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High Speed On/Off Valve Control Hydraulic Propeller

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Abstract: The work-class remotely-operated-underwater-vehicles (ROVs) are mainly driven by hydraulic propulsion system, and the effeciency of hydraulic propulsion system is an important performance index of ROVs. However, the efficiency of traditional hydraulic propulsion system controlled by throttle valves is too low. Therefore, in this paper, for small and medium ROVs, a novel propulsion system with higher efficiency based on high speed on/off valve control hydraulic propeller is proposed. To solve the conflict between large flow rate and high frequency response performance, a two-stage high speed on/off valve-motor unit with large flow rate and high response speed simultaneously is developed. Through theoretical analysis, an effective fluctuation control method and a novel pulse-width-pulse-frequency-modulation (PWPFM) are introduced to solve the conflict among inherently fluctuation, valve dynamic performance and system efficiency. A simulation model is established to evaluate the system performance. To prove the advantage of system in energy saving, and test the dynamic control performance of high speed on/off valve control propeller, a test setup is developed and a series of comparative experiments is completed. The smimulation and experiment results show that the two-stage high speed on/off valve has an excellent dynamic response performance, and can be used to realize high accuracy speed control. The experiment results prove that the new propulsion system has much more advantages than the traditional throttle speed regulation system in energy saving. The lowest efficiency is more than 40%. The application results on a ROV indicate that the high speed on/off valve control propeller system has good dynamic and steady-state control performances. Its transient time is only about 1 s–1.5 s, and steady-state error is less than 5%. Meanwhile, the speed fluctuation is small, and the smooth propeller speed control effect is obtained.On the premise of good propeller speed control performance, the proposed high speed on/off valve control propeller can improve the effeciency of ROV propulsion system significantly, and provides another attractive ROV propulsion system choice for engineers.

Key words: high speed on/off valve, fluctuation control, pulse-width-pulse-frequency-modulation (PWPFM), hydraulic propulsion system, remotely-operated-underwater-vehicle (ROV), energy saving system

1 Introduction

At present, the remotely-operated-underwater-vehicles (ROVs) are important tools for ocean exploration and research[1]. Comparing with equipments on the land, the ROVs are more sensitive on weight. Therefore, the hydraulic actuators with high energy density are widely used.

Normally, there are two methods to regulate propeller speed. One is throttle speed regulation. The other is variable displacement speed regulation. The ROVs need to be kept in the state of low-speed for a long time when they are working. However, the load characteristics of propeller are nonlinear. Namely, that the thrust is 2nd power of propeller speed when the power is 3rd power. So, most energy will be wasted on throttle valves in low-speed condition. Generally speaking, the variable displacement speed regulation is more efficiently[2–3]. However, the minimum variable displacement motor is about 28×10^{-6} m^3/r (28 cc/r), which means that when system pressure is 21 MPa, the power of ROVs with 5 propellers will exceed 70 kW. Therefore, the variable displacement speed regulation propulsion systems are mainly used in large ROVs.

Due to the deep-sea motor which is used to support power for hydraulic system is very expensive, that the studying on energy saving technology for ROVs hydraulic propulsion system is interesting and valuable.

In hydraulic fields, high speed on/off valves and their applications are widely studied for their advantage in energy saving. High speed on/off valves are able to reduce throttle loss. They also can redirect the energy, which dissipated over a control valve in some cases. In more researches and applications, the possible energy savings are already well proved[4–5]. At the same time, the high speed on/off valves are cheap and robust, and they have rapid response speed, so high precision position and speed control can be realized. LAAMANEN, et al[6], proved that

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the high speed on/off valves had potential to replace proportional valves in mobile hydraulic applications. However, the pressure surge and the phsical size are two problems need to be further studied. In Refs. [7–9], the applications of high speed on/off valves on high precision hydraulic cylinder position control have been well studied. And WANG, et al[10], used the high speed on/off valve on the displacement-varying mechanism of pump/motor successfully. Based on the researches above, the effectiveness of high speed on/off valves on position servo system has been proved. For hydraulic motor speed control, QIN, et al[11], realized the speed control of a hydraulic motor with high inertia load based on switch mode hydraulic power supply. The system is a semi closed-loop circuit constructed by a solenoid piloted cartridge valve and a high speed check valve. Although its frequency response performance can meets the requirements of most construction machineries, but not good enough for high-performance servo systems. This is due to the following two challenges.

First is the conflict between large flow rate and high frequency response speed. Second is the conflict among inherently fluctuation, valve dynamic performance and efficiency. In high speed on/off valve control systems, the pressure and speed fluctuation caused by switching cannot be avoided. Generally, the higher the switch frequency is, the smaller the fluctuation will be, however the higher performance of valve is needed and the efficiency of valve will be decreased.

To solve these problems, some researches have been done, and some new valve structures and improved control methods are proposed. TU, et al[12], designed a high speed rotary on/off valve to achieve high respond speed and large flow rate with low actuation power. The efficiency of the valve had been proved through experiments. But, the inner leakage will be large at high pressure for its rotary clearance seal structure. SHI, et al[13], recommended a special two-stage high speed on/off valve, in which a giant magnetostrictive actuator was used as the pilot stage and a ball valve was used as the main stage. Then, flow rate and bandwidth of the valve are greatly increased. But the price of giant magnetostrictive actuator is so high that it couldn't be widely used.

In this research, the fundamental goal is to propose a novel integrated high-efficiency ROV propulsion system based on high speed on/off valve control hydraulic propeller. To achieve this goal, aiming to solve the problems that restrict the practical applications of high speed on/off valve, a new two-stage high speed on/off valve-motor unit and performance optimization methods will be introduced.

The rest of this paper is organized as follows. In section 2, the system working principle and the structure of two-stage high speed on/off valve will be discussed in detail. In section 3, system performance optimization methods will be proposed. In section 4, system simulation model will be established, and some simulation results will be obtained. In section 5, detailed experiment results will be presented, and the application of high speed on/off valve control propeller on a ROV will be introduced briefly. In the last section, some concluding remarks will be presented.

2 Working Principle and Structure

The master-slave control is widely employed in the robot manipulation. In most cases, the joystick or the keyboard is the routine input device for the robot master-slave control system. The system presented in this paper is shown in Fig. 1. A two-stage structure is introduced to get a large flow rate and high frequency response performance simultaneously. The working principle of valve is shown in Fig. 1. The main valve structure and two working states are presented in Fig. 2.

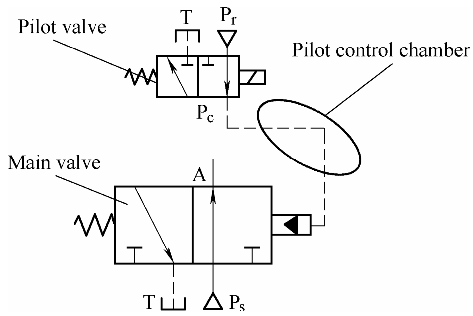

Fig. 1. Working principle of two-stage high speed on/off valve

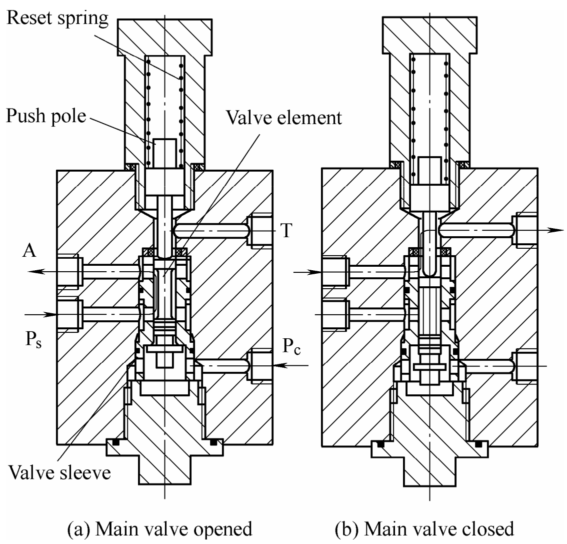

Fig. 2. Structure and working states of main valve

When pilot valve opens, the pilot control chamber pressure p_c will push the valve element up, then port P_s and port A will be connected. When the pilot valve closes, the reset spring will push the valve element down, and then port A will be connected with port T. And one switching cycle is complete.

Two HSV-series high speed solenoid on/off valves are used as the pilot valves. The valve is cheap, robust and has good dynamic response performance. Its open time is only 3.5 ms, while the closed time is 2.5 ms. The nominal working pressure is 7 MPa with 6.67×10^{-5} m^3/s (4 L/min) rated flow.

To improve the efficiency of valve, the energy loss in pilot stage should be as small as possible. Therefore, a slide-valve structure whose open pressure is smaller than other types is chose. In this system, the pilot pressure p_r is only 7 MPa. Therefore, the power loss in pilot stage can be reduced to a low lever.

The diameter of main valve element is 6 mm, and the max working pressure is 31.5 MPa, with 4.17×10^{-4} m^3/s (25 L/min) rated flow.

An integrated valve-motor unit is developed to drive the propeller directly. The integrated valve-manifold is installed in the rear of motor to simplify the system structure and improve the response performance. All the valves are installed into the valve-manifold by screw-in cartridge form. The motor is MCY14-1B axial piston motor with the displacement is 5×10^{-6} m^3/r (5 cc/r), produced by Shanghai High Pressure Hydraulic Pump Factory Company, China.

3 System Performance Optimization

3.1 Speed fluctuation control

3.1.1 Speed fluctuation theoretical analysis

As the hydraulic motor is driven in switching mode, the speed fluctuation cannot be avoided. However, the fluctuation has to be controlled in an acceptable level to improve the control performance. So, it is necessary to analyze the relationships between system parameters and fluctuation degree.

To simplify the analysis, several assumptions are made: 1) the supply pressure p_s is assumed to be constant, 2) the leakage in system and the pressure drop through valves and pipes are neglected, 3) the viscous damping coefficient of motor is neglected, 4) the dynamic characteristics of high speed on/off valves are neglected, which means that the pressure on the motor rises to p_s as soon as high speed on/off valve opens, while it closes, the pressure drops to 0 immediately, 5) using the model of fluctuation process as shown in Fig. 3. The initial speed ω_1 will be accelerated to the ω_2 during DT, and decelerated to the ω_3 after the time from DT to T. The ω_3 is the initial condition in the next period and equals to ω_1.

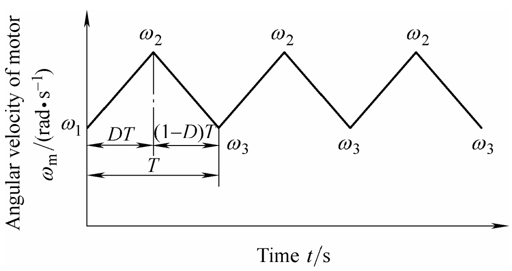

Fig. 3. Speed fluctuation model used in theoretical analysis

When on/off valve opens, the dynamic equation of motor is established as follows:

$$J_m\frac{d\omega_m}{dt}=\frac{D_m p_s}{2\pi}-T_L, \tag{1}$$

where J_m is the moment of inertial of motor and propeller, ω_m is the angular velocity of motor, and D_m is the displacement of motor. The load torque T_L can be approximately simplified as

$$T_L=k_L\omega_m^2, \tag{2}$$

where k_L is the drag coefficient of propeller.

Assuming that the duration time of accelerating stage is DT, where D is the duty cycle and T is the on/off period of valve. From Eq. (1) and Eq. (2), we can get the following ordinary differential equation:

$$\frac{d\omega_m}{\dfrac{D_m p_s}{2\pi k_L}-\omega_m{}^2}=\frac{k_L}{J_m}dt. \tag{3}$$

To integral on both sides of Eq. (3), we get

$$\int_{\omega_1}^{\omega_2}\frac{d\omega_m}{\omega_m{}^2-\left(\sqrt{\dfrac{D_m p_s}{2\pi k_L}}\right)^2}=\int_0^{DT}-\frac{k_L}{J_m}dt. \tag{4}$$

Using the integral formula in Ref. [14]:

$$\int\frac{dx}{x^2-a^2}=\frac{1}{2a}\ln\left|\frac{x-a}{x+a}\right|+C, \tag{5}$$

then it is easily to get

$$\frac{1}{2\sqrt{\dfrac{D_m p_s}{2\pi k_L}}}\ln\left(\frac{\left(\omega_2-\sqrt{\dfrac{D_m p_s}{2\pi k_L}}\right)\left(\omega_1+\sqrt{\dfrac{D_m p_s}{2\pi k_L}}\right)}{\left(\omega_2+\sqrt{\dfrac{D_m p_s}{2\pi k_L}}\right)\left(\omega_1-\sqrt{\dfrac{D_m p_s}{2\pi k_L}}\right)}\right)=-\frac{k_L}{J_m}DT. \tag{6}$$

Similarly, when valve closes, the dynamic equation of motor is established as follows:

$$J_m\frac{d\omega_m}{dt}=-k_L\omega_m^2. \tag{7}$$

The initial angular velocity is ω_2, after the time from DT to T, the angular velocity is ω_3. Solving the Eq. (7), we will get

$$\frac{1}{\omega_3}-\frac{1}{\omega_2}=\frac{k_L}{J_m}(T-DT). \tag{8}$$

Simplifying Eq. (8) and we can obtain the following:

$$\omega_3\omega_2 = \omega_1\omega_2 = \frac{J_{\mathrm{m}}(\omega_2-\omega_3)}{k_{\mathrm{L}}(T-DT)} = \frac{J_{\mathrm{m}}(\omega_2-\omega_1)}{k_{\mathrm{L}}(T-DT)}. \tag{9}$$

From Eq. (6), it is easy to show that

$$\exp\left(2\sqrt{\frac{D_{\mathrm{m}}p_{\mathrm{s}}}{2\pi k_{\mathrm{L}}}}\frac{k_{\mathrm{L}}DT}{J_m}\right) = \frac{\left(\omega_2+\sqrt{\frac{D_{\mathrm{m}}p_{\mathrm{s}}}{2\pi k_{\mathrm{L}}}}\right)\left(\omega_1-\sqrt{\frac{D_{\mathrm{m}}p_{\mathrm{s}}}{2\pi k_{\mathrm{L}}}}\right)}{\left(\omega_2-\sqrt{\frac{D_{\mathrm{m}}p_{\mathrm{s}}}{2\pi k_{\mathrm{L}}}}\right)\left(\omega_1+\sqrt{\frac{D_{\mathrm{m}}p_{\mathrm{s}}}{2\pi k_{\mathrm{L}}}}\right)} = 1-\frac{2\sqrt{\frac{D_{\mathrm{m}}p_{\mathrm{s}}}{2\pi k_{\mathrm{L}}}}(\omega_2-\omega_1)}{\omega_1\omega_2-\sqrt{\frac{D_{\mathrm{m}}p_{\mathrm{s}}}{2\pi k_{\mathrm{L}}}}(\omega_2-\omega_1)-\frac{D_{\mathrm{m}}p_{\mathrm{s}}}{2\pi k_{\mathrm{L}}}}. \tag{10}$$

Substituting Eq. (9) into Eq. (10), we can obtain

$$\exp\left(2\sqrt{\frac{D_{\mathrm{m}}p_{\mathrm{s}}}{2\pi k_{\mathrm{L}}}}\frac{k_{\mathrm{L}}DT}{J_{\mathrm{m}}}\right) = 1+\frac{2\sqrt{\frac{D_{\mathrm{m}}p_{\mathrm{s}}}{2\pi k_{\mathrm{L}}}}(\omega_2-\omega_1)}{\left[\sqrt{\frac{D_{\mathrm{m}}p_{\mathrm{s}}}{2\pi k_{\mathrm{L}}}}-\frac{J_{\mathrm{m}}}{k_{\mathrm{L}}(T-DT)}\right](\omega_2-\omega_1)+\frac{D_{\mathrm{m}}p_{\mathrm{s}}}{2\pi k_{\mathrm{L}}}}. \tag{11}$$

So the speed fluctuation is

$$\Delta\omega_{\mathrm{m}} = \omega_2-\omega_1 = \frac{\frac{D_{\mathrm{m}}p_{\mathrm{s}}}{2\pi k_{\mathrm{L}}}}{\frac{J_{\mathrm{m}}}{k_{\mathrm{L}}(T-DT)}-\sqrt{\frac{D_{\mathrm{m}}p_{\mathrm{s}}}{2\pi k_{\mathrm{L}}}}+\frac{2\sqrt{\frac{D_{\mathrm{m}}p_{\mathrm{s}}}{2\pi k_{\mathrm{L}}}}}{\exp\left(2\sqrt{\frac{D_{\mathrm{m}}p_{\mathrm{s}}}{2\pi k_{\mathrm{L}}}}\frac{k_{\mathrm{L}}DT}{J_m}\right)-1}}. \tag{12}$$

And the average speed can be approximated as

$$\bar{\omega}_{\mathrm{m}} \approx \sqrt{\omega_1\omega_2} = \sqrt{\frac{J_{\mathrm{m}}}{k_{\mathrm{L}}}\frac{\omega_2-\omega_1}{T-DT}} = \sqrt{\frac{J_{\mathrm{m}}}{k_{\mathrm{L}}}\frac{\Delta\omega_{\mathrm{m}}}{T(1-D)}}. \tag{13}$$

Then the fluctuation degree ζ is defined as

$$\zeta = \frac{\Delta\omega_{\mathrm{m}}}{\bar{\omega}_{\mathrm{m}}}\times 100\%. \tag{14}$$

3.1.2 Speed fluctuation control method

According to the analysis above, it is easily to find that the fluctuation degree will reduce significantly when T decreases. However, increasing the switching frequency not only requests a better dynamic performance for high speed on/off valve, but also will decrease efficiency seriously. Therefore, as a tradeoff between performance and efficiency, the highest frequency used in this system is restricted to 20 Hz.

From Eq. (12), it can be known that increasing J_{m} is another effective way to control the fluctuation. Therefore, a flywheel with large inertia is added on the shaft of propeller to filter speed fluctuation.

The comparative simulation response curves of the system with flywheel and without flywheel are displayed in Fig. 4. The simulation model will be established in next section. In simulations, the duty cycle is set as 50% while the frequency is 20 Hz. The speed fluctuation degree is decreased from 69.7% to 5.25%.

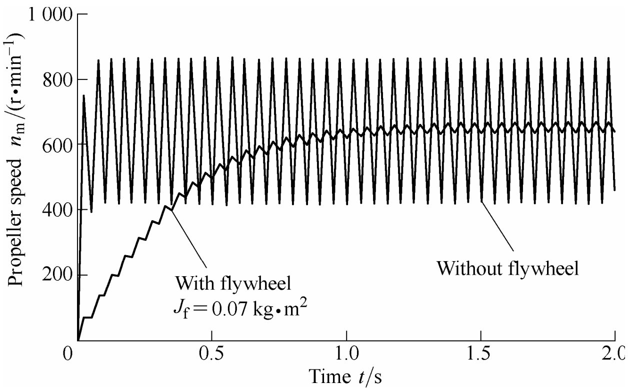

Fig. 4. Fluctuation control effects

This method will increase the response time of the propeller. However, compared with ROVs, whose time constant are normally bigger than 10 s, this tiny response speed decrease is acceptable, and has little influence on ROVs control performance.

3.2 Pulse-width-pulse-frequency modulation

The pulse width modulation (PWM) amplifiers are used in most high speed on/off control systems[15]. The PWM amplifier regulates the duty cycle of pulse signal according to the controller signal, and keeps the frequency constant all the time. In most conditions, the PWM control is good enough, especially in electronic systems. However, there are at least two motivations to introduce a novel PWPFM control method in hydraulic high speed on/off valve control systems.

Firstly, when duty cycle is under 10% or above 90%, the high speed on/off valve cannot be fully opened or closed in a single-pulse period due to the limitation of bandwidth. As a result, the control performance will degrade greatly. Therefore, it is necessary to reduce the frequency properly to make sure the enough switching time when duty cycle is small or big.

Secondly, increase the efficiency. It is easily to understand that the throttling loss in main stage and the power loss in pilot valve are proportional to switching frequency. It is an intuitive idea to reduce the frequency when high frequency is not needed. The problem is how to distinguish when a high frequency is needed and when is not. The most important purpose of high frequency is to reduce fluctuation degree. Therefore, the high frequency is used when small fluctuation degree is wanted. In ROV propulsion system, the moderate speed region is the most commonly used working condition that the small fluctuation degree is needed. And the propeller speed closed-loop control dynamic process can be divided into accelerated/decelerated stage and steady speed stage. At the accelerated/decelerated stage, the rapidity is more important while the fluctuation can be neglected. A lower frequency can be used. At the steady speed stage, the small fluctuation degree and good control performance are more important. Then a higher frequency is needed.

Based on the analysis above, the switching frequency and duty cycle of PWPFM control method are designed as follows:

$$f = -3.2|u|(|u|-5),\ -5 \leqslant u \leqslant 5, \tag{15}$$

$$D = 0.2|u| \times 100\%,\ -5 \leqslant u \leqslant 5, \tag{16}$$

where u is the control input voltage. When it is positive, the valve on the right will be powered and the motor will be accelerated on clockwise. When it is negative, the valve on the left will work and the motor will be accelerated on counter-clockwise.

The PWPFM amplifier has been realized in an analog circuit. Fig. 5 shows the actual measured output performance curves of the PWPFM amplifier.

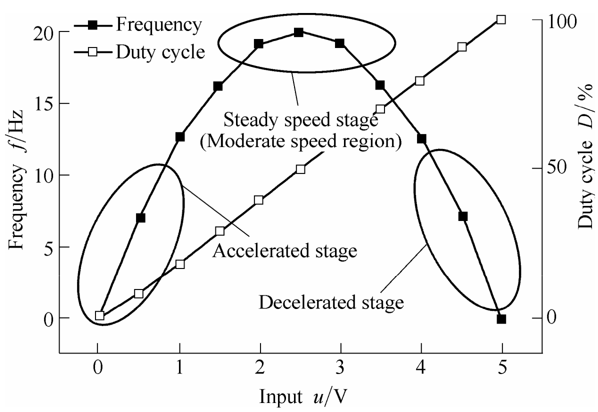

Fig. 5. Actual output performance curves of PWPFM amplifier

4 System Modelling and Simulation Results

The system simulation model is needed to evaluate system performance and optimize design parameters. And to better understand the working principle of high speed on/off valve control propeller system, the detailed system modeling process is presented.

The Fig. 6 is the hydraulic system schematic used in simulation.

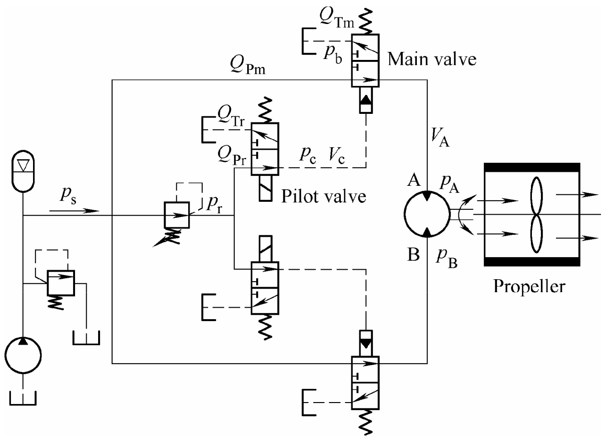

Fig. 6. Hydraulic system schematic

4.1 Assumptions

To simplify the model, several assumptions are made: 1) the effective bulk modulus, β_e, is estimated to be 690 MPa due to the effect of entrained air in the system, 2) the dynamics of relief valve and reducing pressure valve are neglected, which means the system pressure p_s and pilot pressure p_r are assumed to be constant, 3) the leakage and pressure drop through the valves and pipes are neglected.

The definitions and values of notations used in the model can be found in Table.

Table. Simulation model parameters

Parameter	Value
Density of hydraulic fluid ρ/(kg • m^{-3})	870
Effective bulk modulus β_e/MPa	690
Synthesis flow coefficient of pilot valve C_p/(mm^3 • Pa$^{-1/2}$ • s^{-1})	25.2
Synthesis flow coefficient of main valve C_m/(mm^3 • Pa$^{-1/2}$ • s^{-1})	326
Pilot pressure p_r/MPa	7
System pressure p_s/MPa	30
System back pressure p_b/MPa	0.5
Mass of valve element and push pole m_v/g	15
Reset spring stiffness k_s/(N • m^{-1})	5 950
Pre-compression of spring x_{v0}/mm	11.82
Viscosity friction coefficient of valve B_v/(kg • s^{-1})	0.01
Diameter of valve element d_v/mm	6
Cross-sectional area of valve element A_c /mm^2	28.26
Radius of throttling window R/mm	1.2
Actual flow length between ports A and P_s L_1/mm	14
Actual flow length between ports A and T L_2/mm	11
Velocity coefficient C_v	0.98
Initial pilot control chamber volume V_{c0}/cm^3	3.5
Volume between main valve and motor V_A/cm^3	5
Displacement of motor D_m/ (cm^3 • rad^{-1})	0.8
Mechanical efficiency of motor η_m/%	92
Internal leakage coefficient of motor C_{im}/ (m^3 • Pa^{-1} • s^{-1})	2.67×10^{-13}
Viscosity friction coefficient of motor B_m/ (kg • m^2 • rad^{-1} • s^{-1})	0.01
Drag coefficient of propeller k_L/ (kg • m^2 • rad^{-2})	0.002 1
Moment of inertial of motor and propeller J_m/ (kg • m^2)	0.005
Moment of inertial of flywheel J_f/(kg • m^2)	0.07

4.2 Model of pilot valve

The first order model is good enough to describe the dynamics of pilot valve because the pilot valve has excellent dynamic performance. Fig. 7 shows the first order model of pilot valve. The valve displacement is converted into a normalized coefficient K_{vout}, from 0 to 1.

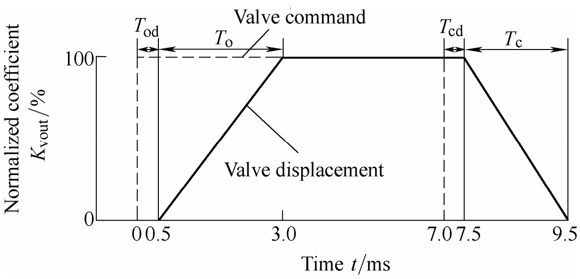

Fig. 7. Linear model of pilot valve

The flow rate from port $\mathrm{P_r}$ to port $\mathrm{P_c}$ can be characterized as

$$Q_{\mathrm{Pr}} = C_{\mathrm{P}} K_{\mathrm{vout}} \sqrt{p_{\mathrm{r}} - p_{\mathrm{c}}}, \tag{17}$$

where C_{P} is the synthesis flow coefficient of pilot valve. And flow rate from port $\mathrm{P_c}$ to port T is

$$Q_{\mathrm{Tr}} = C_{\mathrm{p}} (1 - K_{\mathrm{vout}}) \sqrt{p_{\mathrm{c}} - p_{\mathrm{b}}}. \tag{18}$$

The pressure in pilot control chamber is

$$p_{\mathrm{c}} = \frac{\beta_{\mathrm{e}}}{V_{\mathrm{c}}} \int (Q_{\mathrm{Pr}} - Q_{\mathrm{Tr}}) \mathrm{d}t, \tag{19}$$

where V_{c} is the volume of pilot control chamber, and

$$V_{\mathrm{c}} = V_{\mathrm{c0}} + A_{\mathrm{c}} x_{\mathrm{v}}. \tag{20}$$

where x_{v} is the displacement of main valve element.

4.3 Model of main valve

The main valve is refitted from a pilot-controlled directional valve. The max displacement of valve element is 5 mm. The valve sleeve has round throttling windows. The flow rate characteristics of main valve are nonlinear and subsection functions as follows:

$$Q_{\mathrm{Tm}} = \mathrm{sgn}(p_{\mathrm{A}} - p_{\mathrm{b}}) C_{\mathrm{m}} K_{\mathrm{T}} \sqrt{|p_{\mathrm{A}} - p_{\mathrm{b}}|}, \tag{21}$$

$$Q_{\mathrm{Pm}} = \mathrm{sgn}(p_{\mathrm{s}} - p_{\mathrm{A}}) C_{\mathrm{m}} K_{\mathrm{P}} \sqrt{|p_{\mathrm{s}} - p_{\mathrm{A}}|}, \tag{22}$$

$$K_{\mathrm{T}} = \begin{cases} 1, & x_{\mathrm{v}} < 0.2\ \mathrm{mm}, \\ 1 - \dfrac{S(x_{\mathrm{v}} - 0.2)}{\pi R^2}, & 0.2\ \mathrm{mm} \leqslant x_{\mathrm{v}} \leqslant 2.6\ \mathrm{mm}, \\ 0, & x_{\mathrm{v}} > 2.6\ \mathrm{mm}, \end{cases} \tag{23}$$

$$K_{\mathrm{P}} = \begin{cases} 0, & 0 < x_{\mathrm{v}} < 2.9\ \mathrm{mm}, \\ \dfrac{S(x_{\mathrm{v}} - 2.9)}{\pi R^2}, & 2.9\ \mathrm{mm} \leqslant x_{\mathrm{v}} \leqslant 5\ \mathrm{mm}, \end{cases} \tag{24}$$

$$S(x_{\mathrm{v}}) = \begin{cases} R^2 \arccos\left(\dfrac{R - x_{\mathrm{v}}}{R}\right) - (R - x_{\mathrm{v}}) \times \\ \quad \sqrt{2Rx_{\mathrm{v}} - x_{\mathrm{v}}^2}, \ 0 < x_{\mathrm{v}} < R, \\ \pi R^2 - R^2 \arccos\left(\dfrac{x_{\mathrm{v}} - R}{x_{\mathrm{v}}}\right) + \\ \quad (x_{\mathrm{v}} - R)\sqrt{2Rx_{\mathrm{v}} - x_{\mathrm{v}}^2}, \ R \leqslant x_{\mathrm{v}} < 2R. \end{cases} \tag{25}$$

Where Q_{Tm} is the flow rate from port A to port T, and Q_{Pm} is the flow rate from port $\mathrm{P_s}$ to port A, K_{T} and K_{P} are normalized displacement coefficients from port A to port T and port $\mathrm{P_s}$ to port A individually. The function $S(x_{\mathrm{v}})$ is the calculation formula of valve opening area. The sign of flow rate, which enters into the motor, is defined as positive, while flowing out the motor is negative.

The motion equation of valve element is

$$m_{\mathrm{v}} \ddot{x}_{\mathrm{v}} = F_{\mathrm{c}} - F_{\mathrm{k}} - F_{\mathrm{s}} - F_{\mathrm{d}} - B_{\mathrm{v}} \dot{x}_{\mathrm{v}}, \tag{26}$$

where F_{c} is the driving force, F_{k} is the reset force generated by spring, F_{s} is the steady-state fluid force, F_{d} is the transient fluid force, and B_{v} is the viscosity friction coefficient of valve element:

$$F_{\mathrm{c}} = A_{\mathrm{c}} (p_{\mathrm{c}} - p_{\mathrm{b}}), \tag{27}$$

$$F_{\mathrm{k}} = k_{\mathrm{s}} (x_{\mathrm{v}} + x_{\mathrm{v0}}). \tag{28}$$

The F_{s} and F_{d} are established at two working states respectively:

$$F_{\mathrm{s}} = \begin{cases} \rho Q_{\mathrm{Pm}} v_{\mathrm{P}} \cos 69^{\circ}, \\ \rho Q_{\mathrm{Tm}} v_{\mathrm{T}} \cos 69^{\circ}, \end{cases} \tag{29}$$

$$F_{\mathrm{d}} = \begin{cases} \rho L_1 \dfrac{\mathrm{d}Q_{\mathrm{Pm}}}{\mathrm{d}t}, \\ \rho L_2 \dfrac{\mathrm{d}Q_{\mathrm{Tm}}}{\mathrm{d}t}, \end{cases} \tag{30}$$

$$v_{\mathrm{P}} = C_{\mathrm{v}} \sqrt{\frac{2(p_{\mathrm{s}} - p_{\mathrm{A}})}{\rho}}, \tag{31}$$

$$v_{\mathrm{T}} = C_{\mathrm{v}} \sqrt{\frac{2(p_{\mathrm{A}} - p_{\mathrm{b}})}{\rho}}, \tag{32}$$

where v_{P} is the velocity of flow from port $\mathrm{P_s}$ to port A, v_{T} is the velocity of flow from port A to port T, L_1 is the actual flow length between port $\mathrm{P_s}$ and port A, L_2 is the actual flow length between port A and port T, and C_{v} is the velocity coefficient.

4.4 Model of hydraulic motor and load

For the reason of convenience, p_H and p_L are used to replace p_A and p_B that used above. It will not result in any confusion.

The pressure in the high pressure chamber of hydraulic motor is

$$p_H = \frac{\beta_e}{V_A}\int (Q_{Pm} - D_m\omega_m - C_{im}(p_H - p_L))dt, \tag{33}$$

where C_{im} is the interner leakage coefficient of motor.

The pressure in low pressure chamber is

$$p_L = \frac{\beta_e}{V_A}\int [D_m\omega_m + C_{im}(p_H - p_L) - Q_{Tm}]dt. \tag{34}$$

The motion equation of motor is established as follows:

$$\eta_m D_m(p_H - p_L) = J_L\frac{d^2\theta_m}{dt^2} + B_m\frac{d\theta_m}{dt} + T_L, \tag{35}$$

$$J_L = J_m + J_f, \tag{36}$$

where η_m is the mechanical efficiency of motor, B_m is the viscosity friction coefficient of motor, J_L is the moment of inertial of load, and J_f is the moment of inertial of flywheel. In the one-state model of propeller proposed by Yoerger, the drag torque T_L is[16]

$$T_L = k_L\omega_m|\omega_m|. \tag{37}$$

4.5 Simulation model

According to the analysis above, the system model is established by using Matlab/Simulink® as shown in Fig. 8. The system model is divided into four subsystems: the switching controller, the pilot valves, the main valves and the load.

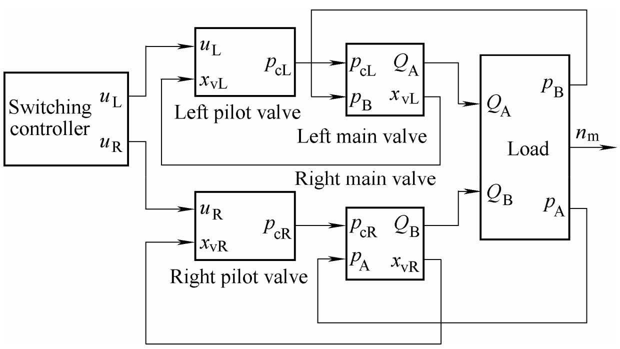

Fig. 8. Simulation model of system

4.6 Simulation results

Fig. 9 shows the simulation curves of some important variables. In the simulations, the switching frequency is 20 Hz and duty cycle is 50%. In Fig. 9, Q_r is the flow rate of pilot valve and Q_m is the flow rate of main valve.

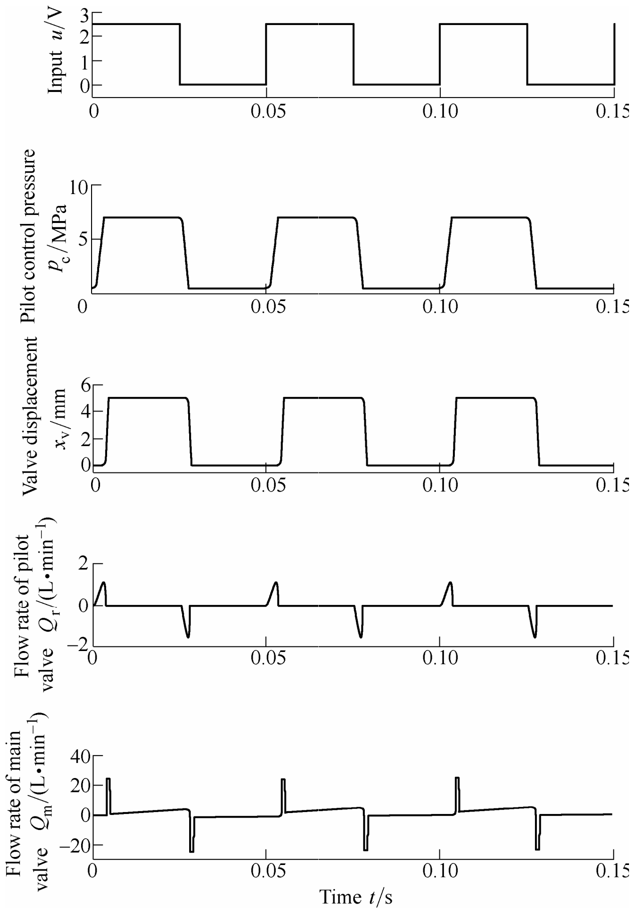

Fig. 9. Dynamic response simulation curves

The simulation results show that the power loss in system is mainly caused by large flow rate of pilot valve and main valve at the moment of opening and closing. Therefore, the peak of flow rate has to be reduced if higher efficiency is wanted.

The output pressure response simulation curve, shown in next section, is compared with experiment result. Although the simulation model cannot describe the micro pressure ripple caused by measurement noise and high frequency unmodeled dynamics, when valve is fully opened, but the model is sufficiently to represent the most important characteristics of the system. Therefore, the validity of the simulation model can be proved.

5 Experiment and Application Results

5.1 Experiment results

5.1.1 Test setup structure

To prove the advantage of the system in energy saving, and test the dynamic control performance of high speed on/off valve control propeller, a test setup is developed as shown in Fig. 10. Two pressure sensors are used to measure the pressure of motor inlet and outlet. Two flowmeters are used to obtain the average flow rate of pilot stage and main stage individually. The torque and rotary speed sensor is used to obtain the motor output torque and speed.

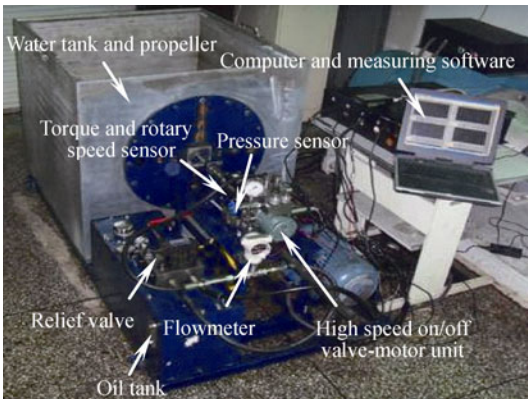

Fig. 10. Test setup overview

The experiment data is acquired by Advantech USB-4716 DAQ card. The sampling interval is 1 ms (1 ks/s). The measuring software is programmed by using LABVIEW 8.2.

5.1.2 System dynamic performance test

The output pressure response curves have been obtained when switching frequency is 20 Hz, and duty cycle is 50%, as shown in Fig. 11. The open dead-zone time T_{do}, is about 5 ms, while the close dead-zone time T_{dc} is about 4 ms.

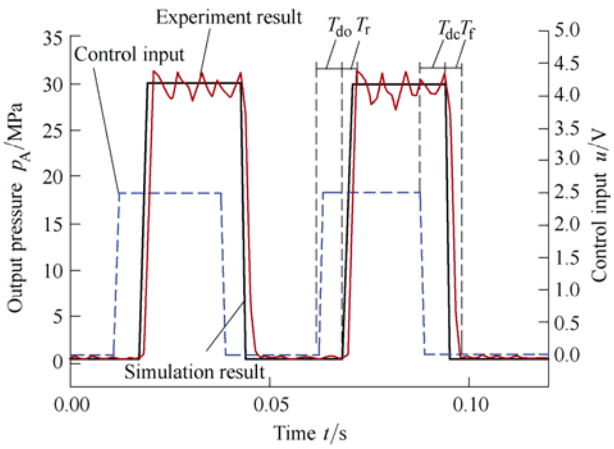

Fig. 11. Output pressure response curves

The dead-zone cannot be avoided in two-stage high speed on/off valves, but it can be reduced by decreasing the pilot control chamber volume, or choosing a bigger pilot valve. Different from the high speed on/off valve control cylinder positioning systems, the dead-zone has a significant influence on control accuracy. The dead-zone has little influence on the control performance of high speed on/off valve control motor speed regulation systems. The dead-zone will only result in a total output phase changing, which will not be summed with the passage of time. It means that the longer the output series is, the smaller the performance degradation will be. Therefore, this little dead-zone time is absolutely acceptable in this application.

The most important performance indexes are pressure rise and fall time. They are directly related to the system dynamic performance and the size of throttling loss. In this system, the pressure rise time T_{r} is only about 2.5 ms, while the pressure fall time T_{f} is about 3 ms, which are very close to the indexes of pilot valve. It proves that the two-stage high speed on/off has an excellent dynamic response performance, and it can be used to realize the high performance speed servo control.

5.1.3 System efficiency comparative experiment result

A comparative test is completed to prove the advantage of the proposed system in efficiency than traditional propeller speed regulation system. In the test, the motor speed is controlled by high speed on/off valve and proportional throttle valve respectively. The efficiency calculation formulas used are as follows:

$$\eta_{\mathrm{on/off}}=\frac{\bar{T}_{\mathrm{m}}\bar{\omega}_{\mathrm{m}}}{p_{\mathrm{s}}\bar{Q}_{\mathrm{Pm}}+p_{\mathrm{r}}\bar{Q}_{\mathrm{Pr}}}\times 100\%, \tag{38}$$

$$\eta_{\mathrm{throttle}}=\frac{\bar{T}_{\mathrm{m}}\bar{\omega}_{\mathrm{m}}}{p_{\mathrm{s}}\bar{Q}_{\mathrm{s}}}\times 100\%, \tag{39}$$

where $\bar{T}_{\mathrm{m}}$ is the average output torque of motor, and $\bar{Q}_{\mathrm{s}}$ is the average flow rate of proportional throttle valve.

It is difficult to estimate the efficiency of motor, coupling and bearings at different working conditions. So, the obtained efficiency is the overall efficiency of valve-motor system.

The comparative experiment results have been shown in Fig. 12. It is clearly that the efficiency of high speed on/off valve control motor system is obviously higher than throttle speed regulation system, especially in medium and low speed. The lowest efficiency of system is more than 40%.

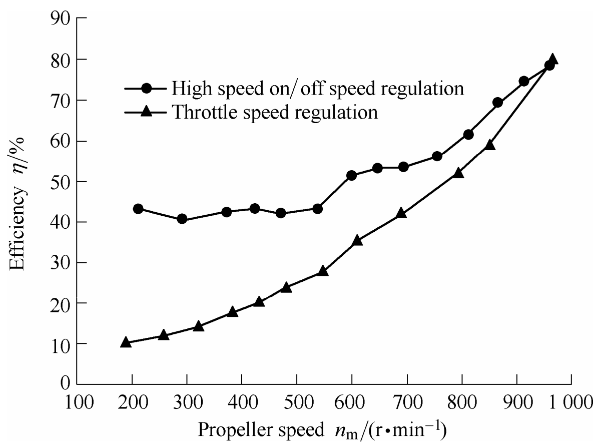

Fig. 12. System efficiency comparative experiment results

5.1.4 Speed fluctuation control effect

The fluctuation degree at different speed is obtained through experiments. It is compared with the theoretical results as shown in Fig. 13. The fluctuation degree in medium and high speed is only about 6% to 7%, which is very close to theoretical results. In low speed, the actual fluctuation degree is bigger than the theoretical results due to the large vibration caused by hydraulic shock.

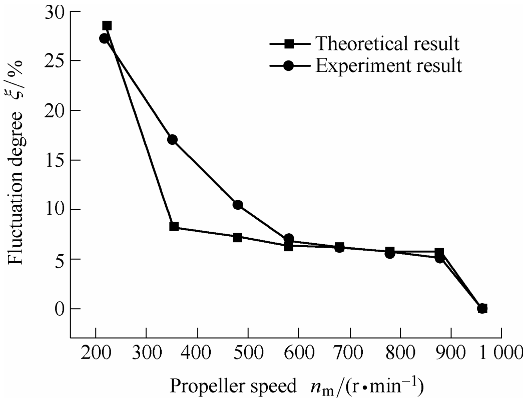

Fig. 13. Speed fluctuation degree control effect

The experiment results prove that the fluctuation control method proposed is effective, and the theoretical results can be used to estimate the size of flywheel.

5.2 Applications on ROV

5.2.1 ROV propulsion system structure

The integrated high-efficiency high speed on/off valve control propeller unit has been successfully used on a 15 kW medium ROV with 5 thrusters as shown in Fig. 14.

Fig. 14. ROV overview

Fig. 15 shows the details of the integrated high-efficiency propulsion unit. The PWPFM amplifier and speed controller are placed in an oil-filled shell which is pressure balanced with water pressure. The circuits can resist the pressure of 45 MPa, corresponding to 4 500 m water depth. Two magnetoelectric sensors, which are placed vertically, are used to measure the speed pulses produced by the magnets fixed on flywheel.

Another advantage of using the integrated propulsion unit is that advanced hydraulic bus structure can be used, as shown in Fig. 16. Compared with the traditional ROV propulsion system, presented in Fig. 17, the propeller valve box can be removed, and the placement of pipes is greatly simplified.

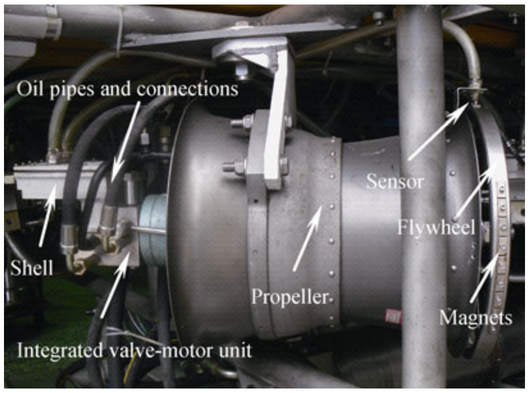

Fig. 15. Integrated high efficiency propulsion unit

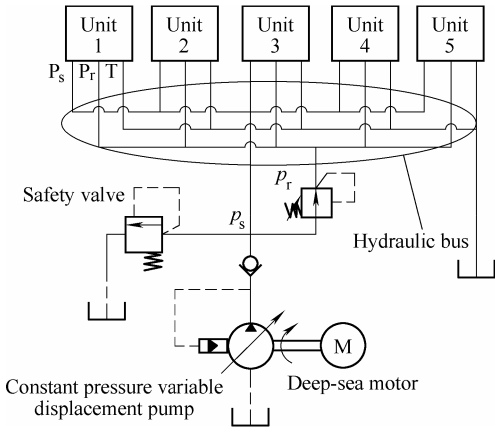

Fig. 16. ROV hydraulic system base on hydraulic bus

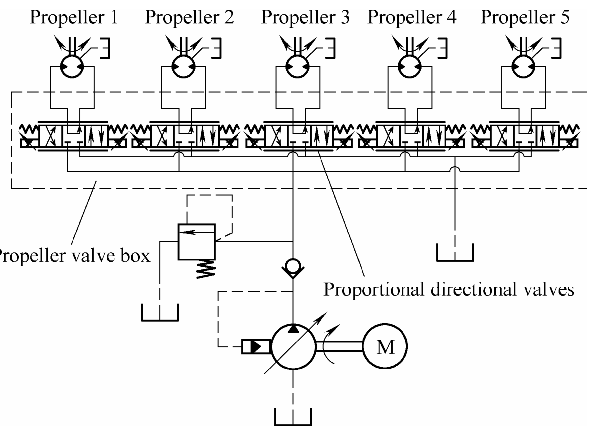

Fig. 17. Traditional ROV hydraulic system schematic

5.2.2 Propeller speed control effect

Because the dynamic characteristics of ROV and disturbances in the ocean are very complicated, the high precision speed control cannot be easily achieved by a simple open-loop control structure. Therefore, a closed-loop control structure is introduced, where an elaborately designed feedback plus feedforward control arithmetic is used. The propeller speed control block diagram is presented in Fig. 18.

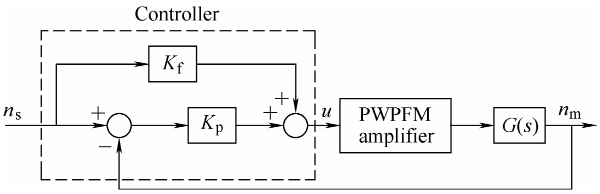

Fig. 18. Speed closed-loop control block diagram

The controller parameters have been turned by Ziegler-Nichols method first, and then by some trial and errors for fine adjusting. At last, the feedback gain K_p is 9 while the feedforward gain K_f is 1.5.

The propeller speed control results have been shown in Fig. 19, where n_m is the measured average speed, and n_s is the speed setpoint. The system has good dynamic and steady-state control performance in the entire working conditions. The transient time is only about 1 s to 1.5 s, while the steady-state error is less than 5%.

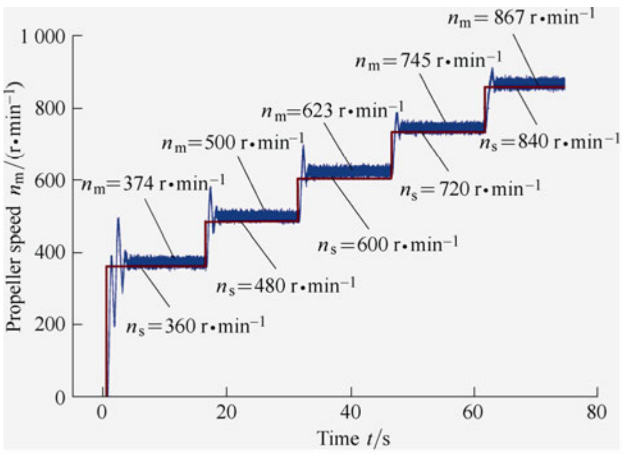

Fig. 19. Speed closed-loop control experiment results

6 Conclusions

(1) By experiment studies and practical applications on a ROV, the advantage of proposed high speed on/off valve control propeller in energy saving has been proved. The lowest efficiency is more than 40%. At the same time, the system also has an excellent speed control performance in the entire working conditions. The transient time is only about 1 s to 1.5 s, while the steady-state error is less than 5%. And the advanced hydraulic bus structure can be used to simplify the placement of pipes significantly.

(2) The experiment results prove that the fluctuation control method by adding a flywheel to filter speed fluctuation is effective. The smooth propeller speed control effect is achieved and the speed fluctuation has little influence on ROV control performance.

(3) The proposed novel PWPFM control method guarantees that the high speed on/off valve has a better frequency respond performance than the traditional PWM control method. And higher system efficiency is obtained by making a good tradeoff between efficiency and speed fluctuation degree.

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