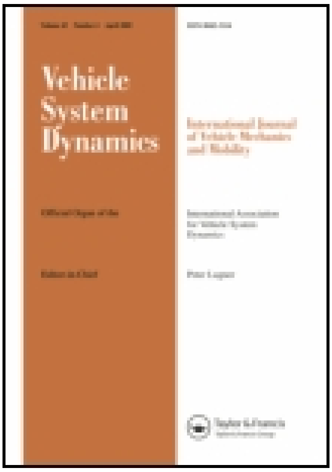

Vehicle System Dynamics: International Journal of Vehicle Mechanics and Mobility

Publication details, including instructions for authors and subscription information:
http://www.tandfonline.com/loi/nvsd20

Dynamic performances of an innovative coupler used in heavy haul trains

Yuan Yao[a], Xiaoxia Zhang[b] & Hongjun Zhang[a]

[a] State Key Laboratory of Traction Power, Southwest Jiaotong University, Chengdu 610031, People's Republic of China

[b] College of Nuclear Technology and Automation Engineering, Chengdu University of Technology, Chengdu 610059, People's Republic of China

Published online: 18 Jul 2014.

Vehicle System Dynamics, 2014
Vol. 52, No. 10, 1288–1303, http://dx.doi.org/10.1080/00423114.2014.938664

Dynamic performances of an innovative coupler used in heavy haul trains

Yuan Yao[a*], Xiaoxia Zhang[b] and Hongjun Zhang[a]

[a]*State Key Laboratory of Traction Power, Southwest Jiaotong University, Chengdu 610031, People's Republic of China;* [b]*College of Nuclear Technology and Automation Engineering, Chengdu University of Technology, Chengdu 610059, People's Republic of China*

(*Received 11 March 2014; accepted 22 June 2014*)

An innovative structure for a heavy haul coupler with an arc surface contact and restoring bumpstop is proposed. This coupler has a small lateral force at a small yaw angle and a limitable yaw angle to ensure an allowable coupler lateral force under intense compressive force. The main structural characteristic of the combined contact coupler is a lateral movable follower with an appropriate friction coefficient of 0.06–0.08 and a slide block with a single freedom of longitudinal movement. In order to verify and simulate the performances, a multi-body dynamics model with four heavy haul locomotives and three detailed couplers was established to simulate the process of emergency braking. In addition, the coupler yaw instability and wheel set lateral forces were tested in order to investigate the effect of relevant parameters on the coupler performances. The combined contact coupler is suitable for heavy haul train for a good dynamic performance.

Keywords: heavy haul train; coupler stability; structure; locomotive; derailment; wheel set lateral force

1. Introduction

A large lateral force on a coupler caused by emergency braking may lead to the derailment of the locomotive in the middle of a train. A smaller lateral force on a heavy haul coupler is required as the train tonnage and speed increase. The repeated occurrence of train derailment caused by emergency braking is a clear indication that both production and operation departments as well as academic institutions require a better understanding of how to prevent these accidents.[1–4] Several researchers previously investigated the longitudinal dynamics of a long train and discussed the effects of buffer parameters on a train's longitudinal impact forces.[5–8] Cole [9–11] conducted many studies on the derailment issues involving heavy haul trains. These included longitudinal dynamics, wagon connection models, comfort, train management, and driving practices. A comprehensive nonlinear model was developed [4,6] to study severe braking conditions and minimise the tendency for train derailment. These studies, however, did not consider the problem of decreased stability when longitudinal compressive forces are applied to a coupler. The current understanding of heavy haul train coupler stability and its effect on derailment is still rudimentary and needs to be studied further.

*Corresponding author. Email: yyuan8848@163.com

Currently, heavy haul couplers in the world can be divided into two types according to their stability principle. The first type of coupler has an arc contact surface at its end.[12] Wu [13,14] used a rotate joint to simulate the yoke key of the coupler, and postulated that friction torques supplied by two end contact surfaces will stabilise the coupler by balancing the yaw moment caused by the coupler's yaw angle. Yao [15] analysed the stability of arc surface couplers with the friction circle theory and illustrated the importance of the coupler's arc friction coefficient and the locomotive's lateral suspension stiffness to the bearing capacity of this type of coupler. The second type of coupler sets an elastic bumpstop which can prevent a larger yaw angle on the coupler. Ma [16] studied the effects of the bumpstop coupler and buffer system on the dynamic performance of heavy haul locomotives to better understand possible derailment issues during braking. In his study, the bumpstop function was simulated by a simplified nonlinear stiffness. Yao [17] analysed the geometric and mechanical characteristics of the bumpstop coupler and proposed a matching principle of the coupler with the secondary suspension parameters of the locomotive to reduce the wheel set lateral forces when the train was braking. Both the couplers have their own characteristics for stabilisation and reducing lateral component force, as well as their own matching principle with the locomotive's suspension. In this study, an innovative structure that integrates the merits of both types of couplers was proposed and the dynamic performances were simulated to verify the feasibility.

2. The structure and mechanical characteristics of coupler

2.1. *The coupler structure*

The first type of coupler adopts an arc surface at its end to achieve the lateral offset of contact points when the coupler rotates. The two unequal radius arc surfaces and flat yoke key are the key characteristics shown in Figure 1. The follower has only a single freedom of longitudinal motion and the rotation centre of the coupler is the contact point at each moment. Thus, it uses the flat yoke key to accommodate the offset of the arc centre at the coupler's end. The flat yoke key transfers the coupler's pull force and the push force is transferred by the arc surface contact. Because the yaw angle of the factual transfer force line through the contact points at either end of the coupler is less than the coupler's yaw angle, the coupler lateral force is reduced and the coupler has a self-stabilisation characteristic within certain yaw angles. However, the arc surface coupler requires a larger locomotive's lateral suspension stiffness to keep stable for a small yaw angle. The coupler stability was improved significantly by improving the secondary structure. When the maximum coupler longitudinal compressive force reaches 2500 kN, the required friction coefficient is reduced from 0.6 to 0.35 relative to the original locomotive.[15] But the increased stiffness of lateral suspensions should affect the vehicle lateral dynamics, such as the ride quality with lateral irregularities and the behaviour on curves or switches. The second type of coupler adopts a bumpstop to limit its yaw angle and its yoke key is round. The rotation centre of the coupler is the rounded yoke key at each moment for the moment equilibrium as shown in Figure 1. The round key transfers the coupler's pull and push force. In the author's studies,[17] it is advised that a smaller secondary lateral stiffness as well as the secondary stopper's lateral free clearance of 35 mm and elastic clearance of 15 mm be selected to match the coupler for a small lateral force. The selection of the locomotive's secondary lateral stiffness is different for the two types of couplers. The bumpstop coupler does not have a higher requirement for the suspension stiffness relative to the arc surface coupler.

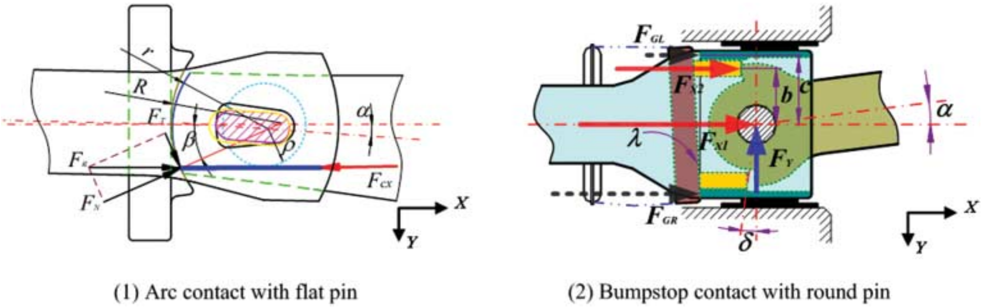

Figure 1. The structure and principle of the two types of coupler.

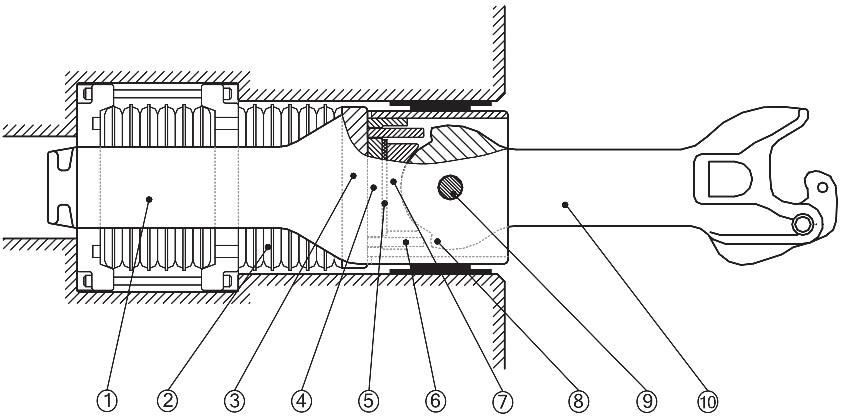

Figure 2. The structure of diesel-electric locomotive coupler with arc surface contact. (1) Coupler yoke, (2) Rubber buffer, (3) Front plate, (4) Slide block, (5) Friction pad, (6) Bumpstop, (7) Follower, (8) Shoulder, (9)Yoke key, and (10) Coupler.

For the first type of coupler, the advantage is that the coupler lateral forces are small, which is conducive to reducing the side wear of rim. The potential coupler instability, however, can cause serious accidents because there is no mechanism to limit the increased yaw angle. A larger secondary lateral stiffness and a high friction coefficient on the arc surfaces are required to ensure the stability of the combined contact coupler. For the second type of coupler, there is a larger lateral component force when the train is braking because the coupler will yaw to one side until the bumpstop contacts easily under a larger longitudinal compression force. The generated wheel set lateral forces cause some undesirable issues such as uneven wear on wheel and rail side wear. However, the bumpstop ensures the safety of the locomotive.

Based on the merits of the two types of couplers, an innovative structure is proposed in this paper to achieve a small coupler lateral force and assured security. The two yoke keys have different functions, but their functions of compensation lateral mobility and as a rotation centre are conflicting. To resolve this conflict is the key problem. The proposed coupler's structure is shown in Figure 2. It is composed of a coupler yoke, rubber buffer, front plate, slide block, follower, bumpstop, round yoke key, and coupler. The slide block, bumpstop, friction pad, and follower are set sequentially from the front plate to the coupler's end. The slide block has a single freedom of longitudinal mobility and the follower can move laterally. The bumpstop passes the slide block to transfer the longitudinal force from the shoulder to the front plate. The friction pad is used to change the follower's friction coefficient. Note that

Table 1. Parameter values.

Parameter	Description	Value
l	Half of the coupler length (m)	0.712
b	Lateral distance from bumpstop to yoke key (m)	0.135
c	Half of the width of the slide block (m)	0.150
R	Arc radius of follower (m)	0.150
r	Arc radius of coupler's end (m)	0.130
l_b	Half of the bogie longitudinal distance (m)	5.075
l_c	Half of the longitudinal distance of coupler mounting seats (m)	17.652

the end of the coupler has a shoulder in contact with the bumpstop which is used to limit the coupler yaw angle, and the arc surface's contact achieves a small coupler lateral force. The coupler and the coupler yoke are connected by a round yoke key. When the coupler is pulled, it pulls the yoke key and moves the coupler yoke, thus compressing the rubber buffer and providing both traction stiffness and damping. When the coupler is compressed, the yoke key cannot transfer the longitudinal compressive force and instead, it is transferred by the arc surface contact to compress the rubber buffer. The combined contact coupler's main feature is the adoption of a movable lateral follower to achieve ideal movement of the arc surface contact. The combined contact coupler does not have as high a requirement for the suspension stiffness as the arc surface coupler. The coupler's parameter values are shown in Table 1.

2.2. *The contact analysis of arc surfaces and its stability mechanism*

The contact analysis of the end arc surface and the lateral mobilisable follower is shown in Figure 3. The contact is a line of contact because of the two arc's different radii. When the coupler rotates and the yaw angle is α, the position of the contact point moves along the arc because of the tangential friction force. The angle between the centre line of the follower and the contact point (called the contact angle) is β. The coupler rotates around the yoke key and the tangential friction force moves the follower laterally with the displacement s. Assuming that the friction of the contact surfaces is not saturated, the two arc contact surfaces experience pure rolling. The angles of α and β and the lateral displacement s satisfy the following relationship:

$$\beta = \frac{r}{R-r}\alpha, \tag{1}$$

$$s = (R-r)\cdot \sin\beta. \tag{2}$$

During braking, the coupler experiences the longitudinal compressive force, Q, and the yaw moment, M, when the coupler has a certain yaw angle. M is equivalent to Q translated in the lateral direction for a distance L, where $L = M/Q$. The moved force line is the actual transfer force line and it passes through the contact point. According to the friction circle theory, a friction circle with a radius ρ is drawn in the circle centre of the arc surface in the coupler end as shown in Figure 3. The ρ and friction coefficient μ of the arc surface have the following relationship:

$$\rho = r\cdot \sin(\tan^{-1}\mu). \tag{3}$$

When L is smaller than ρ, the friction does not reach saturation and the contact is in a pure rolling state. When L is equal to ρ, the contact is in critical state and the force line is at a tangent to the friction circle. When L is larger than ρ, the two arc surfaces slide. The

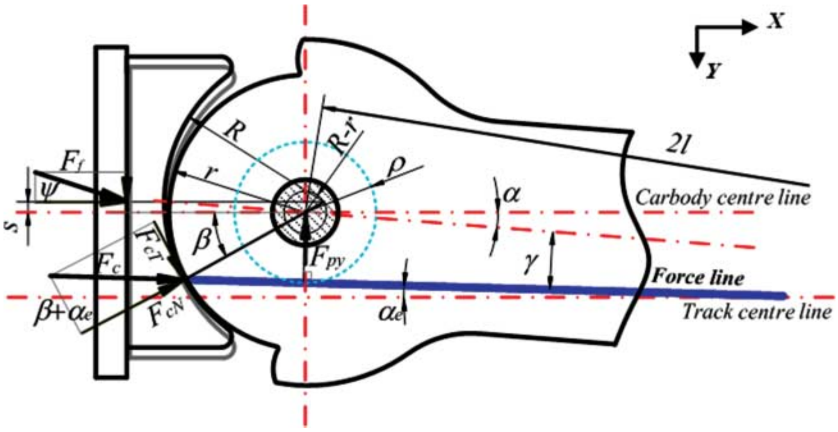

Figure 3. Contact analysis of the end arc surfaces.

Table 2. The geometry parameters of the coupler at different arc friction coefficients.

μ	ρ (mm)	Y_{bc} (mm)	$\gamma_{\max}$ (°)
0.20	25.50	51.00	2.05
0.30	37.36	74.72	3.01
0.40	48.28	96.56	3.89
0.50	58.14	116.3	4.68

coupler experiences the tangent friction and the normal support force at the contact point. The structure is symmetric when the two couplers are connected. The force line is located on the inner common tangent line of the two friction circles at the two ends. The direction of the force line is parallel with the track at a small coupler yaw angle, where the arc friction is not saturated and the coupler lateral force does not occur. In the case of the large coupler yaw angle, the coupler's lateral forces are reduced because the yaw angle of the force line is less than the coupler's actual yaw angle. The maximum difference of the two angles, $\gamma_{\max}$, is proportional to the arc friction coefficient. The larger the arc friction coefficient is, the smaller the yaw angle of force line will be. The relationship between coupler's and force line's yaw angles is shown in Equation (4). The lateral relative displacement, Y_{bc}, between the two car bodies connected by a coupler is defined as the car body critical lateral relative displacement. If the lateral relative displacement of the two car bodies is less than Y_{bc}, there is no coupler lateral force. The geometry parameters of the coupler at different arc friction coefficients are shown in Table 2.

$$\alpha_{ie} \approx \alpha_i - \sin^{-1}\left(\frac{\rho}{l}\right), \quad i = 1, 2. \tag{4}$$

If the friction of arc contact reaches saturation or the yaw angle of force line is larger than the self-locking angle of follower ψ, the coupler will become susceptible to instability easily. Therefore, the selection of an appropriate friction coefficient between the slide block and follower is of importance because if it is too small, it can lead to coupler instability, and if it is too large, the follower cannot move laterally. The non-free lateral movement of the follower causes non-ideal arc surfaces contact. Table 3 shows the yaw angles of the force line at different coupler yaw angles and friction coefficients. The self-locking angle of follower ψ must be greater than the maximum force line's yaw angle. If the coupler's free angle is 6° and the arc friction coefficient is 0.2, the yaw angle of force line is 3.95 and the friction

Table 3. The yaw angles of force line at different coupler yaw angles and friction coefficients.

	$\alpha_{\max}$ (°)			
$\alpha_{\max}$ (°)	$\mu = 0.2$	$\mu = 0.4$	$\mu = 0.3$	$\mu = 0.5$
5	2.95	1.99	1.11	0.32
6	3.95	2.99	2.11	1.32
7	4.95	3.99	3.11	2.32

coefficient of the follower should be larger than 0.07.

$$\mu_f \geq \tan^{-1}\alpha_{e\,\max}. \tag{5}$$

The coupler's lateral forces are balanced by the locomotive's suspension. These lateral forces are transferred to the bogie and increase the wheel set lateral forces. Once the coupler's lateral forces cannot be balanced, the coupler instability causes an instant increase in yaw angle, which would cause derailment. So, the heavy haul trains conservatively use the coupler with restoring bumpstop to limit the maximum coupler yaw angle, but the bumpstop coupler does not have the arc contact coupler's advantages of a small lateral force.

2.3. *The mechanical characteristics of coupler with restoring bumpstop*

When the coupler yaw angle is greater than or equal to its free angle, the one-sided bumpstop experiences a larger coupler longitudinal force and the ideal arc surface contact is destroyed, as shown in Figure 4. Ignoring the arc surface's friction force, a commensurate longitudinal force applied in the coupler end is transferred to the yoke key. The end of the coupler experiences two longitudinal forces F_{X11} and F_{X12} offered by the yoke key and bumpstop, respectively. According to the principle of moment equilibrium, the two forces of F_{X11} and F_{X12} at the rear end are equivalent to a force $F_{cx1}(= F_{X11} + F_{X12})$ in the same direction, where the distance to the yoke key is a_1. The distance a_1 is related to the ratio of the two forces as shown in Equation (5) and its maximum value is the width of shoulder b. Likewise, the two forces of coupler's front end are equivalent to a force F_{cx2} with the distance to the yoke key of a_2. Figure 4 shows the force states of the coupler with shoulder contact in both ends. The action points of F_{cx1} and F_{cx2} at both ends form a line to transfer the coupler force which is known as the force line. The yaw angles of the force line relative to the car body at the two ends are α_{1e} and α_{2e}, respectively, and can be obtained by Equation (6) under the assumption of a small angle. With the same function as the arc surface contact, the reduced yaw angle of the force line decreases the coupler lateral force.

$$a_i = \frac{F_{Xi2}}{F_{Xi1} + F_{Xi2}} b, \quad i = 1, 2, \tag{6}$$

$$\alpha_{ie} \approx \alpha_i - \tan^{-1}\left(\frac{a_1 + a_2}{2l}\right), \quad i = 1, 2. \tag{7}$$

The front plate experiences two-sided distributed force and it is in a balanced state when the coupler is compressed. Because the slide block has only a freedom of longitudinal movement, the distributed force on one side of the front plate will decrease if the front plate tends to rotate. When the bumpstop force increases, the front plate tends to rotate, and the force analysis of critical rotate state is shown in Figure 5. The critical bumpstop force F_{X2} is calculated by Equation (7) and its value is $0.4F_{cx}$ in the case that the lateral distance from bumpstop

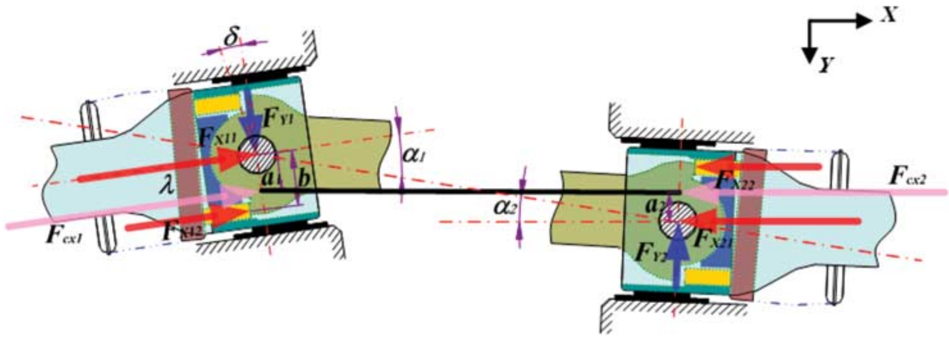

Figure 4. The geometry and force analysis of the coupler.

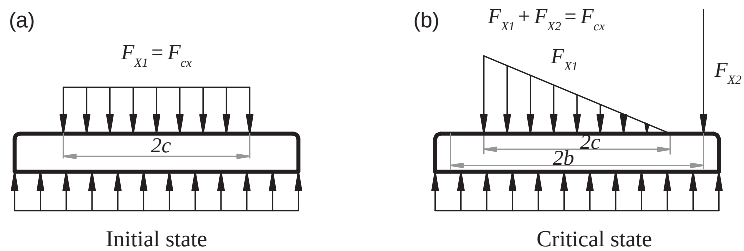

Figure 5. The force analysis of front plate in critical rotate state.

to yoke key b is equal to half of the width of slide block c. This means that only when the bumpstop force is larger than the preload of $0.4F_{cx}$ can the front plate rotate and the coupler's yaw angle increase. Within a larger range of coupler longitudinal forces, the front plate cannot be rotated and the coupler's yaw angle equals the free angle. In this case, the coupler lateral force is calculated by Equation (8) for straight track. It is constant and has nothing to do with the coupler longitudinal force. Reducing the coupler free angle and the locomotive's secondary lateral stiffness tends to reduce the coupler lateral force. The preload of bumpstop is an important factor for limiting the coupler yaw angle. In order to achieve this function, the follower cannot be rotated. This is the reason for setting the slide block with a single freedom of longitudinal movement. Once the front plate rotates, the restoring torque from the rubber buffer prevents increases in the coupler yaw angle. The bend stiffness of the rubber buffer needs to be increased non-linearly with the increase in coupler compressive force. According to the relationship between the bend and longitudinal stiffness of the rubber buffer, when the coupler longitudinal force reaches 2500 kN, the longitudinal stiffness of the rubber buffer should be greater than 100 kN/mm.

$$F_{X2} = \frac{4}{3b/c + 7} F_{cx} \approx \frac{2}{5} F_{cx}, \tag{8}$$

$$F_{cx} = k_{cy} l \cos\alpha, \tag{9}$$

where $k_{cy} = (l_b/l_c)^2 k_s$, k_{cy} is the lateral equivalent stiffness of the coupler mounting seat; k_s is the lateral suspension stiffness of a single bogie which is a series stiffness of the primary and secondary suspension; l_b is half of the bogie longitudinal distance of the locomotive; l_c is half of the longitudinal distance of the coupler mounting seats; and l is half of the coupler length.

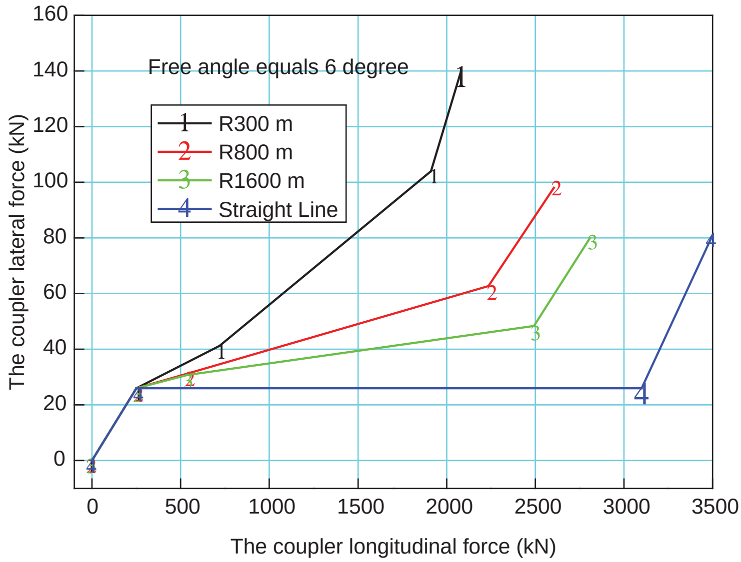

Figure 6. The coupler lateral forces at different tracks.

The restoring torque supplied by the bumpstop is equivalent to the offset of the action points of the coupler force which reduces the yaw angle of the force line for decreasing the coupler lateral force. The coupler lateral force increases with the decrease in curve radius. As shown in Figure 6, the coupler lateral force increases with the longitudinal force in curves. However, the lateral force is a constant within a range of the longitudinal force in a straight line.

2.4. *The geometry and stability analysis of coupler during train braking*

When the locomotive passes the curves, there is a yaw angle α_1 (α_2) between the coupler and the front or rear car body, both of which will be increased by the coupler longitudinal compressive force as shown in Figure 7. The small angles of η_1 and η_2 are the yaw angles of the front and rear car body which can be ignored in the geometry analysis of the coupler. ϕ is the coupler yaw angle caused by the curve. φ is the yaw angle caused by the coupler longitudinal compressive force, and α_1 and α_2 are the yaw angles of the coupler relative to the rear and front car body, respectively. The angles of α_1, α_2, ϕ, and φ have the following relationship:

$$\begin{aligned}\alpha_1 &= \varphi + \phi,\\ \alpha_2 &= \varphi - \phi.\end{aligned} \tag{10}$$

It is equivalent to

$$\begin{aligned}\alpha_1 &= \alpha_2 + 2\phi,\\ \varphi &= \tfrac{1}{2}(\alpha_1 + \alpha_2).\end{aligned} \tag{11}$$

The angle ϕ is dependent on the longitudinal distance of the coupler mounting seat $2l_c$, the length of coupler $2l$, and the curve radius R_t.

$$\beta \approx \sin^{-1}\left(\frac{l_c + l}{R_t}\right). \tag{12}$$

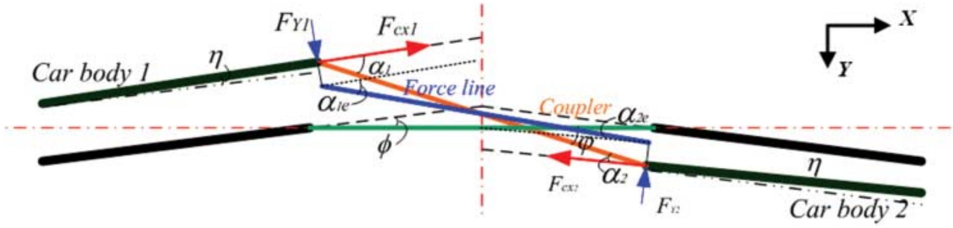

Figure 7. The yaw angle of coupler in curves.

In this study, when R_t is 300, 800, and 1600 m, ϕ is 1.8°, 0.68°, and 0.34°, respectively. Since ϕ is positive, the yaw angle α_1 relative to the rear car body is greater than α_2 relative to the front car body. This will cause the coupler to rotate towards the direction of the curve extending under the compressive force. This is especially true on curves with small or medium radii.

According to the stability mechanism and force analysis of the two types of couplers, the reduced yaw angle of actual transfer force line helps to reduce the coupler lateral force. The yaw angles of the two coupler's force line can be calculated by Equations (4) and (6), respectively. In order to balance or stabilise a coupler under a huge compressive force, the sum of the coupler lateral forces at the two ends must be less than or equal to the restoring force caused by the locomotive's lateral suspension.

$$F_{Y1} + F_{Y2} = F_{cx} \cdot (\sin \alpha_{1e} + \sin \alpha_{2e}) \leq k_{cy} \cdot 2l \sin \varphi. \tag{13}$$

2.5. *Summary*

(1) The arc contact coupler adopts the method of reducing the yaw angle of the actual transfer force line to reduce the coupler lateral force. The arc contact coupler has an advantage of small lateral force and self-stabilisation at a small yaw angle. Increasing the locomotive's secondary suspension stiffness and the coupler friction coefficient is an effective measure to improve the bearing capacity of this type of coupler. The bumpstop coupler adopts the bumpstops to prevent the yaw angle increase. The coupler's yaw angle easily increases to the free angle where the bumpstop contacts when the coupler is compressed. The preload of the bumpstop force needs to be overcome, which is an important factor for limiting the coupler yaw angle. This way the coupler has an advantage to ensure the safety of a locomotive derailed by a huge longitudinal compressive force.

(2) An innovative structure for a heavy haul coupler was proposed in this paper, in which the advantages of the two proposed couplers are combined. The main structural characteristic of the combined contact coupler is a lateral movable follower with an appropriate friction coefficient and a slide block with a single freedom of longitudinal motion. The lateral freedom of the follower ensures ideal movement of the arc's contact points in the coupler's end, and the single longitudinal freedom of the slide block ensures the preload of bumpstop to prevent the increase in coupler yaw angle. The combined contact coupler has a small lateral force in the condition of normal train emergency braking and a limitable coupler yaw angle to ensure an allowable coupler lateral force under a huge coupler compressive force.

3. Multi-locomotive dynamic simulation model

In order to verify the theoretical analysis and accurately calculate the locomotive's wheel set lateral forces with different coupler parameters, the multi-body dynamics software, SIMPACK, was used to build the locomotive and the detailed coupler model. The simple train model composed of four locomotives and three detailed couplers can simulate the processes of emergency braking and coupler overturn with appropriate boundary conditions. The main impact factors of the coupler's bearing capacity and the effect of the suspension parameters of the locomotive on the wheel set lateral forces have been analysed.

3.1. *The locomotive dynamics model*

This locomotive model was established according to an actual heavy haul locomotive in China, as shown in Table 4. The model consists of four locomotives and three couplers as shown in Figure 8. The longitudinal forces F_1 and F_2 are applied to both ends of the locomotives separately, and the braking torques are applied to each wheel set to simulate the longitudinal impulse caused by emergency braking. These studies focus on the middle two locomotives.

Table 4. The main parameters of locomotives.

Axle arrangement	Axle load (T)	Wheel base (m)	Bogie distance (m)	Coupler seats distance (m)
$2B_0$	27	2.6	10.15	17.652
Axle guidance stiffness (signal axlebox) (kN·mm^{-1})			Secondary suspension stiffness (One side in a bogie) (kN·mm^{-1})	
Longitudinal	Lateral	Vertical	Lateral	Vertical
162.6	4.06	1.317	0.58	14

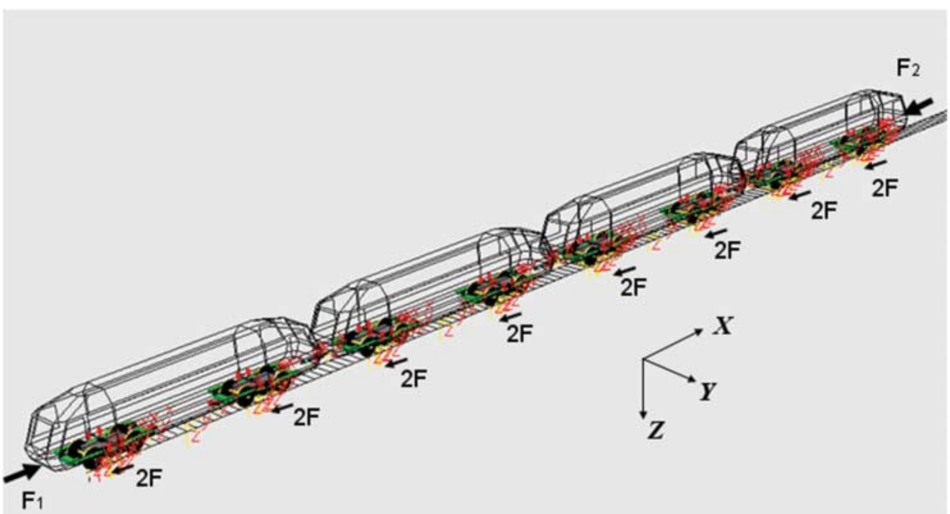

Figure 8. The simple train model composed of four locomotives and three couplers.

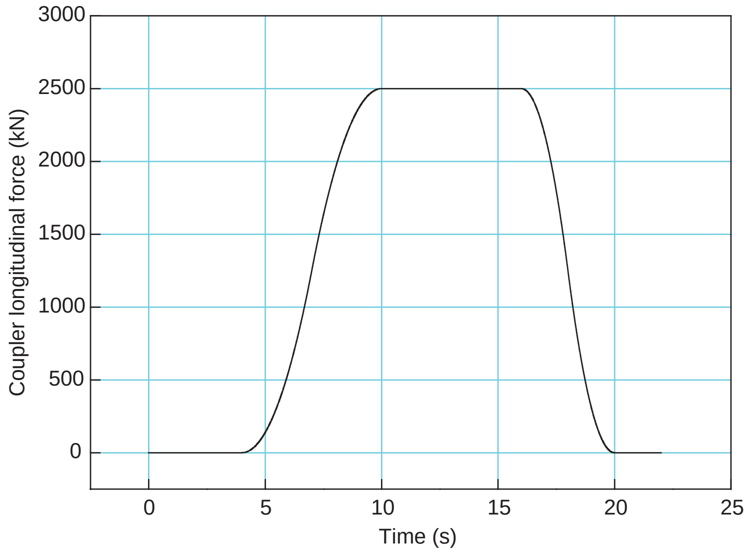

Figure 9. The coupler longitudinal force curve in the time domain.

3.2. *The coupler dynamics model*

The multi-body dynamics model of the coupler includes 9 rigid bodies and 14 degrees of freedom. The follower has the freedom to move longitudinally and laterally. The front plate has the freedom to yaw, and move longitudinally. The coupler yoke has the freedom to move longitudinally, while the coupler can move longitudinally, pitch, and yaw. The bumpstop is modelled by a nonlinear unilateral contact force element with clearance. Similarly, the contact between the follower and the front plate is modelled by the unilateral contact force element.

3.3. *The boundary conditions and calculation conditions*

To simulate the locomotives located in the middle of a 20,000-ton heavy haul train in a straight line, the following boundary conditions were applied. The maximum coupler longitudinal force was set at 2500 kN. The initial speed before braking was 80 km/h. Additionally, the braking force for a single locomotive was set at 255 kN. The track irregularity of AAR5 is used. The coupler force curve in the time domain is shown in Figure 9.

4. The simulation results of the locomotive dynamics

4.1. *The wheel set lateral forces in the straight line*

Larger coupler longitudinal compressive forces which can lead to the derailment of locomotives, especially in the middle of the train, usually occur when a train is emergency braking. This braking scenario has been known to cause several derailment accidents in a straight line. Therefore, there is more concern about the performance of couplers in a straight line. According to the dynamics model and the boundary conditions in Section 3, the wheel set lateral forces of locomotives were simulated at different arc surface friction coefficients. Figures 10 and 11 show the coupler yaw angles and the wheel set lateral forces of locomotives at the coupler longitudinal force of 2500 kN. When the friction coefficient is greater than 0.4, the coupler can be stabilised by the locomotive's lateral suspension through the arc surface contact. Once the coupler cannot be stabilised, the coupler yaw angle suddenly increases to the free angle. There is a distinct impact on the follower at the moment of coupler overturning, which can lead to an instant overshoot of the coupler yaw angle. As shown in Figure 11, the maximum wheel set lateral forces occur at the moment of coupler overturning due to the

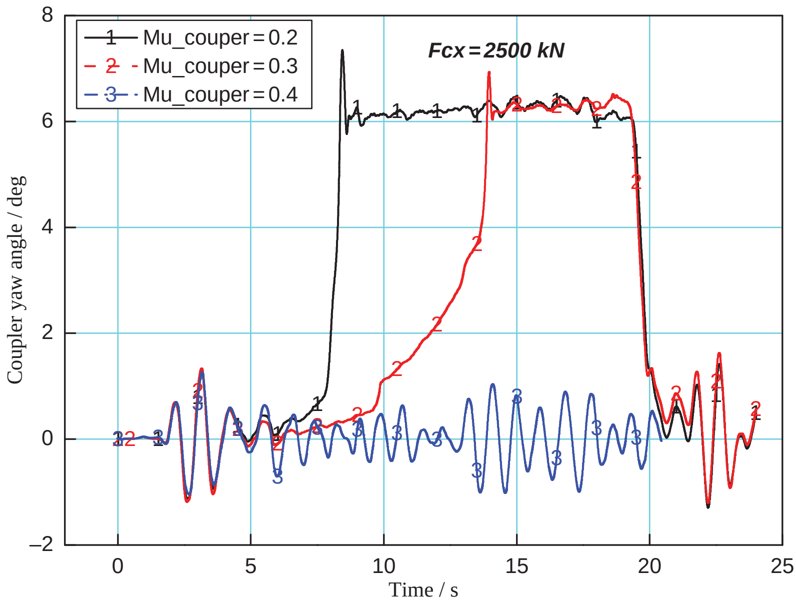

Figure 10. The coupler yaw angle at different arc surface's friction coefficients.

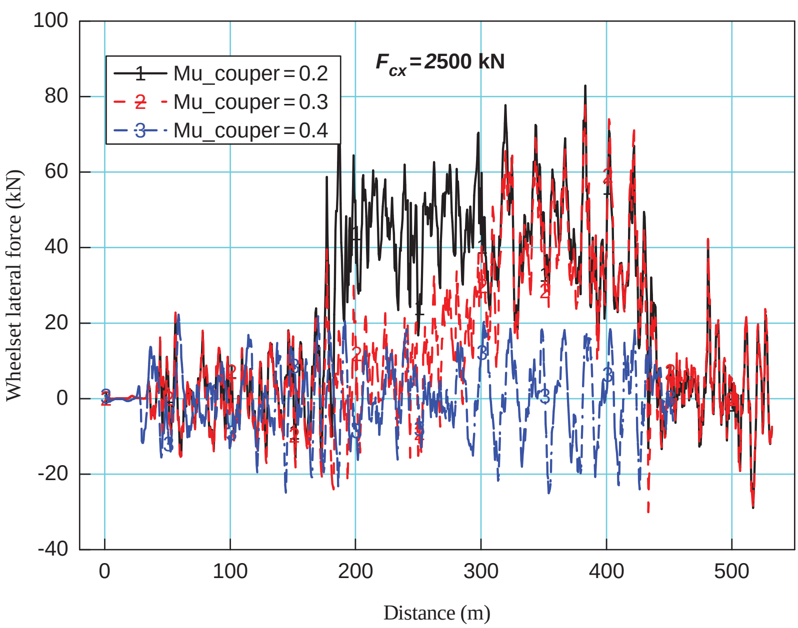

Figure 11. The wheel set lateral force at different arc surface's friction coefficients.

overshoot of the yaw angle. When the coupler longitudinal force or free angle increases, the coupler yaw angle cannot be limited at the free angle because the bumpstop force is greater than the preload of 0.4 times the coupler longitudinal force, and the front plate rotates. At that moment, a large bend stiffness of the rubber buffer is required to limit the coupler yaw angle.

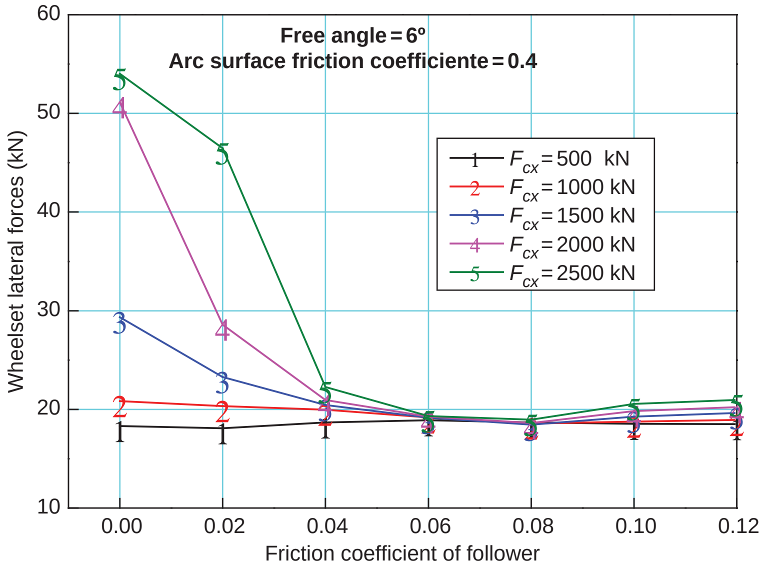

Figure 12. The wheel set lateral forces at different coupler forces and follower's friction coefficients.

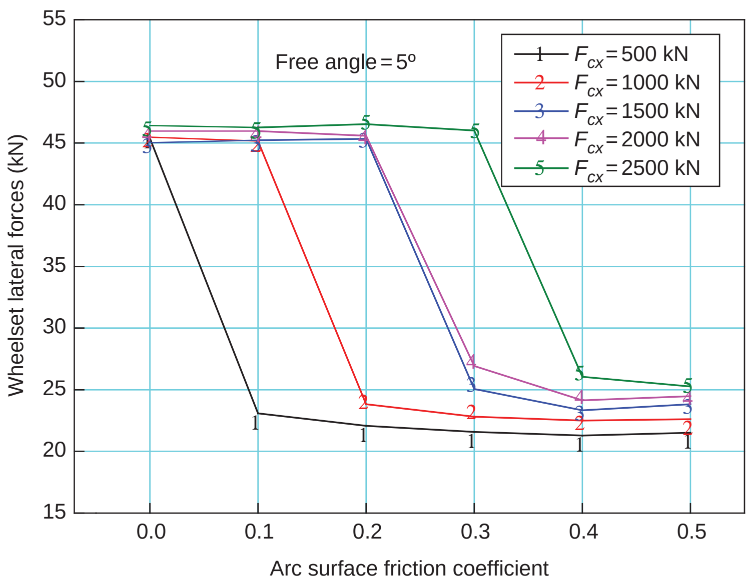

Figure 13. The wheel set lateral forces with different coupler forces at the free angle of 5° .

4.2. *The effect of the follower's friction coefficient on the wheel set lateral forces*

According to force analysis, the selection of an appropriate follower friction coefficient is of importance since one that is too small may lead to coupler instability, and one that is too larger does not allow the follower to move laterally. The self-locking angle of the follower must be greater than the maximum force line's yaw angle. The friction coefficient should satisfy Equation (5). Figure 12 shows the simulation results of the wheel set lateral forces at different coupler forces and follower's friction coefficient. The optimal value of 0.06–0.08 is consistent with the theoretical analysis. When the coupler can be stabilised by the arc contacts, the effect of coupler longitudinal forces on the locomotive's wheel set lateral forces is not obvious.

4.3. *The effect of the free angle and arc surface's friction coefficient on the wheel set lateral forces*

The arc surface the friction coefficient and the free angle of the two types of couplers are important parameters. The former affects the coupler stability and the latter affects the lateral force. Figures 13–15 show the simulation results of the effect of the coupler free angle and arc surface's friction coefficient on the wheel set lateral forces. When the maximum coupler longitudinal forces are 500, 1000, less than 2000, and 2500 kN, the friction coefficient of 0.1, 0.2, 0.3, and 0.4 can stabilise the coupler through the arc surface contact. The wheel set lateral force is small and it is proportional to the coupler longitudinal force if the coupler is stabilised, which cannot lead to abnormal wear of wheel and rail. The coupler yaw angle is limited by bumpstop once the coupler is unstable. The coupler yaw angle is greater than the free angle at a larger longitudinal force or free angle. The simulation results verify the theoretical analysis in Section 2.3, in which the wheel set lateral force is a constant and has nothing to do with the coupler longitudinal force if the coupler yaw angle is equal to the free angle. The wheel set lateral forces increase with the free angle at the same coupler longitudinal force. Once

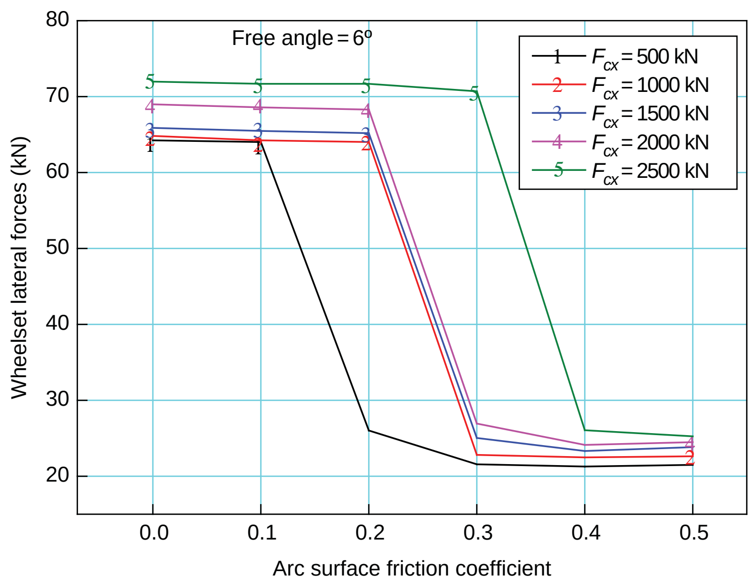

Figure 14. The wheel set lateral forces with different coupler forces at the free angle of 6° .

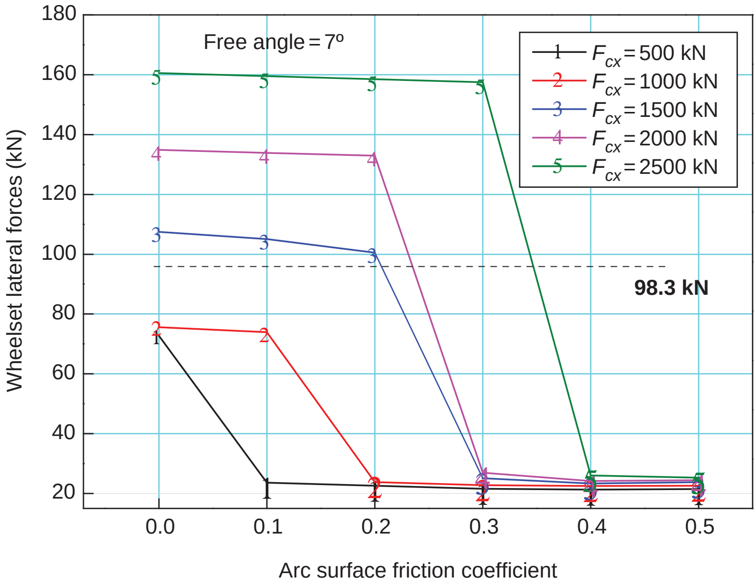

Figure 15. The wheel set lateral forces with different coupler forces at the free angle of 7°.

the follower rotates under a sufficient free angle and coupler longitudinal force, the wheel set lateral forces increase as well. When the coupler free angle is less than 6°, the wheel set lateral forces of the locomotive under the maximal coupler longitudinal force of 2500 kN are less than the allowable value of 98.3 kN calculated with the British standard of EN 14363. However, the wheel set lateral forces cannot meet the required maximal coupler force once the free angle is larger than 7°. It is recommended that the arc surface's friction coefficient is kept greater than 0.4 and the free angle less than 6° under the potential maximum coupler longitudinal compressive force of 2500 kN.

5. Conclusions

(1) An innovative structure of a heavy haul coupler was proposed in this paper, in which the advantages of the two types of couplers are combined. The main structural characteristic of the combined contact coupler is a lateral movable follower with an appropriate friction coefficient and a slide block with a single freedom of longitudinal motion. The lateral freedom of the follower ensures the ideal movement of arc's contact points in the coupler's end, and the single longitudinal freedom of the slide block ensures the preload on the bumpstop to prevent the increase in the coupler yaw angle. The combined contact coupler has a small lateral force in the condition of normal train emergency braking and a limitable yaw angle to ensure an allowable coupler lateral force under a large coupler compressive force.

(2) The combined contact coupler exhibits self-stabilisation tendencies at a small yaw angle. The yaw angle of the force line is less than the coupler's actual yaw angle, which helps to reduce the coupler lateral force. The friction coefficient of the arc contact surface needs to be increased as the longitudinal compressive force of the coupler increases. When the maximum coupler longitudinal forces are 500, 1000, less than 2000, and 2500 kN, the friction coefficient of 0.1, 0.2, 0.3, and 0.4 can stabilise the coupler through the arc

surface contact. If the coupler is stable, the wheel set lateral force is small, which prevents abnormal wear of wheels and rails.

(3) The combined contact coupler adopts the bumpstops to prevent the yaw angle increase once the coupler is unstable. The bumpstop force increases with the increase in the longitudinal force and the coupler yaw angle. Only when the bumpstop force exceeds the preload of 0.4 times the coupler longitudinal force can the front plate be rotated and the coupler yaw angle increased continually. The wheel set lateral force is a constant and has nothing to do with the coupler longitudinal force if the coupler yaw angle is equal to the free angle. The wheel set lateral forces increase with the free angle at the same coupler longitudinal force. When the coupler free angle is less than 6°, the wheel set lateral forces of the locomotive under the maximal coupler's longitudinal force of 2500 kN are less than the allowable value.

Acknowledgements

The authors acknowledge the support of the National Natural Science Foundation of China (No. 51205324, 51375403) and the Fundamental Research Funds for the Central Universities (No. 2682014CX040).

References

[1] European Institute for Railway Research. Etude del probabilities de derailment de trains de marchandises sous l'influence d'efforts longitudinaux de compression eleves; 1999 (ERRI B177.5/RP1).

[2] Transportation Safety Board of Canada. Railway investigation report R02C0050. Ottawa: Transportation Safety Board of Canada; 2002. Derailment, Canadian National Train Number A-442-51-08 Mile 52.1.

[3] Belforte P, Cheli F, Diana G. Numerical and experimental approach for the evaluation of severe longitudinal dynamics of heavy freight trains. Veh Syst Dyn. 2008;46:937–955.

[4] Durali M, Shadmehri B. Nonlinear analysis of train derailment in severe braking. J Dyn Syst Meas Control, ASME. 2003;125:48–53.

[5] Geike T. Understanding high coupler forces at metro vehicles. Veh Syst Dyn. 2007;45:389–396.

[6] Pugi L, Rindi A, Ercole AG, Palazzolo A, Auciello J, Fioravanti D, Ignesti M. Preliminary studies concerning the application of different braking arrangements on Italian freight trains. Veh Syst Dyn. 2011;49:1339–1365.

[7] Pugi L, Fioravanti D, Rindi A. Modeling the longitudinal dynamics of long freight trains during the braking phase. 12th IFTOMMWorld Congress; 2007 June 18–21, p. 1–6; Besançon, France.

[8] Geike T. Understanding high coupler forces at metro vehicles. Veh Syst Dyn. 2007;45:389–396.

[9] Cole C, Mitchell M, Dudley R, Gerhard L, Fanus, W, Ted, M. Heavy haul coal train dynamics simulation comparisons of the COALlink and central Queensland systems. Proceedings of the 7th International Heavy Haul Conference; 2001 June 10–14, p. 217–223; Brisbane, Australia.

[10] Simson SA, Cole C. Simulation of curving at low speed under high traction for passive steering hauling locomotives. Veh Syst Dyn. 2008;46:1107–1121.

[11] Cole C, Sun YQ. Simulated comparisons of wagon coupler systems in heavy haul trains. Proc IMechE, Part F: J Rail Rapid Transit. 2006;220:247–256.

[12] Xu ZQ, Ma WH, Wu Q. Coupler rotation behaviour and its effect on heavy haul trains. Veh Syst Dyn. 2013;51:1818–1838.

[13] Wu Q, Luo SH, Xu ZQ. Coupler jackknifing and derailments of locomotives on tangent track. Veh Syst Dyn. 2013;51:1784–1800.

[14] Wu Q, Luo SH, Wei CF, Ma WH. Dynamics simulation models of coupler systems for freight locomotive. J Traffic Transp Eng. 2012;12:37–43.

[15] Yao Y, Zhang XX, Zhang HJ, Luo SH. The stability mechanism and its application of heavy haul couplers with arc surface contact. Veh Syst Dyn. 2013;51:1324–1341.

[16] Ma WH, Luo SH, Song RR. Coupler dynamic performance analysis of heavy haul locomotive. Veh. Syst. Dyn. 2012;50:1435–1452.

[17] Yao Y, Liu X, Zhang HJ, Luo SH. The stability and mechanical characteristics of heavy haul couplers with restoring bumpstop. Veh. Syst. Dyn. 2014;52:26–44.