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# DESIGN ON KEY PARAMETERS OF DOUBLE DAMPER SYSTEM FOR HYDRAULIC ROCK DRILL

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**Abstract.** The lack of researches on the double damper system seriously restricted the impact power's increase of hydraulic rock drills. The structure and working principle of the double damper system are described. The key parameters such as the unloading damping pressure, damping flow, equilibrium position of the damping piston and double damper clearance are designed. At last, the design method is applied to the YYG170 hydraulic rock drill. Double damper system's advantages of high efficiency and fast response are verified by simulation.

**Key words:** hydraulic rock drill, double damper system, key parameters, design example, dynamic simulation.

## 1. Introduction.

Hydraulic rock drills were widely used in mining, coal mine roadway excavation, railway tunnel, highway tunnel and rock excavation projects with advantages of high-efficiency, clean, safety and so on [1 – 3]. The heavy hydraulic rock drill with high-frequency and high-power had become the first choice facing the large-scale mining and larger-sized tunneling [4 – 6]. Meanwhile, damper system's performance was required higher and higher. As the technological innovation of single damper system, the double damper system emerged as the times require. However, the research on the damper system was less and targeted the single damper system [7 – 10]. So there, were little information about the double damper system which had two damping chambers. As an advanced technology for damping and noise reducing, the double damper system had not been spread and applied in domestic heavy hydraulic rock drills.

The structure and working principle of the double damper system would be described. The key parameters such as the unloading damping pressure, damping flow, equilibrium position of the damping piston and double damper clearance would be designed. At last, the design method would be applied to the YYG170 hydraulic rock drill.

## 2. Structure and Working Principle of the Double Damper System.

As shown in Fig. 1, the impact piston hit the shank adapter with high frequency and high speed under the action of impact system. Its kinetic energy was transmitted to rock through shank adapter, drill rod and drill bit in the form of stress wave. The shank adapter would rebound under the reflected wave because of the difference on wave resistance. The damper system was essential to prevent the shank adapter's direct hit to the block.

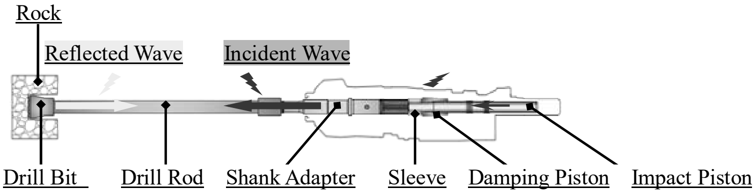

*Fig. 1.* Schematic diagram of the percussive drilling.

As shown in Fig. 2, the double damper system was mainly composed of damping piston, damping valve, damping accumulator. The oil inlet was independent and the flow rate was constant. There were two damping chambers in it which was different from the single damper system.

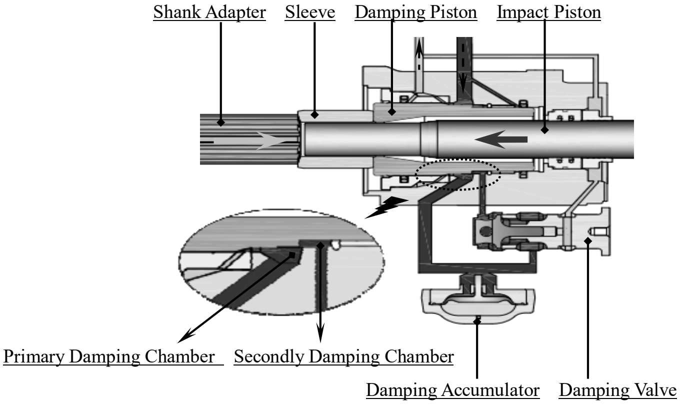

*Fig. 2.* Structure model of the double damping system.

**3. Design on the Unloading Damping Pressure and Damping Flow.**

Fig. 3 was the fuel circuit of the double damper system. In the unloading condition, the damping piston was on the far left, the oil-return hole was fully open, the primary and secondary damping chamber were connected. There was a one-to-one correspondence between the damping pressure and damping flow at the moment. Generally, the unloading damping pressure was assigned to 3.5 MPa to make sure damping chambers' existence. Then, the damping flow could be calculated.

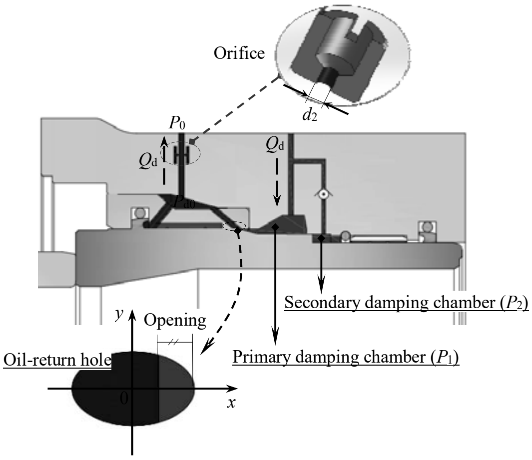

*Fig. 3.* Fuel circuit of the double damper system.

Formula (1) was relation equations of pressure difference and flow rate.

$$\begin{cases} Q_{\mathrm{d}} = C_{\mathrm{d}} \cdot A_{\mathrm{t2}} \cdot \sqrt{\dfrac{2}{\rho} \cdot (P_{\mathrm{d0}} - P_0)}; \\ \dfrac{Q_{\mathrm{d}}}{2} = C_{\mathrm{d}} \cdot A_{\mathrm{t1}} \cdot \sqrt{\dfrac{2}{\rho} \cdot (P_1 - P_{\mathrm{d0}})}. \end{cases} \tag{1}$$

Among them

$$A_{\mathrm{t2}} = \frac{\pi}{4} \cdot d_2^{\,2}; \tag{2}$$

$$P_1 = P_2 = P_{\mathrm{d}}', \tag{3}$$

where $Q_{\mathrm{d}}$ was the damper flow rate. $P_1$ and $P_2$ were the pressure of the primary and secondary damping chamber. $P_{\mathrm{d}}'$ was the unloading damping pressure, $P_{\mathrm{d}}'$ =3,5 MPa. $C_{\mathrm{d}}$ was the flow coefficient, $C_{\mathrm{d}}$ = 0,8. $A_{\mathrm{t1}}$, $A_{\mathrm{t2}}$, were flow areas of the oil-return hole and orifice. $\rho$ was the density of hydraulic oil.

The damping flow could be calculated by solving the simultaneous equations of Eq. (1), Eq. (2) and Eq. (3).

### 4. Determination of the damping Piston's Equilibrium Position.

The double damper system was in equilibrium state before strike. The pressures in primary and secondary damping chambers were equal. Damping piston's statics equation was as follows:

$$F_0 = P_1 \cdot A_{\mathrm{d1}} + P_2 \cdot A_{\mathrm{d2}}, \tag{4}$$

among them

$$P_1 = P_2 , \tag{5}$$

where $F_0$ was the feed force. $A_{\mathrm{d1}}$, $A_{\mathrm{d2}}$, were action areas of primary and secondary damping chamber respectively.

Formula (5) was the expression of $A_{\mathrm{t1}}$ derived by simultaneous equations of (1), (2), (4) and (5).

$$A_{\mathrm{t1}} = \frac{Q_{\mathrm{d}}}{2 \cdot C_{\mathrm{d}} \cdot \sqrt{\dfrac{2}{\rho} \cdot \left(\dfrac{F_0}{A_{\mathrm{d1}} + A_{\mathrm{d2}}} - P_0\right) - \left(\dfrac{4Q_{\mathrm{d}}}{C_{\mathrm{d}} \cdot \pi \cdot d_2^{\,2}}\right)^2}} . \tag{6}$$

Formula (6) was the cross section of oil-return hole. Formula (7) was the relation equation of $A_{\mathrm{t1}}$ and $x$. Among them, $x$ represented the damping piston's equilibrium position.

$$\frac{x^2}{a^2} + \frac{y^2}{b^2} = 1; \tag{7}$$

$$\mathrm{A}_{\mathrm{t1}} = 2 \cdot \int_t^a b \cdot \sqrt{1 - \frac{x^2}{a^2}} dx. \tag{8}$$

### 5. Design on the Double Damping Clearance.

The double damping clearance $\delta_1$ had a great influence on the performance of damper system, which was one of key parameters of the double damper system.

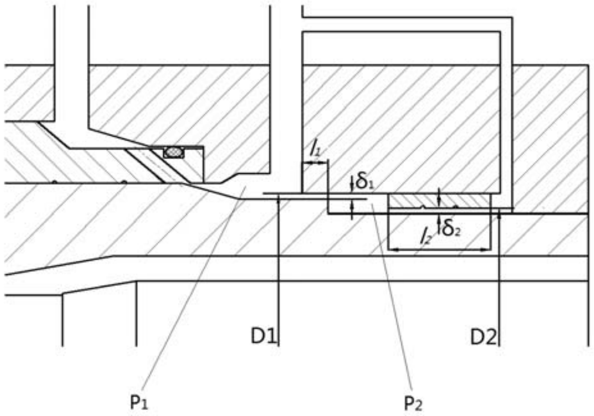

*Fig. 4.* Structure size schematic diagram of the double damper system.

Mathematical model of the double damper system was showed in Eq. (9) referring to Fig. 4.

$$\begin{cases} \dfrac{\mathrm{d}P_2}{dt} = \dfrac{K_e \cdot \left[ A_{d2} \cdot v_p - (\dfrac{\pi D_1 \delta_1^3}{12 \mu l_1} + \dfrac{\pi D_2 \delta_2^3}{12 \mu l_2}) \cdot (P_2 - P_1) \right];}{\mathrm{V}_0 \text{-} A_{d2} \cdot x_p} \\ \dfrac{d^2 x_p}{dt^2} = \dfrac{A_{d1} P_1 + A_{d2} P_2}{-m'}; \\ \dfrac{dx_p}{dt} = v_p; \\ \Delta V = Q_d \cdot t + A_{d2} \cdot x_p; \\ P_1 = P' \cdot (\dfrac{V}{V\text{-}\Delta V})^{1.4}, \end{cases} \tag{9}$$

where $m'$ was the mass of the damping piston and sleeve. $x_p$ and $v_p$ were the displacement and velocity of the damping piston. $V$ was the gas volume of the accumulator. $V_0$ was the initial volume in the secondary damping chamber. $K_e$ was the bulk modulus of oil.

**6. Design example**.

The double damper system of YYG170 hydraulic rock drill was designed by the above method. Parameters were listed in Table 1.

*Table 1.* Parameters of the double damper system

| Variable | Name | Value | Unit |
|---|---|---|---|
| $P'$ | Unloading Damping Pressure | 3.5 | MPa |
| $Q_d$ | Damping Flow | 8 | L/min |
| $V$ | Gas Volume of the Accumulator | 0.12 | L |
| $m'$ | Mass of the Damping Piston and Sleeve | 4 | kg |
| $A_{d1}$ | Action Area of the Primary Damping Chamber | 608.7 | mm$^2$ |
| $A_{d2}$ | Action Area of the Secondary Damping Chamber | 569.4 | mm$^2$ |
| $F_0$ | Feed Force | 10 | kN |
| $t$ | Damping Piston's Equilibrium Position | 0.86 | mm |
| $\delta_1$ | Double Damping Clearance | 0.05,0.06,0.07,0.08 | mm |
| $l_1$ | Fit Length | 2.6 | mm |
| $D_1$ | Fit Diameter | 75 | mm |
| $\delta_2$ | Fit Clearance | 0.03 | mm |
| $l_2$ | Fit Length | 12 | mm |
| $D_2$ | Fit Diameter | 70 | mm |

The damping piston's initial rebound velocity $v_p(0)$ could be calculated according to the referring literature[11 – 15]. Then, the simulation was conducted while the double damping clearance $\delta_1$ was 0,05 mm, 0,06 mm, 0,07 mm and 0,08 mm, respectively

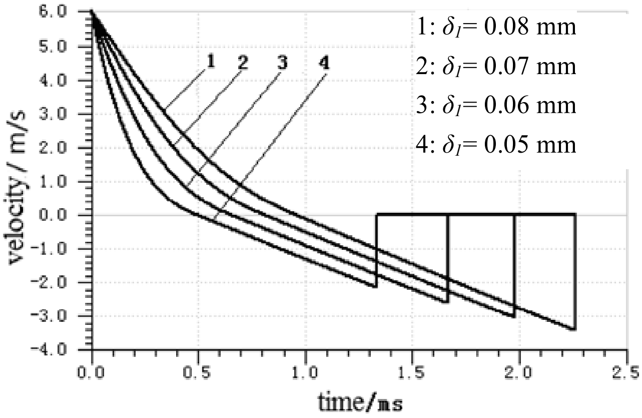

*Fig. 5.* Velocity curve of the damping piston.

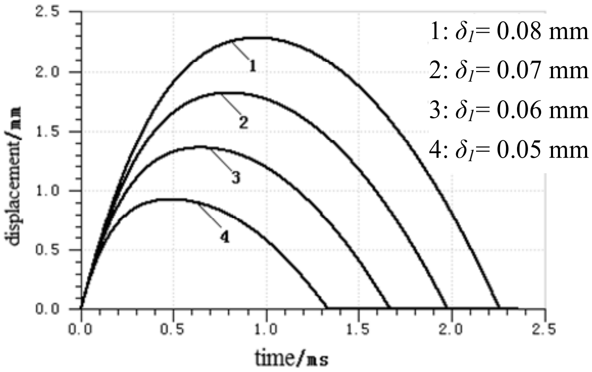

*Fig. 6.* Displacement curve of the damping piston.

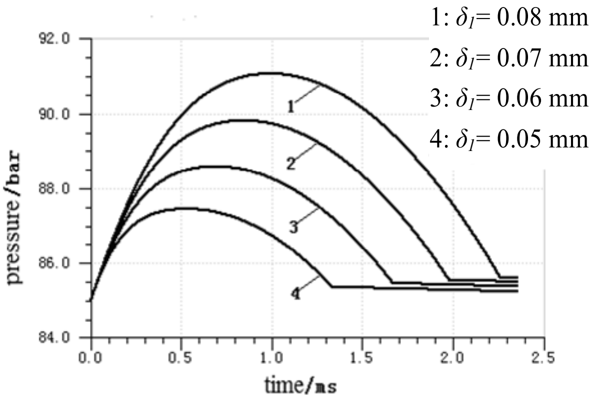

*Fig. 7.* Pressure curve of the primary damping chamber.

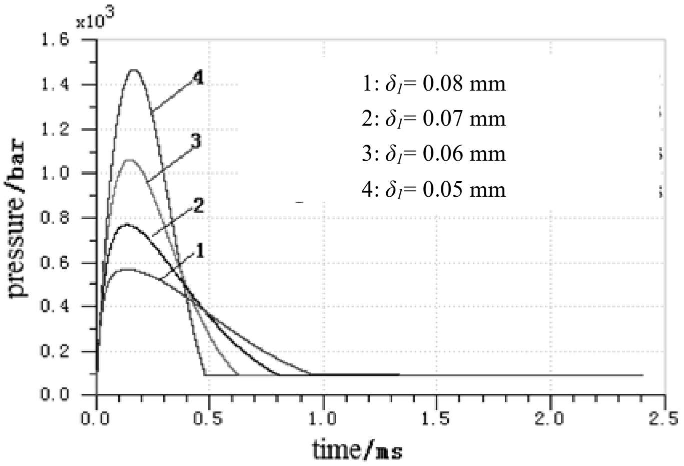

*Fig. 8.* Pressure curve of the secondary damping chamber.

Fig. 5, 6 were the velocity and displacement curve of damping piston. Fig. 7, 8 were the pressure curve of the primary and secondary damping chambers. We could see that:

1) The damping piston's velocity decreased faster and the final return-speed was smaller if the double damping clearance was smaller.

2) The damping piston's displacement was smaller if the double damping clearance was smaller. And the range of the displacement was from 0.9 mm to 2.5 mm.

3) The range of the damping period was from 1.3 ms to 2.5 ms. The shorter damping cycle corresponded to the smaller double damping clearance.

4) The pressure of primary damping chamber changed little and the range was from 8.5 to 9.2 MPa.

5) The double damping clearance had a great effect on the pressure of the secondary damping chamber. The peak pressure ranged from 50 to 150 MPa with different $\delta_l$.

The double damping clearance $\delta_l$ should be 0.07 mm through comprehensive consideration of the spatial structure, processing difficulty and damping performance.

### 7. Conclusion.

Key parameters of the double damper system such as the unloaded damping pressure, damping flow, damping piston's equilibrium position and double damping clearance were designed. The design methods were applied to the YYG170 hydraulic rock drill. The double damper system's dynamic characteristics were simulated. The advantage of responsive and efficient in absorbing the rebounding energy was verified. The research had a guiding significance to the research and development of the hydraulic rock drill with double damper system.

РЕЗЮМЕ. Відсутність досліджень в області подвійної системи демпфування серйозно обмежувало збільшення потужності гідравлічного буріння. Описано структуру і принцип роботи системи подвійного демпфера. Досліджено ключові параметри, такі як розвантаження тиску демпфування, демпфуючий потік, положення рівноваги демпфуючого поршня і подвійний просвіт демпфера. Нарешті, метод розрахунку застосовано для системи гідравлічного буріння YYG170. Переваги подвійної системи демпфування, які полягають у високій ефективності і швидкодії, підтверджуються симуляцією.

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