Modeling and Dynamic Characteristic Research of Hydraulic On/Off Valve

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Abstract

Power losses of a novel high speed hydraulic on/off valve including four transitions during its switching between on and off states are analyzed and modeled using mathematical equations related to the geometric parameters of the on/off valve and hydraulic system parameters. The dynamic response of the on/off valve is investigated by using feedback of axial position of the spool of valve which is driven axially by a gerotor pump. A fuzzy logic parameter self-tuning PID controller is used to overcome the system nonlinearity and simulation result shows that the overshoot is decreased greatly than conventional PID controller.

Keywords: on/off valve, transition, power losses, PID Controller, axial displacement

1. Introduction

Hydraulic systems are widely used in many applications such as mobile equipment and injection machine because of their vast driving ability and high dynamic response characteristic, many of them use throttling valve as flow control unit to adjust speed of the actuator and are low in efficiency. On the other hand, an on/off valve controlled hydraulic system works only in on or off state, which makes it achieve high efficiency and fast dynamic response possible. This paper investigates a novel rotary high speed on/off valve in which the spool of valve turns along one direction at nearly a constant revolution speed and the valve gets on/off cycles. The duty cycle of the on/off valve can be modulated by moving the spool axially through a gerotor pump which is driven by a BLDC (Brushless Direct Current) motor. Compared with conventional linear displacement on/off valves, the novel rotary on/off valve can pass much more flow rate because the revolution of spool is driven by the input hydraulic pressure, not a driving unit such as electromagnetic driver [2], [3].

2. Novel Self-Spinning High Speed Hydraulic On/Off Valve

Figure 1 shows the sketch of the new self-spinning high speed hydraulic 3-way/2-position on/off valve [1], and Figure 2 gives the sketch of the spool of on/off valve and its expansion along circumference. The spool turns only in one direction by the momentum of fluid itself. It consists of a central PWM section and two outlet stages. The central PWM section contains alternating spiral lands overlayed onto the surface of the spool which partition the spool into regions where flow is leaded to load or tank. When the spool turns, the inlet nozzles transition across the barriers and alternate the flow path between two outlet stages, which results in the “on” action and the “off” action. The duty cycle of the on/off valve can be modulated by moving the spool axially through a gerotor pump which is driven by a BLDC (Brushless Direct Current) motor. Compared with conventional linear displacement on/off valves, the novel rotary on/off valve can pass much more flow rate because the revolution of spool is driven by the input hydraulic pressure, not a driving unit such as electromagnetic driver [2], [3].
3. Analysis of Transitions and Throttling Losses of the On/Off Valve

The hydraulic system consisting of the new on/off valve and other components is shown in Figure 3. According to the geometry of valve, there are $N$ (number of pairs of spiral barrier)PWM cycles in a revolution. In one PWM cycle, for the rhombus orifices (nozzles), there are one full open duration to load, one full open duration to tank, two transition durations to load, and two transition durations to tank.

$$Q_{\text{full\_open}} = \Delta p_{\text{full\_open}} Q,$$  \hspace{1cm} (1)
where \( Q \) is fixed displacement pump flow, \( \Delta p_{\text{fullopen}} \) is pressure drop across the rhombus orifice. According to the orifice equation of fluid, \( \Delta p_{\text{fullopen}} \) can be described as [3][4]:

\[
\Delta p_{\text{fullopen}} = \frac{\rho}{2} \left( \frac{Q}{c_d A_{\text{fullopen}}^\eta} \right)^2,
\]

where \( \rho \) is the density of hydraulic oil, \( c_d \) is the orifice discharge coefficient, and \( A_{\text{fullopen}} = 0.5 \cdot R_w \cdot R_h \) is orifice area, \( R_w \) and \( R_h \) are width and height of orifice respectively. According to (2), the power loss for full open duration to tank is the same as for full open duration to load.

![Hydraulic system controlled by a rotary on/off valve](image)

The two transitions to tank include transition during that the orifice opening area to tank is gradually increasing from zero to maximum area (tr1 in Figure 2) and transition during that the orifice opening area to tank is gradually decreasing from maximum area to zero (tr2 in Figure 2). According to Figure 2, The orifice area \( A(\theta) \) is proportional to the rotary angle \( \theta \),

\[
A(\theta) = 0.5 \cdot R \cdot R_h \cdot \theta,
\]

When the orifice opening area to tank increases, The relief valve tends to close. The pump flow \( Q \) is divided into flow \( Q - Q_1 \) which flows to tank via the relief valve and flow \( Q_1 \) which flows to tank via the on/off valve. The rotary angle \( \theta_r \) is the critical angle at which the pressure drop across the relief valve and the on/off valve is still relief valve opening pressure \( P_r \) and it can be calculated according to the following equation:

\[
P_r = \frac{\rho}{2} \left( \frac{Q}{c_d A(\theta_r) N} \right)^2,
\]

From (2), (3), and (4), we obtain:

\[
\theta_r = \frac{R_w}{R} \sqrt{\frac{\Delta p_{\text{fullopen}}}{P_r}},
\]

From (3), The total rotary angle for each transition is:

\[
\theta_{\text{full}} = \frac{A_{\text{fullopen}}}{0.5 \cdot R \cdot R_h} = \frac{R_w}{R},
\]
The energy loss of this transition from rotary angle zero to \( \theta_r \) is calculated as:

\[
E_1 = \int_{t=0}^{t=\theta_r} P_r \cdot (Q - Q_t) \cdot dt + \int_{t=0}^{t=\theta_r} P_r \cdot Q_1 \cdot dt = \int_{\theta=0}^{\theta=\theta_r} P_r \cdot Q \cdot \frac{d\theta}{\omega} + \int_{\theta=0}^{\theta=\theta_r} P_r \cdot Q \cdot \frac{d\theta}{\omega}\\
= \frac{Q \cdot R_v}{\omega \cdot R} \cdot \sqrt{\Delta p_{\text{fullopen}}} \cdot P_r.
\]  

(7)

The energy loss from rotary angle \( \theta_r \) to \( \theta_{full} \) is calculated as:

\[
E_2 = \int_{t=t_{str}}^{t=\text{full}} \Delta P_{\text{tank}} \cdot Q \cdot dt = \int_{\theta=\theta_r}^{\theta=\theta_{full}} \frac{P}{2} \left( \frac{Q}{C_d \cdot N \cdot 0.5 \cdot R \cdot R_h \cdot \theta} \right)^2 \cdot Q \cdot \frac{d\theta}{\omega} + \int_{\theta=\theta_r}^{\theta=\theta_{full}} \frac{P}{2} \left( \frac{Q}{C_d \cdot N \cdot 0.5 \cdot R \cdot R_h \cdot \theta} \right)^2 \cdot Q \cdot \frac{d\theta}{\omega}\\
= \frac{Q \cdot R_v}{\omega \cdot R} \cdot \sqrt{\Delta p_{\text{fullopen}}} \cdot \left( \sqrt{P_r} - \sqrt{\Delta p_{\text{fullopen}}} \right).
\]  

(8)

Therefore, the total energy loss of the transition duration to tank from rotary angle zero to rotary angle \( \theta_{full} \) is:

\[
E_3 = E_1 + E_2 = \frac{Q \cdot R_v}{\omega \cdot R} \cdot \sqrt{\Delta p_{\text{fullopen}}} \cdot \left( 2 \cdot \sqrt{P_r} - \sqrt{\Delta p_{\text{fullopen}}} \right).
\]  

(9)

When the orifice opening area to tank decreases, the relief valve tends to open. According to (7) and (8), the energy losses of this transition to tank from full open area \( \mathcal{A}(\theta_{full}) \) to opening area \( \mathcal{A}(\theta_r) \) is equal to \( E_2 \), and from \( \mathcal{A}(\theta_r) \) to zero is equal to \( E_1 \).

The two transitions to load also include transition during that the orifice opening area to load is gradually increasing from zero to maximum \( \text{tr3 in Figure 2} \) and transition during that the orifice opening area to load is gradually decreasing from maximum to zero \( \text{tr4 in Figure 2} \).

When the orifice opening area to load increases, the relief valve tends to close. The pump flow \( \dot{Q} \) is divided into flow \( \dot{Q} - \dot{Q}_L \), which flows to tank via the relief valve, and flow \( \dot{Q}_L \), which flows to application via the on/off valve. The rotary angle \( \theta_{ra} \) is the critical angle at which the pressure drops across the relief valve is \( P_r \) and across the on/off valve is \( P_r - P_L \) respectively and it can be calculated according the following equation:

\[
P_r - P_L = \frac{P}{2} \left( \frac{Q}{C_d \cdot \mathcal{A}(\theta_{ra}) \cdot N} \right)^2,
\]  

(10)

According to (2), (3) and (10), we obtain:

\[
\theta_{ra} = \frac{R_v}{R} \sqrt{\Delta p_{\text{fullopen}}} \cdot \sqrt{P_r - P_L},
\]  

(11)

The energy loss of this transition to load from rotary angle zero to \( \theta_{ra} \) is calculated as:
The energy loss of this transition to load from rotary angle \( \theta_{ra} \) to \( \theta_{full} \) is calculated as:

\[
E_5 = \int_{\theta_{ra}}^{\theta_{full}} \left( \frac{\Delta P_{fullopen}}{\omega} \right) \cdot \left( P_r - P_L \right) \cdot \frac{d\theta}{\omega}.
\]

Therefore, the total energy loss of the transition to load from rotary angle zero to rotary angle \( \theta_{full} \) is:

\[
E_6 = E_4 + E_5 = \frac{Q \cdot R_w}{\omega \cdot R} \cdot \left( \sqrt{\Delta P_{fullopen}} \cdot \left( \frac{P_r - P_L}{2} \right) + \sqrt{\Delta P_{fullopen}} \cdot \left( \frac{P_r - P_L}{2} - \Delta P_{fullopen} \right) \right),
\]

When the orifice opening area to load decreases, the relief valve tends to open. According to (12) and (13), the energy loss of this transition duration from full open area \( A(\theta_{full}) \) to opening area zero is the same as \( E_6 \).

So, in one complete PWM cycle the transition losses both to load and to tank are as follows:

\[
E_{tran} = 2 \cdot E_5 + 2 \cdot E_6,
\]

And the transition energy losses plus the full open energy losses in one PWM cycle is as follows:

\[
E_{pwm} = \frac{E_{tran}}{t_{transition}} + \frac{P_{fullopen} \cdot (t_{fullopen坦} + t_{fullopen鸟})}{t_{fullopen}} = 2 \cdot \frac{Q \cdot R_w}{\omega \cdot R} \sqrt{\Delta P_{fullopen}} \cdot \left( \frac{P_r - P_L}{2} + \sqrt{\Delta P_{fullopen} \cdot \left( \frac{P_r - P_L}{2} - \Delta P_{fullopen} \right)} \right) + 2 \cdot \frac{Q \cdot R_w}{\omega \cdot R} \sqrt{\Delta P_{fullopen}} \cdot \left( 2 \cdot \sqrt{\Delta P_{fullopen}} + Q \cdot \frac{2\pi}{N} \cdot \frac{A}{\omega} \right),
\]

Among the four transitions of one PWM cycle, the inlet pipe endures pressure increases in transition 2 and transition 4 (t2 and t4 in Figure 2). In transition 2, the pressure increases from \( \Delta P_{fullopen} \) to \( P_r \). In transition 4, the pressure increases from \( \Delta P_{fullopen} + P_L \) to \( P_r \). The increase of pressure in inlet pipe will make the fluid compress and will generate energy loss. Fluid compressibility equation is as follows [4],[5]:

\[
\]
\[ \frac{1}{\beta} = - \frac{dv}{v} \cdot \frac{1}{dp}, \]  

(17)

Where \( \beta \) is fluid bulk modulus and is considered as constant, \( v \) is the initial volume of the inlet pipe.

Energy loss caused by fluid compressibility is calculated as:

\[ E_{\text{comp}} = \left( -\int p \cdot dv \right) = \left( \int_{p_{\text{transition}}}^{p_L} p \cdot v dp / \beta + \int_{p_{\text{transition}} + p_L}^{p_r} p \cdot v dp / \beta \right), \]

\[ = \frac{v}{2\beta} \left( 2 \cdot P_r^2 - \Delta P_{\text{fullopen}}^2 - \left( P_L + \Delta P_{\text{fullopen}} \right)^2 \right), \]  

(18)

The total power losses including transition losses, full open losses and fluid compressibility losses are as follows:

\[ P_{\text{power loss}} = E_{\text{pen}} + E_{\text{comp}} = \frac{N \cdot Q \cdot R_e}{\pi \cdot \omega} - \frac{\Delta P_{\text{fullopen}} \cdot \left( P_r - P_L / 2 \right) \sqrt{\Delta P_{\text{fullopen}} \cdot \left( P_r - P_L \right)} + 2 \left( \frac{\Delta P_{\text{fullopen}}}{P_r} \right)}{\pi \cdot \Delta \beta}, \]

(19)

From equation (19), it can be seen that the total power loss is proportional to system flow rate \( Q \) and spool frequency \( \omega \), and also related to geometric parameters of the on/off valve, full open losses \( \Delta P_{\text{fullopen}} \), load pressure \( P_L \) and relief pressure \( P_r \). The initial inlet pipe volume \( v \) should be decreased to mitigate loss caused by compressibility of fluid.

4. Axial Actuating Model of the On/Off Valve

Supposing that fluid in the gerotor pump circuit is incompressible, then the axial motion of the valve spool can be modeled as follows:

\[ \dot{x} = \frac{Q(u)}{A_s}, \]

(20)

Where, \( x \) is the axial displacement of the valve spool, \( Q(u) \) is the output flow rate of the gerotor pump, \( u \) is the input to the brushless direct current motor control circuit, \( A_s \) is the area of one end of the spool.

A gerotor pump consists of an inner rotor and an outer rotor, and there are \( N \) teeth with the inner rotor and \( N+1 \) teeth with the outer rotor. When the rotors rotate, they act as in the piston within a hydraulic cylinder. When one tooth enters the space of another tooth, fluid is squeezed out of the gerotor pump. Therefore, a continuous flow rate is obtained at the outport of the gerotor pump, and that is proportional to revolution speed of gerotor pump.

Neglecting leakage of the gerotor pump, the approximate displacement per revolution is calculated as follows [6]:

\[ q = BZ \int_{\phi_i}^{\phi_f} \left( \frac{ds}{d\phi} \right) d\phi, \]  

(21)

\[ \phi_i = 0 \]

\[ \phi_f = \frac{2\pi}{N} \]
Where, \( q \) is displacement per revolution, \( B \) is width of the rotors, \( \frac{ds}{d\phi} \) is the change rate of the area surrounded by the inner rotor and the outer rotor, \( Z_1 \) is the number of teeth of the inner rotor, \( \phi \) is the rotary angle of the inner rotor.

The averaging flow rate of the gerotor pump is defined as follows:

\[
Q = qn.
\]  

Where, \( Q \) is theoretical average flow rate, \( n \) is revolution speed.

5. Fuzzy-Logic Parameter Self-Tuning PID Controller

It has proved that closed loop control system of the axial position of spool has less response time than open loop control system, and a not significant rise of ripple of output pressure happens [2]. Because actual flow rate and pulsation of flow rate of the gerotor pump can be influenced by fluid pressure, fluid density, and tooth clearance and so on, control system of axial position of the valve spool is a complicated nonlinear system with time delay. A fuzzy logic model with ability to handle nonlinearity is used to adjust the parameters of the PID controller in the axial position control system [7][8].

The control law of a discrete PID controller is described as follows:

\[
u(k) = u(k - 1) + K_p \left( e(k) - e(k - 1) + \frac{T_i}{T_o} e(k - 1) + \frac{T_d}{T_o} [e(k) - 2e(k - 1) + e(k - 2)] \right),
\]

Where, \( e(k) \) is error, \( K_p \) is proportional coefficient, \( T_o \) is sampling time, \( T_i \) is integral time constant, \( T_d \) is differential time constant, \( u(k) \) is control output at time \( k \).

A fuzzy logic model with inputs error \( e \) and the change rate of error \( ec \) infers real timely the PID parameters \( K_p, T_i, \) and \( T_d \) from a set of fuzzy logic rules which are obtained from expert knowledge and experience. A schematic of control system of the valve spool axial position using fuzzy logic parameter self-tuning PID control algorithm is shown in Figure 4.

According to the range of axial displacement of the spool, the fuzzy grades of inputs error \( e \), the change rate of error \( ec \), and outputs \( K_p, T_i, T_d \) are set to 5, 9 respectively. The fuzzification factors of error \( e \) and the change rate of error \( ec \) are determined by \( K_e = n/\max\text{ }e, K_{ec} = m/\max\text{ }ec \) respectively, where \( m \) and \( n \) are fuzzy grade of \( ec \) and

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respectively. The fuzzification factors affect greatly the dynamic performance of the control system. The bigger the fuzzification factors of error $e$ is, the bigger the overshoot is. The bigger that of the change rate of error $e_c$ is, the smaller the overshoot is, but the response time is bigger. The membership functions are described in Figure 5 and Figure 6, and the fuzzy control rule is shown in Table 1.

![Figure 5. Triangle membership of error $e$, the change rate of error $e_c$](image)

![Figure 6. Triangle membership of $K_p$, $T_i$, $T_d$](image)

### Table 1. Fuzzy control rule

<table>
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<th>$e_c$</th>
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<th>$P2$</th>
<th>$P3$</th>
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<td>$N1/P2/K1$</td>
<td>$N1/P2/ZR$</td>
<td>$N2/P3/ZR$</td>
<td>$N2/P3/ZR$</td>
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<tr>
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<td>$N1/P2/ZR$</td>
<td>$N1/P2/ZR$</td>
<td>$N2/P3/ZR$</td>
<td>$N2/P3/ZR$</td>
<td>$N2/P3/ZR$</td>
</tr>
<tr>
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<td>$N1/P2/P2$</td>
<td>$N2/P3/F3$</td>
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<tr>
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<td>$N4/P4/P4$</td>
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</table>
6. Simulation of Control System of the Axial Position of the Spool

The mathematical model of the control object obtained is described in transfer function as \( G(s) = \frac{10e^{-0.02s}}{2s^2 + 3s + 1} \). Let \( K_c = 0.9 \), \( K_{ec} = 0.05 \), defuzzification factors \( K_p = 5 \), \( K_i = 1.2 \), \( K_d = 0.01 \), initial value of \( K_p, T_i, T_d = 5, 3, 1 \), and sampling time \( T_0 = 0.01 \) s. The control system is simulated using conventional PID controller and the fuzzy logic parameter self-tuning PID controller respectively under MATLAB/Simulink. The result using a step current input is shown in Figure 7. It can be seen that the overshoot is decreased significantly by using the fuzzy logic parameter self-tuning PID controller [9],[10].

![Figure 7. Response curve of the control system to a step input](image)

7. Conclusions

The power loss of a novel rotary on/off valve controlled hydraulic system is deduced in the form of mathematical equations which are suitable for design purpose of the on/off valve and optimization. The power loss is related to both geometric parameters and system parameters. From design point of view, the higher the flow rate \( Q \) and valve switching frequency \( \omega \) are, the better the performance of the on/off valve is, but that is limited by the increase of the power loss, especially in high \( \omega \), the compressibility loss will prevail. Using a feedback control system of axial position of valve spool can obtain good dynamic performance of the on/off valve. It is better using a fuzzy logic parameter self-tuning PID controller than a conventional PID controller because of nonlinearity of the system.

References


