Redundant design and performance analysis of hydraulic power source for actuator

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Abstract
This study explores a high reliability redundancy hydraulic power source, which includes shunt, throttle, relief, and switch modules that can continuously supply constant working pressure and steady flow in the case of input torsion and component failure. Considering the analysis of the working mechanism of each component of the energy control system, this study establishes the mathematical analysis theoretical model and selects the appropriate pressure and flow sensitive parameters, such as structure and spring. Through Amesim software, the physical simulation model is established, and the simulation research is conducted to analyze its static and dynamic characteristics. The redundancy fault tolerance capability of the system is also simulated and verified. Simulation results indicate that the output pressure and flow of the system can maintain stability with the fluctuation of input pressure. When the relief module fails in one branch, the output pressure of power energy remains unchanged, and the flow is halved. Although the maximum speed is attenuated, the system can still be guaranteed to run. The redundancy of energy system is realized, and the reliability of the system is greatly improved. The stable pressure and flow redundancy module designed in this study can keep the pressure and flow of the loop stable in the case of input pressure fluctuation. The system also has a good fault-tolerant function for all faults.

Keywords: Hydraulic power source, Redundant design, Actuator, Theoretical model, Simulation analysis

1 Introduction

Hydraulic power source is an important component for actuator in aerospace, such as aerospace launch vehicle(Nazareth, et al. 2017; Liao, et al. 2017; Yang, et al. 2012). The thrust vector control system of Japanese H2 launch vehicle solid rocket booster (SRB) adopts hydraulic actuator. The pressure regulating valve adjusts the cold helium source of 29.5 MPa to 20.6 MPa and uses the extrusion hydraulic energy as the power source of the hydraulic actuator(Ishibashi, et al. 1995). The US Ariane 5 launch vehicle SRB also uses a similar energy mode(Caye, et al. 1994). In addition, the vehicle makes full use of the engine energy as actuator energy that can greatly reduce the complexity of the servo system and improve its reliability. The Saturn V rocket to the moon and The Falcon V Medium Lift Launch Vehicle use a hydraulic actuator, which is directly used high pressure kerosene from the rocket engine turbine pump as its energy source(Koenigsmann, et al. 2004), in which the flow pressure stability of the power source is the basis of the working stability of the thrust vector servo system.

The thrust vector actuator of aircraft has large power, High power mass ratio, and stable energy supply are the guarantees of servo performance reliability and lightweight design. Making the hydraulic source meet the system use requirements through pressure or flow regulation is a common means of hydraulic system(Pan, et al. 2009; Ma, et al. 2016; Hao, et al. 2019). The performance of power source has greatly influenced the comprehensive performance of hydraulic system(Yan, et al. 2017; Tian, et al. 2017; Jia, et al. 2012). The application of mathematical model analysis
and simulation technology in the design of hydraulic system and components improves the design efficiency and correctness (Alireza and Pedram, 2017; Kim, et al. 2017; Sato, et al. 2017). Researchers have suggested many power source design methods to improve system performance. Yu et al. designed a type of dual power source, which makes the electrohydraulic brake system perform well (yu, 2016). Raade and Kazerooni developed a novel device to supply hydraulic power for anaerobic mobile robotic systems (Raade and Kazerooni, 2005). Chen Zongbin et al. proposed a pressure ripple control strategy on the basis of the auxiliary pump source confluence, thus reducing the structure vibration on the pedestal (Chen, 2019).

In this study, the hydraulic power source of electrohydraulic actuator is come from a high pressure, high flow hydraulic source, and the stability and reliability of the power energy directly affect the function and performance of the system. An energy control system with high reliability redundancy is designed in this study. High reliability redundancy energy control system includes shunt, throttle, relief, and switch modules, which can continuously supply constant working pressure and steady flow in the case of input torsion and component failure. On the basis of the analysis of the working mechanism of each component in the energy control system, this study establishes a mathematical analysis and theoretical model and selects the appropriate pressure and flow sensitive parameters, such as structure and spring. Through Amesim software, the physical simulation model is established, and the simulation research is conducted to analyze its static and dynamic characteristics. The redundancy fault tolerance capability of the system is simulated and verified.

2 Design and mechanism analysis

The common energy control system consists of a throttle valve and a relief valve. The failure of the relief valve will cause pressure fluctuation, and a sharp drop in pressure is a fatal failure of the system. The multi-stage design of large flow relief valve greatly increases the risk and probability of failure. Unlike previous designs, High reliability redundancy energy control system consists of a shunt valve, two throttle valves and two relief valves and two check valves. The shunt valve can evenly diverge the input flow with little impact from the back-end components. When working normally, the flow after the shunt flows through the throttle valve, the relief valve and the check valve and then collects again to jointly drive the actuator to work. This design can ensure that when an relief valve loses pressure due to failure, the output pressure of the system remains unchanged. Although the flow rate is reduced by half at this time, it can guarantee the actuator to work in the down condition and will not cause fatal failure. This design realizes the hardware redundancy of the hydraulic circuit and makes the system have certain fault tolerance ability. An added benefit of this design is that the design complexity of the throttle valve and the relief valve is reduced because the flow rate is superimposed during normal operation. The diagram of high reliability redundancy energy control system is shown in Fig. 1.

![Fig. 1 Actuator system with high reliability redundancy hydraulic power source. The “Hydraulic source” is the hydraulic source input of the system. The section of “Energy control system” is the research object of this paper. The section of “Actuator” is the application object of the “Energy control system”.](image-url)
The structure of high reliability redundant energy control system is illustrated in Fig.2. Each component of the dual redundant throttle, relief, and switch modules are inserted symmetrically on the integrated valve block shell. The two oil inlets of the throttle module are connected with the two oil outlets of the double average shunt module.

Shunt module evenly distributes one oil fluid to two branches. When the load pressure of two outlet oil branches is different, their flow rate can be equal. The shunt module is composed of a shunt element, as illustrated in Fig.3. Fixed throttle orifices a and b and variable throttle orifices g and h are found on the valve spool. Four variable throttle orientations are found on each side. The whole hydraulic valve structure is completely symmetrical, where a and b are two fixed throttling orifices with a symmetrical structure. According to the throttling formula $Q = c_d A \frac{\sqrt{2 \rho \Delta p}}{\rho}$, if the pressure difference added to the two fixed throttling orifices is equal, the flow rate through them can be the same. Given that inlet pressure $p_i$ is the same, the outlet pressures $p_{o1}$ and $p_{o2}$ of the fixed throttle orifice is equal, and the flow through the fixed throttle orifice is the same. In the module outlet, when load pressures $p_{o1}$ and $p_{o2}$ are not equal, spool 4 is shifted accordingly with the action of the pressure difference and is constantly adjusted to ensure that $p_{o1}$ and $p_{o2}$ are equal. After the oil passes through the fixed throttle holes a and b, and then through the variable throttle orifices g and h, the oil flows from the outlet to the subsequent components. The inlet flow is $Q_i$, and the outlet flows of the two branches are $Q_{o1}$ and $Q_{o2}$. Given that the size of the two branches is completely symmetrical,

![Figure 2](image-url)

**Fig. 2** Structure of high reliability redundant hydraulic power source. The system consists of four types of hydraulic components, including one shunt element, two throttle elements, two relief elements and two switch elements. two throttle elements, two relief elements and two switch elements are arranged symmetrically in the shell.

if the load pressure is equal, then $p_{o1}$ and $p_{o2}$ are equal, $Q_{o1} = Q_{o2} = Q_o / 2$, and $p_{o1} = p_{o2}$. If load pressure $p_{o1} > p_{o2}$, then the spool has not yet responded to the motion at the initial moment, and the total resistance of the flow branch is the same. A small pressure difference with a large load leads to a small flow rate, that is, $Q_{o1} < Q_{o2}$. At this time, $p_{o1} - p_{o2} > p_{o1} - p_{o2}$, which can make the outlet pressure $p_{o1} > p_{o2}$ of fixed throttle holes a and b. The flow through fixed throttle orifice a decreases, whereas the flow through fixed throttle orifice b increases. At the same time, the valve spool moves under the action of pressure difference between $p_{o1}$ and $p_{o2}$. Variable throttle orifice g increases, h decreases, $Q_{o1}$ increases, and $Q_{o2}$ decreases until $Q_{o1} \approx Q_{o2}$. The valve spool achieves balance under the comprehensive action of hydraulic pressure, hydraulic power, and spring force. Similarly, the case of $p_{o1} < p_{o2}$ can be analyzed.
The function of the throttle module is to stabilize the oil flow and convert the large flow input into the stable small flow output. When the input source or load fluctuates, the flow is kept stable to compensate for the change of output flow caused by the pressure of hydraulic system or load pressure during the flight of the aircraft. The throttle module is composed of two throttle elements. The diagram of the throttle elements is illustrated in Fig. 4. Assuming that the outlet pressure is a constant value, when the inlet pressure changes, the fixed throttle hole outlet pressure $p_{so}$ also changes. When the liquid pressure on the valve spool overcomes the spring force and causes the movement of the valve spool, the variable throttle orifice opening area changes and the flow changes accordingly. An ideally designed throttling element can ensure constant flow rate when input pressure changes.

Relief module keeps the circuit pressure stable and ensures the continuous and stable pressure supply in the flight process of the aircraft. The relief module is composed of two relief elements. The diagram of relief elements is presented in Fig. 5. Inlet pressure oil is divided into two ways. The first way acts on the main valve spool 7, whereas the second way acts on the loaded side of the main valve spring through the damping hole and acts on the pilot valve spool 5 through the damping hole. When pressure $p_{ri}$ is less than the set value of the pilot valve spring 4, the pilot and main valves are closed. Therefore, no flowing liquid exists in the damping hole, and the pressure on both sides of the main valve spool 7 is equal. The relief valve is in the closed state with the action of the main valve spring 8, where $p_{ri} = p_{pm}$. When input pressure $p_{ri}$ is greater than the set value of the pilot valve spring 4, the oil can drain through the damping hole and the pilot valve opening. The flowing liquid generates pressure drop on both sides of the damping hole. When the pressure drop causes the main spool 7 to bear more force than the set force of spring 8, the main and flow stabilizing valves open, realizing pressure stability. In the figure, $Q_{ni}$, $Q_{no}$, $p_{ni}$, $Q_{mi}$, and $p_{pm}$ are inlet flow, main valve flow, inlet pressure, pilot valve flow, and pilot valve inlet pressure, respectively.
3 Modeling and performance analysis of shunt element

3.1 Theoretical model analysis

The hydraulic bridge diagram of the shunt element is illustrated in Fig.6.

\[
\begin{align*}
Q_{oi} &= Q_{oi1} + Q_{oi2} + \frac{V_i}{\beta} \frac{dp_i}{dt} \\
\frac{\pi D_i^2}{4} (p_{oi1} - p_{of1}) - F_{oi1} + F_{oi2} + k_i(x_{oi1} - x_i) &- k_i(x_{oi1} - x_i) = M_i \frac{dx_i}{dt^2} + B_i \frac{dx_i}{dt} \\
Q_{oi1} - Q_{oi} &= \frac{V_i}{\beta} \frac{dp_{oi1}}{dt} - \frac{\pi D_i^2}{4} \frac{dx_i}{dt} \\
Q_{oi2} - Q_{oi} &= \frac{V_i}{\beta} \frac{dp_{oi2}}{dt} - \frac{\pi D_i^2}{4} \frac{dx_i}{dt} \\
Q_{oi1} &= Q_{oi1} + Q_{oi1} \\
Q_{oi2} &= Q_{oi2} + Q_{oi2}
\end{align*}
\]

where \( Q_{oi} \) is the inlet flow; \( Q_{oi1} \) is the outlet 1 flow; \( Q_{oi2} \) is the outlet 2 flow; \( Q_{oi1} \) is the flow through fixed throttle orifice 1; \( Q_{oi2} \) is the flow through fixed throttle orifice 2; \( Q_{oi1} \) is the flow through variable throttle orifice 1; \( Q_{oi2} \) is the flow through variable throttle orifice 2; \( Q_{oi1} \) is the fixed throttle 1 leakage to the outlet; \( Q_{oi2} \) is the fixed throttle 2 leakage to the outlet; \( p_i \) is the inlet pressure; \( p_{oi1} \) is the fixed throttle orifice 1 outlet pressure; \( p_{oi2} \) is the fixed throttle orifice 2 outlet pressure; \( p_{oi} \) is the outlet 1 pressure; \( p_{oi1} \) is the outlet 2 pressure; \( V_i \) is the inlet volume; \( V_{oi} \) is the volume of shunt chamber 1; \( V_{oi1} \) is the volume of shunt chamber 2; \( F_{oi} \) is the hydrodynamic force of variable throttle 1;
$F_{s2}$ is the hydrodynamic force of variable throttle 2; $D_s$ is the valve spool diameter; $M_s$ is the valve spool mass; $\beta$ is the bulk modulus of elasticity of oil; $k_s$ is the spring stiffness; $x_{o0}$ is the spring precompression length; $x_s$ is the valve spool displacement; and $b_{sv}$ is the viscous damping of spool.

$$Q_{s1} = c_{sv} \frac{\pi d_{s1}^2}{4} \sqrt{\frac{2}{\rho} (p_{o1} - p_{m1})},$$
$$Q_{s2} = c_{sv} \frac{\pi d_{s2}^2}{4} \sqrt{\frac{2}{\rho} (p_{o2} - p_{m2})},$$
$$Q_{s3} = 4c_{sv}s_s(x_s) \sqrt{\frac{2}{\rho} (p_{o2} - p_{m2})},$$
$$Q_{s4} = 4c_{sv}s_s(x_s) \sqrt{\frac{2}{\rho} (p_{o1} - p_{m1})},$$
$$Q_{s5} = 4c_{sv}s_s(x_s) \sqrt{\frac{2}{\rho} (p_{o1} - p_{m1})},$$

where $s_{s1}(x_s)$ is the variable throttle orifice area 1; $s_{s2}(x_s)$ is the variable throttle orifice area 2; $\theta_1$ is the oil ejection angle of variable orifice 1; $\theta_2$ is the oil ejection angle of variable orifice 2; $c_{sv}$ is the fixed orifice flow coefficient; $c_{sv}$ is the variable orifice flow coefficient; $c_{sv}$ is the variable orifice flow coefficient; $d_{s1}$ is the variable orifice velocity coefficient; $d_{s2}$ is the diameter of fixed orifice 1; $d_{s2}$ is the diameter of fixed orifice 2; $\delta_s$ is the spool body fit clearance; $e_s$ is the relative eccentricity; $b_s$ is the oil sealing length; $\rho$ is the oil density; $\mu$ is the dynamic viscosity; and $V_{sb}$ is the volume of shunt chamber when the valve spool is in the middle position.

### 3.2 Simulation analysis

According to the working principle of shunt, the simulation model of shunt element is established by using the hydraulic component design (HCD) of Amesim, as illustrated in Fig.7.

![Simulation model of shunt element](image)

**Fig. 7** Simulation model of shunt element. The values of parameters input pressure, input flow and spool diameter are 550bar, 220L/min and 30mm respectively.

The simulation results of shunt deviation with different load differential pressure, throttle hole diameter, spring stiffness, and initial opening are shown in Fig.8.
Simulation results indicate that the load pressure difference affects the shunt accuracy. When load pressures $p_{1\text{ol}}$ and $p_{2\text{ol}}$ are the same, the shunt accuracy is the highest. When the load pressure difference increases, the shunt accuracy decreases because when the load pressure difference increases, the spool moves for a certain distance to achieve balance with the action of spring force, hydrodynamic force, friction force, and hydraulic pressure. However, a deviation exists in the outlet pressure of fixed throttle orifice, that is, $p_{1\text{os}} \neq p_{2\text{os}}$. Moreover, the greater the outlet pressure difference, the greater the spring compression amount, the greater the difference between $p_{1\text{os}}$ and $p_{2\text{os}}$, and the greater the shunt deviation.

Simulation results also show that fixed throttle hole size $d_f$ affects the shunt accuracy. The smaller the fixed throttle hole, the higher the shunt accuracy because the smaller the fixed throttle hole, the larger the pressure drop. Moreover, when the load is unbalanced, the greater the control force of the valve spool, and the greater the regulating capacity of the valve spool.

Spring stiffness coefficient $k_s$ also affects the shunt accuracy. When the spring stiffness coefficient decreases, the shunt accuracy can be improved because when the stiffness decreases, the sensitivity can increase, and the shunt accuracy can be high. By contrast, low spring stiffness can lead to poor stability and easily cause mechanical stress jitter or flow fluctuations.

Moreover, the variable orifice initial opening $x_i(x_i)$ affects the shunt accuracy. If the initial opening is large, then the shunt deviation can increase because the preset opening of variable throttle orifice is large, and the ratio between the opening size adjusted and the initial opening size by the same distance of spool movement is small, thus increasing shunt deviation. However, too little initial opening can lead to increased pressure loss.

In addition, as the diameter of the valve spool increases, the shunt accuracy of the valve can be improved because increasing pressure area increases control force and reduces the shunt deviation. Moreover, the shunt accuracy of the shunt valve in the rated flow range increases as the input flow increases. When the flow rate is small, the shunt accuracy is significantly reduced. The design selection of parameters is made according to the use of demand, combined with the valve volume and the dynamic characteristics of a comprehensive consideration.

4 Modeling and performance analysis of throttle element
4.1 Theoretical model analysis

The hydraulic bridge diagram of the throttle element is presented in Fig. 9.
The mathematical model of throttle element includes the equations of inlet flow continuity, spool motion balance, internal flow continuity, and outlet flow. These equations are presented as follows:

\[
\begin{align*}
Q_i &= Q_{io} + \frac{V_o}{\beta} \frac{d\rho}{dt} \\
= &\frac{\pi}{4} (p_0 - p_o) (D_i^2 - d_t^2) - F_r - k_s (x_0 - x) \\
= &M_s \frac{d^2x}{dt^2} + B_p \frac{dx}{dt} \\
Q_o &= Q_{so} + \frac{V_o}{\beta} \frac{d\rho}{dt} \frac{\pi D_o^2}{4} \frac{dx}{dt} \\
Q_{io} &= Q_{so} + Q_{t0} \\
\end{align*}
\]

where \( Q_i \) is the inlet flow; \( Q_{io} \) is the flow through a fixed throttle orifice; \( Q_o \) is the leakage; \( Q_{so} \) is the flow through variable throttle orifice; \( Q_{t0} \) is the outlet flow; \( p_0 \) is the inlet pressure; \( p_o \) is the fixed throttle outlet pressure; \( p_w \) is the outlet pressure; \( V_o \) is the inlet volume; \( V_w \) is the volume between fixed throttle orifice and variable throttle orifice; \( F_r \) is the hydrodynamic force of variable throttle; \( d_t \) is the fixed orifice diameter; \( D_i \) is the valve spool diameter; \( M_s \) is the valve spool mass; \( k_s \) is spring stiffness; \( x_0 \) is the spring precompression length; \( x \) is the valve spool displacement; and \( B_p \) is the viscous damping of spool.

\[
Q_{io} = c_{io} \pi \delta_t^2 \frac{2}{\rho} (p_0 - p_w) \cdot Q_o = \frac{\pi D_o \delta_t^2}{12 \mu h} (1 + 1.5 \varepsilon \delta_t) (p_0 - p_w) \cdot F_r = 12c_{io} c_{io} s_j(x) \cos \theta (p_0 - p_w),
\]

\[
Q_{t0} = 4c_{io} s_i(x), \quad V_{w0} = V_{w0} + \frac{\pi D_t^2}{4} x,
\]

where \( s_j(x) \) is the variable throttle orifice area; \( \theta \) is the oil ejection angle of variable orifice; \( c_{io} \) is the fixed orifice flow coefficient; \( c_{io} \) is the variable orifice flow coefficient; \( c_{io} \) is the variable throttle orifice velocity coefficient; \( \delta_t \) is the spool body fit clearance; \( \varepsilon \) is the relative eccentricity; \( h \) is the oil sealing length; \( V_{w0} \) is the volume between fixed throttle orifice and variable throttle orifice at static state.

### 4.2 Simulation analysis

According to the working principle of throttling, the simulation model of throttle element is established by using the HCD of Amesim, as shown in Fig.10.
The simulation results of flow with different fixed throttle orifice diameter, spring pretension, spring stiffness, and initial opening are illustrated in Fig. 11.

Simulation results reveal that fixed throttle orifice and the initial opening of variable throttle orifice affect the flow. The flow increases as the fixed throttle orifice and the initial opening of variable throttle orifice increase. The reason is that when the fixed throttle orifice and the initial opening of variable throttle orifice increase, the flow area increases and the flow increases.

Spring preload and spring stiffness also affect the flow, which increases as spring preload and spring stiffness increase. The reason is that the throttle element is a constant opening element; when the flow increases, the pressure difference to promote the movement of the valve spool and variable throttle mouth small keep the flow steady. When the spring preload or spring stiffness increases, the spool movement needs a great differential pressure.

Appropriate throttling aperture, spring pretension, spring stiffness, and initial opening are selected to ensure that...
the flow meets the requirements.

5 Modeling and performance analysis of relief element

5.1 Theoretical model analysis

The hydraulic bridge diagram of the relief element is shown in Fig.12.

The mathematical equations for relief element are the main valve flow, main spool force balance, damping port flow, pilot port flow, and pilot spool force balance equations. These equations are presented as follows:

\[
\begin{align*}
Q_{m} &= Q_{m0} + Q_{da} + \frac{V_{m}}{2} \frac{dp_{m}}{dt} \\
\frac{\pi}{4} (p_{m} - p_{r}) (D_{m}^{2} + D_{a}^{2}) &= k_{m} (x_{m0} + x_{m}) + F_{m} \\
Q_{da} &= Q_{daI} + Q_{daII} + \frac{V_{da}}{2} \frac{dp_{da}}{dt} \\
\frac{\pi}{4} (p_{m} - p_{r}) D_{a}^{2} &= k_{m} (x_{m0} + x_{m}) + F_{m} \\
Q_{m0} &= Q_{m0I} + Q_{m0II} + Q_{m0III} \\
\end{align*}
\]

where \( p_{m} \) is the inlet pressure; \( p_{r} \) is the main valve damper hole outlet pressure; \( p_{m} \) is the pilot valve inlet pressure; \( p_{r} \) is the relief valve outlet pressure; \( Q_{m} \) is the inlet flow; \( Q_{m0} \) is the total outlet flow; \( Q_{m} \) is the main valve flow; \( Q_{da} \) is the main valve damping hole flow; \( V_{m} \) is the inlet volume; \( Q_{da} \) is the pilot valve flow; \( Q_{m0} \) is the leakage of main valve; \( Q_{m0} \) is the damped posterior chamber volume; \( D_{m} \) is the main valve port diameter; \( x_{m} \) is the main valve opening size; \( d_{a} \) is the main spool damping hole diameter; \( k_{m} \) is the stiffness of main valve spring; \( x_{m0} \) is the precompression of main valve spring; \( F_{m} \) is the steady-state hydrodynamic force of main valve port; \( D_{a} \) is the pilot valve opening diameter; \( x_{a} \) is the pilot valve opening size; \( k_{a} \) is the pilot valve spring stiffness; \( x_{a0} \) is the pilot valve spring precompression value; and \( F_{a} \) is the steady-state hydrodynamic force of pilot port.

\[
\begin{align*}
Q_{m0} &= c_{m} \pi D_{m} \sin \alpha_{m} x_{m} \sqrt{\frac{2}{\rho} \frac{(p_{m} - p_{r})}{\rho}} \\
Q_{da} &= c_{da} \pi d_{a}^{2} \frac{2}{\rho} \frac{(p_{m} - p_{r})}{\rho} \\
Q_{m0} &= c_{m} \pi D_{a} \sin \alpha_{a} x_{a} \sqrt{\frac{2}{\rho} \frac{(p_{m} - p_{r})}{\rho}},
\end{align*}
\]

\[
F_{m} = c_{m} \pi D_{m} \sin (2\alpha_{m}) p_{m},
\]

where \( c_{m} \) is the main valve opening flow coefficient; \( c_{da} \) is the main valve damping hole flow coefficient; \( c_{a} \) is the pilot valve opening flow coefficient; and \( \alpha_{m} \) is the main spool half cone Angle; \( \alpha_{a} \) is the pilot valve half cone angle.

5.2 Simulation analysis

According to the working principle of relief, the simulation model of relief element is established by using the HCD of Amesim, as illustrated in Fig.13.
The function of pilot relief valve is to keep the system pressure constant. Appropriate parameters should be selected to meet the dynamic and static requirements of relief valve.

The simulation results of pressure and flow with different preloading forces of main valve spring, stiffness of main valve spring, diameter of main valve spool, damping hole diameter of main valve spool, preloading force of pilot valve spring, and stiffness of pilot valve spring are illustrated in Fig.14.

Fig. 13 Simulation model of relief element. The values of parameters input pressure is 320bar.

Fig. 14 Performance of relief element with different sensitive parameters. The reference values of parameters main spring preloading force, main spring stiffness, pilot spring preloading force, pilot spring stiffness, main spool diameter and main spool damping hole diameter are 25N, 5N/mm, 197N, 15N/mm, 25mm and 1.8mm respectively.
Simulation results reveal that the main valve spring preload, main valve spring stiffness, pilot valve spring preload, and pilot valve spring stiffness influence flow rate at the rated pressure.

The greater the main valve spring preload, main valve spring stiffness, pilot valve spring preload, and pilot valve spring stiffness, the smaller the flow rate at rated pressure because of the increase in main valve or pilot valve opening pressure.

Moreover, the larger the main spool diameter, the larger the flow rate at the rated pressure because the larger the spool diameter, the larger the hydraulic area and the larger the opening force. At the same time, when the valve spool increases, the opening wet periphery is long, which leads to the flow increase at the rated pressure.

The larger the diameter of the damping hole of the main spool, the smaller the flow under the rated pressure. The reasons are because the main spool damping hole becomes large, pilot valve that opened after the flow of the main spool pressure difference between the two ends becomes small, the main spool opening pressure increases, and the opening degree decreases.

6 Stability and redundancy analysis of hydraulic power loop

The Amesim simulation model of high reliability redundant energy control system is established, as illustrated in Fig.15.

When the input pressure fluctuates greatly, the simulation results of output pressure and flow are shown in Fig.16.

Fig. 15 Simulation model of high reliability redundant hydraulic power. The values of parameters load mass, piston diameter of actuator, rod diameter of actuator and rated flow of servo valve are 15000kg, 130mm, 65mm and 400L/min respectively.

Fig. 16 Output with input pressure fluctuation. The output pressure and flow have good robustness to the fluctuation of input pressure.
When the input pressure fluctuates greatly, the system can obtain stable pressure and flow output.

When the spring failure of relief valve causes the pressure regulation decrease of branch 1, the check valve of branch 1 can be closed, and the oil from branch 1 can overflow from the throttle module of branch 1. Given the function of shunt module, the fault branch is isolated from the system. Branch 2 works normally, and the oil is used as an output to the hydraulic cylinder. In the case of low-frequency signal, the fault has no influence on the system, and the response of the fault is consistent with that of the no-fault case, as illustrated in Fig.17.

![Figure 17](image17.png)

**Fig. 17** Low frequency response with fault. In case of branch failure, the actuator still has good low frequency signal following performance due to redundancy design.

In the case of failure, the phase lag of the system increases by using a high frequency signal input, but the system can still maintain the function. The system high frequency signal output response and flow curve when two branch pressure being both normal, or one branch pressure dropping to 1/2, 1/4, and 0 are illustrated in Fig.18 and Fig.19.

![Figure 18](image18.png)

**Fig. 18** High frequency response with fault. In case of branch failure, the redundant design makes the actuator still respond to the input signal, but the tracking performance becomes worse.

In the case of failure, the phase lag of the system increases by using a high frequency signal input, but the system can still maintain the function. The system high frequency signal output response and flow curve when two branch pressure being both normal, or one branch pressure dropping to 1/2, 1/4, and 0 are illustrated in Fig.18 and Fig.19.

![Figure 19](image19.png)

**Fig. 19** System flow of high frequency signal with the fault of one-way relief valve. The total system flow has different outputs in different fault modes, and the total output flow is minimum when the fault branch pressure is 0.

The system step signal output response and flow curve when two branch pressure being both normal, or one branch pressure dropping to 1/2, 1/4, and 0 are shown in Fig.20 and Fig.21.
When the pressure is low due to the failure of the pressure stabilization module in the case of no shunt module, the oil can leak through the fault branch, and the system cannot supply oil to the hydraulic cylinder to function. The shunt module plays a good role in avoiding this phenomenon. Although the oil is still leaking from the fault branch, another branch of oil can supply the hydraulic cylinder to function, thus ensuring that the hydraulic cylinder haven’t lose its function.

7 Conclusions

This study designs a type of stable pressure and steady flow module with double redundancy; analyzes the theoretical model of each element; establishes the physical model of the element through Amesim; and analyzes and verifies the key influencing factors of component shunt accuracy, flow, and circuit pressure. Amesim simulation software is used to establish a comprehensive simulation model of power and energy system, including shunt, throttle, and relief modules, to simulate the pressure, flow output, and system working condition during power and energy failure. Simulation results indicate that the output pressure and flow of the system can maintain stability with the fluctuation of input pressure. When the relief module fails in one branch, the output pressure of power energy remains unchanged, and the flow is halved. Although the maximum speed is attenuated, the system can still be guaranteed to run. The redundancy of energy system is realized, and the reliability of the system is greatly improved. The stable pressure and flow redundancy system designed in this study can keep the pressure and flow of the loop stable in the case of input pressure fluctuation. This system also has a good fault-tolerant function for all faults.
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