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The Closed-loop Control Strategy for Precise Low Pressure of Powder Pressing Equipment

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Abstract

The multistage compression is usually adopted for large tonnage powder pressing equipment to improve product quality. The first compression is the key process during the powder processing as the target pressure is small and the load is great fluctuant. The pressure in the first compression is likely to be much overshot that makes laminated products if conventional control strategy is applied. The control parameters of routine mode have bad adaptation because they change with the target pressure. In order to improve the accuracy of the pressure and adaptability of control parameters in the first compression, one strategy oriented to engineering application that includes data testing, parameter setting and control optimizing has been proposed. Real-time monitoring system with multi-channels based on LabVIEW is built, and the full-state of equipment can be acquired based on the system. The working statuses of control elements during the pressing process due to inertial force are optimized with the experimental data, and the segmented variable gain control strategy based on parameterization is developed and applied. The experimental results with the strategy during the first compression indicate that the pressure precision is raised from 0.3MPa and above (conventional opened-loop control) to 0.05MPa and below, and the pressing time is controlled in 300ms and below. The control strategy ensures the high accuracy and efficiency of pressing process and it is convenient to be applied in engineering.

***Keywords**: Powder pressing; electro-hydraulic control; pressure closed-loop; data acquisition; segmented variable gain*

1. Introduction

Large tonnage powder pressing equipment (more than 4000 tons), is an important modern industrial production equipment, which is widely used in the field of powder forming composites, metal powders, ceramics, and other wear-resistant materials. The performance of equipment directly determines the quality of the finished products [1-3].

In order to improve the mechanical properties and ensure the quality of the products, the molding process of powder pressing typically requires multistage compression [4]. Pressing process is generally divided into the first compression, secondary compression and final compression (some powder type requires more graded suppression). Among them, the first phase of compression is the basic shaping process of the powder, which is the key link for the pressing process and usually low-pressure compression. The control precision of the pressure in the low-pressure stage directly affects the quality of finished products [5]. However, there are two characteristics in the first compression process: first, the loose powder is gradually becoming compacted that load pressure is small and easy to fluctuate [6]; second, the hydraulic system of large tonnage equipment always needs the larger hydraulic components to match the requirements of high-pressure compression (pressure can be up to 10 times more than the low pressure), such that valve specification cannot optimum matching low pressure requirements in the first pressure compression stage. These characteristics make control precision unstable when open-loop control or

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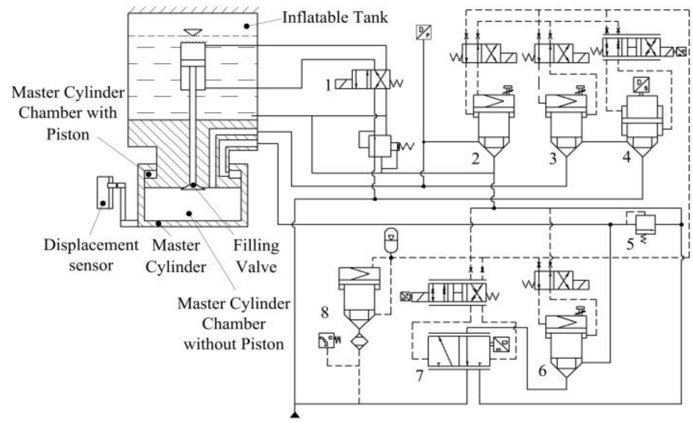
conventional PID pressure control is used, and the adaptability of control parameters is poor in the varying load conditions. Therefore, the study of advanced control strategies with high precision, strong adaptable parameters and large engineering applicable potential is important to ensure high-quality pressed powder molding process.

In this paper, a certain type of large tonnage powder molding equipment was the object for the study. The hydraulic circuit was analyzed to study the key stages that effect the first compression, then, a corresponding mathematical model was built to provide a theoretical basis for the control strategy optimization. Real-time monitoring system with multi-channels based on LabVIEW was built to ensure the effectiveness of subsequent control strategy and the convenience parameter turning. On the basis of the measured data for each state of the press, the operating state of control components in the inertia pressing process was optimized and the segmented variable gain control strategy was designed, which effectively increased the precision of pressure control and efficiency of compression. The control strategy provides an effective method of accurate pressure control for powder pressing equipment in the first pressing process control for practical engineering.

2. Working Principle

The working principle on key parts of a large tonnage powder molding equipment is shown in Figure 1. The basic operation of the device is including: upper-mould down, inertial press, the first compression / the first exhaust, the second compression / the second exhaust, final compression (pre-press, pressure boost, pressure maintaining), pressure relief, unloading and upper-mould return.



1 two position four way reversing valve, 2, 3, 6, 8 two-way cartridge valves, 4 two-way proportional cartridge valves, 7 three-way proportional cartridge valves

Figure 1. Hydraulic Schematic on Key Parts of a Large-Tonnage Powder Pressing Equipment

When the pressing starts, the valve 8 is held state of through the oil to provide pilot pressure. The master cylinder is reset to the upper dead center position. The master cylinder chamber without piston connected with a large inflatable tank due to the valve 1 is operated in the right position and the filling valve is open. Proportional valve 7 is in the left position, the valve 2, valve 3 and valve 4 are all in the closed state, the oil of the small chamber of the master cylinder back into the tank through valve 6 and valve 7, so the master cylinder drops down depend on the pressure of inflatable tank and its gravity. The falling speed is controlled by the amount of proportional valve 7's opening.

After the master cylinder passes the pressurized position, the press will work in the stage of inertia pressing. At the switching moment, the proportional valve 7 is moving to

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the left position with maximum amount of opening, thus the resistance of discharging oil of the master cylinder chamber with piston is minimum. Meanwhile filling valve, the valve 2, valve 3, valve 4 are all closed. The master cylinder is pressing the powder by inertia force until the instruction of first compression is received. At the first compression, the proportional valve 4 and cartridge valve 3 are enabled, the pressure oil is flowing into the master cylinder large chamber through the valve 4 and valve 3 so that chamber's pressure keeps raising. The rate of pressure rising can be controlled by adjusting the opening amount of proportional valve 4. When the pressure reached the set value, the proportional valve 4 and the valve 3 are changing to closed state. Then, the valve 2 is enabled and valve 7 is turned into the right position. The master cylinder move up little to achieve air exhaust.

3. Mathematical Model

The above working principle indicates that the core part of the first compression is the proportional cartridge valve controlled cylinder system. So the low-pressure controlled loop model can be simplified as two-way valve controlled unsymmetrical cylinder shown in Figure 2. Because the powder particles are mostly in elastic deformation during the low-pressure compression stage, the powder load can be replaced by a two-step spring damping load.

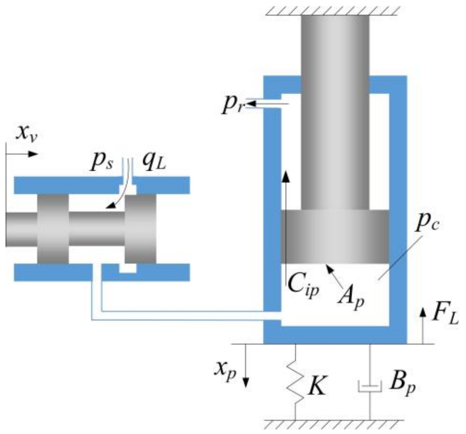

Figure 2. Simplified Model of Low-Pressure Hydraulic Control System

The parameters in Figure 2 are defined as follows: x_v is the spool displacement of two-way valve; x_p indicates the displacement of master cylinder; A_p means the hydraulic effective area of the big chamber in the master cylinder; F_L denotes the external load acting on the master cylinder; K presents the powder load stiffness, B_P describes powder load damping; q_L shows the flow rate that flows through the two-way valve; p_c expresses the pressure in the master cylinder's big chamber; p_s signifies the oil supply pressure; p_r represents the return oil pressure; C_{ip} stands for internal leakage coefficient of the master cylinder.

In addition to the above parameters, the area gradient at valve port of two-way valve, the total mass of the master cylinder, the volume of the master cylinder's big chamber, the flow coefficient, the oil's density; the bulk modulus of oil elasticity are respectively set as w, m, V_0, C_d, ρ, β_e. Besides, the return oil pressure p_r is set to zero. The flow equation for the two-way valve, the flow continuity equation for the master cylinder's big chamber are respectively built as formula (1)-(3) [7], which are based on the principle of the simplified model shown in Figure 2.

$$q_L = C_d w x_v \sqrt{\frac{2(p_s - p_c)}{\rho}} \tag{1}$$

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$$q_L = A_p \frac{dx_p}{dt} + C_{ip} p_c + \frac{V_0}{\beta_e} \frac{dp_c}{dt} \tag{2}$$

$$p_c A_p = K x_p + B_p \frac{dx_p}{dt} + m \frac{d^2 x_p}{dt^2} + F_L \tag{3}$$

The above equations are linearized with small deviation around operated point and used Laplace transformation. Then the following equations are derived.

$$Q_L(s) = K_q X_V(s) - K_c P_c(s) \tag{4}$$

$$Q_L(s) = A_p s X_p(s) + C_{ip} P_c(s) + \frac{V_0}{\beta_e} s P_c(s) \tag{5}$$

$$P_c(s) A_p = (K + B_p s + m s^2) X_p(s) + F_L(s) \tag{6}$$

Where s is the Laplace operator; $Q_L(s)$, $X_V(s)$, $P_c(s)$, $X_p(s)$, $F_L(s)$, respectively, present Laplace transformation for q_L, x_v, p_c, x_p, F_L; K_q and K_c respectively indicate flow gain and flow-pressure coefficient of valve port of two-way valve. The transfer function is built as formula (7), which is based on the transfer function block diagram shown in Figure 3.

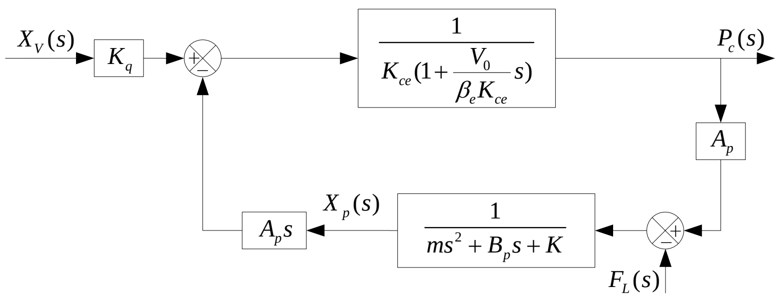

Figure 3. Transfer Function Block Diagram of Two-Way Valve Controlled Asymmetric Cylinder

$$P_c(s) = \frac{\frac{K_q}{K_{ce}}\left(\frac{s^2}{\omega_m^2} + \frac{2\delta_m}{\omega_m} s + 1\right) X_v(s) + \frac{A_p}{K K_{ce}} s F_L(s)}{\left(1 + \frac{s}{\omega_r}\right)\left(\frac{s^2}{\omega_0^2} + \frac{2\delta_0}{\omega_0} s + 1\right)} \tag{7}$$

The total flow-pressure coefficient $K_{ce} = K_c + C_{ip}$;

The hydraulic resonant frequency $\omega_h = \sqrt{\frac{\beta_e A_p^2}{V_0 m}}$;

The hydraulic spring stiffness $K_h = \beta_e A_p^2 / V_0$;

The mechanical resonant frequency $\omega_m = \sqrt{K / m_t}$;

The damping system of load (dimensionless) $\delta_m = \frac{B_p}{2\sqrt{m_t K}}$;

The corner frequency caused by synthetic stiffness $\omega_r = \frac{K_{ce}}{A_p^2\left(\frac{1}{K} + \frac{1}{K_h}\right)}$;

The synthetic resonant frequency $\omega_0 = \omega_h \sqrt{1 + K / K_h}$.

In the actual system, external load disturbance is quite small that it can be ignored and the damping coefficient at the low-pressure stage can be considered to be zero. The

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dynamic response of the two-way cartridge valve can be described as a second-order oscillation link shown in formula (8).

$$\frac{X_v(s)}{U_0(s)} = \frac{K_a}{(\frac{1}{2\pi f})^2 s^2 + 2\frac{1}{2\pi f}\xi s + 1} \tag{1}$$

Where $U_0(s)$ is the voltage signal of proportional valve; f indicates frequency response of proportional valve; K_a presents the gain between displacement of valve spool and input voltage.

The transfer function relationship between the pressure in the master cylinder's big chamber and the voltage signal of proportional valve is shown in formula (9).

$$\frac{P_c(s)}{U_0(s)} = \frac{K_a \frac{K_q}{K_{ce}}(\frac{s^2}{\omega_m^2} + \frac{2\delta_m}{\omega_m}s + 1)}{(1+\frac{s}{\omega_r})(\frac{s^2}{\omega_0^2} + \frac{2\delta_0}{\omega_0}s + 1)(\frac{1}{4\pi^2 f^2}s^2 + 2\frac{1}{2\pi f}\xi s + 1)} \tag{2}$$

Several key parameters of the above model are listed in Table 1.

Table 1. Key Parameters of Mathematical Model

Parameters/Unit	Value	Parameters/Unit	Value
m/kg	13120	β_e/Pa	7×10^8
A_p/m^2	1.227	K_q/ m^3/s	0.024
V_0/ m^3	1.696	K_{ce}/ m^3/(s·Pa)	4.55×10^{-7}
A_2/m^2	1.227	ρ/kg/ m^3	900
f/Hz	20	ξ	0.7

Equation (9) shows the system is zero type high order system of which output is the pressure in the master cylinder's big chamber and input is the voltage signal of proportional valve. If open-loop step control mode is adopted in the system, it will be oscillatory and the control precision isn't ideal. Due to the strong nonlinear characteristics of the hydraulic control system, it is difficult to find the optimal control algorithm by using the mathematics directly. The control parameters can be optimized aiming at the engineering facts to improve the control performance of the control strategy and parameter tuning efficiency based on the mathematical model theory and the measured operating data.

4. The Design of the Monitoring System

The control strategy optimization requires comprehensive and precise press performance data that record pressure, cylinder displacement, solenoid valve and proportional valves signal instructions as well as others essential parameters during equipment is operating as a foundation. The monitor system of traditional powder pressing equipment acquires various sensors' data by PLC transferring to the touching screen to display. Though parts of the equipment performance data can be acquired by this way, which leads the complex performance analysis of control strategy, the difficult parameter tuning and the long R&D cycle due to its large limitation in scanning frequency, visualization and data processing. It is extremely essential to design a full-state and built-in type monitoring system which combines control and measurement specifically for powder suppression equipment [8].

4.1. The Hardware Design of Monitoring System

The hardware acquires physical quantity data from a variety of sensors. The data is transferred to computer in real-time after signal conditioning. According to the signal type and amplitude (Table 2), Advantech USB-4711/A is chosen to be the hardware platform whose sampling rate is 150kS/s and sampling precision is high enough. The general hardware structure of the monitoring system is shown in Figure 4.

Table 2. Signal Parameters from Hardware

Signal source	Signal voltage range	Value range of determined physical quantity
Master cylinder displacement sensor	(4-20)mA	(0-300)mm
Turbocharger displacement sensor	(0-10)V	(0-450)mm
Master cylinder pressure sensor	(0-10)V	(0-40)MPa
System pressure sensor	(0-10)V	(0-60)MPa
Proportional valve 4 instruction	(0-10)V	(0-10)V
Proportional valve 4 feedback	(0-10)V	(0-10)V
Proportional valve 7 instruction	±10V	±10V
Proportional valve 7 feedback	±10V	±10V

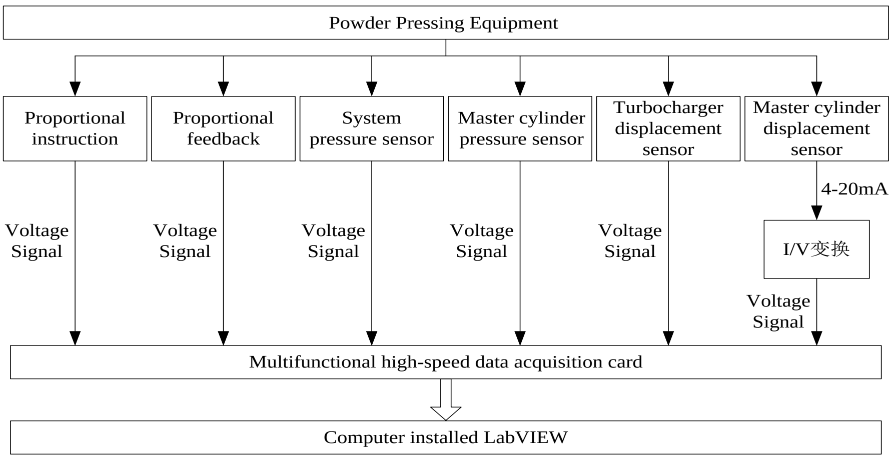

Figure 4. Hardware Structure of the Monitoring System

4.2. The Software Design of Monitoring System

LabVIEW is widely used in engineering application [9-11]. The software platform of monitoring system is based on LabVIEW, which composed of the above hardware platform with multi-channels real-time monitoring system. According to the function requirement, the system is divided into parameter setting, data acquisition, data storage, data playback and data processing *etc*. The software structure of system is shown in Figure 5. The operating panel and main program block diagram are respectively shown in Figure 7 and Figure 8 [8].

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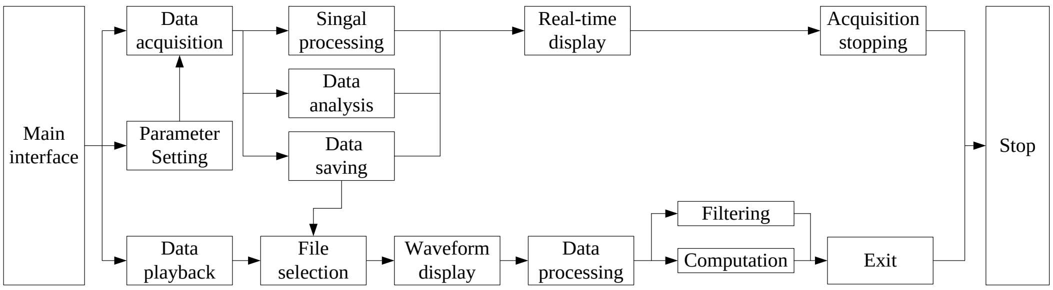


Figure 5. Software Structure of the Monitoring System

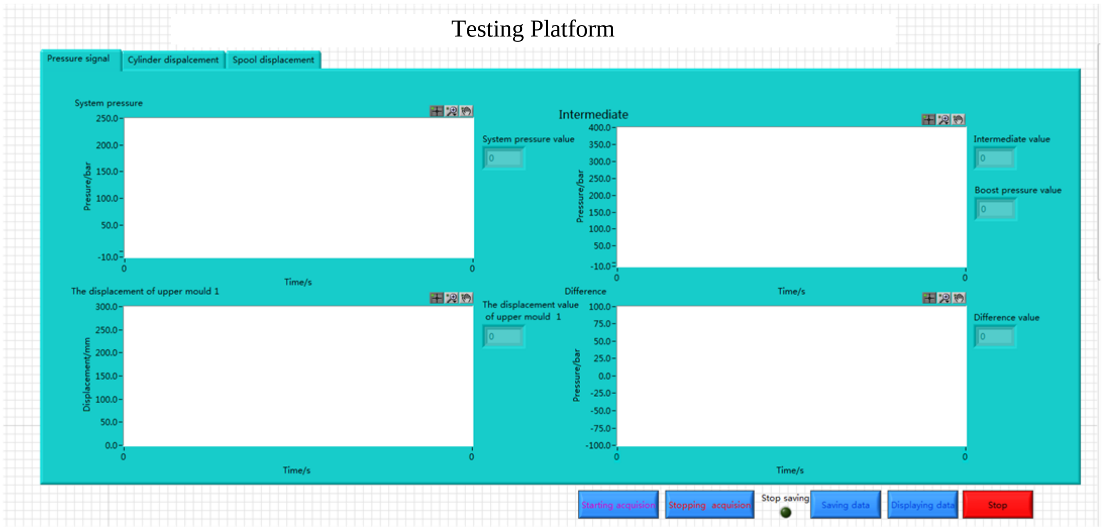


Figure 6. The Front Panel of Monitoring System

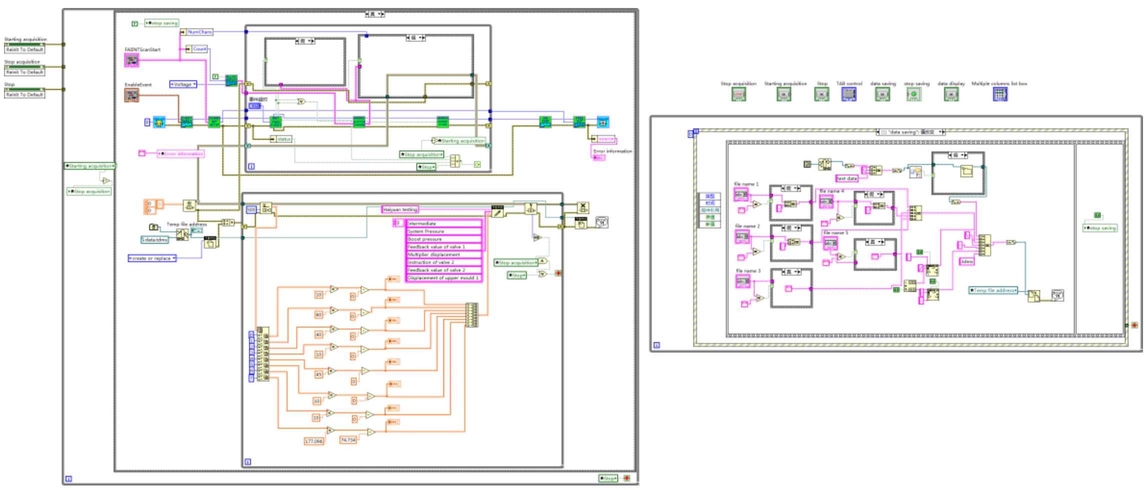


Figure 7. Main Program Diagram of Monitoring System

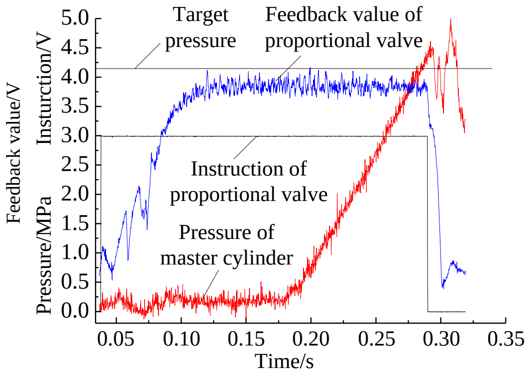
5. Experimental Results

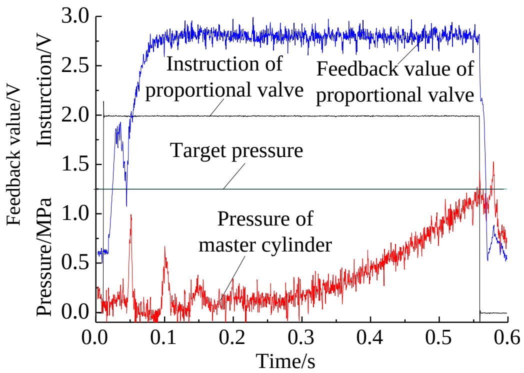
5.1. Control Objective

In this paper, the area of master cylinder patrix installed on powder pressing equipment is 1.2m^2. The pressure of 1MPa in master cylinder's big chamber is equivalent to applying suppressed strength of 120T on powder load. The control requirement of production process is that the suppressed strength accuracy is 6T while the target strength is in the range of 100T-500T (pressure accuracy is 0.05MPa while the target pressure is in the range of 0.8-4.2MPa) with low-pressing time is controlled in 300ms and below. In order to satisfy the above control requirements and accelerate the speed of control strategy development, equipment operation data and control performance with different control modes are acquired and analyzed which are the bases of control strategy optimization.

5.2. Classical Open-Loop Control

Open-loop control is adopted for low-pressure control in various types of equipment currently. Set target pressure as 4.2MPa and 1.25MPa with setting proportional valve voltage as 3V and 2V in a case of actual open-loop mode for low-pressure control. Figure 8 shows the system performance of the first compression under opened-loop control. When the voltage command of first compression is issued, a sudden turn of proportional valve leads to pressure impact. Pressure is beginning to raise after some time that is defined as zero pressure time. According to the analysis of status data from monitoring system, the major cause of the zero pressure time is that master cylinder falls some distance without oil compensating circuit during inertial compression stage. The cause could result in a partial vacuum that it must cost some time to fill oil inside the master cylinder. Proportional valve closes quickly when pressure reaches the target but the pressure is greatly overshooting soon. The delay of the controller instructions inevitably leads to overshoot. The greater the proportional valve opening while the valve suddenly close is, the greater the pressure of master cylinder's big chamber shocks [12]. It can be seen that the traditional mode of open-loop control has many defects.





(a) Target pressure is 4.2MPa and the proportional valve voltage is 3V

(b) Target pressure is 1.25MPa and the proportional valve voltage is 2V

Figure 8. System Performances of the First Suppress Under Opened-Loop Control

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5.3. Optimization of Inertia Compressing Action and Constant Proportion Close-Loop Control

In order to reduce the compression time and increase the pressure precision, it is necessary to fill oil into the big chamber of master cylinder in inertia compressing and adopt the close-loop control strategy in the first compressing. Oil supplement in inertia compressing stage must ensure that the pressure doesn't rise significantly. By analyzing hydraulic circuit, two ways are summarized for oil supplement: a) open valve 2, close proportional valve 4; b) close valve 2, open valve 3 and proportional valve 4. If the former adopted, pressure doesn't raise until valve 2 is completely closed in low-pressure stage. If the latter adopted, not only the oil supplement is ensured, but also valves aren't necessary to switch while suppression state is changing so that pressure jitter is reduced.

To accelerate the overall compression process, the larger constant voltage is giving to proportional valve before pressure raises to certain values. Then the control mode is switch to close-loop control mode. The close-loop control is proportion control that the control voltage is calculated by formula (10).

$$U = K_p \times (Sp - p) \tag{10}$$

Where U is the control voltage; K_p indicates the proportional value; S_p shows the target pressure; p means the real-time pressure. Setting the upper limit (U_{max}) to prevent pressure reaching much overshoot value and lower limit (U_{min}) to skip the proportional valve dead zone. The pressure control precision is quite better through the field commission with the parameters shown in Table 3. The system performance is shown in Figure 9.

Table 3. Parameters of Constant Proportion Closed-Loop Control

Parameters	Values	Parameters	Values
Sp	1.25MPa	U_{max}	2.3V
K_p	8.4	U_{min}	1.2V

Figure 9 indicates that the time pressure raised from the initial to target cost is 320ms that is greater than 300ms. It is quite difficult that the control method uses constant proportional value can satisfy both the rapidity and precision equipment requires. The voltage value calculated according to the formula (10) drives the valve spool moving to position near the dead zone while the pressure is close to the target. The long time that voltage is at the lower limit value can ensure the better control precision but not good rapidity. Moreover, this control method is poor adaptability that is easy to cause the different target pressure corresponding to different control parameters.

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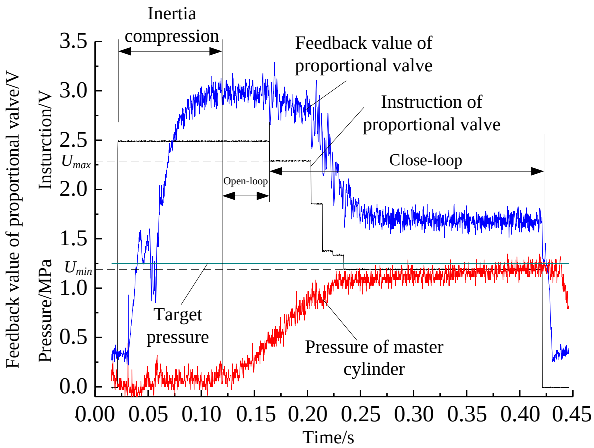

Figure 10. System Performances under Constant Proportion Closed-Loop Control

5.4. Segmented Variable Gain Control Strategy Based On Parameterization

Aiming at the above mentioned problems, segmented variable gain method that proportion is adaptively calculating according to the target pressure and real time pressure conditions is adopted for low-pressure control to improve the flexibility of control parameters of the compression process [13]. The control flowchart is shown in Figure 10. Figure 11 is the simple compression process which shows the relationship of proportional valve voltage and master cylinder's pressure in segmented variable gain control method. The control method includes open-loop control and close-loop control stages. The specific implementation scheme elaborates following: set three certain pressure value symbolized as p_1, p_2 and p_3, set target pressure value is S_p, set four segmented voltage value symbolized as u_0, u_1, u_2 and u_3, and $p_3=S_p-a$, $p_2=b\cdot S_p$, $p_1= c\cdot S_p$. Proportional valve value $u=u_0$ when open-loop control stage. The close-loop control stage's is calculating by formula (11).

$$u = k_p(Sp - p) + \Delta u \tag{11}$$

The value of k_p and Δu are parametric and automatically calculated so that the proportional valve voltage can keep smooth transition while compression stage changing.

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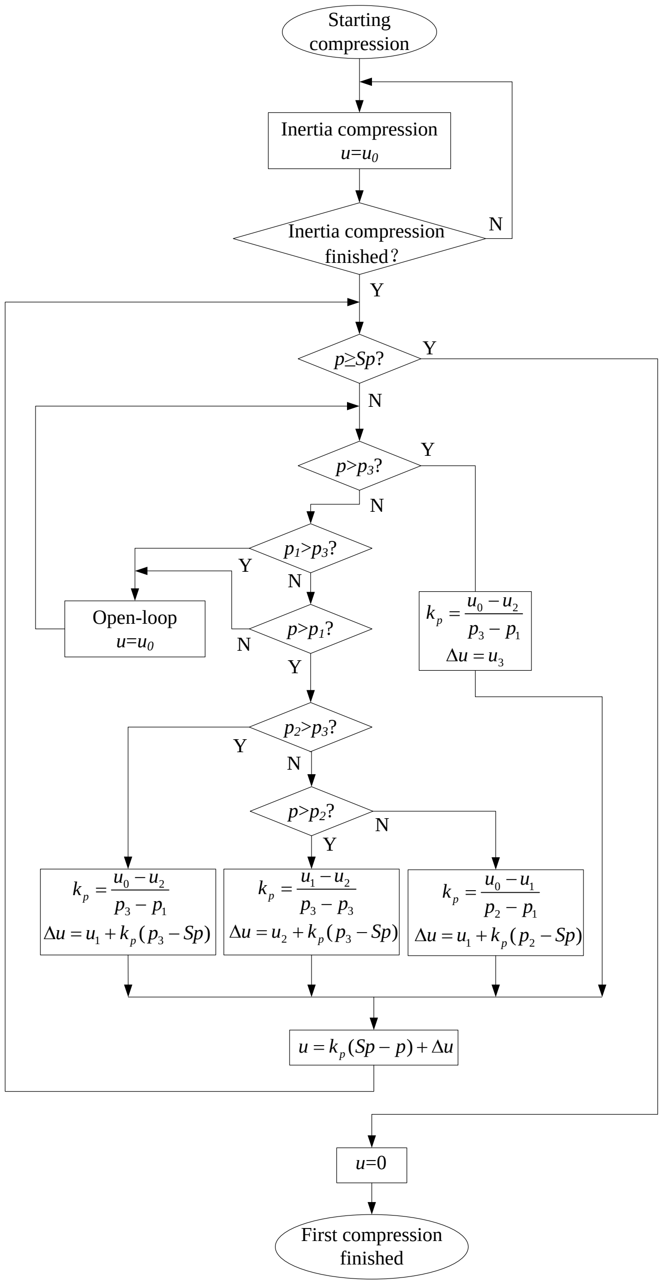

Figure 10. The Flow Diagram of the Closed-Loop Control Strategy with Parameterized Segmented Variable Gain

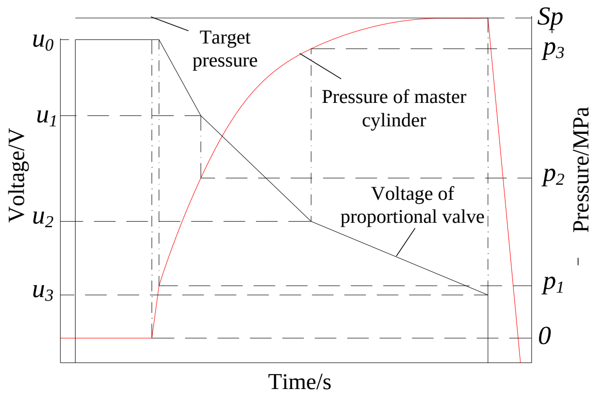

Figure 11. Closed-Loop Control Strategy with Segmented Variable Gain

The values of u_1, u_2, u_3, a, b and c are settled down according to experimental measurement in field commissioning. u_0 is given different value by experimental measurement and linear programming so that it could match to different target pressure. u_0 can be set 2.2V while target pressure is 0.8MPa; u_0 can be set 3.2V while target pressure is 4.2MPa. According to the linear fitting, the function expression for u_0 is shown as following.

$$u_0 = 2.2 + \frac{Sp - 0.8}{3.4} \tag{12}$$

Parameters are set with data in Table 4. Control performance is very nice that both pressure's repeat precision and the time first compression cost meet the requirements of control objectives in field commissioning. The step signal is maximum and the time cost is longest in the situation that target pressure is 4.2MPa and of course it is the most difficult condition of which system performance is shown in Figure 12. The Figure shows that the time that low-pressure control process cost is about 250ms and the pressure control precision is 0.025MPa. The control precision and rapidity are both improved significantly.

Table 4. Closed-Loop Control Parameters with Segmented Variable Gain

Parameters	Value	Parameters	Value
a	0.05	u_1	2.4
b	0.75	u_2	2.0
c	4.2	u_3	1.2

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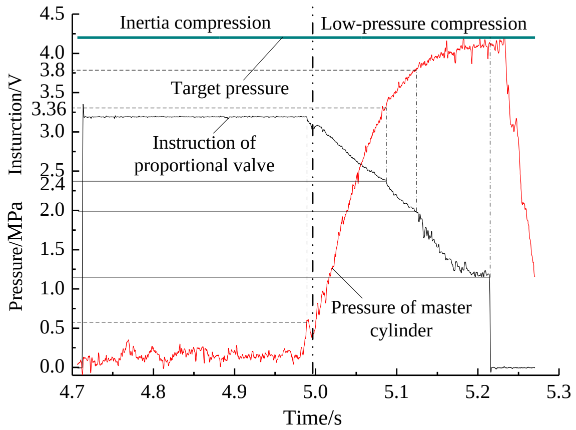

Figure 12. System Performances under the Closed-Loop Control with Segmented Variable Gain

6. Conclusions

Real-time monitoring system of equipment with multi-channels based on LabVIEW is built, and the full-state of equipment can be acquired based on the system. The test data not only provides strong support for control strategy optimization of first compression but also significantly improves R&D efficiency and engineering adaptability of control strategy.

Optimizing the working status of control elements during the pressing process due to inertial force can reduce the running time of the low-pressure compression process. The segmented variable gain control strategy based on parameterization can adapt to the target pressure in the change of 0.8MPa to 4.2MPa and respectively improve the control precision and the compression time within 0.05MPa and 300ms. The control strategy ensures both high precision and high efficiency in compression process.

Acknowledgements

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