Study on Dynamic Characteristics of Water Hydraulic Proportional Control Valve in Nonlinear Region

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Water hydraulic proportional control valves are a novel fluid control device using water as the working fluid. They are very hygienic and eco-compatible, permit high-output control, and have applications in many industries. Previously, the authors expressed its characteristics as a third-order transfer function including a compensation circuit (with spool displacement as a control parameter), solenoid, and pilot valve, and examined the effects of design parameters on frequency characteristics, step response, and system stability. Here, its characteristics are examined in the non-linear region by introducing the non-linearity of the control apertures and damping orifice. The results demonstrate the feasibility of applying a linear model in this region.
Keywords: Water hydraulics, Proportional control valve, Dynamic characteristics, Nonlinear region.
Target audience: Hydraulic components, Industrial machinery, Design Process.

1 Introduction

Water hydraulic proportional control valves are a new kind of fluid control device that employ plain water as the working fluid. In addition to being both very hygienic and eco-compatible, these devices allow systems to achieve high-output control. Their applications are wide-ranging, and include the foods, beverages, semi-conductors, medicine, pharmaceuticals, cosmetics, chemicals, and natural energy industries.

In previous reports which examined a control valve unit using theoretical analysis and empirical verification, the authors defined a transfer function divided into three components—the compensation circuit, with spool displacement as a control parameter; the solenoid; and the pilot valve—and proposed that the valve unit's characteristics could be expressed as a third-order transfer function. Using this transfer function, the effects of various design parameters on frequency characteristics, step response characteristics, and system stability were verified. The valve's transfer characteristics, calculated based on the thrust characteristics of the solenoid and the geometrical structure of the pilot valve, were stable. Its response characteristics were examined by modelling the system as a third-order transfer equation, incorporating a compensation circuit which included unstable factors. It was found that these unstable factors could be minimised and the control valve characteristics stabilised by optimising the proportional gain of the compensatory circuit and the integral time /1–5/. These findings apply to the linear region near the operating point; for practical purposes, the target region corresponds to ±10% of the operating point. When the device is operated in this region, the primary variables to regulate are the force and position of a cylinder displaced over a relatively small range.

However, in the hydraulics industry, there is a need for devices in which cylinder force and position can be controlled in the high-frequency range, in order to accommodate cylinders that are displaced over a large range and handle heavy flow rates. Under these operating conditions, control valve characteristics begin to exceed the linear region, and reflect the influence of nonlinearity. As the previously mentioned transfer function was approximated in the linear region, it may fail to completely reflect a valve's characteristics in the nonlinear region.

In this study, the authors verified the characteristics of this control valve in the nonlinear region by considering the non-linear behaviour of the valve's control apertures and damping orifice, which were approximated as linear in the previous report, and considered the validity of applying the linear model in this region. This task is an extremely important one from a design perspective, allowing us to accurately determine and understand the characteristics of a complex system with as simple a model as possible.

2 Structure and Features of the Water Hydraulic Proportional Control Valve

The structure and major specifications of the water hydraulic proportional control valve are shown in Figure 1 and Table 1, respectively. Due to its low viscosity, the use of tap water as the working fluid limits the formation of a water film within the clearance of the sliding member. In this valve, the spool is supported at both ends by hydrostatic bearings to avoid them coming into contact with the sleeve when displaced; this design reduces sliding wear and friction. The working fluid that passes through the hydrostatic bearings is fed into pressure chambers at both ends of the spool. Damping orifices are located between the pressure chambers at both ends of the spool and the return line; however, effective damping is difficult to achieve with a low-viscosity working fluid like water by throttling alone. The position of the spool is determined by the balance between the solenoid thrust and the spring force. The use of an extension spring frees one end of the spool, which allows the hydrostatic bearings to function more effectively with reduced moment and lateral force. In addition, the valve uses the spool position as a control parameter, as shown in Figure 2, with the aim of improving controllability via feedback control.

Figure 3 shows a schematic of the physical relationship between the spool, hydrostatic bearing orifice, and damping orifice. The hydrostatic bearing orifice, thanks to its location relative to the spool, forms a meter-in valve in the circuit to control the flow, helping to improve response characteristics during spool operation, in addition to effectively reducing wear and friction due to sliding. The damping orifice, on the other hand, helps to attenuate spool motion by forming a meter-out valve in the circuit. Their relative influence can be discussed using Figure 4 /4–5/. Here, C_r is defined as the equivalent diameter ratio of the damping orifice and hydrostatic bearing orifice, T_L is the time constant in the first-order lag transfer function for the pilot valve P(s), and ζ is the damping coefficient in the second-order lag transfer function for the solenoid and pilot valve. For low values of C_r, T_L is increased and the pilot valve is slower to respond, signifying that the damping orifice has meter-out effects on spool motion. For large values of C_r, T_L is decreased and the pilot valve is quicker to respond. In this case, the meter-in effects of the hydrostatic bearing orifice are greater than the meter-out effects of the damping orifice. This design, which combines the functions of hydrostatic-bearing and damping orifices, is a major feature of water hydraulic proportional control valves that use low-viscosity tap water as the working fluid.

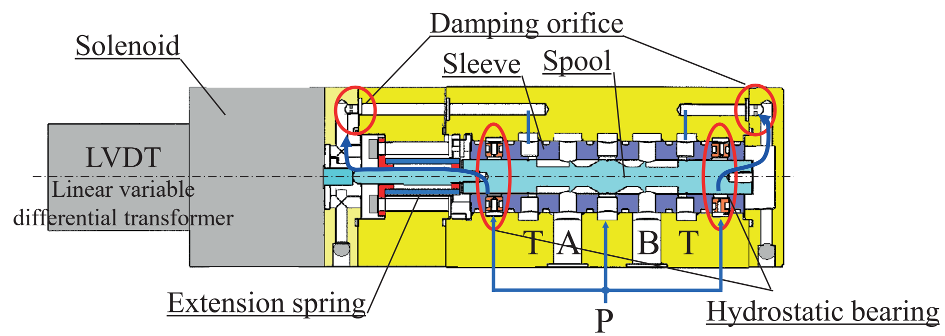

Figure 1: Structure of the water hydraulic proportional control valve.

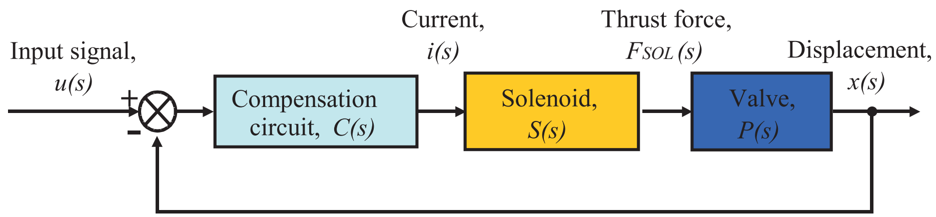

Figure 2: Block diagram of the valve.

ITEM	SPECIFICATION
Rated flow rate	3.3×10^{-4} m^3/s
Rated pressure	14×10^6 Pa
Working fluid	Tap water
Operating pressure range	3.5×10^6 to 14×10^6 Pa
Operating temperature range	2 to 50 °C
Input signal	+/− 10 V

Table 1: Specifications of the water hydraulic proportional control valve.

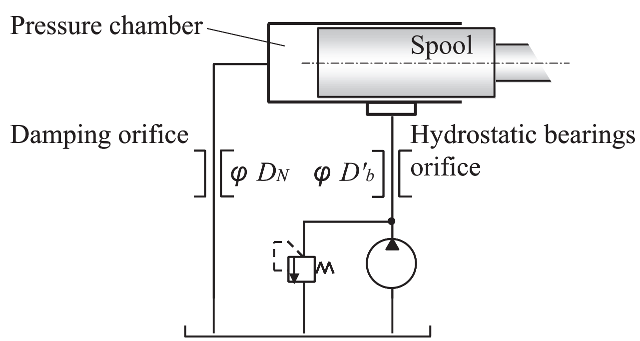

Figure 3: Physical relationships between the spool, hydrostatic bearing orifice, and damping orifice.

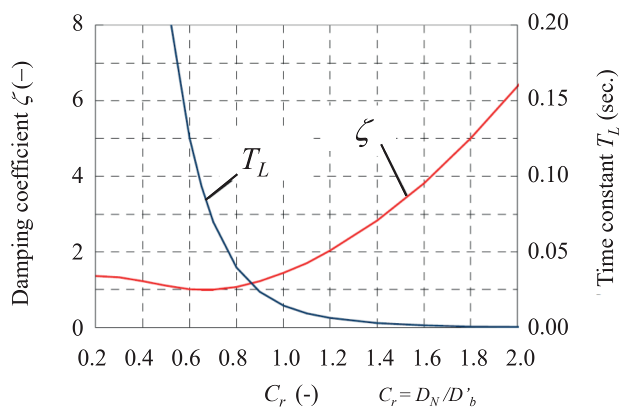

Figure 4: Relationships between the damping coefficient ζ and time constant T_L and the C_r value.

3 Basic Equations of the Valve

Figure 5 shows a schematic of the analysis model with parameters labelled. As stated above, the valve can be thought of as being divided into three components: a compensation circuit, a solenoid, and a pilot valve. Each component is described below /2–3/.

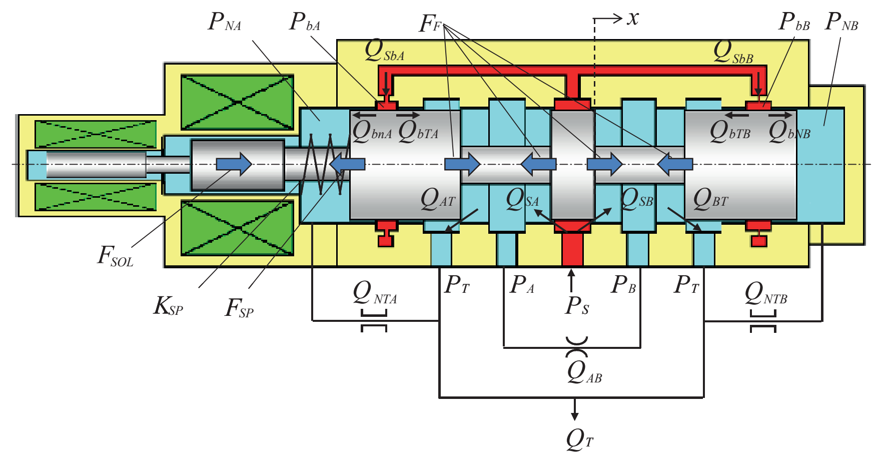

Figure 5: Definitions of valve parameters.

3.1 Compensation Circuit

The compensation circuit is modelled as a general PI controller:

$$C(s) = \frac{K_P T_I\, s + K_P}{T_I\, s}\ . \tag{1}$$

3.2 Solenoid Thrust

The thrust characteristics of the solenoid are defined as a first-order lag transfer function determined empirically in a previous report /3/, where τ_{SOL} is a time constant and K_{SOL} is a proportionality constant:

$$S(s) = \frac{K_{SOL}}{\tau_{SOL}\, s + 1}\ . \tag{2}$$

3.3 Flow Rate Equations

3.3.1 Orifice Flows

The inlet flow rates from the supply port to control ports A and B (Q_{SA}, Q_{SB}) and the outlet flow rates from these ports to the tank (Q_{AT}, Q_{BT}) can be expressed as follows:

$$Q_i = \mathrm{sign}(P_{i\_up} - P_{i\_down}) \cdot C_i \cdot A_i(x) \sqrt{\frac{2\left|P_{i\_up} - P_{i\_down}\right|}{\rho}}\ , \quad A_i(x) = w \cdot x\ , \tag{3}$$

where P_{i_up} and P_{i_down} are the pressures upstream and downstream of the orifice resistance, C_i is the flow coefficient, and A_i is the orifice cross-section, which is a function of orifice width w and spool position x. In orifice flows, pressure and the product of pressure and surface area have nonlinear characteristics.

3.3.2 Choked Flows

Based on empirical data, the flows through the hydrostatic bearings (Q_{SbA}, Q_{SbB}) have Reynolds numbers (Re) of 3500–5300, corresponding to the transition zone between laminar and turbulent flows. The laminar choked flow rate can be expressed as:

$$Q_i = \frac{\pi \cdot d^4 (P_{i\_up} - P_{i\_down})}{128 \cdot \mu \cdot l}, \tag{4}$$

where P_{i_up} and P_{i_down} are the pressures upstream and downstream of the choked orifice, μ is viscosity, d is the inner diameter, and l is the length.

3.3.3 Annular Clearance Flows

The flow rates through the annular clearance between the spool and the sleeve can be expressed as:

$$Q_i = \frac{\pi \cdot D_{SP} (P_{i\_up} - P_{i\_down})}{12 \cdot \mu \cdot l} \delta^3 \left(1 + 1.5\varepsilon^2\right), \tag{5}$$

where ε is the eccentricity of the spool with respect to the sleeve, which varies over 0–1 according to spool position, resulting in a total flow rate of 3.0–8.3×10^{-5} m^3/s /1/. This equation applies to the flows from the hydrostatic bearings to the pressure chambers at both ends of the spool (Q_{bNA}, Q_{bNB}), as well as outflows to the return line (Q_{bTA}, Q_{bTB}).

3.3.4 Pipe Flows

Based on empirical data, the flows through the damping orifices (Q_{NTA}, Q_{NTB}) are turbulent (Re = 6,000–10,000) and can be expressed as flows in a pipe with friction loss factor λ:

$$Q_i = \frac{\pi d}{4} \sqrt{\frac{d}{\lambda \cdot l \cdot \rho} \left| P_{i\_up} - P_{i\_down} \right|}, \tag{6}$$

$$\mathrm{Re} = \frac{v \cdot d}{\nu}, \text{ and} \tag{7}$$

$$\lambda = \frac{0.3164}{\mathrm{Re}^{0.25}}, \tag{8}$$

where Re is the Reynolds number and λ is a coefficient of pipe friction loss approximated by a Blasius equation.

3.4 Pressure Equations

Pressure inside the pressure chambers due to the movement of the piston can be expressed using an equation of continuity that factors in fluid compressibility:

$$\frac{dP_i}{dt} = \frac{K}{V_i \pm \Delta V_i} \left(\sum_j Q_j - \sum_k Q_k \mp A \cdot v \right), \text{ where} \tag{9}$$

$$\Delta V_i = \int A \cdot v \, dt, \tag{10}$$

are Q_j and Q_k are respectively the inflow and outflow. Equation (9) applies to the pressures in the pressure chambers at both ends of the spool (P_{NA}, P_{NB}). In addition, the term ΔV_i in Equation (10) can be ignored for the orifice pressures at the hydrostatic bearings (P_{bA}, P_{bB}), since piston movement does not result in volume change.

3.5 Motion Equations

The equations of motion are as follows:

$$M \frac{d^2 x}{dt^2} = F_{SOL} - F_P - F_F - F_\mu - F_{sp}, \tag{11}$$

$$F_{SOL} = S(s) \cdot i, \tag{12}$$

$$F_P = \left(P_i - P_j\right) A_{SP}, \tag{13}$$

$$F_F = \rho \cdot Q \cdot v \cdot \cos(\theta), \tag{14}$$

$$F_\mu = (L_{bn} + L_{bT}) \frac{2\pi \cdot D_{SP} \cdot \mu}{\delta} \cdot \frac{dx}{dt}, \text{ and} \tag{15}$$

$$F_{SP} = K_{SP} \cdot \left(x - x_0\right), \tag{16}$$

where K_{SP} is the spring constant and x_0 is the initial length of the spring. The flow force (F_F) and viscous force (F_μ) are modelled in Equations (14) and (15) using generally known formulae.

Using the equations above, we investigated various control valve characteristics in the nonlinear region. Table 2 shows the differences between the basic equations for the linear model and those of the nonlinear model, which includes nonlinear terms. In these models, Reynolds numbers were calculated based on empirical measurements of flow rates in each component; in addition, flows through damping orifices were assumed to be turbulent, while flows through hydrostatic-bearing orifices were assumed to be laminar.

ITEM	Basic Equations of the Nonlinear Model	Basic Equations of the Linear Model
Orifice flow	$C_i \cdot A_i(x) \sqrt{\frac{2\left\|P_{i\_up} - P_{i\_down}\right\|}{\rho}}$, $A_i(x) = w \cdot x$	$C \cdot w \sqrt{\frac{2(P_S - P_0)}{\rho}} \Delta x + \frac{C \cdot w \cdot x_{S0}}{\sqrt{2\rho(P_S - P_0)}} \Delta P$
Pipe flow	$P_{i\_up} - P_{i\_down} = \frac{\rho \cdot \lambda \cdot l}{2d} v^2$, $\lambda = \frac{0.3164}{\mathrm{Re}^{0.25}}$	$\Delta P = \left(\lambda \frac{16\rho \cdot L_{NT}}{\pi^2 \cdot D_N^5} Q_0 \right) \Delta Q$, $\lambda = \frac{0.3164}{\mathrm{Re}^{0.25}}$
Inertia term	$M \frac{d^2 x}{dt^2}$	0
Fluid force term	$\rho \cdot Q \cdot v \cdot \cos(\theta)$	$C \cdot w \cdot (P_S - P_L) \cos(\theta) \cdot x$
Spring term	$K_{SP} \cdot (x - x_0)$	$K_{SP} \cdot x$

Table 2: Comparison of nonlinear and linear models.

4 Results and Discussion

4.1 Verification of Model Validity

First, the nonlinear model's validity was tested by comparing its results with those of the previously reported linear model and with experimental data. For the experiments, the input signal to the control valve was set at

50% ± 10%, and load resistance at ports A and B was set at 50% of the supply pressure, P_s = 14 × 10^6 Pa /3/. Figure 6 compares the theoretical results for the linear and nonlinear models along with the experimental results. These data are for an equivalent diameter ratio of C_r = 1.2 between the hydrostatic-bearing and damping orifices, a proportional gain of K_P = 1.9 for the compensation circuit, and an integral time of T_I = 0.03 s. The results indicate that the calculations of the nonlinear model largely agree with those of the linear model and with the results of the experimental data. Since the nonlinear model captured the valve's frequency characteristics in the linear region, we can conclude that the numerical model proposed above is valid.

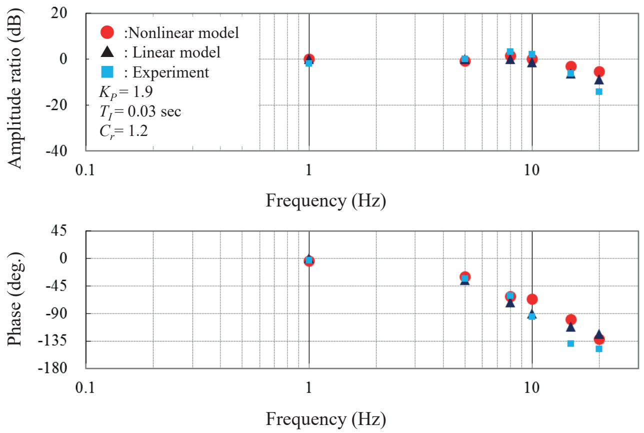

Figure 6: Comparison of results from analytical models and experiment (C_r = 1.2, K_P = 1.9, T_I = 0.03 s).

4.2 Effects of Nonlinearity for Sine-wave Input

Linear models are generally only applicable in the small region near the operating point; in most experiments, the control valve input is kept within a small amplitude range of around ±10% of the rated input. However, the effects of nonlinear terms could begin to manifest at larger inputs. The potential effects are discussed below. Figure 7 shows the response characteristics of the spool in the linear and nonlinear models to a sine-wave input of 1 Hz. Even though the input amplitude has been increased to ±40%, the linear model and nonlinear model match almost completely. Similarly, Figure 8 shows the results when the frequency is set to 10 Hz. With respect to the linear model, the nonlinear model has a smaller amplitude of approximately 5% and is phase-delayed by approximately 5%, but they tend to match overall. The lack of major differences between the linear model and the nonlinear model for sinusoidal input in the high-frequency range, up to ±40% of the rated input, above suggests that the linear model can be safely applied in this range. It is important to note here that using a simple numerical model can improve the expected influence of the design parameters. The findings above suggest that it is possible to use linear models to investigate control system designs in applied settings.

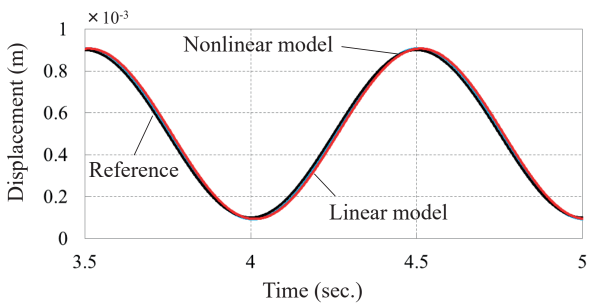

Figure 7: Comparison of linear and nonlinear models in frequency response

(Reference of 50% ± 40%, C_r = 1.2, K_P = 1.9, T_I = 0.03 s at 1 Hz).

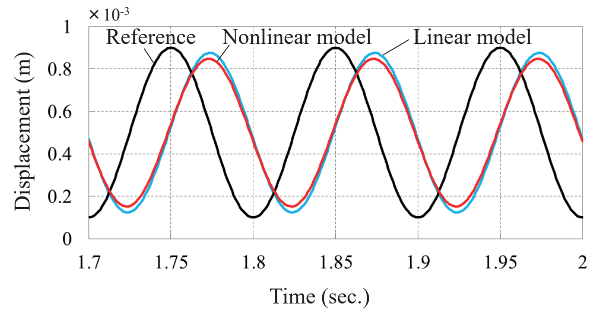

Figure 8: Comparison of linear and nonlinear models in frequency response

(Reference of 50% ± 40 %, C_r = 1.2, K_P = 1.9, T_I = 0.03 s at 10 Hz).

4.3 Effects of Non-linearity for Step Input

Figure 8 shows the response characteristics of the linear and nonlinear models to a step input of 0–50%. While the linear model overshoots the nonlinear model by approximately 8%, they are equal in terms of rise time (0.3 s) and ±1% settling time (0.15 s). The overshoot characteristics can be explained by the rapid movement of the spool during the rise phase, which increases the effects of pipe friction resistance in the damping orifices in the linear model. While the effects of nonlinearity are apparent in the overshoot characteristics, the two models do not differ in any other characteristics there is no difference between the two models. Thus, the results show that evaluations in the linear region can express the characteristics of the control valve well.

The findings above suggest that complex systems, which take into account an actuator's operating characteristics and fluid inertia within pipes, can be modelled more simply by using a linear model to approximate control valve characteristics in a nonlinear region, thus simplifying the design of control systems.

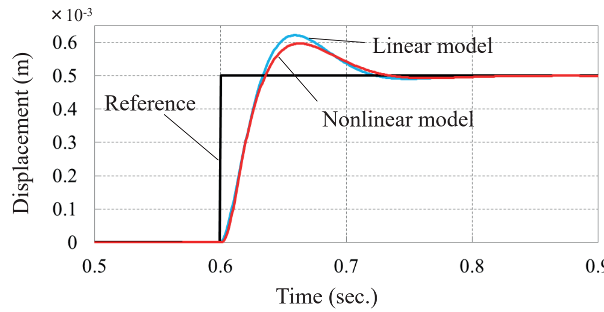

Figure 9: Comparison of linear and nonlinear models in step response characteristics

(C_r = 1.2, K_P = 1.9, T_I = 0.03 s).

5 Conclusion

In this report, the authors verified the characteristics of a water hydraulic proportional control valve in the nonlinear region by considering the potential effects of nonlinearity of the control apertures and damping orifices. The following conclusions were drawn:

- The linear model yields accurate frequency response characteristics over a wide range of input amplitudes (50% ± 40%).

- Some influence of nonlinearity was observed in the system's overshoot response to step input, but it was minor.
- Even a complex system, which take into account the actuator and piping systems, can be modelled more simply by using a linear model to approximate control valve characteristics in the nonlinear region, thus simplifying the design of control systems.

Nomenclature

Variable	*Description*	*Unit*
A_{SP}	Spool cross-sectional area	[m^2]
C	Flow constant	[-]
C_r	Equivalent diameter ratio of the damping orifice and hydrostatic bearing orifice, $C_r = D_N / D'_b$	[-]
D_b	Diameter of hydrostatic bearing orifice	[m]
D'_b	Equivalent diameter of hydrostatic bearing orifice	[m]
D_{SP}	Spool diameter	[m]
D_N	Damping orifice diameter	[m]
F_F	Flow force	[N]
F_{SOL}	Solenoid thrust	[N]
F_{SP}	Spring force	[N]
K	Bulk modulus	[Pa]
K_{SP}	Spring constant	[N/m]
K_{SOL}	Constant of solenoid thrust	[N/A]
L_{bn}, L_{bT}, L_{NT}	Annular clearance length	[m]
M	Mass of moving parts	[kg]
P_s	Supply pressure	[Pa]
P_A, P_B	Pressure in control ports	[Pa]
P_{bA}, P_{bB}	Pressure of hydraulic static bearing	[Pa]
P_L	Load pressure	[Pa]
P_{NA}, P_{NB}	Pressure in chambers at the ends of the spool	[Pa]
P_T	Pressure in return line	[Pa]
Q_{SA}, Q_{SB}, Q_{AT}, Q_{BT}, Q_{AB}	Control flow rate	[m^3/s]
Q_{SbA}, Q_{SbB}	Flow rate through hydro static bearing orifice	[m^3/s]
Q_{NTA}, Q_{NTB}	Flow rate through damping orifice	[m^3/s]
Q_{bnA}, Q_{bnB}, Q_{bTA}, Q_{bTB}	Clearance flow rate between spool and sleeve	[m^3/s]
Q_T	Flow rate in return line	[m^3/s]
T_L	Time constant	[s]
i	Current	[A]
v	Flow velocity	[m/s]
w	Width of control orifice	[m]
s	Laplace operator	[-]
x	Spool displacement	[m]
ζ	Damping coefficient	[-]
λ	Friction factor	[-]
θ	Jet angle	[degree]
δ	Radial clearance	[m]
μ	Viscosity	[Pa s]
ν	Kinetic viscosity	[m$_2$/s]
ρ	Working fluid density	[kg/m$_3$]
τ_{SOL}	Time constant	[s]
K_P	Proportional gain	[-]
T_I	Integral time	[s]

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