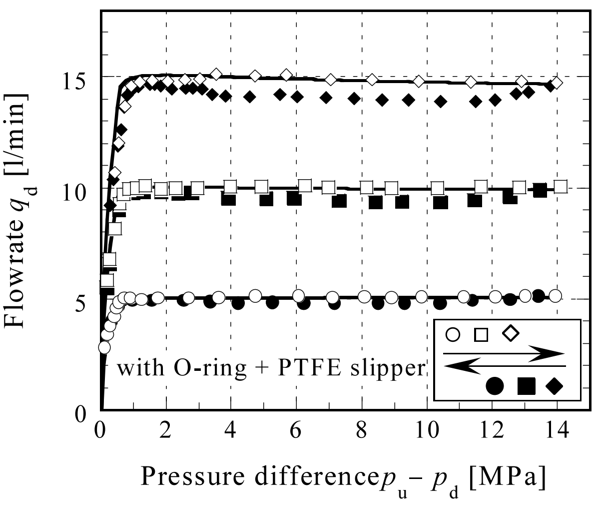


Fig. 9: *Pressure-flow characteristic with O-ring and PTFE slipper for flow force compensation*

The experimental result shown in Fig. 8 to 10 will be estimated by two items; the first is flowrate change from the reference by increasing inlet pressure, and the second is hysteresis.

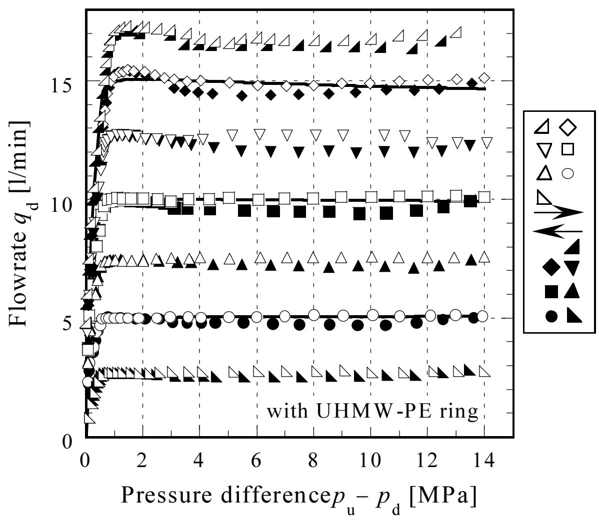


Fig. 10: *Pressure-flow characteristic with UHMW-PE ring for flow force compensation*

The index for the first item is calculated using measured flowrate for increasing inlet pressure (white marks in Figs. 8-10) while the PC valve is operating, and is defined by

$$\Delta Q_+ = \frac{\Delta q_+}{q_\mathrm{I}} \times 100 \tag{15}$$

where Δq_+ is the maximum difference between measured flowrates with the same reference flowrate and q_I is the average of the measured data. Figure 11 shows the estimated result. Without packing, the variation from the reference flowrate increases with the reference flowrate because there is no flow force compensation; it reaches about 8 % for a reference flowrate of 15 l/min. The flow force was compensated for by mounting the O-ring with slipper and by the UHMW-PE ring; the flow force compensation reduced the flowrate variation to 4 %.

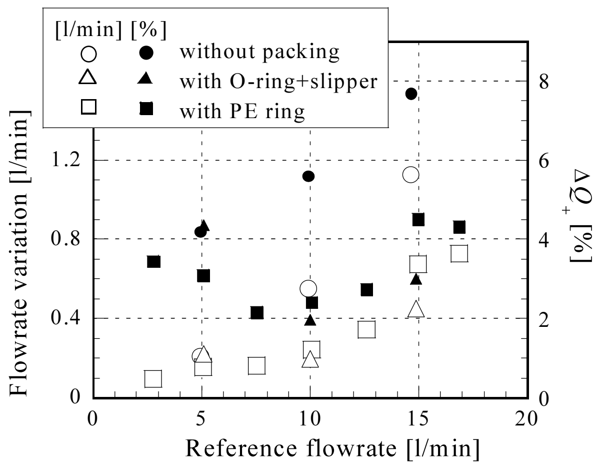


Fig. 11: *Experimental results of flowrate variation for increase of inlet pressure*

The second item, the hysteresis, is defined by

$$H = \frac{|q_\mathrm{I} - q_\mathrm{D}|}{q_\mathrm{I} + q_\mathrm{D}} \times 100 \tag{16}$$

where q_D indicates the flowrate measured for decreasing inlet pressure. Figure 12 shows the measured result.

The hysteresis in the experiment without packing was about 0.5 %. The O-ring with slipper increases the hysteresis to 2.5 %. The UHMW-PE-made ring increased the maximum hysteresis to 2.5 %. The hysteresis for the O-ring with slipper increases with increase of reference flowrates. In contrast, the hysteresis for the UHMW-PE-made ring decreases with increase of reference flowrates; it became 1.2 % for a reference flowrate of 15 l/min. Thus, the flow force compensation tried in this study was effective for increased pressure difference across the flow control valve. Its drawback is the increase of hysteresis caused by friction.

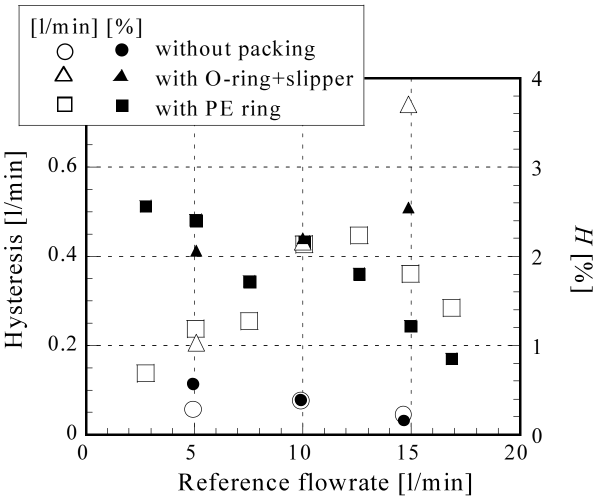


Fig. 12: *Experimental results of hysteresis*

5 Discussion

The experimental result in the previous section revealed that the developed valve does not radiate cavitation noise and has sufficient stability. Output flowrate is fairly constant over a wide range of pressure difference across the flow control valve. Therefore, the double throttles on the PC valve, the installed damper and the flow force compensation ring are regarded as effectively working.

The pressure-drop across the metering orifice is detected by the force balance of the PC valve and the resultant displacement of the PC valve exercise a feedback action to return the pressure drop to its value before the change. Therefore, distribution of the pressure drops among the PC valve throttles has very little influence on the pressure-flow characteristic of the flow control valve. However, the distribution of the pressure drops has an influence on the occurrence of cavitation. Flow and pressure in the valve have various influences on the operating performance of the valve, such as stability, hysteresis and dynamics. Therefore, we will discuss first the distribution of the pressure drops and then the hysteresis that is induced by the force compensation ring.

The PC valve has two throttles to establish two-step pressure drops. The effect of the throttles can be observed through examining the intermediate pressure. The ideal value of the intermediate pressure is just in the middle of the upstream and downstream pressures of the PC valve, namely, $(p_m - p_c)/(p_u - p_c) = 0.5$, regardless of the valve displacement.

Figure 13 shows experimental results for the dimensionless intermediate pressure for typical reference flowrates. The dimensionless intermediate pressure slightly increases with increase of the inlet-outlet pressure difference. It is almost 0.5 for a reference flowrate of 15 l/min, and becomes greater than 0.5 for lower values of the reference flowrate. A cause of this tendency is that the flow velocity becomes higher for a smaller reference flowrate. It is difficult to explain this phenomenon. A possible cause may be the influence of machining and construction error because the valve displacement is small. Another cause may be the pressure distribution in the valve chamber since it is not uniform and changes with flow velocity at the orifice and opening of the valve. When the reference flowrate is 2.7 l/min, the intermediate pressure drastically changes for an inlet-outlet pressure difference of 6 MPa-10 MPa. Above 10 MPa it takes a fixed value because the PC valve is stopped at its right strop end.

Figure 14 shows pressure drop across the metering orifice. If the pressure drop is constant, the flow through the orifice is also constant. The data indicate that the PC valve is at the left stop end when the inlet-outlet pressure difference is less than about 1 MPa. The curves, except for the reference flowrate of 2.7 l/min, show hysteresis similar to those shown in Fig. 12; the cause of the hysteresis is the friction of the flow force compensation ring. For the reference flowrate of 2.7 l/min, the PC valve reaches right stroke end and stops there when the inlet-outlet pressure difference is greater than 10 MPa. The hysteresis in this case is not due to friction because the PC valve is stopping; some hydrodynamic cause is supposed.

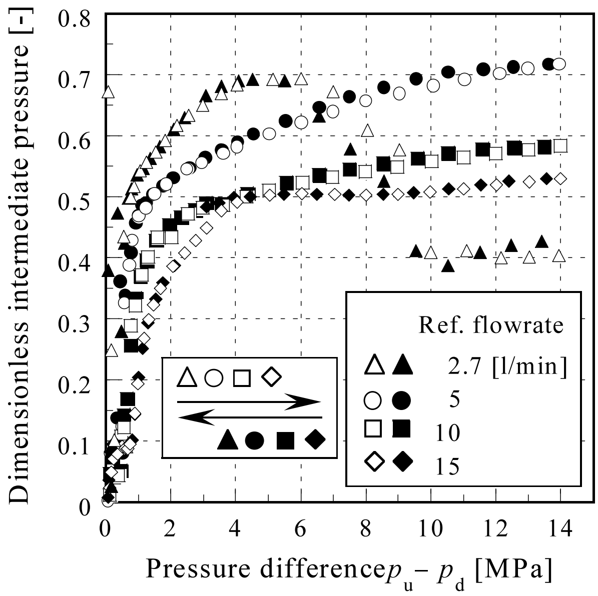


Fig. 13: *Dimensionless intermediate pressure in the space between the two throttles of the PC valve*

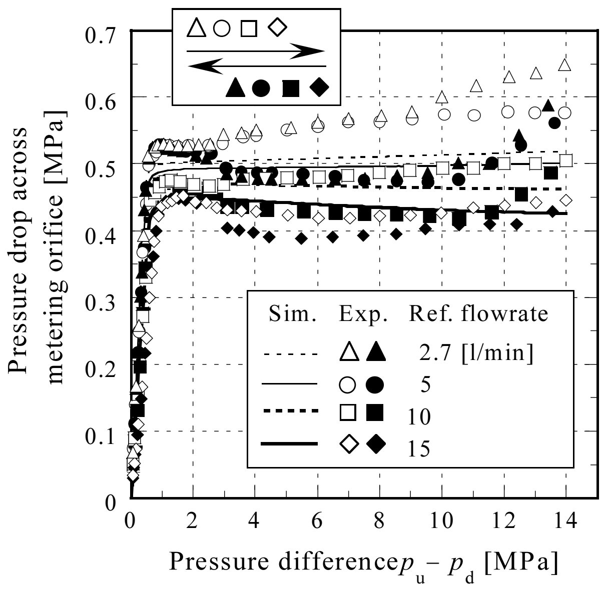


Fig. 14: *Pressure drop across the metering orifice*

Although hysteresis is not considered in the simulation, the pressure-flow relationship of the flow control valve obtained by simulation agrees well with the experimental result. This is a result of the feedback effect produced by the pressure compensation that keeps the pressure difference across the metering orifice constant for a reference flowrate.

6 Conclusions

A pressure-compensated flow control valve for water hydraulics was developed with the aim of: (1) using corrosion preventive materials, (2) using two throttles for the PC valve to prevent cavitation occurrence, (3) installing a viscous damper to compensate for low viscous damping of water, (4) placing the PC valve upstream of the metering orifice to reduce leakage due to low viscosity of water, and (5) mounting a flow force compensation ring to reduce the influence of flow force on the outlet flowrate.

Simulation was carried out to determine the dimensions of the valve and a manufactured valve was experimentally studied. The developed valve was operated for a flow range of 2.7 - 17 l/min with an inlet-outlet pressure difference of 1 - 14 MPa. The produced valve showed pressure-flow characteristics that agree well with those predicted by the simulation. Thus, the designed valve has characteristics suitable for use in water hydraulic systems.

The above-listed countermeasures for the physical properties of water were proved effective for each purpose. However, the intermediate pressure often showed large deviation from the ideal value; hysteresis was induced by the mounting of the flow force-compensating ring. A further study will be necessary to resolve these problems.

Nomenclature

c_{d*}	discharge coefficients of restrictors	[-]
d_*	diameters (see Fig. 3)	[m]
D_0	internal diameter of PC valve sleeve	[m]
D_i	internal diameter of the i-th throttle of PC valve sleeve	[m]
F_c	Coulomb friction acting on PC valve	[N]
F_d, F_s	dynamic and static flow forces acting on PC valve, respectively	[N]
H	dimensionless hysteresis (see Eq. 16)	[-]
h_{xi}	axial opening of the i-th throttle of PC valve	[m]
h_{0i}	initial axial clearance of the i-th throttle of PC valve	[m]
k	spring constant of PC valve	[N/m]
l_*	length (see Fig. 3)	[m]
l_i	damping length of the i-th throttle of PC valve	[m]
m	equivalent mass of PC valve (incl. 1/3 of spring mass)	[kg]
m_f	water mass moving with PC valve	[kg]
p_*	pressures (see Fig. 3)	[Pa]
p_m	intermediate pressure (see Fig. 3)	[Pa]
q_*	flowrate (see Fig. 3)	[m^3/s]
q_D	average flowrate observed by decrease of inlet pressure	[m^3/s]
q_I	average flowrate observed by increase of outlet pressure	[m^3/s]
V_*	volumes (see Fig. 3)	[m^3]
x	displacement of PC valve	[m]
x_0	initial compression of spring	[m]
β	bulk modulus of water	[Pa]
δ_*	radial heights of annular clearances (see Fig. 3)	[m]
ΔQ_+	dimensionless pressure variation (see Eq. 15)	[-]
μ	viscosity of water	[Pa·s]
ρ	density of water	[kg/m^3]
θ_c	half cone angle of control orifice	[-]
θ_f	jet angle of flow through PC valve throttle	[-]

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