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THE EFFECTS OF THE FRICTION BETWEEN FRAME AND ADAPTERS ON THE PERFORMANCE OF THREE-PIECE TRUCK

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ABSTRACT

For the traditional three-piece truck there is no primary suspension and the connection between side frame and adapter (wheelset) is via direct contact with two-dimensional dry friction on the contact surfaces. The dry friction has two modes: stick and slip motion. Knowledge of the friction on the adapter surface is useful as it affects the wagon dynamic performance on both straight and curved track. In this paper we developed a friction element which can be used to describe two-dimensional dry friction with stick-slip modes. Static and kinematic friction coefficients can be used. The successfully implemented this new friction element in a complete wagon model. The model also includes two-dimensional friction on the surfaces of friction wedge dampers in the secondary suspension. Various friction coefficients are selected for the side frame/adapter contact and the dynamic performance of wagon on straight and curved track is investigated.

1 INTRODUCTION

Three-piece truck is widely used in America, Russian, South Africa, China and Australia due to its very simple structure and cheaper price. The typical structure of the traditional three-piece truck is shown in Figure 1.

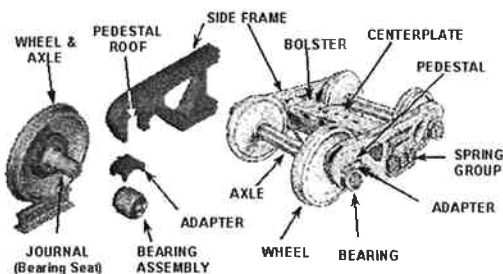


Figure 1. Scheme of three-piece truck

The significant structural characteristics of this kind of truck are that there is no the primary suspension and the connection between side frames and adapters (wheelsets) is via direct contact with two-dimensional dry friction on the contact surfaces; the vertical responses of wagon and bogie warping are damped out by the friction on the surfaces of wedges which are sandwiched between bolster and side frame. There are two types of wedge damper: variable friction force and constant friction force. The first one is mainly used in America and other countries but in Australia the constant friction (it also called ride control) is most widely used. The clearances between a side frame and adapter both in longitudinal and lateral directions are designed to give the truck warp flexibility for curving. As the wagon negotiates a curve the clearances therefore provide wheelset a chance to form a yaw angle for self steering through the curve. When on straight track the friction forces act against the relative displacements between side frame and wheelset both in longitudinal and lateral directions to provide damping of lateral oscillations and hunting motion.

As the contact between side frame and adapter is plane friction the case of two-dimensional dry friction must be analysed. When the external force is less than the friction force there is no relative motion between frame and adapter and vice versa. So there exist two modes: stick and slip motion. The knowledge of the friction on the adapter surface is useful as it affects the wagon dynamic performance on both straight and curved track. A study of this behaviour is important to provide a theoretical background for the optimal truck design.

From mathematical point of view, it is a discontinuous model for the motion from stick to slip [1]. To describe dry friction there are two techniques: direct and indirect methods [2]. Applying the direct method to a complex system such as the railroad wagon system, with dry friction, especially two-

dimensional dry friction is very complicated. The indirect method is where the friction element is developed to approximately deal with the problem, as done in [2][3][4]. In the present paper the indirect method is used and a modified friction element is developed based on the one commonly used in VAMPIRE. The friction element developed is used to describe two-dimensional dry friction with stick-slip modes and static/kinematic friction coefficients.

The wagon model consists of 11 rigid bodies: 1 wagon body, 2 bolsters, 4 side frames and 4 wheelsets, and total system has 66 degrees of freedom. The secondary suspension damping is modelled as the constant force wedge damper system which is widely used in Australia. The model also includes two-dimensional friction on the surfaces of wedges. The research focus in this paper is on the effect of side frame-adapter contact friction on the performance of wagon on straight and curved track.

Questions that can be raised when modelling the contact between side frame and adapter include:

- Can the friction provide enough force to keep the truck stable when truck on straight track?
- During wagon negotiating over curved track, will stick-slip motion between side frame and adapter take place?
- Are the design clearances between the side frame and the adapter in longitudinal and lateral adequate. Should these clearances be larger?

This paper tries to analyse these issues through a detailed wagon model developed in project work for the Australian Rail CRC. This wagon model is designated the C66 model.

Various friction coefficients are selected for the side frame/adapter contact and for the wheel/rail contact and the dynamic performance of the wagon on straight track and curved track is investigated.

2 FRICTION ELEMENT DESCRIPTION

If a mechanical system has some connections which transmit force via friction alone, then the degrees of freedom of the system will vary with the stick and slip of the friction modes. To describe the resulting structure varying system the key point is to determine the friction force both in stick and slip states. When using the direct method, switch conditions are used to determine the motion states. Conversely, when using the indirect method the friction force is approximated using a Coulomb friction element in series with a spring. The friction force in stick state is approximated by the spring force.

The friction element used in Vampire is described as shown in Figure 2 [3][4]. The total force is

$$F_{sc} = F_{ks} + F_{cs}, \quad (1)$$

and the breakout force is

$$F_B = \mu F_n, \quad (2)$$

Finally the effective friction can be expressed as

$$F_f = \begin{cases} F_{sc}, & F_{sc} \leq F_B \\ F_B, & F_{sc} > F_B \end{cases}, \quad (3)$$

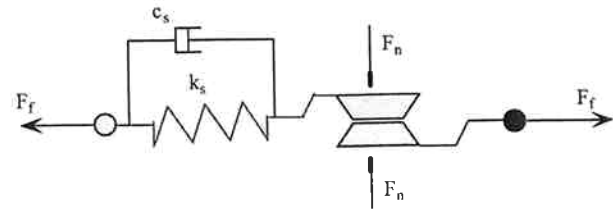


Figure 2. Friction element used in Vampire

It is known that static and kinetic friction coefficients are different, static friction coefficient is normally larger than kinetic friction coefficient. If the friction element in VAMPIRE is used it seems that the kinetic friction force is replaced by breakout force (static friction force). To explain it in another way, the difference between static and kinematic friction is ignored. Furthermore, when the relative velocity is larger than zero (or a certain small value) then the friction force is determined by kinetic friction force. So the friction element in VAMPIRE cannot accurately model static/kinematic friction.

The second friction element was originally developed by Kolsch and later implemented into the ADAMS/Rail [2][5]. The principle of this method is described in Figure 3. The force between the body 1 and body 2 is defined as:

$$F_f = K_0 \Delta d + F_{k1}, \quad (4)$$

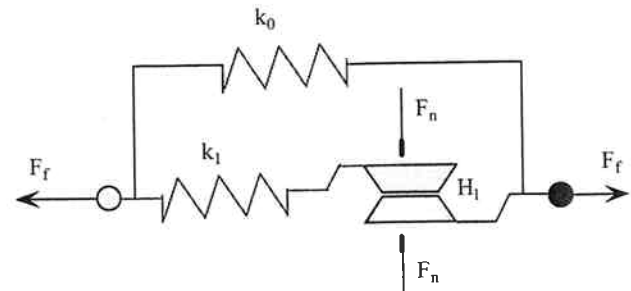


Figure 3. Friction element used in ADAMS/Rail

where Δd denotes the relative displacement between the two bodies and F_{k1} is defined as:

$$\dot{F}_{k1} = K_1 \Delta V \left\{ 1 - 0.5(1 + \text{sign}(\Delta V \times F_{k1})) \left| \frac{F_{k1}}{H_1} \right|^m \right\}, \quad (5)$$

where ΔV denotes the relative velocity between the two bodies; H_1 denotes the maximum static friction force and m is the exponent of the transition. This method is more

accurate than the first one for the friction force calculation during stick mode but it cannot be used to describe the case where there is only dry friction connection between two relative movable bodies. The obvious reason is that the parallel stiffness k_0 limits the free relative motion between the two contacting bodies in the x direction.

For the direct method we take the system shown in Figure 4 as an example, for the slip motion the friction force is determined by

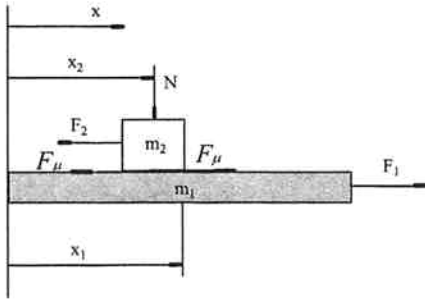


Figure 4. A 2DOFs system with dry friction

$$F_{\mu k} = N\mu_k \text{sign}(\dot{x}_1 - \dot{x}_2). \quad (6)$$

In this case the system has two degrees of freedom. For the stick motion, that is, when the relative velocity between m_1 and m_2 is zero, the friction force is then determined by

$$F_{\mu s} = \frac{1}{m_1 + m_2} [m_2 F_1 + m_1 F_2]. \quad (7)$$

The system has one degree of freedom during the stick mode. The switch conditions that control the motion from stick to slip are

$$\begin{aligned} \dot{x}_1 - \dot{x}_2 &= 0 \\ (m_2 F_1 + m_1 F_2) &\leq (m_1 + m_2) N \mu_s. \end{aligned} \quad (8)$$

If the above conditions are true then the system is stuck, and vice versa.

This method is exact mathematically, but for a wagon system, it will give very complicated system equations [1]. In the present paper we take the method used in Vampire as a starting point, to develop another two-dimensional friction element. The principle can be shortly expressed as (see figure 2):

$$F_f = \begin{cases} k_s \Delta d, & k_s \Delta d \leq F_{fs} \ \& \ |v_r| \leq \Delta v_r, \\ F_{fs}, & k_s \Delta d > F_{fs} \ \& \ |v_r| \leq \Delta v_r, \\ F_{fk}, & |v_r| > \Delta v_r. \end{cases} \quad (9)$$

Where F_{fs} stands for the static friction force; F_{fk} is the kinetic friction force; Δd stands for the relative displacement and v_r is the relative velocity and Δv_r stands for a small value of relative velocity for the numerical analysis requirement.

3. A COMPLETE WAGON MODEL

Roughly speaking, a wagon consists of 11 bodies: 1 wagon car body, 2 bolsters, 4 side frames and 4 wheelsets. One body in space has 6 degrees of freedom so there is a total of 66 degrees of freedom for a wagon system. As the connection between wagon car body and the two trucks is through two centre bowls, there are at least two constraints in the vertical direction. This kind of constraint also exists between side frames and adapters (wheelsets) as shown in Figure 5, giving at least 8 constraints in the vertical direction. If these constraints are included in our wagon model then the system can be modeled using differential algebraic equations (DAE), [6]. This method is rather complicated from the numerical point of view. As an alternative, the constraints can be replaced by spring connections with suitable stiffness as shown in Figure 6 [4]. In this way the mathematical equation of the wagon system becomes a simple system of ordinary differential equation (ODE), which is much easier to solve. In this paper we will use this technique.

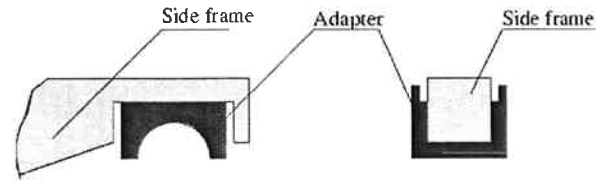


Figure 5. Contact between side frame and adapter

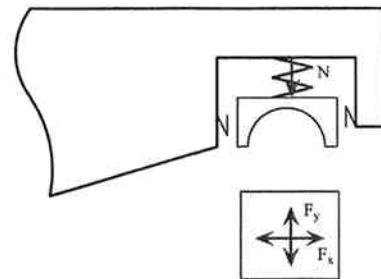


Figure 6. Replacement of direct contact with a spring

In order to analyze the motion of wagon on curved track, it is a good idea to know the relationships between the coordinate systems. To simulate the motion of wagon on both straight and curved tracks three coordinate systems are needed: inertial coordinate system, translating (moving) coordinate system and body connected system. Normally, the parasitic motions of wagon are expected and the absolute motion is not meaningful for the wagon stability and hunting motion analysis. The wagon parasitic motion is described with respect to its translating (moving) coordinate system.

For a wagon on curved track the translating coordinate systems are shown in Figure 7.

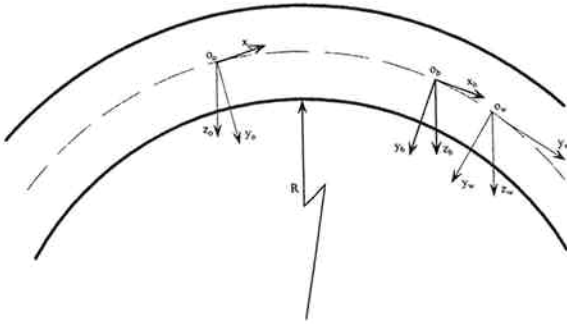


Figure 7. Translating coordinate systems on curved track

The motion of wagon body is described with respect to the moving system (O_o, x_o, y_o, z_o) and the motion of bolster is described with respect to the moving system (O_b, x_b, y_b, z_b). The complete transformation from (O_b, x_b, y_b, z_b) to (O_o, x_o, y_o, z_o) can be expressed as [7]

$$\begin{bmatrix} x_o \\ y_o \\ z_o \end{bmatrix} = \begin{bmatrix} a_b \\ \frac{1}{2} \kappa_c a_b^2 \cos \varphi_c \\ -\frac{1}{2} \kappa_c a_b^2 \sin \varphi_c \end{bmatrix} + \begin{bmatrix} 1 & -\kappa_c a_b \cos \varphi_c & \kappa_c a_b \sin \varphi_c \\ \kappa_c a_b \cos \varphi_c & 1 & -a_b (\varphi'_c - \kappa_t) \\ -\kappa_c a_b \sin \varphi_c & a_b (\varphi'_c - \kappa_t) & 1 \end{bmatrix} \begin{bmatrix} x_b \\ y_b \\ z_b \end{bmatrix} \quad (10)$$

Where a_b is the half distance between the mass centre of wagon body and the mass centre of bolster; κ_c stands for the curvature of track; φ_c is cant angle of track; κ_t is the torsion and φ'_c stands for derivative of φ_c with respect to distance.

The extra relative displacements between wagon body and bolster at center plate for wagon on curve will be

$$\begin{bmatrix} d_{xob} \\ d_{yob} \\ d_{zob} \end{bmatrix} = \begin{bmatrix} 0 \\ -\frac{1}{2} \kappa_{co} a_b^2 \cos \varphi_{co} - a_b r (\varphi'_{co} - \kappa_{to}) \\ \frac{1}{2} \kappa_{co} a_b^2 \sin \varphi_{co} \end{bmatrix}, \quad (11)$$

In which the sub-index o in κ_{co} stands for the curvature of the track at the mass center of wagon body, the same sub-index is used for the following equations, (12)

Similarly, the extra relative displacements between side frame and adapter can be written

$$\begin{bmatrix} d_{xfw} \\ d_{yfw} \\ d_{zfw} \end{bmatrix} = \begin{bmatrix} \mp b_w \kappa_{cb} a_f \cos \varphi_{cb} \\ -\frac{1}{2} \kappa_{cb} a_f^2 \cos \varphi_{cb} - a_f r (\varphi'_{cb} - \kappa_{tb}) \\ \frac{1}{2} \kappa_{cb} a_f^2 \sin \varphi_{cb} \mp a_f (\varphi'_{cb} - \kappa_{tb}) \end{bmatrix}, \quad (12)$$

Where b_w is the half distance between left and right adapters; a_f stands for wheelset half axle spacing; the sub-index b in κ_{cb} stands for the curvature of track at the mass center of bolster.

With these relationships the corresponding extra spring forces or friction forces can be determined.

The equation of the wagon system can be written as

$$[M] \ddot{X} = F_n + F_t + F_w + F_g + F_c + F_d + F_s + F_f, \quad (13)$$

The symbols in equation are below:

M :	System mass matrix
F_n :	Normal wheel rail contact force vector
F_t :	Tangential wheel rail contact force vector
F_w :	Weight vector
F_g :	Gyroscopic force vector
F_c :	Centrifugal force vector
F_d :	Damping force vector
F_s :	Spring force vector
F_f :	Friction force vector

Firstly, the kinematical wheel rail contact parameters were calculated prior to simulation by the program WRKIN to form the wheel rail contact table which includes the static wheel normal force as a function of the lateral and yaw of the wheelsets. The wheel rail contact parameter table is then used as a look-up table during the simulation. The effective normal wheel force can be determined by:

$$F_{nd} = \left(F_{n0}^{\frac{2}{3}} + K_h^{\frac{2}{3}} q_d \right)^{\frac{3}{2}} \quad (14)$$

Where q_d is the dynamic penetration and F_{n0} is the static wheel load. The dimensions of the updating contact ellipse are then given by

$$a_d = a_0 \left[\frac{F_{nd}}{F_{n0}} \right]^{\frac{1}{3}}, \quad b_d = b_0 \left[\frac{F_{nd}}{F_{n0}} \right]^{\frac{1}{3}}, \quad a_d b_d = a_0 b_0 \left[\frac{F_{nd}}{F_{n0}} \right]^{\frac{2}{3}} \quad (15)$$

As the contact dimensions are known, the tangential wheel rail contact force can be determined by Kalker theory based formulae. e.g., SHE's formulae [8].

4 CASE STUDIES

A hopper wagon is used for this investigation. Some parameters of the wagon are listed below.

Semi-spacing of truck	5.18m
Half axle spacing of wheelset	0.838 m
Lateral semi-spacing of primary suspension	0.8 m
Wheel radius	0.425 m
Semi spacing of side support	0.616 m
Truck distance	14.820 m
Wedge static friction coefficient	0.4
Static load on wedge friction surface	20.0 KN
Car body mass(empty/loaded)	8.1/66.10 Mg
Car body roll inertia(empty/loaded)	10.4/85.58 Mgm ²
Car body pitch inertia	79.3/647.18 Mgm ²
Car body yaw inertia	80.0/652.98 Mgm ²
Side frame mass	0.447 Mg
Side frame roll inertia	0.101 Mgm ²
Side frame pitch inertia	0.1156 Mgm ²
Side frame yaw inertia	0.1156 Mgm ²
Bolster mass	0.465 Mg
Bolster roll inertia	0.175 Mgm ²
Bolster pitch inertia	0.115 Mgm ²
Bolster yaw inertia	0.176 Mgm ²
Wheelset mass	1.12 Mg
Wheelset roll and yaw inertia	0.4201 Mgm ²
Wheelset pitch inertia	0.1 Mgm ²

4.1 Wagon on straight track with irregularities.

Firstly, straight track with cross level sinusoidal irregularities is evaluated for the running speed of 20 m/s. The static/kinetic friction coefficients are 0.4/0.3 for the contact between side frame and adapter. Wheel/rail contact friction coefficient is 0.3.

Figures 8 and 9 show the contact friction force on the surface of the right hand side adapter of the leading wheelset of the leading truck. In this case the frame and adapter are locked most of time because the resultant friction force is less than kinetic friction or equal static friction force in most of interval.

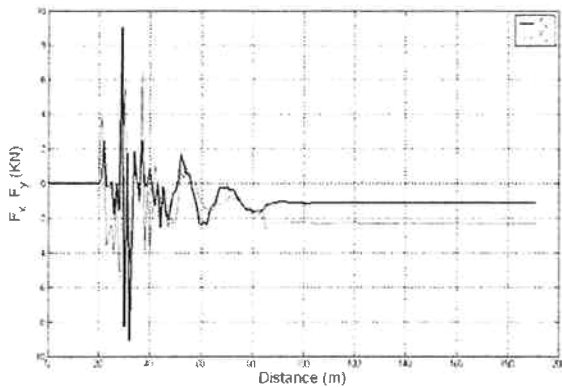


Figure 8. Longitudinal and lateral friction forces

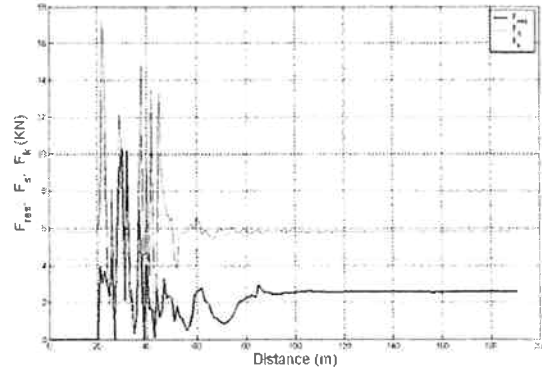


Figure 9. Resultant friction force, static and kinetic friction forces

To consider the effect of changed friction coefficients on the motion states of the frame-adapter contact, smaller friction coefficients, 0.25/0.2 are used, wheel rail friction coefficient remains 0.3. Figure 10 shows longitudinal and lateral friction force components. Figure 11 shows resultant friction, static and kinetic friction forces. In this case, the resultant friction force is almost larger than kinetic or equal static friction force in the interval of 20~110m. As a consequence, sliding motion will take place, such that relative displacement between frame and adapter is obviously larger than zero as shown in Figure 12.

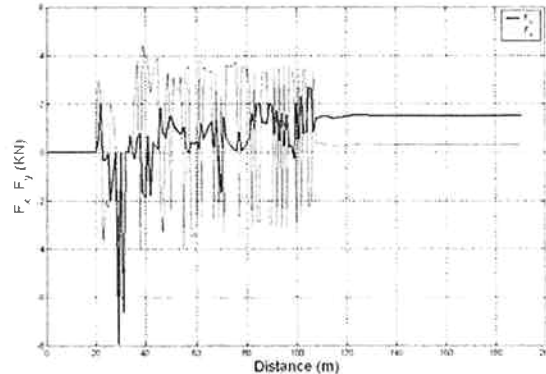


Figure 10. Longitudinal and lateral friction forces

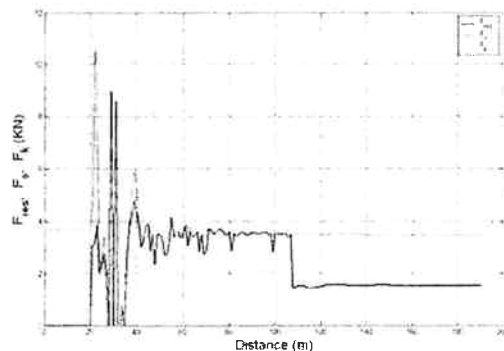


Figure 11. Resultant friction force against static and kinetic friction force

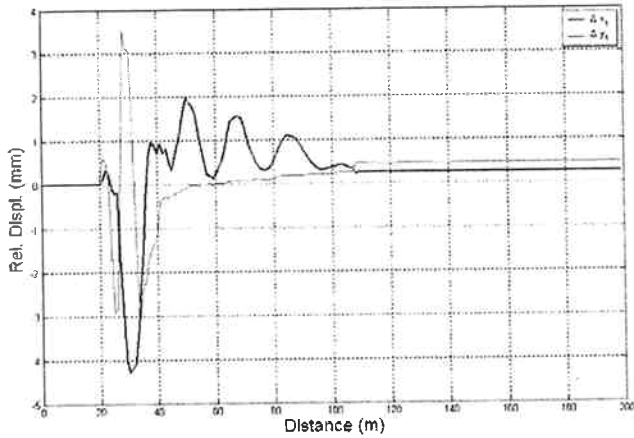


Figure 12. Relative displacement between frame and adapter of right hand side

4.1 More complex irregularity track

For the study of more complex responses track with irregularities transferred from Power Spectral Density (PSD) available from Federal Railroad Administration (FRA) are used [1]. The frequency range is from 0.05 rad/m to 2π rad/m (Wavelength from 1 m to 125 m). The static/kinetic friction coefficients are 0.4/0.3. Running speed is 20m/s. Figures 13 shows the contact friction forces components.

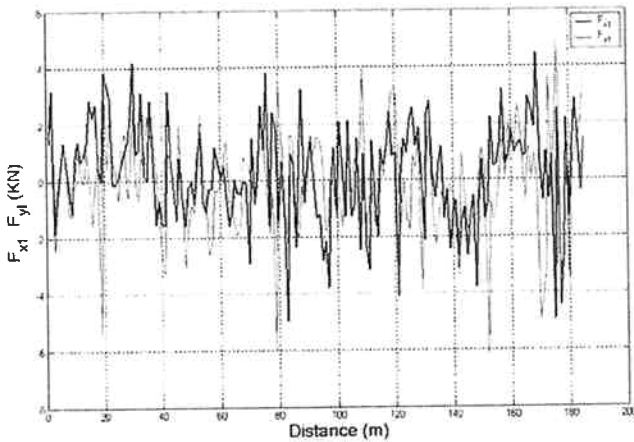


Figure 13. Longitudinal and lateral friction forces

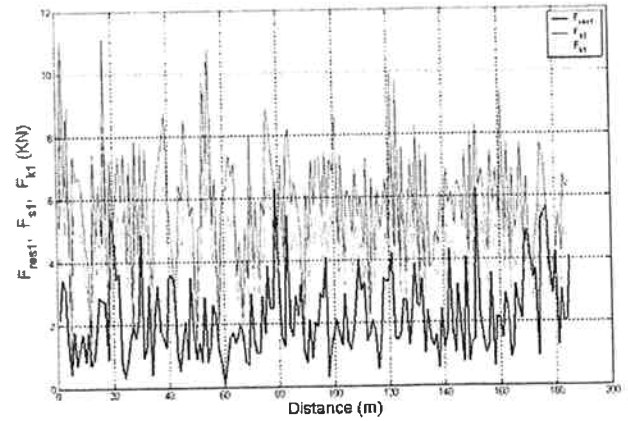


Figure 14. Resultant friction force, static and kinetic friction forces

In this case, most of resultant friction force is less than static or kinetic friction forces as shown in Figure 14. That means that the frame and adapter locks together and the relative displacements between frame and adapter are small as shown in Figure 15.

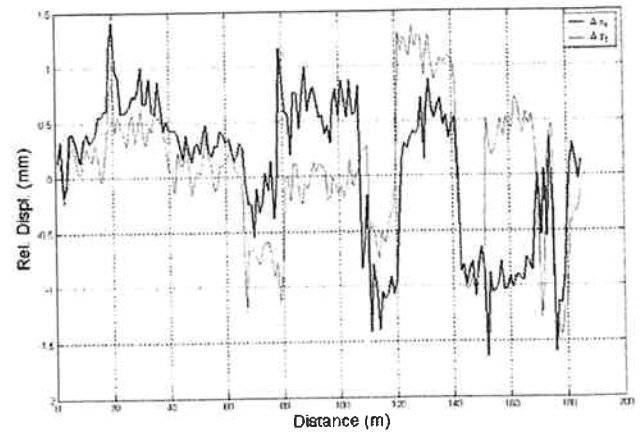


Figure 15. Relative displacement between frame and adapter of right hand side

4.3 Wagon on curved track without irregularity

The curve track parameters used are: linear transient length of 60 m. The curve radius is 500m and super elevation of outside rail is 100mm. Running speed is 20m/s. Static/kinetic friction coefficients are 0.4/0.3.

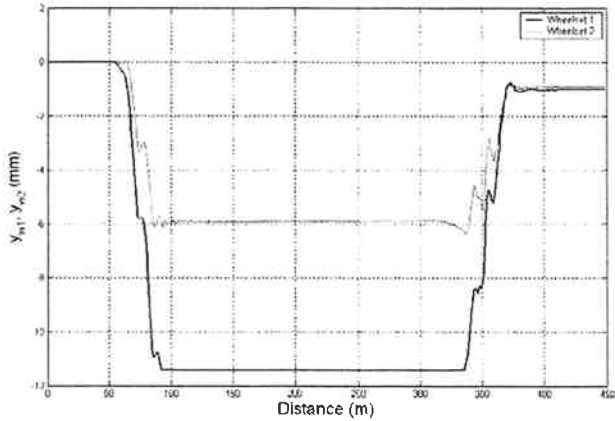


Figure 16. Lateral displacements of the first and second wheelsets on curve, C66 model.

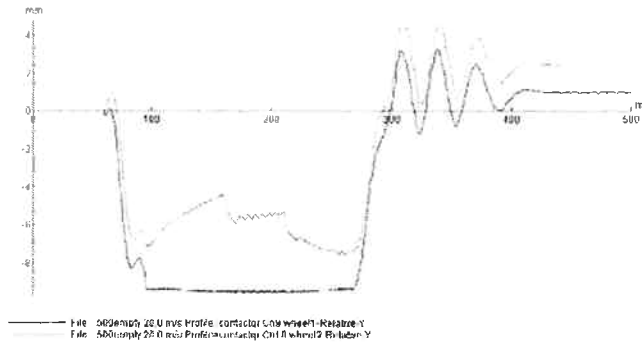


Figure 17. Lateral displacements of the first and second wheelsets on curve, VAMPIRE model.

In the present paper we don't want give detail verification of the C66 model but here just show few comparisons of results from the C66 and VAMPIRE models. Figures 16 and 17 show the comparison of the wheelset displacements and Figures 18 and 19 show the comparison of the friction force components.

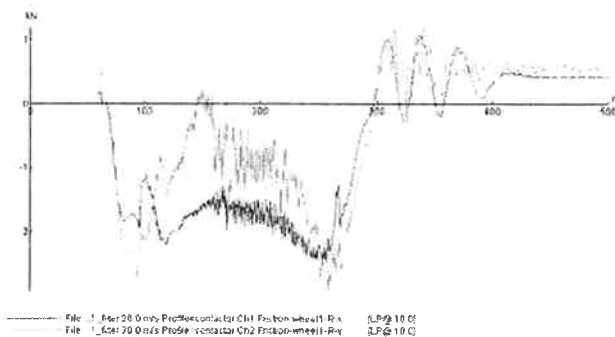


Figure 18. Longitudinal and lateral friction forces on wheelset at right hand side from VAMPIRE model

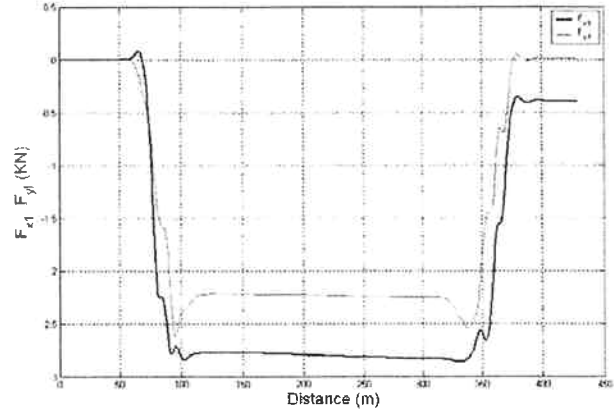


Figure 19. Longitudinal and lateral friction forces on wheelset at right hand side

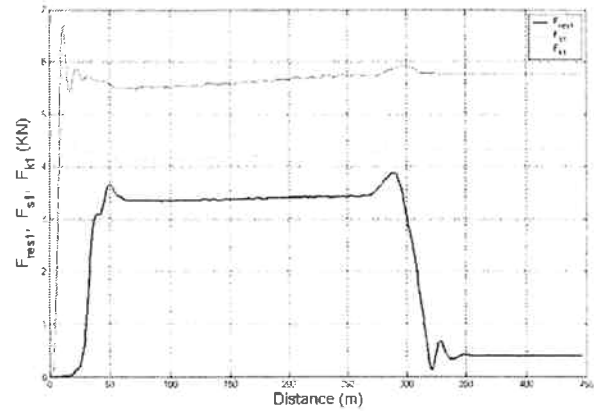


Figure 20. Resultant friction force, static and kinetic friction forces at right hand side

From Figure 20 we can see that in this case there is no relative motion between frame and adapter. The connection is locked or stuck because the resultant friction force is always less than even the kinetic friction force. It should be noted that if wagon tracks over a certain small radius curve or if there is a decrease in the static/kinematic friction coefficients the slip motion will take place.

4.4 The effect of coefficients on the motion stability of wagon

Here we don't want to go through the general determination of critical speed but just using an example to see how different combinations of static/kinetic friction affect the performance of wagon. For this purpose, a sinusoidal alignment irregularity track is designed as shown in figure 21. We do the simulation with an increasing running speed from low to high until hunting motion appears.

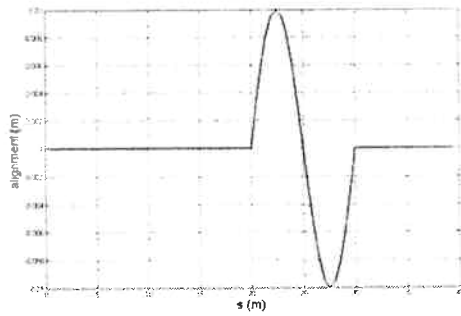


Figure 21. Straight track with alignment irregularity

If the static/kinetic friction coefficients are 0.4/0.3 then the stable running speed can reach 26 m/s (90 km/h) for an empty wagon. Figure 22 shows the motion of the leading wheelset. When the leading wheelset motion is stable then the motions of the other bodies of the wagon are also stable.

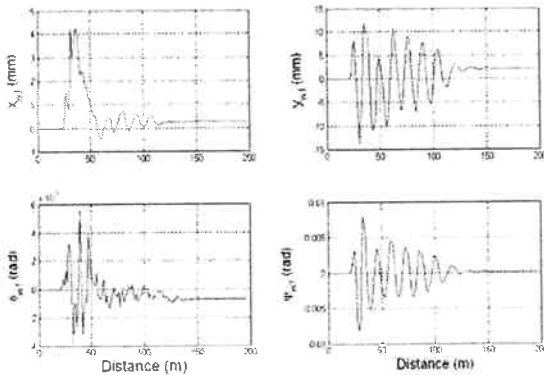


Figure 22. Stable motion of wagon with speed 26m/s

If we decrease the static/kinematic friction coefficients to 0.25/0.2 then the hunting motion will appear even for the running speed 23m/s as shown in figure 23.

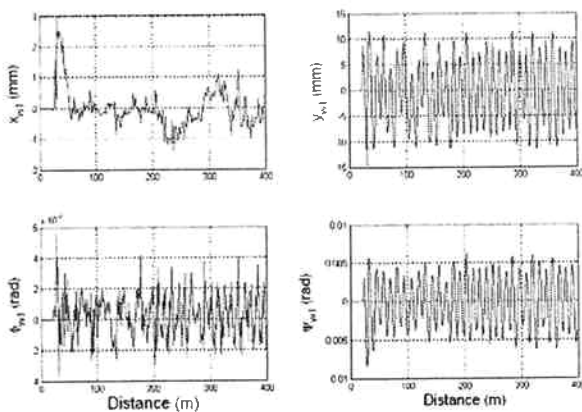


Figure 23. Hunting motion of wagon with speed 23m/s

As expected, this demonstrates that decreasing friction coefficients on the surface of adapters will decrease the maximum stable running speed.

5 CONCLUSIONS

A friction element which models static and kinematical friction was developed to describe two-dimensional dry friction. The friction element was successfully used in the simulation modelling of three-piece truck.

The friction between side frame and adapter is thoroughly investigated for selected conditions. The results show that in the general case, both stick and slip modes exist for the side frame and adapter contact. The relative displacements allowed by the bogie clearances are about ± 6 mm in longitudinal direction and about ± 5 mm in the lateral direction.

Whilst the truck is negotiating curved track of 500 m radius (or larger), the capability of the steering is not limited by the clearances between side frame and adapter. For curved track of 500 m radius with static/kinematical friction coefficients of 0.4/0.3, the side frame and adapter connection remains in stick mode.

The stability of the wagon was shown to decrease with the decreasing static/kinematical friction coefficients on the adapter contact surface. The reduction of 0.4/0.3 to 0.25/0.2 decreased the stable running speed by 13.4%.

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