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Simulation on Hydraulic Control Characteristics of Regulator System in Bent-Axis Type Piston Pump

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Abstract: Variable displacement axial piston pumps are widely used for raising the energy level of the fluid in hydraulic systems. And the regulator is the device which regulates the discharge flow of the piston pump by controlling the swivel angle. The regulator receives the hydraulic pilot pressure and controls the pump output flow depending on the machine load and engine speed. This work deals with constant power control (horsepower control) in the design of a regulator by using a bent-axis type piston pump. In order to effectively use engine power, we must keep the horsepower from the engine to the pump constant. Therefore the regulator operates the constant power control. As a result, optimum power usage is obtained by accurately following the power hyperbola. This study focused on developing a simulation model of a regulator. First, the governing equations of the regulator are derived, and analysis is performed by computer simulation, which can identify significant parameters of regulator. As a result, the variation of the swivel angle, flowrate, hyperbolic curve, inner leakage and responsibility are simulated, and significant parameters of a regulator are identified.

Key words: regulator, swivel angle, bent-axis type piston pump, constant power control, power hyperbola, simulation, leakage, flowrate, control piston, spool

Introduction

A variable displacement bent-axis type axial piston pump which receives power from engine or motor and has the cylinder block tilted to a certain angle is rotating on the fixed spherical valve plate is the precise device which converts power into pressure forms of fluid. Currently, construction machineries require quick model changes, better functions and performances than traditional types. Therefore a piston pump has to be superior in the levels of reliability, controllability and responsibility. Also low noise level, and high efficiency have been key design points

Accordingly, as the regulator controls the swivel angel to adjust the flowrates according to the variation of discharge pressure in the system by load variation, it can improve the performance such as energy saving, and fine controllability of actuator. In order to use engine power effectively, the constant power control is a function of maintaining the pump input torque constant in combination with a constant input speed. While the operating-pressure is dependent on the variation of the load, the change in the swivel angle can vary the flowrates. That is, this controlling multiplies pressure and flowrate continuously, and compares the pressure with the set spring force.

The variation of displacement of bent axis type piston pump can be solved by swivel angle (g) shown in Eq. (1).

$$Q_{th} = \frac{\pi}{4} d_p^2 \cdot Z \cdot D_1 \sin \gamma \tag{1}$$

Consequently, the prompt variation of the swivel angle and operating condition is needed, if possible vibration must not exist. Therefore knowledge about the static and dynamic characteristics of pump is important in designing a precision fluid power system, but currently papers and design data of control system of the pump are scant in the literature.

This study builds up a mathematical model of constant power control regulator of bent axis type axial piston pump used in construction machinery. It also applies theoretical analysis by computer simulation before developing domestic test products.

Basic Structure and Operation

Fig. 1 is the hydraulic circuit of pump; the structure of regulator is schematically shown in Fig. 2. The constant power control regulator makes the products of discharge pressure (P) and discharge flowrate (Q) constant within the control limits. That is, it maintains power in order to be constant.

If the discharge pressure of the pump exceeds constant power curve of P-Q plot shown Fig. 3, the pressurized pilot piston connected with control rod acts as a control spring, port 1 and port 2 of the spool is opened, and then the working fluid from the pump goes into the control piston. Finally pump is

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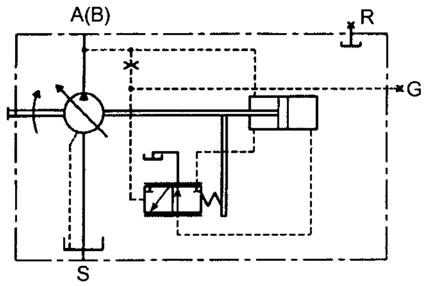

Fig. 1. The hydraulic circuit of the regulator.

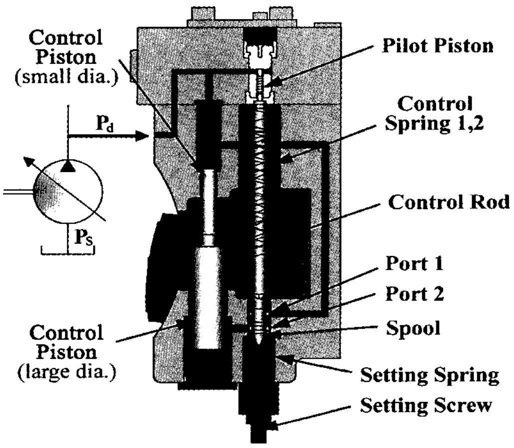

Fig. 2. The schematic diagram of the regulator.

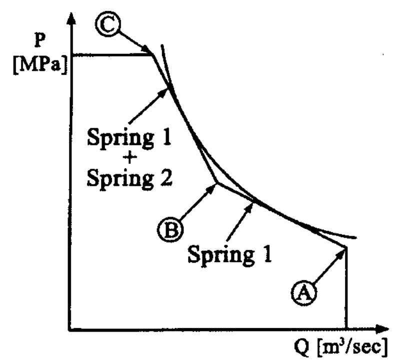

Fig. 3. Constant power curve.

automatically swiveled back to a smaller angle, as a result of pushing the control piston up against the forces of the springs. The swivel stroke is reduced until the resulting hydraulic power is once again equal to the given mechanical power. On the other side, if discharge pressure of pump is reduced, the pilot piston is pressurized and the port 2 is opened by control springs, and then the working fluid in the large diameter part of the control piston is drained to a reservoir. Fig. 3 shows that spring 1 in the range from (A) to (B) and spring 1,2 in the range from (B) to (C) is operated.

Mathematical Model

The characteristics of pressure, flowrate and behavior of regulator is decided by numerical parameters, a mathematical model related with these is solved as follows.

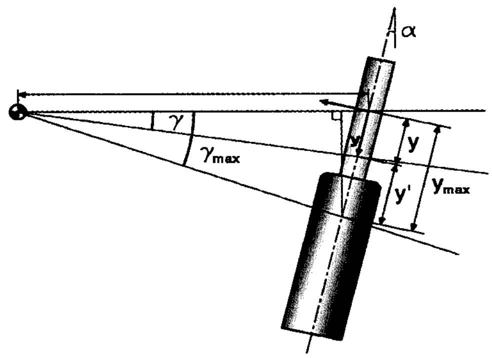

Fig. 4. The free body diagram of the control piston.

y - γ Equation

The Fig. 4 Shows the free body diagram of the control piston and the equation of y can be translated into the equation of γ.

$$y = \frac{L\sin\gamma_{max}}{\cos(\alpha-\gamma_{max})} - \frac{L\sin\gamma}{\cos(\alpha-\gamma)} \tag{2}$$

so that, the first and second derivatives are

$$\dot{y} = -\frac{L\cos\alpha}{\cos^2(\alpha-\gamma)}\dot{\gamma}$$

$$\ddot{y} = -\frac{L\cos\alpha}{\cos^2(\alpha-\gamma)}\ddot{\gamma} + \frac{2L\cos\alpha\cdot\sin(\alpha-\gamma)}{\cos^3(\alpha-\gamma)}\dot{\gamma}^2$$

The Equation of Motion of Control Piston

It is important in respect of responsibility how to slide between the valve plate and the regulator. Then, for solving friction torque (T) between two parts, Eq. (3) with the experimental value of friction coefficient (μ_s) is considered as follows. The effect of tangential force of fluid is ignored.

$$T^* = \mu_s L_v\left[\frac{zAP_d}{2}\right]SIGN(-\dot{\gamma}) \tag{3}$$

Where μ_s is the friction coefficient of sliding surfaces.

The equation of motion of control piston is represented in

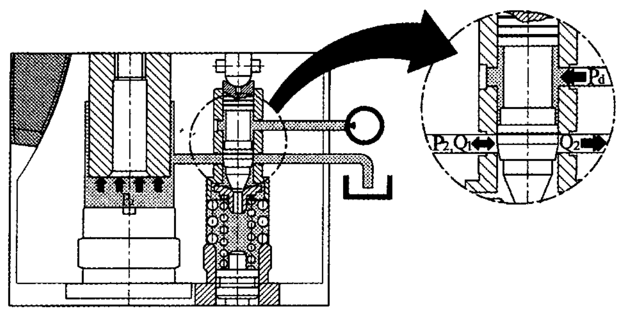

Fig. 5. The schematic diagram of the spool.

consideration of control spring stiffness, damping and the equivalent moment of Inertia as follows:

$$[A_{cpl} \cdot P_2 - A_{cps} \cdot P_d]L = \tag{4}$$

$$L[M_{cp}\ddot{y} + C_{cp}\dot{y} + k_1(y + \delta_1) + k_2(y - \delta) \cdot u(y - \delta)] + I_e\ddot{\gamma} + T^*$$

Where Eq. (4) will be shown as follows:

$$\left[I_e + m_{cp}L^2\frac{\cos\alpha}{\cos^2(\alpha - y)}\right]\ddot{\gamma} + \left[C_{cp}L^2\frac{\cos\alpha}{\cos^2(\alpha - \gamma)}\right]\dot{\gamma}$$

$$= 2m_{cp}L^2\left[\frac{\cos\alpha \cdot \sin(\alpha - \gamma)}{\cos^3(\alpha - \gamma)}\right]\dot{\gamma}^2 - [P_2A_{cpl} - P_dA_{cps}]$$

$$+k_1\left[\left(\frac{L\sin\gamma_{max}}{\cos(\alpha - \gamma_{max})}\right) - \frac{L\sin\gamma}{\cos(\alpha - \gamma)} + \delta_1\right] \tag{5}$$

$$+k_2\left[\left(\frac{L\sin\gamma_{max}}{\cos(\alpha - \gamma_{max})}\right) - \frac{L\sin\gamma}{\cos(\alpha - \gamma)} + \delta_1\right]u(\gamma - \delta) + T^*$$

In the case considering all the nonlinear terms included in Eq. (5), the calculation time increases. Therefore, the linearization about a part of terms is applied.

$$B_1 = \frac{\sin\gamma_{max}}{\cos(\alpha - \gamma_{max})} - \frac{\sin\gamma}{\cos(\alpha - \gamma)} = -\gamma + 0.4014$$

$$B_2 = \frac{\cos\alpha \cdot \sin(\alpha - \gamma)}{\cos^3(\alpha - \gamma)} = -1.04\gamma + 0.17$$

$$B_3 = \frac{\cos\alpha}{\cos^2(\alpha - \lambda)} \approx 1$$

The final equation of motion of control piston is as follows:

$$\ddot{\gamma} = \frac{1}{I_e + m_{cp}L^2}\begin{bmatrix} -C_{cp}L^2\dot{\gamma} - L^2(k_1 + k_1u(\gamma - \delta))\gamma \\ +2m_{cp}L^2(-1.04\gamma + 0.17) + k_1L(0.04L + \delta_1) \\ +k_2(0.04L - \delta)u(\gamma - \delta) - (P_2A_{cpl} - P_dA_{cps}) + T^* \end{bmatrix} \tag{6}$$

The Equation of Motion of Pilot Piston

As the inflow of working fluid toward the large diameter chamber is diverted by displacement of the pilot piston, the behavior of the pilot piston is the influential parameter on response. The equation of motion of the pilot piston will be shown as formula Eq. (7).

$$\ddot{x} = \frac{1}{m_{pi}}\begin{bmatrix} -C_{pi}\dot{x} - k_3x + k_3\delta_3 - k_1(y + \delta_1) \\ +k_2(y - \delta)u(y - \delta) + P_oA_p \end{bmatrix} \tag{7}$$

The Cross Sectional Area of Spool

The flow coming into the large diameter chamber of the regulator passing through the spool is not proportional to movement of spool, because the shape of the port is circular.

Therefore, for the calculation of flowrate and pressure in the chamber, the variation of the opening area of the port is

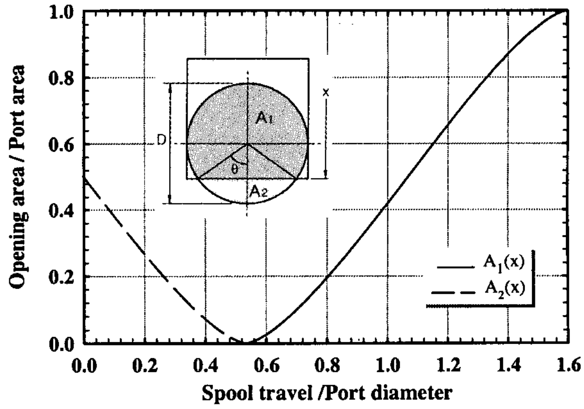

Fig. 6. The variation of cross sectional area.

considered as a non-dimensional parameter as follows in Fig.6. The datum point is 1.214mm under center point of the circular port.

The Flow Equations And The Continuity Equations

Each of the algebraic equations about flowrate (Q_1,Q_2) passing through the opening area(A_1, A_2) are assumed to be turbulent and are expressed as follows:

$$Q_1 = KA_1(x)\sqrt{\frac{2|P_d - P_2|}{\rho}}SIGN(P_d - P_2) \tag{8}$$

$$Q_2 = KA_2(x)\sqrt{\frac{2|P_2 - P_s|}{\rho}}SIGN(P_2 - P_s) \tag{9}$$

Also, the flowrate passing through the orifice of the pilot piston is as follows:

$$Q_0 = KA_0\sqrt{\frac{2|P_d - P_s|}{\rho}}SIGN(P_d - P_s) \tag{10}$$

In order to find P_2, P_o and Q_a,

$$P_{new} \cdot V_{new} = P_{old} \cdot V_{old} \tag{11}$$

The variation of inner control volume of spool and pilot piston and the flowrates of inlet and outlet are considered as follows:

$$Q_a = Q_{th} - Q_1 - \frac{V_1}{K_f}\dot{P}_d - L_1(P_d - P_s) \tag{12}$$

$$P_2 = \frac{K_f}{V_2}\int(Q_1 - Q_2 - A_{cpl} \cdot \dot{y} - L_2(P_2 - P_s))dt \tag{13}$$

$$P_0 = \frac{K_f}{V_p}\int(Q_o - A_p \cdot \dot{x} - L_p(P_o - P_s))dt \tag{14}$$

Then, assuming that leakage is laminar flow, leakage

$$Ql_{ab} = \frac{\pi DC^3}{12\mu l}(P_a - P_b) \tag{15}$$

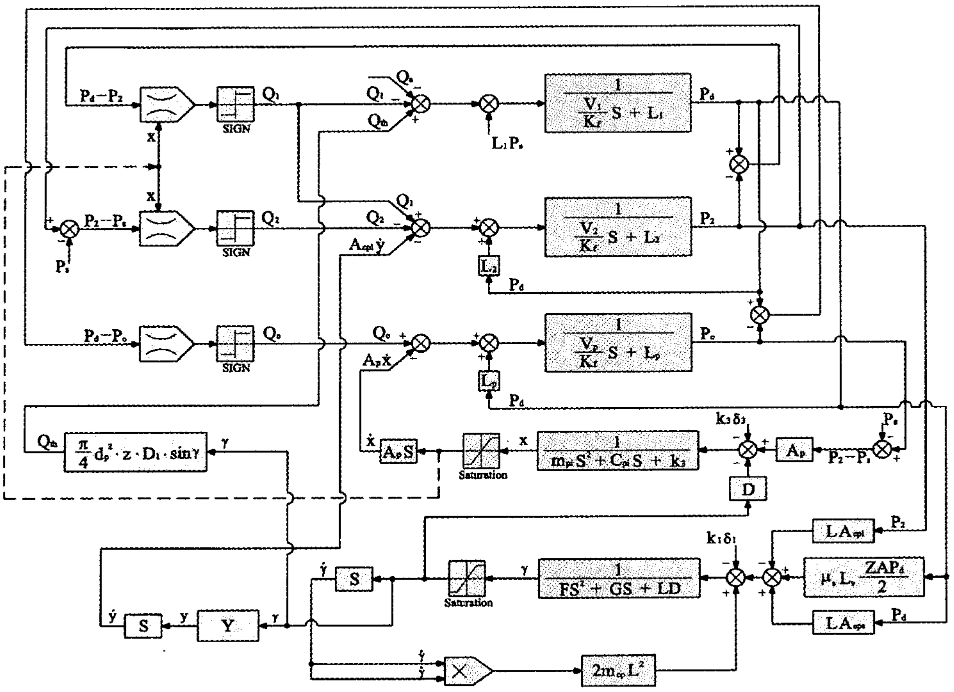

Fig. 7. Block Diagram for simulation.

Simulation

The pulsation which been closely related with air mixed in pipeline is ignored in the simulation. The state variables of the dynamic system are displacement, velocity, and acceleration. Each equation of motion is composed of state equation

The differential equations are computed with time step Δt by Rung-Kutta 5th. The Fig. 7 shows the block diagram of the fluid flow circuit of pump. The operating conditions for simulation are shown as Table 1.

The dynamic behavior of the regulator is presented in the form of a response function to ramp input and step input. The inputs mean the discharge pressure of pump. (Relief pressure 34 MPa, 1.0 Hz). Fig. 8 shows delivery flowrate of the pump. If the discharge pressure exceeds the set value of the setting screw, discharged working fluid passes through port 1 of the spool by the pilot piston and flows in large diameter chamber of the control piston. As a result the pump is automatically swiveled back to a smaller angle and then the flowrate decreases. On the other side, in the case of discharge pressure decrease, the force of the control springs and setting spring is large than one of the pilot piston, then the spool moves downward. Consequently the working fluid of the large chamber is drained to a reservoir and the swivel angle increases. It is noted that the variation of delivery flowrate shows two slopes by two stiffnesses of control spring, and constant power control as Fig. 8.

When the input is the discharge pressure, Fig. 9 shows the variation of the swivel angle. The behavior of spool converges on a neutral point(1.6 mm) within 0.15 sec.

In the case of the hydraulic system related to dynamic characteristics or high-pressure fluid, it is necessary that the working fluid deal with the compressible fluid. The Fig. 10

Table. 1 Specification of regulator.

Working Fluid	VG32	
Temperature	55°C	
Revolution	2000 rpm	
Spring Constant (N/m)	Spring 1	$5.60\ 310^{3}$
	Spring 2	$1.14\ 310^{4}$
	Spring 3	$1.17\ 310^{5}$
Swivel Angle	Max 23° ↔ Min 7°	
Suction Pressure	0.1 MPa	
Moment of Inertia	0.346 N · m^2	
Coef. of Discharge	0.63	

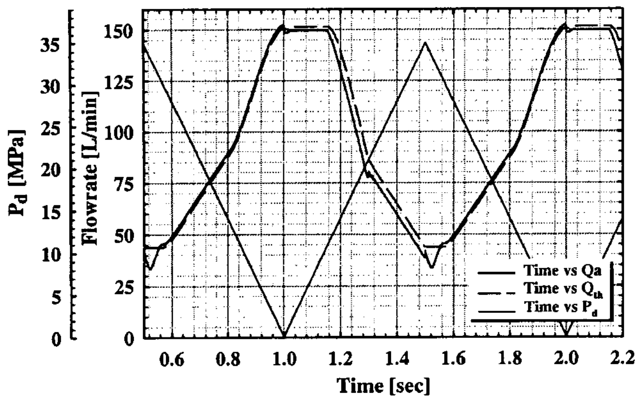

Fig. 8. The delivery flowrate of the pump (ramp).

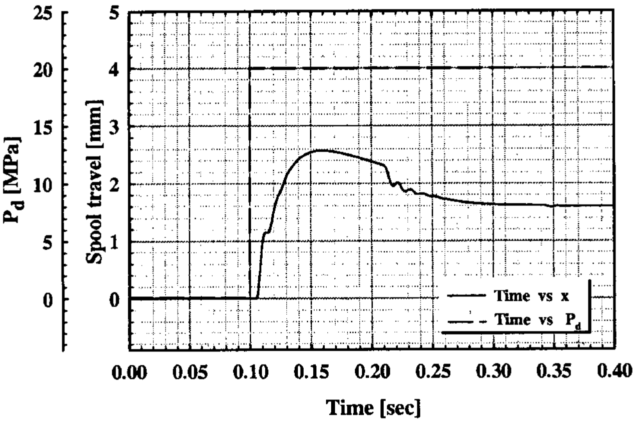

Fig. 9. The variation of spool travel with time (step).

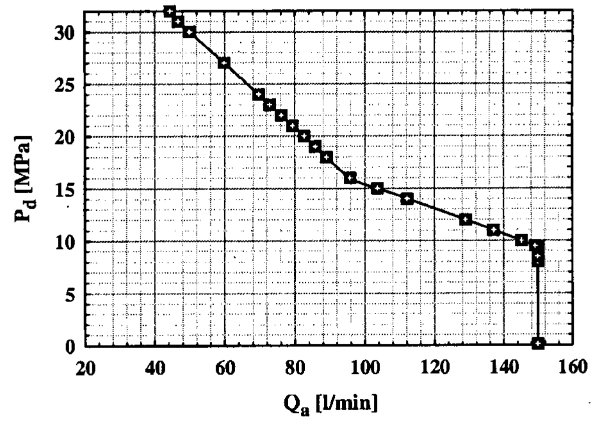

Fig. 11. The Hyperbolic curve of the regulator.

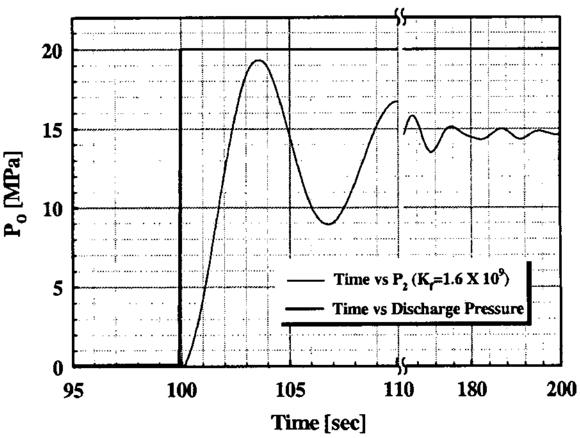

Fig. 10. The effect of bulk modulus (step).

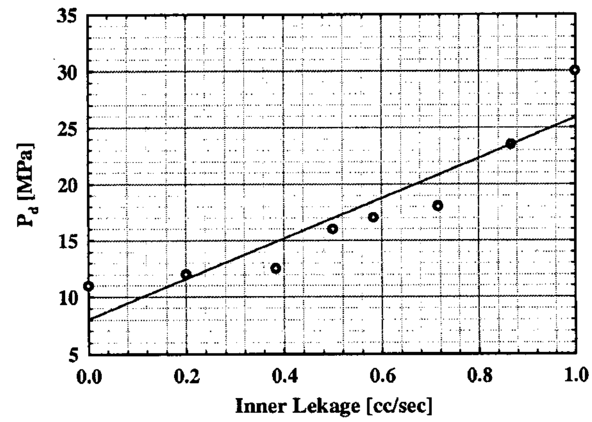

Fig. 12. The inner leakage of regulator.

shows high responsibility of the regulator.

It is important that the static characteristics as well as dynamic one of the regulator are understood. Therefore the step input of a certain pressure is applied. Then the time taken to get to the steady state is taken about 0.15 sec and by using the flowrate of the steady state, the power hyperbola of the simulated regulator is shown in Fig. 11. Approximately in 10MPa the regulator starts running, while in 15 MPa both spring 1 and spring 2 are running.

In the side of responsibility of the constant power control the over-torque is generated by the operation leakage of the control device. Finally a larger flywheel is needed for preventing a drop in the engine revolution, therefore on the side of energy saving and cost it makes a problem. Actually it is difficult to measure inner leakage of the regulator. The inner leakage of regulator computed at this simulation is shown in Fig. 12.

Conclusions

This paper has considered a simulation of a variable displacement bent-axis type piston pump subjected to hydraulic characteristics. The conclusions from the modeling and simulation of this paper can being stated as follows.

In the case that discharged pressure being given as ramp input, due to the stiffness of the control piston springs, the variation of the swivel angle shows the two slopes and we find that they are important parameters.

Generally, the settling time of regulator is taken about 0.15sec against reference input and the amount of leakage of the system is found, the higher the discharge pressure is, the more leakage is remarkably generated. This affects response, energy saving, and control performance of the system.

The modeling and simulation procedures given in this paper will provide reasonable basis for a rational design of the others as well as the constant power control of the piston pump.

Acknowledgments

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Nomenclature

P_s	Suction pressure [MPa]
P_d	Pump delivery pressure [MPa]
P_o	Orifice pressure acting on pilot piston [MPa]
P_2	Pressure of the small diameter part of the control piston [MPa]
δ	Interval between spring 1 and spring 2 [m]
δ_1	Initial deflection of spring 1 [m]
δ_2	Initial deflection of spring 2 [m]
$\gamma, \dot{\gamma}, \ddot{\gamma}$	Swivel angle, angular velocity and acceleration
$\gamma_{max}, \gamma_{min}$	Max. and Min. of the swivel angle [rad]
α	Regulator angle of inclination [rad]
$x, \dot{x}, \ddot{x}$	Travel, velocity, acceleration of the spool
$y, \dot{y}, \ddot{y}$	Travel, velocity, acceleration of the control piston
A_{cps}	Area of the small diameter part [m^2]
A_{cpl}	Area of the large diameter part [m^2]
A_1	Area between chamber 1 and chamber 2 [m^2]
A_2	Area between chamber 2 and reservoir [m^2]
Q_a	Actual delivery flowrate[m^3/s]
L_1, L_2	Leakage coefficient of the spool [$m^5/N \cdot s$]
L_p	Leakage coefficient of the pilot piston [$m^5/N \cdot s$]
K	Discharge Coefficient
V_1, V_2	Volume of chamber 1 and chamber 2 [m^3]
K_f	Fluid bulk modulus [MPa]
m	Mass [N]
k	Stiffness of the spring [N/m]
C	Coefficient of viscosity resistance [$N \cdot s/m$]
D	Diameter of the piston
μ	Viscosity of working fluid [$N \cdot s/m^2$]
ρ	Density of working fluid [$N \cdot s/m^4$]
L	Distance between center piston spherical joint and control piston [m]
L_v	Distance between center piston [m]
Z	Number of the piston of axial piston pump
G	$= C_{cp} + L^2$
F	$= I_e + m_{cp}$
$D=$	$k_1 y + k_{2y}(y-\delta)u(y-\delta)$

- Subscripts -

cp	Control piston
pi	pilot piston