



THE MECHANISM OF DRIVE SYSTEM FLEXIBLE SUSPENSION AND ITS APPLICATION IN LOCOMOTIVES

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Abstract. The drive system flexibly suspended on the bogie frame is conducive to the lateral dynamic performances of locomotive. In order to clarify the mechanism and optimize the suspension parameters using this model, a bogie dynamics model with 10 degrees of freedom (including the drive system) was established. The lateral dynamic performances were analyzed with the different suspension parameters. The mechanism was determined from the dynamic vibration absorber and the best suspension frequency of the drive system was put forward. The multi-body-dynamics model with two types of drive systems was simulated in locomotives to verify this theoretical analysis. When the suspension frequency is close to the hunting motion frequency of the wheelset, the locomotive dynamics performed the best. The length of the swing rods at the front and rear drive systems were different on the same bogie, which improved the locomotive dynamic performances within a wider range of speed and the wheel conicity. The track shift forces of the locomotive were reduced by 45% and 34% respectively with the swing rods on the motor side and the non-motor side relative to the non-flexible suspension.

Keywords: locomotive; dynamics; drive system; flexible suspension; lateral stability.

Introduction

The drive system is suspended flexibly under the bogie frame or under the car body through the swing rods or other elastic. This is a new technology that has been used on high speed trains in recent years (Middendorf 1996; Hödl, Haigermoser 1992; Alfi *et al.* 2008; Lata 2008). This approach can improve the lateral dynamics performances of locomotives or motorized cars at high running speeds. For example, this technology has been implemented in 160 km/h locomotives or high speed motorized cars, such as the CRH3, 'Blue Sword' and the 'China Star' in China. Many 200 km/h class locomotives are based on the E120 locomotive in Europe, but their drive systems are changed to flexible suspension (Luo 2005). Alfi *et al.* (2008) analyzed the effects of motor connection on the critical speed of high-speed railway vehicles. Luo *et al.* (Luo 2005; Luo, Jin 2007; Luo *et al.* 2005), Zhang *et al.* (2007, 2008) have made significant contributions to flexible suspension locomotive dynamics and their engineering applications in China. It has long been understood that the mechanism of flexible suspension of the drive system combined with the elastic connection can decouple the mass and inertia between the drive system and bogie frame. This in turn, isolates the lateral movement of the drive system from the bogie

frame (Luo 2005; Segieth 1996). This qualitative explanation is not sufficient because the flexible suspension parameters cannot be given out accurately. In this paper, a bogie dynamics model with 10 degrees of freedom, including the elastic connection of drive system, is established. The lateral dynamic performances are analyzed with different suspension parameters. This phenomenon has been made clear from the dynamic vibration absorber and the best suspension frequency is put forward according to the anti-resonance theory. Finally, two types of multi-body-dynamic models of locomotives with different drive system structures were simulated to verify the theoretical analysis. The results provide a theoretical basis for the engineering applications of the drive system suspension optimization parameters of the railway locomotive and power cars.

1. The Mechanism of Flexible Suspension of Drive System

1.1. Dynamics Model of the Flexible Suspension Drive System Bogie

For a high-speed locomotive, the drive systems are suspended under a bogie frame through a single rubber joint and two swing rods. The single rubber joint and the two swing rods are arranged on the two sides



of drive systems separately along track direction. The two swing rods can be arranged on the side of the motor or the non-motor end. Relative to the bogie frame, the drive systems possess the lateral and yaw degrees of freedom on the horizontal direction. In this paper, a bogie is equipped *Bo'* axle arrangement and the impact of car body movement is ignored. The bogie frame is assumed to be connected with the fixed coordinate system through the secondary suspension. The dynamic model is shown in Fig. 1.

Calculating the mass and yaw inertia of the flexible suspended part of drive system, the non-flexible suspended part was added to the bogie frame. Taking the

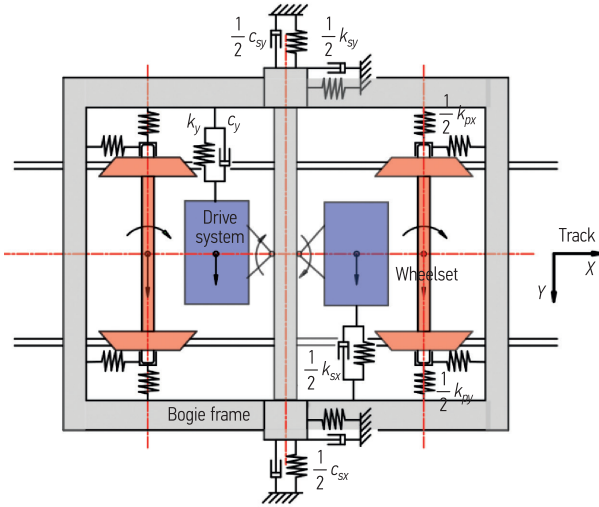


Fig. 1. Dynamics model of the bogie with flexible suspension of drive system

lateral and yaw vibrations of the bogie frame, wheelset and drive systems into account, the dynamics model was found to have five rigid bodies and 10 degrees of freedom. The dynamics equation is:

$$M\ddot{X} + C\dot{X} + KX = Q, \quad (1)$$

where: the M , C and K are the mass, damping and stiffness matrix of the system separately; the Q is the external force vector. The parameters of the dynamics model are shown in Table 1.

For the motion equation of the wheelset, the lateral and yaw movement of wheelset are coupled by the wheel-rail tangential forces. The lateral and yaw frequencies of the wheelset are the same and their phase angles have a certain difference. Ignoring the spin creep forces of the wheelset, the stiffness matrix and damping matrix K_w , C_w of the linear dynamics equation of the wheelset with elastic positioning in a straight line are as follows:

$$K_w = \begin{bmatrix} k_{py} + k_{gy} & -2f_{\eta} \\ 2\lambda_e l_0 f_{\xi} / r_0 & l_1^2 k_{px} + k_{g\psi} \end{bmatrix}; \quad (2)$$

$$C_w = \frac{1}{v} \begin{bmatrix} 2f_{\eta} & \\ & 2l_0^2 f_{\xi} \end{bmatrix}. \quad (3)$$

For the asymmetric stiffness matrix K_w , there are complex eigenvalues for the corresponding system matrix. If any real part of the eigenvalues is positive, the system is unstable. For the wheel-rail system, the non-diagonal items in K_w represent the coupling of the wheelset's lateral and yaw movement, which input energy into the system while the C_w consumes energy. From the above equation, C_w is related to the speed v . As v increases, the

Table 1. The parameters of model

Symbol	Physics mean	Value	Symbol	Physics mean	Value
m_w	Mass of wheelset	3085 kg	$2b$	Wheel base	2.5 m
I_w	Yaw inertia of wheelset	2024 kg·m ²	$2l_0$	Distance of contact spot	1493 mm
m_f	Mass of bogie (include non-flexible suspend part of drive system)	7186 kg	$2l_1$	Lateral spacing of primary suspend	2200 mm
I_f	Yaw inertia of bogie (include non-flexible suspend part)	9751 kg·m ²	$2l_2$	Lateral spacing of secondary suspend	2200 mm
m_d	Mass of flexible suspend part of drive system	1765 kg	r_0	Wheel rolling radius	0.625 m
I_d	Yaw inertia of flexible suspend part	1019 kg·m ²	λ_e	Wheel-rail contact conicity	0.15
k_{px}	Primary longitudinal stiffness per axle	40 kN/mm	f_{ξ}	Longitudinal creep coefficient	8.624e6
k_{py}	Primary lateral stiffness per axle	6 kN/mm	f_{η}	Lateral creep coefficient	8.144e6
k_{sx}	Secondary longitudinal stiffness	0.8 kN/mm	k_{gy}	Gravitational Stiffness	41.4 kN/m
k_{sy}	Secondary lateral stiffness	0.8 kN/mm	$k_{g\psi}$	Gravitational angular stiffness	-23 kN/rad
c_{sx}	Secondary longitudinal damping	1000 kN·s/m	k_r	Contact stiffness of wheel-rail	50 kN/mm
c_{sy}	Secondary lateral damping	60 kN·s/m	v	Speed	-m/s

damping decreases. In a period of hunting motion, if the input and the consumed energy are equal, the linear system is in the critical stability state. The corresponding speed v is the linear critical speed.

The lateral and yaw vibration of drive system have the same frequency and attenuation coefficient. The expression of lateral stiffness k_y and damping c_y of the drive system suspension are defined as follows:

$$\begin{aligned} k_y &= (2\pi f_y)^2 m_d; \\ c_y &= \xi_y 2m_d (2\pi f_y), \end{aligned} \quad (4)$$

where: f_y is the drive system lateral suspension frequency; ξ_y is the drive system lateral suspension-damping ratio; m_d is the mass of flexible suspended parts of drive system. Similarly, we can get the expression of yaw stiffness and damping between the drive system and bogie frame.

1.2. Effect of Flexible Suspension on the Linear Stability of the Bogie

For the rigid wheelset on the rail vehicle, the wheelsets represent the hunting motion on the track because of the conicity of the wheel profile. As the primary guide stiffness and the gravity, the hunting motion of the wheelset is stable within a certain range of speeds. That is, the wheelset tends to the center of the track. When the vehicle speed exceeds a certain value (called critical speed), the maximal lateral displacement of the wheelset tends towards a nonzero value (the wheel-rail gap usually). In engineering, when the lateral displacement of the wheelset can't converge quickly, it known as 'vehicle lateral instability'. Due to the changes of some parameters, such as the increases in the wheel conicity caused by wear, the vehicle will reduce the lateral stability significantly. The lateral stability will worsen the vehicle dynamic performances or even cause derailment. Thus, the rail vehicle requires the lateral stability to be routinely checked to ensure a high safety margin. The lateral stability of the rail vehicle has been one of the important and difficult problems in vehicle system dynamics (Sun 1993).

Checking the lateral stability of the vehicle includes both linear and nonlinear stability. For the linear stability, you can estimate the vehicle's critical speed and optimize some parameters quickly, while identifying the instability mode through the rigid body vibration frequencies and the corresponding mode shapes.

The eigenvalues and eigenvalue vectors of the homogeneous items of equation (1) can be calculated in the way of state space for the non-symmetrical stiffness and non-proportional damping matrix. The imaginary and real part of each eigenvalue denote the circular frequency of this vibration and its attenuation coefficient. The eigenvalue vector denotes the rigid mode shapes of the corresponding frequency. The root locus curves of the linear system are composed of all eigenvalues at all calculated speeds. If the real part of all eigenvalues are negative, the motion of the system is stable.

In order to discuss the effect of the structure form of drive systems and the suspension parameters on the

bogie lateral stability, the linear stability of 4 types of models categorized according to 6 conditions and analyzed:

1. no drive system model;
2. drive system fixed to the bogie frame non-flexibly;
3. single wheelset with elastic positioning, that when equivalent to the bogie frame has only one degree of freedom with forward direction;
4. drive system flexible suspension, $f_y = 3$ Hz, $\xi_y = 1.0$;
5. drive system flexible suspension, $f_y = 3$ Hz, $\xi_y = 0.3$;
6. drive system flexible suspension, $f_y = 4$ Hz, $\xi_y = 0.3$.

Fig. 2 shows the calculated linear critical speeds of the different models and suspension parameters of the drive systems. The lateral stability of the drive system is worst when fixed to the bogie frame non-flexibly because the drive system increases the mass and yaw inertia of the bogie frame. The single wheelset with flexible positioning means the bogie frame does not participate in hunting motion and its linear critical speed is the largest. Fig. 3 shows the root locus curves of the three models. For the bogie model, the smaller imaginary part are the frequencies of the wheelset hunting (1÷4 Hz) and the bigger are the lateral vibration frequencies of the bogie frame (5÷10 Hz). The critical speeds of the bogie

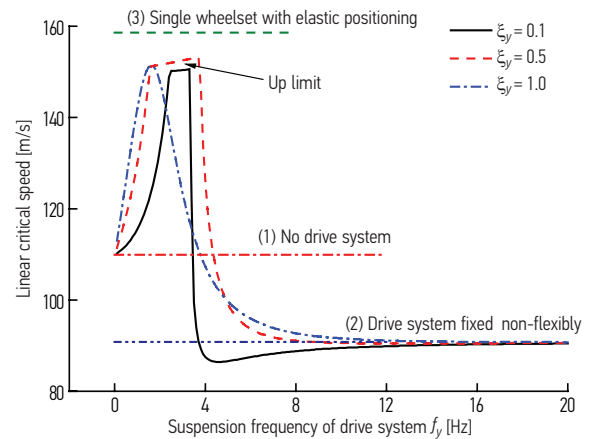


Fig. 2. The linear critical speed

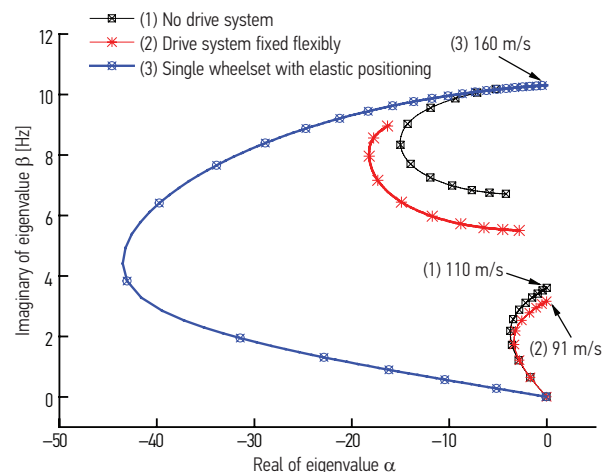


Fig. 3. The root locus curves

with the flexible suspension are related to the suspension frequency f_y . When the f_y is 0, it is equivalent to no drive system. The large f_y means that the drive system is fixed to the bogie frame non-flexibly. When f_y is close to the hunting frequency of wheelset, the lateral stability of the bogie with drive system flexible suspension is significantly better than no drive system and the critical speed is related to suspension damping ξ_y . The flexible suspension of the drive system is equivalent to the dynamic vibration absorber and its lateral and yaw vibrations absorb the vibration of the bogie frame. When f_y is close to the excitation frequency, the vibration amplitude of the bogie frame is at its minimum and that enhances the lateral stability of the bogie. In Fig. 4, the corresponding lateral displacement of the drive system is at its lowest when the critical bogie speed is at its maximum, rather than the relative displacement between drive system and bogie frame (interpreted as dissipating energy of suspension damping). Thus, this exercise verifies the explanation of the dynamic vibration absorber.

The first instable mode of rail vehicles is usually the bogie hunting motion, that is the frequencies and phases of the wheelsets and frame motions are the same. Except for the curve (5) in Figs 3 and 5, the first instable mode is bogie hunting as the speed increases. The first instable mode of curve (5) is the lateral vibration of the bogie frame. The critical speed of the bogie with drive system flexible suspension are the upper limits of curves in the Fig. 2. If the instability of the rail vehicle occurs,

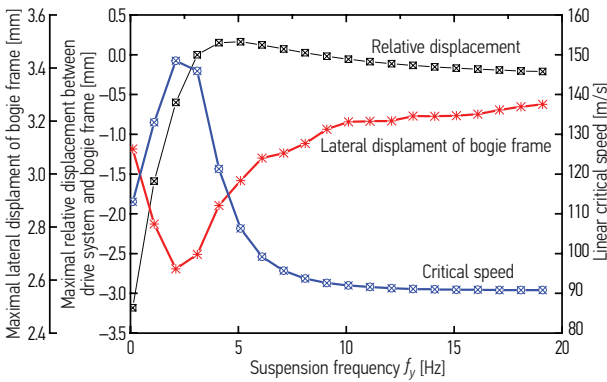


Fig. 4. The simulation results

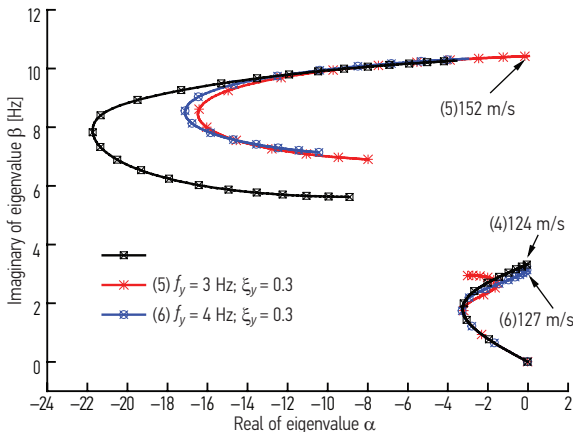


Fig. 5. The root locus curves

the main vibration frequencies of all rigid bodies in the vehicle is the unstable mode frequency. The corresponding lateral vibration frequency of the bogie frame is high and can lead to a severe vibration. Thus, the suspension parameters of the vehicle should first be optimized to avoid bogie frame lateral instability.

1.3. Effect of Flexible Suspension on the Track Shift Forces

There is a large dynamic force between the wheel and rail due to the track irregularity. The track shift forces (sum of the guiding forces on the two wheels of the same axle) is an important indicator of a vehicle's dynamic performance. As the track shift forces increase with the increasing running speed, the high-speed vehicle attention is affected by the low dynamic action force. In order to study the effects of flexible suspension on the track shift forces, the dynamic model of the bogie in equation (1) which takes the horizontal track irregularity and the wheel-rail gap into account was simulated, and the flange contact are used the unilateral contact stiffness k_r and damping to simulate. The numerical method of Runge-Kutta is used to get numerical results and statistically evaluate the maximal track shift forces.

The simulation results of different suspension frequencies f_y on the AAR5 class track irregularity are shown in Figs 6 and 7. The effect of flexible suspension

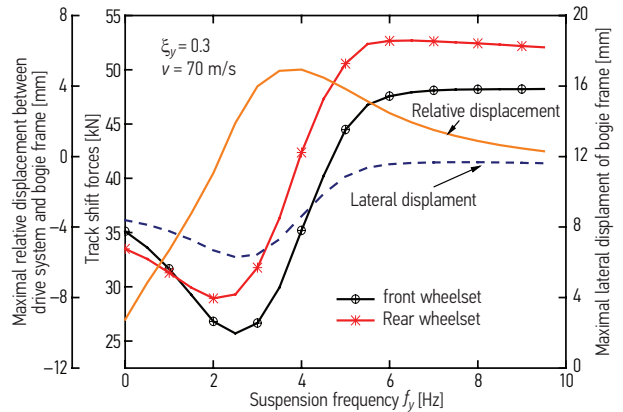


Fig. 6. The simulate results with different f_y

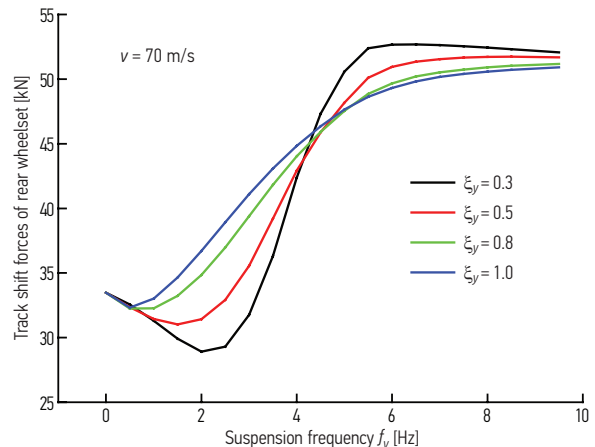


Fig. 7. The effect of ξ_y on the track shift forces

on track shift forces are the same as the lateral stability. The flexible suspension of drive system reduces the track shift forces and the lateral and yaw displacement of the bogie frame. The track shift forces and the bogie frame displacement are lowest when the suspension frequency is close to the hunting frequency. In Fig. 7, when the range of hunting frequency is small, a smaller suspension damping ξ_y is beneficial to reduce the track shift forces.

The rear wheelset of the track shift force is usually bigger than the front wheelset on a straight line. The simulation results show that the suspension frequency f_y has influences on the ratio of the track shift forces of the rear and front wheelsets. The bogie lateral acceleration on the front and rear wheelsets is shown as follows respectively in the paper by Alfi *et al.* (2008):

$$\ddot{y}_1 = \ddot{y}_f + p\ddot{\psi}_f = -\omega^2 (y_0 + p\psi_0) e^{i\omega t}; \quad (5)$$

$$\ddot{y}_2 = \ddot{y}_f - p\ddot{\psi}_f = -\omega^2 (y_0 - p\psi_0) e^{i\omega t}. \quad (6)$$

The bogie lateral acceleration on the front and rear wheelsets is the vector sum of the lateral and yaw accelerations of the bogie. The acceleration vector is shown in Fig. 8. The analysis of the linear vehicle system shows that the vibration frequency and phase of the wheelsets and bogie frame in one bogie (caused by the wheelset hunting motion) are the same, and the yaw phase angle of the bogie frame is about 90 degrees from the lateral vibration of bogie frame. As shown in Fig. 8, when the phase difference is 90 degrees, the lateral acceleration of the front and rear wheelsets on the bogie are equal.

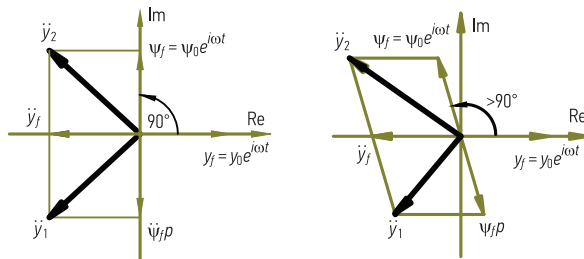


Fig. 8. The vector relationship of acceleration

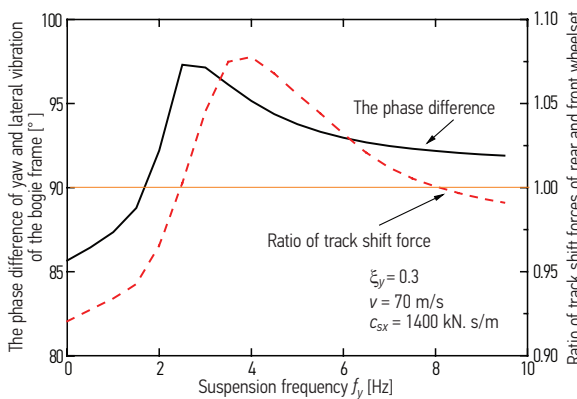


Fig. 9. The effect of f_y on the ratio of track shift forces and phase difference

When the phase difference is greater than 90 degrees, the lateral acceleration of the rear wheelset position on bogie is bigger than the front wheelset position. These are the intuitionistic explanations to explain the different track shift forces of the front and rear wheelsets in one bogie on a straight line.

The phase difference of the yaw and lateral vibrations of the bogie frame is related to the mass and yaw inertia and the horizontal suspension parameters of the bogie frame. The different suspension parameters of drive systems can affect the bogie frame's natural frequency of lateral vibration and the phase difference. Fig. 9 shows the effect of the suspension frequency on the ratio of track shift forces. The simulation results show that although the relationship of the phase difference and the ratio of track shift forces is not in strict accordance with the vector analysis in Fig. 8, the overall trend is roughly consistent. The smaller lateral suspension stiffness of the drive system is beneficial to balance the track shift forces of the front and rear wheelsets.

2. The Theoretical Explanation of the Flexible Suspension

For the locomotive with flexible suspension, when the suspension frequency of drive system is close to the wheelset hunting frequency, the locomotive will have the best dynamic performance. In order to explain this phenomenon, the simplified model of the bogie dynamic vibration absorber is established, using the anti-resonance theory to explain the mechanism and put forward the optimal value of the suspension parameter of flexible suspension. For the simplified model as shown in Fig. 10, the system's DOFs are equivalent lateral movements, and the dynamics parameters are equivalent parameters. Assuming the lateral position of car body is fixed, this model takes into account the lateral vibration degrees of freedom of the bogie frame and drive system. The horizontal simple harmonic motion of the wheelsets serves as the system's excitation.

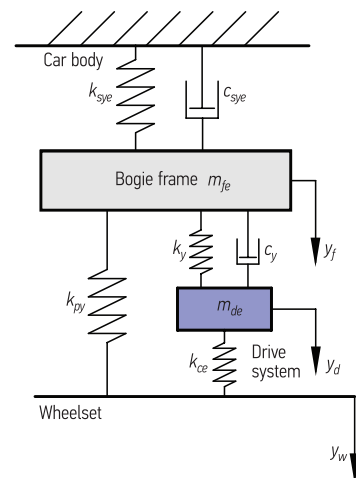


Fig. 10. The lateral simplified model of the railway vehicle

Ignoring the damping, the transfer function of the ratio of lateral amplitude of the bogie frame y_f and the wheelset y_w is as follow:

$$G_{11}(\omega) = \frac{y_f}{y_w} = \frac{k_{py}(k_y + k_{ce} - m_{de}\omega^2) + k_{ce}k_y}{(k_{sye} + k_{py} + k_y - m_{fe}\omega^2)(k_y + k_{ce} - m_{de}\omega^2) - k_yk_y}, \quad (7)$$

where: k_{py} is the primary lateral stiffness; k_{ce} is the equivalent combination lateral stiffness of the transmission of double hollow shaft; k_{sye} is the equivalent secondary lateral stiffness; m_{fe} is the equivalent mass of the bogie frame; m_{de} is the equivalent mass of flexible suspended parts of the drive system; ω is the excitation frequency. When the system parameters and ω are certain, the ratio of vibration amplitudes of the bogie frame and the wheelset is a constant. Equation 7 equals 0 when $\omega = \omega_n$ satisfies the following:

$$k_{py}(k_y + k_{ce} - m_{de}\omega_n^2) + k_{ce}k_y = 0, \quad (8)$$

when the bogie frame does not move, it is known as anti-resonance. The corresponding excitation frequency ω_n is known as the ‘anti-resonance frequency’. Anti-resonance frequency is different from the resonant frequency. When the denominator of $G_{11}(\omega)$ is zero, the corresponding excitation frequency is the resonant frequency, while the anti-resonance frequency is when the numerator is zero. For the vibration system with n degrees of freedom, there are n resonance frequencies and n anti-resonance frequencies, respectively (Sun 1993).

The hunting frequency is the main frequency within the lower frequency region of the lateral wheelset movement. When the speed is constant, the excitation frequency ω_n is the hunting frequency ω_0 . So the optimal lateral suspension stiffness k_y of drive system is as follow:

$$k_y = \frac{k_{py}}{k_{py} + k_{ce}}(m_{de}\omega_0^2 - k_{ce}). \quad (9)$$

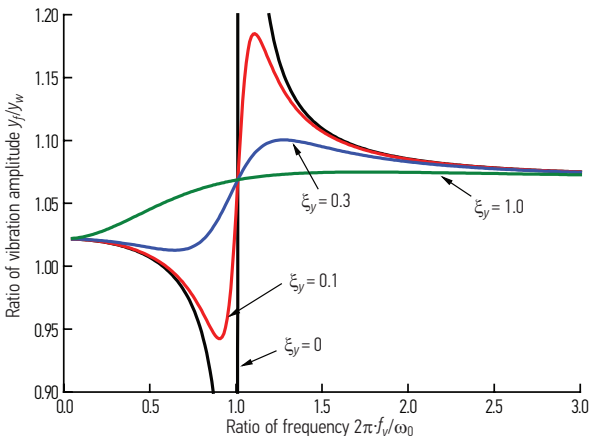


Fig. 11. The vibration amplitude ratio of the bogie frame and wheel-set with different f_y

From the above equation, the corresponding k_y increases as the ω_0 and m_{de} increase and the k_y decreases as k_{ce} increases. As the k_{ce} becomes smaller than the k_{py} (Zhang et al. 2008), we conclude that:

$$k_y \approx m_{de}\omega_0^2 - k_{ce}. \quad (10)$$

As the hunting frequency of wheelset is relevant to the running speed, a reasonable value of k_y can be selected based on the common speed. To weaken the lateral vibration and reduce the resonance amplitude of drive system, the drive system is connected with the lateral damper. Fig. 11 shows the vibration amplitude ratio between the bogie frame and wheelset with the different ξ_y . The horizontal coordinate is the ratio of suspension frequency of the drive system and the wheelset hunting frequency. As the effect of suspension dampens, when the ratio of suspension frequency and hunting frequency is slightly less than 1, the vibration amplitude ratio of the bogie frame and wheelset is lowest.

3. The Application of Different Suspension Frequencies on the Same Bogie

There is no optimal suspension frequency of the drive system for all conditions due to changes in hunting frequencies with the changes in running speeds and wheel conicity. The hunting frequencies of the wheelset usually increase with the increase of the running speeds and the wheel conicity λ_e as shown in Fig. 12. From the mechanism of flexible suspension, the suspension frequencies of the drive system should be close to the hunting frequency to support better dynamic performances of the rail vehicle. The suspension frequencies cannot be changed for the mechanical connection without a control system. Thus, the application of different suspension frequencies of the drive system on the same bogie is an effective problem-solving measure.

Figs 13–15 show the simulation results of the bogie model shown in Fig. 1 with different suspension frequencies and different wheel conicity. The 3 Hz suspension frequency of the front and rear drive system reduces the track shift forces when the bogie runs at high speed, and

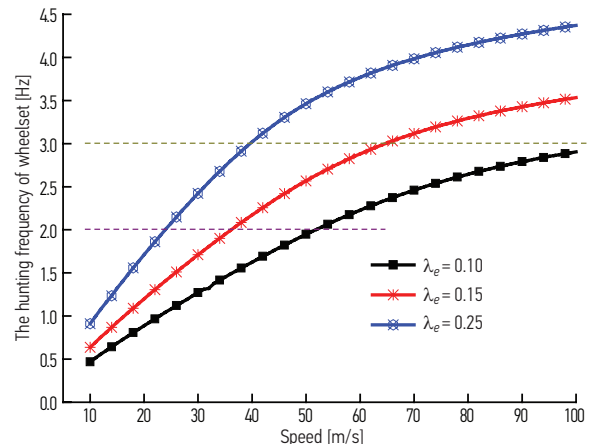


Fig. 12. The hunting frequencies of wheelsets with the different speed and wheel conicities λ_e

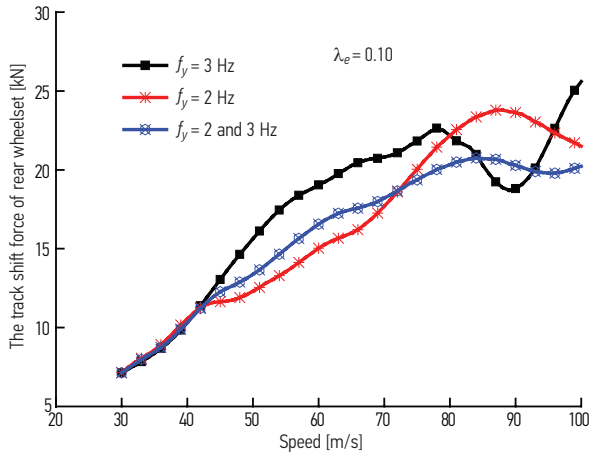


Fig. 13. The track shift forces with the different suspension frequencies at λ_e is 0.1

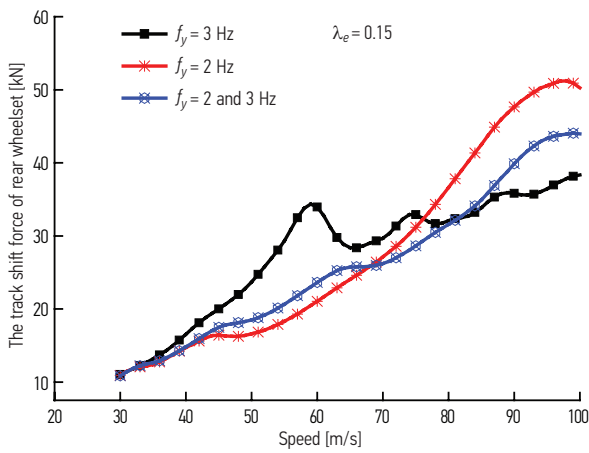


Fig. 14. The track shift forces with the different suspension frequencies at λ_e is 0.15

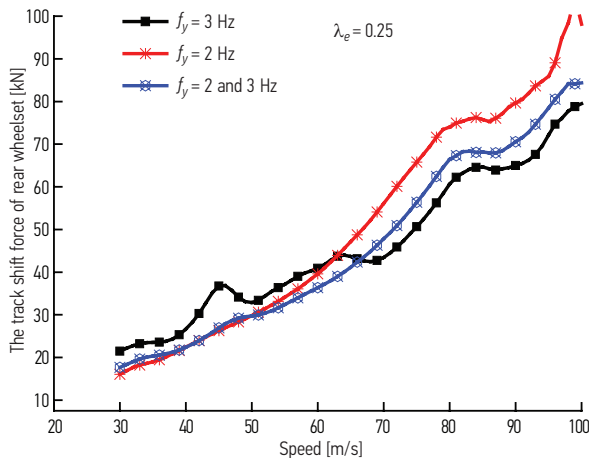


Fig. 15. The track shift forces with the different suspension frequencies at λ_e is 0.25

the 2 Hz suspension frequency reduces the track shift forces at lower speed. The suspension frequencies are 2 and 3 Hz of the front and rear drive system respectively and can improve the dynamic performances for a wide range of speeds. The appropriate running speeds for the flexible suspension decrease with an increase in wheel conicity.

4. The Suspension Frequencies of Two Types of Drive Systems

For a high-speed locomotive, the drive systems are suspended under a bogie frame through a single rubber joint and two swing rods. The single rubber joint and the two swing rods are arranged on the two sides of the drive systems separately along the track direction. The swing rods can be arranged close either to the motor, or to the other side, as shown in Figs 16 and 17 respectively. The wheelset and the drive system are connected by the transmission of double hollow shafts. The combined lateral stiffness is k_c as shown in the figures. A lateral damper shown as lateral damping c_y in the figures is arranged between the drive system and the bogie frame. The positions of the mentioned components above are shown in Figs 16 and 17.

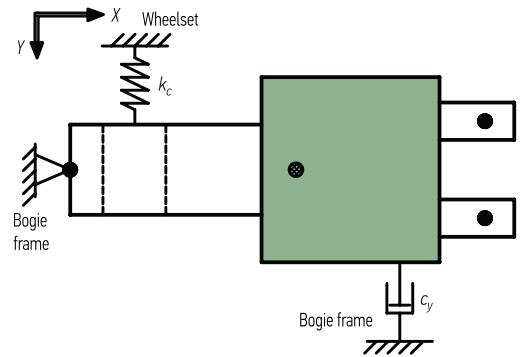
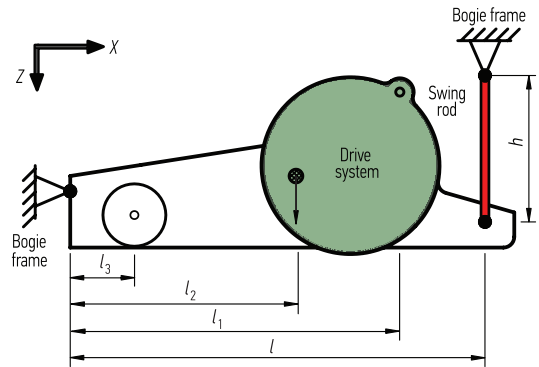


Fig. 16. The flexible suspension drive system A

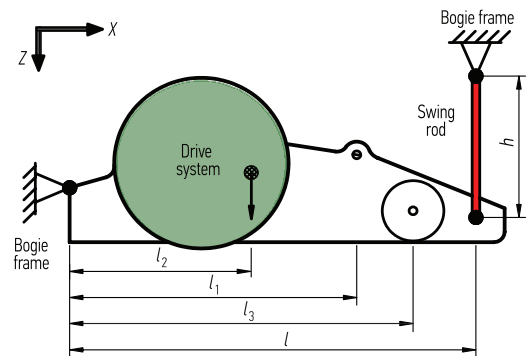


Fig. 17. The flexible suspension drive system B

The lateral natural frequency of the drive system is defined as the suspension frequency, which depends on the length of the swing rods h , the k_c and the corresponding positions. Assuming that the bogie frame and wheelset are fixed and the angular displacement of the swing θ is small, the horizontal motion equation for the drive system is shown as:

$$\begin{aligned} & \left(m_d i_2^2 h^2 + I_d h^2 / l^2 \right) \ddot{\theta} + c_y i_1^2 h^2 \dot{\theta} + \\ & \left(m_d g i_2 h + k_c i_3^2 h^2 \right) \theta = 0. \end{aligned} \quad (11)$$

The parameters of m_d and I_d are the mass of the drive system and the yaw inertia respectively. The g is the gravity acceleration. The ratio of the corresponding position parameters to l is $i_k = l_k / l$, $k = 1, 2, 3$.

The suspension frequency of the drive system is as follows:

$$f_y = \frac{1}{2\pi} \sqrt{\frac{m_d g i_2 h + k_c i_3^2 h^2}{m_d i_2^2 h^2 + I_d h^2 / l^2}} \approx \frac{1}{2\pi} \sqrt{\frac{lg}{l_2 h} + \frac{k_c}{m_d} \left(\frac{l_3}{l_2} \right)^2}. \quad (12)$$

The f_y decreases when the h and l_2 increase and the k_c and l_3 decrease. Namely the connecting lateral stiffness between the drive system and the frame decreases.

This paper selected two different locomotives in the 200 km/h speed class as the analysis objects, of which drive systems were flexibly suspended on the frame respectively in the two structural forms mentioned above. The structural parameters of the two drive systems are shown in Table 2.

Table 2. The parameters of the drive system

Parameters	h [m]	l [m]	l_1 [m]	l_2 [m]	l_3 [m]
Drive system A	0.47	1.7655	1.384	1.078	0.612
Drive system B	0.485	1.812	1.622	0.808	1.2

Parameters	k_c [kN/mm]	c_y [kN·s/m]	m_d [kg]	I_d [kg·m ²]
Drive system A	0.34	40	4000	1070
Drive system B	0.34	40	4146	683

According to Equation 7, the f_y reduces with increasing h and decreasing k_c . The suspension frequencies of two types of drive system are shown in Figs 18 and 19. The shorter length of the swing rod has a greater impact on f_y . When the h is greater than 100 mm, the f_y is less than 3 Hz. It's not obvious if one is able to reduce the lateral stiffness of the drive system by continuing to increase the length of the swing rod. On the drive system B, because the swing rod and the transmission of the double hollow shafts are on the same side, the k_c has an obvious impact on the f_y and the f_y is larger. When the hunting frequency of the wheelset is 2 Hz, for the k_c in Table 1, the optimal length of the swing rod on drive system A is 0.1 m, while the longer length of the

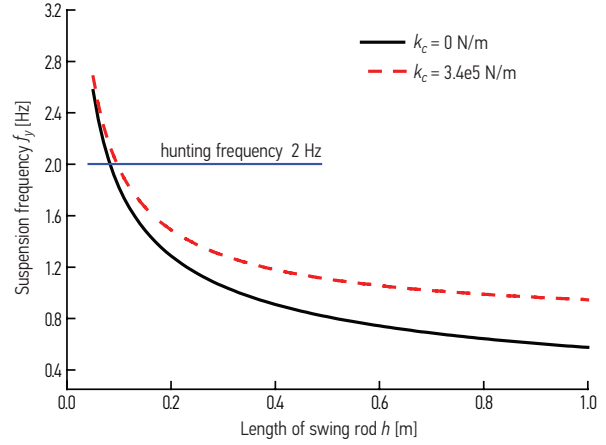


Fig. 18. The suspension frequency of drive system A

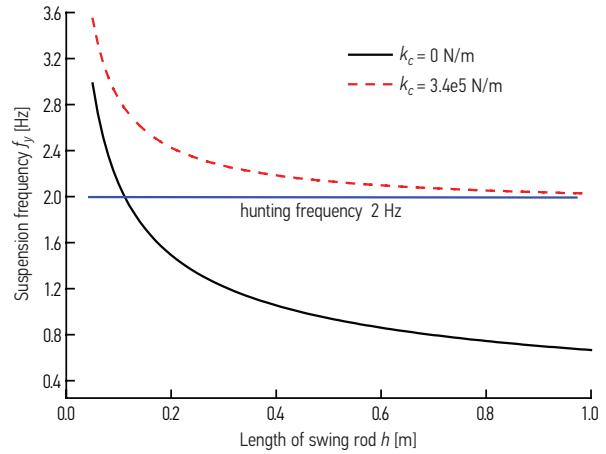


Fig. 19. The suspension frequency of drive system B

swing rod on the drive system B is better in the calculation range. But the optimal length is 0.1 m if the k_c equals 0 m.

5. Vehicle Dynamics Simulation and Verification

The dynamics analyses were carried out on two different locomotives in order to verify the effect of drive system flexible suspension on the vehicle system dynamics. The two locomotives dynamics models were established by the multi-body-dynamics software SIMPACK, which is shown in Fig. 20 and the main parameters are given in Table 3. The drive systems A and B were applied to the locomotives 1 and 2 with the standard gauge respectively. The wheel profile was of the wear type JM3. The track irregularity of DB high class are adopted in a straight line for simulation, in which the speed is 200 km/h and the corresponding hunting frequency of the wheelset is 2 Hz.

The track shift force curves with different swing rod lengths are shown in Figs 21 and 22 for locomotive 1. The track shift force of the rear wheelset is greater than that of the front wheelset in one bogie. The maximum displacements of the bogie and the track shift forces have the same rules with the changing swing rod. The corresponding minimum track shift forces and lateral displacement appear when the length of the swing rod is 0.1 m for the

Table 3. The main parameters of locomotives

Parameters	Axle arrangement	Axle load [t]	Motor arrangement	Wheel base [m]	Bogie base [m]
Locomotive 1	2C _O	21	Sequence	2.0÷2.0	13.05
Locomotive 2	2C _O	21	Sequence	2.15÷2.0	11.76

Parameters	Primary longitudinal stiffness [kN/mm]	Primary lateral stiffness [kN/mm]	Second lateral stiffness [kN/mm]
Locomotive 1	28.67	3.67	0.428
Locomotive 2	28.7	3.74	0.426

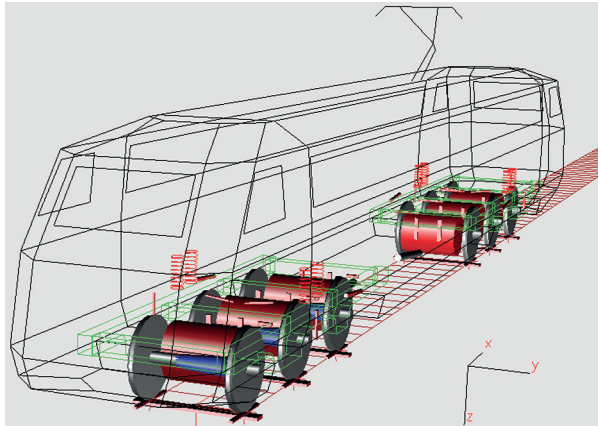


Fig. 20. The multiple-body-dynamics model of locomotive

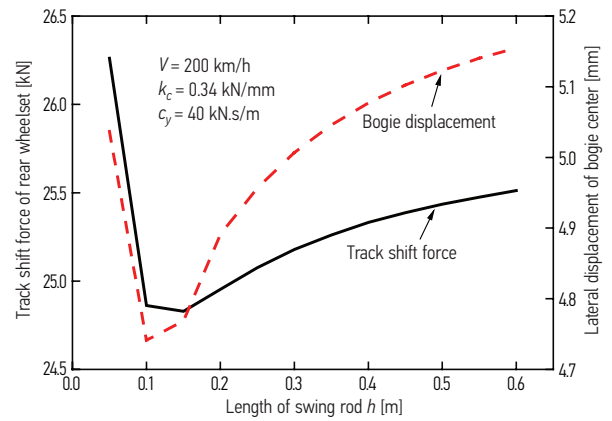


Fig. 22. The simulation results of locomotive 1 when the k_c is 0.34 kN/mm

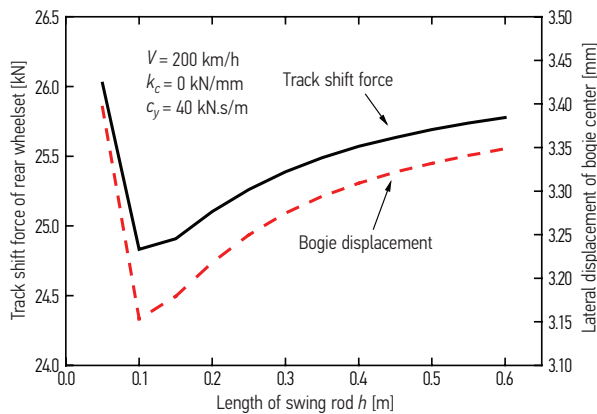


Fig. 21. The simulation results of locomotive 1 when the k_c is 0 kN/mm

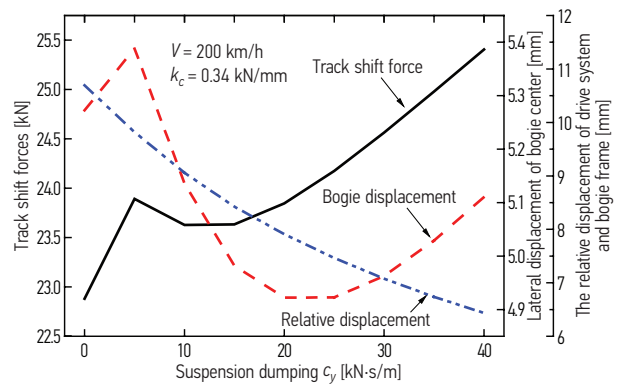


Fig. 23. The simulation results of locomotive 1 with different c_y

the k_c is 0 or 0.43 kN/mm. This is the same as the demonstration in Fig. 18. The maximum track shift forces and displacement of the bogie are 48 kN and 9.3 mm respectively when the length of the swing rod is 0, which means the connection between the drive system and the frame is non-elastic. The lateral force of locomotive 1 is reduced by 45% when the drive system is in flexible suspension compared to the non-elastic connection.

Simulation results of the different c_y of locomotive 1 on the track shift forces and lateral displacements of the bogie frame and drive system are shown in Fig. 23. The track shift forces and the lateral displacement of the bogie frame are minimal when the c_y equals 10 to 20 kN·s/m, while the higher c_y is favorable for reducing the relative displacement between the drive system and bogie frame.

Figs 24 and 25 show the effect of differing swing rod lengths on the track shift forces of the locomotive 2. The track shift forces and lateral displacement of the bogie are minimal when the length of swing rod is 0.1 m. When the k_c is 0.34 kN/mm, the increasing length of the swing rod is better for dynamic vehicle performance, as is demonstrated in Fig. 9. The track shift force of locomotive 2 reduces 34% when the drive system is in flexible suspension compared to the non-elastic connection.

Simulation results of the different c_y of locomotive 2 on the track shift forces and lateral displacements of the bogie frame and drive systems are shown in Fig. 26. The track shift forces and the lateral displacement of bogie frame are minimal when the c_y equals 5 to 10 kN·s/m, while the higher c_y is favorable for reducing the relative displacement between the drive system and bogie frame.

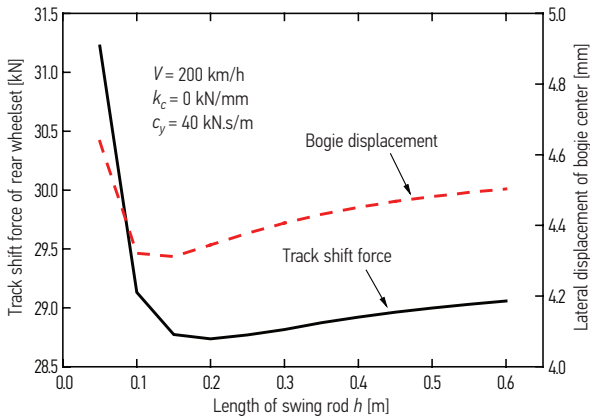


Fig. 24. The simulation results of locomotive 2 when the k_c is 0 kN/mm

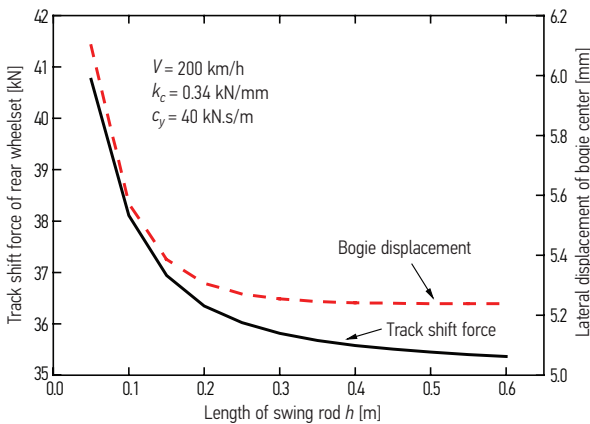


Fig. 25. The simulation results of locomotive 2 when the k_c is 0.34 kN/mm

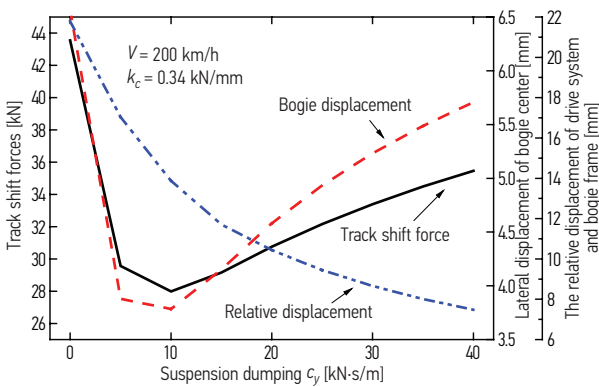


Fig. 26. The simulation results of locomotive 2 with different c_y

The nonlinear critical speeds of the locomotives are calculated according to the EN14363 (Ploach 2006; Yao *et al.* 2011). The optimal lengths of the swing rod for locomotives 1 and 2 whose drive systems are suspended flexibly are different. Variations in critical speed with the changes in length of the swing rod are shown in Fig. 27.

There is no optimal swing rod length for any drive system due to the changes in hunting frequencies and the changes of the running speeds and wheel conicity. The vehicle dynamics analysis shows that when the

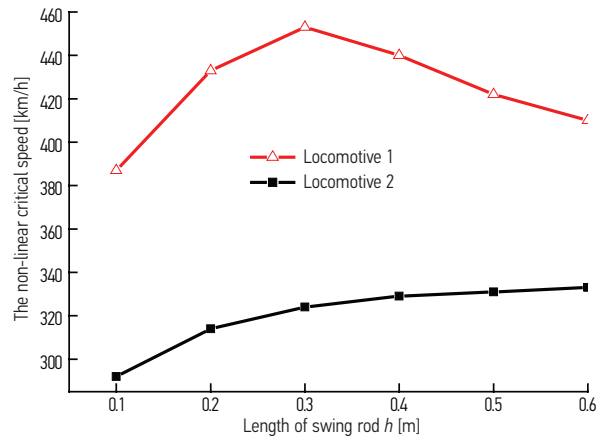


Fig. 27. The effect of length of swing rod on the non-linear critical speed

length of the swing rod is longer than 0.1 m, the suspension frequency of the drive system is lower than 3 Hz. Therefore, the flexible suspension of the drive systems has an obvious role in enhancing the vehicle dynamics performances, while no apparent influence can be made by further increasing the length of swing rod.

Conclusions

1. The drive system mounted on the bogie frame flexibly will help to improve locomotive dynamic performances. When the suspension frequency of the drive system is close to the wheelset hunting, the dynamic performance will be the best. The relative movement of the drive system to the bogie frame absorbs the movement energy of the frame that reduces the amplitude of hunting on the frame, which improves the lateral stability and reduces the track shift forces of locomotives.
2. The suspension frequency of the drive system depends on the length of the hanging rods, and the lateral synthesis stiffness of the double hollow shaft transmission with the corresponding position parameters. When the swing rods and the transmission are located on the same side, the effect of the lateral synthesis stiffness due to the double hollow shaft transmission equipment on the suspension frequency is obvious. When the length of the swing rods is greater than 100 mm, the suspension frequency of the drive system is less than 3 Hz.
3. When the swing rods and transmission equipment are located on the same side, the improvement of the dynamic performance of the locomotive is more visible. The axle lateral forces of locomotives whose hanging rods are located on the motor side and non-motor side are decreased by 45% and 34% respectively, relative to the non-flexible suspension of the drive system.
4. When the length of the swing rods of the front and rear drive systems are different in the same bogie, the locomotive dynamic performances are better within a broader range of speeds and the slopes of the tread.
5. The suspension frequency of the drive system should be close to the hunting frequency of the wheelset.

The appropriate suspension damping of the drive system is conducive to improve the dynamic performance of the locomotive, but in order to reduce the relative lateral displacement of the drive system to the bogie frame, it is necessary to increase the suspension damping.

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