Dynamics of the differential coupling independently rotating wheels for railway vehicle

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Received: 5 December 2019; Revised: 12 March 2020; Accepted: 2 July 2020

Abstract
In this investigation, the planetary gear differential is intended to the independently rotating wheels as a passive control device of the left and right wheels. Based on dynamics and kinematics analysis of the railway vehicle system and differential system, the differential coupling wheels vehicle (DWV) model and the comparative models, including rigid-wheelset vehicle (RWV) model and the independently rotating wheels vehicle (IRWV) model, are built. Through numerical studies, it can be concluded that the longitudinal creep forces of independently rotating wheels disappear for the separation of the wheels. But in the coupling effect of differential on wheels, the differential coupling wheels vehicle regains longitudinal creep forces and the resetting capability on straight lines. Compared with the rigid-wheelset vehicle, the differential coupling wheels vehicle has superior dynamics, including safety, guiding performance and wear performance on sharp curves. However, due to lack of sufficient longitudinal creep forces, the dynamics of the differential coupling wheels vehicle is slightly worse than that of the rigid-wheelset vehicle on medium radius curves. In general, the differential coupling wheels vehicle solves the problem of guiding and safety of the independently rotating wheels vehicle, and has better dynamic performance than the rigid-wheelset vehicle on sharp curves, which indicates that the differential coupling wheels vehicle is applicable to the urban railway transit which contains many sharp curves.

Keywords: Independently rotating wheel, Differential coupling, Safety, Guiding performance, Wear performance, Sharp curves

1. Introduction

In the urban railway transport, low-floor vehicles are needed in order that passengers get on and off conveniently. The smallest curve radius of the tram lines is almost 20m at present. Independently rotating wheels vehicle which can save the space of axle, reduce the height of vehicle floor and negotiate the sharp curves is widely used without the need for high platform, and can eliminate the hunting instability of the traditional rigid wheels bogie, (Goodall and Li, 2000; Winter, 2012; Zhang et al., 2016; Courtois, 1994). The independently rotating wheels wheelset refers to a pair of left and right wheels which can keep parallel to each other and rotate freely around their axles, respectively, as there is no rigid connection between the wheels. However, due to the separation of the left and right wheels of the independent rotating wheel, there is no longitudinal creep force and creep torque of the independently rotating wheels theoretically. So, the automatic resetting and curve guiding capability disappear on the independently rotating wheels, (Yerpes et al., 2012). Furthermore, the lateral displacement of the independently rotating wheels increases on straight track and it may run along one side track. This would result in wheel-rail two-point contact. On the curves, the independently rotating wheels would be guided by flange, which leads to serious rim wear and even causes derailment accident accordingly, (Xu and Zeng, 2011). In order to solve the guiding and safety problem of the independently rotating wheels, numerous kinds of methods have been considered in the latest studies, (Cho and Kwak, 2012; Wang, 2001; Matsumura, et al, 2011; Bruni, 2002; Wickens and Nagy, 1999; Powell, 1999; Mei and Goodall, 1999; Shen et al., 2004; Suda et al., 2001). Suda Y
proposed a self-steering reverse taper independent rotating wheel mechanism, (Suda et al., 2012). Pérez added a differential active control strategy to optimize vehicle curving performance, (Pérez et al., 2002). Michitsuji studied the power-steering railway bogie with the independently rotating wheels, (Michitsuji and Suda, 2006). Shi proposed the friction coupling bogie with independently rotating wheels, and the whole vehicle wear power of the friction coupling bogie vehicle is about 2/3 of the rigid axle level, (Shi et al., 2016; Shi et al., 2017). Ji developed a simulation model of active steering control of an independently rotating wheel in an in-wheel motor to investigate effects of torque ripple, (Ji et al., 2018). At present, the control methods of the independently rotating wheels are mostly realized through control of rotation speed or the torque of left and right wheels in the electrical and mechatronic active control method perspective. And these methods rely on the accurate measure of the rotating speed difference between the left and right wheels heavily. There are few researches about the independently rotating wheels controlled by mechanical structures passively which does not need the measure of rotation speed of wheels.

This paper intends the planetary gear differential which is usually used in the automobile for the independently rotating wheels as a kind of mechanical passive control device. With this device, left and right wheels can rotate relatively independently and the guiding problems of the independently rotating wheels are solved by the coupling effect of planetary gear differential simultaneously. The dynamics, especially the safety, guiding performance and wear performance of differential coupling wheels vehicle is studied in this paper. A two-wheelsets test vehicle is designed and running on a special line with tiny curves, so the models in this paper are built based on this vehicle. The planetary gear differential is only intended for powered wheels.

2. Structure and Dynamic Equation
2.1 The structure of DWV

The test vehicle is a two-axle vehicle composed of 1 car body, 1 frame and 2 wheelsets. There is no second suspension, thus the car body is fixed on the frame, as shown in Fig. 1. The car body fixed on frame has 6 DOF (degrees of freedom), namely transversal, longitudinal and vertical displacements, rolling, pitching and yawing. The two wheelsets have only 4 DOF, namely transversal and longitudinal displacements, pitching and yawing. However, the left and right wheels are independent, thus 2 more DOF are assigned to the wheelsets. So, the test vehicle in this paper has 16 DOF in total. The vehicle parameters are showed in Table 1.

| Parameters                               | Value          |
|-----------------------------------------|----------------|
| Wheelbase [mm]                          | 1750           |
| Rolling radius [mm]                     | 300            |
| Wheelset Mass [kg]                      | 200            |
| Wheelset Moment of Inertia $I_x, I_z$ [kg·m²] | 150,400        |
| Tread form                              | LM             |
| Car body Mass [kg]                      | 1125           |
| Car body Mass center x, y, z [mm]       | -32, 0, 1210.7 |
| Frame Mass [kg]                         | 4935           |
| Frame Mass center x, y, z [mm]          | 0, 0, 500      |
| Moment of Inertia $I_x, I_y, I_z$ [kg·m²] | 1870.3, 8127.3, 9903 |
| Stiffness $K_x, K_y, K_z$ [MN·m⁻¹]      | 2.4, 1.0, 0.5  |
| Damping $C_x, C_y, C_z$ [N·s·m⁻¹]       | 400, 8000, 10000 |
| Rail profile                            | UIC 60         |
| Gauge [mm]                              | 1435           |

Table 1. Main parameters of test vehicle
2.2. Dynamic Equation

The planetary gear differential coupling left and right wheels is a symmetrical planetary gear differential that is composed of differential gear housing, planetary gears, planetary gear shaft and sun gears. The differential housing, the final drive driven gear and planetary gear shaft are fixed. Meanwhile, they will rotate around the same axle in the impact of the driving torque from final drive and the interior friction torques. The frictions of bearings are ignored. Left and right wheels are driven by output torques of differential, wheel-rail creep forces and wheel-rail creep moments respectively. The forces and torques of the wheels and differential components are shown in Fig. 2 and Fig. 3.

![Symmetrical planetary gear differential](image)

Fig. 2 Symmetrical planetary gear differential

![Forces of the wheels and differential components](image)

Fig. 3 Forces of the wheels and differential components
The driving force that differential housing acts on the planetary gears can be written as

\[
F = F_1^* + F_2^* + \frac{I_2 \dot{\omega}_h}{r} = \frac{T_0 - I_0 \dot{\omega}_h}{r}
\]  

(1)

Where \( F_1^* \) and \( F_2^* \) are the reaction forces acting on planetary gear by sun gears. \( \dot{\omega}_h \) is the rotation rate of differential housing. \( I_0 \) is the inertia moment of rotation of differential housing. \( I_2 \) is the moment of inertia of revolution of planetary gear. \( T_0 \) is the driving torque of the differential housing from final drive. \( r \) is the pitch radius of the addendum line at the midpoint of sun gears.

The driving forces that the planetary gears act on the left and right sun gears are different, the difference can be written as

\[
F_1 - F_2 = \frac{T_{fp} + I_1 \dot{\omega}_h}{r_p}
\]  

(2)

Where \( F_1 \) and \( F_2 \) are the driving forces acting on the left and right sun gear, respectively. \( J_1 \) is the moment of inertia of rotation of planetary gear around their own axles. \( r_p \) is the pitch radius of the addendum line at the midpoint of planetary gears. \( T_{fp} \) is the friction torque between the back of the planetary gears and differential housing, the additional torques acting on the sun gears generated by the inertia resistance moments of planetary gears that have the same rotating direction with inner sun gear and different rotating directions with outer sun gear. \( \dot{\omega}_3 \) is the rotation rates of planetary gear.

The friction torques generated by the relative motions between the back of the planetary gears and differential housing can be written as

\[
T_{fp} = \mu_{fp} \tan \alpha \frac{\sin \theta_p (T_0 - I_0 \dot{\omega}_h - I_1 \dot{\omega}_h)(d_p^3 - d_i^3)}{12rR_s (\sqrt{4R_s^2 - d_i^2} - \sqrt{4R_s^2 - d_p^2})}
\]  

(3)

Where \( \mu_{fp} \) is the friction coefficients between back surface of planetary gears. \( \alpha \) and \( \theta_p \) are pressure angle and pitch angle of planetary gears, respectively. \( d_i \) and \( d_p \) are the mounting hole diameter and spherical outer circle diameter of planetary gears, respectively. \( R_s \) is the radius of back spherical surface of planetary gears.

The friction torques generated by the relative motion between the backs of the lower speed sun gear and differential housing can be written as

\[
T_{fs1} = \mu_{fs} \tan \alpha \cos \theta_p \left( \frac{D_k^2 + DD_k + D_i^2}{6(D + D_k)} \right) \frac{T_0 - I_0 \dot{\omega}_h - I_1 \dot{\omega}_h}{r} \frac{T_{fp} + I_1 \dot{\omega}_h}{r_p}
\]  

(4)

Where \( \mu_{fs} \) is the friction coefficients between the back of sun gears and differential housing. \( D \) and \( D_k \) are the mounting hole diameter and spherical outer circle diameter of sun gear, respectively.

The friction torque generated by the relative motion between the back of the higher speed sun gear and differential housing can be written as

\[
T_{fs2} = \mu_{fs} \tan \alpha \cos \theta_p \left( \frac{D_k^2 + DD_k + D_i^2}{6(D + D_k)} \right) \frac{T_0 - I_0 \dot{\omega}_h - I_1 \dot{\omega}_h}{r} \frac{T_{fp} + I_1 \dot{\omega}_h}{r_p}
\]  

(5)

From the formulas as of \( T_{fp} \), \( T_{fs1} \) and \( T_{fs2} \), it can be seen that the difference between \( T_{fs1} \) and \( T_{fs2} \) will be greater as the \( T_{fp} \) increases.

When the DWV is running on straight line, the rotation rates of left and right wheels are the same, and the differential housing and the sun gears rotate at a same speed. Moreover, the planetary gears do not rotate, thus the internal friction torques, including \( T_{fp} \), \( T_{fs1} \), and \( T_{fs2} \), equal 0. The driving torque is equally distributed to the left and right sun gear by the differential. The driving torques acting on sun gears can be written as.
\[ T_{Li} = T_{Ri} = \frac{\eta(T_o - I_o \dot{\phi}_b - I_z \dot{\phi}_b)}{2} \]  

(6)

\[ T_{Li} \text{ and } T_{Ri} \text{ are the driving torques of left and right wheels, respectively. } \eta \text{ is the transfer efficiency.} \]

When the DWV is running on curves, the rotation rates of left and right wheels are different. The driving torques acting on sun gears are composed of three parts: \( T_{vo}/2 \), which is the torque averagely transferred from differential housing by the planetary gears and \( T_{fo} \). Thus final driving torques of the left and right wheels can be written as

**Inner wheel**

\[ T_{di} = r \left[ 1 + \mu_p \tan \alpha \cos \theta_p \left( D^2 + DD_k + D_p^2 \right) \right] \left( \frac{T_o - I_o \dot{\phi}_b - I_z \dot{\phi}_b}{2r} + \frac{T_{fo} + I \dot{\phi}_i}{2r_p} \right) + T_{fi} \]  

(7)

**Outer wheel**

\[ T_{d2} = r \left[ 1 - \mu_p \tan \alpha \cos \theta_p \left( D^2 + DD_k + D_p^2 \right) \right] \left( \frac{T_o - I_o \dot{\phi}_b - I_z \dot{\phi}_b}{2r} + \frac{T_{fo} + I \dot{\phi}_i}{2r_p} \right) - T_{fi} \]  

(8)

It can be inferred from the formulas of \( T_{d1} \) and \( T_{d2} \) that the difference between the driving torques of left and right wheels is mainly caused by the moments of rotation of planetary gears and the friction forces among gears.

The dynamic equations of left and right wheels can be written as

**Inner wheel**

\[ I_{wy} \ddot{\beta}_{wLi} = r_i T_{Li} + a \psi_{wLi} \left( T_{Li} + N_{Li} \right) + M_{T_{Li}} + T_{Li} \]  

(9)

**Outer wheel**

\[ I_{wy} \ddot{\beta}_{wRi} = r_i T_{Ri} + a \psi_{wRi} \left( T_{Ri} + N_{Ri} \right) + M_{T_{Ri}} + T_{Ri} \]  

(10)

Where \( T_{Li} \) equals \( T_{d1} \), and \( T_{Ri} \) equals \( T_{d2} \) when DWV turns left while \( T_{Li} \) equals \( T_{d2} \), and \( T_{Ri} \) equals \( T_{d1} \) when DWV turns right.

Where \( \beta_{wLi} \) and \( \beta_{wRi} \) represent the pitching angle of left and right wheels, respectively. \( I_{wy} \) represents the inertia moment of wheels around Y axis. \( \psi_{wLi} \) and \( \psi_{wRi} \) represent the yawing angles of left and right wheels, respectively. \( r_{Li} \) and \( r_{Ri} \) represent the rolling radius of left and right wheels, respectively. \( T_{Li}, T_{Ri}, T_{Ri} \) and \( T_{Li} \) represent the X-axle and Z-axle component forces of wheel-rail creep forces acting on left and right wheels, respectively. \( N_{Li} \) and \( N_{Ri} \) represent the Z-axle component forces of wheel-rail normal force acting on left and right wheels, respectively. \( M_{T_{Li}} \) and \( M_{T_{Ri}} \) represent the Y-axle component moments of wheel-rail creep moments acting on the left and right wheels, respectively. \( i = 1,2 \), which represents the front and rear wheel respectively.

The equations of vehicle system dynamic can be written with mass matrix \( M \), stiffness matrix \( K \) and damping matrix \( C \) as

\[
\begin{bmatrix} M \\ C \end{bmatrix} \ddot{u} + \begin{bmatrix} C \end{bmatrix} \dot{u} + \begin{bmatrix} K \end{bmatrix} u = \{ P \}
\]  

(11)

Where \( \{ \ddot{u} \} \) represents the accelerations matrix of vehicle system, \( \{ \dot{u} \} \) represents the velocities matrix of vehicle system, \( \{ u \} \) represents the displacements matrix of vehicle system, \( \{ P \} \) represents the generalizing forces matrix of vehicle system.

3. Dynamic model and Verification

3.1 Dynamic models

To contrast the dynamics of DWV with that of RWV and IRWV, the dynamic model of these three types of vehicles were built in dynamic software Universal Mechanism (Developer: Laboratory of Computational Mechanics, Bryansk State Technical University, Version: UM7.6, Major features: is a multi-body software package used for the simulation of kinematics and dynamics of mechanical systems, and includes a number of task-oriented modules specific to the
simulation of railway vehicle dynamics: UM Loco, UM Train, UM Train3D, UM RCF, and UM Wheel-Rail Wear). The DWV model was built based on the actual two-wheelsets test vehicle. And the RWV and IRWV model were built as the contrast models. They had the same vehicle parameters and tread profiles, as showed in Table 1. The only difference between them were the wheelset structures. The dynamic model of DWV, IRWV and RWV are showed in Fig. 4.

![Dynamic models of three types vehicles: DWV, IRWV, RWV](image)

**Fig. 4** The dynamic models of three types vehicles: (a) DWV; (b) IRWV; (c) RWV

### 3.2 Test and verification

To verify the model of DWV, the accelerations of the test vehicle when it ran in the curves of 8m radius at the speed of 3.6km/h were measured. The acceleration sensor was fixed on the longitudinal and lateral center of vehicle body top surface, as shown in Fig. 5, and the three-direction accelerations of the vehicle body were measured. The accelerations of vehicle body in the three directions measured and the simulated of the DWV are contrasted in the Fig. 6 and the root mean square (RMS) of accelerations are shown in the Table 2.

![Acceleration sensor position](image)

**Fig. 5** The three-direction acceleration sensor on vehicle

From the Fig. 6, it can be seen that the time histories of three direction simulation accelerations are so close to the measured accelerations. Table 2 illustrated the difference between the root mean square (RMS) of simulation accelerations and measured accelerations is less than 10%. So the simulation model of the vehicle (DWV) is verified and suitable to be used in research of the dynamics of the DWV.

|               | Longitudinal | Lateral | Vertical |
|---------------|--------------|---------|----------|
| **Simulation RMS [m/s²]** | 0.5142       | 1.0492  | 1.7391   |
| **Test RMS [m/s²]**      | 0.5685       | 1.0913  | 1.7574   |
| **Difference [%]**        | 9.55         | 7.41    | 3.23     |

**Table 2** The contrast between test result and simulation result
3.3 The rotation rate of wheels on curved track

The main difference among independently rotating wheels, differential coupling wheels and rigid wheelset is that the left and right wheels of IRW and differential coupling wheelset have different rotation rates while the left and right wheels of rigid wheelset can only rotate synchronously. The rotation rates of the independently rotating wheels and the differential coupling wheels were calculated when the DWV and IRWV negotiated three types of sharp curves, including three types of simulation conditions (Condition1- Condition3) in Table 3. To make the result clearer, no track irregularity was applied to all the tracks. The simulation results are shown in Fig. 7.

| Condition | Radius[m] | Velocity[km/h] |
|-----------|-----------|----------------|
| Condition1| 20        | 18             |
| Condition2| 50        | 36             |
| Condition3| 100       | 54             |
| Condition4| 500       | 72             |

![Fig. 6 The three-direction acceleration sensor on vehicle](image)

![Table 3 The simulation conditions](image)

![Fig. 7 The rotation rates of the differential coupling wheels (First row) and the independently rotating wheels(Second row) in three conditions: (a)Condition1; (b)Condition2; (c)Condition3;](image)
In terms of the differential coupling wheels and the independently rotating wheels, it can be seen from Fig. 7 that the rotation rates of right wheel (the outer wheel) are always bigger than that of the left wheel (the inner wheel) on the entire curve section in all the three conditions. The rotation rates of both left and right wheel of the differential coupling wheels and the independently rotating wheels almost have the same amplitudes and time history curves on the curve section. After negotiating the curve section, the rotation rates of left and right wheels of the differential coupling wheels can become the same rapidly, but the left and right wheels of the independently rotating wheels cannot recover synchronous rotation, especially in condition 3. This indicates that the differential coupling wheels can achieve the relative rotation as the independently rotating wheels on the curves and the coherent rotation as rigid wheelset on the tangent.

4. The dynamic performance
4.1 The guiding performance on curved track

To investigate the guiding performances, the angles of attack of the first wheelset of DWV and RWV were calculated in all the three conditions. In addition, the guiding performances of those vehicles on negotiating the medium radius curves in Condition 4. The simulation results are shown in Fig. 8. The longitudinal creep forces which were necessary for guiding were also calculated and showed in Fig. 9.

As shown in Fig. 8, the angles of attack of the first wheelset of DWV almost showed the same time history curves with those of RWV on the entire curved track in condition 1, condition 2 and condition 3. All the differences of the angles of attack are negligible, which indicates that the two vehicles have the close guiding performance on sharp curves at the low speed. From Fig. 9, it can be seen that when passing sharp curves, the longitudinal creep forces of DWV were not equal to the RWV, with the maximum amplitude of 30.2kN, 29.3kN and 45.1kN respectively that were not equal to the maximum amplitude of 63.4kN, 68.5kN and 64.5kN of the longitudinal creep forces of RWV. However, the DWV and RWV contained the close guiding capability in the sharp curves, thus the longitudinal creep forces of DWV were sufficient for guiding. When the RWV negotiated the medium radius curve at higher speed, it had a smaller maximum amplitude of angle of attack than that of DWV. So, the RWV had better guiding capability. In this condition, the longitudinal creep force with the maximum amplitude of 39.8kN was not sufficient for guiding, which resulted in the poor guiding capability of DWV in contrast with RWV. Through comparison, it can be summarized that the DWV has the same guiding performance as RWV on sharp curves at low speed but poorer guiding performance than RWV on medium radius curves at relatively higher speed.

![Fig. 8 The wheel-rail angles of attack of DWV and RWV: (a)Condition1; (b)Condition2; (c)Condition3; (d)Condition4;](image-url)
Fig. 9 Longitudinal creep forces of the first wheelset of DWV and RWV: (a) Condition 1; (b) Condition 2; (c) Condition 3; (d) Condition 4;

4.2 The resetting capability on straight track

When DWV, RWV, and IRWV negotiated 500 meters radius curve at the speed of 72 km/h, the wheels displaced outward to the curve and the outer wheels were against the track. When vehicles passed the curve section and entered the tangent part, the lateral displacement and the longitudinal creep forces of the first wheelset of vehicles were investigated to determine their resetting capabilities on straight track. The simulation results are shown in the Fig. 10.

Fig. 10 The lateral displacements and longitudinal creep forces of the first wheelset of DWV, RWV, and IRWV

Through the comparison of the lateral displacements of wheelsets in Fig. 10, it can be seen that only wheelsets of DWV and RWV eventually returned back to the middle position of the track. The maximum of lateral displacement on the sharp curve are almost same with each other, and the maximum of the difference among them is little than 1%. After passing the curves, the lateral displacement of wheelsets of IRWV keeps -6.15mm, and left and right wheels rotated at different speeds. Without longitudinal creep force, the IRWV have no automatic resetting capability. The maximum of longitudinal creep force of RWV is 60.8kN, and the maximum of longitudinal creep force of DWV is 39.8kN on the curves. However, the longitudinal creep force of IRWV keeps the value of zero. Since the DWV have smaller longitudinal creep force than that of RWV on the curved track, the wheelset of DWV returned back to the center position of the track,
but its resetting time is 6.4s later than that of RWV wheelset after passing the curve. But the longitudinal creep force of DWV is sufficient for resetting of the wheel. In the coupling effect of differential to wheels, the DWV retains longitudinal creep forces and gains the resetting capability on straight track.

4.3 The safety on curved track

On negotiating curves, the safeties of the DWV, RWV and IRWV were studied through comparison of wheel-rail lateral forces and derailment coefficients of first wheelset of these vehicles in the four conditions.

![Fig. 11 The wheel-rail lateral forces of the first wheelset of DWV, RWV and IRWV](image1)

![Fig. 12 The derailment coefficients of the first wheelset of DWV, RWV and IRWV](image2)
Fig. 13 The lateral displacements of the first wheelset of DWV, RWV and IRWV on circular curves: (a) Condition 1; (b) Condition 2; (c) Condition 3; (d) Condition 4.

It can be seen from Fig. 11 and Fig. 12 that the wheel-rail lateral forces and derailment coefficients of the first wheelset of DWV and IRWV have almost the same time history curves on negotiating curve in all the conditions, which indicates that they have almost the same running safety on the curves. In Fig. 11, the maximum of wheel-rail lateral forces of RWV are 17.60kN, 19.35kN and 17.62kN in condition 1, condition 2 and condition 3 separately, which are obviously bigger than that of DWV and IRWV, but the maximum of wheel-rail lateral forces of RWV is 6.67kN which is smaller than that of DWV and IRWV in condition 4. Similarly, in Fig. 12, the maximum of derailment coefficients of RWV are 0.5762, 0.6263 and 0.6181 in condition 1, condition 2 and condition 3 separately, which are obviously bigger than that of DWV and IRWV, but the maximum of derailment coefficient of RWV is 0.2825 which is smaller than that of DWV and IRWV in condition 4. In the sharp curves conditions, including condition 1, condition 2 and condition 3, for the separation of the wheelset, the longitudinal creep forces of IRWV and DWV which would impede the relative motions of left and right wheels disappear or decline on the curved tracks. Meanwhile, it can be observed from Fig. 13 that the lateral displacement of wheel of RWV is 9.55mm, 18.16mm and 17.55mm on the curves in condition 1, condition 2 and condition 3 separately, which are bigger than that of DWV and IRWV. The main reason is that the relative rotation of left and right wheel of DWV and IRWV weakens the lateral movement and lateral displacement of wheels. As a result, the wheel-rail lateral forces and derailment coefficients of the first wheelset of DWV and IRWV are smaller than those of the RWV, which indicates that the DWV implies superior safety on negotiating sharp curves compared with RWV. On the contrary, when the three types of vehicles negotiate medium radius curves, without sufficient guiding capability, the lateral displacements of wheels of DWV and IRWV are bigger. The lateral movements result poorer safety of DWV on the entire curved tracks compared with RWV on negotiating medium radius curves.

4.4 The wear performance of DWV

The independently rotating wheels are mainly guided by the flange on the curves, thus the flange wear is serious. To compare the wear performances of DWV with RWV and IRWV, the wear powers of DWV, RWV and IRWV were calculated and shown in Fig. 14.

The wear power of wheels $P$, including the wear powers of tread and flange, can be calculated as follows, (Zhang and Qi, 2012)

$$P = v \cdot (F_x \xi_x + F_y \xi_y)$$  

(12)
Where $F_x, F_y$ represent the longitudinal and lateral creep forces respectively, $\xi_x, \xi_y$ represent the longitudinal and lateral creepages respectively. The creep forces and creepages are calculated by FASTSIM algorithm, the simplified theory of Kalker.

As shown in Fig. 14, the whole vehicle wear power of DWV and that of IRWV are almost the same in all conditions. Comparing the wear powers of the vehicles on the curves, it can be seen that the DWV and IRWV both have smaller wear powers than RWV on the sharp curves but bigger wear powers than RWV on the medium radius curves. With the separation of wheelset, the creep forces and creepages of wheels of DWV and IRWV decline on the curve section which decrease the wear of wheels of DWV and IRWV. However, on the medium radius curves, the DWV and IRWV would be guided by flanges, thus the wear of the flanges of wheels of the differential coupling wheels and the independently rotating wheels are more serious, which leads to the more serious wheel-rail general wear of the differential coupling wheels and the independently rotating wheels.

5. Conclusion

The planetary gear differential is employed as a passive control device of the independently rotating wheels in this paper. After the dynamics and kinematics analysis, the dynamics of DWV, including the safety, guiding performance and wear performance, is compared with that of IRWV and RWV, the following conclusions are made from the numerical studies:

1. With the coupling of planetary gear differential, DWV can attain the relative rotation as the independently rotating wheels on the curved track and the coherent rotation as rigid vehicle on the tangent track. In terms of guiding performance on curves, DWV has the same guiding performance with RWV on sharp curves at low speed but poorer guiding performance compared with RWV on medium radius curves.

2. For the separation of the wheelset, the longitudinal creep forces of IRWV disappear and the longitudinal creep forces of DWV are also smaller than those of RWV. However, in the coupling effect of differential to wheels, the DWV retains longitudinal creep forces and gains the resetting capability on straight track as RWV, which solves the problem that IRWV runs against one said of the track after negotiating the curved sections.

3. With smaller lateral displacements and movements, DWV inheriting the merit of IRWV has superior safety on negotiating sharp curves than RWV. On the contrary, when vehicles negotiate medium radius curves, the safety of DWV is worse than that of RWV.

4. On the sharp curved tracks, with smaller lateral displacements and wheel-rail angles of attack of wheels, the
wear powers of DWV and IRWV are both smaller than that of RWV. By contrast, for lack of sufficient guiding capabilities on curves, DWV would be guided by flanges. Thus, the wear of wheels of DWV is more serious than RWV on the medium radius curves. In general, the dynamics of DWV is better than RWV on the sharp curves, which indicates that the DWV is applicable to the urban railway transit which contains many sharp curves. For the special structure of the DWV, small radius curve is conducive to reflect its feature.

Further research

In this paper, a two-wheelsets vehicle model with planetary gear differential was built based on the parameters of actual vehicle to investigated the influence of planetary gear differential to the railway vehicles. We only concerned that the planetary gear differential was used in the powered wheels. The further research would be investigated including the dynamics of the non-powered wheels coupled by the planetary gear differential and the dynamics of low-floor tram with the differential coupling wheels rather than the two-wheelsets vehicle. This paper only compared the dynamics of IRWV, DWV and RWV, the comparison of the performance improvement of the proposed passive measures with some control methods should be conducted in the future. The design parameters of differentials are really important to the performance of DWV. In the test vehicle, we just choose a mature technology differential. In the further research the influence of design parameters of differentials to the performance of DWV should be studied respectively.

Acknowledgements

The author(s) disclosed receipt of the following financial support for the research, authorship, and/or publication of this article: This work has been supported by the National Natural Science Foundation of China (grant number 61174214).

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