Thermodynamic Modeling and Simulation Analysis of Double-tube Damper

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Abstract. At present, most of the special vehicles were also designed with a double-tube damper, which was characterized by convenient installation and low cost. However, due to the design of the oil storage cavity outside the working cylinder, the heat dissipation effect was not ideal, and the working temperature of the oil too high will accelerate the aging of the sealing member, which was also a key problem that plagued the reliability design of the damper. Aiming at the structural characteristics of the double-tube damper, the heat generation mechanism and energy conversion research of the oil were carried out, and the heat transfer coefficient and heat conduction equation of the inner and outer surfaces of the cylinder were derived. The comprehensive heat transfer expression was solved for the working medium, and the thermodynamic model of the double-tube damper was constructed. Finally, the simulation analysis of the influence of different structural parameters on the heat balance temperature were carried out, which could effectively guide the design and development of the damper.

1. Introduction
When the vehicle is driving under various road conditions, in order to reduce the influence of ground excitation on the ride comfort, the vibration damping device is usually installed between axle and frame. The current application is more widely used as a double-tube damper, mainly through the damping valve to generate damping force on the oil throttling to attenuate the body vibration, the vibration energy consumed is converted into heat energy, which causes the temperature of the cylinder to rise continuously, which has great influence on the seal. The double-tube damper is simple in process and low in cost, but it is difficult to dissipate heat. When the temperature of the cylinder reaches and exceeds the limit temperature that the seal can withstand, it is easy to cause oil leakage. Based on this situation, this paper systematically studies the thermodynamic model of the double-tube damper [1-2].

2. Thermodynamic physical model of double barrel damper
The thermodynamic physical model of the established double cylinder hydraulic damper is shown in Figure 1. The two cylinders are set together, and a working chamber of the damper is formed in the inner cylinder, and an oil storage chamber is arranged between the inner and outer cylinders for storing and compensating the oil in the working chamber. The oil storage chamber is usually filled with a certain pressure of air, according to the conventional design, the oil can be set to account for half of the volume of the oil storage chamber. A piston is arranged at the lower end of the piston rod, which supports and guides the inner cylinder, and isolates the working chamber into two upper and lower chambers. The compression and recovery valve are mounted on the piston to attenuate the
The vibration of the vehicle body by damping the oil throttling. The bottom valve isolates the lower chamber from the oil reservoir, and has a function of draining and replenishing the working fluid to prevent the damper from being subjected to aerodynamic distortion.

![Shock absorber thermodynamic physical model](image)

**Figure 1.** Shock absorber thermodynamic physical model

### 3. Thermodynamic Mathematical Model of Double-tube Damper

The heat transfer mode of the damper is usually dominated by heat conduction and heat convection. Heat conduction, also referred to as heat conduction, refers to the transfer of heat from the high temperature part of the object to the low temperature part, or the transfer of a high temperature object to the low temperature object in contact therewith. When the temperature of the inner and outer walls of the cylinder is different, heat conduction occurs inside the solid. The thermal differential equation of the cylindrical coordinate system is as follows [3-4]:

\[
\rho c \frac{\partial t}{\partial \tau} = \frac{1}{r} \frac{\partial}{\partial r} \left( \lambda r \frac{\partial t}{\partial r} \right) + \frac{1}{r^2} \frac{\partial}{\partial \varphi} \left( \lambda \frac{\partial t}{\partial \varphi} \right) + \frac{\partial}{\partial z} \left( \lambda \frac{\partial t}{\partial z} \right) + q_v
\]

(1)

Where \( \rho \) is the cylinder density, \( c \) is the specific heat capacity of the cylinder material, \( \lambda \) is the thermal conductivity of the cylinder material, \( q_v \) is the heat generation rate of the internal heat source, \( t \) is the Celsius temperature, and \( \tau \) is the time of heat transfer.

In the operating temperature range of the damper, once the cylinder material is determined, its thermal conductivity can be set to a constant value, \( q_v=0 \) when there is no internal heat source; in addition, the temperature field of the cylinder heat conduction is one-dimensional heat conduction symmetrically distributed along the axis. So equation (1) can be simplified as follows:

\[
\frac{1}{r} \frac{d}{dr} \left( r \frac{dt}{dr} \right) = 0
\]

(2)

According to the heat conduction differential equation and Fourier's law, the heat conduction equations of the inner and outer cylinder walls are obtained [5-6].

\[
Q_2 = -2\pi L_2 \lambda g \frac{T_{w3} - T_{w2}}{\ln (r_{no} / r_{w2})}
\]

(3)

\[
Q_3 = -2\pi L_2 \lambda g \frac{T_{w3} - T_{w4}}{\ln (r_{no} / r_{w4})}
\]

(4)
Where $Q_w$ is the heat transferred by the cylinder wall, $r_{in}$ is the inner cylinder radius, $r_{hw}$ is the inner cylinder outer radius, $r_{int}$ is the outer cylinder inner radius, and $r_{out}$ is the outer cylinder outer radius.

In the oil reservoir, the upper part is the air with pressure and the lower part is the oil. Since the liquid level does not fluctuate greatly when the damper is working, the medium in the oil storage chamber mainly radiates heat to the outside in the way of heat conduction. According to formula (2) and Fourier's law, the heat conduction of air and oil can be separately derived:

$$Q_{ck} = -\pi L_{c} \lambda_{k} g \frac{T_{u2} - T_{u3}}{\ln \left( \frac{r_{uw}}{r_{uw}} \right)}$$

$$Q_{cy} = -\pi L_{c} \lambda_{y} g \frac{T_{u2} - T_{u3}}{\ln \left( \frac{r_{uw}}{r_{uw}} \right)}$$

Where $\lambda_{k}$ is the thermal conductivity of air, $\lambda_{y}$ is the thermal conductivity of the oil, $Q_{ck}$ is the heat transferred by the air in the reservoir, and $Q_{cy}$ is the heat transferred by the oil in the reservoir.

The simultaneous expressions (3)-(6) are combined according to the heat transfer relationship to obtain a comprehensive expression of heat conduction:

$$Q_{w} = \pi L_{c} \left( T_{u4} - T_{u4} \right) \left[ \frac{\ln \left( \frac{r_{uw}}{r_{uw}} \right)}{2\lambda_{k}} + \frac{\ln \left( \frac{r_{uw}}{r_{uw}} \right)}{2\lambda_{y}} + \frac{\ln \left( \frac{r_{uw}}{r_{uw}} \right)}{\left( \lambda_{k} + \lambda_{y} \right)} \right]$$

According to the theory of heat transfer, the development of the boundary layer of air near the wall of the cylinder is not disturbed by space constraints, and it belongs to the natural convection heat transfer in large space. The expression of the heat transfer coefficient on the outer surface of the cylinder in the vertical state can be derived by the computational correlation proposed by S W Churchill [7]:

$$h_{w0} = \frac{\lambda_{k}}{L_{c}} \left\{ 0.825 + \frac{0.387 \left( Gr \cdot Pr_{k} \right)^{1/6}}{1 + \left( 0.492 / Pr_{k} \right)^{9/16}} \right\}^{2}$$

Prandt number:

$$Pr_{k} = \frac{v_{k}}{\lambda_{k}} \rho_{k} c_{k}$$

Grachev number:

$$Gr = \frac{g \beta \left( T_{u4} - T_{u4} \right) L_{c}^{3}}{v_{k}^{2}}$$

Where $c_{k}$ is the specific heat capacity of air, $\rho_{k}$ is the air density, and $v_{k}$ is the kinematic viscosity of air. $\beta = \frac{1}{V} \left( \frac{\partial V}{\partial T} \right)_{p}$ is the volume expansion coefficient, also known as the body expansion coefficient.

Heat exchange between the studio fluid and the inner cylinder is forced convection, so the forced convection heat transfer coefficient of the inner surface of the inner cylinder under the laminar flow state is derived as follows:

$$h_{w0} = \frac{\lambda_{k}}{L_{c}} \left\{ 3.66 + \frac{0.0688 V D_{int}^{2} Pr_{y}}{v_{k} L_{c} + 0.04 L_{c} \left( V D_{int} Pr_{y} \left( D_{int} / L_{c} \right) \right)^{2/7}} \right\}$$
Where $\lambda$ is the thermal conductivity of the oil, $\nu$ is the kinematic viscosity of the oil, and $Pr_y$ is the Prandtl number of the oil.

And the forced convection heat transfer coefficient of the inner surface under turbulent flow:

$$h_{iyl} = R \cdot \frac{\lambda}{L_y} \cdot \frac{M \cdot N \cdot Pr_y}{1 + B \left(Pr_y^{2/3} - 1\right)}$$

Where:

$$R = 1 + \left(\frac{D_{hn}}{L_y}\right)^{0.7}, \quad M = \frac{1}{8} \left(0.79 \ln \left(\frac{VD_{hn}}{L_y}\right) - 1.64\right)^2$$

$$N = \frac{VD_{hn}}{L_y} - 1000 \cdot B = 25.4 \sqrt{2} \cdot \left[0.79 \ln \left(\frac{VD_{hn}}{L_y}\right) - 1.64\right]^{1/3}$$

From the formula (7) and the Newtonian cooling formula, the heat expression of the oil to the external environment can be derived:

$$Q_n = (T_y - T_x) \cdot \left[\frac{\ln \left(\frac{r_{in}}{r_{in1}}\right)}{2\lambda \pi L_y} + \frac{\ln \left(\frac{r_{out1}}{r_{out}}\right)}{2\lambda \pi L_y} + \frac{\ln \left(\frac{r_{out1}}{r_{out}}\right)}{\pi L_y \left(\lambda_k + \lambda_y\right)} + \frac{1}{h_{iyl} A_n} + \frac{1}{h_{w0} A_{w1}}\right]^{-1}$$

Where $A_{in}$ is the inner surface area of the inner cylinder and $A_{out}$ is the outer surface area of the outer cylinder.

Taking oil as the research object and deriving according to the principle of conservation of energy:

$$m_y c_y p \frac{dT_y}{dt} + B_y T_y = Y + B_T T_w$$

Where:

$$B_y = \left[\frac{\ln \left(\frac{r_{in}}{r_{in1}}\right)}{2\lambda \pi L_y} + \frac{\ln \left(\frac{r_{out1}}{r_{out}}\right)}{2\lambda \pi L_y} + \frac{\ln \left(\frac{r_{out1}}{r_{out}}\right)}{\pi L_y \left(\lambda_k + \lambda_y\right)} + \frac{1}{h_{iyl} A_n} + \frac{1}{h_{w0} A_{w1}}\right]^{-1}$$

Damping valves are calculated by orifice and slit throttling:

$$Y = \frac{12 \mu_{eff} (A_v V)^2}{\pi \delta^3 B_j} e^{\lambda_j (t_y - t_{in})} + \left(\frac{A_v V}{2(C_d A_k)^2 \rho_y}\right)^{3/2}$$

Where $m_y$ is the oil mass, $c_y p$ is the constant pressure specific heat capacity of the oil, $\rho_y$ is the oil density, $A_v$ is the orifice area, the flow coefficient $C_y = 0.62$, $\delta_j$ is the width of the gap, $B_j$ is the wet circumference of the gap, $l_j$ is the length of the gap, $\mu_{in}$ is the initial dynamic viscosity of the oil, $\lambda_j$ is the viscosity temperature index of the oil, and $A_k$ is the effective working area of the piston.

The simultaneous (3) to (16) form a thermodynamic mathematical model of the double barrel damper.

If the outer wall of the damper is painted black, the amount of heat transfer from the heat radiation must also be considered. The expression of the amount of radiated heat transfer is directly listed below:

$$Q_{of} = e C_v \left[\frac{T_{w4}}{100}\right]^4 - \left(\frac{T_{w1}}{100}\right)^4\right] A_{w1}$$

Where Blackbody emissivity is $C_v = 5.67 W/m^2 \cdot K^4$, $\varepsilon$ is the rate of emissivity for thermal radiation.
The expression of the composite heat transfer coefficient of the outer wall of the outer cylinder is derived as follows:

\[ h_{wof} = \varepsilon C_b \left( \frac{T_w^2 + T_{\infty}^2}{(100)^4} \right) + h_{wo} \]  

Equation (18)

Combined with equation (7) and Newtonian cooling formula, the comprehensive heat expression of oil released to the outside world considering heat radiation is obtained:

\[ Q_w = (T_y - T_{\infty}) \left[ \frac{\ln\left(\frac{r_{bw}}{r_{wo}}\right)}{2\lambda\pi L_y} + \frac{\ln\left(\frac{r_{in1}}{r_{in2}}\right)}{2\lambda\pi L_y} + \frac{\ln\left(\frac{r_{in2}}{r_{inw}}\right)}{\pi L_y \left(\lambda_y + \lambda_{in}\right)} + \frac{1}{h_{wof} A_{in}} + \frac{1}{h_{wo} A_{in}} \right]^{-1} \]

Equation (19)

The above expressions constitute a thermodynamic mathematical model of a double-tube damper considering thermal radiation.

4. Thermodynamic model simulation analysis

The thermodynamic model of the damper is iteratively operated by software Matlab programming. The relevant parameters are as follows: \( \delta_f = 0.1 \times 10^{-3} \text{m} \), \( B_f = 0.25 \text{m} \), \( C_s = 0.62 \), \( I_f = 1 \times 10^{-3} \text{m} \), \( \lambda = 23 \text{W/(m\cdot\text{k})} \), \( T_s = 300 \text{K} \), \( \rho_w = 890 \text{kg/m}^3 \), \( \lambda_w = 0.02 \), \( \rho_h = 1.293 \text{kg/m}^3 \), \( T_h = 273 \text{K} \), \( \mu_{in} = 8.9 \times 10^{-7} \text{kg/(m\cdot\text{s})} \), the initial temperature of the system is equal to the ambient temperature.

Input harmonic excitation \( A = 0.05 \text{m} \), \( f = 1 \text{Hz} \), set the damper to exchange energy with the outside world every time it works, cycles \( X = 5000 \), single cycle time \( t = \frac{1}{f} \).

![Figure 2. Different cylinder lengths](image1)

![Figure 3. Different outer cylinder diameters](image2)

![Figure 4. Different inner cylinder diameters](image3)

![Figure 5. Different piston rod diameters](image4)
Figure 6. Thermal radiation oil temperature comparison

Through the above figures 2-6, the following conclusions can be drawn: under the condition that the structure size of the damping valve is constant, as the axial dimension of the cylinder and the radial dimension of the piston rod increase, the temperature rise of the damper oil is gradually reduced in the same time. With the increase of the radial dimension of the inner and outer cylinders, the temperature rise of the damper oil gradually increases in the same time; when the outer surface of the damper is sprayed with dark or black paint, the surface thermal radiation emissivity increases, and the temperature rise of the damper oil gradually decreases.

5. Conclusion
(1) When the structural parameters of the damping valve are constant, increasing the axial dimension of the cylinder and the radial dimension of the piston rod, or reducing the radial dimension of the inner and outer cylinders, can reduce the temperature rise of the damper oil.
(2) In order to increase the heat dissipation of the system, dark or black paint can be sprayed on the outer surface of the damper to increase the emissivity of the heat radiation, thereby effectively slowing the temperature rise of the oil.
(3) As time goes by, the system will reach the temperature balance point. It can avoid the leakage phenomenon by adjusting the relevant parameters of the damper and controlling the balance point within the limit temperature range of the seal.

References
[1] Everaert K 2006 Heat transfer from a single tube to the flowing gas-solid suspension in a CFB riser Heat Transfer Engineering 27 66-70
[2] Khan W A, Culham J R and Yovanonich M M 2006 Fluid flow and heat transfer in power-law fluids across circular cylinders Journal of Heat Transfer 128 870-8
[3] Morales A L, Nieto A J and Chicharro J M 2015 An adaptive pneumatic system for the attenuation of random vibrations Journal of Vibration and Control 21 907-18
[4] Konieczny L, Burdzik R and Wegrzyn T 2016 Analysis of Structural and Material Aspects of Selected Elements of a Hydropneumatic Suspension System in a Passenger car Archives of Metallurgy & Materials 61 79-83
[5] Li Z L, Li X Q and Chen X D 2014 Generic vibration criteria-based dual-chamber pneumatic spring vibration isolation table design P I Mech Eng B-J Eng. 228 1621-29
[6] Theron N J and Els P S 2007 Modeling of a Semi-active Hydro-pneumatic Spring-damper Unit International Journal of Vehicle Design 45 501-21
[7] Chai T, Dreyer J T and Singh R 2015 Nonlinear dynamic properties of hydraulic suspension bushing with emphasis on the flow passage characteristics P I Mech Eng D-J Aut. 229 1327-44