Modelling and Experimental Validation of the Dynamic Damping Characteristics of A High-speed Train Hydraulic Damper

Youquan Fan¹, Shan Zhu²,³, Yongming Wu² and Wenlin Wang³*,

¹Zhuzhou Lince Group Shock Absorber Co., Ltd, Zhuzhou 541004, China.
²School of Electromechanical Engineering, Guangdong University of Technology, Guangzhou 510006, China.
³School of Mechanical Engineering, Dongguan University of Technology, Dongguan 523808, China.

*Corresponding author’s e-mail: pianowwl@163.com

Abstract. Aiming to understand the effects of damper parameter on vehicle system dynamics in the context of railway vehicle system design, a full parametric damper model, which includes the complete valve system dynamics, the dynamic force equilibrium equation, the flow continuity equation, the flow losses model and the comprehensive stiffness model, is established in this work. Both numerical simulation using the established damper model and bench test of dynamic performance of the damper are carried out, the comparison shows that the maximum relative error between simulation and bench test is only 2.6%, in addition, the rates of asymmetry are near and both small. Therefore, the simulation result agrees well with the test result, the established full parameter damper model is validated, the model would be accurate in predicting damper parameter effects in further high-speed train dynamics research and vehicle design.

1. Introduction

Hydraulic dampers [1] are crucial devices for operation stability and ride comfort of modern railway vehicle systems. In the context of railway vehicle system design, understanding the effect of damper parameters on vehicle system dynamics under normal [2] or extreme weather conditions [3] is always the research theme. However, the existing model [1-2, 4-5] for dynamic damping characteristics of hydraulic damper is more in a macro sense level, i.e., the model does not contain concrete physical parameters of the damper, so it is limited in predicting the effect of damper parameters on vehicle system dynamics. In previous research works, Ou [6] and Zhou [7] respectively performed experimental study and theoretical calculation approach on dynamic damping characteristics of the railway hydraulic damper, but the two literatures both used the macro level damper model introduced in the standard [1] or text book [4]; Mellado [8] has built a simple parameter model in predicting the remarkable effect of small displacement on dynamic damping characteristics, Wang [9] and Huang [10] both have built complex parameter damper models in depicting dynamic behaviour of railway hydraulic dampers, but the parameters involved are limited and the research works should be pushed forward. In this work, mathematical modelling of the dynamic damping characteristics in full parameter of a hydraulic damper used in the Chinese CRH 380 high-speed train is conducted, in modelling the complete
valve system, valve parameters are fully considered in predicting the dynamic behaviour of the valve system, which will have dominant effect on the outer performance of the damper. Both numerical simulation and bench test of dynamic performance of the damper are carried out, and the comparison shows that the simulation result agrees well with the test result, therefore, the established full parameter damper model is validated and it can be used further damper parameter sensitivity analysis and design procedure of railway vehicle systems.

2. Mathematical modelling

The cross-section of a hydraulic damper used in the Chinese CRH 380 high-speed train is illustrated in Figure 1, it shows that the fluid will circulate in one direction during vibration input, so the complete valve system embedded in the guide seat is crucial in damping force generating, in addition, the check valves in the piston and the foot valve are also of importance to the dynamic characteristics of the hydraulic damper.

In modelling the full parameter damper model, the dynamic force equilibrium equation, the flow continuity equation, the flow losses model and the comprehensive stiffness model can all be inherited from previous work [9], thus, in this paper, the full parameter model of the complete valve system, as shown in Figure 2(a), will be introduced in detail.

Figure 2(a) shows that under low speed conditions only the constant orifice in the valve 4 works, thus, the pressure-flow characteristics of the complete valve system can be formulated by:

\[ Q_v = \frac{\pi}{4} d_4^2 c_d \sqrt{\frac{2}{\rho}} (p - p_h) \times \left\{ \begin{array}{ll} 0 & \text{if } 0 < |v| < v_1 \end{array} \right. \]

(1)

where \( v, v_1 \) are respectively vibration speed and the first typical speed of the damper, \( Q_v \) is flow discharged by the complete valve system, \( d_4 \) is diameter of the constant orifice in valve 4, \( C_d \) is discharge...
coefficient of the orifice, $\rho$ is oil density, $p$ and $p_b$ are respectively pressure of the inner tube and back pressure in the outer tube.

When the vibration speed increases, valve 1 will be blown-off, so this action makes valve 1 join valve 4 to work, therefore, the pressure-flow characteristics of the complete valve system would be

$$Q_v = \frac{\pi}{4} d_4^2 c_d \sqrt{\frac{2}{\rho}} (p - p_b) + \frac{\pi}{4} d_i^2 c_d \sqrt{\frac{2}{\rho}} \left[p - (1 + E_i) p_i - p_b \right], \quad \text{if } v_1 < |v| < v_2$$

where $v_2$ is the second typical speed of the damper, $d_i$ is diameter of the constant orifice in the seat of valve 1, $p_i$ and $E_i$ are respectively set pressure and set pressure deviation of valve 1.

The valve 2 and 3 are both relief valves, so if the vibration speed continues to increase, valve 2 and 3 would be blown-off sequentially, therefore, the pressure-flow characteristics of the complete valve system would be

$$Q_v = \frac{\pi}{4} d_4^2 c_d \sqrt{\frac{2}{\rho}} (p - p_b) + \frac{\pi}{4} d_i^2 c_d \sqrt{\frac{2}{\rho}} \left[p - (1 + E_i) p_i - p_b \right]$$

$$+ k_3 \delta_3^3 \left[(1 + E_2) p_2 - p_b \right], \quad \text{if } v_2 < |v| < v_3$$

and

$$Q_v = \frac{\pi}{4} d_4^2 c_d \sqrt{\frac{2}{\rho}} (p - p_b) + \frac{\pi}{4} d_i^2 c_d \sqrt{\frac{2}{\rho}} \left[p - (1 + E_i) p_i - p_b \right]$$

$$+ k_2 \delta_2^3 \left[(1 + E_2) p_2 - p_b \right] + k_3 \delta_3^3 \left[(1 + E_3) p_3 - p_b \right], \quad \text{if } v_3 < |v| < v_4$$

respectively. Where $v_3, v_4$ are respectively the third and fourth typical speeds of the damper, $k_2, k_3$ are respectively the spring stiffnesses of valve 2 and valve 3, $\delta_2, \delta_3, p_2$ and $E_2$ are respectively opening height, set pressure and set pressure deviation of valve 2, $\delta_3, \delta_4, p_3$ and $E_3$ are respectively opening height, set pressure and set pressure deviation of valve 3.

The valve 4 is actually a combined valve of a constant orifice $d_4$ and a relief valve, so at extreme high vibration speed situations, the relief valve in valve 4 would be blown-off, and the flow state of the damping valve system can be illustrated by Figure 2(b), the pressure-flow characteristics of the complete valve system would be formulated by:

$$Q_v = \frac{\pi}{4} d_4^2 c_d \sqrt{\frac{2}{\rho}} \left[(1 + E_4) p_4 - p_b \right] + \frac{\pi}{4} d_i^2 c_d \sqrt{\frac{2}{\rho}} \left[p - (1 + E_i) p_i - p_b \right]$$

$$+ k_2 \delta_2^3 \left[(1 + E_2) p_2 - p_b \right] + k_3 \delta_3^3 \left[(1 + E_3) p_3 - p_b \right]$$

$$+ k_4 \delta_4^3 \left[(1 + E_4) p_4 - p_b \right], \quad \text{if } |v| > v_4$$

where $k_4$ is the spring stiffness of valve 4, $\delta_4, p_4$ and $E_4$ are respectively opening height, set pressure and set pressure deviation of valve 4.

Finally, the above complete valve system model is coupled with the dynamic force equilibrium equation, the flow continuity equation, the flow losses model and the comprehensive stiffness model to obtain a full parametric model, the full model depicts the inner dynamic damping characteristics of the hydraulic damper.

### 3. Simulation and experimental validation

Numerical simulation on the dynamic damping characteristics of the hydraulic damper is performed in MATLAB environment, by using the full parametric model established in the last section. Bench test of the dynamic damping characteristics of the hydraulic damper, as shown in Figure 3, is also carried out by using a MTS test stand with hydro-servo excitation system.

Figure 4 compares the simulation result and the test result of the dynamic damping characteristics of
Figure 3. Bench test of the dynamic damping characteristics of the hydraulic damper.

the hydraulic damper. Figure 4(a) shows the dynamic Force-displacement characteristics at typical speed of 0.01 m/s, which is the rated damping characteristics of the damper. Figure 4(a) demonstrates that the simulation result agrees well with the test result.

Figure 4(b), which shows the dynamic Force-displacement characteristics under excitation frequency @ 3 Hz and amplitude @ 1 mm, also demonstrates that the simulation result agrees with the test result in a macro sense, although there exist some discrepancies when the damper changes its directions, the errors are tolerable in engineering.

Figure 4. A comparison of simulation result and test result of the dynamic damping characteristics of the hydraulic damper under conditions (a) excitation frequency @ 2 Hz and amplitude @ 1 mm, (b) excitation frequency @ 3 Hz and amplitude @ 1 mm.

Table 1. A comparison of main damping characteristics indices and its relative errors obtained respectively from simulation and bench test.

| Damping Performance Indices            | Simulation Result | Test Result | Relative Error (%) |
|----------------------------------------|-------------------|-------------|--------------------|
| The maximum extension force (N)        | 10805             | 10923       | 1.09               |
| The maximum compression force (N)      | 10547             | 10712       | 1.56               |
| The absorption work (Nm)               | 23.41             | 24.02       | 2.60               |
| The rate of asymmetry (%)              | 2.45              | 1.97        | --                 |
Table 1 compares the main damping characteristics indices and its relative errors obtained respectively from simulation and bench test. The datum in Table 1 prove that the simulation result agrees well with the test result, the maximum relative error between the both results is only 2.6%, in addition, the rates of asymmetry are near and both small. Therefore, the consistency of simulation result and test result proves that the established dynamic damping characteristics model of the high-speed train hydraulic damper is validated, the model would be accurate in predicting damper parameter effects in further high-speed train dynamics research and vehicle design.

4. Conclusions
(1) The existing model for dynamic damping characteristics of hydraulic damper is more in a macro sense level, the model does not contain concrete physical parameters of the damper, so it is limited in predicting the effect of damper parameters on vehicle system dynamics.
(2) A full parametric damper model, which includes the complete valve system model, the dynamic force equilibrium equation, the flow continuity equation, the flow losses model and the comprehensive stiffness model, is established.
(3) Both numerical simulation and bench test of dynamic performance of the damper are carried out, the comparison shows that the maximum relative error between simulation and bench test is only 2.6%, in addition, the rates of asymmetry are near and both small. Therefore, the simulation result agrees well with the test result, the established full parameter damper model is validated and it would be accurate in further damper parameter sensitivity analysis and design procedure of railway vehicle systems.

References
[1] European Committee for Standardization 2013 EN13802: 2013 Railway applications–Suspension Components–Hydraulic Damper. CEN-CENELEC Management Centre, Brussels, Belgium.
[2] Qin Z, Zhou S X, Sun C L, Chen J X, Long W B, Zhang X J, Wang C G and Luo J L 2017 Influence of hydraulic shock absorber characteristic parameters on the critical speed of high-speed trains. Chinese Journal of Mechanical Engineering, 53(6):138–144. (In Chinese)
[3] Wang W L, Liang Y W, Zhang W H and Iwnicki S 2020 Experimental research into the low-temperature characteristics of a hydraulic damper and the effect on the dynamics of the pantograph of a high-speed train running in extreme cold weather conditions. Proceedings of the Institution of Mechanical Engineers, Part F: Journal of Rail and Rapid Transit, 234(8):896–907.
[4] Yang G Z and Wang F T 2003 Hydraulic Damper of Railway Vehicle. China Railway Press, Beijing, China. (In Chinese)
[5] Dixon J C 2007 The Shock Absorber Handbook (2nd Edition). Professional Engineering Publishing Ltd. and John Wiley & Sons, Ltd., UK.
[6] Ou H B, Wang Y and Huang C 2016 Experimental study on dynamic characteristics of yaw damper. Mechanical Engineering and Automation, 198(5):51–52,55. (In Chinese)
[7] Zhou X Z, Chi M R, Gao H X, Yang D X and Qin J S 2018 Research on calculation method of hydraulic damper dynamic characteristics. Electric Drive For Locomotives, 4:88–91. (In Chinese)
[8] Mellado A C, Gomez E and Viñolas J 2006 Advances on railway yaw damper characterisation exposed to small displacements. International Journal of Heavy Vehicle Systems, 13(4):263–280.
[9] Wang W L, Huang Y, Yang, X J and Xu G X 2011 Nonlinear parametric modeling of a high-speed rail hydraulic yaw damper with series clearance and stiffness. Nonlinear Dynamics, 65(1-2):13–34.
[10] Huang C H and Zeng J 2018 Dynamic behavior of a high-speed train hydraulic yaw damper. Vehicle System Dynamics, 56(12):1922–1944.

Acknowledgements
The authors thank financial support from the National Natural Science Foundation of China (NSFC) under Grant No. 11572123 and the Research Fund for High-level Talent of Dongguan University of Technology under Project No. KCYXM2017026.