DEVELOPMENT OF A HIGH-SPEED ON/OFF DIGITAL VALVE FOR HYDRAULIC CONTROL SYSTEMS USING A MULTILAYERED PZT ACTUATOR

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Abstract

In this study, a high-speed on/off digital valve was developed for use in a hydraulic control system. The device basically consists of a poppet valve acting as the main valve, and a multilayered piezoelectric (PZT) actuator for driving the poppet valve. A hydraulic amplifier was adopted to increase the actuation of the PZT actuator to the poppet valve. A compensation mechanism was set up to reduce this temperature effect in the hydraulic actuation of the PZT actuator. This problem arises when the oil temperature increases and causes the valve displacement to fluctuate slightly.

The static and dynamic characteristics of the device were investigated by experiment and computer simulation. As a result, it was found that the switching time of the valve is less than 0.7 ms. Moreover, the valve can be driven by a PWM carrier wave using frequencies of up to 500 Hz. Additionally, the validity of the temperature compensation mechanism was confirmed. Hence, this valve may be determined as feasible device to be used in hydraulic systems.

Keywords: hydraulic actuator, piezo-element, digital control, PWM control, digital valve

1 Introduction

A digital valve used in hydraulic control systems can be a simple and logical choice of interface between a computer and hydraulic systems, introducing both low cost and high reliability to the system. On the other hand, the performance of these systems is unsatisfactory when compared to analogue-type hydraulic systems. This fact is encountered in a servo-system installed with an electrohydraulic servo valve. Hence, it is a matter of importance to develop a digital valve capable of similar high performance. Since the dynamic performance of digital valves is generally estimated by the speed of its on/off action, it is expected to develop a high-speed digital valve with very fast switching. It can be said that most of digital valves existing in the market use electromagnetic actuator, as is the one developed by Tanaka (1984). This type of the valve, however, has a limit of speediness due to its actuating principle. Recently, digital valve utilizing PZT actuator is receiving much attention as a next generation valve. The attractiveness of PZT actuator lies in its very high speed action, small size and strong actuation force. Lühmann et al (1982) and further Yokota et al (1990) independently developed high speed digital valve which is directly driven by PZT actuator. However, in this kind of digital valve, the drawback is that the PZT actuator cannot produce enough displacement to actuate the main valve. For this reason, it is necessary, in general, to adopt a certain kind of displacement amplifier. For example, Yokota et al (1990) developed PZT valve using lever or two stage servo mechanism in order to increase flow rate of the valve.

In this study, a high-speed switching valve using a multilayered PZT actuator was developed. It acts as the first-stage amplifier and its displacement is multiplied by a hydraulic amplifier (Wennmacher et al, 1993). A compensation mechanism for reducing the temperature effect in the hydraulic amplifier was also implemented, since the valve causes a displacement fluctuation as the oil temperature increases. The static and dynamic characteristics of this valve are investigated by experiment and computer simulation.

2 Principle of the Digital Valve

The schematic diagram of the digital valve developed in this study is shown in Fig. 1. The valve basically consists of (a) the piezoelectric actuator, (b) oil cham

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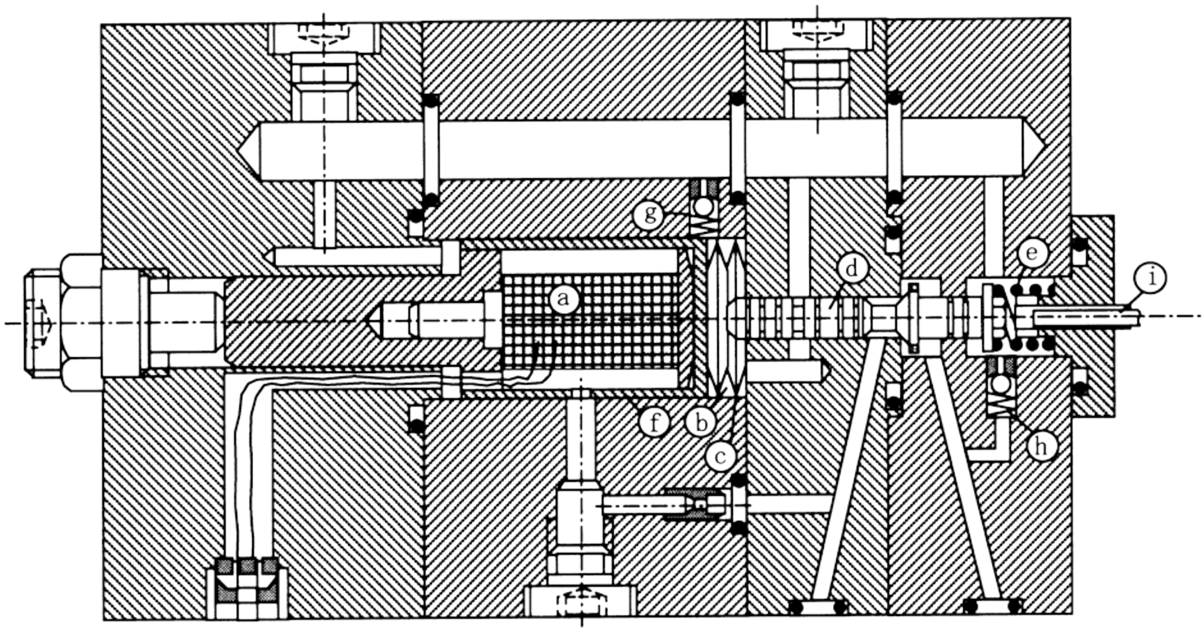


Fig. 1: *Schematic diagram of the designed valvedue to its actuating principle.*

ber, (c) spring, (d) poppet valve, (e) valve spring, (f)press piston, (g) check valve 1, (h) check valve 2 and (i) linear variable-difference transformer (LVDT). When the PZT actuator in Fig. 1 is switched on, the press piston (f) moves towards the right. At the same time, oil in the oil chamber (b) is compressed and the poppet valve (d) is pushed by the pressure in the oil chamber. The displacement of the poppet valve is amplified by this mechanism. The amplifying factor is the ratio of cross sectional areas of the press piston to the valve α ($=a_p/a_v$), when the compressibility of oil is neglected.

Increase in temperature causes an expansion of the oil in the hydraulic oil chamber. It also causes an undesirable change in valve displacement. In order to counteract this effect, the poppet valve and the press piston were designed to have a clearance to drain some of the heated oil, thereby decreasing the volume. Check valve 1 (g) is for resupplying the drained oil when necessary.

When the valve is in the ON position, the displacement of the poppet valve decreases because of leakage, and finally the poppet stops at the zero position. Afterwards, if the valve is switched OFF, the pressure of the oil chamber drops and oil is recharged through check valve 1. When the valve is operated at a certain duty, it acts in this way. However, with an increase in duty and a decrease in the time when the increase in duty and a decrease in the time when the valve is in the OFF position, there is less time than required to recharge the oil through the check valve. In this case, the displacement of the valve becomes smaller and finally the valve will be closed. Therefore, the viable range of duty is limited. The actual duty range of this valve will be described later.

The PZT actuator heats up with high-speed ON/OFF operation, and requires repolarization if the Curie temperature is exceeded. Oil flow around the PZT actuator accomplishes some of this cooling down process. That is, part of the oil from the pressure supply port will pass through the PZT chamber and exhaust through the drain port. Furthermore, for the purpose of safety, a temperature sensor is attached to switch off the device when the temperature exceeds a certain, set safety limit.

Table 1: Specifications of the PZT actuator (TOKIN NLA-18x10x10)

Max. operating voltage	100	[V]
Displacement	15.0 (±10%)	[μm/100V]
Force generation	3430 (±20%)	[N]
Resonant frequency	75 (±20%)	[kHz]
Static capacitance	6500	[nF]
Curie temperature	145	[°C]

Table 2: Dimensions and parameters of the valve

Diameter of press piston	$d_p = 20 \cdot 10^{-3}$	[m]
Diameter of poppet valve	$d_v = 5 \cdot 10^{-3}$	[m]
Spring coefficient of PZT	$k_{p1} = 2.287 \cdot 10$	[N/m]
Spring coefficient of valve	$k_v = 2.5 \cdot 10^4$	[N/m]
Mass of press piston	$m_{p1} = 28.8$	[g]
Mass of poppet valve	$m_v = 3.799$	[g]
Clearance of press piston	$d_{cp} = 5.5 \cdot 10^{-6}$	[m]
Clearance of poppet valve	$d_{cv} = 5.5 \cdot 10^{-6}$	[m]
Bulk modulus of fluid	$K = 1.58 \cdot 10^9$	[Pa]
Mass density of fluid	$\rho = 8.6 \cdot 10^2$	[kg/m^3]
Viscosity of fluid	$\eta = 1.2 \cdot 10^{-2}$	[Pas]
Sampling time for simulation	$t_s = 1 \cdot 10^{-6}$	[s]

Check valve 2 (h) acts as a relief valve to avoid cavitation in the oil chamber (b). The line pressure in the valve is kept under 6 MPa by this check valve 2. Sealing is used to close up the chamber (b) tightly. The valve body, fixed with 4 bolts, consists of 4 blocks in order to facilitate assembly. Spring (c) is attached to strengthen the body stiffness (the displacement of the poppet valve is reduced by about 10% without this spring). The specifications of the PZT actuator are shown in Table 1. The dimensions of the valve's parameters are given in Table 2.

3 Mathematical Model and Simulation of the Valve

In order to optimize the design parameters of the valve, and to analyze the dynamic performance of the valve, a simulation analysis was conducted. In this chapter the mathematical model of the system is discussed. The following assumptions are made for deriving the mathematical model of the system.

- The supply pressure from the pump p_s is constant.
- Compressibility of oil is taken into consideration only in oil chamber (b).
- Piezo actuator is considered as mass and spring system, and force of piezo F_p is regarded as an extra force.
- Mass of piezo actuator (a) and press piston (b) are regarded as unified.
- Influence of increase in temperature is neglected.

Equation of motion of piezo actuator with press piston;

$$m_p \ddot{x}_p + c_p \dot{x}_p + k_p x_p = F_p - a_p p_a \qquad (1)$$

$$(\text{if } F_p = 0 \text{ then } k_p = 0)$$

Equation of motion of poppet valve;

$$m_v \ddot{x}_v + c_v \dot{x}_v + k_v (x_v + x_{v0}) = a_v p_a \qquad (2)$$

Compressibility of oil in the oil chamber;

$$(v_a / K) \dot{p}_a = a_p \dot{x}_p - a_v \dot{x}_v - q_{lp} - q_{lv} - q_{cv} \qquad (3)$$

Volume of oil chamber;

$$v_a = v_{a0} - a_p x_p + a_v x_v - v_l \qquad (4)$$

where,

$$v_l = \int_0^t q_{lp} dt + \int_0^t q_{lv} dt + \int_0^t q_{cp} dt \qquad (5)$$

Leakage from oil chamber to press piston (Merritt, 1967);

$$q_{lp} = \frac{\pi \, d_p \, d_{cp}^3 \, p_a}{12 \, \mu \, l_p} \qquad (6)$$

Leakage from oil chamber to poppet valve;

$$q_{lv} = \frac{\pi \, d_v \, d_{cv}^3 \, p_a}{12 \, \mu \, l_v} \qquad (7)$$

Equation of motion of check valve;

$$m_{cv} \ddot{x}_{cv} + c_{cv} \dot{x}_{cv} + k_{cv} (x_{cv} + x_{cv0}) = -a_{cp} \Delta p_{cv} \qquad (8)$$

Flow rate through check valve;

i). $\Delta p_{cv} \geq 0$

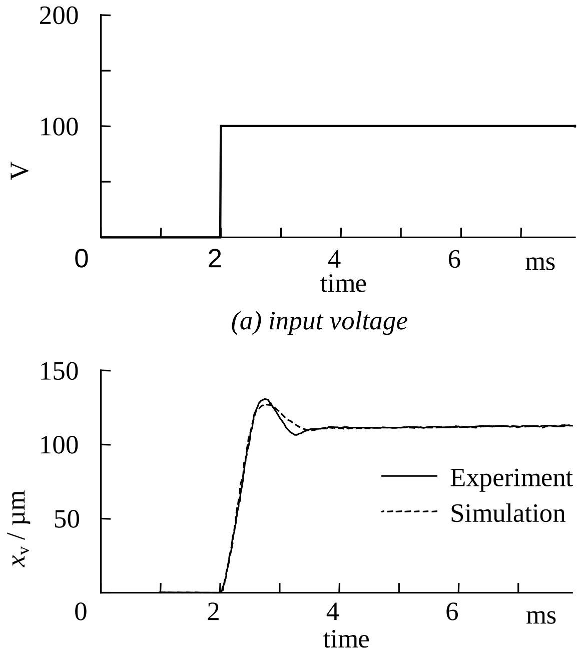
$$q_{cv} = c_c a_{cv} \sqrt{\frac{2}{\rho} \Delta p_{cv}} \qquad (9)$$

ii). $\Delta p_{cv} < 0$

$$q_{cv} = -c_c a_{cv} \sqrt{\frac{-2}{\rho} \Delta p_{cv}} \qquad (10)$$

where

$$a_{cv} = 2\pi \, r_{cv} \, x_{cv} \qquad (11)$$



(a) input voltage

(b) poppet displacement

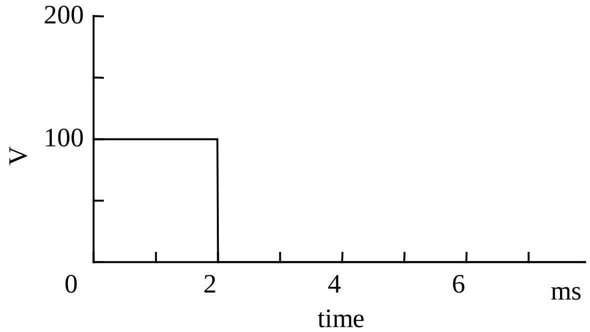
Fig. 2: *Response curve of poppet valve (OFF to ON switching, p_s = 0 MPa, T = 30°C)*

4 Performance of the High-speed Digital Valve

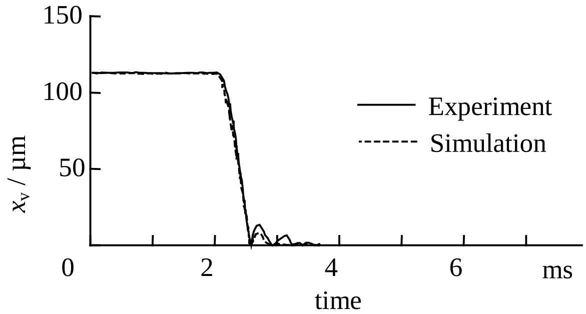
The design parameters of the valve were optimized by evaluating the simulated results of valve performance in advance. The valve block is immersed in fluid oil when it is assembled in order to exclude air from the oil chamber. Figure 2 shows response curves of experimental and simulated results when step voltage input (100 V) is applied to the valve. Figure 2 (a) shows voltage input and Fig. 2 (b) shows displacement of poppet valve x_v which is measured by LVDT. In this case, the supply pressure is set at 0 MPa. As shown in this figure, displacement of the poppet valve quickly converges to the maximum displacement within 0.7 ms. Though overshoot appears after reaching the maximum displacement, it quickly settles down to a constant value (ca. 100 μm). By comparing the experimental curves with the simulated ones, we were able to confirm that the simulated model properly represented the experiment results.

Figure 3 shows experimental and simulated results of ON to OFF switching. The displacement of the poppet falls to zero within 0.4 ms after the input voltage is reduced to 0 V. Although there were minor oscillations after closing, it settled down quickly.

This valve uses temperature compensation by oil leakage as mentioned before. In order to examine the influence of leakage, the displacement of the poppet was measured when the input voltage was held at 100 V, as shown in Fig. 4.



(a) input voltage



(b) poppet displacement

Fig. 3: *Response curve of poppet valve (ON to OFF switching, p_s = 0 MPa, T = 30°C)*

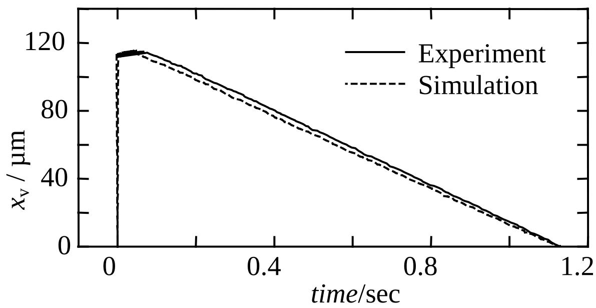


Fig. 4*:* *Response curve of poppet valve (Holding input voltage at 100 V)*

After reaching the maximum position, the poppet displacement constantly decreased due to oil leakage, and totally closed after approximately 1.13 s. Therefore, the total volume of leakage from the oil chamber can be calculated as ca. 1.7×10^{-3} cm^3/s. Simulated results also agreed well with experimental ones, and thus the validity of the mathematical model is confirmed.

The flow vs. duty characteristics when the valve is driven at f_c = 100, 200 and 500 Hz are shown in Fig. 5.

The broken lines were measured when the duty was increased from 0 % to 100 % and the solid lines were measured when the duty was decreased from 100 % to 0 %. When the valve is driven at f_c = 100 Hz, flow rate increases in proportion to the duty from *Duty* = 0 % to 75 %. However, it decreases over *Duty* = 75 %, because of a decrease in the period of the valve in the OFF position. This is, because there is not enough time to recharge the oil through the check valve 1, and the volume of oil in the oil chamber decreases, as mentioned before. The curves for which duty was decreased

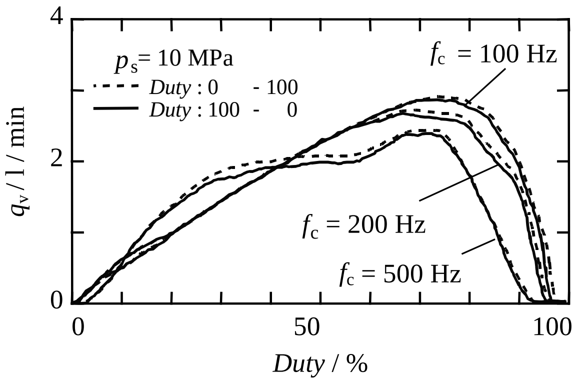


Fig. 5: *Flow vs. Duty characteristics.*

from 100 % to 0 % nearly agrees with the ones when duty was increased. This means that the poppet displacement recovered even after the poppet displacement reached x_v = 0 mm because oil is recharged through the check valve 1 when duty decreases. In case of the valve being driven with a carrier wave frequency f_c = 100 Hz, it can be said that the available duty range is 0 % to 75 %. When the valve is driven with f_c = 200 Hz, it works similarly as with f_c = 100 Hz, though each of them has different available duty ranges. In case of f_c = 500 Hz, it has nonlinear characteristics. However, this kind of nonlinearity can be linearized sufficiently by using linearization method (Ye, 1992) for practical use.

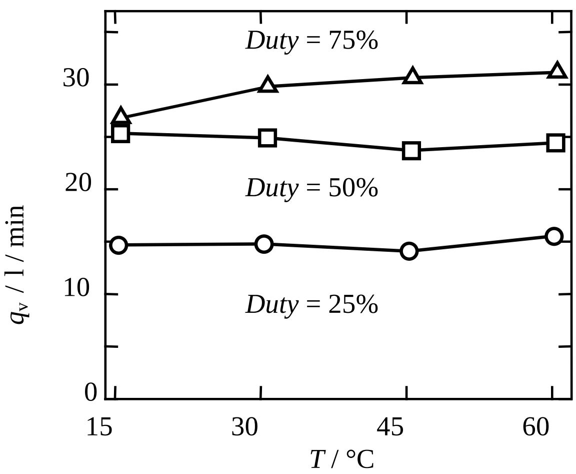


Fig. 6: *Flow rate vs. Temperature characteristics.*

Next, flow rate vs. temperature characteristics was measured in order to evaluate the validity of the temperature compensation, as shown in Fig. 6. In this measurement, the valve was driven with *Duty* = 25, 50 and 75 %. It should be noted that the valve has nearly constant flow rates even when the fluid temperature ranged from 15°C to 60°C. Figure 7 shows the flow vs. duty characteristics under the condition that the fluid temperature range was set from 15°C to 60°C. The lines indicating the fluid at different temperatures are fairly coincident. Thus it is can be said the temperature compensation mechanism was effective.

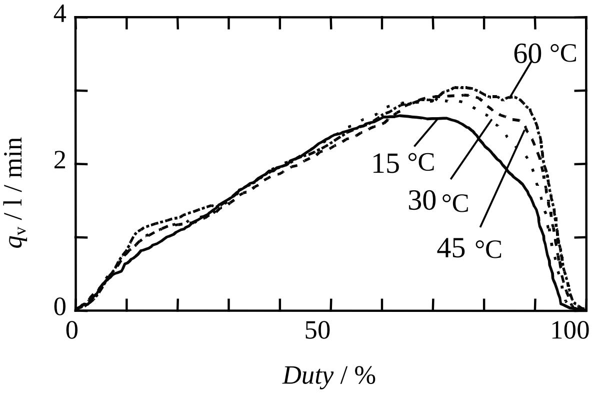


Fig.7: *Flow vs. Duty characteristics (15°C to 60°C).*

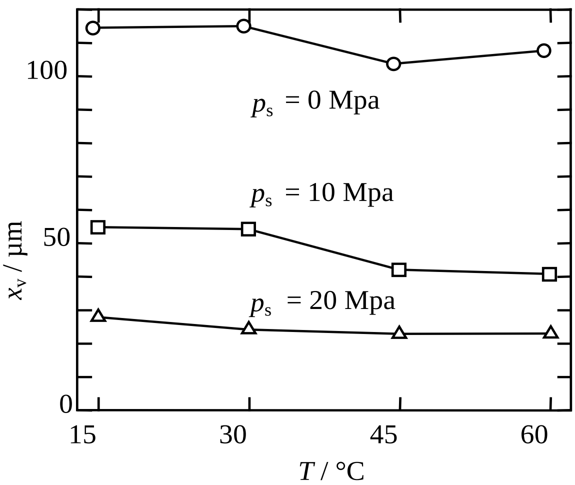


Fig. 8: *Maximum poppet displacement vs. Temperature characteristics.*

Figure 8 shows the characteristics of maximum poppet displacement vs. temperature. Without temperature compensation, poppet displacement fluctuation is estimated as Δx_v = 22.4 µm/K. It is observed from the figure that maximum displacement is almost constant with changes in temperature. However, it was verified that the displacement was inversely proportional with the supply pressure, p_s. It is presumed that flow force acting on the poppet is the source of this influence. The problem of flow force has been studied by many researchers. Tanaka (1984) implemented flow force compensation by attaching an orifice to the upper line of the poppet valve. This kind of compensation reduces the influence of flow force, and a sufficient flow rate can be obtained.

5 Conclusions

In this study, a high-speed switching valve using a multilayered-piezoelectric actuator with an hydraulic amplifier was developed. Static and dynamic characteristics of the valve were investigated by experiment and simulation. The results obtained are summarized as follows:

- The measured ON to OFF switching time is ca. 0.7 ms, and OFF to ON switching time is ca. 0.4 ms. The valve could be driven at up to f_c = 500 Hz carrier wave of PWM modulation. Additionally, the efficiency of the displacement amplifier was confirmed.
- The measured flow rate through the valve changed little with rises in temperature. Therefore, the validity of the temperature compensation was confirmed.

Nomenclature

a_{cp}	cross-sectional area of check valve poppet	[m^2]
a_{cv}	port area of check valve	[m^2]
a_p	cross-sectional area of press piston (f)	[m^2]
a_v	cross-sectional area of poppet valve	[m^2]
c_{cv}	viscous damping coefficient of check valve poppet	[Ns/m]
c_c	flow coefficient of check valve	[-]
c_p	viscous damping coefficient of piezo actuator with press piston	[Ns/m]
c_v	viscous damping coefficient of poppet valve	[Ns/m]
d_{cp}	clearance of press piston	[m]
d_{cv}	clearance of poppet valve	[m]
d_p	diameter of press piston	[m]
d_v	diameter of poppet valve	[m]
Duty	input duty to valve [= (acting time)/T_c]	[%]
f_c	frequency of PWM-carrier wave (f_c= 1/T_c)	[Hz]
F_p	maximum force of piezo actuator	[N]
K	bulk modulus of fluid	[Pa]
k_{cv}	spring coefficient of check valve	[N/m]
k_p	spring coefficient of spring with piezo actuator (c)	[N/m]
k_v	spring coefficient of valve spring	[N/m]
l_p	leakage length of press piston	[m]
l_v	leakage length of poppet valve	[m]
m_{cv}	mass of check valve poppet	[kg]
m_p	effective mass of piezo actuator with press piston	[kg]
m_v	mass of poppet valve	[kg]
p_a	pressure of oil chamber (b)	[Pa]
p_s	supply pressure	[Pa]
q_{cv}	flow rate through check valve	[m^3/s]
q_{lp}	leakage from oil chamber (b) to press piston	[m^3/s]
q_{lv}	leakage from oil chamber (b) to poppet valve	[m^3/s]

r_{cv}	radius of check valve poppet seat	[m]
v	volume of oil chamber (b)	[m^3]
v_{a0}	initial volume of oil chamber	[m^3]
v_1	volume of leakage	[m^3]
x_{cv}	displacement of check valve poppet	[m]
x_{cv0}	initial compression of check valve spring	[m]
x_p	displacement of press piston	[m]
x_v	displacement of poppet valve	[m]
x_{v0}	initial compression of valve spring	[m]
Δp_{cv}	pressure difference of check valve	[Pa]
η	viscosity of fluid	[Pas]
ρ	mass density of fluid	[kg/m^3]

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