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Design and performance characteristic analysis of servo valve-type water hydraulic poppet valve †

Sung-Hwan Park*

School of Mechanical Engineering, Pusan National University, San 30, Jangjeon-dong, Geumjeong-ku, Busan, 609-735, Korea

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Abstract

For water hydraulic system control, the flow or pressure control using high-speed solenoid valve controlled by PWM control method could be a good solution for prevention of internal leakage. However, since the PWM control of on-off valves cause extensive flow and pressure fluctuation, it is difficult to control the water hydraulic actuators precisely. In this study, the servo valve-type water hydraulic valve using proportional poppet as the main valve is designed and the performance characteristics of the servo valve-type water hydraulic valve are analyzed. Furthermore, it is demonstrated through experiments that a decline in control chamber pressure that follows the change of pilot flow is caused by the occurrence of cavitation around the proportional poppet, and that fundamental characteristics of the developed valve remain unaffected by the occurrence of cavitation.

Keywords: Water hydraulics; Hydraulics; Servo valve-type poppet valve; Proportional poppet; Flow control valve; PWM control; Cavitation

1. Introduction

Ancient historical accounts show that water was used for centuries to produce power by means of water-wheels. However, these early uses of fluid power required the movement of huge quantities of fluid because of the relatively low pressures provided by nature. Based on Pascal's law, the first technical use of water hydraulics which transmits power by means of compressed water began in 1795 by Joseph Bramah after acquisition of a patent for a water hydraulic press. After this the water was mainly used as a power transmission medium of fluid power systems. In the beginning of the 1900's, the use of water as a power transmission medium declined by powerful electric technology. At this time, oil hydraulics was introduced for the first time. Afterwards, oil hydraulics was used in a wide range of engineering fields in the aircraft, marine, mobile and manufacturing sectors because of its rapid response and high power densities.

Recently, however, there has been an upsurge of interest in novel water hydraulics, largely as a result of environmental and safety concerns. But this novel water hydraulics is different to that of 1800's. In the modern sense, water hydraulics aims to realize environmentally friendly power transmission system through the use of advanced technologies which are linked to the use of new materials such as improvements in the technology of manufacture, awareness of tribological issues and developments in electronics and microprocessors. The water hydraulic systems in the modern sense consist of specially designed pumps, actuators and several control valves, which can be obtained already in the market area [1]. In the fields that safety, sanitation and pollution problem are emphasized, the water hydraulic systems have already been essential equipment [2].

One of the most important components in the constitution of water hydraulic systems is a control valve. High performance water hydraulic flow control

† This paper was recommended for publication in revised form by Associate Editor Haecheon Choi
*Corresponding author. Tel.: +82 51 510 3053, Fax.: +82 51 582 9164
E-mail address: shpark01@pusan.ac.kr

valves such as servo valves and proportional valves for water hydraulic use have already been developed, although maintenance and price-related problems in their practical use still remain [3, 4]. Accordingly, the flow or pressure control using high-speed solenoid valve controlled by PWM control method could be a good solution for these problems.

The author has already developed a directly-driven water hydraulic high speed solenoid valve using a seat-type ball valve [5]. And by applying this high-speed solenoid valve as a pilot valve and using a poppet as the main valve, a novel water hydraulic high-speed solenoid valve with a two-stage mechanism was developed [6]. Both of developed valves can be used as flow control valves of water hydraulic systems. However, these valves only have an on-off function, so a control method such as PWM control has to be used for proportional control of the flow rate. However, since the PWM control of on-off valves causes extensive flow and pressure fluctuation, it is difficult to control the water hydraulic actuators precisely.

Therefore, it is essential to develop the servo valve-type proportional control water hydraulic valve using seat-type valve as the main valve because the slide-type flow control valves such as servo valve and proportional valve for water hydraulic use have a very complex structure in order to solve the problems concerned with leakage and tribologies of water.

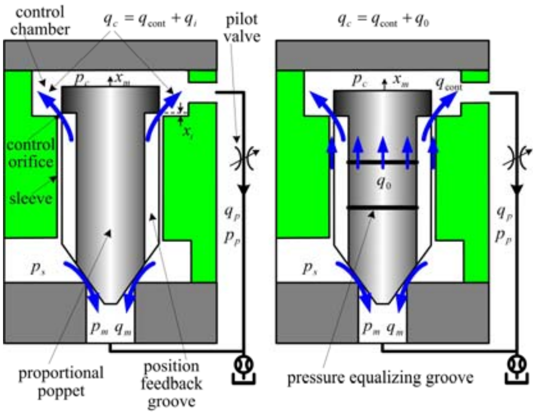
In 1984, Andersson first suggested the servo valve-type proportional poppet valve for oil hydraulic use [7]. In Japan, Wu reported the characteristics of the oil hydraulic poppet valve, which has a position feedback groove [8-10]. Afterward, Luomaranta reported the results on the control of an oil hydraulic single rod cylinder by applying the servo valve-type proportional poppet valve [11]. In addition, Sato also reported that an oil hydraulic motor can be controlled to have desirable dynamic characteristics with the use of an oil hydraulic servo valve-type proportional poppet valve with a balance spool [12]. However, there is no research on the servo valve-type water hydraulic poppet valve.

In this study, the servo valve-type water hydraulic valve using proportional poppet as the main valve is designed and the performance characteristics of the servo valve-type water hydraulic valve are analyzed. The proposed servo valve-type water hydraulic valve using proportional poppet has a simple structure with a control orifice which passes through the pilot orifice at both sides of the conventional poppet. In spite of its simple structure, in addition to the general advantages of poppet valves, which have no leakage, the displacement of a proportional poppet can be operated proportional to pilot flow rate. The experiment verified the validity of design method of the servo valve-type water hydraulic valve by showing the linear relation between the main flow rate and pilot flow rate. Furthermore, it is also demonstrated experimentally that a decline in control chamber pressure that follows the change of pilot flow is caused by the occurrence of cavitation around the proportional poppet, and that fundamental characteristics of the developed valve remain unaffected by the occurrence of cavitation.

2. Design of a servo valve-type water hydraulic valve using proportional poppet

Fig. 1(a) shows the fundamental structure and operating principle of the servo valve-type oil hydraulic proportional poppet valve suggested by Andersson [7]. The proportional poppet part is composed of three essential parts: the proportional poppet, the position feedback groove installed on both sides of the proportional poppet to form a control orifice, and the pilot valve series-connected to the control chamber. Here, the proportional poppet has simple structures, in which the grooves at both sides of the proportional poppet to feedback the position of the proportional poppet mechanically. By using this proportional poppet, the control orifice known as the variable orifice, is formed between the top of the groove and sleeve. By the role of this variable orifice, the servo valve-type proportional poppet valve produces a displacement proportional to the pilot flow rate. In addition, the output flow rate from the proportional poppet changes continuously in response to the change of the pilot flow rate. On the other hand, in order to guarantee the stable initial operation of the oil hydraulic proportional poppet, it is necessary to design under lap x_i at a control orifice as shown in Fig. 1(a).

In this study, the servo valve-type water hydraulic valve using proportional poppet is developed for water hydraulic systems. In the design process of water hydraulic servo valve-type proportional poppet valve, although it is assumed that corrosion-resistant materials are used, many problems caused by the characteristics of water need to be solved in order to realize the structure and operational principle of the proportional poppet-type valve as shown in Fig. 1(a).



(a) Hydraulic use (b) Water hydraulic use

Fig. 1. Comparison of structure of the proportional poppet valve.

There are two major problems to be solved in order to realize the design principal of the servo valve-type oil hydraulic valve in the water hydraulics. Because of the low viscosity of the water, water leaks inevitably in large amounts from the clearance between the proportional poppet surface and the sleeve. Also, since water has insufficient lubrication and lateral force is also applied to the proportional poppet, hydraulic lock can easily occur.

With consideration of these problems concerning water hydraulics, the inevitable leakage at the clearance between the proportional poppet and the sleeve is used as the initial flow to stabilize the initial operation of the proportional poppet as shown in Fig. 1(b). Also, a pressure equalizing groove is added to the proportional poppet to prevent the hydraulic lock of the proportional poppet [13].

2.1 Theoretical analysis

In this section, a theoretical analysis to design a servo valve-type water hydraulic valve, which uses a proportional poppet as the main valve, as well as the reflection method on typical characteristics of water are explained. As shown in Fig. 2, the main flow rate from the proportional poppet and the pilot flow rate from the pilot valve can be expressed as Eqs. (1), (2) by using the orifice equation.

$$q_m = C_m W_m x_m \sqrt{2(p_s - p_m)/\rho} \tag{1}$$

$$q_p = C_p A_p \sqrt{2(p_c - p_p)/\rho} \tag{2}$$

where, $W_m = \pi d_{m1} \sin\theta$, $A_p = \pi d_{p1} x_p$

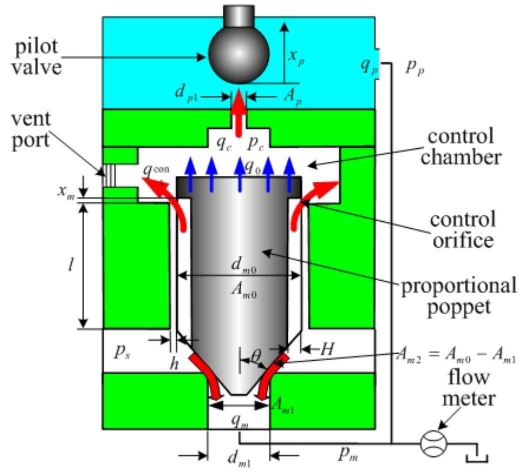


Fig. 2. Simplified model of proportional poppet of the servo valve-type water hydraulic valve.

The open area of the control orifice can be expressed as Eq. (3)

$$A_c = nW_c x_m \tag{3}$$

Since water viscosity is low in the servo valve-type water hydraulic valve using proportional poppet, the leakage flow of q_0 occurs from the clearance between the proportional poppet and the sleeve, as shown in Fig. 2. In this study, this leakage flow q_0 is regarded as the initial flow that stabilizes the initial operation of the proportional poppet. Therefore, the proportional poppet of a novel servo valve-type water hydraulic valve proposed in this study is designed as a critical lap. Meanwhile, the flow rate from the clearance between the proportional poppet and the sleeve can be expressed as Eq. (4) by using the leakage equation of the annulus clearance because the flow is characterized by a small Reynolds coefficient, which in turn allows the flow to be regarded as laminar flow [14].

$$q_0 = \frac{p_s - p_c}{R_c} \tag{4}$$

where, $R_c = 12\mu l / \pi d_{m0} h^3$

Thus, the control flow rate and the pilot flow rate of a servo valve-type water hydraulic valve can be expressed as Eqs. (5) and (6), respectively.

$$q_{cont} = C_c A_c \sqrt{2(p_s - p_c)/\rho} \tag{5}$$

$$q_c = q_{\text{cont}} + q_0 \tag{6}$$

The report of Wu [8] confirms through an experiment that the steady state flow force acting on the oil hydraulic proportional poppet is very weak compared with the force due to the pressure (even in the worst case of the experiment, the steady state flow force was below 7% of the force produced by pressure).

Based on the experimental result of Wu [8], it is possible to neglect the steady state flow force due to the control flow from the control orifice and the main flow from the proportional poppet because the developed valve operates basically under the same principle of oil hydraulic proportional poppet by Wu.

Therefore, the force equilibrium equation related to the top and bottom of the proportional poppet can be derived as Eq. (9) and expressed as Eq. (10).

$$p_c A_{m0} = p_m A_{m1} + p_s A_{m2} \tag{7}$$

$$p_c = S_1 p_m + S_2 p_s \tag{8}$$

where, $S_1 = A_{m1}/A_{m0}$, $S_2 = A_{m2}/A_{m0}$, $S_1 + S_2 = 1$

In the practical use of the servo valve-type water hydraulic valve, because the paths of the pilot flow and main flow are joined to reduce the wasteful flow as shown in Fig. 2, it can be assumed that $p_p = p_m$. Substituting Eq. (8) into Eq. (2), Eq. (9) can be derived from Eq. (1).

$$q_m = \frac{C_m W_m}{C_p A_p \sqrt{S_2}} \cdot x_m \cdot q_p \tag{9}$$

Applying the continuity equation at the pilot flow rate yields Eq. (10) and the displacement of the proportional poppet x_m can be derived as Eq. (11).

$$q_c = q_p \tag{10}$$

$$x_m = \frac{C_p A_p}{C_c n W_c}\sqrt{\frac{S_2}{S_1}} - \frac{\sqrt{\rho\,(p_s - p_c)/2}}{C_c n W_c R_c} \tag{11}$$

Substituting Eq. (11) into Eq. (9), it is possible to obtain the relation between the main flow rate and the pilot flow rate as Eq. (12).

$$q_m = \frac{C_m W_m}{C_c n W_c \sqrt{S_1}}(q_p - q_0) \tag{12}$$

where, as expressed in Eq. (4), q_0 is the leakage flow from clearance between the proportional poppet and the sleeve, namely, the flow passing through the fixed orifice.

If the supply pressure p_s and load pressure p_m at the bottom side of the servo valve-type water hydraulic valve are constant, the control chamber pressure p_c is also constant in the steady state because p_c is determined by the cross-section area A_{m0}, as shown in Eq. (7). Also, q_0 is constant. In addition, it can be proven from Eq. (4) that q_0 is proportional to the cube of h. If h is very small, the leakage flow q_0 is also small.

Based on the above theoretical analysis, the main flow rate q_m of a servo valve-type water hydraulic is almost linearly related to the pilot flow rate q_p. Therefore, all parameters necessary to design a prototype of a water hydraulic proportional poppet-type valve, such as the width W_c of the position feedback groove, the depth H of the position feedback groove, the number n of the position feedback groove, and the size of clearance between the proportional poppet and sleeve h, can be determined from the above analysis.

Substituting Eqs. (2), (6) into Eq. (10) and abridging it with the use of Eqs. (4), (5), (7) and (8), the displacement x_p of the pilot valve can be expressed as Eq. (13). When the supply pressure p_s and the load pressure p_m at the bottom side of the proportional poppet are constant, the second term on the right is constant in the steady state. Therefore, the displacement of the pilot valve x_p and the open area of the control orifice to the control chamber $nW_c x_m$ have a linear relation.

$$x_p = \frac{C_c n W_c x_m}{C_p W_p}\sqrt{\frac{S_1}{S_2}} + \frac{1}{C_p W_p R_c}\sqrt{\frac{S_1}{S_2}}\sqrt{\frac{\rho}{2}(p_s - p_c)} \tag{13}$$

2.2 Manufacture of a prototype valve

In this study, the validity of the theoretical analysis and design principle of the proposed valve was reviewed, and the performance characteristics of the servo valve-type water hydraulic valve were confirmed by several experiments.

A prototype of the servo valve-type water hydraulic valve using proportional poppet was fabricated. Fig. 3 is a photo view of the servo valve-type water hydraulic valve using proportional poppet and actual structure of the prototype valve. In this study, by considering the practicality and commercialization of

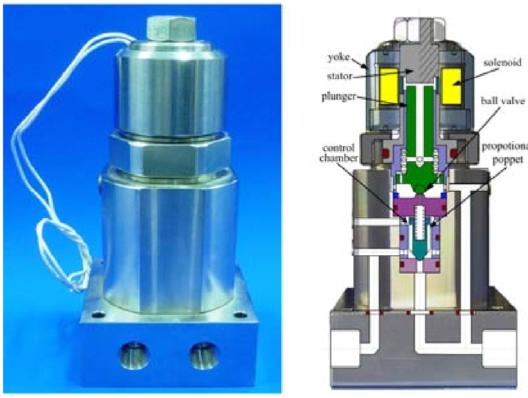


Fig. 3. Photo view of the servo valve-type water hydraulic valve using proportional poppet.

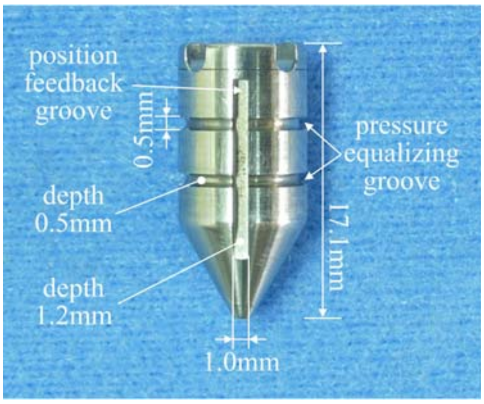


Fig. 4. Photo view of the proportional poppet with a position feedback groove and two pressure equalizing grooves.

the proposed valve, the prototype of the servo valve-type water hydraulic valve using proportional poppet was designed to have self-contained pilot valve that is operated by high speed solenoid as shown in Fig. 3. All components of the valve are made from precipitation hardening stainless steel AISI630. In particular, the proportional poppet and the valve seat are given appropriate hardness by solution heat treatment H900.

In the actual fabrication of the prototype servo valve-type water hydraulic valve, the width of position feedback groove W_c is designed to 1.0 mm and installed at both sides of the proportional poppet symmetrically. Also, by installing two pressure equalizing grooves having a width of 0.5 mm and depth of 0.5 mm at the circumference of the proportional poppet as shown in Fig. 4, it is possible to prevent a malfunction caused by hydraulic lock between the proportional poppet and sleeve, as well as to reduce friction [13].

Table 1. Specific value of the design parameters.

Max. supply pressure	$p_{s\max}$	14 MPa
Max. main flow rate	$q_{m\max}$	11 L/min
Diameter of the proportional poppet	d_{m0}	7.99 mm
Diameter of the seat of the proportional poppet	d_{m1}	2.35 mm
Cross-sectional area of the proportional poppet	A_{m0}	50.14 mm^2
Cross-sectional area of the seat of the proportional poppet	A_{m1}	4.34 mm^2
Cross-sectional area for supply pressure	A_{m2}	45.8 mm^2
Clearance between the proportional poppet and sleeve	h	10 µm
Depth of the position feedback groove	H	1.2 mm
Length of clearance h	l	9 mm
Number of the position feedback groove	n	2
Width of the position feedback groove	W_c	1.0 mm
Max. stroke of the proportional poppet	$x_{m\max}$	0.55 mm

Table 1 shows the design parameters for the above design.

3. Performance evaluation of a prototype valve

To evaluate the performance characteristics of the novel servo valve-type water hydraulic valve using proportional poppet, an experimental system is set up as shown in Fig. 5. The pressure supplied from the water hydraulic power unit to the experimental system is maintained at 7 MPa by using the relief valve. All the signals are acquired, sent, displayed and saved by using Matlab/Simulink Real-Time Windows Target running on a PC in real time.

Fig. 5 shows an experimental setup to evaluate the performance characteristics of the servo valve-type water hydraulic valve using proportional poppet. In the practical use of a servo valve-type water hydraulic valve using proportional poppet, it is desirable to use overall flows. Therefore, the paths of main flow and pilot flow are joined to eliminate wasteful flow. However, in the experiment shown in Fig. 5, two flow paths are separated and the changes of the main flow rate against the changes of the pilot flow rate are measured to investigate the relationship between the main flow rate and the pilot flow rate.

In the meantime, the prototype valve has a self-contained pilot valve operated by high speed solenoid as shown in Fig. 3 and Fig. 5. In practical use, the pilot valve operated by solenoid is generally

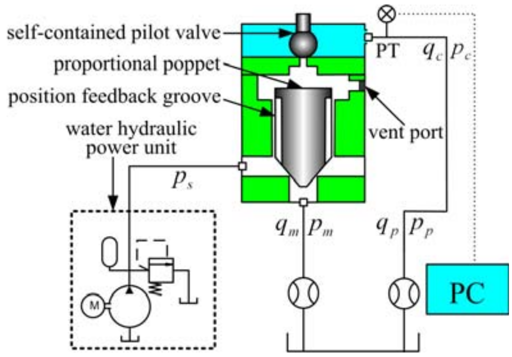
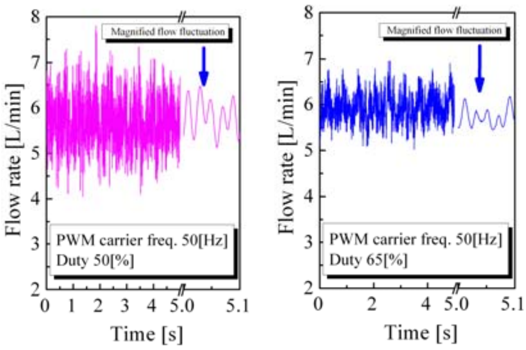


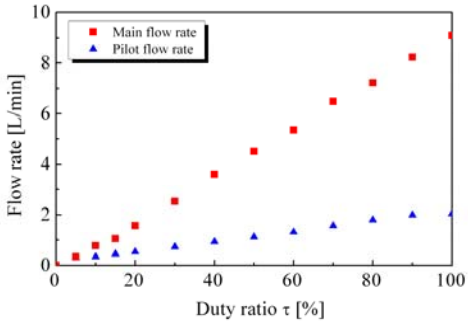
Fig. 5. Experimental setup for comparison of flow characteristic between main flow rate and pilot flow rate.



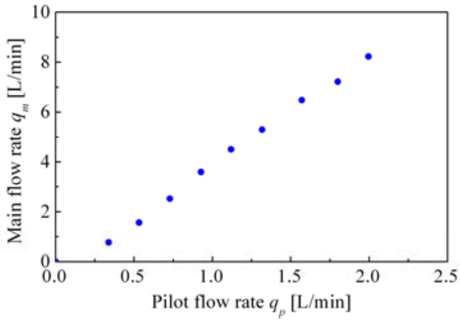
(a) Ordinary poppet (b) Proportional poppet

Fig. 6. Comparison of flow fluctuation.

controlled by PWM control method. When the pilot valve is controlled by PWM control method, the flow fluctuation of proportional poppet occurs by the on-off operation of the pilot valve as shown in Fig. 6. Fig. 6 shows a comparison of the flow fluctuation of the main flow rate between a simple two stage-type water hydraulic valve using conventional poppet [6] and the servo valve-type water hydraulic valve using proportional poppet. To compare the similar mean flow rate of two kinds of valves, the reference input pulse was applied to the pilot valve in the form of the carrier wave of 50 HZ and the duty was 50% and 65%, respectively. The main flow rate from the simple two stage-type water hydraulic valve using conventional poppet fluctuates violently because of the simple on-off operation. However, in case of the servo valve-type water hydraulic valve using proportional poppet, the main flow rate is proportional to the duty ratio of PWM and has comparatively small fluctuation because the main flow is proportional to the mean pilot flow in this valve.



(a) Mean value of main flow rate and pilot flow rate according to the change of duty ratio



(b) Relation between the main flow rate and pilot flow rate

Fig. 7. Mean flow rate characteristic of prototype valve.

If the theoretical analysis and design principle of the servo valve-type water hydraulic valve using proportional poppet is valid, the mean flow rates from the main and pilot valve have to have proportional relationship.

Fig. 7(a) and Fig. 7(b) show the relationship between the mean value of the main flow rate and that of the pilot flow rate. The experimental results show a linear relation in the overall range as a derived relationship between the main flow rate and the pilot flow rate in Eq. (14) in the theoretical analysis. Therefore, the theoretical analysis and design method for the servo valve-type water hydraulic valve were validated by these experiments.

4. Analysis of occurrence of cavitation through experiment

In the theoretical analysis, the control chamber pressure p_c always has a constant value when the supply pressure p_s and load pressure p_m are constant as shown in Eq. (8). To identify this characteristic, the servo valve-type water hydraulic valve using proportional poppet is driven by PWM control under the no load condition as shown in Fig. 5. The reference input pulse was applied to the self-contained pilot valve in the form of the carrier wave of 2 HZ and the duty was 40%. In this experiment, the frequency of the carrier wave for PWM control is very low because when the frequency of the carrier wave is high, a normal change of the pressure cannot be observed.

Fig. 8 shows the experimental result on the control chamber pressure p_c. When the valve is switching, there is a pressure fluctuation of approximately 100 Hz. It can be considered that a 1.5m high pressure hose between the water hydraulic pump and the valve causes this pressure fluctuation because the frequency of the pressure fluctuation decreases when the length of the high pressure hose is lengthened. Accordingly, this pressure fluctuation can be neglected in the process of analyzing performance characteristic of the servo valve-type water hydraulic valve. The proportional poppet moves to the position which the force by supply pressure p_s and by control chamber pressure p_c take equilibrium due to the ratio of the upper and lower part area of the proportional poppet defined as S_2.

In Fig. 8, the control chamber pressure p_c rises up promptly against the input pulse and it converges to 3.8 MPa within approximately 0.06 s.

However, as a result of this experiment, a strange phenomenon that cannot be explained solely by theoretical analysis was discovered. The theoretical value that the control chamber pressure p_c converges is approximately 6.3 MPa from Eq. (8) and Table 1 because the load pressure p_m is zero under no load condition and the supply pressures p_s and S_2 are 7 MPa and 0.91, respectively. However, the experimental results shown in Fig. 8 demonstrate that the control chamber pressure p_c converges to 3.8 MPa, which is much lower than 6.3 MPa.

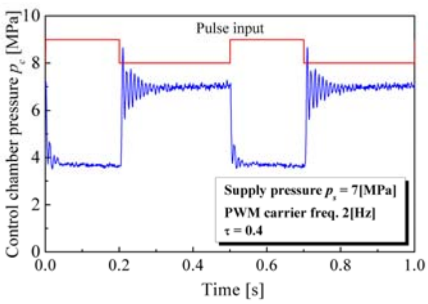


Fig. 8. Change of the control chamber pressure p_c.

Accordingly, the pressure change of the control chamber according to the change of the pilot flow is checked over by using the experimental equipment as shown in Fig. 9. In this experimental setup, the self-contained pilot valve is turned off and the manual valve is connected to the vent port of the servo valve-type water hydraulic valve as a remote control pilot valve to investigate the change of control chamber pressure p_c in detail without pressure fluctuation. In addition, both flow rates from the main and pilot valve are measured under no load condition.

Fig. 10 shows the change of control chamber pressure p_c according to the change of main flow rate q_m and pilot flow rate q_p. As shown in Fig. 10 the main flow rate changes linearly along with the change of the pilot flow rate. The pressure in the control chamber p_c is decreasing along with an increasing pilot flow rate despite the fact that the pressure in the control chamber has to remain at a constant value (6.3 MPa) from Eq. (8).

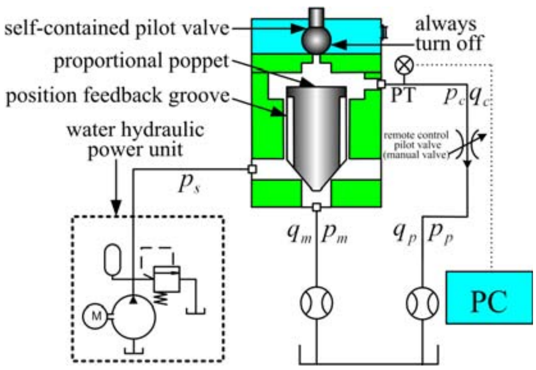


Fig. 9. Experimental setup for checking over the change of control chamber pressure p_c according to the change of main and pilot flow rate.

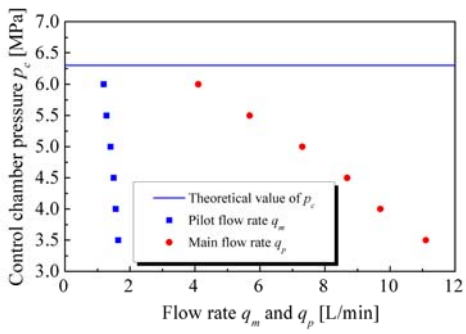


Fig. 10. Change of control chamber pressure p_c according to the change of flow rate.

Since the viscous force of water acting on the proportional poppet is smaller than that of oil (even in the worst case of the experiments, the viscous force of water was below 7% of the force produced by pressure) [8] and steady flow force is small compared with the force caused by the pressure, it does not have an effect on the decrease of the pressure in the control chamber as shown in Fig. 10.

Therefore, under no load condition, to satisfy Eq. (8) by using pressure values of p_s, p_c, p_m which are measured through an experiment shown in Fig. 9, there is no way to assume that the effective cross-section area of an actually pressured area A_{m0} or A_{m2} is changed by an unknown cause. Accordingly, it can be assumed that one of the primary causes of this discordance between the theoretical analysis and experiment is the occurrence of cavitation.

Since A_{m0} is almost unaffected structurally by cavitation, it can be regarded that the cavitation occurring at the orifice around the proportional poppet as shown in Fig. 11 causes the change of the effective cross-section area of A_{m2}.

Fig. 12 shows the change of the effective cross-section area A_{eff} of A_{m2} calculated by Eq. (8) based on the measured pressure values in the experiment shown in Fig. 10. From this result, it is regarded that according to the increase of flow rate, the occurrence areas of the cavitation increase and A_{eff} decreases. However, this result is only calculated under the assumption that the cavitation occurs around the part of the orifice of the proportional poppet and causes the change of A_{eff}. Accordingly, the cause of the pressure drop in the control chamber is not obvious only by this fact.

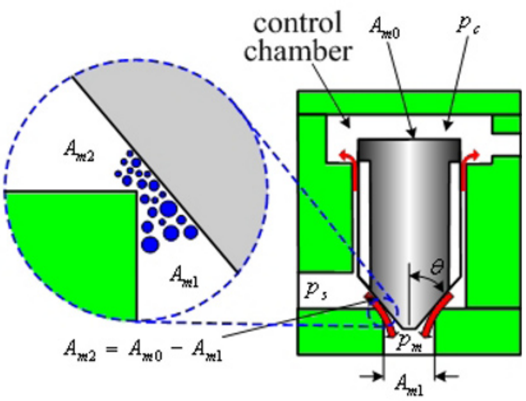


Fig. 11. Change of effective cross-section area A_{eff} of A_{m2} by the occurrence of cavitation.

If the cause of the pressure drop in the control chamber is due to the change of effective cross-section area of A_{m2} caused by cavitation, it is regarded that the pressure in the control chamber p_c will recover to the theoretical value by adjusting the load pressure p_m at the bottom of the proportional poppet, which leads to the prevention of cavitation. Therefore, in order to prove this theoretical deduction, the experiments changing the load pressure p_m from 0 to 6.5 MPa are carried out by using the experimental setup shown in Fig. 13.

Fig. 14 shows the experimental results of the experimental setup shown in Fig. 13. The changes of the control chamber pressure p_c are expressed in the form of the cavitation coefficient.

In general, the cavitation coefficient at the orifice of valves can be defined as Eq. (14),

$$k = \frac{p_2 - p_g}{p_1 - p_2} \tag{14}$$

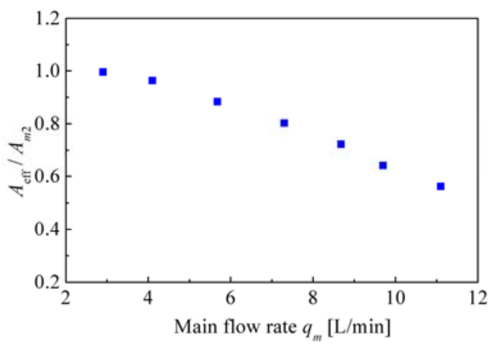


Fig. 12. Calculated change of effective cross-section area A_{eff}.

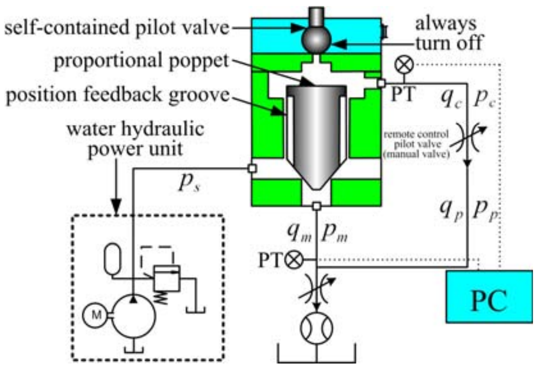


Fig. 13. Experimental setup for the investigation of cavitation around the proportional poppet.

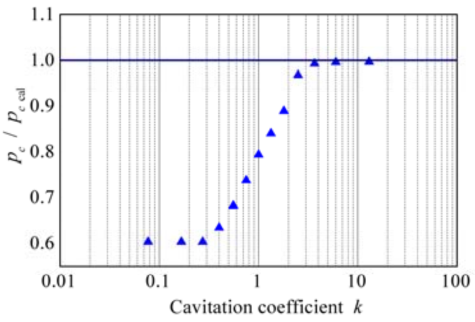
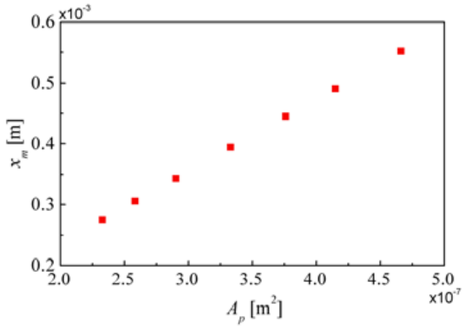
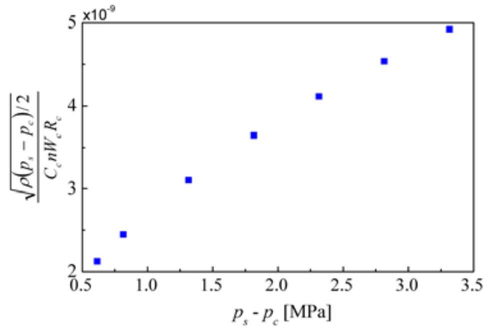


Fig. 14. Change of control chamber pressure according to cavitation coefficient.



(a) Relationship between the open area of orifice and displacement of proportional poppet



(b) Changing rate of the second term of Eq. (13) according the change of p_c

Fig. 15. Calculation results of Eq. (13).

where, p_1 and p_2 represent pressure of the top and bottom sides of the orifice, respectively, and p_g is a saturated vapor pressure which the bubbles in water start to produce.

A saturated vapor pressure p_g at 20 °C is approximately 2.34 kPa. Since this value is exceedingly small compared with the p_2, Eq. (14) can be expressed as Eq. (15) by using the variables used in this study.

$$k \approx \frac{p_m}{p_s - p_m} = \frac{p_m / p_s}{1 - (p_m / p_s)} \tag{15}$$

As shown in Fig. 14, the control chamber pressure p_c is maintained at the theoretical value until the cavitation coefficient k is 3.7 (load pressure is 5.5 MPa). From this result, it is identified that the cavitation does not occur around the proportional poppet. In addition, the effective cross-section area of A_{m2} is not changed. Meanwhile, it is identified that the cavitation coefficient starts to decrease abruptly around 3.7 and continuously decreases between the range in which the cavitation coefficient k is 3.7 (load pressure is 5.5 MPa) to 0.4 (load pressure is 2 MPa). From this experimental result, cavitation can be considered to start to occur when k is 3.7. Also, the area where cavitation is occurring is increasing according to the decrease of cavitation coefficient k. Accordingly, it can be supposed that the effective cross-section area A_{m2} is also decreasing in this range. Within the range where the value of the cavitation coefficient is less than 0.4, the control chamber pressure is maintained at a constant value (3.8 MPa). This means that the area where cavitation is occurring is saturated in this range. From these results, it is possible to relate the cause of an unknown pressure drop in the control chamber with the occurrence of cavitation. Therefore, based on the relationship between the pressure drop in the control chamber and the cavitation coefficient, it can be concluded that occurrence of cavitation causes changes in the effective cross-section area A_{m2} and the pressure drop in the control chamber.

The problem related to cavitation around the proportional poppet orifice is predicted with the advent of a novel water hydraulic system. In this study, since the proportional poppet and valve seat of the prototype valve are given appropriate hardness by solution heat treatment H900, cavitation erosion did not occur in spite of long-term use. However, it is necessary to examine the material of the proportional poppet which can prevent cavitation erosion for the commercialization of the developed valve.

In the meantime, when the control chamber pressure is changed by the occurrence of cavitation, it is expected that the linear relation between the main flow rate and pilot flow rate will be broken up by Eq. (13). However, as shown in Fig. 7 and Fig. 10, in spite of the pressure drop in the control chamber, the experimental results concerned with the relation between the main flow rate and pilot flow rate show a linear relation.

To verify the above experimental results, the displacement of the proportional poppet x_m in Eq. (13) and the value of the second term of Eq. (13) are calculated based on a measured value of the control chamber pressure p_c in Fig. 10 and shown in Fig. 15.

As shown in Fig. 15(a), the open area of orifice A_p of the pilot valve and displacement x_m of the proportional poppet have a linear relation. This means that the changing rate of the second term of Eq. (13) is very small and it can also be identified from Fig. 15(b).

Therefore, although the pressure in the control chamber drops due to the occurrence of cavitation, it does not affect the performance of the servo valve-type water hydraulic poppet valve and it can be confirmed that the theoretical analysis of this study is appropriate.

5. Conclusions

By making the best use of the leakage characteristic of water due to the low viscosity, a novel servo valve-type water hydraulic poppet valve, which uses the leakage from the clearance between the proportional poppet and sleeve as the initial flow to stabilize the initial operations of the proportional poppet, is developed. Also, the linear relation between the main flow rate and pilot flow rate and the adaptability of the servo valve-type water hydraulic poppet valve to PWM control are confirmed through experiments.

Finally, the theoretical analysis and design method for the servo valve-type water hydraulic poppet valve were validated by experiments. The developed servo valve-type water hydraulic poppet valve was found to be desirable and sufficient for practical use.

Moreover, it is confirmed through experiments that the occurrence of the pressure drop in the control chamber according to the change of the pilot flow is due to the occurrence of cavitation around the orifice of the proportional poppet. Finally, it is also proved that the pressure drop in the control chamber does not have an influence on fundamental characteristics of the developed valve.

Nomenclature

A_c : Open area of the control orifice (for water hydraulic systems with critical lap)
A_{coil} : Open area of the control orifice (for oil hydraulic systems with under lap)
A_{eff} : Effective cross-sectional area
A_{m0} : Cross-sectional area of the proportional poppet
A_{m1} : Cross-sectional area of the seat of the proportional poppet
A_{m2} : Cross-sectional area for supply pressure
A_p : Open area of the pilot valve
C_c : Flow coefficient of the control orifice
C_m : Flow coefficient of the proportional poppet
C_p : Flow coefficient of the pilot valve
d_{m0} : Diameter of the proportional poppet
d_{m1} : Diameter of the seat of the proportional poppet
d_p : Diameter of the pilot valve
D : Displacement volume of the water hydraulic motor
h : Clearance between the proportional poppet and sleeve
H : Depth of the position feedback groove
k : Cavitation coefficient
l : Length of the clearance between the proportional poppet and sleeve
n : Number of the position feedback groove
p_c : Control chamber pressure
$p_{c\,\mathrm{cal}}$: Theoretically calculated control chamber pressure
p_g : Saturated vapor pressure
p_m : Load pressure applied to the proportional poppet
p_p : Lower chamber pressure of the pilot valve
p_s : Supply pressure
p_1 : Pressure of top side of orifice
p_2 : Pressure of bottom side of orifice
q_c : $q_c = q_{\mathrm{cont}} + q_0$
q_{coil} : Control flow rate from control orifice (for oil hydraulic systems with under lap)
q_{cont} : Control flow rate from control orifice
q_0 : Leakage flow between the proportional poppet and sleeve
q_m : Main flow rate from the proportional poppet
q_p : Pilot flow rate from the pilot valve
R_c : Coefficient related on annulus leakage
S_1, S_2 : Area coefficient, $S_1 = A_{m1}/A_{m0}$, $S_2 = A_{m2}/A_{m0}$
W_c : Width of the position feedback groove
W_m : Coefficient related on open area of proportional poppet, $\pi d_{m1} \sin\theta$
W_p : Coefficient related on open area of pilot valve
x_i : Under lap length of the proportional poppet
x_m : Displacement of the proportional poppet
x_p : Displacement of the pilot valve

θ : Conical half-angle of the proportional poppet
μ : Dynamic viscosity
ρ : Density of water
τ : Duty ratio

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Sung-Hwan Park received the B.S. and M.S. in precision mechanical engineering from Pusan National University in 1990 and 1992, respectively. He received the Ph.D. in precision mechanical engineering from Pusan National University in 1996, and the Dr. Eng. in control systems engineering from Tokyo Institute of Technology in 2005. He is a research assistant professor in the School of Mechanical Engineering, Pusan National University, Korea. His research interests include hydraulic control systems, design of electro-hydrostatic actuators, water hydraulic control systems and brake systems for railway rolling stocks