5th ADAMS/Rail User's Conference
Haarlem 10-12 May 2000

Dynamic Modeling and Simulation of Three-Piece North American Freight Vehicle Suspensions with Non-linear Frictional Behaviour Using ADAMS/Rail

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ABSTRACT

At present there are approximately 2.5 million three-piece bogies operating under North American freight vehicles. Although US bogie manufacturers offer a variety of designs to vehicle builders, central to all of these bogie designs are the non-linear frictional characteristics inherent in the bogie suspension. These characteristics derive from the dynamic behaviour of the friction wedges. Friction wedges are small triangular metal elements that are held into position via spring loading, and function as the primary mechanism for vibrational energy dissipation between the bolster and side frame. Therefore, as a prerequisite to obtaining a realistic dynamic model of a typical three-piece bogie, an accurate model of the load-sensitive behaviour of these dry friction devices is required. Using design specifications from US bogie manufacturers and experimental data from the open literature, a load-dependent friction wedge model was developed using the ADAMS/View environment. After this model was validated against theoretical benchmarks, it was imported into an ADAMS/Rail vehicle model of a typical North American three-piece bogie. The results of model development for both the friction wedge and three-piece bogie will be presented, as well as ADAMS/Rail simulation results of the bogie dynamics under a variety of operating conditions.

INTRODUCTION

Three-piece bogies are a standard feature of virtually all US freight vehicles, as well as rail cars manufactured in Russia, India, South Africa, Brazil and Australia. The three main "pieces" which comprise the bogie structure are one bolster and two sideframes. As shown in Figure 1, the bolster is coupled to the two sideframes via the secondary suspensions, which in turn are connected to the wheelsets via the axle bearing connections (primary suspensions). Since the sideframe-axle bearing connection typically includes little or no resilient material, the soft suspension required for good curve negotiation (via wheelset yaw flexibility) is absent, and this connection is modelled using very high stiffness values for each degree of freedom.

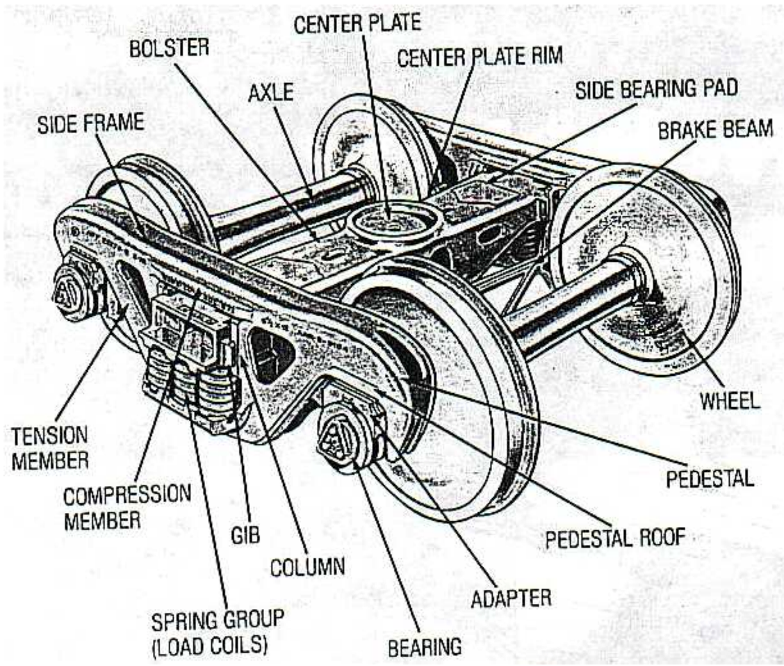

Figure 1. Typical three-piece freight vehicle bogie.

The secondary suspensions consist of a group of vertical nested springs in parallel with non-linear frictional damping elements known as “friction wedges”. While the spring nest provides vertical support between the sideframe and bolster, the friction wedges are spring loaded into an approximately conformal space between the vertical sideframe friction plates and the bolster to provide damping. The two main types of friction wedge suspensions differ by the method in which the wedges are spring loaded. The constant-damping suspension incorporates preloaded control springs that hold the wedges in place via a constant force. The variable-damping suspensions are typified by a set of independent support springs that supply the wedge compression forces as a function of the displacement between the bolster and sideframe [1]. Figure 2 presents a detailed view of these two different wedge configurations.

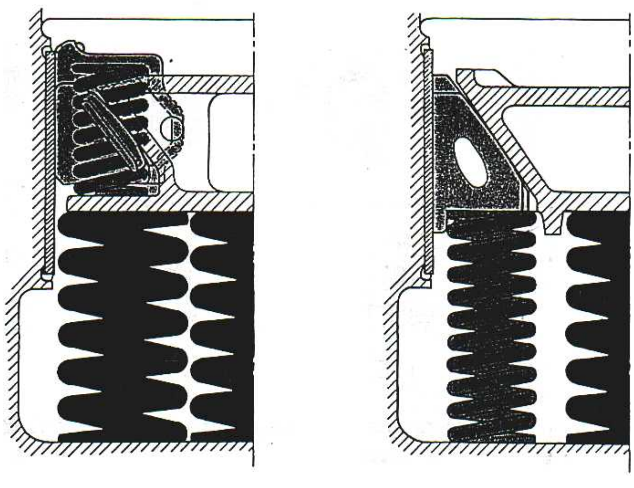

Figure 2. Secondary suspension configurations for (a) constant and (b) variable damping.

SUSPENSION MODEL DEVELOPMENT IN ADAMS VIEW

The objective of this study was to develop a virtual prototype of the friction wedge suspension for the purpose of better understanding the way in which different suspension parameters influence bogie dynamic behavior. In order to accomplish this task, the non-linear behaviour of the friction wedge damping needed to be understood and described mathematically, only then could its effect be incorporated along with equivalent spring nest stiffness into a dynamic model. It was decided that a friction wedge suspension having the constant-damping type of spring loading should be modelled first. A free-body diagram of the friction wedge is shown below in Figure 3 for both the case of suspension loading and unloading [2]. During the loading condition, the bolster is moving downward with respect to the sideframe, while during unloading, the bolster moves up. As indicated, the two frictional surfaces of the wedge are assumed to be in relative motion with respect to the sloped bolster surface and vertical wear plate of the sideframe under both conditions.

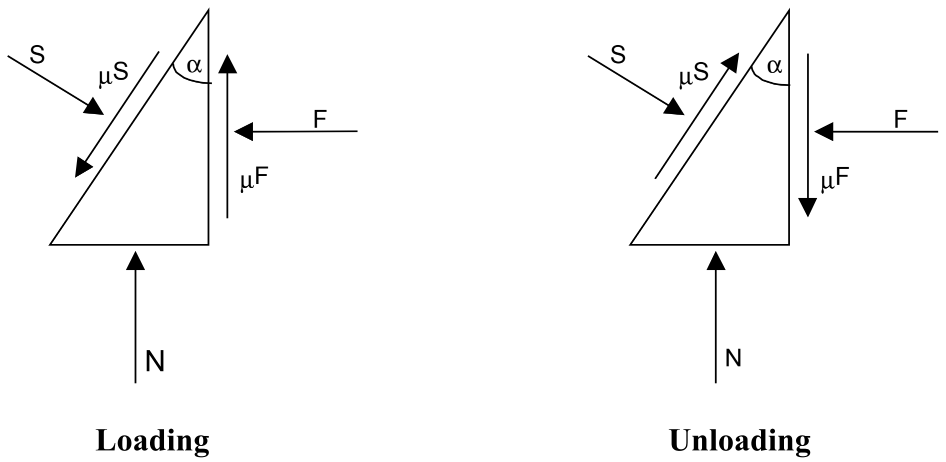

Figure 3. Free-body diagrams of friction wedge element forces during loading and unloading conditions

Loading Case:

$$S\sin\alpha + \mu S\cos\alpha = N + \mu F$$

and

$$F = S\cos\alpha - \mu S\sin\alpha$$

thus

$$\mu F = \mu N \frac{\cot\alpha - \mu}{1+\mu^2} \qquad (1)$$

Unloading Case:

$$S\sin\alpha - \mu S\cos\alpha = N - \mu F$$

and

$$F = S\cos\alpha + \mu S\sin\alpha$$

thus

$$\mu F = \mu N\frac{\cot\alpha + \mu}{1+\mu^2} \qquad (2)$$

Equations (1) and (2) were incorporated in a force limited, linear damper approximation similar to that used by Fries, *et al* [3]. For the case of "constant-damping" (fixed pre-load control spring loading) of Figure 2 (a), the force between the vertical wear plate of the sideframe and the friction wedge is described by μF. The wedge vertical force N is the result of the parallel combination of the control spring stiffness (PCS) and linear damper that are together connected in series to the wedge (to account for the effects of control spring pre-load and damping). A schematic representation of this combination is shown in Figure 4.

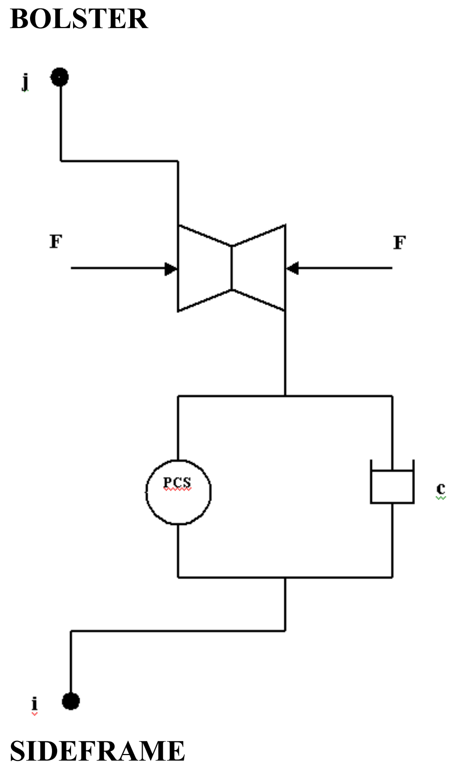

Figure 4. Friction wedge element schematic force diagram

As indicated by equations (1) and (2), the sideframe-bolster connection accommodated by the friction wedge, changes form depending on whether the bolster is in a loading or unloading condition. In order to develop the friction wedge model in ADAMS/View, it was necessary to describe the "switch" between these two conditions using ADAMS functions. The switch needed to be "toggled" between a "+" and "-" value, and it was decided that the relative vertical velocity between the bolster and sideframe would be the best measure for the independent variable. So a function which described the behavior shown in Figure 5 was developed using the STEP5 function and then incorporated as an ADAMS state variable into the wedge model.

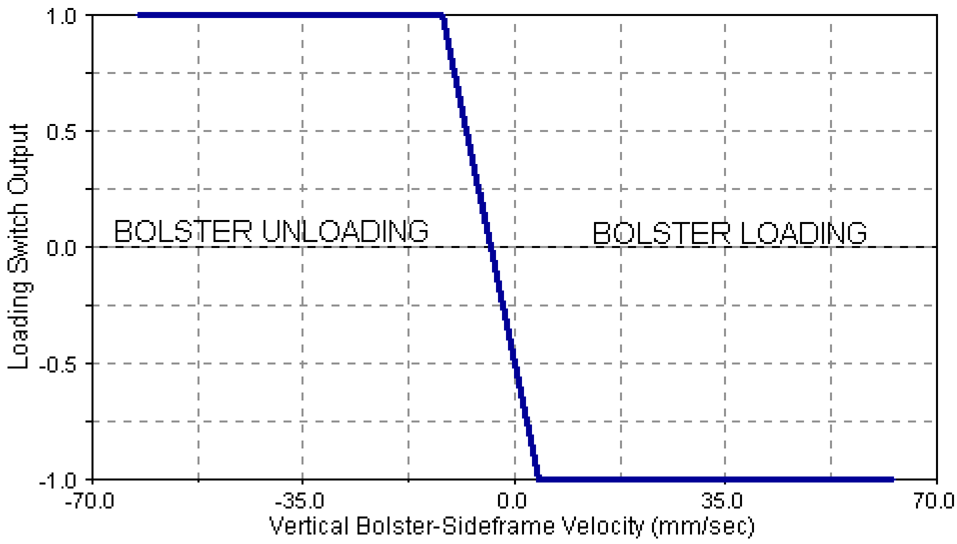

Figure 5. Loading Switch Function

The four state variables and single component force developed for implementing the complete friction wedge (includes two wedge elements) are given below for the ADAMS/View model, MODEL_2, which was a simple two mass system (representing a 1/4 vehicle loaded bolster and a single sideframe). For this model, an equivalent spring nest force was used to model the secondary suspension stiffness and five design variables were used to describe: coefficient of friction, wedge angle, pre-load of the control springs, spring nest stiffness (secondary suspension), and damping (for the control spring). Design variables were incorporated so that an ADAMS dialog box could be used to easily vary each term for the purpose of parametric investigation. An illustration of the ADAMS/View model is shown in Figure 6, where the vertical driving motion was input via the linear translational motion generator at a rate of 1.0 Hz.

State Variables and Single Component Force Definition:

Object Name : .MODEL_2.**LoadingSwitch**
Function : STEP5(VZ(.MODEL_2.PART_1.MAR_2,.MODEL_2.PART_2.MAR_4), -0.1, 1.0, 0.1, -1.0)

Object Name : .MODEL_2.**DampingForce**
Function : (.MODEL_2.Damping)*-VZ(.MODEL_2.PART_1.MAR_2,.MODEL_2.PART_2.MAR_4)

Object Name : .MODEL_2.**FrictionForce**
Function : (.MODEL_2.Mu)*(((1/TAN((.MODEL_2.WedgeAngle))) - ((.MODEL_2.Mu) * VARVAL(.MODEL_2.LoadingSwitch))) / (1+ (.MODEL_2.Mu)**2))* 2.0 * (.MODEL_2.PCS + VARVAL (.MODEL_2.DampingForce))

Object Name : .MODEL_2.**SpringNestForce**
Function : (.MODEL_2.SpringNestStiffness) * (.MODEL_2.SpringFreeLength - DM(.MODEL_2.PART_1.MAR_2,.MODEL_2.PART_2.MAR_4))+.MODEL_2.PRELOAD

Object Name : .MODEL_2.**FORCE_1**
Function :VARVAL(.MODEL_2.FrictionForce)*VARVAL(.MODEL_2.LoadingSwitch)+ VARVAL(.MODEL_2.SpringNestForce)

Values used for the initial design variables were:

mu = 0.4
wedge angle = 37.5
PCS = 1.6E4 N.
Spring Nest Stiffness = 4533.2 N/mm
Damping = 200. Ns/mm

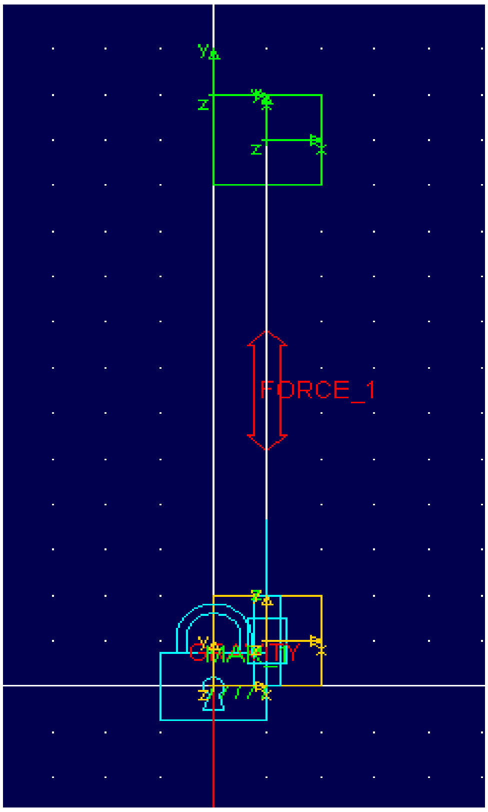

Figure 6. ADAMS/View model used for friction wedge simulation

SUSPENSION MODEL SIMULATION RESULTS

A sample output from this model is shown in Figure 7. The data are presented as a hysteresis plot where the equilibrium location of the bolster with respect to the sideframe is identified at a position of zero on the abcissa, and 235,360 Newtons on the ordinate (suspension pre-load condition). The result in Figure 7 is for an initial bolster unloading condition, and so the hysteresis output proceeds from the origin, generally down and to the right until the bolster reaches its peak unloading condition. At this point the loading switch function reverses polarity (due to the onset of bolster decent), and a vertical "jump" results as the friction wedge force output of Figure 7 begins to move generally up and to the left as the bolster continues to deflect the suspension spring nest. When the maximum loading condition is reached the output once again produces a vertical "drop" in force, and the bolster begins to rise as the force produced tracks along the lower force "unloading" curve continuing in a counter-clockwise fashion.

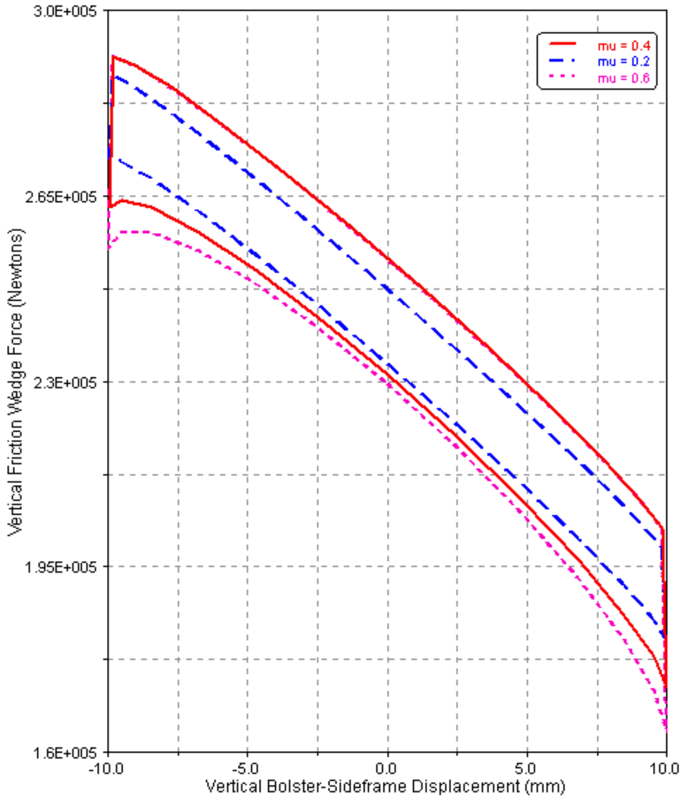

Figure 7. ADAMS/View friction wedge force hysteresis plot with mu as a parameter

BOGIE MODEL DEVELOPMENT WITH ADAMS/RAIL

A typical three-piece freight vehicle bogie model was developed in the ADAMS/Rail environment using geometry, material properties and connection data as specified via several different sources [4-7]. The model had approximately 34 degrees of freedom, 11 moving parts and 4 revolute joints. A rendered representation of this ADAMS/Rail vehicle model is shown in Figure 8.

The sideframes each had a mass of 525.8 kg and were connected to the wheelsets at each end by way of bushing connectors which were attached to small cylindrical solid "link" elements. There were two link elements attached to the wheelset (one at each end) via revolute joints, so as to facilitate wheelset roll freedom. These bushing connections had relatively high stiffness values in all directions, as the physical connection at the axle box – sideframe is basically metallic. The bolster – sideframe connections were the most complicated to model, as the literature indicated that in most directions, these connections were non-linear. Two bushing elements were used to accommodate the vertical and longitudinal stiffnesses at these connections, the vertical accounting for the nest of secondary suspension springs (4.5332E+06 N/m) and the longitudinal accounting for a metallic contact. Additional bolster - sideframe connections were modelled using two G-Forces to describe lateral and yaw stiffness, and lateral damping. All three of these connections were non-linear and modelled using the Akima spline function (AKISPL). An example is given below in Figures 9 and 10 for the lateral stiffness and damping connection.

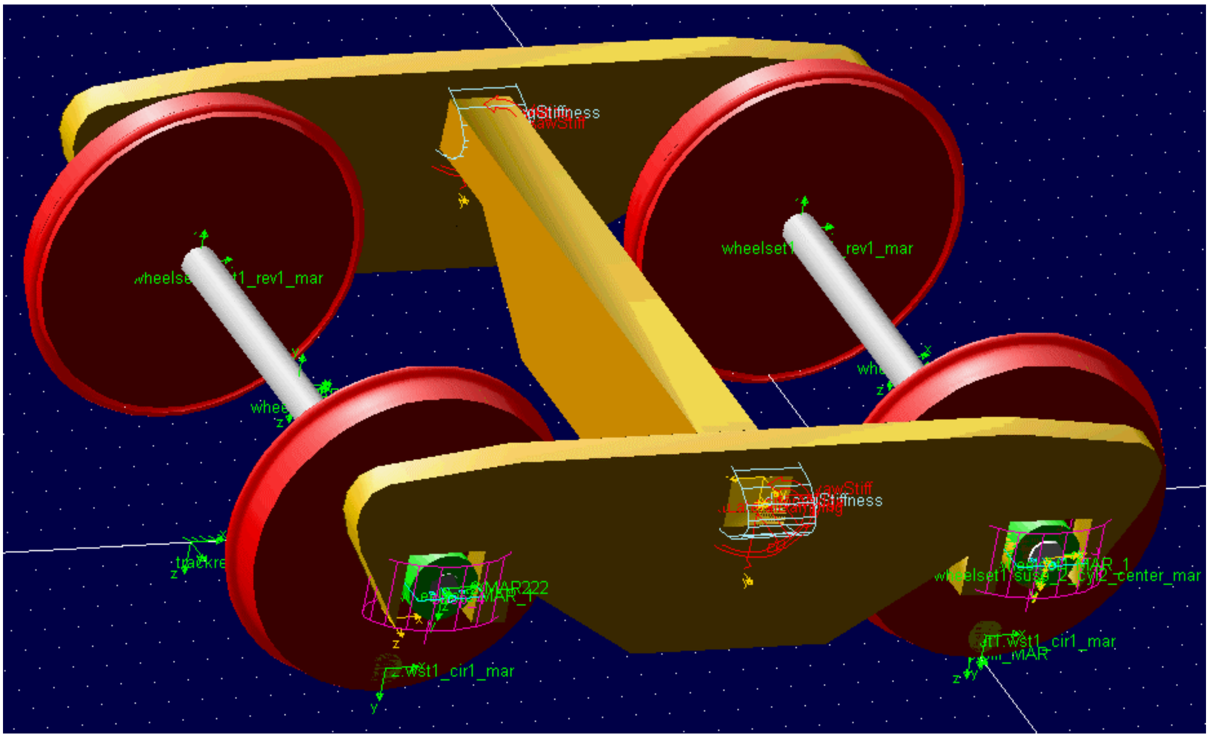

Figure 8. Model of three-piece bogie developed in this study.

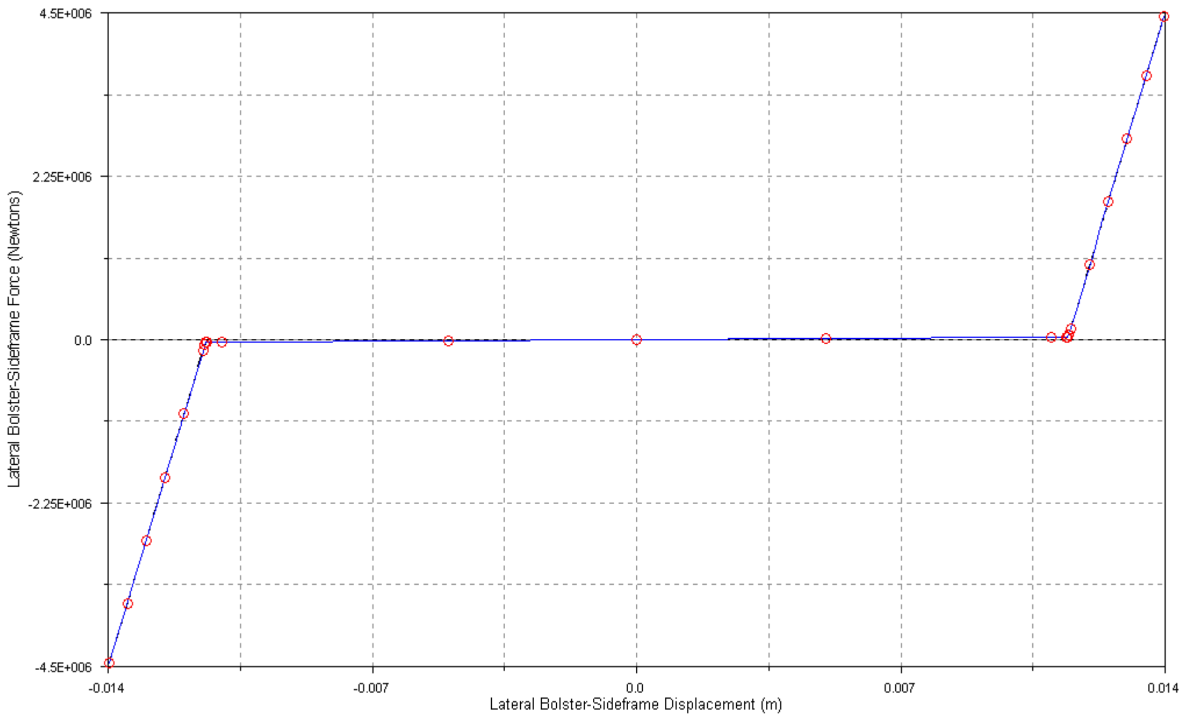

Figure 9. Spline used for bolster-sideframe lateral force–displacement behaviour

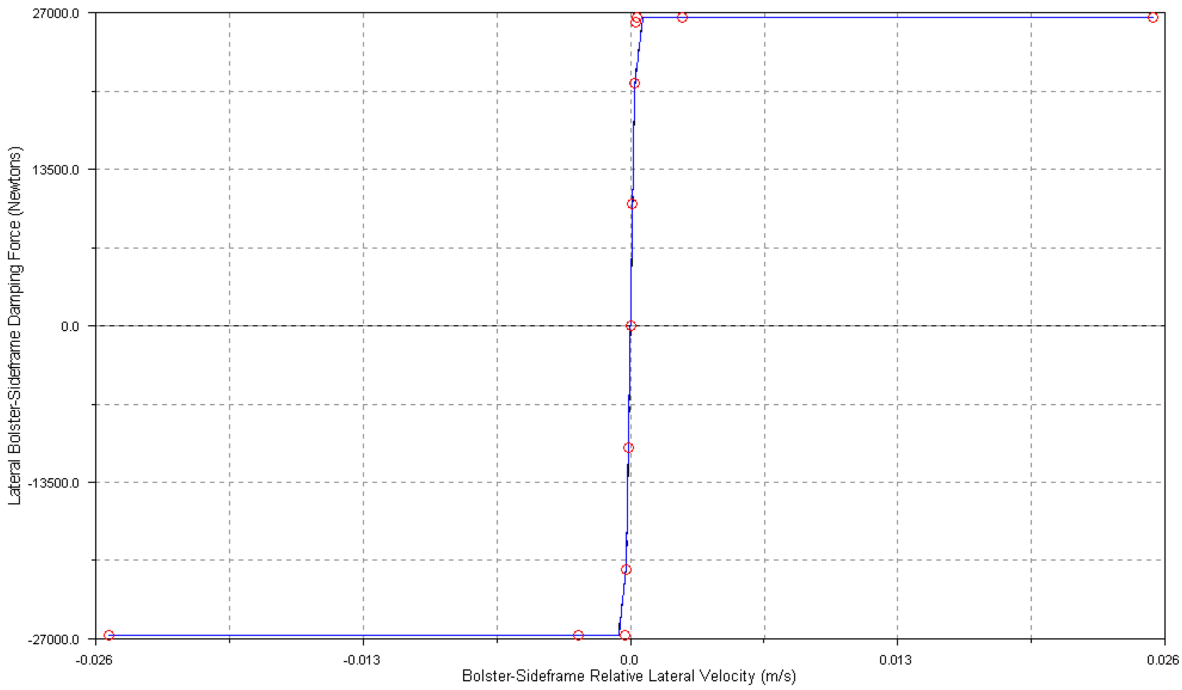

Figure 10. Spline used for bolster-sideframe lateral damping behaviour.

In addition to these connections, the friction wedge model developed in ADAMS/View was imported and modified so as to account for the correct marker numbers and directional sense between the bolster and sideframes.

THREE-PIECE BOGIE MODEL SIMULATION RESULTS

After some difficulty, the three-piece bogie model was successfully run using a level I contact model, however, the real interest was in obtaining results for a level III contact. This proved to be quite difficult, as numerous errors and warnings kept arising on account of all the non-linear connections (e.g. spline limits exceeded), bushing element orientations, and the use of an S-Force in describing the friction wedge force contribution. This was especially true when attempts were made to run the model with level III on curved track. Several modifications were made to allow the bogie to run with level III. Two of the main improvements made, were that of choosing a significant increase in the number of integration time steps during the analysis phase, and the addition of four separate “link connections” (and their respective revolute joints), which facilitated the axle rotation degree of freedom needed for level III. Prior simulation attempts had failed in part due to the bushing “wind-up” that resulted from a direct sideframe – wheelset, bushing connection. Regarding time step changes, it was found that with about 1000 to 1500 steps per second, the model was able to converge to a solution for level III on curved track (initial velocity of 7 m/s), in about 200 seconds of cpu time running on an Intergraph TDZ 425, parallel P3 processor computer.

A sample set of simulation results for the three-piece bogie model (with 24.24 Ton axle loads) traveling into an S-curve at an initial velocity V_{ox} = 9.0 m/s (20 mph) are included below in Figures 11-14. The wheel/rail profile was that of a S1002/UIC60 and a level III contact model was used for the analysis.

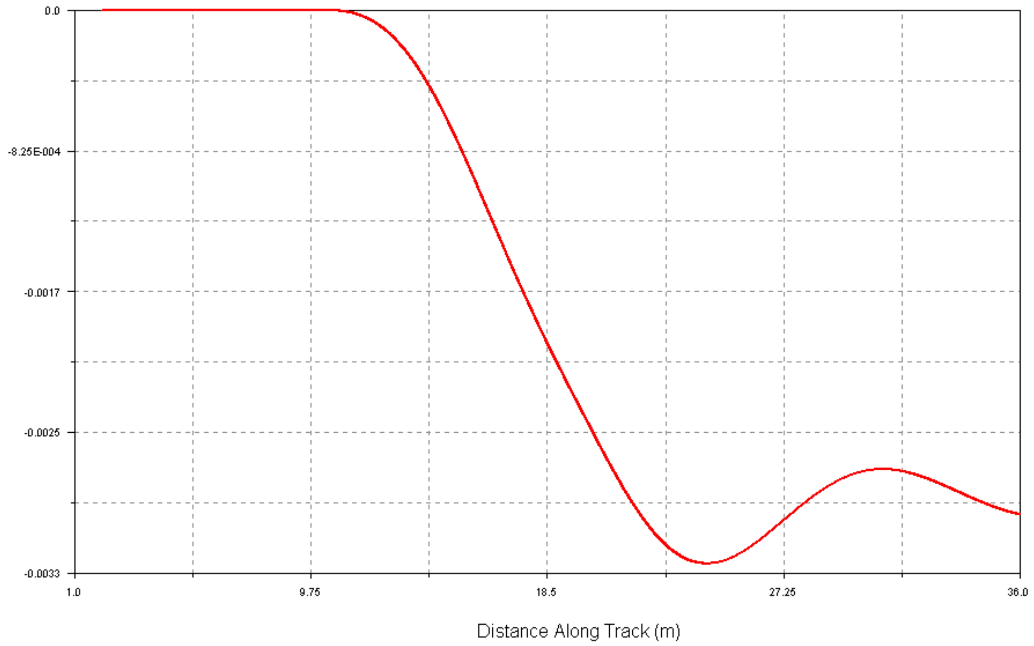

Figure 11. Lateral (y) displacement of wheelset 1 with respect to track centerline

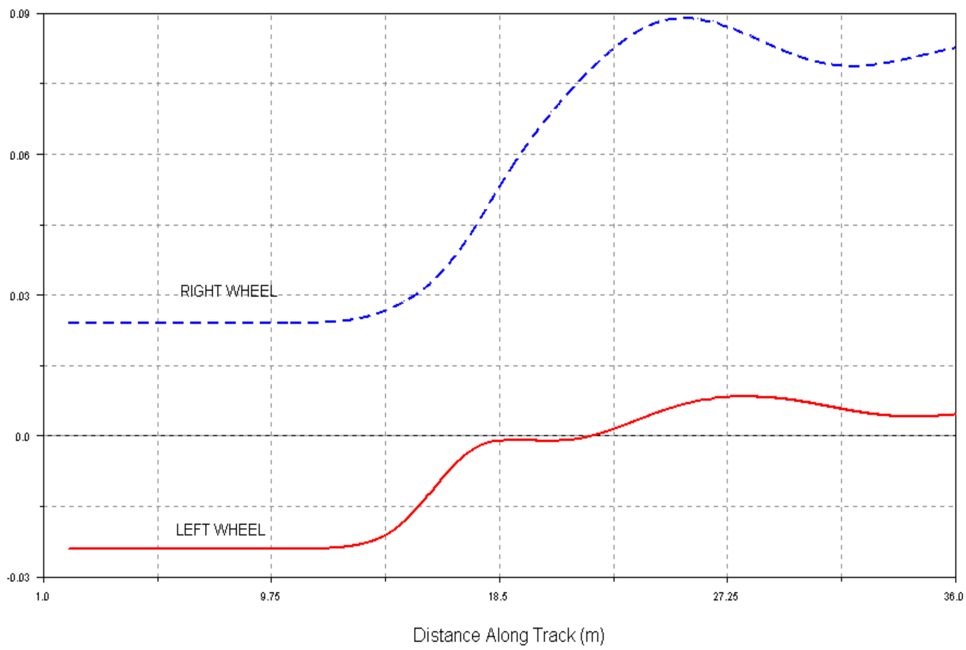

Figure 12. L/V ratio for left and right wheels of wheelset 1

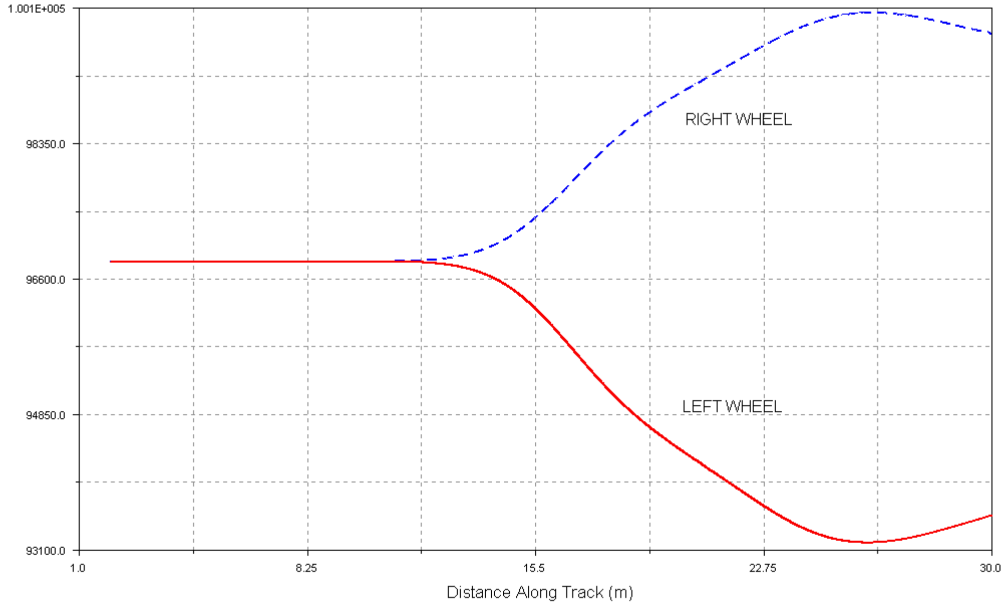

Figure 13. Vertical Forces on wheelset 1

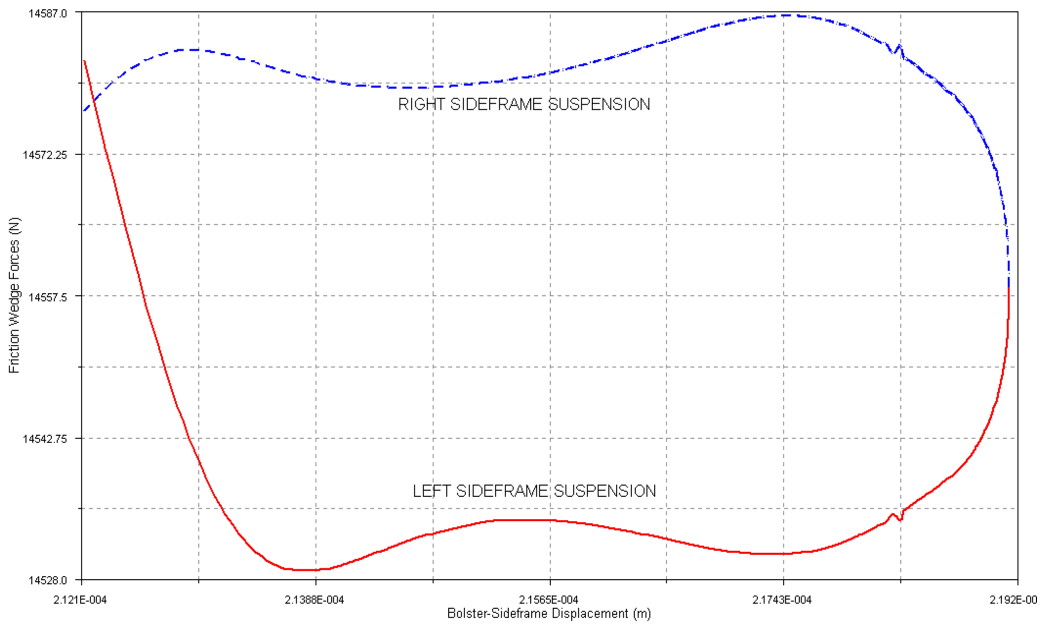

Figure 14. Forces in friction wedge suspensions at bolster-sideframe connections

CONCLUSIONS

A friction wedge model was developed using ADAMS/View and imported into ADAMS/Rail. The results of preliminary testing indicate that the bogie's two friction wedge suspension units coupled with six other highly non-linear connections located between the bolster and sideframe present significant modelling challenges, especially when level III contact is desired for curving studies. The results obtained seem reasonable for the overall bogie dynamic behaviour, however, future analysis will focus on refining the model in light of an exhaustive parametric study and comparison with experimental data.

ACKNOWLEDGEMENTS

The author is grateful to the following individuals for providing assistance with this study: Gabriele Ferrarotti, Chiara Bogo, Karl Bangert of MDI Germany, and Greg Saxton, Daniel Militaru, and Gary Kaletta of Gunderson, Inc, USA, also, Steve Mace, of Progressive Rail Technologies, Inc, USA.

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