Research on Equivalent Thickness of Oil and Gas Spring Damping Valve

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Abstract. Analytical algorithm for deflection analysis of superimposed throttle plate of oil and gas spring damping valve was studied, the theory of equivalent thickness of superposed valve plates was put forward, it can accurately calculate the equivalent thickness of a single valve plate corresponding to the superimposed valve plate groups, which greatly facilitates engineering calculations. After that, the boundary flow theory was used to derive the flow expression of the gap under the turbulent state. The relationship between the output force and displacement of the oil and gas spring was simulated and analyzed, compared with experimental data, the correctness of the mathematical model was verified.

1. Introduction
The damping device often uses superimposed valve plates of different thicknesses to generate damping force for oil throttling. This assembly method has the following advantages: 1. Throttle valve plate is processed by elastic steel plates of different standard thicknesses, which is conducive to product standardization and serialization, and is easy to achieve mass production to reduce costs; 2. It can reduce the concentration of stress and increase the service life of the damper valve by reducing the thickness of a single throttle valve plate; 3. By adjusting the thickness and number of superimposed valve plates, the purpose of changing the amount of deformation of the valve plate and adjusting the damping force can be achieved.[1-2]

In view of this situation, a theoretical study on the equivalent thickness of the valve plate is carried out, so that the same deformation amount is produced when different valve combinations are subjected to the same load, to ensure the stability of the damper valve opening, and to improve the design accuracy of the vibration reduction device.

2. Structure diagram
Figure 1 is simplified structural diagram of oil and gas spring. The throttle plate is installed at the end of the piston. During the reciprocating movement of the piston, pressure difference occurs between the upper and lower chambers to deform the valve plate, and oil flows through the annular gap between the deformed valve plate and the piston body. The nitrogen and oil are separated by floating piston, and the length of the gas chamber changes with the change of oil and gas spring stroke.

Figure 2 is assembly schematic diagram of throttle plate. In the figure 2, Δp is the pressure on the thin plate, δ0 is the initial gap width, Δδ is the dynamic gap width after the throttle valve is deformed, r_a is the inner radius of the valve plate, r_o is the outer radius before the valve plate is deformed, and r is the outer...
radius after the valve plate is deformed.

**Figure 1.** Oil and gas spring diagram

**Figure 2.** Schematic diagram of throttle valve deformation

### 3. Equivalent thickness of superposed throttle plate

The inner ring of the annular thin plate is fully constrained, and the free uniform load deformation of the outer ring is: [3]

\[
f = C_o \frac{\Delta p R^4}{E h^3}
\]

(1)

Where \( f \) is the deformation of the outer edge of the throttle plate, \( R \) is the outer radius of the valve plate, \( C_o \) is the deformation coefficient, \( E \) is the elastic modulus of the valve plate, and \( h \) is the thickness of the valve plate.

According to the parallel and split principle of superimposed valve plates, the equivalent thickness relationship of superimposed valve plates under different thicknesses can be derived:

\[
h_i = \left( h_i^3 + h_i^3 + h_i^3 + \cdots + h_i^3 \right)^{\frac{1}{3}}
\]

(2)

Where \( h_i \) is the thickness of single valve plate, \( i=1,2, \ldots, n \).

And the equivalent thickness relationship of valve plate superposition under the same thickness:

\[
h = h_p (n_p)^{\frac{1}{3}}
\]

(3)

Where \( h_p \) is the thickness of single superimposed valve plate, and \( n_p \) is the number of superimposed valve plates.

If there are multiple groups of different thickness of valve plates, the equivalent thickness is:

\[
h_e = \left( n_1 h_1^3 + n_2 h_2^3 + n_3 h_3^3 + \cdots + n_n h_n^3 \right)^{\frac{1}{3}}
\]

(4)

Where \( n_1 \ldots n_n \) is the number of valve plates in different groups.

For example, from the above formulas, it can be estimated that under the same pressure, when 2 pieces of \( 4 \times 10^{-4} \) m thick pieces or 3 pieces \( 3.5 \times 10^{-4} \) m thick pieces are stacked together, the deformation is basically equivalent to the single piece of \( 5 \times 10^{-4} \) m thick.

According to the corresponding relationship between the deformation size of the throttle plate, the dynamic gap width is derived:

\[
\Delta \delta = (r_b - r_e) - \left[ (r_e - r_e)^2 - f^2 \right]^{\frac{1}{2}}
\]

(5)

Then the total width of the gap is:

\[
\delta = \delta_0 + \Delta \delta
\]

(6)

Derive the outer radius of the valve plate after deformation:
\[ r = r_a + \left[ (r_b - r_a)^2 - f^2 \right]^{\frac{1}{2}} \]  

In addition, the stress expression of the valve plate is: [4]

\[ \sigma_{\text{max}} = C \cdot \frac{F}{2h^2} \]  

Where the subscript is \( i = 1,2,3 \), \( F \) is the total load under different stress modes.

It can be seen from the above derivation that the elastic modulus \( E \) is equal when the radial structure size and material properties of the superimposed valve plate are the same.

Sorted out:

\[ (F : F_1 : F_2 : \cdots : F_n) = (h_1^3 : h_1^3 : h_2^3 : \cdots : h_n^3) \]  

\[ (\sigma_{\text{max}} : \sigma_{\text{max}1} : \sigma_{\text{max}2} : \cdots : \sigma_{\text{max}n}) = (h_1 : h_1 : h_2 : \cdots : h_n) \]  

Where \( \sigma_{\text{max}} \) is the maximum stress value of the valve plate equivalent thickness, \( \sigma_{\text{max}1} \cdots \sigma_{\text{max}n} \) are the maximum stress value of different valve plates.

Let the thickness relationship of the valve plates are:

\[ h_1 > h_2 > \cdots > h_n \]  

\[ h_1 > h_1 > h_2 > \cdots > h_n \]  

\[ F > F_1 > F_2 > \cdots > F_n \]  

Then

\[ \sigma_{\text{max}} > \sigma_{\text{max}1} > \sigma_{\text{max}2} > \cdots > \sigma_{\text{max}n} \]  

It is not difficult to see that when the throttle plates of different thicknesses are stacked together, the thickest one is subjected to the largest partial pressure, and the stress value generated is also the largest compared to other valve plates. In other words, when valve plates of different thicknesses are superimposed, it is only necessary to check whether the thickest one of the plates exceeds the strength limit of the material, if its normal use no problem, the strength of the remaining valve plates can also be used.

4. Analytical calculation of gap flow

At present, the more commonly used is the gap flow analysis formula based on the laminar flow state:

\[ Q_{fc} = \frac{\Delta p B \delta^3}{12 \mu l} \]  

Where \( \mu \) is the dynamic viscosity of the fluid, \( l \) is the length of the gap, and \( B \) is the width of the gap.

In order to meet the design requirements of damping valves, the fluid flow boundary layer theory is used to derive the flow rate analysis formula of the gap in the turbulent state as follows.

\[ Q_f = 2\pi \rho \left[ \frac{\tau}{\mu} y_0^2 + A(y_1 - y_0) + 5 \left( \frac{\tau}{\rho} \right)^{\frac{1}{2}} y_1 \ln y_1 - 10 \left( \frac{\tau}{\rho} \right)^{\frac{1}{2}} y_0 \ln y_0 + 5 \left( \frac{\tau}{\rho} \right)^{\frac{1}{2}} \frac{\delta}{2} \ln \frac{\delta}{2} + B \left( \frac{\delta}{2} - y_1 \right) \right] \]  

\[ A = 10 \left( \frac{\tau}{\rho} \right)^{\frac{1}{2}} \ln \left( \frac{\tau}{\rho} \right)^{\frac{1}{2}} - 16.1 \left( \frac{\tau}{\rho} \right)^{\frac{1}{2}} \ln \nu \]
Where $Q_f$ is the gap flow, $\tau_w$ is the wall shear stress, $\delta$ is the width of the single-sided gap, $\rho$ is the fluid density, and the thickness of the viscous bottom layer is:

$$y_0 = 5\nu \left( \frac{\tau_w}{\rho} \right)^{\frac{1}{2}}$$  \hspace{1cm} (17)$$

The thickness of the transition layer is:

$$y_1 = 30\nu \left( \frac{\tau_w}{\rho} \right)^{\frac{1}{2}}$$  \hspace{1cm} (18)$$

Where $\nu$ is the kinematic viscosity of the fluid.

Based on the principle that the pressure difference between the two ends of the gap is balanced with the frictional force of the fluid on the wall, the formula for the wall shear stress of the gap is derived:

$$\tau_w = \frac{\Delta p \delta}{2l}$$  \hspace{1cm} (19)$$

Where $\Delta p$ is the pressure difference across the gap.

Analytical formula for turbulent flow of annular gap is derived:

$$Q_f = 2\pi r \left[ -127.8\nu + \left( 2.5 \ln \delta + 3 - 2.5 \ln \nu + 2.5 \ln \left( \frac{\Delta p \delta}{2l \rho} \right)^{\frac{1}{2}} \right) \delta \left( \frac{\Delta p \delta}{2l \rho} \right)^{\frac{1}{2}} \right]$$  \hspace{1cm} (20)$$

$$A = 10 \left( \frac{\tau_w}{\rho} \right)^{\frac{1}{2}} \ln \left( \frac{\tau_w}{\rho} \right)^{\frac{1}{2}} - 16.1 \left( \frac{\tau_w}{\rho} \right)^{\frac{1}{2}} - 10 \left( \frac{\tau_w}{\rho} \right)^{\frac{1}{2}} \ln \nu$$  \hspace{1cm} (21)$$

$$B = 6 \left( \frac{\tau_w}{\rho} \right)^{\frac{1}{2}} - 5 \left( \frac{\tau_w}{\rho} \right)^{\frac{1}{2}} \ln \nu + 5 \left( \frac{\tau_w}{\rho} \right)^{\frac{1}{2}} \ln \left( \frac{\tau_w}{\rho} \right)^{\frac{1}{2}}$$  \hspace{1cm} (22)$$

Where $Q_f$ is the gap flow, $\tau_w$ is the wall shear stress, $\delta$ is the width of the single-sided gap, $\rho$ is the fluid density, and the thickness of the viscous bottom layer is:

$$y_0 = 5\nu \left( \frac{\tau_w}{\rho} \right)^{\frac{1}{2}}$$  \hspace{1cm} (23)$$

The thickness of the transition layer is

$$y_1 = 30\nu \left( \frac{\tau_w}{\rho} \right)^{\frac{1}{2}}$$  \hspace{1cm} (24)$$

Where $\nu$ is the kinematic viscosity of the fluid.

Based on the principle that the gap pressure difference is balanced with the frictional force of the fluid on the wall, the formula for the shear stress of the gap is derived:

$$\tau_w = \frac{\Delta p \delta}{2l}$$  \hspace{1cm} (25)$$

Where $\Delta p$ is the pressure difference across the gap.

Get the annular gap flow expression through the above:
\[ Q_f = 2\pi r \left[ -127.8\nu + \left( 2.5\ln \frac{\delta}{2} + 3 - 2.5\ln \nu + 2.5\ln \left( \frac{\Delta p\delta}{2L\rho} \right)^{\frac{1}{2}} \right) \right] \] (26)

5. Mathematical model of oil and gas spring

The output force of the oil and gas spring is:[6]

\[ F_z = F_d + F_t \] (27)

Where \( F_z \) is the total output force of the oil and gas spring, \( F_d \) is the damping force, and \( F_t \) is the elastic force.

Simultaneously, the above equations can be used to solve the pressure difference across the gap. Also consider the local resistance loss of the oil passage:

\[ \Delta p_j = \xi \frac{\rho (A_h V)^2}{2A_j^2} \] (28)

Where \( \xi \) is the local loss coefficient, \( A_h \) is the effective working area of the piston, \( A_j \) is the cross-sectional area of the oil passage, and \( V \) is the working speed of the oil and gas spring.

The resistance value of the oil and gas spring is:

\[ F_d = (\Delta p + \Delta p_j) A_h \] (29)

The actual gas pressure expression is

\[ p = \frac{R_g T m_q - a m_q^2}{V_q - m_q b - V_q^2} \] (30)

Where \( p \) is the absolute gas pressure, \( R_g \) is the gas constant, \( T \) is the thermodynamic temperature, \( m_q \) is the gas mass, \( a \), \( b \) is the gas constant, and the volume of the gas is:

\[ V_q = L_j - \frac{A_{gw}s}{A_{gw}} \] (31)

Where \( L_j \) is the length of the air chamber at the static equilibrium position, \( A_{gw} \) is the external cross-sectional area of the piston rod, \( A_{gw} \) is the internal cross-sectional area of the piston rod, and \( s \) is the displacement of the oil and gas spring.

Then the pressure of the oil and gas spring in the static equilibrium position is:

\[ p_j = \frac{2m_q g \xi}{\pi D_{gw}^2} \] (32)

Where \( m_q \) is the sprung mass of single wheel, \( \xi \) is the suspension lever ratio at the static equilibrium position, and \( D_{gw} \) is the outer diameter of the piston rod.

The elastic force of the oil and gas spring is:

\[ F_t = p A_{gw} \] (33)

6. Simulation and experimental research

Input sinusoidal excitation \( f_z = 1 \) Hz, \( A_z = 0.04 \) m, for external characteristic test of oil and gas spring, the test and simulation curves are shown in Figure 3, the force value pairs of each key point are shown in Table 1.
In the table, $F_{z\text{max}}$ is the maximum output force of the oil and gas spring, $F_{z\text{min}}$ is the minimum output force, $F_{z\theta}$ is the return stroke output force at zero displacement, and $F_{z\text{y}}$ is the compression stroke output force at zero displacement.

It can be seen that the simulation curve basically agrees with the experimental data, which proves the correctness of the mathematical model and can provide a reference for the design of the oil and gas spring and its damping valve.

### 7. Conclusion

The mathematical model of the oil and gas spring was simulated by software MATLAB, compared with the external characteristic test data of the developed product, verified the correctness of the equivalent thickness rule for superimposed valve plates, provides convenient conditions for engineering design and theoretical analysis.

### References

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