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of Transportation

**Federal Railroad
Administration**

SAFETY OF RAILROAD PASSENGER VEHICLE DYNAMICS

Final Summary Report

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This report is a summary of all the work done by Foster-Miller on the passenger rail vehicle dynamic safety under the contract awarded by the FRA. The report presents key issues and findings in the safety assessments and a safety assessment methodology for rail vehicles negotiating tracks with geometry perturbations. Key issues include definition of appropriate derailment criteria, track perturbations to cover a variety of scenarios, development of a simulation tool and a framework for safety assessment. The simulation tool OMNISIM and its test validations have been presented. The required parameters in the simulation tool and the measurement techniques for their quantification are presented. A safety assessment methodology is formalized, which has been validated through full-scale field testing.

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SI* (MODERN METRIC) CONVERSION FACTORS

APPROXIMATE CONVERSIONS TO SI UNITS

Symbol	When You Know	Multiply By	To Find	Symbol
		LENGTH		
in	inches	25.4	millimeters	mm
ft	feet	0.305	meters	m
yd	yards	0.914	meters	m
mi	miles	1.61	kilometers	km
		AREA		
in^2	square inches	645.2	millimeters squared	mm^2
ft^2	square feet	0.093	meters squared	m^2
yd^2	square yards	0.836	meters squared	m^2
ac	acres	0.405	hectares	ha
mi^2	square miles	2.59	kilometers squared	km^2
		VOLUME		
fl oz	fluid ounces	29.57	milliliters	ml
gal	gallons	3.785	liters	l
ft^3	cubic feet	0.028	meters cubed	m^3
yd^3	cubic yards	0.765	meters cubed	m^3

NOTE: Volumes greater than 1000 l shall be shown in m^3.

Symbol	When You Know	Multiply By	To Find	Symbol
		MASS		
oz	ounces	28.35	grams	g
lb	pounds	0.454	kilograms	kg
T	short tons (2000 lb)	0.907	megagrams	Mg
		TEMPERATURE (exact)		
°F	Fahrenheit temperature	5(F-32)/9 or (F-32)/1.8	Celcius temperature	°C
		ILLUMINATION		
fc	foot-candles	10.76	lux	l
fl	foot-Lamberts	3.426	candela/m^2	cd/m^2
		FORCE and PRESSURE or STRESS		
lbf	poundforce	4.45	newtons	N
psi	poundforce per square inch	6.89	kilopascals	kPa

APPROXIMATE CONVERSIONS FROM SI UNITS

Symbol	When You Know	Multiply By	To Find	Symbol
		LENGTH		
mm	millimeters	0.039	inches	in
m	meters	3.28	feet	ft
m	meters	1.09	yards	yd
km	kilometers	0.621	miles	mi
		AREA		
mm^2	millimeters squared	0.0016	square inches	in^2
m^2	meters squared	10.764	square feet	ft^2
m^2	meters squared	1.195	square yards	ac
ha	hectares	2.47	acres	mi^2
km^2	kilometers squared	0.386	square miles	
		VOLUME		
ml	milliliters	0.034	fluid ounces	fl oz
l	liters	0.264	gallons	gal
m^3	meters cubed	35.71	cubic feet	ft^3
m^3	meters cubed	1.307	cubic yards	yd^3
		MASS		
g	grams	0.035	ounces	oz
kg	kilograms	2.202	pounds	lb
Mg	megagrams	1.103	short tons (2000 lb)	T
		TEMPERATURE (exact)		
°C	Celcius temperature	1.8C + 32	Fahrenheit temperature	°F
		ILLUMINATION		
lx	lux	0.0929	foot-candles	fc
cd/m^2	candela/m^2	0.2919	foot-Lamberts	fl
		FORCE and PRESSURE or STRESS		
N	newtons	0.225	poundforce	lbf
kPa	kilopascals	0.145	poundforce per square inch	psi

* SI is the symbol for the International System of Units

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PREFACE

This report presents a summary of the work done by Foster-Miller on the Safety of Passenger Rail Vehicle Dynamics program. It summarizes the technical reports previously prepared on the subject for the Federal Railroad Administration (FRA). These are entitled:

1. Passenger Rail Vehicle Safety Assessment Methodology, Volume I: Summary of Safe Performance Limits – This report presents a methodology based on the OMNISIM simulation program that assesses the safe performance limits of rail passenger vehicles.

2. Passenger Rail Vehicle Safety Assessment Methodology, Volume II: Detailed Analyses and Simulation Results – This report presents detailed results from analytic tools and the OMNISIM program on dynamic response which are used to develop the safe dynamic performance limits of rail passenger vehicles.

3. A Review of Current Technology and Analysis used in Assessing Safety in Rail Passenger Vehicles – This report presents a critical review of literature to identify previous approaches used for the evaluation of rail passenger vehicle safety and the extent of validation of the computer programs through tests conducted.

4. Safety of Passenger Rail Vehicle Dynamics: Summary of OMNISIM Simulation and Test Correlations for Passenger Cars – This report validates OMNISIM and the safety assessment methodology developed for vehicle dynamics.

This final report also presents a formalized safety assessment methodology for passenger rail vehicles.

The work reported here has been performed under the contract DTFR53-95-C-00049 from the FRA. Dr. Thomas Tsai of the FRA is the Technical Monitor. The authors wish to thank Dr. Thomas Tsai and Dr. Herbert Weinstock of the Volpe Center for their inputs during the program.

1. INTRODUCTION

The Federal Railroad Safety Authorization Act of 1994 requires that the Federal Railroad Administration (FRA) establish regulations for minimum safety standards for passenger rail vehicles. Passenger rail vehicles have to safely operate over specified speed ranges on a variety of track geometries: tangent track, curves and spirals. Under a contract with the FRA, Foster-Miller has conducted research on the rail passenger vehicle safety. This research work resulted in four technical reports by the FRA (references 1-4) on the development of the OMNISIM (OMNI simulation) vehicle-track interaction computer code and a generalized safety assessment methodology for new and existing vehicles. The four technical reports referred to deal with the review of literature, development of analytic and computer simulation tools, vehicle safe and unsafe dynamic behavior, vehicle parameter characterization procedures, full-scale testing and correlations with the simulation code, OMNISIM, and validation of the safety assessment methodology. The intent of this final report is to provide a summary of all the work performed as well as to present a finalized version of the safety assessment methodology.

Despite significant published work on vehicle-track interaction, a comprehensive approach for the passenger rail vehicle safety assessment is seldom addressed in the literature. There are several computer codes available, but many of them are "proprietary" to the developer and can be used only as "black boxes," with limited explicit information on their mathematical formulations. The accuracy of these codes is not adequately discussed in the literature. Therefore, Foster-Miller undertook the development of OMNISIM with its source code and mathematical formulation to be made available in the public domain. The results of the code are compared with those from full-scale field testing on vehicles for many track conditions and the correlations are presented in a comprehensive technical report for understanding and quantification of the discrepancies between the code and the test. The use of the code for an assessment of vehicle dynamic safety has been discussed in detail.

This final report is organized as follows. Section 2 gives a summary of the literature reviews conducted. Section 3 presents a summary of research conducted, which includes the development of the OMNISIM simulation code and full-scale tests to validate the simulation code.

Section 4 presents a finalized version of the safety assessment methodology for new and existing rail passenger vehicles. The methodology discusses car parameter characterization, critical track scenarios, and application of the OMNISIM code to derive safe regimes of operations.

Section 5 presents conclusions based on the data generated from the code and the tests and recommendations for easy implementation of the safety assessment methodology.

2. LITERATURE REVIEW

A literature review (3) was conducted with focus on the following issues related to the vehicle-track interaction.

1. Vehicle Derailment Criteria.
2. Available Simulation Codes.
3. Vehicle Testing Procedure.
4. Vehicle Specifications.

The findings from the review are briefly summarized here.

2.1 Rail Vehicle Derailment Criteria

Vehicle derailment is possible due to several track structure failure modes such as gage widening, rail rollover, track shift and buckling. However, attention is focussed on unsafe wheel excursions such as from hunting, wheel climb or lift. For a given type of track, these excursions are attributable to vehicle design.

The wheel climb is an important safety issue in vehicle dynamic behavior. The Nadal criterion is one of the criteria usually applied in the literature for this. It is based on simple equilibrium of the forces between a wheel and rail at the single point of flange contact just prior to derailment, as shown in Figure 1.

The maximum ratio of lateral force, L, to vertical force, V, or maximum L/V on any individual wheel, is generally used in assessing proximity to wheel climb or derailment. Although the subject of much study, its limitations are often not well understood. It is excessively conservative under some conditions, which has led to other criteria being examined.

Nadal assumed that the wheel would be initially in two-point contact with the flange point leading the tread and that the flange contact point would move downwards due to wheel rotation about the tread contact. Derailment is deemed to occur when the downward force of the flange point is equal to the friction force, and the downward motion would cease.

At the point on the flange, the forces acting, L, and V, are applied by the wheel to the rail, and the reactive forces R and F are in the coordinates of plane of contact of the flange with the rail. The equilibrium of the forces at the rail can be analyzed as follows to give the Nadal criterion:

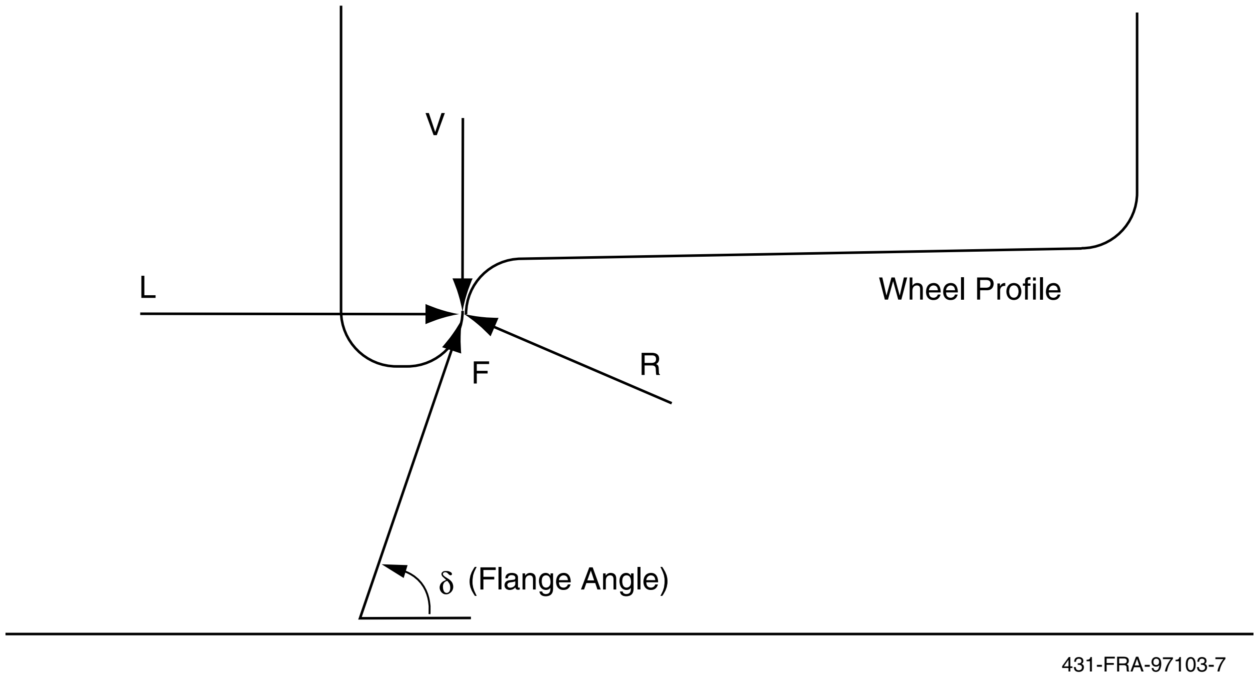


Figure 1. Forces at incipient derailment on the flange

$$\frac{L}{V} = \frac{\tan(\delta) - \mu}{1 + \mu \tan(\delta)}$$

where:

$L = R\sin(\delta) - F\cos(\delta)$

$V = R\cos(\delta) + F\sin(\delta)$

For full slip across the flange, $F = \mu R$

The absolute value of the non-flanging wheel L/V increases to the coefficient of friction as the angle increases, while the flanging wheel approaches the value given by Nadal. The flanging wheel value becomes very large for large negative angles of attack. The conservative nature of the Nadal criterion for small and negative angles of attack can lead to large, but safe, values.

Values recorded during tests regularly exceed those suggested by the Nadal criterion as close to derailment without any apparent danger. It is probable that this has been responsible for much of the subsequent work aimed at improving the correlation between the measured single wheel

L/V and the Nadal value. In particular, the measurements made by Japanese National Railways may well have caused their search for a "duration dependent" relationship. There are, however, other reasons for both the conservative nature of the criterion and its apparent time dependency. The examination of the Nadal criterion reported by Gilchrist and Brickle (5) shows that the Nadal criterion is conservative even in steady-state conditions.

2.2 Conclusions on Rail Vehicle Derailment Criteria

In steady curving, the most dangerous condition appears to occur with no longitudinal creepage across the contact patch of the derailing wheel or during braking. The works of Gilchrist and Brickle ref. (5) and of Weinstock ref. (6) show clearly the dependency of the condition of incipient derailment upon the effective angle of attack of the wheel set. Nadal's criterion is very conservative where the effective angle of attack of the wheel to the rail is small or negative. This may be contributed to by the lateral velocity of the wheel set.

In general, tests have not included measurements of both the lateral wheel set velocity and yaw angle, to permit identification of the true limit for incipient derailment for the circumstance in which the L/V was measured. Much of the published work contains approximations concerning the true angle of the plane of contact with increasing yaw angle of the axle and lift of the derailing wheel. This also tends to render the analysis conservative.

The true measure of proximity to derailment is the wheel set displacement relative to the rails. Although this displacement can be readily predicted in simulation studies, it is not currently possible to measure it accurately during field tests. Wheel/rail forces can be measured with current instrumented wheel technology. The L/V ratios are used as a measure of safety in field testing and evaluation of potential for damage to the track. The L/V ratios do provide an ability to indicate that a derailment will not occur. However, an exceedance of criteria does not necessarily indicate an imminent derailment. Therefore, the Nadal limit is used in the experimental work, and vertical displacement limits in the simulation work to predict the derailment in the vehicle-track interaction studies presented here.

2.3 Available Simulation Models

The following codes are available for use in the United States, although some do not provide source code for the examination of the algorithms they use.

2.3.1 ADAMS/Rail

The ADAMS (Automatic Dynamic Analysis of Mechanical Systems) software package dates back to the early 1970s. In the beginning, ADAMS concentrated mostly on linear analysis, with no real industry demand for nonlinear capabilities. In the early 1990s, with a joint effort between the developer of ADAMS and the Nederlanse Spoorwegen railway, ADAMS/Rail was developed to provide rail vehicle modeling capabilities. The ADAMS/Rail module provides various levels of dynamic analysis, ranging from Level I linearized with no rail wheel set capabilities to Level III including wheel sets and full nonlinear creep contact theory. At the present time, two point contact analysis is not available. ADAMS/Rail is based on an open architecture format, and is customizable to each user's specific needs. The Rail module is in its developmental stage. ADAMS/Rail requires the basic ADAMS module to operate.

2.3.2 A'GEM

The A'GEM program (7), an acronym for Automatic Generation of Equations of Motion, was developed by the Mechanical Engineering Department of Queens University in Kingston, Ontario, Canada. The rail vehicle model is built using a graphical user interface of AutoCad. The program exists in the DOS shell for execution of the processing modules. The post-processing features include wheel unloading, vehicle stability, vehicle curving, and ride quality. The software includes nonlinear analysis. There are doubts about the accuracy of the track profile input. Strengths include time and frequency domain based analysis, good graphics, and animation capabilities.

2.3.3 MEDYNA

MEDYNA (8), for MEhrkorper DYNAmik, is an integrated interactive program containing many model options. It establishes the equations of motion, determines the nominal interaction forces or the static equilibrium and solves the resulting equations in time or frequency domain. A key feature of MEDYNA is a multibody formalism used to generate the equations of motion from user data, provided interactively, describing the mechanical system. The total motion of the bodies is composed of a large motion of one or more reference frames, small rigid body motions relative to these frames, and if necessary small elastic deformations. The motion of the reference frames is of particular value when simulating vehicles driving along any curved guideways.

In the development of MEDYNA wheel/rail-vehicle modeling was a main objective. Three methods are provided for modeling:

1. Quasilinear wheel/rail connecting element.
2. Nonlinear wheel/rail connecting element.
3. Substructure wheel set/track element.

Using method 1 or 2 the computational model of any configuration of wheel set and track can be generated. In contrast to method 1 and 2, the configuration of the wheel/rail model is fixed in method 3. Four different substructure models considering the nonlinearities of the contact physics have been implemented.

2.3.4 NUCARS

NUCARS (New and Untried Car Analytic Regime Simulation) (9) is a general-purpose program modeling for rail vehicle transient and steady-state responses. NUCARS has made significant advances over existing models in providing a single means to predict vehicle response in such varied regimes as hunting, rock and roll, curve entry, and steady-state curving. The flexible structure of the program allows the user to easily model a variety of new or existing vehicle designs. Recent improvements make the model increasingly general and suitable as a universal modeling tool for dynamic systems. Validation includes comparison of test results with a lightweight, two-axle, intermodal car, trailer on standard flatcars, and a track research vehicle with a special track loading axle.

NUCARS version 2.1 includes automatic two-point contact calculation, detection of wheel drop derailment, improved wheel lift force calculation, corrected acceleration spikes, and calculation of pitch degree of freedom during spiral negotiation. The current version 2.2 includes improved flange climb predictions.

2.3.5 VAMPIRE

The VAMPIRE (Vehicle dynAmic Modeling Package In a Railway Environment) software package was developed by the British Rail Research group over a 25 year period. Although VAMPIRE has a highly developed air spring element and models for most passenger equipment well, it does not offer friction wedge elements that are useful to model North American freight trucks. The TTCI (Transportation Technology Center, Inc.) has compared VAMPIRE and NUCARS and discovered that similar simulation results were obtained for a European type passenger vehicle, but significant differences were obtained for a freight vehicle. The program is currently licensed to approximately 25 users in 10 countries.

2.3.6 Model Validations

Although the literature is extensive in regard to theoretical models, and testing performed on a variety of cars, a detailed correlation between the theory and the test data is lacking. Often testing was aimed at generating some sort of data for vehicle acceptance. Validation of generalized system simulation models is involved. This is because the generalized model should be applicable for all track scenarios and types of vehicles. The effort involved in testing all possible scenarios is expensive. There was no simulation model, which was fully demonstrated and correlated with tests with the exception of NUCARS. Although many full-scale tests have been performed at TTC in the past, most of the information on NUCARS is kept proprietary. The British Rail Researchers who developed VAMPIRE have acknowledged that the code needs to be validated. Likewise, the German researchers who developed MEDYNA stated that attempts are being made to validate the code by scale models in the laboratory prior to any full-scale testing.

2.4 Vehicle Testing

The report (3) presents a brief review of rail vehicle testing in the United States. The testing includes:

- Canadian Light Rapid Comfortable (LRC) train set tests to evaluate the passenger train tilting suspension safety.

- Perturbed track tests on locomotives with known history of derailments, to understand the major factors contributing to their unsafe behavior.

- Perturbed track tests on freight cars with the objective of validating an early simulation model named the SIMCAR model.

- Tests on CSX at Starr, OH which were intended to examine forces on weak tracks with the resulting effect of wheel drop.

- Tests at Bennington, NH on curves with sinusoidal alignment variations. These tests were intended to validate SIMCAR and evaluate potential vehicle derailments.

The results of all the testing culminated in the form of a set of track-worthiness tests to evaluate freight vehicle operational safety. These tests are specified in Chapter XI of AAR Specification M-1001 requirements for dynamic testing to assure track worthiness of new and untried freight cars. Hunting, rock and roll, pitch and bounce, yaw and sway, curve entry, steady-state and dynamic curving performance were all included in track-worthiness testing. This experience in freight vehicle testing has been extended to the passenger car safety assessments as described in the subsequent sections.

2.5 Vehicle Specifications

Since 1989, there have been a number of orders for the manufacture of passenger vehicles in the United States. Several of the orders were reviewed in the literature search report (3). These included Mass Bay Transit Authority (MBTA) Redline (1989) and Greenline (1993), Los Angeles Greenline (1992), Metro North (1994), Long Island Railroads (LIRR 1994), New York City Transit Authority (NYCTA 1997), and also AMTRAK high-speed train sets (1996).

The specifications are often incomplete in regard to the safety requirements. The specifications cover design requirements and certain selected safety criteria, as stated in reference (3). The specifications include testing requirements for stability and ride comfort. Very little is mentioned on the safety analysis requirement. Consequently, the manufacturer may utilize empirical or questionable safety analysis at the design stage, and the safety problem may surface during the testing stage after the car is manufactured. Often it is too late or extremely involved to correct the carbody structure then. Hence, the importance of concurrent safety assessments at the design stage cannot be overemphasized for the benefit of all parties involved, namely, car user, manufacturer and the safety enforcing Government agency. A formalized safety assessment methodology for passenger vehicles did not exist prior to the research work initiated by the FRA as presented in the later sections.

3. SIMULATION TOOL

In the application of the Safety Methodology presented in Section 4, it is necessary to perform dynamic simulations in the preliminary assessment of safety of vehicle operations on track with perturbations. It was previously considered that simple analytic solutions would be adequate to predict the vehicle dynamic behavior. Extensive analysis performed in references (1,2) and elsewhere revealed that simple analytic solutions are very restrictive for many situations in revenue service conditions. Hence, a general-purpose code, OMNISIM, has been developed to analyze the vehicle dynamics problems. The specific advantages of this code include:

- The source code will be in the public domain.
- The code accounts for the track flexibility in the lateral and the vertical plane.
- The code has been validated as shown later under a variety of vehicle/track scenarios, and its accuracy has been established.

3.1 OMNISIM

OMNISIM (10) is a multi-body system simulation program, modeling both vehicle and supporting structures in a generalized manner. Each system modeled is represented as a group of bodies, each having its own inertial properties and position in space. These bodies are connected by appropriate interconnections, which may be defined as having special properties, such as suspensions or the rolling connection between the wheel and rail, or being very stiff such as metal-to-metal contact. The program can predict the behavior of the bodies in transient and steady-state response in the time domain. OMNISIM also permits the bodies to be represented as having simple flexible properties. This is useful, for example, to simplify the representation of the torsional rigidity of a vehicle body when negotiating track crosslevel variations.

OMNISIM can work in FPS (foot, pound, second system) or metric units and with measured or analytically constructed inputs or a combination of both. It presents a unified approach to predicting rail vehicle response to a variety of inputs, such as those from the track, actuator or wind forces. Vehicle ride quality may also be assessed. The flexible structure of the input allows the user to model any new or existing vehicle design. In addition to the main processor, pre- and post- processing programs have also been created. Each system is defined in a text file called the definition file, using an appropriate word processor. This file is then preprocessed to the required format and units by the preprocessing program DEFINE. This program rearranges the data and the system units and permits the user to see the system in diagrammatic form, displaying its geometry and characteristics. DEFINE will also display previously pre-processed files.

Means are provided in the definition file to identify the degrees of freedom for each body required in the model. The potential choices include all translational and rotational rigid body motions and the first beamlike free-free flexible modes in twist and in vertical and lateral bending. The interaction of rigid or flexible bodies is defined through hard or soft connections (e.g., metal to metal or suspension elements). The program requires the user to define a vehicle and track system model with inertial and geometric properties, connection characteristics, wheel/rail geometry data, and displacement or force inputs.

There are a number of different types of track and vehicle interbody connections available. Their characteristics range from simple spring and damper pairs in parallel or in series to more complex friction elements. The characteristic of each spring and damper is defined using piecewise linear functions of displacement and velocity, respectively. Hysteresis requires two piecewise linear functions that represent the asymptotic loading and unloading curves. Additional information, such as that which controls the speed of closure to the asymptote in hysteresis, may also be specified.

The present wheel/rail connection assumes no roll rotation of the rail, with the vehicle and track system in the same moving coordinates. This is equivalent to a track model that generates the same behavior at the wheel as the vehicle moves down the track. Although useful in identifying rail motions, further improvements are contemplated. These will allow the rails to be modeled as a stationary continuum with a potential reduction in the number of degrees of freedom, and will release the rail support model from moving with the vehicle.

Each individual wheel/rail connection uses a look-up table representing the required variables at the point of contact between the wheel and rail so that the rolling contact forces may be calculated for the steel wheels on steel rails. The profile data tables are precomputed using a more flexible version of Law and Cooperrider's program WHRAILA (11), and named PROFIT (PROfile FIT). A four-dimensional look-up table of creep force coefficients, according to Kalker (12), is used in determining the forces and moments on each wheel. The rotational speed of the wheel or axle, which may be a solid or have independently rotating wheels, is regarded as a special variable and is required to obtain the wheel/rail forces. The method assumes that the dominant changes in the wheel/rail contact geometry are those due to local relative displacement between each wheel and the rail to which it is connected.

The inputs to the system under study may be measured or analytically constructed in segments using several optional functions. Those representative of laboratory simulation, generally as a function of time, can be formed in the input text file that is read directly by the stepping processor at commencement. A swept frequency sine wave allows vibration testing of a stationary vehicle. However, at the option of the user, the input file may request some or all of the data from a file of either measured or analytically defined histories, formed using the preprocessor called INFORM. This may be filtered and is formatted as digital information in steps along a chosen path or track. If measured data is to be used, it is called into INFORM, from a measured track geometry file. INFORM uses a text setup file to identify the source and preprocess the path and input data that may be of mixed measured and analytic origins.

The short wavelength inputs are regarded as local perturbations, and are introduced as variations in lateral or vertical position of the rails or guideway. For the analytically defined inputs, a repeated shape and amplitude for a segment of the rail may be chosen from a combination of cusps, bends, or sine waves. The long wavelength variations define the overall path and are linearly interpolated from positions along the track at which curvature and superelevation are either chosen analytically or taken from the measured data set. These are transformed into components of the connection strokes, so that the degrees of freedom for each body remain those relative to its local inertial coordinate system. Provision is made to allow both external displacement and forcing inputs to the model. Rail perturbations are an example of displacement inputs; whereas coupler loads due to train action is an example of a forcing input.

For post-processing, PLOTS produces graphs of the output for monitor display or for hardcopy output. TEXTS produces numerical information for viewing or passing to other post-processors, such as a spreadsheet, for further manipulation. Much of the work in this report was postprocessed using a spreadsheet program.

3.2 Description of Rail Test Vehicles

3.2.1 The Rail Test Vehicles

The rail vehicles used for test validation of OMNISIM and the safety methodology are modern bi-level passenger cars. The first has non-equalized trucks while the second has equalized trucks.

The vehicle with non-equalized trucks uses an H truck frame and bolster with outside journal bearings. A schematic of a generic non-equalized truck is shown in Figure 2. The frame is welded steel and consists of two box sections for the side beams and two circular sections for the lateral beams. The truck bolster is a welded box structure that is also used as an auxiliary air supply for the air springs. A center pivot provides the interface between the frame and truck bolster with a nylon bushing.

A radius arm between the truck frame and the journal bearing provides wheel set guidance. The primary suspension is a set of steel coil springs supported on the journal bearing through a rubber pad.

The vehicle uses an air bag secondary suspension. An air spring with a back-up rubber spring is used as an emergency if air is lost. There are stops to limit both vertical and lateral movement. The lateral stops are on the truck bolster and the vertical stops are between the carbody and truck bolsters. Rotary dampers provide damping in the lateral and vertical directions and are connected between the truck bolster and carbody.

The vehicle with equalized trucks uses a rigid H truck frame and bolster with outside journal bearings. A schematic of a generic equalized truck is shown in Figure 3. The frame is cast steel with a center hole and side bearing to accommodate the truck bolster. The truck bolster is a welded structure with a center pin arrangement.

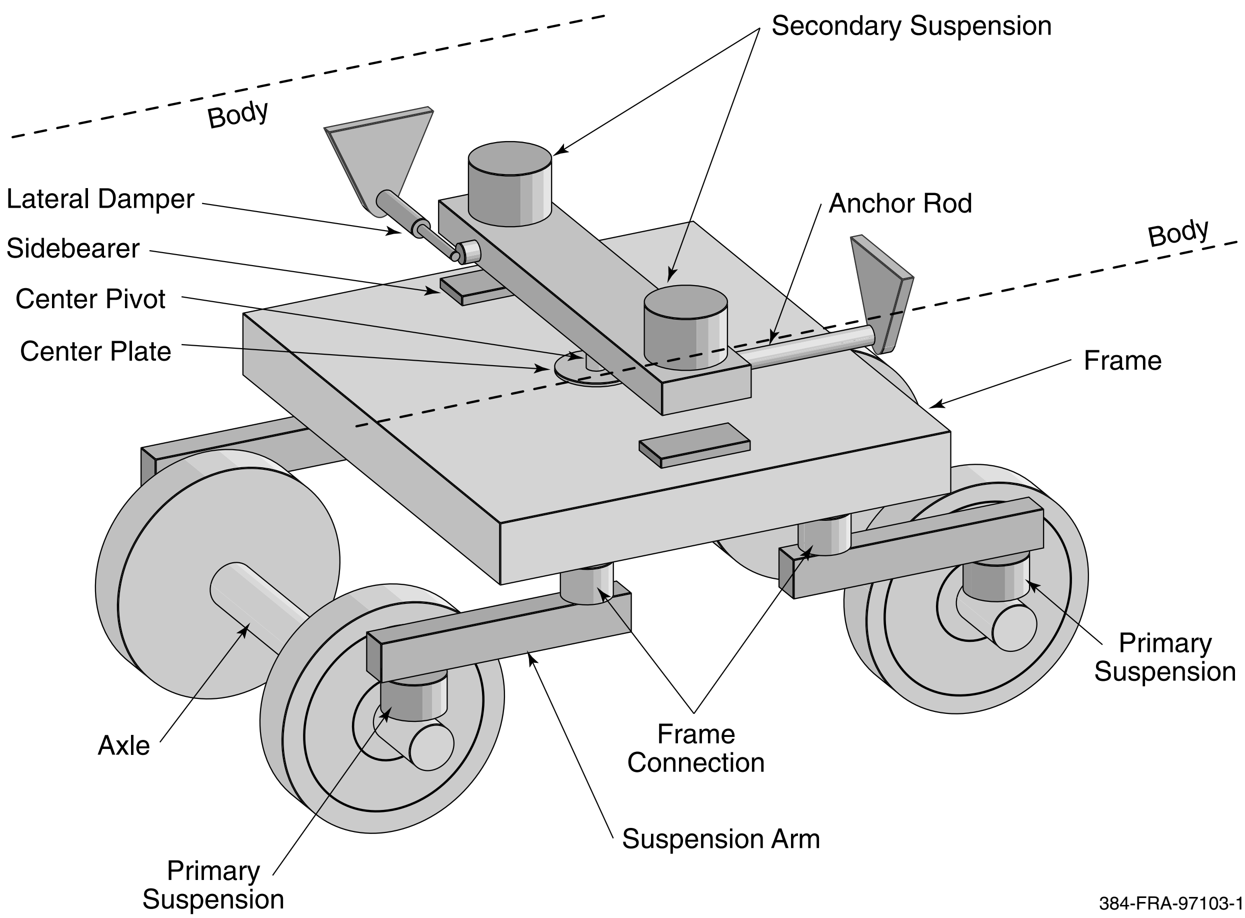


Figure 2. Schematic of non-equalized truck

The pedestal guides provide wheel set guidance for the journal bearing housings. The primary suspension is a pair of steel coil springs supported between each equalizer beam and side of the truck frame. The pedestal guides provide lateral and longitudinal wheel set restraint.

The vehicle uses two coil spring packs for the secondary suspension. The coil springs are between the truck bolster and the underframe of the car. There are lateral and vertical dampers connected between the end of the truck bolster and the carbody for a total of four per truck.

3.2.2 The Rail Vehicle Models

Each rail vehicle is represented by a multi-body model consisting of springs, dampers and masses that represent the carbody, primary and secondary suspensions, trucks, axles, and wheels. The track structure is also represented with springs, dampers and masses. These parameters are identified on the basis of manufacturer's data or measured by testing as explained in Section 4.

Each carbody is represented with lateral, vertical, pitch, yaw, roll, torsional and bending degrees of freedom (DOF). The suspension between the carbody is represented with

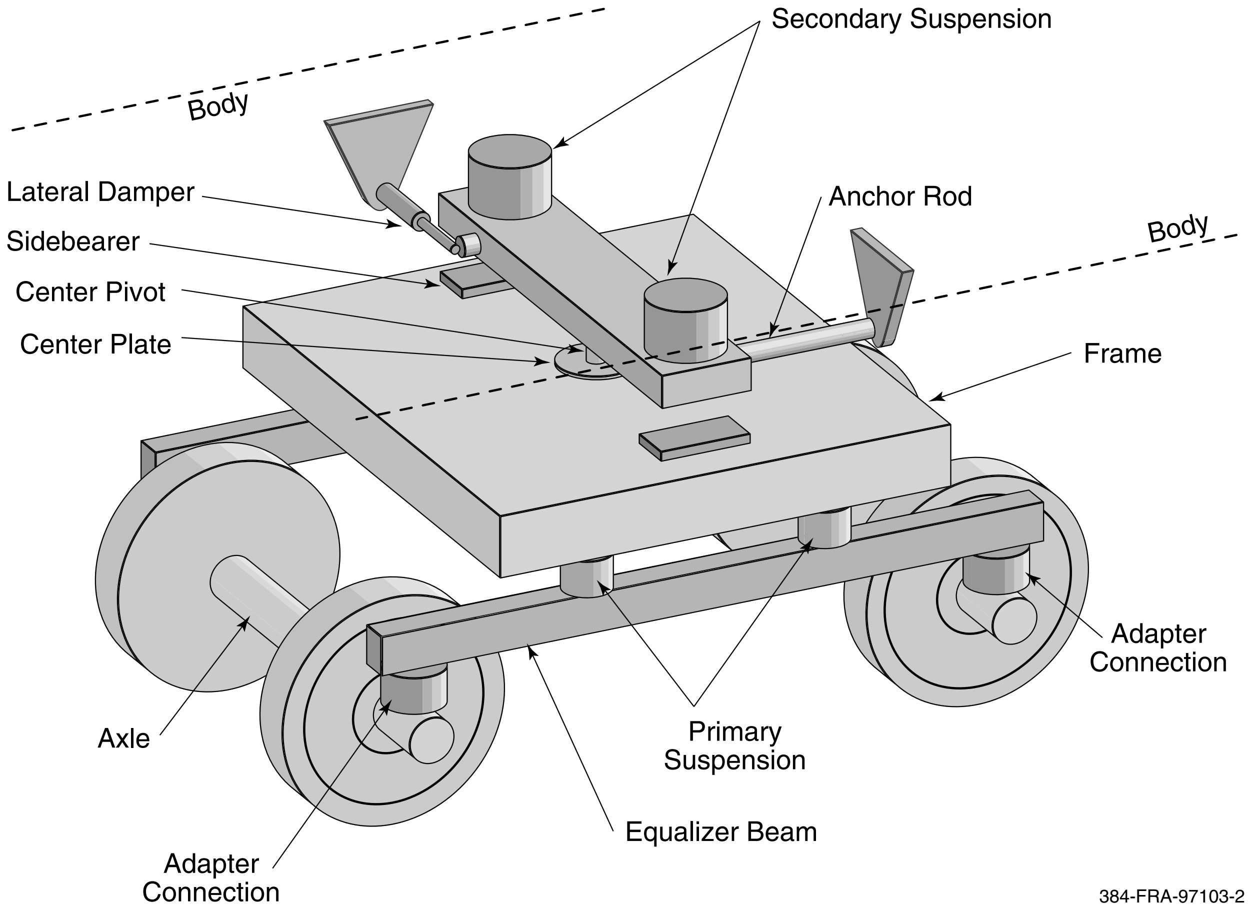


Figure 3. Schematic of equalized truck

longitudinal, lateral, and vertical DOF. The truck is represented with longitudinal, lateral, vertical, pitch, yaw and roll DOF. The primary suspension between the truck and axle is represented with longitudinal, lateral, and vertical DOF. Each wheel set has longitudinal, vertical, lateral, pitch and roll DOF. The wheel/rail contact model uses the Kalker formulation as described in reference (12). The parameters are summarized in Tables 1 and 2 for the test vehicles with non-equalized and equalized trucks, respectively. These parameters are assembled from the manufacturer's data and from car characterization tests as discussed in subsection 4.1.

3.3 Test Scenarios and Procedures

The dynamic tests discussed in the following sections were performed on various test tracks located at the TTC (Transportation Technology Center). Simulations were run using the computer program OMNISIM on a Pentium PC. The track scenarios were modeled and the program was exercised to produce lateral and vertical forces. Time history plots were developed for the simulation and compared to time histories of the test data. Comparisons were made for the maximum lateral force and the minimum vertical force. These were chosen because the

Table 1. Test car characteristics (non-equalized trucks)

Unit	Parameter Description	Value
lb-s^2/in.	Carbody mass	257.91
lb-s^2/in.	Truck bolster mass	5.24
lb-s^2/in.	Truck frame mass	15.86
lb-s^2/in.	Wheel set mass	11.33
in.	Truck wheel base	102
in.	Truck center spacing	714
in.	Wheel radius	18
in.	Carbody center of gravity from top of rail	84.80
in.	Bolster center of gravity from top of rail	30.21
in.	Truck frame center of gravity from top of rail	23.40
in.	Wheel set center of gravity from top of rail	18.00
in.	Transverse secondary spring spacing	79.02
in.	Transverse secondary damper spacing	107.01
in.	Transverse bolster anchor rod spacing	107.01
in.	Transverse wear plate spacing	45.67
in.	Transverse primary spring spacing	79.02
in.	Center of air spring height from top of rail	40.04
in.	Center of lateral damper height from top of rail	33.10
in.	Center of bolster anchor rod height from top of rail	20.95
lb-s^2-in.	Carbody roll moment of inertia	9.89E+05
lb-s^2-in.	Carbody pitch moment of inertia	2.70E+07
lb-s^2-in.	Carbody yaw moment of inertia	2.70E+07
lb-s^2-in.	Truck bolster roll moment of inertia	5.98E+03
lb-s^2-in.	Truck bolster pitch moment of inertia	2.21E+02
lb-s^2-in.	Truck bolster yaw moment of inertia	5.78E+03
lb-s^2-in.	Truck frame roll moment of inertia	1.31E+04
lb-s^2-in.	Truck frame pitch moment of inertia	1.56E+04
lb-s^2-in.	Truck frame yaw moment of inertia	2.83E+04
lb-s^2-in.	Wheel set roll moment of inertia	8.03E+03
lb-s^2-in.	Wheel set pitch moment of inertia	1.49E+03
lb-s^2-in.	Wheel set yaw moment of inertia	8.03E+03
lb/in.	Primary longitudinal stiffness (per wheel)	5.60E+04
lb/in.	Primary lateral stiffness (per wheel)	4.20E+04
lb/in.	Primary vertical stiffness (per wheel)	4.49E+03
lb/in.	Secondary suspension lateral stiffness (per spingset)	1.29E+03
lb/in.	Secondary suspension vertical stiffness (per spingset)	3.82E+03
lb-s/in.	Secondary lateral damping (per truck)	5.60E+02
lb-s/in.	Secondary vertical damping (per truck)	2.80E+02

Table 2. Test car characteristics (equalized trucks)

Unit	Parameter Description	Value
lb-s^2/in.	Carbody mass	302
lb-s^2/in.	Truck bolster mass	5.18
lb-s^2/in.	Truck frame mass	12.69
lb-s^2/in.	Equalizer beam mass	1.10
lb-s^2/in.	Wheel set mass	11
in.	Truck wheelbase	102
in.	Truck center spacing	714
in.	Wheel radius	18
in.	Carbody center of gravity from top of rail	88.9
in.	Bolster center of gravity from top of rail	26.5
in.	Truck frame center of gravity from top of rail	27.89
in.	Equalizer beam center of gravity from top of rail	14.84
in.	Wheelset center of gravity from top of rail	18
in.	Transverse secondary spring spacing	86.04
in.	Transverse secondary damper spacing	107.04
in.	Transverse bolster anchor rod spacing	107.04
in.	Transverse primary spring spacing	86.04
in.	Center of bolster anchor rod height from top of rail	18.96
lb-s^2-in.	Carbody roll moment of inertia	1.40E+06
lb-s^2-in.	Carbody pitch moment of inertia	3.28E+07
lb-s^2-in.	Carbody yaw moment of inertia	3.28E+07
lb-s^2-in.	Truck bolster roll moment of inertia	4.40E+03
lb-s^2-in.	Truck bolster pitch moment of inertia	2.49E+02
lb-s^2-in.	Truck bolster yaw moment of inertia	4.54E+03
lb-s^2-in.	Truck frame roll moment of inertia	1.35E+04
lb-s^2-in.	Truck frame pitch moment of inertia	1.12E+04
lb-s^2-in.	Truck frame yaw moment of inertia	2.38E+04
lb-s^2-in.	Wheel set roll moment of inertia	8.33E+03
lb-s^2-in.	Wheel set pitch moment of inertia	1.16E+02
lb-s^2-in.	Wheel set yaw moment of inertia	8.33E+03
lb/in.	Primary longitudinal stiffness (per wheel)	6.45E+03
lb/in.	Primary lateral stiffness (per wheel)	6.45E+03
lb/in.	Primary vertical stiffness (per wheel)	1.00E+04
lb/in.	Secondary suspension lateral stiffness (per spingset)	4.91E+03
lb/in.	Secondary suspension vertical stiffness (per spingset)	4.13E+03
lb-s/in.	Secondary lateral damping (per truck)	5.60E+02
lb-s/in.	Secondary vertical damping (per truck)	2.85E+02

maximum lateral force coupled with the minimum vertical force produce the largest L/V ratio. Also, maximum carbody lateral accelerations were correlated for the range of test speeds.

3.3.1 Dynamic Test Scenarios (Vehicle with Non-Equalized Trucks)

The car tested was the cab car in a three car consist. The car was tested in both pull and push modes. When the car was pulled the instrumented wheel set was trailing and conversely when the test car was pushed the instrumented wheel set was leading. Data including wheel vertical and lateral forces were measured on each of the two AAR instrumented wheel sets.

3.3.1.1 Vehicle Response to Variations in Vertical Alignment

One of the tests with variations in curved track vertical alignment was conducted to measure the capability of the car to operate safely on low speed curves at permissible speeds and to predict the potential of wheel lift. This test is also referred to as the vertical dip test. Test runs were made on the 5 deg portion of the Wheel/Rail Mechanisms (WRM) loop. The 5 deg curve has a 20 mph balance speed and has concrete ties on granite ballast. A vertical perturbation of 2 in. on the outer rail was installed on the track. Figure 4 shows the vertical dip that was installed in the 5 deg curve. The test was run for a range of speeds (5 to 22 mph) in forward and reverse directions. A video camera was also deployed on the carbody focussing on the primary suspension to capture its movement under potential wheel lift situations.

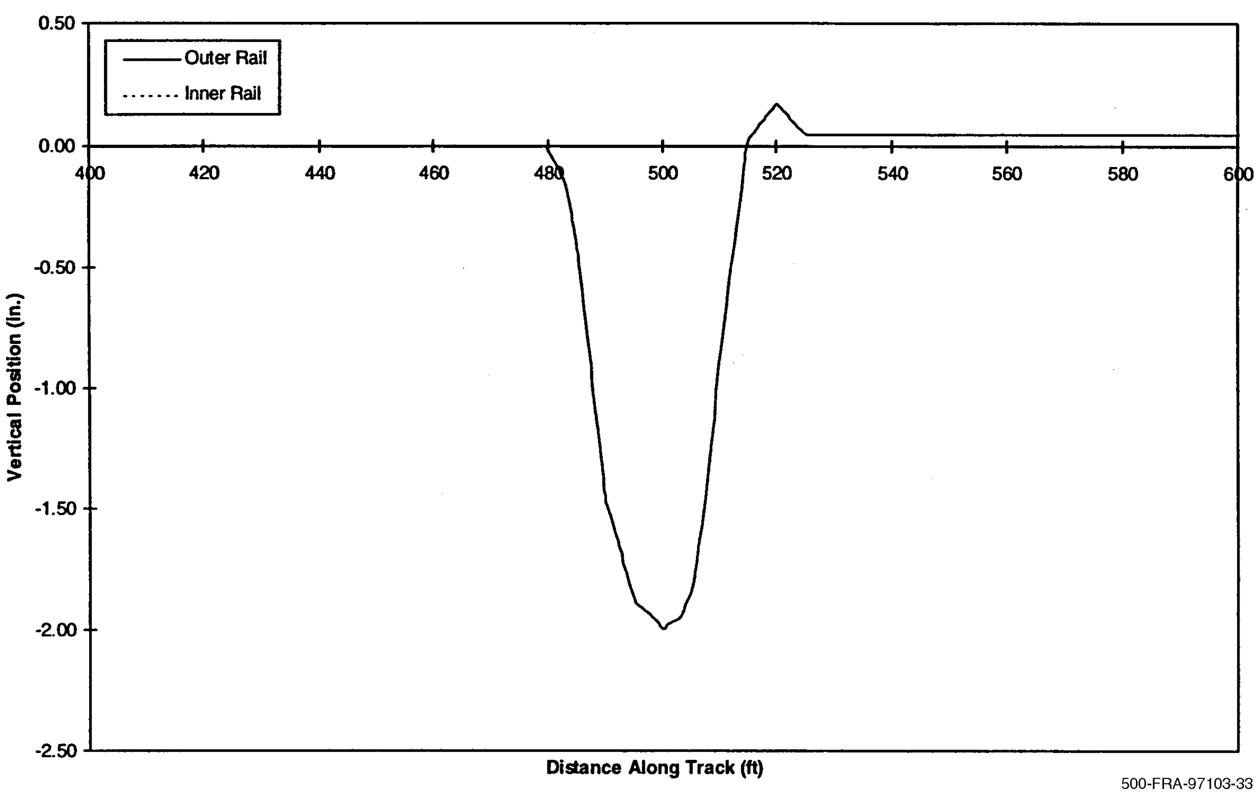


Figure 4. Vertical dip in the 5 deg curve

3.3.1.2 Steady Curving with Spirals

Steady curving tests were conducted to measure the test car capability to operate safely on high-speed curves. The test consist was operated at speeds from the balance speed up to an unbalanced (cant deficiency) condition of ~7 in. Unbalance is defined as the additional height in inches, which if added to the rail in a curve at a certain car speed would provide a single resultant force, (combined effect of weight and centrifugal force on the car) in a direction perpendicular to the plane of the track. A constant 1 deg, 15 min reverse curve with 6 in. superelevation, on the Railroad Test Track (RTT) was used for all tests. Test runs were performed over Class 5 and Class 6 tracks, at speeds of 84 mph (balance speed) to 124 mph (~7 in. unbalance) on the RTT. The tracks have AREA 136 rail and wood ties with cut spike construction on slag ballast.

3.3.1.3 Dynamic Curving

The test for dynamic curving was designed to evaluate safety of the car as it negotiates combinations of vertical profile irregularities and crosslevel in jointed tracks. The resulting forces between the wheel and rail should have an adequate margin of safety against any tendency of the wheel to climb. The 10 deg curved track for dynamic curving consists of five staggered vertical perturbations over a wavelength of 39 ft, with a crosslevel of 0.5 in. (see Figure 5). The latter was achieved by appropriately shimming the rails, which also creates combined gage and alignment variations. The maximum gage of 57.5 in. corresponds to the low points of the outer rail. The minimum gage of 56.5 in. corresponds to the low points on the inner rail. This is shown in Figure 6. The tests were performed at speeds in the range of 10 to 32 mph.

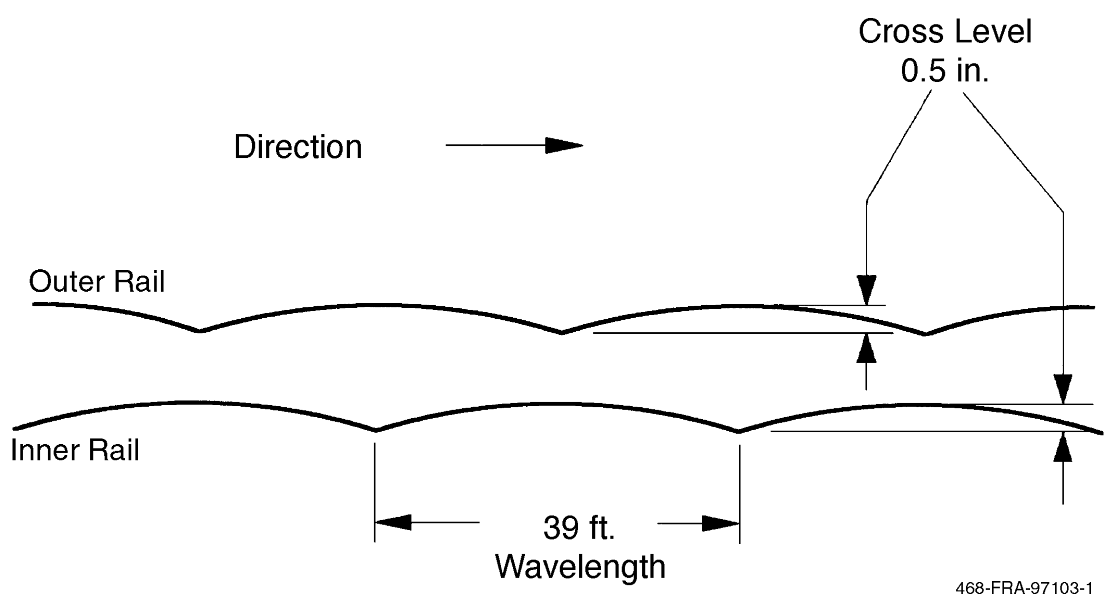


Figure 5. Crosslevel variation for dynamic curving

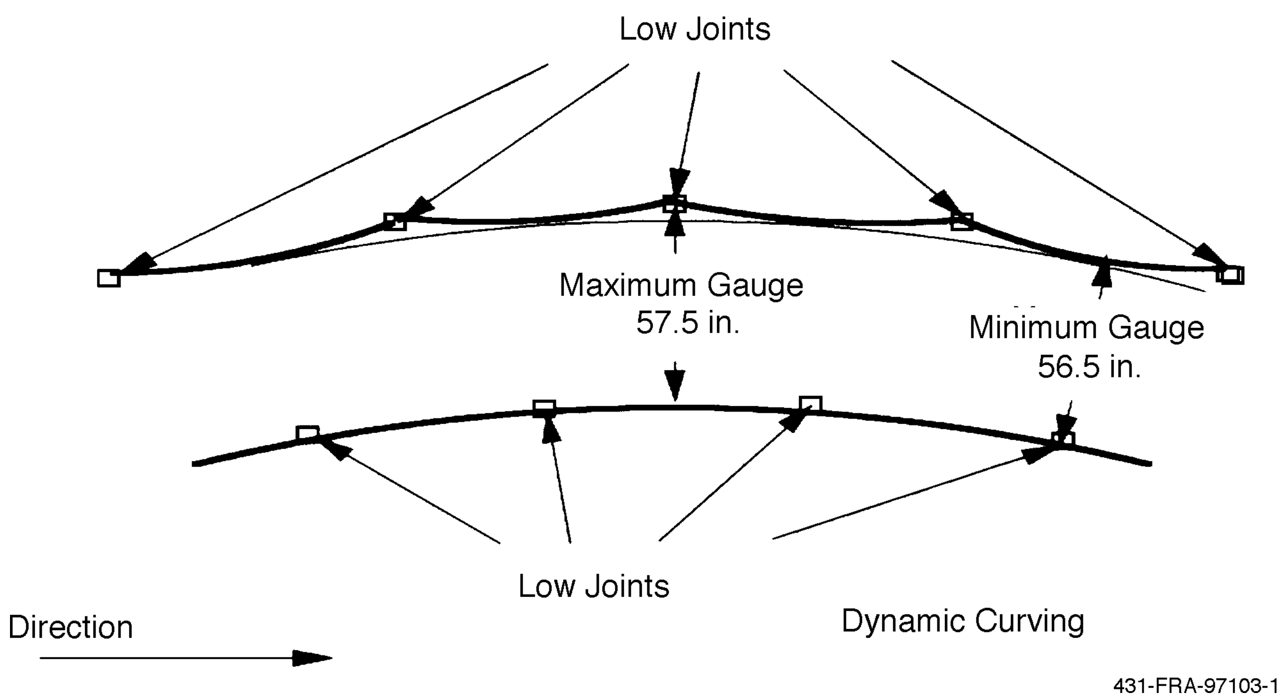


Figure 6. Gage and alignment variation for dynamic curving

3.3.1.4 Yaw and Sway

This test was designed to evaluate vehicle safety in its negotiation of track perturbations that generate yaw and sway oscillations. The resulting forces between the wheel and rail should have an adequate margin of safety against any tendency for the car to derail. The car was excited by a symmetric, sinusoidal track alignment deviation with a wavelength of 39 ft on tangent track. Each simulation included five parallel, lateral perturbations with sinusoidal double amplitude of 1.25 in. peak to peak on both rails and a constant wide gage (see Figure 7). The tests were performed at speeds in the range of 15 to 90 mph, with the intent to capture the resonant speed.

3.3.1.5 Twist and Roll

Successive crosslevel excitation of cars may lead to large car roll and twist amplitudes, which should be limited for car safety assurance. The analyses and tests are required to evaluate the margin of safety against derailment. The test and simulation track sections included 10 vertical perturbations 39 ft apart, staggered each with amplitude of 0.75 in. (see Figure 8). The cusp shaped perturbations were located on each rail to generate the lower and upper roll and twist resonance modes. The tests were performed at speeds in the range of 10 to 70 mph.

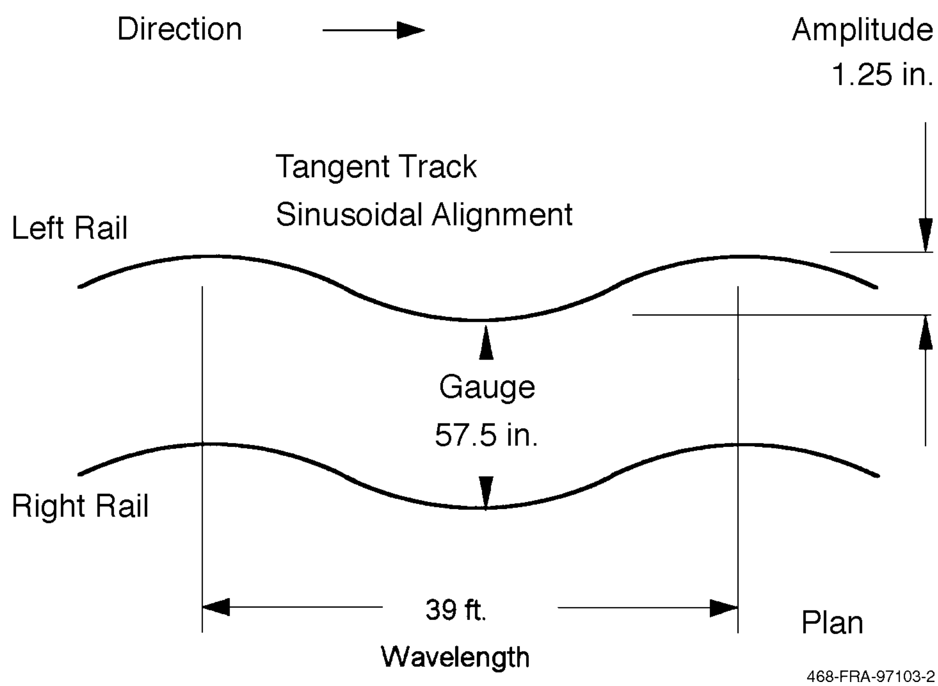


Figure 7. Track alignment variation for yaw and sway

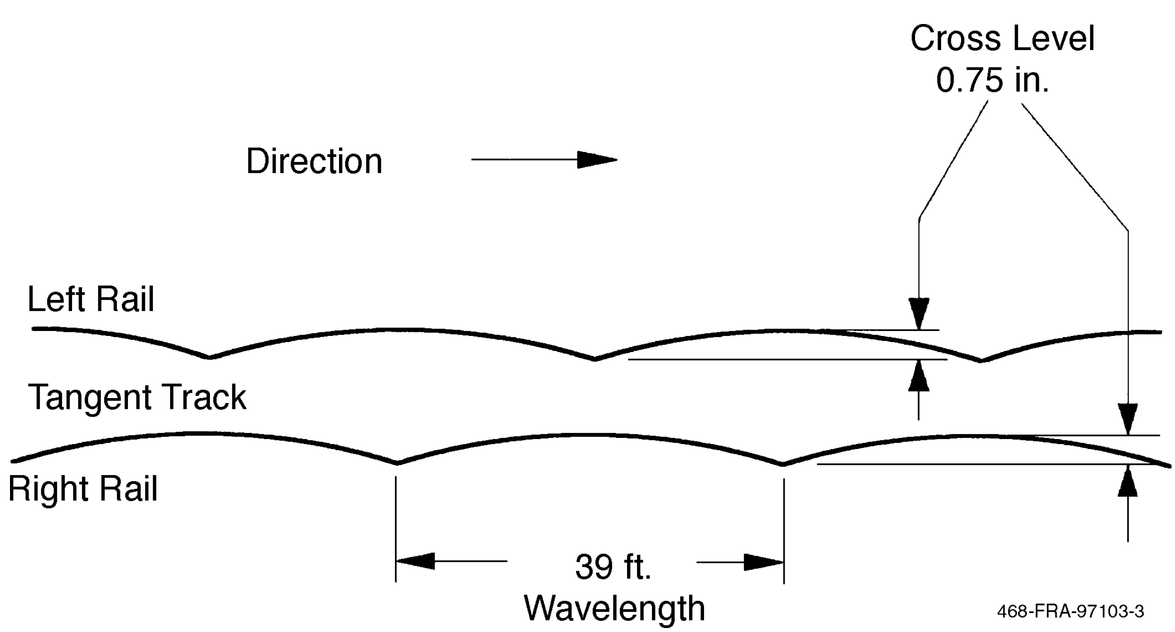


Figure 8. Crosslevel variation for twist and roll

3.3.1.6 Pitch and Bounce

This test was designed to evaluate the car safety as it negotiates track perturbations, which generate pitch and bounce oscillations. An example is a track constructed with parallel joints and/or track structure with changes in the vertical track stiffness. The analyses and tests show the margin of safety in the wheel/rail forces against any tendency for the car to derail. The track included 10 parallel, vertical perturbations, 39 ft apart, with amplitude of 0.75 in. (see Figure 9). The tests were performed at speeds in the range of 10 to 70 mph, ensuring the capture of the resonant speed.

3.3.1.7 Hunting Test with Initial Alignment Defects

The hunting test was conducted to provide information on lateral vehicle stability at various operating speeds on tangent track. Tests were conducted on the Railroad Test Track (RTT) during dry conditions while recording carbody accelerations and wheel/rail forces. The tests were performed at speeds in the range of 80 to 130 mph. The test vehicle was operated over the test track through a single lateral perturbation of 9/16 in. with a 22 ft wavelength to initiate a lateral dynamic response. The installed lateral perturbation is equal on both rails and is shown in Figure 10.

3.3.2 Dynamic Test Scenarios (Vehicle with Equalized Trucks)

For the curving tests (vertical bump, constant curving and dynamic curving) the car tested was the cab car in a four car consist. For the remaining tests (yaw and sway, twist and roll, and pitch and bounce tests) the same car was tested but in a three car consist. The car was tested in both pull and push modes. When the car was pulled the instrumented wheel set was leading and conversely when the test car was pushed the instrumented wheel set was trailing. Data including wheel vertical and lateral forces were measured on each of the two AAR instrumented wheel sets.

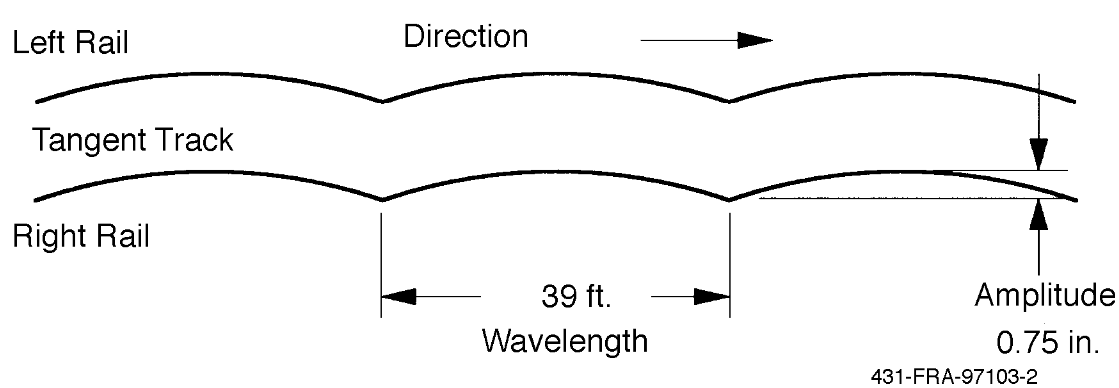


Figure 9. Track surface variation for pitch and bounce

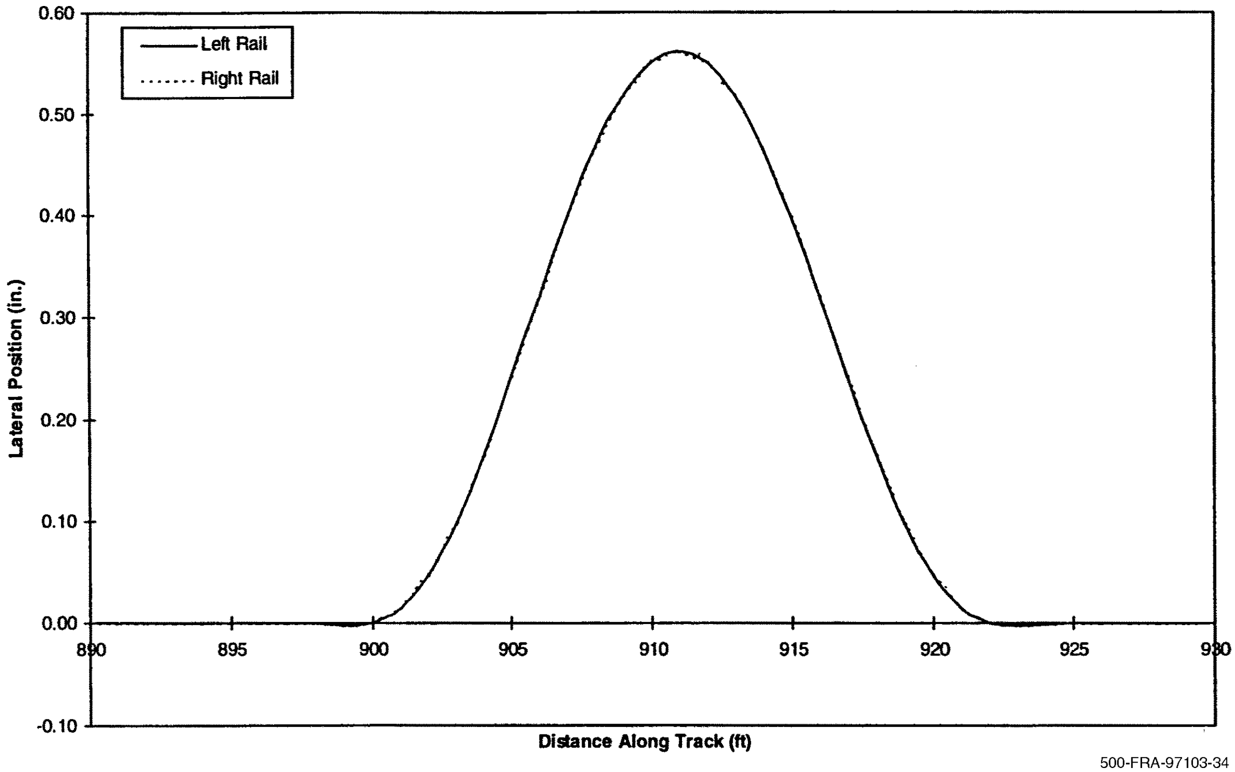


Figure 10. Lateral perturbation for hunting test

3.3.2.1 Vehicle Response to Variations in Vertical Alignment

A test with variations in curved track vertical alignment was conducted to measure the test car capability to operate safely in low speed curves at permissible speeds and to predict the potential of wheel lift. This test is also referred to as the vertical bump test. Test runs were performed on the 7.5 deg curve of the Wheel/Rail Mechanisms (WRM) loop. The 7.5 deg curve has a 24 mph balance speed and has concrete ties on granite ballast. A vertical perturbation of 2 in. amplitude (bump) on the inner rail was installed on the track. Figure 11 shows the vertical bump that was installed in the 7.5 deg curve. The test was run for a range of speeds (10 to 24 mph) in forward and reverse directions. A video camera was also deployed on the carbody focussing on the primary suspension to capture its movement under potential wheel lift situations. A wayside video camera was deployed to capture the occurrence or potential of wheel lift.

3.3.2.2 Constant Curving with Spirals

Constant curving tests were conducted to measure the test car capability to operate safely on tight curves. The test consist was operated at speeds from below balance speed up to an unbalanced (cant deficiency) condition of ~3 in. The tests were run on the WRM loop and consisted of a 7.5 deg curve, a 12 deg curve and a 10 deg curve. All curves have a balance speed of 24 mph. The 7.5 deg curve has 3 in. superelevation, the 12 deg curve has 5 in. superelevation

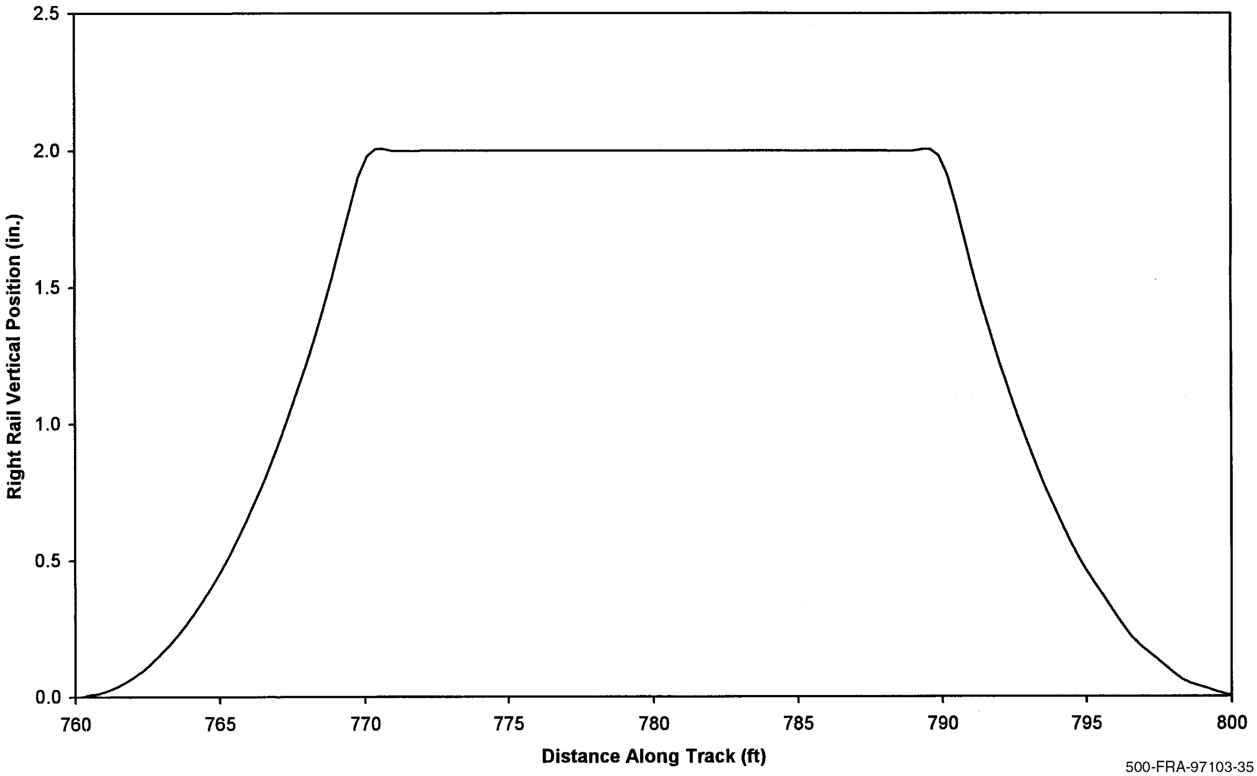


Figure 11. Vertical bump in the 7.5 deg curve

and the 10 deg curve has 4 in. superelevation. Test runs were performed at speeds of 12 mph (except the 12 deg, which was performed at 15 mph), 24 mph (balance speed) and 32 mph (~3 in. unbalance) on the WRM. The tracks have AREA 136 rail and concrete ties on granite ballast.

3.3.2.3 Dynamic Curving, Yaw and Sway, Twist and Roll, Pitch and Bounce

These are the same as described in previous subsections 3.3.1.3 through 3.3.1.6.

3.4 Correlations Summary

Detailed correlations of OMNISIM simulation results with the test data are presented in reference (5). The dynamic response characteristics are correlated for both types of cars and for all the track scenarios in the tests. These include:

- Maximum vertical wheel forces.
- Maximum lateral wheel forces.
- Minimum vertical wheel forces.
- RMS values for the vertical and lateral forces in the zone of track irregularity.
- Maximum carbody accelerations.

The maximum vertical and lateral wheel forces are often used as a measure of quantitative validation of the theory. The maximum lateral force is also a measure of potential unsafe behavior of the car. The minimum vertical wheel force is a measure of potential unsafe wheel lift behavior of the car. The RMS value over the zone of interest represents the overall agreement between simulation and test data. The carbody acceleration is important for ride comfort assessment and is also useful for quantitative validation of theory.

The correlations presented in reference (4) cover all instrumented wheels. For brevity, the results presented in this volume are for the lead outer wheel. Similar correlations are found for other wheels.

3.4.1 Maximum Absolute Forces

The maximum absolute forces are summarized in Tables 3 and 4, respectively, for the vehicles with non-equalized and equalized trucks. The summary results are in the form of maximum absolute vertical and lateral forces of the lead outer wheel for a few speeds in each dynamic scenario. The lateral force levels in pitch and bounce are too small to be of any practical significance and are not shown.

The results in Tables 3 and 4 show good correlation (under 10 percent) in a majority of the test scenarios, especially for the vertical force. In every case, except the lateral force in the dynamic curving scenarios, the simulation is conservative in its prediction.

3.4.2 Minimum Vertical Forces

Tables 5 and 6 show the theoretical and test data for the vehicles with non-equalized and equalized trucks, respectively. Wheel lift is clearly seen in the case of the non-equalized truck car at a speed of 20 mph. The simulation showed that the dynamic wheel load is reduced to about 37 percent of the static value.

3.4.3 RMS Forces

The RMS results are summarized in Tables 7 and 8 for the vehicles with non-equalized and equalized trucks, respectively. The summary results are in the form of RMS vertical and lateral forces of the lead outer wheel for a few speeds in each dynamic scenario. As in the maximum absolute force tables of subsection 3.4.1, the lateral force levels in pitch and bounce are too small to be of any practical significance and are not shown.

3.4.4 Summary

The simulation tool, OMNISIM has been exercised to predict the dynamic response of a vehicle negotiating various track scenarios including vertical and lateral perturbations in the track alignment, steady-state curving, dynamic curving, and truck hunting (vehicle with non-equalized trucks only). Detailed simulation and test correlations are provided in reference (4).

Table 3. Lead outer wheel maximum absolute forces (non-equalized trucks)

Test	Speed (mph)	Curve (deg)	Vertical Force (kips)		Lateral Force (kips)	
			Test	Simulation	Test	Simulation
Vertical Dip	10	5	21.17	19.50	10.15	7.86
Vertical Dip	15	5	23.82	21.49	11.65	7.98
Vertical Dip	20	5	24.57	22.55	13.59	8.11
Vertical Dip	22	5	24.58	22.25	13.29	8.17
Steady Curving	84	1.25	20.67	17.41	7.22	4.32
Steady Curving	90	1.25	21.85	18.03	7.50	4.65
Steady Curving	114	1.25	26.20	21.13	7.37	5.85
Steady Curving	124	1.25	29.86	23.07	7.32	6.63
Dynamic Curving	20	10	19.10	19.31	6.94	13.95
Dynamic Curving	28	10	22.96	21.18	11.12	17.12
Twist and Roll	18.8	tangent	21.40	20.39	1.74	1.24
Twist and Roll	60	tangent	22.04	20.82	2.42	1.97
Pitch and Bounce	20	tangent	17.95	17.62	Negligible	
Pitch and Bounce	60	tangent	19.19	17.91		
Yaw and Sway	20	tangent	20.73	20.33	5.09	1.26
Yaw and Sway	60	tangent	21.66	17.32	6.74	0.46
Hunting	80	tangent	18.85	17.25	0.92	0.12
Hunting	100	tangent	19.26	17.27	1.90	0.19
Hunting	130	tangent	19.55	17.28	3.79	0.31

The following conclusions are reached on the basis of correlations between test and computer simulation:

1. In the case of the vehicle with non-equalized trucks, the vehicle response to a variation in vertical alignment shows that simulation results for vertical forces agree closely with test data at all speeds. The simulation predicts wheel unloading at a certain vehicle speed in close agreement with test data. In the case of the vehicle with equalized trucks, the simulation shows very good correlation with the measured data on the vertical force, as the vehicle negotiated the vertical bump in the rail at different speeds. Consistent with the test observations, no wheel unloading is predicted in the simulation.

2. For both types of vehicles, in the steady curving tests with spirals the simulation has good correlation with the measured vertical forces at all speeds. The correlation for the lateral force on individual wheels is not as good, however, the lateral net axle

Table 4. Lead outer wheel maximum absolute forces (equalized trucks)

Test	Speed (mph)	Curve (deg)	Vertical Force (kips) Test	Vertical Force (kips) Simulation	Lateral Force (kips) Test	Lateral Force (kips) Simulation
Vertical Bump	10	7.5	23.50	20.09	10.46	9.12
Vertical Bump	18	7.5	25.87	22.99	11.14	9.57
Vertical Bump	24	7.5	28.48	23.25	12.73	10.04
Steady Curving	12	7.5	22.67	19.65	8.17	8.42
Steady Curving	24	7.5	23.66	20.72	9.51	9.31
Steady Curving	32	7.5	25.41	22.49	10.68	10.20
Steady Curving	12	10	22.63	20.33	9.27	8.47
Steady Curving	24	10	24.50	21.64	11.51	9.82
Steady Curving	32	10	27.19	23.78	13.38	11.01
Steady Curving	15	12	21.71	20.39	7.82	8.59
Steady Curving	24	12	23.08	21.44	9.79	9.94
Steady Curving	32	12	25.93	24.15	13.08	11.36
Dynamic Curving	18.6	10	24.08	20.88	13.93	17.61
Dynamic Curving	28	10	28.01	26.85	16.84	24.85
Twist and Roll	19.2	tangent	24.08	21.95	3.53	1.32
Twist and Roll	60	tangent	24.66	22.42	4.21	2.02
Pitch and Bounce	30	tangent	22.00	20.57	Negligible	
Pitch and Bounce		tangent	22.08	19.87		
Yaw and Sway	20	tangent	21.31	19.31	4.15	0.36
Yaw and Sway	60	tangent	22.07	19.45	4.33	0.50

forces are in relatively good agreement between simulation and test results both at balance and over balance speeds.

3. In the dynamic curving tests the simulation is able to accurately predict the shape and amplitude of the vertical forces for both vehicles. The simulation has very good correlation for the vertical forces. The correlation for the lateral force is not as good. The distribution of predicted lateral forces have the same shape as test lateral forces but are larger in amplitude, at all test speeds. Both the simulation and test data indicate that a wheel climb condition has not been approached for the conditions studied.

4. In yaw and sway tests, the simulation has very good correlation with test data for the vertical forces for both vehicles. The correlation for the lateral force is not as good. The simulation is able to predict the shape and amplitude of the vertical forces. The

Table 5. Lead outer wheel minimum vertical forces (non-equalized trucks)

Test	Speed (mph)	Curve (deg)	Vertical Force (kips)	
			Test	Simulation
Vertical Dip	10	5	8.78	11.16
Vertical Dip	15	5	7.35	9.37
Vertical Dip	20	5	0.00	6.27
Vertical Dip	22	5	0.00	6.07
Steady Curving	84	1.25	12.97	15.44
Steady Curving	90	1.25	12.68	15.61
Steady Curving	114	1.25	12.97	15.48
Steady Curving	124	1.25	11.52	15.25
Dynamic Curving	20	10	12.51	13.84
Dynamic Curving	28	10	14.85	15.83
Twist and Roll	18.8	tangent	12.98	13.93
Twist and Roll	60	tangent	12.42	13.53
Pitch and Bounce	20	tangent	15.83	16.61
Pitch and Bounce	60	tangent	15.10	16.43
Yaw and Sway	20	tangent	14.01	13.87
Yaw and Sway	60	tangent	15.35	16.76
Hunting	80	tangent	14.78	16.86
Hunting	100	tangent	14.35	16.86
Hunting	130	tangent	13.66	16.83

predicted lateral force distributions have similar shape as in test lateral forces but the simulation under predicts the amplitude.

5. In twist and roll tests, the simulation has very good correlation with test data for the amplitude of the vertical and lateral forces generated by both vehicles.

6. For both vehicles, in pitch and bounce tests, the simulation has good correlation with test data for the vertical force throughout the speed range. The simulation predicts the shape and amplitude of the vertical forces. The predicted lateral forces are small, as are the test results.

7. The vehicle with non-equalized trucks did not show any truck or body hunting oscillations up to the speed limits achieved in the test program (130 mph). This is consistent with the theory, which predicted a hunting speed of well over 200 mph. No hunting test was performed on the vehicle with equalized trucks.

Table 6. Lead outer wheel minimum vertical forces (equalized trucks)

Test	Speed (mph)	Curve (deg)	Vertical Force (kips)	
			Test	Simulation
Vertical Bump	10	7.5	14.18	15.06
Vertical Bump	18	7.5	14.49	15.02
Vertical Bump	24	7.5	16.44	16.12
Steady Curving	12	7.5	15.22	16.29
Steady Curving	24	7.5	16.42	17.61
Steady Curving	32	7.5	17.06	18.04
Steady Curving	12	10	14.83	16.51
Steady Curving	24	10	16.24	18.17
Steady Curving	32	10	17.46	18.56
Steady Curving	15	12	13.32	15.28
Steady Curving	24	12	14.41	16.94
Steady Curving	32	12	16.14	17.65
Dynamic Curving	18.6	10	13.56	14.67
Dynamic Curving	28	10	16.10	16.03
Twist and Roll	19.2	tangent	15.40	15.71
Twist and Roll	60	tangent	15.24	15.53
Pitch and Bounce	30	tangent	17.80	17.14
Pitch and Bounce	60	tangent	17.43	18.02
Yaw and Sway	20	tangent	17.56	18.48
Yaw and Sway	60	tangent	16.97	18.26

Table 7. Lead outer wheel RMS forces (non-equalized trucks)

Test	Speed (mph)	Curve (deg)	Vertical Force (kips) Test	Vertical Force (kips) Simulation	Lateral Force (kips) Test	Lateral Force (kips) Simulation
Vertical Dip	10	5	14.09	15.50	4.88	5.03
Vertical Dip	15	5	14.55	15.75	4.93	5.14
Vertical Dip	20	5	14.55	16.09	7.01	5.28
Vertical Dip	22	5	15.00	16.24	5.80	5.31
Steady Curving	84	1.25	16.95	16.45	1.64	2.59
Steady Curving	90	1.25	17.56	16.80	1.35	2.78
Steady Curving	114	1.25	18.67	18.29	1.61	3.51
Steady Curving	124	1.25	17.35	19.05	1.80	3.91
Dynamic Curving	20	10	16.38	17.04	3.74	7.18
Dynamic Curving	28	10	19.68	18.39	6.04	7.86
Twist and Roll	18.8	tangent	17.26	17.14	0.59	0.50
Twist and Roll	60	tangent	17.10	17.18	0.84	0.82
Pitch and Bounce	20	tangent	17.08	17.10	Negligible	
Pitch and Bounce	60	tangent	17.12	17.10		
Yaw and Sway	20	tangent	17.62	17.10	0.67	0.18
Yaw and Sway	60	tangent	17.48	17.10	0.72	0.14
Hunting	80	tangent	16.58	17.10	0.26	0.05
Hunting	100	tangent	16.92	17.10	0.35	0.05
Hunting	130	tangent	16.58	17.10	0.50	0.06

Table 8. Lead outer wheel RMS forces (equalized trucks)

Test	Speed (mph)	Curve (deg)	Vertical Force (kips) Test	Vertical Force (kips) Simulation	Lateral Force (kips) Test	Lateral Force (kips) Simulation
Vertical Bump	10	7.5	18.75	17.78	6.28	6.97
Vertical Bump	18	7.5	19.36	18.61	6.75	7.46
Vertical Bump	24	7.5	20.42	19.52	6.94	7.83
Steady Curving	12	7.5	18.72	17.94	4.95	7.72
Steady Curving	24	7.5	20.48	19.62	5.50	8.38
Steady Curving	32	7.5	22.36	21.33	6.08	9.06
Steady Curving	12	10	18.51	17.47	6.21	7.81
Steady Curving	24	10	20.92	19.83	7.07	9.00
Steady Curving	32	10	23.56	22.24	7.92	10.05
Steady Curving	15	12	17.86	17.66	5.15	7.49
Steady Curving	24	12	19.61	19.58	6.31	8.45
Steady Curving	32	12	22.30	21.99	7.30	9.52
Dynamic Curving	18.6	10	18.69	18.50	8.45	6.38
Dynamic Curving	28	10	21.73	19.93	9.37	7.03
Twist and Roll	19.2	tangent	19.99	18.90	1.47	0.47
Twist and Roll	60	tangent	19.83	18.92	1.49	0.60
Pitch and Bounce	30	tangent	19.97	18.88	Negligible	
Pitch and Bounce	60	tangent	19.55	18.86		
Yaw and Sway	20	tangent	19.57	18.86	1.27	0.11
Yaw and Sway	60	tangent	19.27	18.86	1.22	0.11

4. SAFETY ASSESSMENT METHODOLOGY FOR PASSENGER RAIL VEHICLES

On the basis of analysis and test research conducted in this program, a safety assessment methodology for rail passenger vehicles has been formalized. Figure 12 shows schematically the six steps involved in the process. These are described in detail in the following paragraphs.

4.1 Assemble Vehicle Parameters

The required vehicle parameters are shown in Table 9. The table also includes the method of measurement (direct measurement or manufacturer's data) or using characterization tests. Rail vehicle suspension characterization tests and rail vehicle rigid body modal tests are particularly complicated and will be described in the following paragraphs.

4.1.1 Rail Vehicle Suspension System Characterization

The purpose of this test is to measure the load-displacement characteristics for the primary and secondary suspensions. Load measuring instrumented rails combined with displacement transducers can be used to obtain stiffness data (force versus displacement) for each suspension element. The method typically used to measure the vertical suspension characteristics is shown in Figure 13, where the carbody is unloaded and deflections are measured on the primary and secondary suspension elements. Unloading of the wheels is achieved using pneumatic floor jacks and overhead cranes. Load cells mounted in-line with the applied force, and displacement transducers mounted across each suspension element can be used to obtain stiffness data (force versus displacement). The lateral and longitudinal suspension characterization tests can be determined similarly. In the longitudinal test (see Figure 14), the load is applied longitudinally at the truck for the primary suspension only. In the lateral test, the load is applied laterally at the truck and reacted on the other side with a large reaction mass. Prior to the execution of the truck characterization tests, all dampers (vertical and lateral) are to be removed from both trucks. Additional truck modifications are also required to secure movement of the spring plank on the truck under test.

4.1.2 Rail Vehicle Rigid Body Modal Characteristics

The modal characterization tests can be conducted to obtain rigid body modal frequencies and damping for each dynamic vehicle mode. Excitation for all modes are to be performed manually at selected locations, collecting data from strategically placed accelerometers. Carbody mass moments of inertia and the center of gravity height can be calculated using the

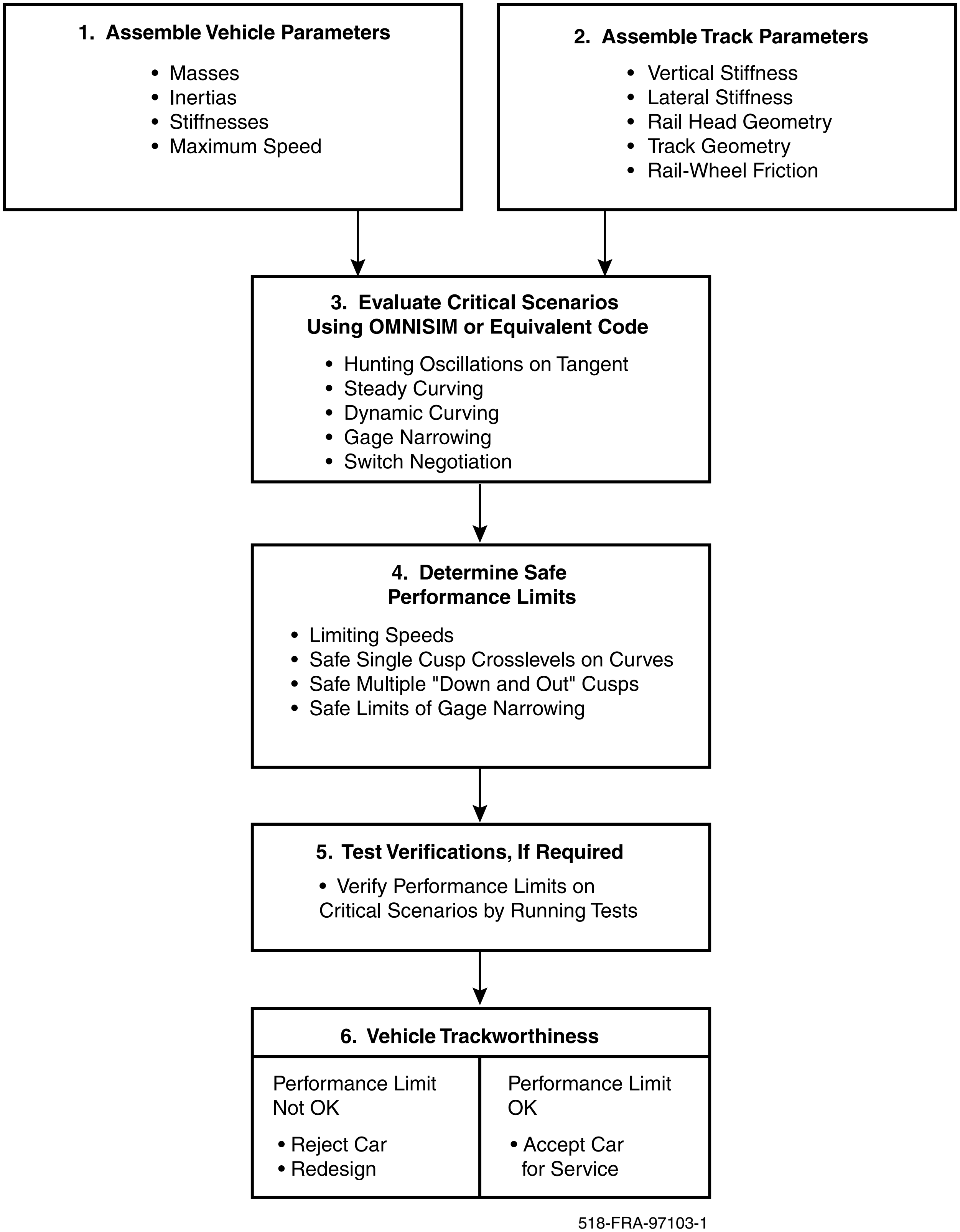


Figure 12. Overall safety assessment methodology for passenger vehicles

Table 9. Required vehicle parameters

Parameter Description	Directly Measured (D)/Manufacturer Data (M)	Characterization Test
Carbody mass	D/M	
Truck bolster mass	D/M	
Truck frame mass	D/M	
Equalizer beam mass	D/M	
Wheel set mass	D/M	
Truck wheelbase	D	
Truck center spacing	D	
Wheel radius	D	
Carbody center of gravity from top of rail		Rigid Body Modal Test
Bolster center of gravity from top of rail	D	
Truck frame center of gravity from top of rail	D	
Equalizer beam center of gravity from top of rail	D	
Wheelset center of gravity from top of rail	D	
Transverse secondary spring spacing	D	
Transverse secondary damper spacing	D	
Transverse bolster anchor rod spacing	D	
Transverse primary spring spacing	D	
Center of bolster anchor rod height from top of rail	D	
Carbody roll moment of inertia		Rigid Body Modal Test
Carbody pitch moment of inertia		Rigid Body Modal Test
Carbody yaw moment of inertia		Rigid Body Modal Test
Truck bolster roll moment of inertia	M	
Truck bolster pitch moment of inertia	M	
Truck bolster yaw moment of inertia	M	
Truck frame roll moment of inertia	M	
Truck frame pitch moment of inertia	M	
Truck frame yaw moment of inertia	M	
Wheel set roll moment of inertia	M	
Wheel set pitch moment of inertia	M	
Wheel set yaw moment of inertia	M	
Primary longitudinal stiffness (per wheel)		Longitudinal Stiffness Test
Primary lateral stiffness (per wheel)		Lateral Stiffness Test
Primary vertical stiffness (per wheel)		Vertical Stiffness Test
Secondary suspension lateral stiffness (per spingset)		Lateral Stiffness Test
Secondary suspension vertical stiffness (per spingset)		Vertical Stiffness Test
Secondary lateral damping (per truck)		Rigid Body Modal Test
Secondary vertical damping (per truck)		Rigid Body Modal Test
Wheel Profile Measurement		Wheel Profilometer

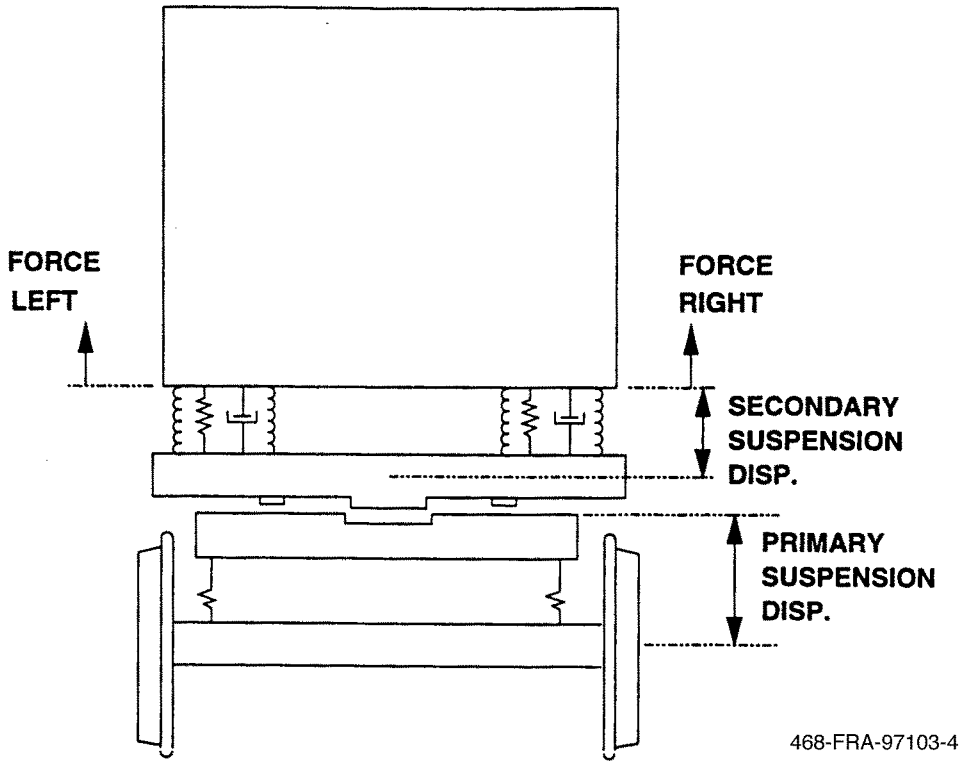


Figure 13. Vertical suspension characterization test

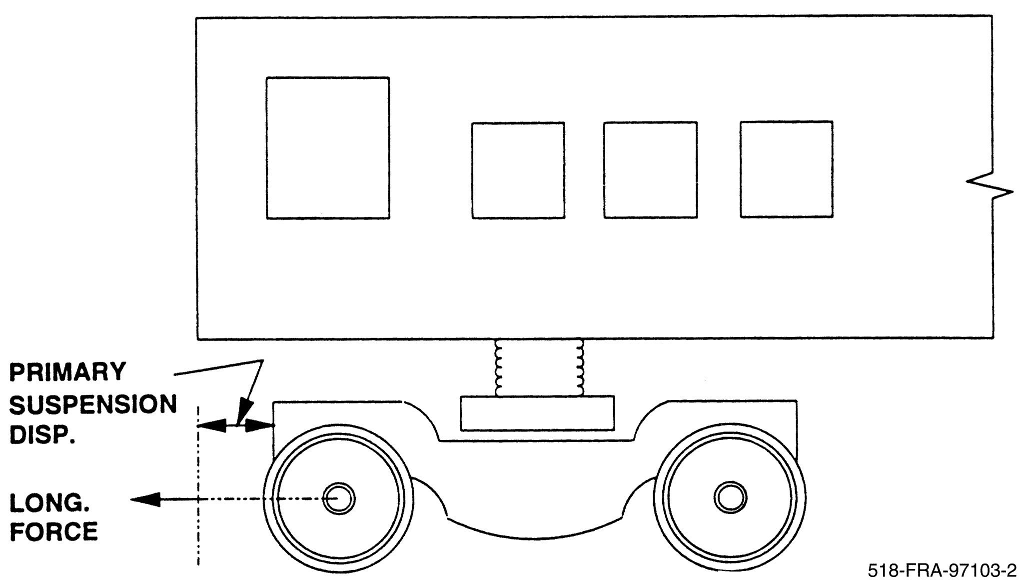


Figure 14. Longitudinal suspension characterization test

measured natural frequencies and a parameter identification algorithm. The primary dynamic modes of vibration tested are shown in Figure 15. By exciting the carbody at selected locations, these vibration modes can be generated and the frequency response measured. The damping coefficients can be evaluated by measuring the hysteresis of force versus displacement plots for each suspension element.

4.1.3 Wheel Profile Measurement

For accurate inputs to OMNISIM or other equivalent code, representative wheel profile shapes are required which can be measured using a portable profilometer. The wheel profilometer is magnetically attached to the wheel while a digitization probe rolls over the wheel surface. Data is obtained using a notebook PC for graphical display and data processing for modeling requirements. Wheel profile processing also includes measurements of wheel diameter.

4.2 Assemble Track Parameters

The required characteristics shown in Table 10 include vertical track stiffness, lateral track stiffness, lateral rail stiffness, wheel and rail friction, track geometry and rail profile. The tests are required at an adequate number of locations.

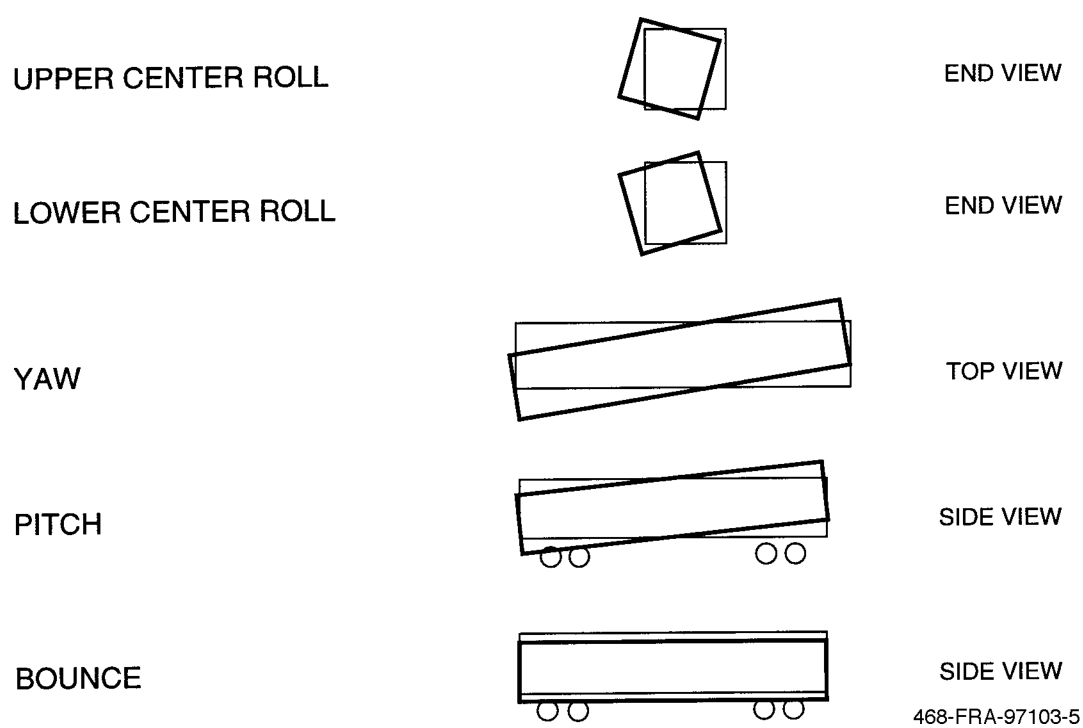


Figure 15. Rigid body modes

Table 10. Required track parameters

Parameter	Assumed	Special Test
Vertical Stiffness		Rail deflection under the wheel load
Lateral Stiffness		Use Single Tie Push Test (STPT)
Track Imperfection/Geometry	From Class of track, assume allowable profile variations, misalignments and crosslevel	Track Geometry Car
Rail-wheel Friction		Tribometer
Rail Profile		Profilometer

4.2.1 Track Vertical Stiffness

The vertical track stiffness can be measured using any car with known axle load. The vertical track stiffness is calculated by using the measured vertical track deflection under the applied loads.

4.2.2 Track Lateral Stiffness

The lateral resistance of the ties can be measured using the Single Tie Push Test (STPT) fixtures for wood and concrete ties. The measurements are to be made on the test tracks in various locations. A single tie push measurement involves removing the spikes, tie plates, and anchors from the selected test tie. Next, a hydraulic actuator is attached to the tie and pushes laterally against the rail until approximately a 2 in. displacement between tie and rail is obtained. The displacement is measured with a string pot. In order to determine the tie to ballast friction, single tie push measurements should be made at several locations on ties with vertical load.

4.2.3 Track Imperfections/Geometry

The allowable profile variations, lateral misalignments and crosslevels are defined in the FRA specifications for each class of track. These can be assumed in the safety evaluations. Alternately, track geometry car can be used to map the track geometry along a given route from which the maximum track irregularities can be determined.

The track geometry data related to curvature, superelevation, profile and gage variations, lateral misalignments and crosslevels are important inputs in the construction of limiting track scenarios to be examined for safety. These will be discussed in detail in subsection 4.3.

4.2.4 Rail-Wheel Friction

The friction coefficient between the wheel and the rail is an important parameter in the vehicle dynamic behavior, and it can vary depending on wet and dry conditions and rail lubrication if used on curved tracks. The friction coefficient can be measured at selected locations using a tribometer.

4.2.5 Rail Profile

Rail profile which depends on the type of rail and wear under service conditions needs to be defined in the vehicle safety assessment methodology. The profile can be measured at selected locations using a profilometer.

4.3 Evaluate Critical Scenarios using OMNISIM or Other Code

In the application of the safety assessment methodology, several scenarios should be studied. As a minimum, the scenarios in Table 11 should be included. Some of them may be found "critical" in the sense that the vehicle operations under the scenarios may be unsafe or marginally safe. Vehicle testing on such critical scenarios may be required to validate the simulation code predictions.

The following describe the scenarios listed in Table 11 which must be analyzed in the safety assessment.

4.3.1 Hunting Oscillations on Tangent

The maximum allowable vehicle speed should be under the critical speed that can trigger truck/carbody oscillations. The critical speed can be evaluated using OMNISIM. Alternately, the vehicle dynamic response at its maximum speed under a maximum allowable lateral misalignment can be simulated, and the rate at which the peak lateral forces decay with the distance from the misalignment location can be studied, whether the hunting oscillations are sustained or not. It is desirable to simulate not only the new vehicle characteristics (suspension and wheel profile) but also the anticipated degraded suspension characteristics and worn wheel profile due to service conditions.

4.3.2 Lateral Misalignment Negotiations on Tangent

Vehicle response under a lateral misalignment (with allowable amplitude and wavelength) should be studied to assure that L/V generated at the maximum vehicle speed does not exceed the Nadal limit.

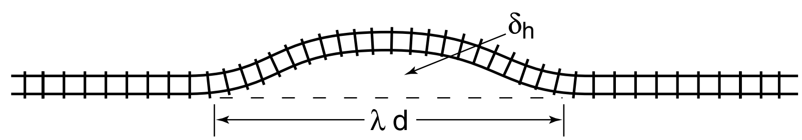
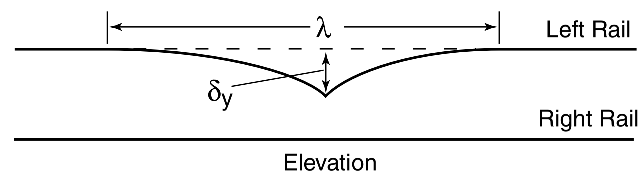
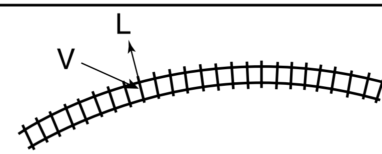
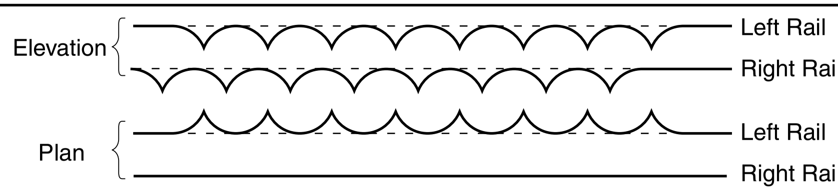
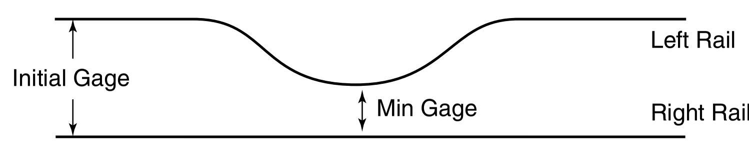
4.3.3 Single Cusp Negotiation on Tangent

Vehicle response under a vertical cusp or dip of one of the rails (with allowable amplitude and wavelength) should be studied to assure that the wheel lift or a significant reduction in wheel vertical force (~10 percent of static load) will not occur.

4.3.4 Steady Curving

The L/V generated by vehicles running at maximum speed on curves should be evaluated to assure that the ratio does not exceed Nadal's limit on wheel climb.

Table 11. Scenarios for safety assessment

	Scenario	Track Shape	Safety Concern
1.	Hunting (Tangent)		Large Lateral Excursions High L/V
2.	Lateral Misalignment Negotiation on Tangent		High L/V
3.	Single Cusp on Tangent		Wheel Lift
4.	Steady Curving		L/V Exceeding Nadal's Limit
5.	Dynamic Curving		Wheel Climb and Lift
6.	Gage Narrowing		Wheel Climb
7.	Switch Negotiation	AREMA No. 8,10 as the Case Maybe	Wheel Climb

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4.3.5 Dynamic Curving

Dynamic response of vehicles negotiating single or multiple "down and out" cusps (vertical and lateral irregularities, see Table 11) must be evaluated to assure that wheel climb or lift does not occur.

4.3.6 Gage Narrowing

Passenger rail vehicles will have to safely negotiate gage variations that can occur on revenue lines. For a given variation of gage, there is a maximum speed limit above which wheel climb can occur. This must be evaluated.

4.3.7 Switch Negotiation

The geometry of a switch on the revenue lines is defined by its type and number (e.g., AREMA No. 8). The safe maximum speed of the vehicle as it negotiates the switch should be evaluated using the OMNISIM.

4.4 Determine Safe Performance Limits

It is important to perform parametric studies, varying vehicle speed, track curvature, amplitude and wavelength of track scenarios to facilitate the development of safe performance limits in the form of graphical charts or tables for all the track scenarios in subsection 4.3. The limits can be expressed in terms of maximum safe speeds, safe amplitudes, wavelengths of cusps, misalignments at given speeds, or margin of safety on Nadal's limit.

4.5 Test Verifications, if Required

Vehicle running tests on critical track scenarios in which safety is not assured by analytical predictions must be conducted in vehicle speed increments from a low value. The predictions at lower speeds must be confirmed and the test results extrapolated to unsafe high speeds and correlated with the results. Some vehicle qualification tests may still be performed even if they do not predict unsafe vehicle behavior. The data from the vehicle qualification tests can be used to validate the predictions.

4.6 Vehicle Trackworthiness

If the test validations confirm the potential unsafe behavior of the vehicle, it must be considered not trackworthy. The vehicle must not be allowed on revenue lines and the manufacturer should redesign, which may include change of suspension characteristics. Sometimes, simple changes in suspension may not improve the car safe performance. Hence, it is prudent to stipulate to the manufacturer to apply the safety assessment methodology during the design stage rather than wait till the vehicle is manufactured.

5. CONCLUSIONS AND RECOMMENDATIONS

5.1 Conclusions

1. The literature review revealed that the buyer specifications to the rail vehicle manufacturers are often incomplete in regard to the assurance of vehicle dynamic safety.

2. The safety assessment methodology for passenger rail vehicles requires evaluation of dynamic behavior under several track scenarios. The scenarios include allowable track perturbations in the lateral and vertical planes per the existing FRA track specification. Vehicle safety against hunting, wheel climb, wheel lift, excessive carbody pitch, yaw, and roll oscillations as well as lateral and vertical accelerations need to be evaluated from the vehicle dynamic response.

3. The dynamic behavior evaluation should be performed using a general purpose simulation code such as the OMNISIM. This code has certain advantages which include: a) it is validated for a variety of track scenarios, b) its code is available in the public domain, and c) it has a flexible track model. A general purpose simulation code provides a convenient tool in the analysis of several track scenarios, which may be difficult to accomplish by conventional analytic methods.

4. The application of OMNISIM or equivalent code requires several vehicle and track parameters. Some of the parameters can be readily measured and others need special tests, which are reasonably standardized. Certain vehicle parameters can be obtained from the manufacturer's published data.

5. From the simulation results, one can determine the problem areas of the vehicle, if any. If the vehicle has already been built, the problem areas can be confirmed by testing at reduced speed and extrapolating the test data to the maximum speed of interest. If the car is already built, it is often very difficult to change its suspension and/or other parameters. Therefore, the design should ideally be conducted with an iterative process using OMNISIM or other code until satisfactory results are obtained prior to manufacturing.

6. The overall accuracy of OMNISIM when compared with the test data seems to be reasonable. Generally, excellent agreement is obtained in the vertical response and wheel lift can be predicted fairly accurately. Predictions of lateral force are less accurate, although the forces involved are small to be accurately measured even by advanced instrumented wheel sets.

5.2 Recommendations

1. The FRA should make available to the rail car buyers the safety analysis methodology as presented here for usage in their procurements.

2. The OMNISIM code is in the DOS operating system. It should be upgraded to the Windows operating system for ease of use.

3. The Windows version should have a library of vehicle parameters, wheel profiles and built-in track scenarios. The program should be automated for ease of application as an expert system.

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