J. Mod. Transport. (2016) 24(3):159–165
DOI 10.1007/s40534-016-0116-4

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A new lever-type variable friction damper for freight bogies used in heavy haul railway

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Received: 26 October 2015 / Revised: 12 July 2016 / Accepted: 14 July 2016 / Published online: 30 July 2016

Abstract To overcome defects caused by the complex structure and unstable damping performance of the wedge-type damper, a new lever-type friction damper has been developed for use in freight bogies; the design allows the advantages of traditional three-piece bogies to be retained. A detailed description of the structure and mechanism of the lever-type damper is provided, followed by a stress analysis using the finite element method. Dynamic performance characteristics of the lever-type damper and the wedge-type damper are compared in terms of the nonlinear critical speed, riding index, and curve negotiation. The results indicate that the maximum stress of the lever remains below its yield limit. The lever-type car has higher running performance reliability, and achieves similar nonlinear critical speed, riding index, and curve negotiation when compared with the wedge-type car.

Keywords Lever-type damper · Wedge-type damper · Dynamic performance

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1 Introduction

With the rapid development of Chinese heavy haul railway, it is increasingly necessary to improve transportation efficiency. Like many developed countries that use heavy haul technology, China works to improve its railway transportation efficiency by increasing axle load, decreasing the empty weight, raising running speed, and adding more vehicles to each train. Three-piece bogies are normally used in heavy haul transportation because they are cheap and robust. The side-frame cross-bracing bogie [1–5] and sub-frame radial bogie (e.g. bogie K7) [5–11] are two kinds of three-piece bogies widely used in China, both of which adopt friction wedge dampers; however, the sub-frame radial bogie has better dynamic performance [5]. In the wedge-type bogies, the structure at the two ends of the bolster features four small holes for holding the wedges, which complicates the design, increasing the costs of the damper and making it difficult to manufacture and maintain [12]. According to statistics from 2001 [13], of the 8,120 bolsters that were overhauled in Zhanjiang Depot, 2,368 bolsters were flawed in 2,815 faulty bolsters, and the bolster with flaws at the end structure is 2,137, accounting for 75.9 % of all flawed bolsters. This indicated that the use of a wedge-type damper reduces the reliability of the bolster.

Vibrational energy is dissipated through friction produced between the wedges, bolster, and side frames [14, 15]. The damping force and anti-warp performance are directly related to the support force provided by the secondary suspension and the degree to which the wedges are worn. After wear, the wedges would move upwards, resulting in a decrease in the support force provided by the wedges, and worsening their surface conditions. Consequently, the damping force and anti-warp performance change significantly, which ultimately affects the reliability of the running performance of the cars [14–17]. It is therefore necessary to design a new damper with more reliable performance and a simpler structure.

This article introduces a new lever-type damper with a simple structure for use in three-piece freight bogies. First, the structure and mechanism of the lever-type damper are introduced in detail. Then, the static stress of the lever is modelled using finite element method (FEM). Finally, the dynamic performance of a lever-type car is analysed and compared with a wedge-type car to verify its feasibility.

2 Mechanism of the new lever-type variable friction damper

The schematic diagram of the damper is shown in Fig. 1a. The coil springs of the secondary suspension are divided into the side-frame end coil springs and the lever end coil springs. The side-frame end coil springs are located on the side-frame spring seat, and the lever end coil springs are located directly on the lever, generating vertical forces F_{v1} and F_{v2}. The pivot of the lever in the side frame is the centre of rotation; the distance between the two rows of lever end coil springs and the pivot is L_1 and L_2, respectively. The vertical forces F_{v1} and F_{v2} are transmitted to the upper end of the lever through the damper mechanism to form the reaction force F_N; the vertical distance from the pivot is L_3. There is also friction at the pivot that generates a moment M_p. The friction coefficient is μ, and the damping force induced by the longitudinal force F_N is F_μ, as shown in Fig. 1b. A friction pair is formed between the upper end of the lever and the bolster. Because of the horizontal force generated by the lever acting on the non-lever side of the bolster, another friction pair is generated between the bolster and the side frame. Hence, there are two friction pairs on each end of the bolster. The lever-type damper can absorb vibration in both the vertical direction and the horizontal direction.

The damping force and anti-warp performance of lever-type cars are related to the support force of the secondary suspension and the surface conditions of the levers. The support force changes little after lever wear. The only factor that would affect the damping force and anti-warp performance is the worn surface condition. The lever-type damper can thus ensure the running stability of lever-type cars.

When the lever rotates around the pivot, line-surface contact can occur between the two friction plates which results in a sharp reduction of damping force; a rotatable friction plate is thus set at the top of the lever. The plate ensures that the lever mechanism maintains face-to-face contact after rotation. The working principle of the rotatable friction plate is shown in Fig. 1b.

According to the moment balance theory and the Coulomb-friction law, Eqs. (1) and (2) are derived as follows:

$$F_{v1} \times L_1 + F_{v2} \times L_2 = F_N \times L_3 \pm M_p, \tag{1}$$

$$F_\mu = F_N \times \mu. \tag{2}$$

Damping force is usually described by the relative friction coefficient φ, which is defined as the ratio of the frictional force F to the vertical force P of the suspension system:

$$\varphi = \frac{F}{P} = \frac{2F_\mu}{P}, \tag{3}$$

where P is the sum of the vertical spring force on each end of the bolster, including the side-frame end coil springs and the lever end coil springs. F_{v1} and F_{v2} can be expressed as follows:

$$F_{v1} = F_{v2} = \frac{1}{3}P. \tag{4}$$

Considering that M_p is small, by combining Eqs. (1)–(4), we obtain

$$\frac{L_1 + L_2}{L_3} \approx \frac{3\varphi}{2\mu}. \tag{5}$$

By defining the length of L_1, L_2, and L_3, the relative friction coefficient can be determined to define the

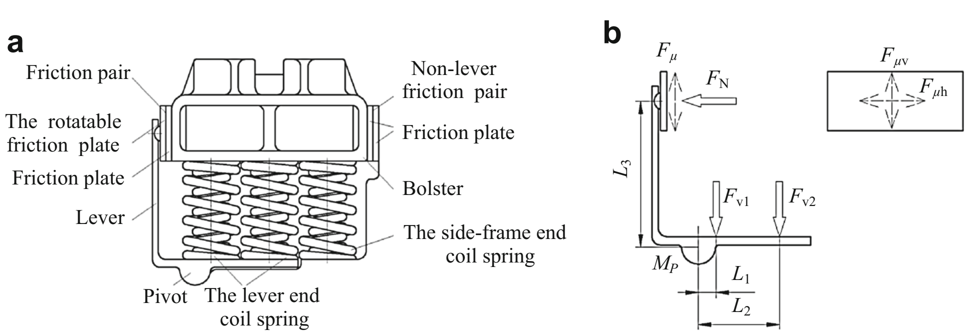

Fig. 1 Principles of a lever-type variable friction damper. **a** Schematic diagram of lever-type damper. **b** Force analysis of lever-type damper

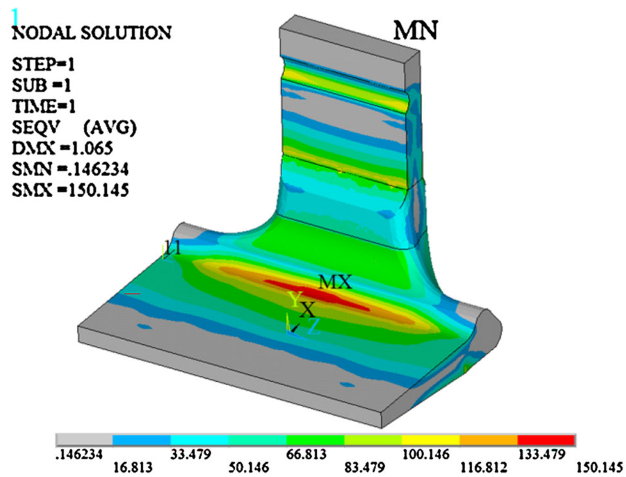

Fig. 2 Stress nephogram of the lever-type damper

damping capacity of the lever-type damper to be the same as the wedge-type damper.

3 Feasibility analysis of the lever-type damper

After assembly of the bogie, the upper end of the lever-type damper fits closely with one side of the bolster. In application, the lever-type damper rotates by a slight angle around the pivot because of irregularities in the track, and thus only requires a small space for installation. In practice, cracks in the side frame mainly occur on the top of the side frame pedestal and the brake chutes [18]. The lever-type damper is located on the side-frame spring seat, which, whilst it would slightly decrease the static strength of the spring seat, would not significantly weaken the fatigue strength of the side frame.

A lever-type damper has lower strength and stiffness than a wedge-type damper, but these characteristics can be greatly improved by rational structural design. To check the strength of the lever, we adopted Grade B + steel as its material and took bogie K7 with a 25 t axle load as an example. According to TB/T 1335-1996, the maximum vertical force on the spring site is 1.5C, where C is the axle load:

$$C = (G - T)g, \tag{6}$$

G is the axle load of 25 t, T is the 1.2 weight of a wheelset, and g is the gravitational acceleration.

Using Eq. (6), C can be obtained as follows:

$$C = 233,478 \text{ N} \tag{7}$$

The load on the lever was two-thirds of the load on the whole spring seat, so the maximum vertical force on the lever was C. The finite element model of the lever was produced using ANSYS; the maximum Von Mises stress was 150.15 MPa, which is less than the yield stress of 340 MPa and the allowable stress of 151 MPa on the bolster and side frame using Grade B + steel. The maximum vertical displacement of the end of the spring seat was 1.46 mm. The stress nephogram is shown in Fig. 2. The maximum stress reduced and the stress distribution of the lever increased even after the optimization.

4 The dynamic performance of lever-type and wedge-type damper

In theory, a lever-type damper can be applied to any traditional three-piece bogie. The C_{80C} car equipped with the three-piece bogie K7, has been in operation for many years and has excellent dynamic performance. The performance of an empty car is usually worse than that of a heavy car; therefore, the empty C_{80C} was taken to be the subject of our comparison. Dynamic models equipped with two types of dampers differ only in the structure of the damper; the suspension parameters and inertial parameters are the same. A key performance of the damper is its damping capacity, namely the relative friction coefficient φ, which can be determined by defining the lengths L_1, L_2, and L_3 to ensure the same damping capacity for the two types of damper.

Table 1 DOFs of the dynamic model

Component	DOFs					
	Longitudinal	Lateral	Vertical	Rolling	Yawing	Pitching
Car body	X_C	Y_C	Z_C	θ_C	β_C	φ_C
Bolster (i = 1, 2)	–	–	–	–	β_{Bi}	–
Side frame (i = 1,2,3,4)	X_{Fi}	Y_{Fi}	Z_{Fi}	Z_{Fi}	β_{Fi}	φ_{Fi}
Deputy frame (i = 1,2,…,8)	–	–	–	–	–	φ_{Di}
Wheelset (i = 1,2,3,4)	X_{Wi}	Y_{Wi}	Z_{Wi}*	Z_{Wi}*	β_{Wi}	φ_{Wi}
Lever (i = 1,2,3,4)	–	–	–	–	–	φ_{Li}

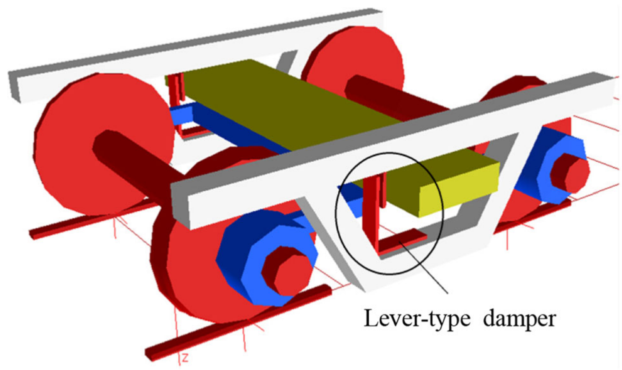

Fig. 3 Dynamic model equipped with lever-type damper

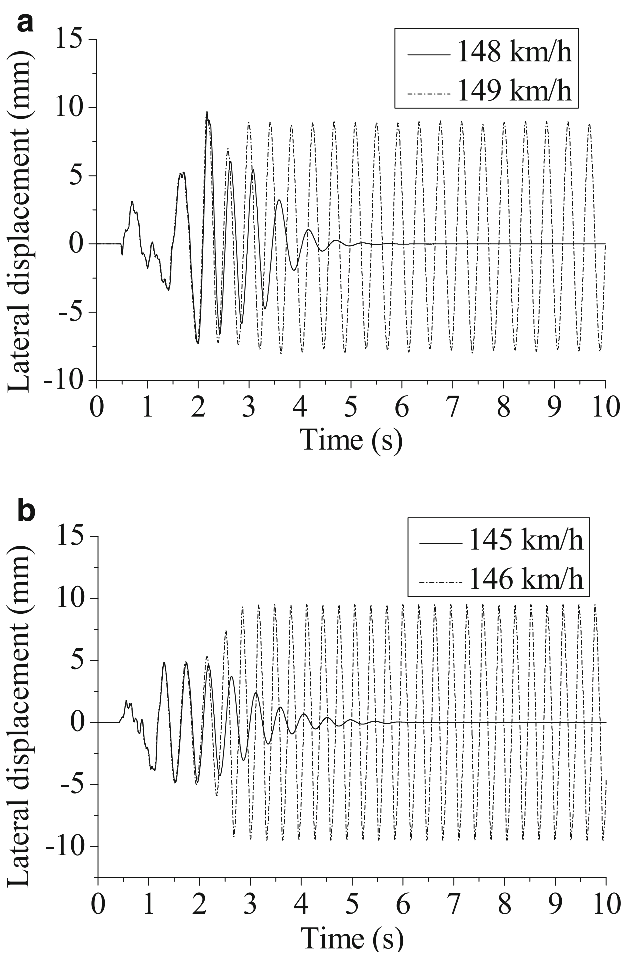

Fig. 4 Results of nonlinear critical speed of the two types of dampers. **a** The wedge-type damper. **b** The lever-type damper

The dynamic models were separately modelled using Simpack software, and both adopted the 60 kg/m rail. The degrees of freedom (DOFs) of the two dynamic models and nonlinear characteristics such as gaps, stops, friction forces, and wheel-rail contact geometry were all taken into consideration. The DOFs of the dynamic models are shown in Table 1, the model equipped with the lever-type damper is shown in Fig. 3.

All components were considered to be rigid. The car body, side frame, and wheelset all had six DOFs, the bolster only had yawing, and the deputy frame and lever both only had pitching. Because of the different structures of the two types of dampers, the number of rigid components was different; thus there were 60 DOFs for the dynamic model equipped with the lever-type damper, and 52 DOFs for the model equipped with the wedge-type damper. The vertical motion and rolling of the wheelset is dependent on the lateral motion and yawing. Thus, the vertical motion and rolling were independent DOFs, denoted with an asterisk (*).

The dynamic performance indexes of the two types of dampers equipped in C_{80C} were separately analysed and compared in detail, based on straight line travel and curve negotiation.

4.1 The nonlinear critical speed

The anti-warp performance of a bogie equipped with a wedge-type damper was poor and would worsen after wedge wear, which would result in a poor riding index and damping force reliability [13]. The normal force of the friction pair of the wedge-type bogie is equal to that of the lever-type bogie with the same φ, where the width of the upper end of the lever is the same as the wedge. Therefore, the nonlinear critical speed of the lever-type car should be similar to that of the wedge-type car.

The results of the two models are shown in Fig. 4, where the fifth grade track irregularity power spectral density of U.S. railways is considered, indicating that the nonlinear critical speed of the wedge-type car was 148 km/h, and 145 km/h for the lever-type car. The data show that the lever-type damper can reach a nonlinear critical speed similar to that of the lever-type damper.

4.2 The riding index

As lever-type and wedge-type cars have the same relative friction coefficient φ, the same suspension parameters and the same vehicle weight, the dynamic models equipped with the two types of dampers should theoretically have similar riding indexes.

Figure 5 shows the riding indexes W_y and W_z, and the maximum acceleration $A_{y\max}$ and $A_{z\max}$ of the two models with a change in velocity. As seen in Fig. 5, all four evaluation indexes were almost the same and met the requirements of GB/T 5599-1985.

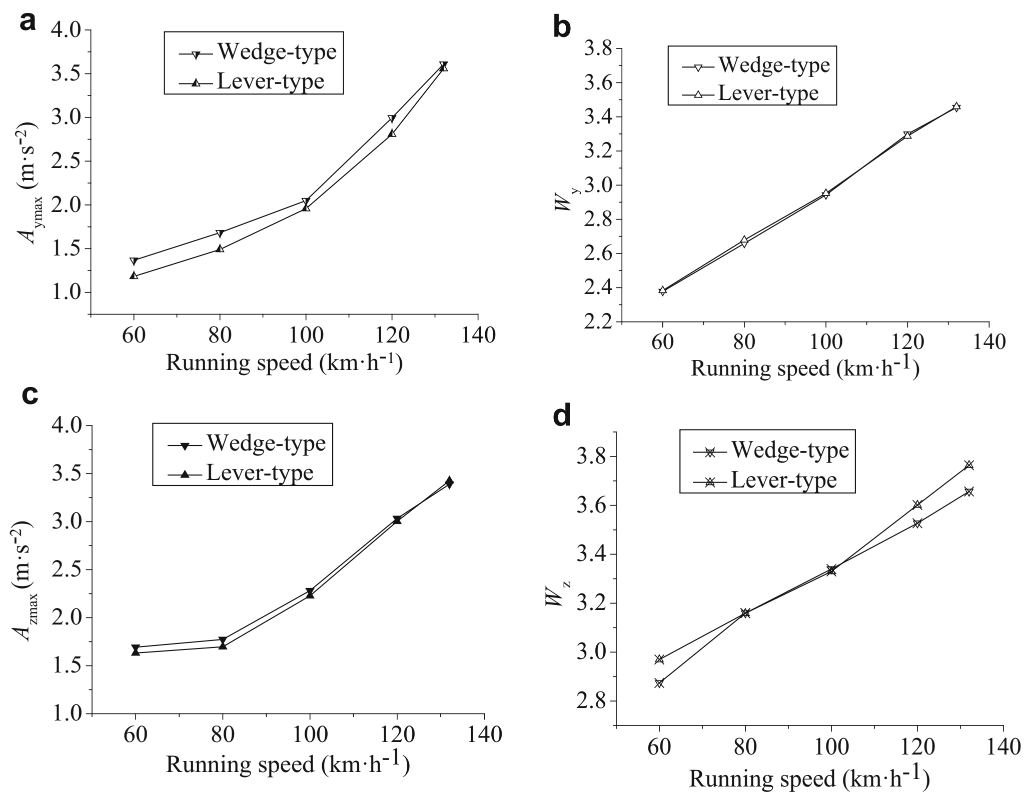

Fig. 5 Comparison of riding indexes between wedge-type and lever-type vehicles. **a** The maximum lateral acceleration. **b** The lateral riding index. **c** The maximum vertical acceleration. **d** The vertical riding index

Table 2 Curve conditions

Case no.	L_t (m)	R (m)	h (mm)	V (km/h)
1	70	300	100	65.7
2	70	400	100	75.9
3	70	600	80	87.3
4	50	800	60	93.9
5	50	1,200	50	110.5
6	35	1,600	35	119.3

4.3 Curve negotiation

To compare the curve negotiation of the dynamic models equipped with the two types of dampers with a change of curve radius as the vehicle passes through a smooth curve, the maximum deficient superelevation of all different curve radiuses was 70 mm. The running speed V is given by

$$V = \sqrt{\frac{R(h + h_d)}{11.8}}, \tag{8}$$

where R is the curve radius, h is the superelevation, and h_d is the deficient superelevation.

According to Eq. (8), the detailed calculation conditions are shown in Table 2, where L_t is the length of transition curve.

Figure 6 shows the results of these curve conditions. In Fig. 6, we can see that the derailment coefficient, rate of wheel load reduction, lateral wheel-rail force, and lateral wheelset-rail force for the two types of dampers were almost the same as the vehicle passed through a smooth curve with a change of curve radius, all of which decreased with the increase of curve radius. The capacity for curve negotiation is directly related to the longitudinal stiffness of the primary suspension. The suspension parameters of the two types of dynamic models were the same. The capacity for curve negotiation of the vehicles equipped with the two types of dampers was thus almost the same. The simulation result shows that the derailment coefficient for all cases was less than 1.0; the rate of wheel load reduction was less than 0.6. The lateral wheel-rail force and the lateral wheelset-rail force both met the requirements of GB 5599-1985

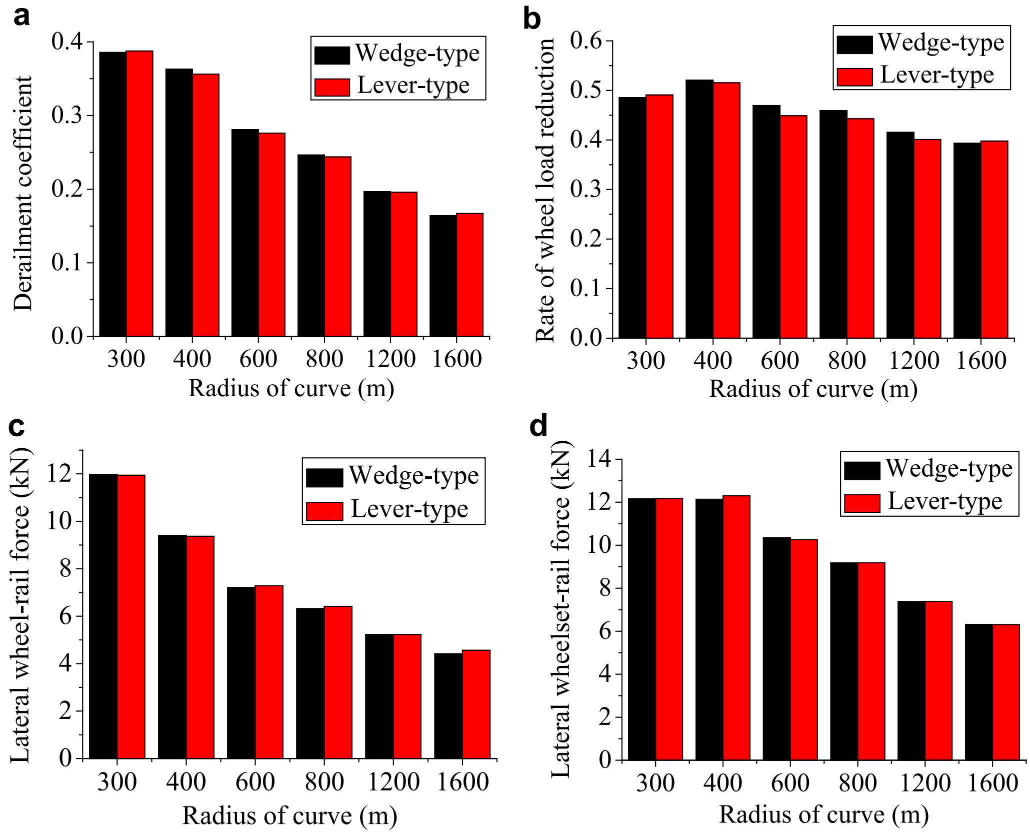

Fig. 6 Curve negotiation for the two types of dampers. **a** Derailment coefficient. **b** Rate of wheel load reduction. **c** Lateral wheel-rail force. **d** Lateral wheelset-rail force

5 Conclusion

Motivated by the development of Chinese freight railway, we proposed a new lever-type damper. Through theory analysis, strength analysis of the lever, and the comparison of the dynamic performance of vehicles equipped with the two types of dampers, conclusions are as follows:

(1) Compared with a traditional wedge-type damper, the lever-type damper can significantly simplify the structure of the bolster, and improve its reliability.

(2) The steady running performance of the lever-type car is less sensitive to damper wear than the wedge-type car.

(3) The lever-type car can achieve a nonlinear critical speed similar to the wedge-type car because of the similar anti-warp performance.

(4) The lever-type car has riding index and capacity for curve negotiation similar to the wedge-type car because of similar relative friction coefficients and suspension parameters.

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