Transient performance analysis of the master cylinder hydraulic system of a 6.3 MN fineblanking press

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Abstract. The master cylinder hydraulic system is the core component of the fineblanking press that seriously affects the machine performance. A key issue in the design of the master cylinder hydraulic system is dealing with the heavy shock loads in the fineblanking process. In this paper, an equivalent model of the master cylinder hydraulic system is established based on typical process parameters for practical fineblanking; then, the response characteristics of the master cylinder slider to the step changes in the load and control current are analyzed, and lastly, control strategies for the proportional valve are studied based on the impact of the control parameters on the kinetic stability of the slider. The results show that the kinetic stability of the slider is significantly affected by the step change of the control current, while it is slightly affected by the step change of the system load, which can be improved by adjusting the flow rate and opening time of the proportional valve.

1. Introduction

Fineblanking is a kind of rapid processing method for punching out high-precision and high-quality parts from sheet metal under a three-dimensional compressive stress state. Due to the high pressure and frequency of the fineblanking, the load on the master cylinder hydraulic system varies drastically and continuously during the process, which seriously affects the machining quality. Therefore, a key step in the fineblanking machine development process is establishing the simulation model for the master cylinder hydraulic system based on the fineblanking process and analyzing the dynamic performance of the system, especially the transient response under the step changes in the system load and control current, to lay the foundation for further design.

In hydraulic system modeling, some mathematical models are presented to study the dynamic characteristics of fluid in hydraulic pipes [1]-[3], and some simulation models based on commercial software are proposed for dynamic performance analysis of hydraulic systems based on different parameters [4],[5]. In hydraulic system analysis, the methods of finite element, finite volume, computational fluid dynamics, multi-body dynamics, and numerical computation are used to research hydraulic characteristics [6]-[11]. In hydraulic system control, electrical position feedback, built-in control rod, speed control, and real-time sensing and actuation are applied to the proportional valve, cylinder, and pump [12]-[15].

Due to the step change characteristics of the fineblanking load, it is necessary to study the transient performance of the master cylinder hydraulic system. On the basis of the abovementioned study, this study designed the typical process parameters of a B630T fineblanking press (rated load of 6.3 MN), established the equivalent models of the hydraulic components and master cylinder system, analyzed the response characteristics of the slider motion to the step change of the load and control signal, and studied the influence of the proportional valve parameters on the motion stability of the slider as well as the control strategy.
2. Equivalent model of the master cylinder hydraulic system

The master cylinder hydraulic system model is shown in Figure 1.

![Figure 1. Master cylinder hydraulic system model.](image)

3. Simulation and analysis of the transient performance of the master cylinder hydraulic system

3.1. Response of the master cylinder slider to the step change of the load

3.1.1. Simulation. At the end of the fineblanking process, the part is cut from the material, and the master cylinder load undergoes an approximate step change from large to small at this point. The simulation settings are set as follows: the load of the master cylinder is 3000 kN at the beginning of the simulation then jumps to 0 at 0.5 s; the control current of the proportional valve is constant; all solenoid valves are in the energized state. The simulation result is shown in Figure 2.

3.1.2. Analysis. In the simulation, when the load drops from 3000 N to 0 N in a very short time, the slider rushes forward 0.9 mm, and the proportional valve appears to be in the unstable state for 20 ms, with a rapid response in the inlet and outlet pressures. However, in practice, the load is unloaded step by step when the slider reaches the top dead center, and the solenoid valve is triggered at the same time to stop the slider quickly. Thus, the abovementioned simulation settings are extreme. Even so, the variation of the slider speed and acceleration are still controllable, and the master cylinder still runs stably under the step load using some optimized control methods.

3.2. Response of the master cylinder slider to the step change of the control signal

3.2.1. Simulation. At the beginning and the middle stage of the blanking, the slider speed is controlled by the proportional valve. At the end stage, the response of cartridge valve 5 is uncontrollable due to the damping effects of 10-Y2 and 10-Y4, so stopping the slider at top dead center mainly relies on the joint effect of proportional valves 10-Y5 and 10-Y9. Therefore, to obtain the system performance, it is necessary to analyze the response of the slider to the proportional valve control signal. The simulation settings are as follows: the master cylinder runs steadily under the load of 4000 kN; the control current of the proportional valve changes from 700 mA to 200 mA at 0.5 s; other solenoid valves are in the energized state. The simulation result is shown in Figure 3.

3.2.2. Analysis. When the control current jumps from 700 mA to 200 mA, it takes the slider 100 ms to reach the set position. However, because the blanking time is only 1.5 s, it is difficult to control the slider effectively using only the solenoid valve. A feasible solution to achieve the desired performance is collaborative control with other components.
3.3. Control performance analysis of the pressure reducing cartridge valve

In the master cylinder hydraulic system, cartridge valve 3 and control cover 2 are used to keep the differential pressure between the inlet and outlet of the proportional valve within a certain range, which is adjusted by the direct relief valve on the cover. The stability of the slider speed is affected directly by the differential pressure between the inlet and outlet when the proportional valve control current is constant. Taking cartridge valve 3 and control cover 2 as the research objects, the factors influencing the pressure difference are analyzed. For ease of analysis, the proportional valve is assumed to be equivalent to the damping, and the load pressure is assumed to be equivalent to the pressure source.

If the orifice diameter is small, and it is easy to keep the differential pressure between the inlet and outlet stable, which means that the follow-up performance of the outlet pressure to the load pressure is better. The main influencing factor is the stroke of the cartridge valve spool. When the flow is small, the spool stroke is short, and when the flow is large, the spool stroke is long, which takes some time to complete.

The velocity of the valve spool can be adjusted by changing the size of the two orifices on the control cover, but this increases the risk of vibration. This variation increases the inlet pressure of the direct acting relief valve on the control cover, the differential pressure of the pressure reducing valve, and the flow of the load through the direct acting relief valve.

The opening speed of the proportional valve can be related to the change in the orifice diameter and the variations of the outlet pressure at three speeds. At speed 1, the orifice opens 1% in 10 ms and then to 100% in 50 ms; at speed 2, the orifice opens from 0 to 100% in 60 ms; at speed 3, the orifice opens with a step change.

3.4. Opening time control strategy for the proportional valve

At the moment the punch just touches the workpiece, the proportional valve of the master cylinder hydraulic system starts, reducing the impact load and optimizing the operating condition. In the transient analysis of the proportional valve opening, the master cylinder system is simplified as follows: the electric control pilot valve of the proportional valve is replaced by a pressure source; the step cylinder is replaced by a velocity source; and the times of turn-on and turn-off of the valves are ignored.

The master cylinder slider is driven by the step cylinder in the first 0.5 s and touches the workpiece at 0.5 s, and then, the load rises from 0 to 1000 kN in 50 ms. The state of each valve is changed at 0.5 s, and the operating conditions of the components are shown in Figure 4. There are crests in the flow and velocity at position 1 because the proportional valve opening is constant at this moment, and the differential pressure between the two ports of the proportion valve is much larger than the design value due to the pressure valve lag, which results in the flow increasing. There are troughs in the pressure
and velocity at position 2 because the slider has an initial speed at this moment and because the charging valve is closed, but the oil from the proportional valve has not arrived yet.

**Figure 4.** Operating conditions of the master cylinder and the proportional valve.

Based on the flow crest at position 1 in Figure 4, the cartridge valve in the main oil circuit is opened at 0.49 s in advance, and the result is shown in Figure 5. The velocity and acceleration of the slider are significantly improved.

**Figure 5.** Operating conditions of the master cylinder and the proportional valve (the cartridge valve is opened in advance).

The negative velocity at position 2 in Figure 4 is due to the low instantaneous flow of the proportional valve. On the basis of the cartridge valve opening in advance, the proportional valve is opened 0.49 s in advance, and the result is shown in Figure 6. Although the negative speed is eliminated, the acceleration of the slider is increased.

**Figure 6.** Operating conditions of the master cylinder and the proportional valve (the proportional valve is opened in advance).
In summary, the pressure variation in the cartridge valve is affected by changing the speed of flow and pressure. Therefore, stabilities of the slider velocity and displacement are difficult to take into account at the same time.

4. Conclusions
- The step change of the master cylinder hydraulic system load from large to small leads to a controllable variation in the slider velocity and acceleration, and some appropriate optimization controls can keep the master cylinder running stably.
- The stepped change of the control current of the master cylinder hydraulic system from large to small causes the slider to not be well-controlled when relying only on the solenoid valve, but the desired performance can be achieved by controlling other components.
- The stability of the slider velocity is directly affected by the differential pressure of the proportional valve, and a smaller orifice is conducive to maintaining a constant pressure differential, which causes better follow-up performance of the outlet pressure to the load pressure.
- The changing speeds of the flow and pressure affect follow-up performance of the pressure variations of the cartridge valve, which leads to difficulty taking the stabilities of velocity and displacement of the slider into account at the same time, but the velocity and acceleration of the slider are significantly improved by opening the cartridge valve in advance.

5. References
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