Design, Analysis and Testing of a Hydraulic Catapult System

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ABSTRACT A hydraulic catapult system is proposed to simulate the movement of the missile in the launch tube, which mainly consists of a high-speed hydraulic cylinder, an accumulator group, a main valve, a servo valve and a cable pulley mechanism. The key technical difficulties are that how to improve the response speed of main valve of big flux and increase the catapult speed in limited stroke. An ingenious controlled valve is designed and an cable pulley mechanism is applied to the hydraulic lever. The system working principle and design method are discussed. Then a system mathematical model is established, and the characteristics of the controlled valve and ejection process are further investigated through numerical simulation. Theoretical results are finally compared with the experimental ones, and it is found that they are in agreement. Finally, PID control is applied to the hydraulic catapult system, and the experimental results show that the hydraulic catapult can accurately track the velocity commands. This paper provides a valuable reference for improving high performance hydraulic catapult system in the aerospace field.

INDEX TERMS Hydraulic catapult, controlled valve, cable pulley mechanism, ejection process, numerical simulation.

I. INTRODUCTION
In general, a ground-based testing is necessary to predict the performance indicators of missile launch before the practical application [1]. By simulating the actual working environment, the working status of the system can be detected. To research and detect the performance of missile launch, many simulating technology have been investigated [2]–[5]. The core part of the simulation equipment is the design of high-performance catapult. Currently, the main methods are cartridge catapult, slide catapult, pneumatic catapult and hydraulic catapult [6], [7]. As the aerospace industry evolves, missile launches require higher speed performance and responsiveness [8]. The ways of cartridge catapult and slide catapult can not achieve the required speed [9], [10]. Pneumatic catapult has characteristics of fast response and large output force, but the sealing of high-pressure cylinder brings difficulties to its control and maintenance work [11], [12]. Due to the advantages like large power-to-weight ratios, high response, high control precision and high load capability [13], [14], hydraulic catapult has been widely used in industrial production, such as crash test [15], [16] and UAV launch [17], [18]. However, the employment of the hydraulic catapult onto the simulation of missile launch is rather new.

Usually, the basic idea of hydraulic catapult is to drive the missile through a hydraulic cylinder controlled by a servo valve. Zhejiang Institute of Technology designed a high-performance hydraulic catapult with an speed of 8 m/s and a load of 230 kg, and the feasibility of the principle was verified by a prototype [19], [20]. [10] proposed a hydraulic catapult mechanism, whose design index was that the speed of the piston reached more than 2 m/s at the initial 20 mm strokes and the maximum speed reached 15 m/s. Beihang University proposed a high-performance hydraulic ejection mechanism consisting of high-speed hydraulic cylinder, two-bar mechanism, accumulator, main valve, sensor, etc. The ejection speed can reach 11 m/s within 63 ms [21]. EDO company produced the AVEL catapult with a folding arm, it was driven at a speed of 7.6 m/s by a hydraulic cylinder within 0.1 s [22]. COBHAM company produced DMMEL catapult, its ejection speed was about 8.9 m/s within 0.3 s [23]. Although these studies provide some key informations about improving the speed of the catapult, they still cannot meet the speed requirements of “the missile” of 16 m/s.

To improve the speed of hydraulic catapult, the key technologies are the design of the servo valve of big flux and fast response and the design of the high-speed hydraulic...
cylinder. In 1946, Tinsley in the UK registered the first patent for a two-stage valve, which overcame the bottleneck of insufficient thrust in a single-stage structure [24]. Bell Aircraft has subsequently developed two-stage valves and the structural performance of the two-stage valve has been improved [25]. In 1957, Atchley designed a servo valve based on the principle of jet pipe and the reliability of the valve has been improved [26], [27]. At the same time, a three-stage structure servo valve appeared to meet the requirements of the system for large flow [28], [29]. Moog has developed a three-stage jet pipe servo valve with 0-100% input signal step response time of 20 ms and rated flow of 3900 L/min. BoschRexroth has developed a nozzle-flapper servo valves with 0-100% input signal step response time of 70 ms and rated flow of 2800 L/min. Parker has developed a moving coil solenoid proportional valve with a step response time of 20 ms and rated flow of 6800 L/min. These mentioned servo valves are two- or three-stage valves which increase flow and response speed by improving the performance of the pilot valve. And its high manufacturing and maintenance costs limit their wider application. For hydraulic cylinder, sealing technology and high speed buffer structure are the main factors that limit the speed of cylinder piston. When the cylinder moves faster, the sealing element is easier to be extruded out of the sealing groove. Therefore, the limit speed of cylinder is 15 m/s [30]. At present, the high speed hydraulic cylinder is basically multi-stage hydraulic cylinder. Zhejiang Provincial Key Laboratory of Modern Textile Machinery Technology designed a multi-stage synchronous hydraulic cylinder for high speed hydraulic catapult [31]. The speed of hydraulic cylinder can reach 20 m/s, but the multi-stage structure is complex and the processing cost is high.

This paper proposes a hydraulic catapult system that combines hydraulic system and cable pulley mechanism. And its manufacturing and maintenance costs are greatly reduced. In order to improve the performance of hydraulic catapult, an innovative controlled valve is proposed, which uses two spool valve ports in parallel to increase the area gradient of the valve port and solve the contradiction between high frequency response and large flow. And the valve dynamic characteristics are studied through numerical simulation. Cable pulley mechanism is connected to the hydraulic cylinder. It is used in reverse to enlarge the speed of the hydraulic cylinder. And the speed of catapult is much greater than the speed of the hydraulic cylinder. Therefore, the catapult scheme solves the problem of the hydraulic cylinder limiting the catapult speed to less than 15 m/s. The dynamic simulation of cable pulley system is addressed with accurate modeling the motion of the flexible cable with time-varying length and coupling motions between the cable and the pulleys instead of eliminating pulleys or simplifying as a point [32], [33]. Based on this, parameters influencing the catapult speed are analyzed. Finally, the hydraulic catapult is built to test and PID control is applied to track the missile velocity commands. Hydraulic catapult mechanism belongs to the high-speed and high-power hydraulic system. As hydraulic equipment is developing towards high-speed, high power, low noise, the study on the above key technologies will have a wide range of application prospect.

II. METHODS
Hydraulic catapult is composed of hydraulic system, cable pulley mechanism and hydraulic braking mechanism. Hydraulic system acts as the drive of the catapult system, continuously accelerating the load during the ejection. The catapult speed is determined by the flow rate of rodless cavity of the hydraulic cylinder. The greater the flow, the faster the catapult speed. To increasing the flow of rodless cavity, the controlled valve adopts two on-off valves of big flux in parallel, which are controlled by a servo valve. And the solenoid valve is specifically use for adjusting the initial position and the final unloading of hydraulic cylinder. Hydraulic cylinder is connected with cable pulley mechanism to enlarge the catapult speed. The principal diagram is shown in figure 1.

A. CONTROLLED VALVE
The controlled valve with high frequency response and large flow rate is a key control component in hydraulic catapult system. Generally, high frequency response and large flow are common contradiction in hydraulic valve design. Theoretically, there are two ways to increase the valve flow rate: increasing the working stroke of the spool or increasing the diameter of the spool. However, both methods reduce the response speed of the valve. Therefore, this paper proposes an innovative controlled valve, as shown in Figure 2, in which the main valve controlled by a servo valve adopts a double throttle parallel output structure. The oil flows into two slide valve port at the same time, doubling the flow capacity without increasing spool stroke, so that the response speed of the valve is almost unaffected. In addition, a dead zone is designed for the valve, and the length of the dead zone not only plays the role of the spool sealing gap, but gives the spool a pre-acceleration process to increase the opening speed of the valve port and the valve outlet pressure. As shown in figure 2, the cavity $A_c$ and $B_c$ are controlled by a servo valve.
In order to improve response speed, the effective action area of cavity $A_c$ is larger than that of cavity $B_c$. Thus, the force applied in cavity $A_c$ is larger and the spool responds more quickly.

**B. CABLE PULLEY MECHANISM**

Cable pulley mechanism consists of a fixed pulley group, a movable pulley group, a guide pulley and a steel wire rope. The movable pulley group is connected to the hydraulic lever. It moves along with the mechanical connector, and drives the load through a wire rope. As shown in figure 3, cable pulley mechanism expands the stroke route of the wire rope, thus the speed of the hydraulic cylinder is amplified.

**C. HYDRAULIC BRAKING MECHANISM**

During the ejection process, the hydraulic cylinder must be properly cushioned and braked to avoid strong impact and vibration. An internal cushion structure increases the length of hydraulic cylinder and also requires higher manufacturing costs. Therefore, this paper adopts an external cushion structure as shown in figure 4. It is composed of two hydraulic dampers and two polyurethane buffers. The external cushion work with the hydraulic cylinder to achieve cushioning braking.

**III. MATHEMATICAL MODEL**

**A. CONTROLLED VALVE**

The equivalent diagram is shown in Figure 5, where the servo valve acts as the pilot valve, while the main valve can be equivalent to two on-off valves in parallel. In the movement of the spool, it will be subject to steady-state flow force. The force is directly related to the diameter and flow rate of the spool, so for on-off valve with large flow rate, the steady-state flow force cannot be ignored. Steady-state flow force can be described by

$$F_p = \rho Q v \cos \theta,$$

where $\rho$ stands for the density of hydraulic oil, and $Q$ stands for total flow rate of control cavity. $\theta$ stands for jet angle.

When flow coefficient is 0.65, $\theta$ is 69°, steady state flow forces is

$$F_p = 0.43\omega x_p \Delta p = k_f x_p,$$

where $\omega$ stands for area of the gradient, $\Delta p$ is differential pressure at both ends of valve port, $k_f$ stands for steady state hydrodynamic stiffness.

According to Newton’s inertial law, the dynamic equation can be written as

$$P_m A_m - P_n A_n = m_n \ddot{x}_p + B_c \dot{x}_p + (k_f + k_n) x_p,$$

where $A_m$ and $A_n$ stand for the action areas of the control cavity $A$ and control cavity $B$. $m_n$ stands for the mass of the spool, respectively; $P_m$ and $P_n$ stand for the pressures of the control cavity $A$ and control cavity $B$, respectively; $B_c$ stands for the equivalent damping coefficient; $k_n$ stands for the spring stiffness; $x_p$ stands for the spool displacement.

According to the motion relation and the leakage relation, the flow continuity equation of the available hydraulic oil can
TABLE 1. Parameters of the controlled valve.

| Parameters                      | Symbols | Value | Unit |
|---------------------------------|---------|-------|------|
| Spool mass                      | \(m_s\) | 3.0   | kg   |
| Action areas of the control cavity \(A\) | \(A_s\) | 213.6\*2 | mm\(^2\) |
| Action areas of the control cavity \(B\) | \(A_n\) | 703.7\*2 | mm\(^2\) |
| Spring stiffness                | \(K_s\) | 40    | N/mm |
| Maximum displacement            | \(x_p\) | 10    | mm   |

be expressed as

\[
\begin{align*}
Q_m &= A_m \dot{x}_p + \frac{V_m}{E} \dot{P}_m \\
Q_n &= A_n \dot{x}_p - \frac{V_n}{E} \dot{P}_n,
\end{align*}
\]

(4)

where \(Q_m\) and \(Q_n\) stand for the flow rates of control cavity \(A\) and control cavity \(B\), respectively; \(V_m\) and \(V_n\) stand for the controlled volumes of control cavity \(A\) and control cavity \(B\), respectively, \(E\) stands for effective volume modulus of hydraulic oil.

Assume the back pressure is zero. When \(x_v \geq 0\), according to the thin-walled orifice throttling principle, the flow equation can be written as

\[
\begin{align*}
Q_m &= C_d \omega \sqrt{\frac{2}{\rho} (P_s - P_m) \cdot x_v} \\
Q_n &= C_d \omega \sqrt{\frac{2P_n}{\rho} \cdot x_v},
\end{align*}
\]

(5)

where \(\omega\) stands for the opening gradient; \(C_d\) stands for the shape coefficient and \(\rho\) stands for the density of hydraulic oil.

According to equations (2)–(5), a set of the first-order differential equations for the controlled valve are derived.

\[
\begin{align*}
\frac{V_m}{E} \dot{P}_m &= -A_m \dot{x}_p + C_d \omega \sqrt{\frac{2}{\rho} (P_s - P_m) \cdot x_v} \\
\frac{V_n}{E} \dot{P}_n &= -A_n \dot{x}_p - C_d \omega \sqrt{\frac{2P_n}{\rho} \cdot x_v} \\
m_s \ddot{x}_p &= P_m A_m - P_n A_n - B_c \dot{x}_p - (k_f + k_n) x_p,
\end{align*}
\]

(6)

Since the displacement and speed of the spool directly determine the ejection speed and response speed, the main factors affecting the displacement are analyzed next.

**B. ANALYSIS OF SPOOL DISPLACEMENT**

The structure parameters of the controlled valve is shown in Table 1. Through simulation, the effect of pilot valve response time on on-off valve is obtained and the simulation results are shown in figure 6. In the simulation, \(P_s\) is 15 Mpa, the step response time of servo valve are 10 ms, 15 ms, 20 ms, respectively. It can be seen that the response time of the servo valve has a significant impact on that of the on-off valve.

The response time of servo valve is 10 ms, working pressure are 10 Mpa, 15 Mpa, 20 Mpa, respectively. As shown in figure 7, the greater working pressure, the faster response of on-off valve. And along with the increasing of pressure, the influences become smaller.

The response time of servo valve is 10 ms, working pressure is 15 Mpa, masses of valve spool are 2.0 kg, 3.0 kg, and 4.0 kg, respectively. As shown in figure 8, spool mass has little influence on dynamic characteristics of on-off valve.

The purpose of setting dead zone is to reduce the leakage of the valve at the zero displacement, and to improve the pressure gain of the valve outlet, which is very necessary in the catapult with the initial speed requirement. However, the dead zone also prolongs the response time of the valve. For
In view of the tension of wire rope is uneven, wire rope is divided into several sections with the pulley as the dividing line. Besides, the rope between two pulleys is also cut into several sub-parts and each sub-part moves independently along the rope axis. As shown in figure 11, the wire rope is cut into several sub-parts and each sub-part moves independently along the rope axis. The longitudinal dynamic model of wire rope established is based on the following conditions: air flow is not considered; regard a multi-strands wire rope as a single-strand wire rope with the equivalent cross-sectional area. According to these assumptions, the instantaneous values for total stiffness, viscous friction and inertia of the rope depend on the length of the rope is

$$\begin{align*}
    k_i &= k_0/(l_i + l_1 + l_2) \\
    b_i &= b/(l_i + l_1 + l_2) \\
    m_r &= \rho_i(l_i + l_1 + l_2) \\
    l_1 &= f \int r_i \omega_i dt \\
    l_2 &= f r_i \omega_i dt,
\end{align*}$$

(8)

where $k_0$ stands for the initial stiffness of the unit length of rope, $b_0$ refers the initial viscous friction of the unit length of rope; $\rho_i$ denotes mass of the rope per unit meter; $l_i$ denotes the length between movable and fixed pulleys; $l_1$ and $l_2$ stand for uncoiling and coiling length of rope end, respectively. $\rho_i$ refers mass of the rope per unit meter.

The velocity of each subpart is estimated as

$$\dot{v}_i = n(F_i + 1 - F_i)/m_r - g\sin(\varepsilon),$$

(9)

where $v_i$ and $F_i$ stand for the velocity and the force of the subpart $i$, respectively; $m_r$ refers the mass of the rope; $g$ denotes the acceleration due to gravity; when the inclination of the rope is 90° the input port is the highest, $\varepsilon > 0$; otherwise the input port is the lowest, $\varepsilon < 0$; when the inclination of the rope is 0, $\varepsilon = 0$. The lengthening between 2 sub-parts is a state variable, which derivative depends on the relative velocities.

$$\begin{align*}
    \dot{x}_i &= v_i - v_i \\
    \dot{x}_1 &= v_{out} - v_1 \\
    \dot{x}_{i+1} &= v_i - v_{in},
\end{align*}$$

(10)

where $x_i$ is the lengthening of the subpart $i$. 

Cable pulley mechanism is used to enlarge the hydraulic cylinder speed. Assume that there is no relative slip between the rope and pulleys, the motion of the contact segment of the rope can be described by the kinematics parameters of pulleys. As shown in figure 10, when movable pulleys are driven by cylinder, the rope tension between two different pulleys are not equal, thus the speed and lengthening are also inequality. The ranges of forces that rope acting on four movable pulleys are gradually degressive or progressive. The pulleys cannot be ignored when modeling the motion of the time-varying length rope.

The mathematical model of the motion characteristics of the contact segment with movable pulley and fixed pulley can be obtained as

$$\begin{align*}
    v_{i1} &= v_{i3} + r_i \omega_i \\
    v_{i2} &= v_{i3} - r_i \omega_i \\
    F_{i1} &= 0.5F_{i3} + 0.25m_r r_i \omega_i - m_i \dot{v}_{i3} \\
    F_{i2} &= 0.5F_{i3} - 0.25m_r r_i \omega_i - m_i \dot{v}_{i3},
\end{align*}$$

(7)

where $v_{i1}, v_{i2}, v_{i3}$ and $\omega_i$ stand for uncoiling, coiling, input and angular rotation speed of fixed and movable pulleys, respectively; $F_{i1}, F_{i2}, F_{i3}$ stand for the uncoiling, coiling and input force of fixed and movable pulleys, respectively; $m_i$ stands for pulley mass; $a$ stands for the input acceleration of movable pulleys; $r_i$ stands for the radius of the pulleys.
According to these lengthenings, the spring and damper forces are computed as
\[ F_i = -(K_i x_i + B_i \dot{x}_i). \]  
(11)

The force at input and output can be deduced by the following relation
\[
\begin{align*}
F_{\text{in}} &= F_n \\
F_{\text{out}} &= F_1
\end{align*}
\]  
(12)

When the force acting on the load is greater than the pretightening force, the catapult speed is
\[ F_{\text{in}} + Mg = Mv_{\text{end}}. \]  
(13)

D. DYNAMIC CHARACTERISTICS OF CABLE PULLEY MECHANISM

For the cable pulley mechanism transfer process, based on above motion models of the movable pulley, fixed pulley, wire rope and the load, a full nonlinear transfer process model is established as shown in figure 12.

Dynamic characteristics of cable pulley are mainly related to \( F_p \), \( v_h \), \( a_h \), \( l_{\text{in}} \), \( l_{\text{e}} \), and \( n \). The speed signals \( v_h = a_0 t \), \( a_h = g \sin 5\pi t \). The total length of the rope \( l = l_e + 8l_{\text{in}} \).

As shown in figure 13, the end speed is directly determined by the total length \( l \) and is almost unaffected by the distribution of the rope length. The smaller the value of \( l \), the faster the end speed. From figure 14, it can be intuitively seen that: the longer the rope, the heavier hysteresis and the more violent the acceleration curve fluctuations.

The preload is applied to the end of the rope. As shown in Figures 15 and 16, the greater the preload, the slower the response speed. In the initial stage, the speed decreases with the increase of the preload, but after a period of time, the greater the preload, the faster the speed.

The value of \( l_e \) is 5 m and value of \( l_{\text{in}} \) is 0.3 m, each section of the rope is cut into \( n \) sub-parts. When \( n = 0 \), the rope is treated as a pure spring, the inertial of the rope is ignored. As shown in figure 17 and figure 18, value of \( n \) has little influence on end speed and response time. When \( n \) is not equal to 0, a larger fluctuation of acceleration occurs. And the higher value of \( n \), the heavier computation.
E. VALVE CONTROLLED CYLINDER

The dynamic characteristics of hydraulic system directly determine the performance of the catapult system. The simplified schematic diagram is shown in figure 19.

According to Newton’s inertial law, the force balance equation of the cylinder and the load is

\[
\begin{align*}
    P_1A_1 - P_2A_2 &= m_h\ddot{x}_h - B_h\dot{x}_h + F_t \\
    F_t &= F_{13} + F_{33} + F_{53} + F_{73},
\end{align*}
\]

where \(A_1\) and \(A_2\) stand for the pressure action areas of the rodless and rod chamber of the hydraulic cylinder, respectively; \(P_1\) and \(P_2\) stand for the working pressures of the rodless and rod chamber of the hydraulic cylinder, respectively; \(m_h\) stands for the equivalent load mass; \(B_h\) stands for the equivalent damping coefficient; \(x_h\) stands for the piston displacement.

Assuming that there is no leakage, the flow continuity equation of the available hydraulic oil is

\[
\begin{align*}
    Q_1 &= A_1\dot{x}_h + \frac{V_1}{E}\dot{P}_1 \\
    Q_2 &= A_2\dot{x}_h - \frac{V_2}{E}\dot{P}_2,
\end{align*}
\]

for the equivalent load mass.
where $Q_1$ and $Q_2$ stand for the flow rates of the rodless and rod chamber of the hydraulic cylinder, respectively; $V_1$ and $V_2$ stand for the controlled volumes of the corresponding cavity, respectively.

Assume the back pressure is zero. According to the thin-walled orifice throttling principle, the flow equation can be written as

$$
\begin{align*}
Q_1 &= C_d \omega_1 \sqrt{\frac{2}{\rho} (P_s - P_1) \cdot x_p}, \\
Q_2 &= C_d \omega_2 \sqrt{\frac{2}{\rho} (P_2 - P_r) \cdot x_p}.
\end{align*}
$$

The opening gradient $\omega_i$ and shape coefficient $C_d$ of controlled valve are known constants and $\rho$ stands for the density of hydraulic oil.

Suppose that the gas in the accumulator satisfies the adiabatic equation during the impact process, the equation is

$$
P_s \cdot V_s = P_{s0} \cdot V_{s0} = P_0 \cdot V_0^r = C_a,
$$

where $V_0$ and $P_0$ are the effective volume and precharging pressure of the accumulator; $V_{s0}$ and $P_{s0}$ stand for the gas volume and pressure of the accumulator after filling liquid, respectively; $V_s$ and $P_s$ stand for the gas volume and pressure of the accumulator during the impact, respectively; $r$ stands for the specific heat capacity of the gas; $C_a$ stands for an adiabatic constant.

After linearizing the equation (18), substitution (18) can be rewritten

$$
\frac{dP_s}{dt} = -\frac{C_a r}{V_s^{r+1}} \frac{dV_s}{dt}.
$$

**F. INFLUENCE ANALYSIS OF ACCUMULATOR PARAMETERS**

The research of high-power catapult system mainly focuses on high-speed, light-load or low-speed, large-load. For system of high-speed and large-load, one of the difficulties is how to achieve the instantaneous energy supply. This section...
is to study the variation rules of the dynamic characteristics at different working pressures, volumes, gas pressures and number of accumulators. As shown in figure 20 and 21, simulation results indicate that accumulator working pressure has a great influence on the catapult performance. As working pressure increases, the velocities and accelerations basically increase in equal proportion. Figure 21 shows cable pulley starts work since 0.01 s, working pressure cannot improve the hysteresis of the rope.

As shown in figure 22 and 23, the results of simulation are that gas precharge pressure and volume have little influence on catapult performance.

According to the analysis above, working pressure, precharging pressure and volume of accumulator are set to 150 bar, 100 bar, 100 L, respectively. Since single large-capacity accumulator requires more manufacturing costs, this paper applies an accumulator group instead of a single one. By comparison, the simulation results show that catapult speed with an accumulator group is faster. Figure 24 and 25 shows that the time of rope acting on load is nearly unchanged before 0.01 s.

IV. EXPERIMENT AND SIMULATION

Overall considering the characteristics of response time, catapult speed and acceleration, this article adopts an accumulator group composing of four 25 L accumulators in parallel. The initial working pressure, precharging pressure and volume are 150 bar, 100 bar and 100 L, respectively. The step response time of the servo valve is 10 ms. The values of $I_e$, $I_m$ are 5 m, 0.3 m. The system adopts two wire ropes to drive in parallel. The value of $K$ is approximately equal to 1000000, nodes of each section is 1 and $F_p$ is 3000 N. The test bench is established as shown in figure 26. System is powered by a motor-driven hydraulic plunger pump. Accumulator group is used as energy storage device for supplying fluid. Hydraulic
cylinder is controlled by a controlled valve of big flux, cable pulley mechanism is combined to the hydraulic lever.

To verify the effectiveness of the proposed design, the experiments are conducted under different input signals and compared with the simulation. The catapult platform is first tested for a step signal $u = 10$, and the comparison results are shown in Figure 27. The simulated results of displacement, velocity and acceleration are in good agreement with the measured results.

To further test the effectiveness, the ramp signal $u = 40t$ is used. The comparison results are shown in Figure 28. The simulated results of displacement, velocity and acceleration are also in good agreement with the measured results.

Based on the comparison results, the correctness of the simulation modeling in this article can be proved. Since the experimental results are more stable in the ramp signal case, PID control is used to control the slope of the ramp signal to accurately track the velocity commands. The motion trajectory is $v_{end} = -14sin(5.03t+2.618)+6.963$, PID parameters are set as follow: $k_p = 1.6$, $k_i = 1$, $k_d = 0$, which represent the P-gain, I-gain, and D-gain of velocity. The experimental results in PID are shown in figure 29.

V. CONCLUSION

An experimental setup has been developed and validated. Dynamic characteristics of the catapult are analyzed by simulation, and parameters that affect the performances of the catapult are optimized. Conclusions are drawn as follows:

1. The experimental results are consistent with the simulation ones. The hydraulic catapult program and mathematical model proposed in this paper are demonstrated to be effective.

2. The main valve designed in this paper adopts double throttle parallel output structure and is controlled by a servo valve, which can effectively increase the flow of the valve. And the response time of main valve with big flux is approximately 10 ms.

3. Accumulator group as the energy source, can afford the instantaneous high-power source with compact structure. Compared with an single accumulator, accumulator group can improve the response time. And increasing the initial working pressure of the accumulator group is one of the effective ways to increase the velocity of the catapult, while its precharging pressure and volume have little effect on the performance of the catapult.

4. Coupling motions between the cable and the pulleys are established, and tension of the wire rope is modeled in sections with the pulley as the dividing line. The modeling method will be beneficial to the intending theoretic research of rope driving.

All of these results will be beneficial to develop hydraulic technology in high performance catapult. At the same time, they provide some useful ideas and means for the next research about vibration suppression and control.

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