Analysis of dynamic behaviour of high-speed railway vehicle with faulty anti-hunting damper

Wang Yi-Xuan*, Chen En-Li, Qi Zhuang, Liu Peng-Fei, Zhang Lin

School of Mechanical Engineering, Shijiazhuang Tiedao University, Hebei Provincial Key Laboratory of Traffic Safety and Control, Shijiazhuang 050043, Hebei, China

(Manuscript Received: 2 July 2017; Revised: 25 August 2017; Accepted: 9 September 2017)

Abstract

With factors such as temperature, pressure and the percentage of the air in oil taken into consideration, the relation between the force of the faulty damper and the gap of oil seal is analyzed by observing the change of the oil stiffness while the oil seal was worn continually. And the influence of the faulted damper on the stability, running performance and the curve negotiation capability of high-speed train is analyzed by using the Adams/Rail software package. The full vehicle models with different degrees of faulty anti-hunting dampers were simulated. Results show that the excessive worn of the oil seal of the anti-hunting damper can lead to the decrease of its oil stiffness, which reduces its actual damper force. The working condition of anti-hunting damper has great effect on train stability. If the worn oil seal deteriorates continually, the nonlinear critical velocity of the high-speed vehicle would be reduced seriously. When the oil seal gap wears to 0.2 mm, the critical velocity of the vehicle with faulty anti-hunting damper decreases by 21.4% compared to the normal vehicle. And the lateral axial force would be increased by 4.2% when the oil seal gap is worn to 0.2 mm on curve track. However it can hardly affect the vertical running performance.

Keywords: Anti-hunting damper; Faulted damping characteristic; Oil stiffness; Multi-body dynamic simulation

1. Introduction

As the critical component of high-speed vehicle to cushion and absorb vibration, the accuracy of hydraulic damper model and the selection of critical parameters were brought into focus [1-4]. Much fault has been found while the hydraulic damper was used frequently, especially the fault of oil leakage, which was the one of the most common faults. Aiming at the problem of the fault of oil leakage of hydraulic damper, Fan You-quan, has built a dynamic mathematical model of damper modulation system by a lot of experimental data., which the influence of various of parameters on damping property [6]. The performance of hydraulic damper and the nodal points of traction rods has been researched by Jiang Gui-shan. And he has also analyzed the relation between the wear of the piston and the oil seal lip and the fault of oil leakage. The relation of the percentage of the air in oil and the damper force has been researched by measuring the dynamic pressure and observing the performance of the squeeze film damper (SFD) by Andres [8].

The dynamic performance of the hydraulic damper has been researched by comparing between the new and the old damper by Xu [9]. These studies pay more attention to the research of the theoretical model, the failure mechanism of damper’s internal leakage and the results of the measurement of the damper’s external leakage. The study of the theoretical model of damper’s external leakage was researched by few researcher. Then, the study about it was carried out in this article. The changes of the damper’s oil stiffness and damping performance were analyzed while the oil seal was worn continually. And the dynamic behaviour of high-speed vehicle with faulty anti-hunting damper was analyzed further. The results of this study can be referred to the design of the hydraulic damper and the condition monitoring of the high-speed vehicle.

2. The model of faulty anti-hunting damper

2.1. The equivalent model of damper

Figure 1. The model of anti-hunting damper
There are rubber nodes at both ends of the damper, especially there are little bubble mixed in oil when the damper is working with a high frequency, the damper would have elastic characteristics. Then, the anti-hunting damper was abstracted to a model with a nonlinear damper and stiffness in series. The equivalent mechanics model of anti-hunting damper is shown in Fig. 1. The dynamic oil stiffness, rubber stiffness and the seat stiffness were equivalent to $K_e$.

$$K_e = \frac{K_{seat}K_{rubber}K_{oil}}{K_{seat}K_{rubber} + K_{seat}K_{oil} + K_{rubber}K_{oil}}$$

(1)

The sine excitation was applied to the end of the damper, $F_0 = F(t) = F_0 \sin(\omega t)$. The displacement of $A$ was $x(t) = x_0 \sin(\omega t)$, Then the displacement of $B$ can be set to $y(t) = y_0 \sin(\omega t + \varphi)$, the force analysis was:

$$c(x - y) + K_e y = 0$$

(2)

The actual amplitude of the damper was:

$$F_0 = K_e y_0 = \frac{K_e \omega}{\sqrt{K_e^2 + (\omega t)^2}} x_0$$

$$c = \frac{K_e}{\sqrt{K_e^2 + (\omega t)^2}}$$

where $c$ was the equivalent damping coefficient; $c_i$ was the equivalent damping coefficient with series stiffness.

2.2. Model of Oil leakage parameters

There is very high pressure in the high pressure oil chamber of the damper. Then, if there are fault that the leakage of oil in the damper, its damping performance must be affected. The dynamic model of the damper’s oil leakage is shown in Fig.2.

![Figure 2. The dynamic flow of damped system [6]](image)

The input of the system in Fig.2 is the $v$, which is the compression or tension velocity of the piston. And the output of the system is the $F$, which is the damping force of the hydraulic damper. The feedback of the system is the $P$, which is the pressure in the high pressure oil chamber of the damper. $f_i$ is the fiction of the system, and the $A$ is the effective area of the piston. The total flow of the system $Q_{all}$ the effective working flow is $Q_{valve}$. And $Q_{loss}$ is the loss flow of the system, which was caused by the oil leakage of the damper. The loss flow of the system consists of volume compression of the oil, internal and external leakage of oil in the damper. And then the external leakage of oil is just the key point in this paper.

The oil stiffness of the damper [10]:

$$K_{oil} = \frac{\beta_e (\pi d^2)^2}{4 V_{oil}} = \frac{\beta_e \pi^2 d^4}{16 V_{oil}}$$

(4)

where $\beta_e$ is the bulk elastic modulus; $V_{oil}$ is the volume of the high pressure oil chamber;

The bulk elastic modulus of the oil in the high pressure oil chamber:

$$\text{Voil} = \left\{ \begin{array}{ll} \frac{\pi}{8}(D^2 - d^2)[H - H_p - 2|x(t)|] & ; x(t) < 0 \\ \frac{\pi}{8}(D^2 - d^2)[H - H_p + 2|x(t)|] & ; x(t) > 0 \end{array} \right.$$

(6)

where $d$ is the diameter of the piston; $H$ is the total height of the damper’s internal cylinder;

$H_p$ is the total height of the piston; $x(t)$ is the displacement...
of the piston.

The dynamic viscosity of the oil:

\[ \mu_t = \frac{\mu_0}{2} [1 + 1.5 \varepsilon + e^{\alpha_p (P - P_t) - \lambda (T - T_0)}] \]  

(7)

where \( \mu_0 \) is the dynamic viscosity when the temperature is \( T_0 \) under standard atmospheric pressure; \( \alpha_p \) is the pressure-viscosity coefficient of the oil; \( \lambda \) is the viscosity-temperature coefficient.

The model of the flow of the external leakage of oil:

\[ Q_{es} = 1.75 \frac{\pi d \delta^3 (P - P_e)}{12 \mu_t} \]  

(8)

where \( \delta \) is the gap between the piston and the oil seal.

The damping valve system of the anti-hunting damper mainly consists of three damping valves, which open in sequence according to the pressure of the high pressure oil chamber. The damper valve 1 contains a normally open throttle valve and a relief valve. And the damper valve 2 and 3 are both relief valves. The opening pressure of the relief valves 2, 3 and 1 are respectively \( P_2 \), \( P_3 \) and \( P_1 \). And they meet the condition: \( P_2 < P_3 < P_1 \). Therefore, there are four different pressure-flow equations [11].

When the pressure of high pressure oil chamber during the operation of damper \( P < P_2 \):

\[ Q_{all} = C_{d0} A_0 \sqrt{\frac{2(P - P_0)}{\rho}} \]  

(9)

When \( P_2 < P < P_3 \),

\[ Q_{all} = C_{d0} A_0 \sqrt{\frac{2(P - P_0)}{\rho}} + C_{d1} A_1 \sqrt{\frac{2(P - P_1)}{\rho}} \]  

(10)

When \( P_3 < P < P_1 \),

\[ Q_{all} = C_{d0} A_0 \sqrt{\frac{2(P - P_0)}{\rho}} + C_{d1} A_1 \sqrt{\frac{2(P - P_1)}{\rho}} + k_{q2} \frac{h_2^3}{\mu_t} P \]  

(11)

When \( P > P_1 \),

\[ Q_{all} = C_{d0} A_0 \sqrt{\frac{2(P - P_0)}{\rho}} + C_{d1} A_1 \sqrt{\frac{2(P - P_1)}{\rho}} + k_{q2} \frac{h_2^3}{\mu_t} P + k_{q3} \frac{h_3^3}{\mu_t} P \]  

(12)

where \( P_e \) is the pressure between the valve bottom and the spool after the damping valve 2 opened; \( C_{d0} \) is the flow coefficient of the normally open throttle valve of damping valve 1; \( C_{d1} \) is the flow coefficient of the damping hole of the damping valve 2; \( A_0 \) is the sectional area of the normally open damping hole of damping valve 1; \( A_1 \) is the sectional area of the damping hole of damping valve 2; \( \rho \) is the density of the oil; \( \rho_e = 841 \); \( k_{q2} \) is the internal leakage coefficient of the damping valve 2; \( k_{q3} \) is the internal leakage coefficient of the damping valve 3; \( h_2 \) is the oil film thickness of the opening damping valve 2; \( h_3 \) is the oil film thickness of the opening damping valve 3.

2.3. The analysis of damping performance of the anti-hunting damper

The material of the oil seal is special-material, which will swell with oil. Therefore, the oil can be prevented from oozing externally by the oil seal when the hydraulic damper working normally. But there will be external leakage of the oil when the oil seal is worn excessively. The relation of the oil stiffness and the gap between the piston and the oil seal is shown in Fig.3, and the relation of the series stiffness and the gap is shown in Fig.4.

In Fig.3, where \( K_{oil} \) is the oil stiffness of the anti-hunting damper; \( v \) is the velocity of the piston of the anti-hunting damper; \( \delta \) is the gap between the piston and the oil seal. And the relation between the oil stiffness of the high pressure oil chamber and the velocity of the piston is analyzed when the \( \delta \) is 0 mm, 0.02 mm, 0.04 mm, and 0.06 mm. The oil stiffness in the high pressure oil chamber decreases with the increase of the gap \( \delta \), which causes the decrease of the series stiffness \( K_e \) of the hydraulic damper. We can know that in Fig.3 and Fig.4, the lower the velocity of the damper’s piston, the more seriously the oil stiffness of the damper will decrease, after the oil seal wearing excessively.

![Figure 3. Anti-hunting damper’s oil stiffness under different oil seal gaps](image-url)
The relation between the loss of the oil stiffness and the velocity of the piston is shown in Fig. 5.

The unloading speed of the anti-hunting damper in this paper is 0.0033 m/s. Therefore, in the section where the speed is lower than the unloading speed, the relation between the oil stiffness and the gap of the oil seal can be achieved by analyzing the condition that the velocity of the piston is 0.003 m/s. The result is shown in Fig. 6.

From the Figs.3-6, it is known that the oil stiffness will reduce after the oil seal wearing excessively. And the decrease of the oil stiffness can cause the decline of the series stiffness of the damper, which can reduce the damping force of the damper. The more seriously the oil seal of the damper wear, the more seriously the damping force decline. The damping performance of the hydraulic damper with no considering oil stiffness, considering oil stiffness, and serious oil leakage are shown in Fig.7.

In Fig.7, we can know that unloading speed of the anti-hunting damper is about 0.0033 m/s, and the unloading force is about 8100 N without considering oil stiffness. And the damping force declined slightly with considering oil stiffness. But if the oil seal of the damper worn excessively, the damping force would decline sharply.

3. Establishing dynamics model

The high-speed vehicle model and the vehicle models with different levels of fault anti-hunting damper are set up in the software Adams/Rail. The model mainly consists of one car body, two bogies, four wheel-sets, and eight axle-boxes. The car body, bogies, wheel-sets, and the traction rods have six DOF, and the axle-boxes have one DOF. Therefore, the full vehicle has sixty-two DOF in total. The model of the bogie and the car body of the high-speed vehicle are shown in Fig.8.

And the critical parameters of the dynamic simulation are listed in Table 1 and Table 2.
Table 1. The structural parameters of main parts

<table>
<thead>
<tr>
<th>The name of the items</th>
<th>Size /mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>Size of the car body</td>
<td>24500×3380×3700</td>
</tr>
<tr>
<td>Center distance of bogie</td>
<td>17500</td>
</tr>
<tr>
<td>Wheel base</td>
<td>2500</td>
</tr>
<tr>
<td>Lateral span of primary suspension</td>
<td>2000</td>
</tr>
<tr>
<td>Lateral span of air spring</td>
<td>2460</td>
</tr>
<tr>
<td>Lateral span of anti-hunting damper</td>
<td>2700</td>
</tr>
</tbody>
</table>

Table 2. The main inertial parameters of the simulation

<table>
<thead>
<tr>
<th>Mass /t</th>
<th>Rotational inertia / (t.m^2)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Car body 28.8</td>
<td>I_{xx} = 93.312</td>
</tr>
<tr>
<td></td>
<td>I_{yy} = 1411.12</td>
</tr>
<tr>
<td></td>
<td>I_{zz} = 1331.712</td>
</tr>
<tr>
<td></td>
<td>I_{xx} = 2.106</td>
</tr>
<tr>
<td>The bogie 2.6</td>
<td>I_{yy} = 1.424</td>
</tr>
<tr>
<td>The wheel set 1.97</td>
<td>I_{zz} = 2.600</td>
</tr>
<tr>
<td></td>
<td>I_{xx} = 0.623</td>
</tr>
<tr>
<td></td>
<td>I_{yy} = 0.078</td>
</tr>
<tr>
<td></td>
<td>I_{zz} = 0.623</td>
</tr>
</tbody>
</table>

4. The analysis of the simulation results

4.1. The analysis of stability

The stability is one of the most critical factors of the high speed vehicle [12-14]. For high speed vehicle, any small change may make a great impact on quality of vehicle in motion.

The track with a lateral ramp excitation was modeled in software Adams/Rail. The distance between excitation and the starting point is 150 m. And the amplitude and the width of the excitation are respectively 9 mm and 5 m. The comparison between the critical velocity of the high-speed vehicle in normal condition and the condition that vehicle with serious fault anti-hunting damper is shown in Fig.9 and Fig.10.

From the Fig.9, in normal condition of the vehicle, the lateral displacement of wheel set is divergent when the speed is 502 km/h, which means that there is hunting instability in the operation. And the lateral displacement of the wheel set begins to be convergent when the speed reduces to 501 km/h, which means the vehicle return to steady state. Then, the nonlinear critical speed of the vehicle is 501 km/h. And in Fig.10, in the condition that the vehicle runs with extreme fault anti-hunting damper, its nonlinear critical speed that can be obtained in the same way is 395 km/h.

The relation of the gap between the piston of the damper and the oil seal and the critical velocity of the high speed vehicle is shown in Fig.11.

The critical velocity of the vehicle doesn’t have obvious changes when the gap less than 0.1 mm. And the critical velocity begins to decrease gradually when the gap is between 0.1 mm and 0.16 mm. Once the gap exceeds 0.16 mm, the critical of the vehicle begins to decrease sharply. When the gap wears to 0.2 mm, the critical velocity of the vehicle with faulty anti-hunting damper decreases 21.4% compared with
the normal vehicle.

By the results above, a conclusion can be drew is that the fault of oil leakage of anti-hunting damper can seriously affect the stability of vehicle. Especially in the condition that the oil seal wear badly, the nonlinear critical speed of the vehicle will decrease sharply. Then, as for high speed vehicle, the oil leakage of the anti-hunting damper of vehicle cannot be ignored. The running speed of the vehicle should be decreased if there is oil leakage of the anti-hunting damper in the operation. And the faulty damper must be repaired or replaced as soon as possible.

4.2. The analysis of safety

The curve track with 7000 m radius, 150 mm super elevation, and 670 m transition curve was modeled in Adams/Rail software. And the safety performance of the vehicle with different levels fault anti-hunting dampers was analyzed at 300 km/h. The simulation results are shown in Figs. 12-14.

From the Figs. 12-14, the maximum unloading rate decreases, while the maximum derailment coefficient and the maximum lateral axial force increase when the gap increases from 0 mm to 0.2 mm. However, the amplitude of the maximum derailment coefficient and the maximum reduction rate change a little. The maximum of the maximum unloading rate is 0.4019, and the maximum of the maximum derailment coefficient is 0.186 when the gap changes from 0 mm to 0.2 mm. According to The Standard for Whole Vehicle of High Speed EMU the maximum derailment coefficient and the maximum unloading rate are both within the allowable range, which means that the two indexes can be affected by the oil leakage of the anti-hunting damper slightly. The decrease of the maximum unloading rate indicates that the oil leakage of the anti-hunting damper causes the decrease of its damping force and then the hunting motion of the car body becomes easier. But as for the maximum lateral axial force, the maximum of the maximum lateral axial forces is 16860 N. And the increase rate of the maximum compared to the minimum 16175 N is about 4.2 %. Therefore, the conclusion we can draw is that the oil leakage of the anti-hunting damper can cause the decrease of its damping force, which may reduce the inhibiting effect of the hunting motion of car body from anti-hunting damper. And then, the increase of the hunting motion would cause the increase of the lateral axial force, which may lead to the increase of the wear of the matching surface between the wheel and the rail. Further, the increase of the lateral axial force can seriously affect the quality of vehicle in motion.

4.3. The analysis of running performance

The straight track with Germany railway spectrum of low irregularity (GRSLI) was modeled and the dynamic performance index of the high-speed vehicle was analyzed in Adams/Rail software. The lateral acceleration of the car body and the lateral ride comfort index are shown in Fig. 15 and Fig. 16. And the vertical ride comfort index is shown in Fig. 17.
The nonlinear critical velocity will be accelerated with the gap gradually increasing. However, the effect of the oil stiffness to the actual damping force comprehensively. The conclusions we can draw from the analysis of the stability, safety, and the running performance of the high speed vehicle are as follows:

1. If the oil seal of the anti-hunting damper worn excessively, the oil stiffness would decrease. The more seriously the oil seal wear, the more sharply the oil stiffness decrease. And the fastest decrease of the oil stiffness occurs in the no unloading section.

2. It is critical to the quality of vehicle in motion that if the anti-hunting damper can work normally. If the oil seal of the anti-hunting damper worn excessively, the nonlinear critical velocity of the vehicle would decrease. And the reduction of the nonlinear critical velocity will be accelerated with the gap gradually increasing.

3. The oil leakage of the anti-hunting damper has an effect on the maximum derailment coefficient, the maximum unloading rate, and the maximum lateral axial force when the vehicle runs in curve track. However, the effect of the oil leakage on the first two factors is not obvious. But the decrease of damping force of the anti-hunting damper will reduce the inhibiting effect of the hunting motion of car body, which makes the lateral axial force increase sharply with the gap gradually increasing.

4. The fault of the oil leakage of the anti-hunting damper can hardly affect the vertical ride comfort index of the high speed vehicle.

5. Conclusion

The results above show that the oil leakage of the anti-hunting damper can cause obvious increase of the lateral acceleration of the car body and the lateral ride comfort index. However, it can affect the vertical ride comfort index hardly.

Acknowledgement

This research reported was supported by the National Natural Science and Foundation of China (No.11172183; No.11572206).

Reference


