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Influence of Curve Geometric Parameters on Curving Performance of Sub-frame Radial Bogie

Chunlei YANG[1*], Yunhua HUANG[2], Junjun DING [2]

1. School of Information Engineering, Hubei Minzu University, Enshi 445000, Hubei, China;
2. School of Mechanical Engineering, Southwest Jiaotong University, Chengdu 610031, Sichuan, China

***Corresponding Author:** Chunlei YANG, School of Information Engineering, Hubei Minzu University, Enshi 445000, Hubei, China 2015017@hbmy.edu.cn*

Abstract:
Based on the theory of vehicle-track coupling dynamics, the coupling dynamic model of the freight car mounted with the sub-frame bogies and the numerical model of curved track were established, utilizing the fast numerical integration method, the curving performance of the vehicle was simulated to study the influence of the curve geometric parameters such as curve radius, transition curve length and superelevation of outer rail on the wheel-rail dynamic interaction. The simulation results indicate that: (1) Increasing the curve radius can decrease the wheel-rail wear, but the effect will weaken while the curve radius is greater than 800m. (2) If the transition curve length is less than 30m, vibrations will appear at the transition-circle connecting point, and the smaller the transition length, the bigger the vibrations, the worse the wheel-rail wear, but when the length is bigger than 50m, its further variation has very little effect on wheel-rail wear. (3) The superelevation of outer rail can affect the distribution and difference of the inner and outer wheel-rail forces, and too large deficient or excessive superelevation will worsen the wheel-rail wear either. However, an appropriate deficient superelevation of outer rail (e.g. <20mm) is helpful to reduce the wheel-rail wear, which is consistent with the engineering practice of settling a certain deficient superelevation value.

Keywords: *heavy-haul freight car; sub-frame radial bogie; curve geometric parameters; wheel-rail wear*

Heavy-haul transportation is an effective way to enhance the railway transport capacity, but the problem of wheel-rail wear and rail fatigue defects caused by heavy load is a common engineering challenge for all operating heavy-haul railway transport countries, and the wheel-rail wear of curve segment is particularly serious [1]. From the actual operation results of Chinese Da-Qin and Shuo-Huang coal transport lines, with the increase of axle load, the side abrasion, corrugation and peeling and dropping of rail occurred often, and the damage of wheel and rail was significantly increased [2-3]. Predictably that the wheel-rail wear will be more serious with the increase of axle load to 27-30t in China.

Aiming at the wheel-rail abrasion, many scholars and engineers have conducted a lot of researches and experimental explorations, and made some certain achievements [4-7]. However, most of researchers focused on the high-speed railway and urban rail system in China [8-12], paid less attention to the heavy-haul railway, while which had more serious wheel-rail abrasion problems. Besides, most of the dynamic researches were usually simulated by foreign commercial software such as SIMPACK, NUCARS or UM etc [13-19].

The wheel-rail abrasion is caused by many factors, such as wheel-rail matching profile, wheel-rail contact state, parameters of vehicle and track etc., it is the result of the comprehensive interaction between the vehicle and track, so it is necessary to analyze from the perspective of overall vehicle-track coupled system, while it has little effects by analyzing the performance of vehicle or track separately. Therefore, based on the actual structures of 27t axle load freight wagon and heavy-haul track in China, the vehicle-track coupled dynamic model and the mathematical model of curved track were established, and applying the Hertz nonlinear contact theory and Kalker linear wheel-rail creep theory [20], adopting the fast numerical integration method [21], the curving performance of the heavy load vehicle was simulated to mainly explore the influence law of geometric parameters such as curve radius, transition curve length and superelevation of outer rail on wheel-rail wear, to provide certain technical support for the design and type selection of the heavy-haul railway vehicle and track in

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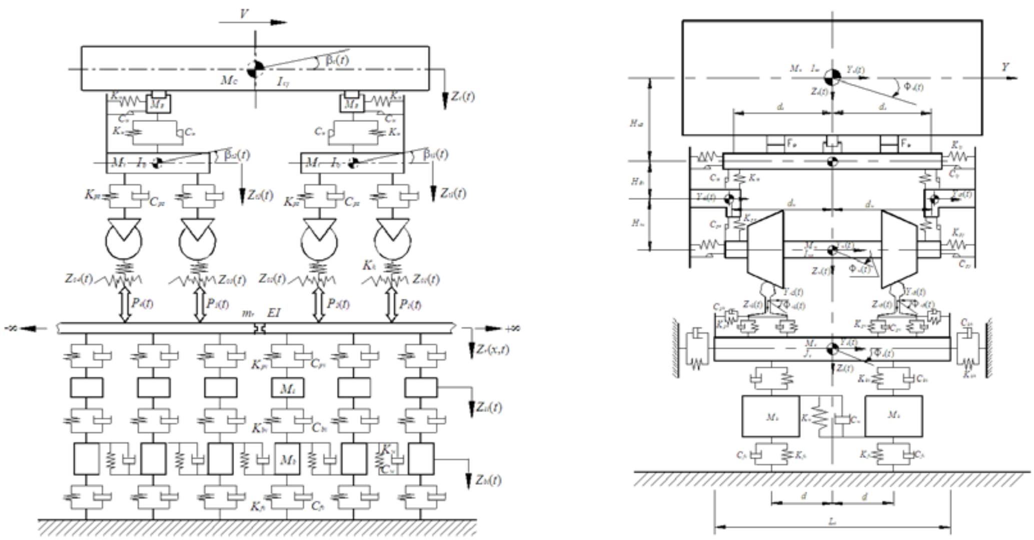
China.

1 Simulation model

1.1 Heavy-haul vehicle-track coupled dynamic model

$C80_F$ type wagon is a 27t axle load gondola developed in China, mounted with two sub-frame radial bogies. The front and the rear wheelset of the bogie are connected with a u-shaped sub-frame and an elastic crossing bar. Its primary suspension adopts a rubber-metal pad, and the second suspension consists of several sets of two-stage stiffness springs and a combined friction wedge [22]. Considering the specific structural characteristics of heavy-haul freight car and its nonlinear links, the vehicle-track coupled dynamic model is established and shown in Fig. 1.The model is mainly composed of the car body, bolsters, side frames and wheelsets. Between the car body and the bolsters, the rotary moments of the central plates and the friction moments of the elastic side bearings are considered. The whole vehicle suspension system consists of the primary and the secondary suspension system. The primary suspension is mainly the three-direction stiffness and damping provided by the rubber-metal pad, the secondary suspension is the three-direction stiffness provided by the two-stage stiffness springs and the nonlinear friction damping characteristics of the combined wedge, the connecting crossed rod is simulated with the longitudinal and lateral stiffness. The track is simulated as a three-layer continuous elastic discrete points supporting infinite long Euler beam model. Considering its vertical, transverse and torsional degrees of freedom of the rail, the sleeper is regarded as a rigid body, considering its vertical, lateral and rotation movement, the ballast is discretized into rigid mass blocks, connected by the shear stiffness and shear damping elements, and the ballast bed and roadbed are connected by linear springs and damping elements, and only consider the vertical vibration of the ballast bed. The whole degree of freedom of the vehicle-track coupled model is 56 (shown in Table 1). The detailed descriptions of symbols in Fig. 1 and Table 1 can be found in the reference [23].



(a)Front view (b) End view

Fig.1 Heavy-haul vehicle-track coupled dynamic model

Table 1 DOFs of heavy-haul vehicle-track coupled dynamic model

DOFs	Longitudinal	Lateral	Vertical	Rolling	Yawing	Pitching
Car body	Xc	Y_c	Z_c	φ_c	ψ_c	β_c
Bolster(i=1,2)	X_{Bi}	Y_{Bi}	Z_{Bi}	φ_{Bi}	ψ_{Bi}	—
Side-frame(i=1,2)	$X_{t(L,R)i}$	$Y_{t(L,R)i}$	$Z_{t(L,R)i}$	—	$\psi_{t(L,R)i}$	$\beta_{t(L,R)i}$
Wheel set(i=1~4)	—	Y_{wi}	Z_{wi}	φ_{wi}	ψ_{wi}	β_{wi}
Rail	—	$Y_{r(L,R)}$	$Z_{r(L,R)}$	$\varphi_{r(L,R)}$	—	—
Sleeper	—	Y_s	Z_s	φ_s	—	—
Ballast	—	—	$Z_{b(L,R)}$	—	—	—

1.2 Mathematical model of heavy-haul curved track

The typical curved track plane is shown in Fig. 2 [25], generally is composed of straight line--transition curve--circular curve--transition curve--straight line. When the vehicle enters the curve with a radius of R_0 (shown in fig. 2) from a straight line, the curvature changes from zero to k_0 ($k_0=1/R_0$), and an outward inertial centrifugal force will appear. To reduce or balance this centrifugal force, a certain superelevation (h_0) of the outer rail is needed. In order to maintain the smoothness of vehicle curve negotiating and alleviate the impact caused by the superelevation of the outer rail and the sudden change of the curvature, between the straight line and the circular curve, a space curve (called transition curve) with continuous changes of curvature and superelevation is usually provided. The transition curve is generally divided into the cubic and the high order spiral line, while Chinese railway usually adopts the cubic type as the transition curve [16].

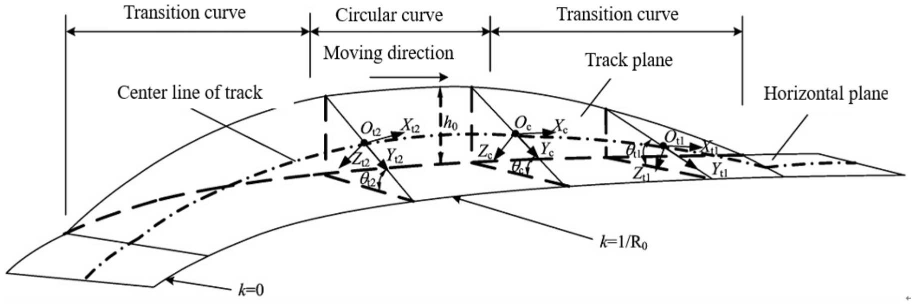


Fig.2 Schematic diagram of curved track

The linear slope requires both the superelevation (h) of outer rail and the curvature (k) of transition curve should change linearly with the length of transition curve, means that:

$$\mathrm{h} = \frac{l}{l_s} h_0 \tag{1}$$

$$\mathrm{k} = \frac{l}{l_s} k_0 = \frac{l}{l_s R_0} \tag{2}$$

Where l is the vehicle running distance on transition curve, l_s is the total length of transition curve, h_0, R_0 and k_0 are the superelevation, radius and curvature of the circular curve. Then, the plane equation of the cubic spiral line can be derived as:

$$\begin{cases} x = l - \dfrac{l^5}{40R^2{l_s}^2} \\ y = \dfrac{l^3}{6Rl_s} - \dfrac{l^7}{336R^3{l_s}^3} \end{cases} \tag{3}$$

At the junction point of transition and circular curve:

$$\begin{cases} x_0 = l_s - \dfrac{{l_s}^3}{40{R_0}^2} \\ y_0 = \dfrac{{l_s}^2}{6R_0} \end{cases} \tag{4}$$

Where R is the radius of transition curve.

From the force analysis of the vehicle passing through a curve, the formula of the superelevation of outer rail is given by:

$$h = \frac{s}{g} \frac{v^2}{R} \tag{5}$$

Where s is the gauge of the track, g is the constant acceleration of gravity. For Chinese 1435mm standard gauge, while the units of speed v, superelevation h and radius R are respectively km/h, mm and m, then the formula can be simplified as:

$$h \approx 11.8 \frac{v^2}{R} \tag{6}$$

If the actual superelevation is greater or less than this calculated value, then it is considered that the vehicle runs at an excessive or inadequate superelevation. *Chinese railway works regulations* rules that the maximum designed superelevation is 150mm.

Due to the superelevation of outer rail and the curve curvature change, the track plane will be distorted, to cause the additional longitudinal, lateral, vertical, rolling, yawing and pitching movement of the rigid body parts of vehicle, lead to relative displacement and relative velocity happen at each suspension point, thus to affect the suspension forces [24-25].The specific calculation method of relative displacement and relative velocity caused by the variations of superelevation and curvature is detailed in the ref. [24].

2 Simulation conditions and calculation method

Based on the specific structure and actual parameters of 27 t axle load freight wagon mounted with two sub-frame radial bogies, the simulation parameters are chosen according to its actual parameters, e.g. the wheel profile is LM wear type tread, and the diameter of wheel is 915mm. The track parameters are obtained using the parameters provided in appendix 4 of ref. [20]. Curve parameters are chosen according to the first level mix railway of passenger and cargo transport in China, and some requirements for Chinese new heavy-haul railway line in *the design specifications for heavy-haul railway* (TB 10625-2017). In order to clearly compare the influence of curve geometric parameters on wheel-rail wear, the track geometric irregularities excitation is not considered in the calculation. The detailed simulation conditions is shown in Table 2.

Table 2 Simulation conditions of the curve line

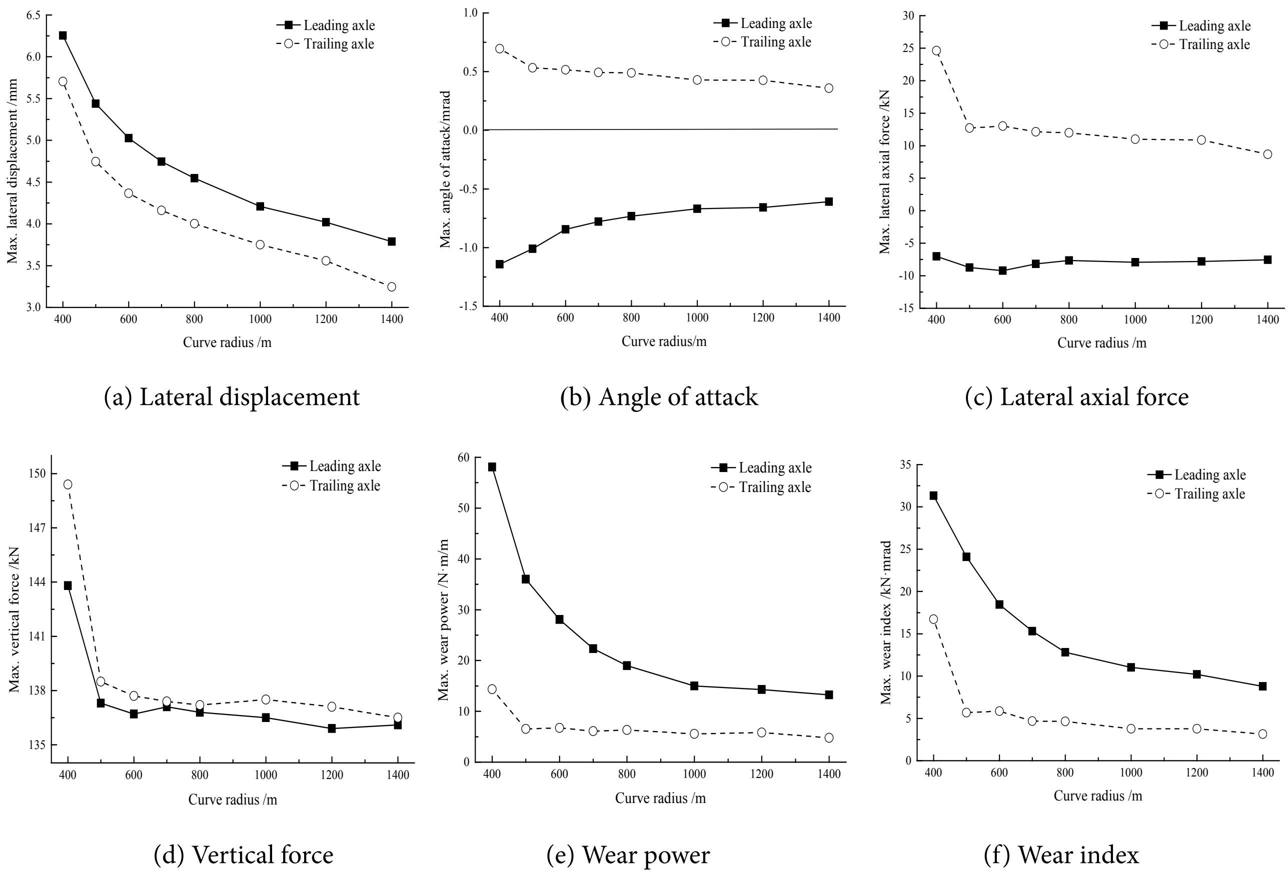
Simulation conditions	Curve radius (m)	Superelevation of outer rail (mm)	Length of transition curve (m)	Length of circular curve (m)	Running speed (km • h^{-1})
1	350~1400	$h = 11.8v^2/R$	75	50	
2	800	95	20-150	50	80
3	800	5-150	75	50	

Due to a large number of nonlinear factors and degrees of freedom in the vehicle-track coupled model, it is difficult to solve analytical solutions directly. Therefore, using the numerical integration method is a common effective way. Based on the vehicle-track coupled dynamic model and curve mathematical model, the forces on the moving parts of the system were analyzed and the motion equations of the parts were established. The wheel-rail normal forces are solved by Hertz nonlinear contact theory, the wheel-rail creep forces/torques are solved by Kalker linear creep theory, and revised by "Shen's theory" [20], while the suspension forces are calculated by the system coordinate transforming method [24]. The wheel-rail wear can be compared and analyzed with the common evaluation indexes such as the wheel-rail vertical force, wheel-rail lateral force and wheel-rail wear power (defined as the product of wheel-rail creep force and wheel-rail creepage) [20].

3 Simulation results

3.1 Effects by the curve radius

Fig.3 shows the comparative variations of some dynamics indicators between the leading and trailing axle of the 27t axle load sub-frame radial bogie as the curve radius increases. As shown in Fig.3 (a), the lateral displacement of wheelset decreases as the curve radius increases, and that of the leading axle is slightly greater than that of the trailing one(< 1mm). The directions of angle of attack of the leading and trailing axle are opposite, the amplitudes of them become smaller especially within the curve radius of 600m as the curve radius increases, but after the curve radius is greater than 600m, they change very slightly (Fig. 3(b)). The lateral axial force and vertical force decrease as the curve radius increases, especially change very fast within the curve radius of 500m. After that, the variations and magnitudes of the lateral and vertical force of the both axles have only slightly differences, but the values of the trailing axle are corresponding greater than that of the leading axle at the whole range (Fig.3 (c) and (d)). Fig.3 (e) and (f) indicate that the wheel/rail wear decreases as the curve radius increases, and the magnitude of decline is more significant on more severe curves. However, after the curve radius is greater than 1,000 m for the leading axle, 500m for the trailing axle, the changes slow down obviously. Comparing the two wear indexes between the leading and the trailing axle of the bogie, both of the leading axle are always greater than that of the trailing one, which means that the leading wheelset will wear faster and have relatively shorter service life.



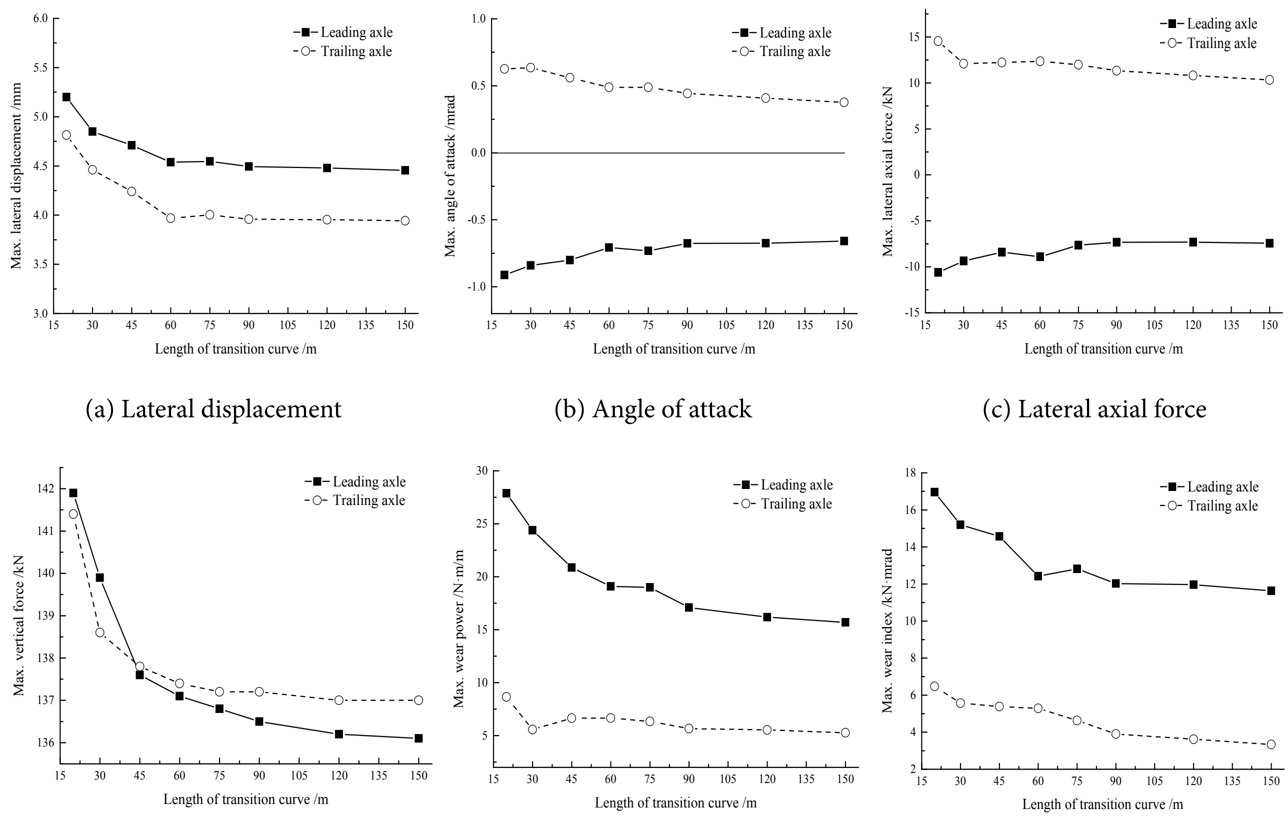
(a) Lateral displacement (b) Angle of attack (c) Lateral axial force

(d) Vertical force (e) Wear power (f) Wear index

Fig.3 Effects of curve radii to wheel-rail dynamic interaction

3.2 Effects by the transition curve length

The influence of length of transition curve on the dynamic interaction of the bogie is shown in Fig.4. The lateral displacements and the angles of attack of both the leading and the trailing wheelset decrease as the length of transition curve increases within 60m, but they change little after the length is greater than 60m (Fig.4 (a) and (b)). The lateral axial forces of both wheelset decrease slowly with the increase of the length of transition curve. But after the length of transition curve is greater than 90m, the lateral axial force of the leading axle change little, and that of the trailing has very slightly variations when the length is longer than 30m (Fig.4 (c)). The length of transition curve has much effects on the vertical force: as seen from Fig. 4 (d), the vertical forces of the leading and the trailing wheelset decrease rapidly with the increase of length of transition curve especially at the range of 45m and 30m separately, but they decrease slowly after the length is greater than 90m and 60m accordingly. The wear power and the wear index of both wheelsets continuously decrease with the increase of the length of transition curve, and they decrease relatively faster within the transition curve length of 90m. Between the two wheelsets, the wheel/rail wear of the leading wheelset is obviously larger than that of the trailing, indicates that the leading wheelset of the bogie is easier to be worn (Fig.4 (e) and (f)). From the above, it can be concluded that properly increasing the transition curve length within a certain range is useful to reduce the wheel/rail wear. To this sub-frame radial bogie, generally the minimal length of transition curve should be 90m, and preferably not less than 60m even in a difficult engineering practice.



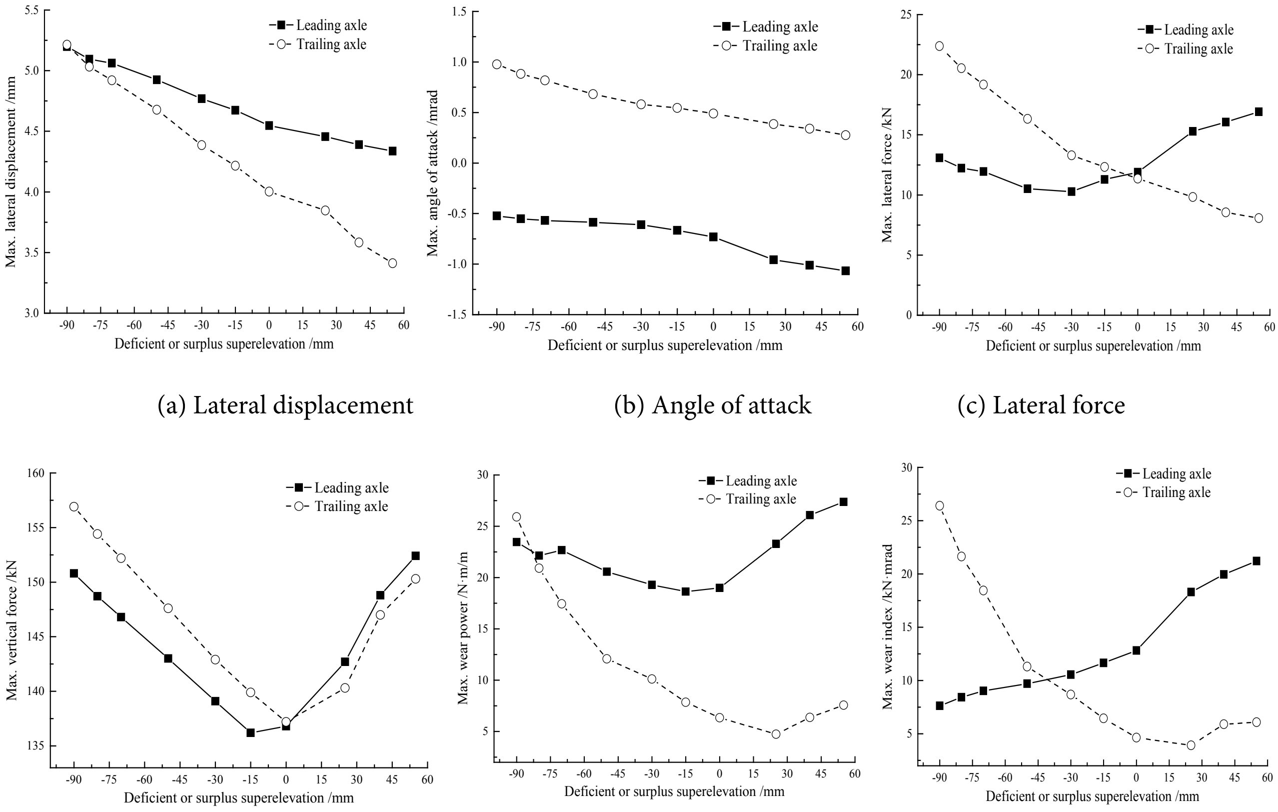
(a) Lateral displacement (b) Angle of attack (c) Lateral axial force

(d) Vertical force (e) Wear power (f) Wear index

Fig.4 Effects of transition curve length to wheel-rail dynamic interaction

3.3 Effects by the superelevation of outer rail

The influence of rail superelevation on dynamic interactions is shown in Fig.5. It can be seen that: the lateral displacement of wheelset decreases as the rail superelevation increases, that of the leading axle is larger than that of the trailing one, and the higher the superelevation, the greater the difference (Fig.5 (a)). With the increase of rail superelevation, the angle of attack of the leading axle increases, while the trailing one decreases accordingly, and they almost both change with a similar linear gradient (Fig.5 (b)). The two changing curves of the lateral forces of the leading and the trailing axle are shown as an approximately "X" shape (Fig.5(c)), that of the leading axle decreases with the increase of rail superelevation within 65mm (deficient superelevation -30mm), then converts to rise continuously, while that of the trailing axle continuously decreases. Within the deficient superelevation range, the lateral force of the trailing axle is greater than that of the leading one, but at the range of surplus superelevation, the lateral force of leading axle is larger conversely. The vertical force curves of the leading and the trailing axle are shown as a "V" shape (Fig.5 (d)), each curve has a turning point, before this point, the vertical force decreases with the increase of superelevation, after this point, it converts to increase. Thus, nearly at the turning point, the vertical force is minimal. For the leading axle, the turning point is about at the deficient superelevation of -15mm, for the trailing one, it is about at the zero (equilibrating superelevation). The wear power curve with the superelecation shows a parabolic shape, at the range of deficient superelevation, the wear power of the leading decreases slowly, while that of the trailing decreases faster relatively; at the range of surplus superelevation, that of the leading begins to increase, and changes obviously rapidly, while that of the trailing converts to increase at the surplus superelevation of 20mm (Fig.5 (e)). At the range of deficient superelevation, the wear index of the leading axle increases slowly with the increase of the superelevation, but that of the trailing axle decreases significantly. At the range of surplus superelevation, the wear index of the leading increases faster, that of the trailing decreases slowly, even when the surplus superelevation is greater than 20 mm, it begins to increase with the superelevation (Fig.5 (f)). Therefore, comprehensive analysis of all these evaluating indicators, much too deficient or surplus superelevation can both worsen the wheel/rail interactions, but a proper deficient superelevation (about -20~0mm) is relatively beneficial to lessen the wheel/rail dynamic response, especially to that of the leading wheelset.



(a) Lateral displacement (b) Angle of attack (c) Lateral force

(d) Vertical force (e) Wear power (f) Wear index

Fig.5 Effects of superelevation of outer rail to wheel-rail dynamic interaction

4 Conclusions

By establishing the vehicle-track coupled dynamic model and the curve track mathematical model, adopting numerical calculation method to program to simulate the influence of geometric parameters such as curve radius, transition curve length, and superelevation of outer rail on the dynamic interaction performance of the heavy-haul freight wagon, and to analyze and evaluate the curving performance of vehicle by the wheel-rail vertical force, wheel-rail lateral force and wheel-rail wear power. Based on the calculation results, the following conclusions can be drawn:

(1) Increasing the curve radius can reduce the wheel-rail wear, but with the increase of the curve radius, the effect is gradually weakened. According to its influence level and effect, the curve radius can be divided into three segments: the sharply descending segment (<400m), transition segment (400~800m) and moderating segment (>800m).

(2) Too small transition curve length (e.g. <20m) will arouse vehicle vibration at the transition-circle connection point, thus increase the wheel-rail dynamic interaction and wheel-rail wear. Increasing the transition curve length can reduce the wheel-rail wear, however, when the length is greater than 50~60m, its change has little influence on the wheel-rail wear.

(3) Too large deficient or excessive superelevation will increase the wheel-rail dynamic interaction and wear, but an appropriate deficient superelevation (e.g. <20mm) is conducive to reduce the wheel-rail wear, which is consistent with the engineering practice of setting a certain deficient superelevation value.

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