Modelling and linear analysis of high-speed articulated trainsets

Imtiaz Haque

Department of Mechanical Engineering, Clemson University, Clemson, SC, USA
Fax: 867-656-4435 E-mail: sih@eng.clemson.edu

Mark Nagurka

Department of Mechanical and Industrial Engineering, Marquette University, Milwaukee, WI, USA
Fax: 414-288-7790 E-mail: nagurka@marquette.edu

Abstract: The success of high-speed rail systems in Europe and Japan has led to the consideration of deployment of such systems in the United States. Since operating conditions in the United States are different from those in Europe and Japan, questions arise as to the safety-related behaviour of such systems. To address questions of dynamic behaviour, linear simulation models have been developed to compute the lateral stability of articulated trainsets on tangent track. These models include the essential features of articulated trainsets such as shared trucks and suspension characteristics such as car-to-car connections and car-to-truck yaw dampers. Parameter studies have been conducted for a trainset consisting of ten vehicles and having a critical speed of approximately 310 km/h. The studies show that consist stability is sensitive to the yaw damping between the trucks and the carbody, the conicity of the wheel profile, and the primary suspension. Other parameters affect the stability but to a lesser degree. The modal behaviour of the consist suggests that instabilities can occur in the form of whole consist modes, in some cases a wave-like motion, especially at low conicities.

Keywords: Articulated vehicle, lateral stability, trainsets, vehicle dynamics.

Reference to this paper should be made as follows: Haque, I. and Nagurka, M. (2001) 'Modelling and linear analysis of high-speed articulated trainsets', *Int. J. of Vehicle Design*, Vol. 26, Nos. 2/3, pp. 249–263.

Nomenclature

a	track semi-gauge (m)
a_t	longitudinal distance between truck cg and secondary vertical suspension (m)
b	semi-distance between primary suspension (m)
c	longitudinal distance between vertical side spring suspension and car cg (m)
C_{cx}	inter-body roll damping (N·m·s/rad)
C_{cy}	inter-body pitch damping (N·m·s/rad)

C_{cz}	inter-body yaw damping (N·m·s/rad)
C_{jtf}	yaw damping for female joint-truck connection (N·m·s/rad)
C_{jtm}	yaw damping for male joint-truck connection (N·m·s/rad)
C_{px}	primary longitudinal suspension damping coefficient (N·s/m)
C_{py}	primary lateral suspension damping coefficient (N·s/m)
C_{pz}	primary vertical suspension damping coefficient (N·s/m)
C_{sy}	joint-to-truck lateral suspension damping coefficient (N·s/m)
C_{sz}	secondary vertical damping (N·s/m)
C_s	joint-to-truck vertical suspension damping coefficient (N·s/m)
d_f, d_r	distance between truck cg and front and rear wheelsets respect (m)
f_{11}	lateral creep coefficient (N/wheel)
f_{12}	lateral-spin creep coefficient (N·m/wheel)
f_{22}	spin creep coefficient (N·m^2/wheel)
f_{33}	longitudinal creep coefficient (N/wheel)
F_{sy}	joint suspension lateral force on truck (N)
F_{py}	primary suspension lateral force on truck (N)
h_3	vertical distance of joint lateral suspension to truck cg (m)
h_4	vertical distance of primary lateral suspension to truck cg (m)
I_{cx}	carbody roll moment of inertia (kg·m^2)
I_{cz}	carbody yaw moment of inertia (kg·m^2)
I_{cx}	carbody pitch moment of inertia (kg·m^2)
I_{tz}	truck frame yaw moment of inertia (kg·m^2)
I_{tx}	truck frame roll moment of inertia (kg·m^2)
I_{wy}	wheelset pitch moment of inertia (kg·m^2)
I_{wz}	wheelset yaw moment of inertia (kg·m^2)
I_{wx}	wheelset roll moment of inertia (kg·m^2)
k_{cx}	inter-body roll stiffness (N·m/rad)
k_{cy}	inter-body pitch stiffness (N·m/rad)
k_{cz}	yaw stiffness of car-to-car connection (N·m/rad)
k_{jtm}	male joint-to-truck yaw suspension stiffness coefficient (N/m)
k_{jtf}	female joint-to-truck yaw suspension stiffness coefficient (N/m)
k_{px}	primary longitudinal suspension stiffness coefficient (N/m)
k_{py}	primary lateral suspension stiffness coefficient (N/m)
k_{pz}	primary vertical suspension (N/m)
k_{sy}	joint-to-truck lateral suspension stiffness coefficient (N/m)
k_{sz}	inter-car yaw stiffness (N/m)
k_s	vertical stiffness between articulation joint and truck (N/m)
m_c	mass of carbody (kg)
m_J	mass of joint (kg)
m_t	mass of truck frame (kg)
m_w	wheelset mass (kg)
p	semi distance between secondary suspension (m)
r_l, r_r	rolling radii, left and right wheels respectively (m)
S_f	distance from car cg to male joint (m)
S_r	distance from car cg to female joint (m)
V	forward speed (m/s)
W_{app}	applied load (N/wheel)
λ	conicity

Γ roll angle coefficient
Δ contact angle coefficient
δ_0 contact angle offset
$\delta_l \delta_r$ contact angles, left and right wheels, respectively (rad)
ω angular velocity (rad/s)

1 Introduction

A number of High Speed Rail (HSR) systems have been developed in Europe and Japan that can operate over 300 km/h. Examples include the French TGV, German ICE, Swedish X2000, Italian ETR, and Japanese Shinkansen trains. The success of these systems in Europe and Japan has led to the consideration of deployment of such systems in the United States. Since operating conditions in the United States are different from those in Europe and Japan, questions arise as to the safety-related behaviour of such systems.

Some HSR features and configurations are fundamentally different from those of conventional railway vehicles. For example, the TGV uses an innovative articulated arrangement between adjacent cars in the consist that is distinct from the car-to-car connections in conventional vehicles. With this arrangement there is a shared truck between adjacent cars. The kinematics of the articulation and the associated car-to-car and car-to-truck connections produce coupled dynamic interactions between the vehicles in the train that cannot be accurately accounted for by considering single vehicle models. Therefore a multi-vehicle model is needed in the analysis and simulation of articulated trains to describe and evaluate their dynamic performance.

Stability of rail vehicles usually refers to lateral stability and identification of speeds at which the 'hunting' phenomena occurs in rail vehicles. Hunting is a self-excited lateral-yaw oscillation that is produced above a specific forward speed by the wheel-rail forces. The speed at which this oscillation is initiated is called the 'critical speed'. It is characterized by violent oscillations of the wheelsets and truck assemblies. Severe hunting is detrimental to good dynamic performance of the vehicle and poses significant safety problems. It can lead to track damage and derailment. Thus, the study of lateral stability is of particular importance for a high speed passenger train. The purpose of a lateral stability analysis is to determine the safe operating speed of the vehicle for a given set of parameters and to delineate safe and unsafe regions of behaviour.

2 Scope of work

This paper presents the results of the use of linear models of articulated trainsets for investigating lateral stability. Stability is restricted to linear eigenvalue-eigenvector analyses for determining a 'critical speed' of the trainset for a given set of parameters. A forced response model for predicting pitch and bounce behaviour in response to vertical track irregularities (Haque, Liu, and Zhu, 1995), as well as a curving model (Ahmed, Haque, and Liu, 1996) have also been developed but are not reported here.

Figure 1 Articulated joint and shared truck concept of the French TGV[1].

3 Model development

For an articulated train, the key element of focus is the articulation design for support and stability of two consecutive vehicles. The purpose of the articulation is to couple consecutive vehicles together more tightly than is typically the case for conventional arrangements. Articulated trainsets generally contain conventional components such as wheelsets, truck frames and car bodies as well as primary and secondary suspensions which are modeled in much the same way as with conventional trains. A typical carbody/truck/vehicle assembly, shown in Figure 1, consists of conventionally attached trucks in the front and rear, and shared trucks in between. The conventional trucks are connected to the carbody through lateral, yaw and roll suspension elements. The shared trucks are connected directly to the carbody through roll suspension elements and through the articulation joints. A key element of articulated train modelling is the representation of the articulation connections between the carbodies and the interconnections with the trucks. In the model developed here the articulation joint model has been made as generic as possible to allow the analyst to model different designs by picking the suspension elements appropriately. It consists of yaw and roll suspension connections between the male and female ends of the joint, and yaw suspension elements between the female and male ends and the trucks individually. These elements represent the car-to-truck yaw dampers that are normally employed in high speed vehicles. Lateral and vertical suspension elements also exist between the joint and the truck frame.

[1]Picture obtained from web site http://mercurio.iet.unipi.it

3.1 The wheelset model

The wheelset model development follows previous work, and is described by Shah, Thirumalai, Cui, and Haque (1997). It consists of two degrees-of-freedom, i.e. lateral and yaw, linear wheel-rail geometry, and linear creep force characteristics based on Kalker's linear creep theory. Wheelset angular motions as well as angles of contact between the wheel and the rail are assumed to be small.

3.2 The truck model

Each truck consists of two wheelsets, a truck frame, and primary and secondary suspension components. The truck frame is considered to have lateral, yaw, and roll degrees-of-freedom. The primary suspension contains longitudinal, lateral, and vertical stiffness and damping elements. The secondary suspension components include connections between the truck frame and carbody and connections between the truck frame and the articulation joint. The presence of the articulation joint between two adjacent car bodies and the connections between the car bodies and the shared truck preclude the necessity of having lateral and longitudinal suspensions between the carbody and the trucks. The vertical suspension characteristics between truck and carbody consist of linear stiffnesses and damping. The lateral and yaw suspensions between the car bodies and the trucks are accounted for through suspension elements that connect the articulation joint to the truck.

3.3 The articulation joint model

The modelling of the articulation joint has been done to accommodate a variety of possible designs. The articulation joint is assumed to have a mass even though this mass may be small compared to the carbody mass. The joint is modelled as kinematically constraining adjacent car bodies in the lateral and vertical planes. Connections between a joint and a truck are through suspension elements. The joint has a female end and a male end. The male end is fixed in front of carbody, and the female at the rear of the carbody. The articulation joint allows relative yaw and roll motions between the bodies. The suspension characteristics consist of:

1. Yaw and roll stiffness and damping between the male and female ends;
2. Yaw stiffness and damping between the male end and the truck frame as well as between the female end and the truck frame; and
3. Lateral stiffness and damping between the whole joint and truck frame.

3.4 The carbody model

Typical models of rail vehicles that assess lateral stability assign three degrees-of-freedom to the carbody. These are lateral displacement and yaw and roll rotations. In the case of articulated trains, the presence of the articulation joint provides a constraint that the lateral motion of the points of interconnection of two adjacent bodies move equally in the lateral plane thereby reducing the degrees-of-freedom of the system. The generalized coordinates chosen here to represent the motion of the bodies consist of the lateral displacements of the articulated joints and the roll angles of each of the bodies. The lateral translations of ith and $(i+1)$th articulation

Table 1 Degrees of freedom for the ten vehicle trainset.

Definition of DOFs	*Specification*
1st wheelset lateral displacement	For ith truck $i = 1, 2, \ldots, 11$
1st wheelset yaw displacement	
2nd wheelset lateral displacement	
2nd wheelset yaw displacement	
ith truck frame lateral displacement	
ith truck frame yaw angle	
ith truck frame roll angle	
Lateral displacement of 1st to 11th joint	For articulated joint motion
Roll displacement of 1st to 10th carbody	For carbody motion

joint are sufficient to represent the ith carbody yaw around its cg and the lateral translation of the ith carbody's cg.

3.5 Equation formulation

The equations of motion for an articulated train set containing n bodies are developed using the methodology outlined by Shah, Thirumalai, Cui, and Haque (1997). Symbolic equations of motion utilizing the symbolic computation program MAPLE are developed, automatically written out in Fortran, and solved through MATLAB. Different models with different numbers of bodies can be easily generated using this formulation.

3.6 Method of solution

The equations of motion are written in state variable form and the eigenvalues found by solving the characteristic equation. The critical speed for hunting stability is found by running the program sequentially through a number of forward speeds until the real part of any eigenvalue becomes a positive number and the corresponding damping ratio becomes zero or negative. The least damped hunting mode is then obtained from the corresponding eigenvector.

The model used for the purpose of this study consists of ten vehicles. Each truck is shared by contiguous vehicles except at the ends of the train set where the truck is placed under the carbody and connected to a lateral, yaw, and roll suspension connection located in the transverse plane of the cg of the truck frame. The total degrees-of-freedom (DOF) for n articulated vehicles are $9n + 8$, i.e. 98 DOF for a 10 articulated-vehicle trainset. These are shown in Table 1.

4 Results

A typical set of baseline parameters of this model is given in Table 2. This parameter set represents a hypothetical trainset and was chosen to provide a critical speed of

Table 2 Baseline parameters for a ten vehicle consist.

Symbol	*Value*	*Unit*	*Symbol*	*Value*	*Unit*
m_c	25000	kg	k_{cz}	0.0	$N \cdot m/rad$
m_t	5000	kg	C_{jtf}	2.40×10^5	$N \cdot m \cdot s/rad$
m_w	2000	kg	C_{jtm}	2.40×10^5	$N \cdot m \cdot s/rad$
m_J	250	kg	C_{cx}	0.0	$N \cdot m \cdot s/rad$
I_{cx}	5.625×10^5	$kg \cdot m^2$	C_{cz}	0.0	$N \cdot m \cdot s/rad$
I_{cz}	6.625×10^5	$kg \cdot m^2$	f_{11}	6.550×10^6	N/wheel
I_{tx}	1740	$kg \cdot m^2$	f_{12}	2.396×10^4	$N \cdot m$/wheel
I_{tz}	7630	$kg \cdot m^2$	f_{22}	0.0	$N \cdot m^2$/wheel
I_{wx}	1500	$kg \cdot m^2$	f_{33}	8.150×10^6	N/wheel
I_{wy}	140	$kg \cdot m^2$	λ	0.025	
I_{wz}	1500	$kg \cdot m^2$	Δ	0.025	
k_{sz}	2.936×10^6	N/m	Γ	0.025	
C_{sz}	3.930×10^4	$N \cdot m \cdot s/rad$	δ_0	00	
k_{px}	6.800×10^6	N/m	W_{app}	1.5×10^5	N/wheel
k_{py}	3.920×10^6	N/m	a	0.7175	m
k_{pz}	5.756×10^5	N/m	b	0.95	m
C_{px}	2.500×10^4	$N \cdot s/m$	p	0.95	m
C_{py}	2.500×10^4	$N \cdot s/m$	S_f	10.8	m
C_{pz}	3.920×10^4	$N \cdot s/m$	S_r	10.8	m
k_{sy}	3.570×10^5	N/m	d_f	1.25	m
C_{sy}	7.845×10^4	$N \cdot s/m$	d_r	1.25	m
k_{jtf}	1.760×10^5	$N \cdot m/rad$	h_3	0.365	m
k_{jtm}	1.760×10^5	$N \cdot m/rad$	h_4	0.365	m
k_{cx}	0.0	$N \cdot m/rad$	h_f, h_r	0.36	m
C_{srf}	0.0	$N \cdot s/m$	c_f, c_r	0.36	m
C_{srr}	0.0	$N \cdot s/m$	T_f, T_r	5.00	m
r_o	0.45	m	a_{tf}, a_{tr}	1.25	m

approximately 300 km/h for a ten car consist with conventional truck configurations at the ends. Some of the sources for these data are available in the open literature, see Iguchi (1993). The data are presented in Table 2.

The above parameter set was used as a baseline parameter set to investigate articulated vehicle stability behaviour. All the vehicles in the consist were assumed to have identical suspension characteristics and were assumed to be symmetrical about the longitudinal axis.

The stability characteristics of an articulated trainset differ considerably from that of a single vehicle. With the number of elements connected together through kinematic constraints, there is the possibility of numerous modes of instability. The presence and characteristics of these modes depend primarily on the suspension parameters, the wheel/rail geometry parameters, and the masses and inertias of the interconnected elements. Hence, a different combination of data could give different results. This parameter study focused on a select group of suspension, geometry, and inertial parameters.

The following aspects were investigated: (a) Typical modes of instability or the nature of the eigen solution for the nominal vehicle; (b) the influence of suspension parameters on critical speed; (c) the influence of wheel tread conicity on critical speed; (d) the influence of wheelset and truck frame mass on critical speed, and (e) the influence of consist length and arrangement on critical speed.

The results obtained are discussed in Table 2.

5.1 Nature of the eigen solution

The parameters presented in Table 2 give a critical speed of 311 km/h for the ten vehicle consist. The natural frequency of the least damped mode at this speed is 1.567 Hz. This is well below the free wheelset kinematic frequency (3.83 Hz) but closer to the rigid truck kinematic frequency (1.9 Hz) for this speed and conicity. The least damped or unstable mode resembles the carbody hunting mode seen in conventional vehicles. The eigenvector is characterized by large lateral motions of the wheelsets, truck frames, and the articulation joints. Since the number of elements is large, a graphical representation of select components of the eigenvector is presented in Figure 2. Only linear displacements are plotted. The vertical location of each of the elements in the plot is obtained by plotting the real part of the eigenvector which only gives a sense of the phase relationships between the elements. The largest element in this case is the tenth truck. In the front of the trainset, the wheelsets, trucks and carbody move in-phase. The middle portion shows out-of-phase motions between the trucks and

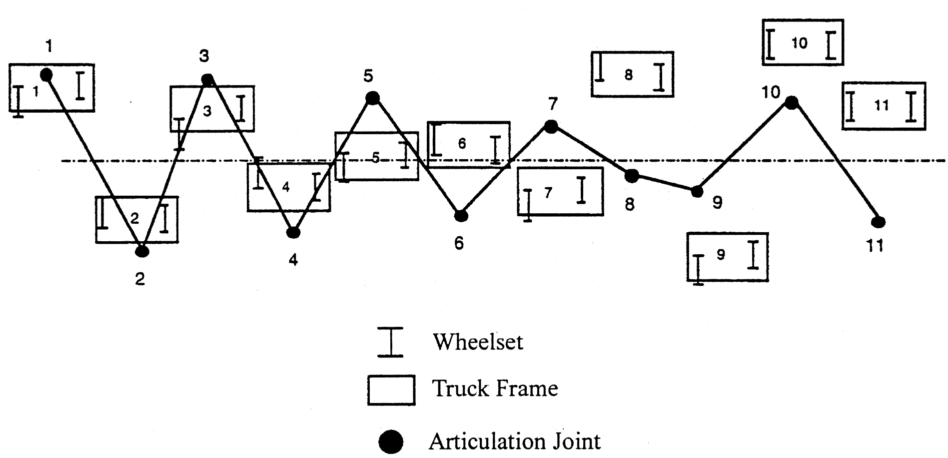

Figure 2 Mode shape (linear displacements only) for the least damped mode using nominal parameters.

the articulation joints with in-phase motions again at the rear of the trainset. The wheelsets follow closely the behaviour of the trucks. Though not obvious in Figure 2, the elements of the eigenvector show that the magnitude of wheelset, truck and joint motions grow progressively as one approaches the rear of the consist except for the motion of the last truck. The largest motions for the joints occur at the rear end of the train with the exception of the last articulation joint. The mode shape is affected by a number of parameters including wheel conicity.

5.2 Influence of the carbody-truck connections

The secondary lateral and vertical damping and the secondary yaw stiffness are known to have a significant influence on the stability of a conventional vehicle. In the case of the articulated train models, the secondary suspension elements of interest are the lateral suspension between the articulation joint and the truck, the roll suspension between the carbody and the truck, and yaw suspensions between the male and female joints and the truck.

Figures 3 through 6 show the influence of these suspension parameters on the stability of the baseline vehicle. Figure 3 shows that a decrease in the nominal lateral joint-to-truck stiffness results in a slight increase in critical speed with a maximum value being reached at approximately 2.1×10^5 N/m, a further decrease leading to a reduction in critical speed. This indicates that an optimum value for the secondary lateral stiffness may exist, the value of which undoubtedly depends on the other suspension parameters of the consist. Increasing the stiffness above the nominal value results in a decrease in critical speed. The influence of lateral damping (Figure 4) is more pronounced than that of the stiffness. A decrease in damping increases the critical speed.

Figure 5 shows the influence of the secondary yaw stiffness on the critical speed. Increasing the yaw stiffness increases the critical speed. Zero stiffness lowers the frequency of the least damped mode but only lowers the critical speed slightly. The yaw dampers play a strong role in the lateral stability of the articulated train model. Figure 6 shows that removal of the yaw dampers reduces the critical speed of the

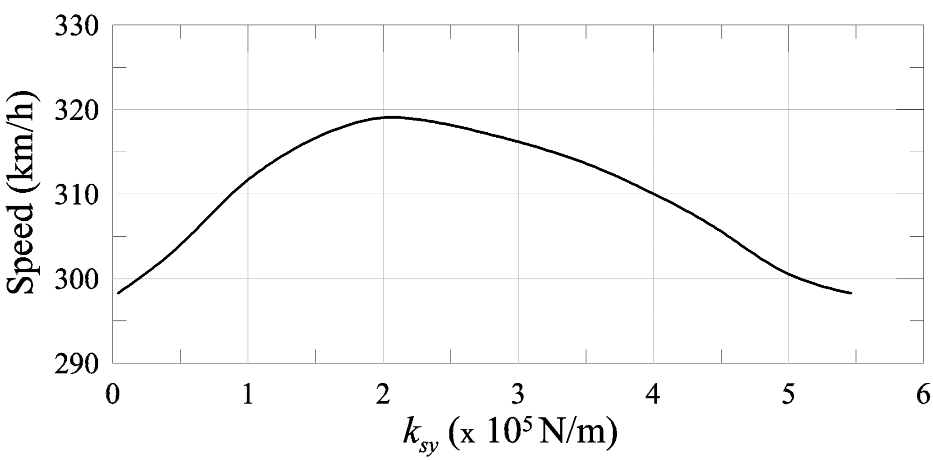

Figure 3 Critical speed versus joint-truck lateral stiffness.

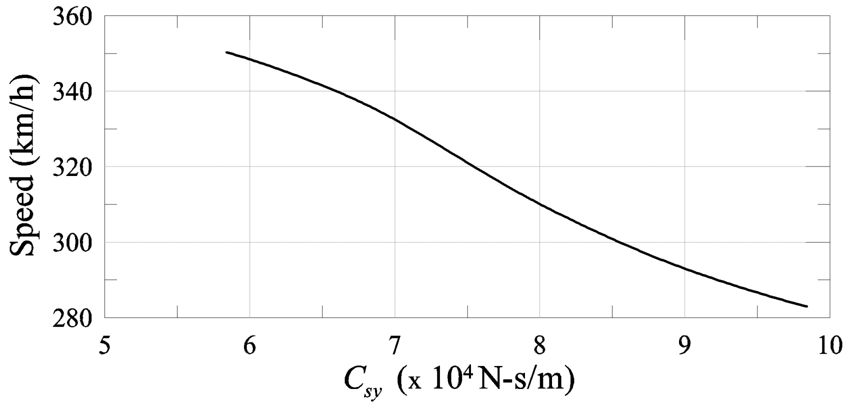

Figure 4 Critical speed versus joint-to-truck lateral damping.

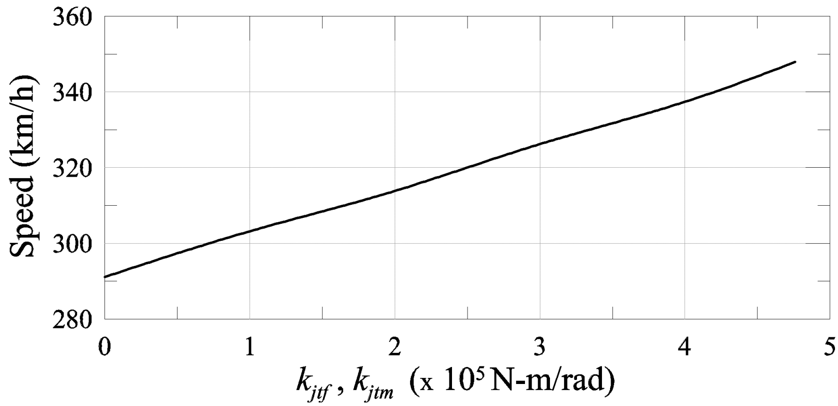

Figure 5 Critical speed versus car-to-truck yaw stiffness.

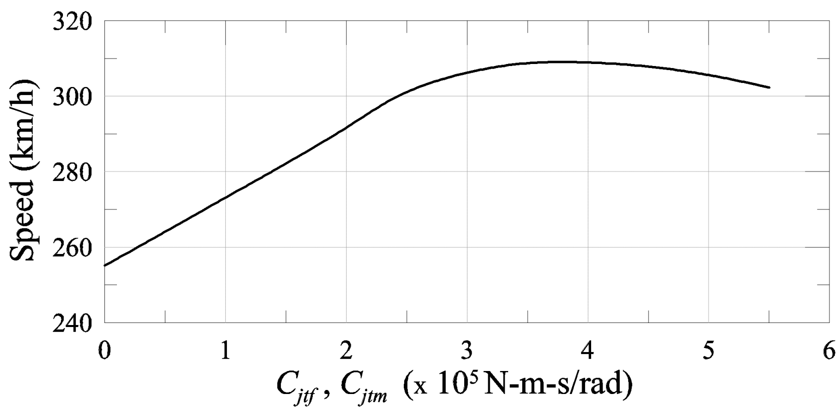

Figure 6 Critical speed versus car-to-truck yaw damping

nominal vehicle but an increase in yaw damping increases the critical speed to a point past which the speed starts to decrease. This again indicates an optimum value for the damping that may depend on suspension parameters of the consist. The case where

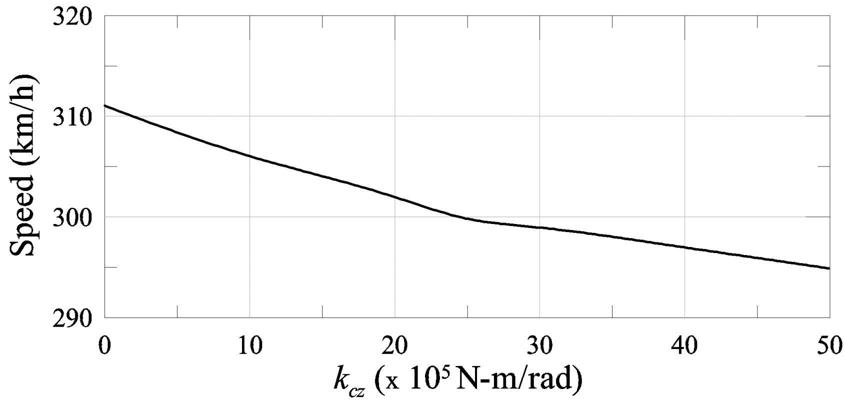

Figure 7 Critical speed versus car-to-car roll stiffness.

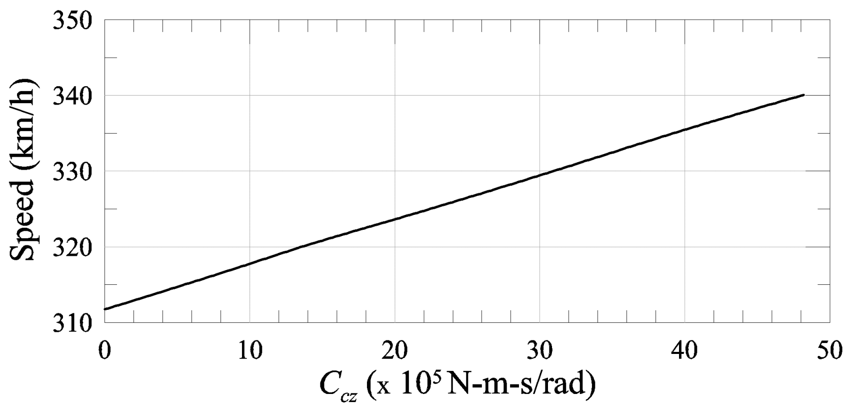

Figure 8 Critical speed versus car-to-car yaw damping.

both the stiffness damping were removed from the model is not shown here. The influence on critical speed is more pronounced with the removal of the damping playing a bigger role. The critical speed in this case is 241 km/h as opposed to 291 km/h for the zero stiffness and 256 km/h for the zero damping case.

5.3 Influence of the inter-car connection – yaw and roll dampers

Some articulated passenger trainsets have inter-body connections to reduce inter-car, roll and yaw. The inter-car roll and yaw damping and stiffness between two consecutive car bodies have some influence on the lateral stability. Results are shown in Figures 7 and 8. Increasing the yaw stiffness has an adverse affect on system stability. Yaw damping has the opposite effect with an increase in yaw damping increasing the critical speed. The influence of inter-car roll damping is minimal.

5.4 Influence of sprung and unsprung mass

The unsprung mass has a strong influence on the lateral stability of the consist. Reducing the weight of wheelset from 2000 kg to 1000 kg increases the critical speed

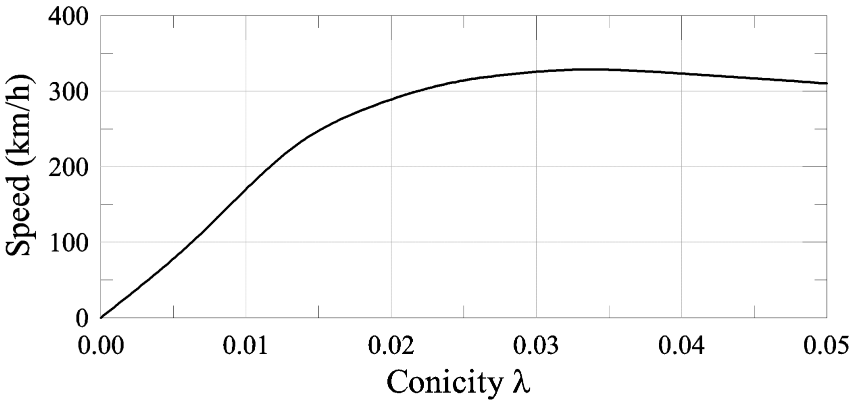

Figure 9 Critical speed versus conicity.

significantly. For the ten car consist, this results in an increase in speed from 310 km/h to about 378 km/h. Similarly, reducing the weight of the truck frame increases the critical speed.

5.5 Influence of conicity

For conventional vehicles it is well known that, in general, decreasing conicity increases truck hunting stability, with the speed versus conicity curves typically approximating quadratic hyperbolae, and with maximum achievable critical velocities occurring at the lowest conicities. The body hunting modes are similarly affected by decreasing conicity with the accompanying effect that more damping is needed to control body hunting at lower conicities. However, it has been found for steered vehicles that this is not necessarily the case (see Smith and Anderson, 1987). Steered axle trucks can exhibit low conicity instabilities, i.e. areas of instabilities at the low conicity end of the spectrum.

Articulated vehicles share similar characteristics. Figure 9 shows the effect of conicity on critical speed. As expected, the equivalent conicity strongly affects the lateral stability. Conicities below the nominal value result in lower critical speeds. The critical speed is higher above the nominal value, increasing with conicity until a value of 0.035. Above this value, the critical speed decreases. Table 3 shows some of the

Table 3 Influence of conicity on stability.

Conicity	0.015	0.025	0.035	0.045	0.055
Critical speed (km/h)	240	311	330	321	306
Largest element of eigenvector	Lateral of 10th articulation joint	Lateral of 10th truck frame	Lateral of 9th truck frame	Front wheelset lateral of 11th truck frame	Front wheelset lateral of 11th truck frame
Natural frequency of least damped mode (Hz)	1.02	1.57	1.92	2.46	2.67

results. An inspection of the eigenvectors shows that at the low conicities, the consist executes a wave-like motion with the small motions occurring at the front and the larger motions occurring at the end of the train. Again, the eigenvector is dominated by the lateral motions of the components with very small angular motions. The modal behaviour is similar to the carbody hunting modes seen in conventional vehicles. Increasing conicity results in large truck lateral motions as the modal frequency gets closer to the free truck hunting frequency. At and above conicities of 0.045, the modal frequency coincides with the free truck kinematic frequency resulting in fully developed truck hunting for one truck in the consist. In this type of mode, the motion is dominated by large wheelset and truck lateral and yaw motions and smaller articulation joint motions at the 11th truck. All other motions are very small.

5.6 Influence of primary suspension

Variations in the parameter values of lateral stiffness and damping in the primary and secondary suspensions affect the lateral stability of the consist. Figures 10–13 show the influence of the primary suspension parameters on the critical speed. The lateral primary suspension, for the nominal parameters chosen for this model, has an influence on vehicle behaviour that has been known previously. There exists, for this set of

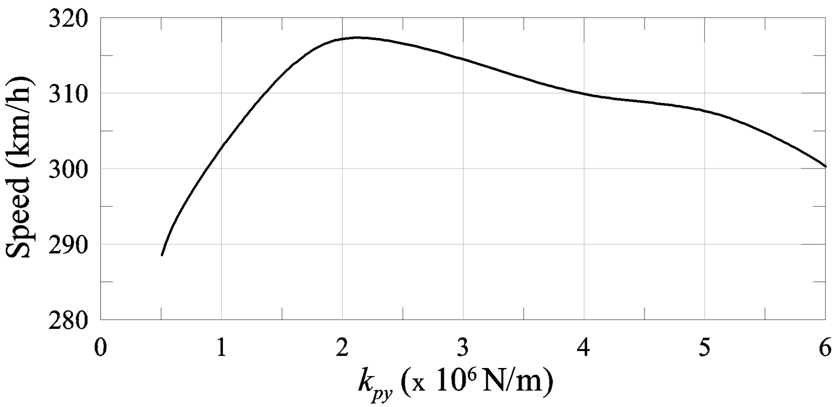

Figure 10 Critical speed versus lateral primary stiffness.

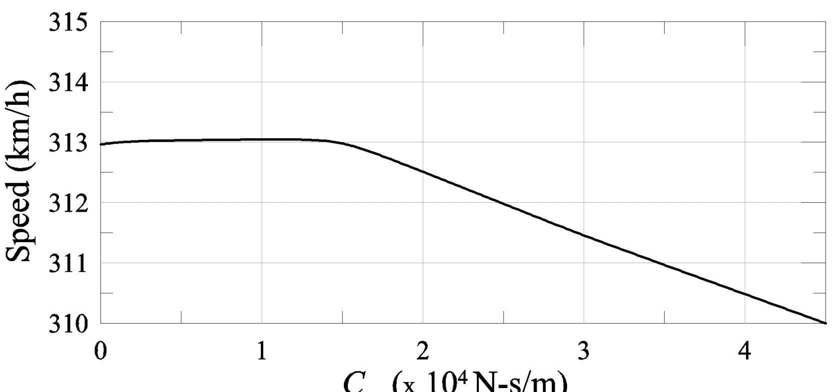

Figure 11 Critical speed versus lateral primary damping.

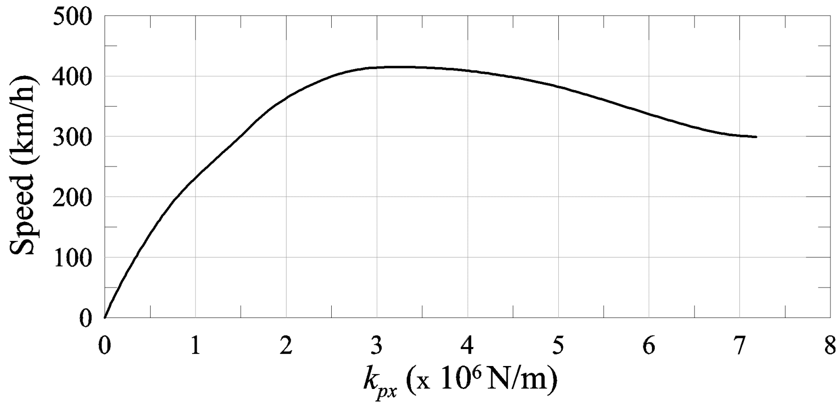

Figure 12 Critical speed versus longitudinal primary stiffness.

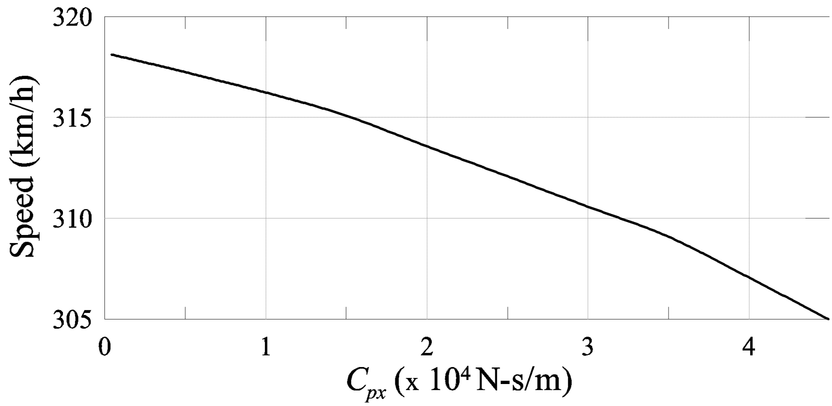

Figure 13 Critical speed versus longitudinal primary damping.

parameters, a range of optimum values of the primary lateral stiffness, below and above which the critical speed of the consist is lowered (Figure 10). The influence of the primary lateral damping as well as the longitudinal stiffness and damping is shown in Figures 11–13.

6 Summary

This paper has discussed the development of a computer model for predicting the lateral stability of articulated vehicles. A set of parameters representing a baseline consist consisting of ten cars and giving a critical speed of approximately 310 km/h was assembled. Parameter studies were conducted mainly to assess the influence of car-to-car and car-to-truck suspension parameters on consist stability. The studies show the results of the model are sensitive to the parameter values chosen. In addition, for the parameters chosen here, optimum values of suspension characteristics exist that produce the highest critical speeds. It is especially noted that consist stability is sensitive to the yaw damping between the trucks and the carbody, the conicity of the wheel profile, with low conicities below a certain value resulting in lower critical speeds, and the primary suspension. Results were found to be fairly insensitive to the inter-body yaw and roll connections.

Acknowledgements

This work was supported by the U.S. Department of Transportation through the Volpe National Transportation Systems Center (VNTSC), Cambridge, MA.

References

Haque, I., Liu, C. and Zhu, G. (1995) *Modeling and Linear Analysis of High Speed Articulated Trainsets*, Report submitted to Volpe National Transportation Centre.

Ahmed, A. K.W., Haque, I. and Liu, C. (1996) 'Computer simulation of steady state performance of high speed articulated train sets', *Transactions of the CSME*, Vol. 20, No. 4, pp. 365–389.

Iguchi, M. (Ed.) (1993) *JSME Proceedings of the International Conference on Speedup Technology for Railway and Maglev Vehicles*, Vol. 2, Yokohama.

Shah, M., Thirumalai, M., Cui, K. and Haque, I. (1997) 'Symbolic methods for modeling rail vehicle systems', *Special Issue of International Journal of Vehicle Design*, Vol. 18, No. 5, pp. 487–517.

Smith, R. E. and Anderson, R. J. (1987) 'Low conicity instabilities in forced-steering railway vehicles', *Supplement to Vehicle System Dynamics*, Vol. 17, pp. 17–28.

Biographical notes

Imtiaz Haque is Professor of Mechanical Engineering at Clemson University in Clemson South Carolina. He has been with the Department since 1982. His teaching and research interests lie in the general areas of dynamics and vibrations of mechanical systems with special emphasis on modeling and simulation of Vehicle Dynamics, Multi-body Systems, Mechanisms and Machines, and Manufacturing Processes. He is a member of the American Society of Mechanical Engineers (ASME) and currently serves on the Vehicle Design Committee of the Asme Design Division. He is a member of the editorial board of the International Journal of Heavy Vehicle Systems. Dr. Haque has served as consultant to numerous private and Federal Agencies and in 1996–97 spent hsi sabbatical year at the BMW Research and Engineering Center in Munich, Germany.

Mark L. Nagurka holds B.S.E. and M.S.E. degrees in mechanical engineering and applied mechanics from the University of Pennsylvania (1978 and 1979, respectively) and a Ph.D. degree in mechanical engineering from MIT (1983). He taught at Carnegie Mellon University (1983–94) and worked as a Senior Research Engineer at the Carnegie Mellon Research Institute (1994–96) in Pittsburgh, PA., where he conducted research on high speed rail and maglev vehicles. He is currently an associate professor in the Department of Mechanical and Industrial Engineering at Marquette University in Milwaukee, WI, where he pursues research in vehicle dynamics, human/machine interaction, and control system design, and teaches courses in "Mechanical Measurements & Instrumentation," "Dynamics of Mechanical Systems," and "Mechatronics".