A study on the wear allowance of a railway vehicle hydraulic damper

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Abstract. Railway vehicle hydraulic dampers are usually subject to wear-induced premature failures, so it is meaningful to evaluate the wear allowance of the hydraulic damper. Mathematical modelling of damping characteristics of a railway vehicle hydraulic damper under wear conditions is carried out, the model describes the functions of main damping characteristics indices vs. inner tube wear $M_1$ and rod wear $M_2$. Simulation is then performed using the model, and indicates that increase of $M_1$ would lead to significant drop of the maximum extension damping force, fading of the damping coefficient and significant rising of the damping force asymmetry rate, but the maximum compression damping force keeps to be constant, while increase of $M_2$ would lead to significant drop of almost all of the indices except for the asymmetry rate. A wear allowance line for the damper is finally obtained when both of $M_1$ and $M_2$ work in real situations, the area enclosed by the wear allowance line and two axles is a safe area, any dot with a combination of $M_1$ and $M_2$ outside the safe area means the damper has exceeded its wear allowance and should be maintained or replaced. The mathematical model and the wear allowance line obtained in this work can be used for health status evaluation and design optimization of railway hydraulic dampers.

1. Introduction

Hydraulic dampers [1-2] are key components for running stability and ride comfort of modern railway vehicle systems [3], however, as shown by Figure 1, hydraulic dampers are usually subject to wear-induced premature failures. Therefore, it is meaningful to establish damping characteristics model under wear conditions and evaluate the wear allowance of the hydraulic damper.

Figure 1. Wear-induced premature failure of a railway vehicle hydraulic damper.
In previous works, Gallardo et al [4] investigated the service failure of an automobile shock absorber with welding imperfections, and found that poor positioning of the items to be joined and prior contamination of the surfaces to be welded would lead to premature failures of the damper attachment. Reza et al [5] studied the fretting fatigue failure and crack under cyclic loading of the shim valve of an automotive shock absorber, Zheng [6] found that the main cause of sealing failure is due to vibration with high frequency and low amplitude, Wang et al [7] built fluid leakage mathematic model to illustrate the effect of fluid leakage on damping characteristics. In addition, simple parametric model [8] and complex parametric models [9,10] were built to describe the dynamic damping characteristics in recent years, however, research work on modelling damping characteristics under wear conditions and evaluate the wear allowance of the damper shows inadequate.

In this work, mathematical modelling of the damping characteristics of a railway vehicle hydraulic damper under wear conditions is firstly carried out, then numerical simulation is performed using the established model, the influence of inner tube wear $M_1$ and rod wear $M_2$ on main damping characteristics indices is obtained and analysed, therefore, a wear allowance line for the hydraulic damper is finally obtained to evaluate the health status of the hydraulic damper.

2. Modelling the damping characteristics under wear conditions

Figure 2 illustrates the physical wear models of the two friction pairs in a railway vehicle hydraulic damper under high-frequency but low-amplitude vibration conditions. Fig. 2(a) shows the wear and gap of the inner tube which would cause inner leakage of the fluid, while Fig. 2(b) shows the wear and gap of the rod which would cause outer leakage of the fluid.

![Figure 2. Physical wear models of (a) the friction pair of piston and inner tube and (b) the friction pair of rod and guide seat in a railway vehicle hydraulic damper.](image-url)
If only considering pressure leakage between the friction pairs, the leakage coefficients of the inner tube and the rod can be respectively formulated by

\[ t_1(M_1) = \frac{\pi(D + 2M_1)(\delta_1 + M_1)^3}{12c_1 \mu l_{s1}} \]  

and

\[ t_2(M_2) = \frac{\pi(D - 2M_2)(\delta_2 + M_2)^3}{12c_2 \mu l_{s2}} \]

where \( M_1, M_2 \) are respectively wear-induced gaps in the inner tube and the rod, \( D, d \) are respectively inner diameter of the inner tube and diameter of the rod, \( \delta_1, \delta_2 \) are respectively initial clearance between the inner tube and piston and that between the guide seat and rod, \( l_{s1}, l_{s2} \) are respectively widths of the piston seal and rod seal, \( c_1, c_2 \) are respectively correction factors for laminar flow at the initial section of the two friction pairs, \( \mu \) is dynamic viscosity of the fluid.

Similarly, the leakage coefficient at the two end of the inner tube can be formulated by

\[ t_3 = \frac{\pi \delta_3^3}{6c_3 \mu \ln(R/r)} \]

where \( \delta_3 \) is initial clearance between the inner tube end and the seal, \( c_3 \) is correction factor for laminar flow at the initial section of the inner tube end seal, \( R \) and \( r \) are respectively outer radius and inner radius of the inner tube end.

Therefore, for a hydraulic damper in which the fluid circulates in one direction [9], the total fluid leakage coefficients can be formulated by

\[ K_{hy}(M_1, M_2) = t_1(M_1) + t_2(M_2) + t_3 \]  

for the extension stroke, and

\[ K_{hc}(M_2) = t_2(M_2) + 2t_3 \]

for the compression stroke, thus, the fluid continuity equations of the damper are formulated by

\[ vA = C_d A \sqrt{\frac{2 F_{\text{max}}}{\rho A} + K_{hy}(M_1, M_2) \frac{F_{\text{max}}}{A}} \]

for the extension stroke, and

\[ vA = C_d A \sqrt{\frac{2 F_{\text{max}}}{\rho A} + K_{hc}(M_2) \frac{F_{\text{max}}}{A}} \]

for the compression stroke, where \( v \) is vibration speed, \( A \) is effective action area of the piston, \( C_d \) is discharge coefficient, \( A_1 \) is discharge flow area of the damping valve, \( \rho \) is density of the fluid, \( F_{\text{max}} \), \( F_{\text{cmax}} \) are respectively maximum damping forces at the extension stroke and the compression stroke.

The relation between damping force and wear can be deduced from Equations (6) and (7), i.e., the functions are

\[ F_{\text{max}} = F_{\text{max}}(M_1, M_2) \]

for the extension stroke, and

\[ F_{\text{cmax}} = F_{\text{cmax}}(M_2) \]

for the compression stroke.

Therefore, according to Equations (8) and (9), the damping coefficient and the asymmetry rate of the hydraulic damper can be respectively formulated by

\[ C(M_1, M_2) = \frac{|F_{\text{max}}(M_1, M_2)| + |F_{\text{cmax}}(M_2)|}{2v} \]
\[ A_{\text{sy}}(M_1, M_2) = \frac{\left| F_{\text{cmax}}(M_2) \right| - \left| F_{\text{xmax}}(M_1, M_2) \right|}{\left| F_{\text{cmax}}(M_2) \right| + \left| F_{\text{xmax}}(M_1, M_2) \right|} \times 100\% \]  

(11)

3. Wear allowance evaluation

Simulation is performed using the above established damping characteristics model under wear conditions, the geometric value of main parameters include \( D = 70 \text{ mm} \), \( d = 28 \text{ mm} \), \( A = 3231.06 \text{ mm}^2 \), \( A_v = 3.14 \text{ mm}^2 \), \( R = 38 \text{ mm} \), \( r = 35 \text{ mm} \), \( \delta_1 = 0.1 \text{ mm} \), \( \delta_2 = 0.1 \text{ mm} \), \( \delta_3 = 0.01 \text{ mm} \), \( l_x = 2.5 \text{ mm} \), \( l_z = 11 \text{ mm} \), \( C_d = 0.82 \), \( \rho = 840 \text{ kg/m}^3 \), \( \mu = 1.127 \text{ Pas} \).

Figure 3 shows the effect of inner tube wear \( M_1 \) on damping characteristics of the damper, and indicates that increase of inner tube wear leads to significant drop of the maximum extension damping force, while the maximum compression damping force keeps to be constant, this is because fluid leakage at the inner tube is inner leakage, it does not affect the damping force of the compression stroke. Therefore, as shown in Fig. 3, that increase of inner tube wear would cause fading of the damping coefficient and significant rising of the damping force asymmetry rate.
Similarly, Figure 4 shows the effect of rod wear $M_2$ on damping characteristics of the damper, and indicates that increase of rod wear would lead to significant drop of all of the maximum extension damping force, the maximum compression damping force and the damping coefficient, this is because fluid leakage at the rod is outer leakage, it would affect the damping force all of the times. Therefore, as shown in Fig. 4, that increase of inner tube wear has almost no effect on the asymmetry rate, for both of the maximum extension damping force and the maximum compression damping force drop simultaneously.

Figure 5 gives the result about effect of both $M_1$ and $M_2$ on damping characteristics in real situations. According to the Chinese Railway Standard [2], the allowed maximum rate of damping force drop is 10%, thus, the real line in Fig. 5 is a critical line for the maximum extension damping force $F_{\text{xmax}}$, the area enclosed by the critical line and two axles is a safe area, if the rod wear is outside the safe area, the damper would be regarded as in failure.

Similarly, the dotted line in Fig. 5 is a critical line for the maximum compression damping force $F_{\text{cmax}}$, the area enclosed by the critical line and two axles is a safe area, if the inner tube wear is outside the safe area, the damper would be regarded as in failure.

![Figure 5](image.png)

Figure 5. The effect of both $M_1$ and $M_2$ on damping characteristics in real situations.

Therefore, the shaded area in Fig. 5 is the final obtained safe area when both of $M_1$ and $M_2$ works, which is true in real situations. In other words, the real line in Fig.5 is the wear allowance line for the hydraulic damper, any dot with a combination of $M_1$ and $M_2$ outside the wear allowance line means that the damper has exceeded its wear allowance and should be maintained or replaced.

4. Conclusions
(1) Mathematical modelling of the damping characteristics of a railway vehicle hydraulic damper under wear conditions is carried out, the model describes the functions of main damping characteristics indices vs. inner tube wear and rod wear, then simulation is performed using the established model.

(2) Increase of inner tube wear $M_1$ would lead to significant drop of the maximum extension damping force, while the maximum compression damping force keeps to be constant, so $M_1$ would lead to fading of the damping coefficient and significant rising of the damping force asymmetry rate; increase of rod wear $M_2$ would lead to significant drop of all of the maximum extension damping force, the maximum compression damping force and the damping coefficient, but has almost no effect on the asymmetry rate.
(3) A wear allowance line for the hydraulic damper is obtained when both of $M_1$ and $M_2$ works in real situations, the area enclosed by the wear allowance line and two axles is a safe area, any dot with a combination of $M_1$ and $M_2$ outside the safe area means the damper has exceeded its wear allowance and should be maintained or replaced.

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