D. Gray

Contract No. DTRS-57-D-00027 TTD No. VA3204

REPORT

Safety-Related Analysis and Simulation of High Speed, Guided, Ground Transportation Systems

Task 1, Volume I: State of the Art

To

The Volpe National Transportation Systems Center

Cambridge, Massachusetts

March 9, 1995

Battelle

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12 - Safety

Technical Report

TTD No. VA 3204

Contract No. DTRS-57-D-00027

Safety-Related Analysis Speed, Guided, Ground Transportation Systems

Task 1, Volume I: State of the Art

by

Battelle
CMRI
Clemson

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Acknowledgement

This report was prepared by Battelle Memorial Institute and its subcontractors under the Volpe National Transportation Systems Center Contract No. DTRS-57-93-D-00027, entitled Vehicle Guideway and Terminal Systems. Ongoing technical support to Battelle is being provided by Carnegie Mellon Research Institute (CMRI), Clemson University, and Bechtel Corporation. The scope of this work is to develop, apply, and maintain analytical models of HSGGT systems to support the FRA's safety mission.

The present document is submitted in fulfillment of base task requirements, as defined under Technical Task Directive (TTD) No. VA3204, "Dynamic Analysis Support for HSGGT Systems." A primary requirement of this TTD is that the "contractor... shall define modeling and analytical requirements for computational tools that will predict the safety-related dynamic performance of a vehicle or train operating over flexible guideways having irregularities and changing alignments."

The work has benefited from technical contributions and valuable insights of numerous individuals, including Mr. Donald Ahlbeck, Mr. Jeffery Hadden, Mr. James Tuten, and Mr. Richard Rice of Battelle (Columbus, OH), Dr. Michel Thomet of Bechtel (San Francisco), Dr. Imtiaz Haque and Dr. Harry Law of Clemson University (Clemson, SC), Dr. Mark Nagurka and Dr. Richard Uher of CMRI (Pittsburgh, PA), and Dr. Herbert Weinstock and Mr. David Tyrell of VNTSC (Cambridge, MA).

Executive Summary

High Speed Guided Ground Transportation (HSGGT) vehicle/guideway systems—both high speed rail (HSR) and magnetically levitated (Maglev)—are being considered for deployment in the United States. The Federal Railroad Administration (FRA) has the mission of ensuring the safety of such systems. To support this mission, it is essential that credible models and analytical procedures be available and readily accessible for predicting safety-related dynamic behavior. The purpose of this document is to define requirements for computational tools that will predict the safety-related dynamic performance of high speed rail or Maglev trains operating over flexible guideways having irregularities and changing alignments. Battelle is under contract* with the Volpe National Transportation Systems Center to perform Technical Task Directive (TTD) No. VA3204, "Dynamic Analysis Support for HSGGT Systems." The present document is submitted in partial fulfillment of base task requirements, as defined under Task 1 of VA3204. The scope of this work is to develop, apply, and maintain analytical models of HSGGT systems to support the FRA's safety mission.

The HSGGT systems under consideration for operation in the U.S. differ from conventional passenger trains in two important respects: they operate at speeds considerably higher than conventional rail vehicles, and some of their design features are fundamentally different from those of conventional vehicles. The combination of higher operating speeds and new vehicle designs raises the critical safety issue of how existing safety standards must be extended to ensure safe operation in the United States.

For the purposes of this work, HSGGT systems are defined as HSR and Maglev systems designed for operation at speeds of at least 150 mi/h (240 km/h). Examples of HSR systems are the French TGV, German ICE, Swedish X2000, Italian ETR 450 and ETR 500, and Japanese Shinkansen trains. Examples of Maglev systems are the German TR07, Japanese MLU series, and the four System Concept Designs (SCDs) proposed under the National Maglev Initiative (NMI).

There are three basic goals that must be met to achieve adequate safety-related dynamic performance:

1) Safety against vehicle ***loss of guidance*** (e.g., derailment),

2) ***Ride quality*** that ensures passenger safety and comfort, and

3) ***Structural integrity*** that ensures adequate durability and reliability of the hardware comprising the vehicle/guideway system.

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* Vehicle Guideway and Terminal Systems Contract No. DTRS-57-93-D-00027.

Numerous evaluations must be performed to qualify a HSGGT system as meeting these goals. Modeling and dynamic simulation are an important part of these evaluations. The purpose of this document is threefold:

- to define modeling and simulation requirements for HSGGT systems,
- to provide an overview of available computational tools that meet these requirements, and
- to provide recommendations on a means for making the appropriate and/or recommended tools available for safety-related analysis.

Section 2 addresses "Safety-Related Dynamic Performance Considerations", including a wide variety of critical issues regarding HSGGT dynamic analysis and simulation. This Section defines the analysis objectives for the modeling effort by defining the dynamic performance objectives to be examined.

Section 3 addresses the "System Configuration" part of the methodology; it provides examples of present-day HSGGT systems, including general information on their vehicle and truck designs, primary and secondary suspension systems, and control algorithms. This Section provides the link to the physical systems that are to be modeled. Appendices A and B provide details of the systems, configuration descriptions, and physical parameters for candidate HSGGT systems.

Section 4 addresses the "Definition of Modeling Requirements"; it reviews the requirements for modeling high-speed guided ground transportation systems while providing insight into the process of creating a mathematical representation of the physical systems described in Section 3.

Section 5 represents the critical, but often overlooked "Model Validation" part of the methodology, in which practical approaches for assessing the adequacy of specific models are reviewed. Gaining confidence in the performance of a model is a basic part of the interpretation of the data developed using the model. This confidence is necessary before any presentation of data or use of modeling results is attempted.

Section 6 focuses on the computational "engine" or "solver" of the toolkit, in that it describes specific models useful for dynamic analysis and simulation of HSGGT systems. In some cases these models also provide a large part of the data output capability and provide the means for data presentation.

Section 7 addresses the logistical issues that must be managed in order to implement a modeling capability.

Section 8 includes general recommendations for acquisition, development, and maintenance of HSGGT modeling tools. Several different levels of implementation are presented.

For the purposes of this study, the dynamic responses of HSGGT systems were grouped into three regimes: stability, curving, and dynamic forced response. Vehicle/guideway response in accident situations (e.g., collisions with other vehicles and fully developed derailment scenarios) are beyond the scope of this work. It is evident that no single analytical tool meets all of the FRA's modeling and simulation requirements for assessing the dynamic performance of HSGGT systems.

The primary conclusions from this work are summarized below.

1. The modeling/analysis of HSGGT systems is a complex process that requires a high level of expertise and credible modeling techniques. There are no shortcuts in this process. The modeling of the complete vehicle/rail or vehicle/guideway system is still a challenging and complex simulation problem. The capabilities for modeling such systems are dispersed throughout the United States at a variety of different organizations. The formation of an HSGGT Modeling and Simulation Center consisting of a team of experts with a "Toolkit" of available vehicle dynamics models is a proposed approach to meeting the modeling and analysis requirements for introducing HSGGT systems to the US..

2. A wide variety of truck designs, suspension systems, and car bodies (tilt versus non-tilt, for example) are currently used in high-speed rail applications. Variety also exists in track structures, ranging from wood crossties on ballast to direct-fixation track on slab structure. The capability is needed to support the following analysis and simulation activities for this wide range of vehicles and track:

    - Vehicle/guideway dynamic response studies for both the typical installed U.S. track environment and for newer track structures

    - Parameter studies of vehicle sensitivity to the US operating environment

    - Quick-response forensic engineering to support accident investigations

    - Vehicle/System certification and validation studies.

3. A similar variety of Maglev vehicle/guideway combinations has been identified, falling into two categories: the electromagnetic system (EMS), which is based on magnetic attraction between vehicle and guideway; and the electrodynamic system (EDS), which is based on eddy current-based magnetic repulsion. With Maglev designs, the vehicle and guideway are even more intimately coupled as a total system and must be analyzed as such.

4. Frequency domain, time domain, and iterative solutions of models of varying complexities will be required to support both HSR and Maglev system evaluations. Each of these solution techniques provides unique strengths and efficiencies that can be used to advantage in the different parts of a dynamic analyses.

5. The models and the modeling approach must be tailored to the problem at hand. Models used for HSR vehicle response are much different from the models used for Maglev vehicle evaluations. There are three general modeling approaches that are used in current practice:

    - Original, usually specialized programs (written, for example, in FORTRAN or MATLAB) that focus on more-or-less specific analytical problems and solutions.

- Multi-body numerical platform codes such as ADAMS/Rail, MEDYNA, NUCARS, and VAMPIRE. The codes provide a dynamic model without having to manually derive equations of motion.

- Symbolic platform codes, such as AUTOSIM (which has been used primarily for automotive vehicle modeling). These codes develop symbolic models for compilation that can be compiled and linked with other software.

6. No single computer program or modeling platform currently provides an adequate modeling means for all aspects of either HSR or Maglev systems. The multi-use "all purpose" multi-body codes reviewed in Section 6 have many strengths, but all also have some weaknesses, ranging from limited solution capabilities to size (computer system requirements) and cost. Most of the multi-body codes have certain "black box" aspects that make a detailed evaluation of the code difficult, present input parameter determination difficulties, and limit the confidence of the results. Several currently available modeling platform codes can prove useful for certain applications and should be included in an analysis "Toolkit". The specialized codes already developed by the team represent a set of effective tools that meet many of the modeling requirements for HSGGT systems.

7. The logistical considerations associated with acquiring, developing, maintaining and applying the required modeling and simulation tools are formidable. These issues often tend to get neglected as mundane, but the success or failure of a modeling endeavor often can be traced to the implementation and concern with the logistical issues. Many factors affect the selection, implementation, and maintenance of modeling tools. A balance must be reached between the various items that affect the choices of modeling implementation. ***A successful, useful model is the blend of engineering and computer sciences for a particular set of hardware and software platforms that is constrained by user and cost limitations.*** Choices in these central four issues, two scientific and two facility related, are limited in their selections by logistical elements of cost, user training, and available technical support. Only when all elements are joined into a functional model is success realized. To ignore any element and make poor selections will doom a model to failure or marginal acceptance.

8. The state of HSR and Maglev system modeling in North America reflects the lack of emphasis placed on our HSR rail systems. Many of the existing analysis codes are old and were written for previous generations of computers and operating systems. Many of these codes are in need of rewrites and conversion to modern programming methods and I/O methodologies. The rapidly ongoing conversion of the computer industry to visual GUI (Graphical User Interface) based operating systems will be placing added emphasis on this conversion in the coming years. There are several commercial rail modeling packages including NUCARS, A'GEM, and, and ADAMS/Rail. NUCARS is in need of a major rework due to the limitations imposed during its original derivation. In its current configuration it is also a "bad" computing corporate citizen. It must be run stand-alone due to hardware interactions that prevent its being run without excessive computer reboot and resets. A'GEM is not widely accepted and needs added fidelity of the wheel/rail interface before it can be considered for the safety related analysis required by the Volpe Center. ADAMS/Rail has promise as a complete modern implementation of a MBS type code. Unfortunately this package is in its early stages of commercial introduction. This package was developed in conjunction with the Dutch Railways and the European Partner of ADAMS (Mechanical Dynamics). Therefore most of the technical expertise for this package resides in Germany, making use and support difficult.

The work performed to date on VA3204 has made significant progress toward the establishment of an HSGGT analytical toolkit and associated HSGGT Modeling and Simulation Center. The Battelle team's experts in vehicle dynamics and control, computer hardware and software engineering has worked together since the fall of 1993 on this program. During this time an extensive assessment of HSGGT models and general-purpose modeling tools has been made, new codes have been developed for Maglev and HSR applications, and comparative evaluations between specialized codes and a commercial code (NUCARS) performed.

1.0 Introduction

High Speed Guided Ground Transportation (HSGGT) vehicle/guideway systems—both high speed rail (HSR) and magnetically levitated (Maglev)—are being considered for deployment in the United States. HSGGT systems offer the promise of fast, safe, comfortable, and cost-effective transport of people between large urban areas with less pollution than is commonly generated by cars and airplanes.

To support the mission of the Federal Railroad Administration (FRA) to ensure the safety of such systems, it is essential that credible analytical procedures be available and readily accessible for predicting safety-related dynamic behavior. A first step in providing such analytical tools is to determine the modeling and simulation requirements for HSGGT systems. Based on these requirements, a set of analytical tools can be developed, acquired, and maintained as an analytical "toolkit" for evaluating the safety-related dynamic behavior of candidate HSGGT systems under intended in-service operating conditions.

1.1 Motivation

From the standpoint of dynamic behavior, the HSGGT systems under consideration for operation in the U.S. differ from conventional passenger trains in two important respects: they operate at speeds considerably higher than conventional rail vehicles, and some of their design features and configurations are fundamentally different from those of conventional vehicles. The combination of higher operating speeds and new vehicle designs raise the critical safety issue of how existing safety standards must be extended to ensure safe HSR operation in the U.S.

The differences between high-speed and conventional speed vehicle/guideway systems raise a fundamental question with respect to modeling and simulation: to what extent can existing models of more conventional vehicle/guideway systems be used to evaluate high-speed systems? A large number of modeling and simulation tools have been developed since the 1960s for predicting the dynamic performance of guided ground transportation systems. Many of these were developed by researchers for specific vehicles, vehicle classes, and operating/loading environments. More recently, several commercial, general-purpose dynamic simulation codes have been developed for simulating a relatively wide range of vehicle/guideway systems. These modeling tools are the foundation for the development of an analytical toolkit for accurate and comprehensive assessments of the safety-related dynamic performance of HSGGT systems.

1.2 Scope

For the purposes of this work, HSGGT systems are defined as HSR and Maglev systems designed for operation at speeds of at least 150 mi/h (240 km/h). Examples of HSR systems are the:

- French TGV
- German ICE
- Swedish X2000
- Italian ETR 450 and ETR 500
- Japanese Shinkansen trains.

Examples of Maglev systems are the:

- German TR07
- Japanese MLU series
- Four System Concept Designs (SCDs) proposed under the National Maglev Initiative (NMI).

Modeling and simulation requirements for these HSGGT systems are based on three basic goals for safety-related dynamic performance:

1. Safety against vehicle ***loss of guidance*** (e.g., derailment)

2. ***Ride quality*** that ensures passenger safety and comfort

3. ***Structural integrity*** that ensures adequate durability and reliability of the hardware comprising the vehicle/guideway system.

Also, for the purposes of this study, the dynamic response of HSGGT systems has been grouped into three regimes:

1. ***Stability*** — self-excited vehicle and guideway oscillations; resonance responses; control system performance

2. ***Curving*** — curve entry/exit; steady-state curve negotiation, including wind loading and propulsive and braking forces

3. ***Dynamic Forced Response*** — Vehicle/guideway dynamic interaction; response to forces induced by track geometry effects, wheel profile and track anomalies, special trackwork, adjacent cars; aerodynamic effects; pantograph/catenary interaction.

These dynamic performance goals must be evaluated for each regime of dynamic response. For example, each cell of the evaluation matrix in Figure 1-1 may require a different computational tool to perform the associated analysis. This naturally leads to a "toolbox" approach to modeling and simulation for these systems.

The overall modeling and simulation requirement for the VA3204 task is to provide all of the necessary analytical tools for predicting HSGGT vehicle/guideway response in a manner that can be applied to safety-related dynamic performance goals as presented. Vehicle/guideway response in accident situations (e.g., collisions with another vehicle and fully developed derailment scenarios) are considered beyond the scope of this work.

		Response Evaluation Regime		
		Stability	Curving	Dynamic Forced Response
Safety-Related Performance Criteria	Loss of Guidance			
	Ride Quality			
	Structural Integrity			

Figure 1-1. Matrix of response evaluation regimes versus performance requirements.

1.3 Organization of Document

This report defines currently envisioned requirements for dynamic analysis and simulation of HSGGT systems in the U.S. However, FRA requirements for modeling of HSGGT systems will undoubtedly change as HSGGT systems are brought closer to routine service in the U.S. and as the range of tools available to respond to those requirements continues to broaden.

The contents of Sections 2 through 8 are interrelated, as shown in Figure 1-2, and they represent a methodology for developing the analytical toolkit. The essential elements of a modeling effort are presented in Figure 1-3. The four major steps to modeling are:

- Define the vehicle, object, or system to be modeled and define the modeling goals
- Prepare a model by idealizing the vehicle, object, or system and creating a mathematical description of the vehicle, object, or system

- Execute a mathematical algorithm, or series of algorithms to "solve" or exercise the mathematical description of the vehicle, object, or system
- Interpret the results of the algorithmic execution to relate them to the physical vehicle, object, or system.

These processes are represented as the four major boxes across the center of the Figure 1-3. Each step of the modeling process is influenced by both the scientific issues surrounding the modeling and the logistical issues of implementation. Additionally, there are personnel, equipment, and cost, issues that must be addressed. These issues will be addressed in the following sections of this report.

Section 2 addresses "Safety-Related Dynamic Performance Considerations", including a wide variety of critical issues regarding HSGGT dynamic analysis and simulation. This Section defines the analysis objectives for the modeling effort by defining the dynamic performance objectives to be examined.

Section 3 addresses the "System Configuration" part of the methodology; it provides examples of present-day HSGGT systems, including general information on their vehicle and truck designs, primary and secondary suspension systems, and control algorithms. This Section provides the link to the physical systems that are to be modeled.

Section 4 addresses the "Definition of Modeling Requirements"; it reviews the requirements for modeling high-speed guided ground transportation systems while providing insight into the process of creating a mathematical representation of the physical systems described in Section 3.

Section 5 represents the critical, but often overlooked "Model Validation" part of the methodology, in which practical approaches for assessing the adequacy of specific models are reviewed. Gaining confidence in the performance of a model is a basic part of the interpretation of the data developed using the model. This confidence is necessary before any presentation of data or use of modeling results is attempted.

Section 6 focuses on the computational "engine" or "solver" of the toolkit, in that it describes specific models useful for dynamic analysis and simulation of HSGGT systems. In some cases these models also provide a large part of the data output capability and provide the means for data presentation.

Section 7 addresses the logistical issues that must be managed in order to implement a modeling capability.

Section 8 includes general conclusions pertaining to the acquisition, development, and maintenance of HSGGT modeling tools. .

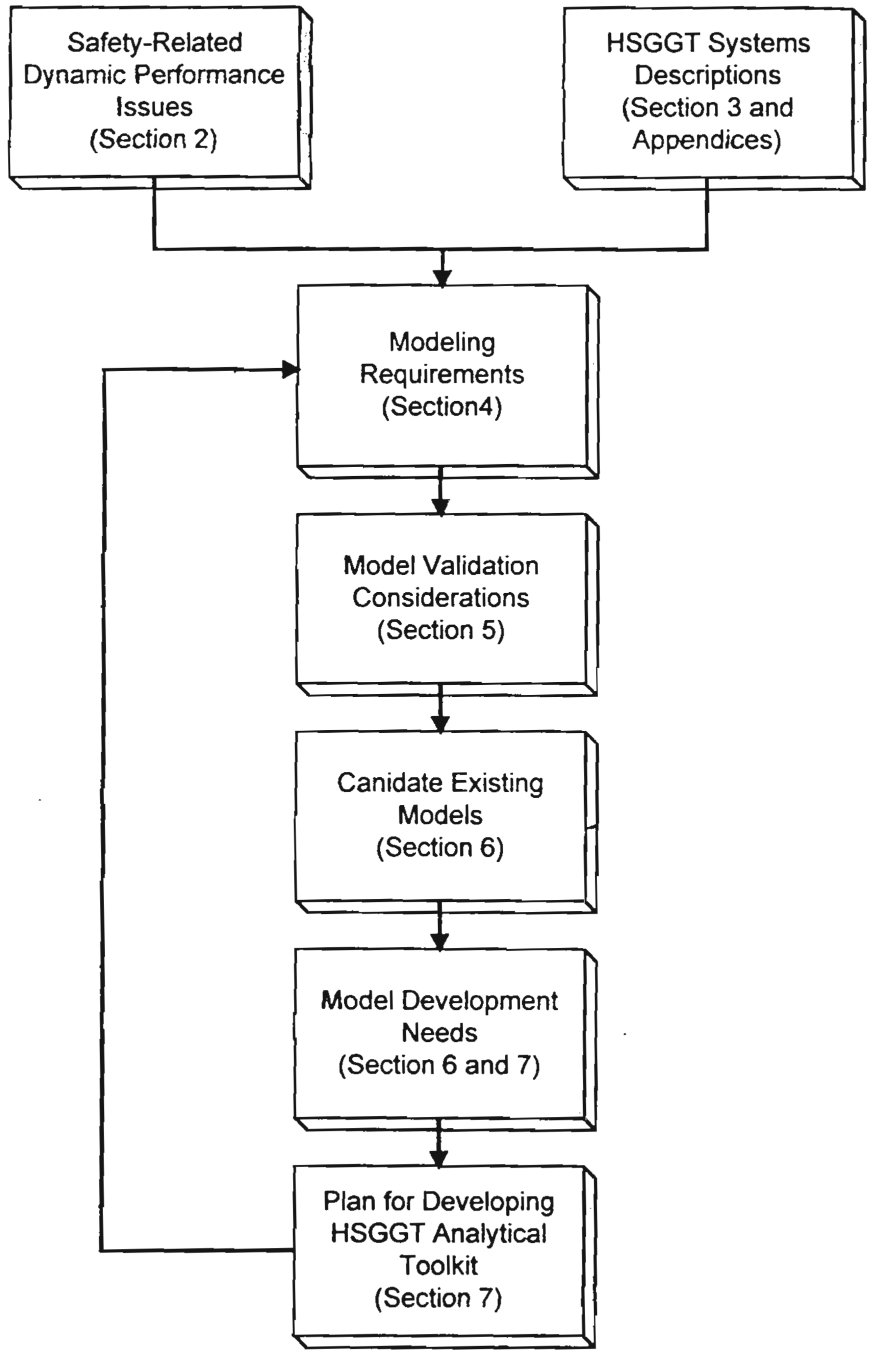
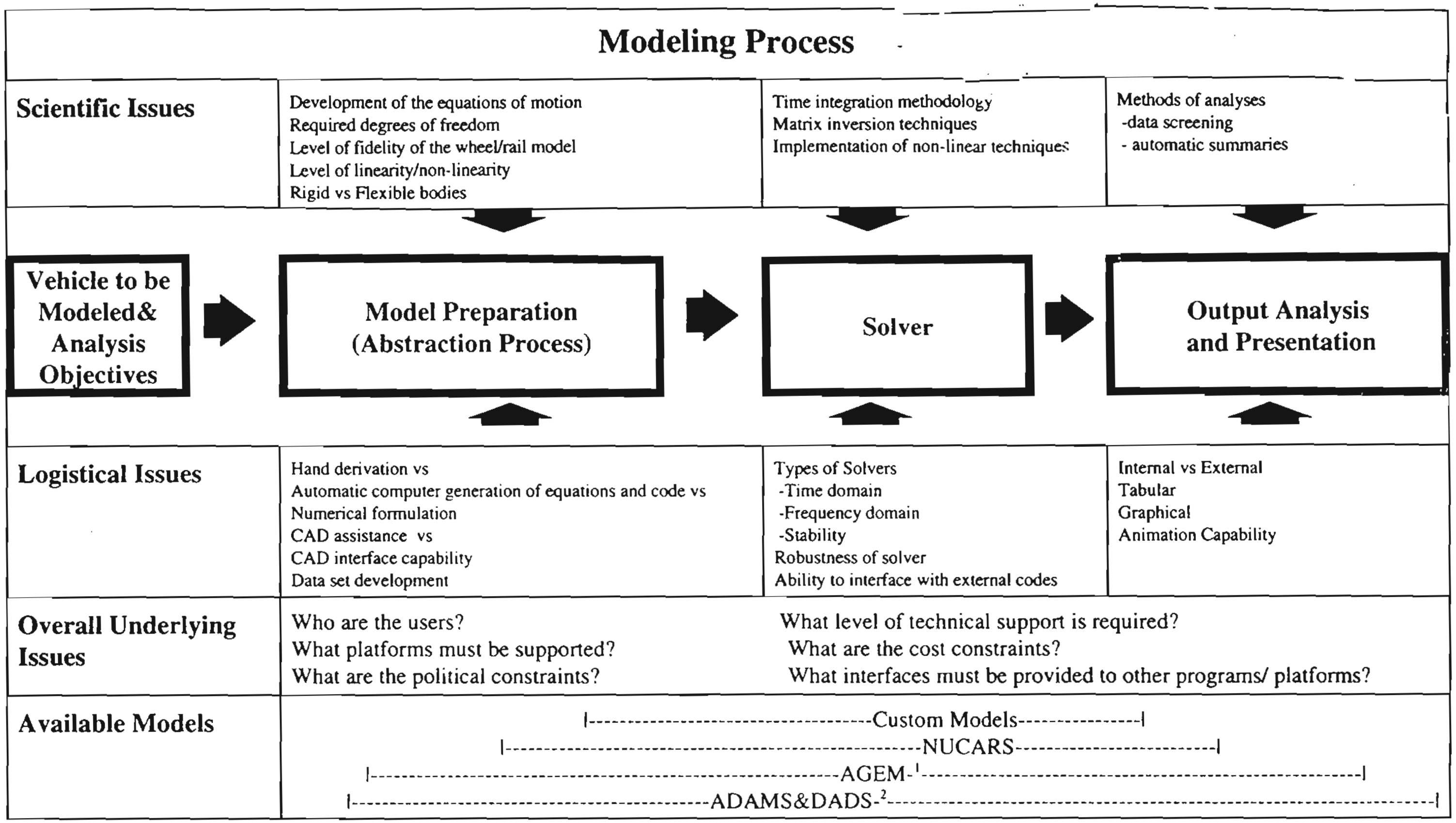


Figure 1-2. Methodology for developing HSGGT analytical toolkit, and structure of VA3204 Task 1 report.



1- AGEM accomplishes a full range of modeling needs but may need updating wrt the w/r model 2- DADS is also a complete package but currently contains no w/r model

Figure 1-3. Essential Elements of the Modeling Process

2.0 Safety-Related Dynamic Performance Considerations for HSGGT Systems

This section provides an overview of the safety-related dynamic performance issues associated with HSGGT systems. HSR and Maglev systems are discussed separately in the context of safety-critical events that affect loss of control, poor ride quality, and degradation in structural integrity.

2.1 Safety-Related Dynamic Performance of HSR Systems

2.1.1 Performance Objectives for HSR Systems Derailment safety, ride quality, and structural integrity can all be challenged by events related to stability, curving, and forced response. Table 2-1 relates the critical performance objectives to the dynamic response events that challenge the objectives.

2.1.2 Critical Events for HSR Systems

HSR Derailment Safety: Derailments are most often caused by combinations of track misalignments and vehicle motions, producing forces that defeat the guidance restraints of the rail on one or more wheelsets. Among predictably dangerous conditions are wheelset instability, excesses in the ratios of lateral to vertical wheel/rail forces leading to wheel climb or rail rollover, car body roll, and longitudinal interaction between cars.

This subject is treated in detail by the Report entitled "Review of Current Derailment Criteria," which has been submitted as a Technical Note under VA3204. The study defines the following modes of derailment:

1. Wheel climb derailment
2. Rail rollover
3. Gauge widening
4. Track panel shift
5. Overspeed on curves.

Criteria used by various rail authorities to determine derailment likelihood are discussed for each of the potential derailment modes, as listed in Table 2-2.

Table 2-1. Rail vehicle performance objectives vs. governing modes of dynamic response.

Performance Issues	How Responses Affect the Objectives		
	Stability	**Curving**	**Forced Response**
Derailment Safety	Critical speed for wheelset hunting must be > maximum train speed under worst case wear conditions.	Curving affects: ▪ Wheel climb, flanging ▪ Rail rollover ▪ Track shift ▪ Wheel/rail loads, and ▪ Car body rollover likelihood.	Danger is increased when unbalanced speed is combined with traction/braking, wind loads, track irregularities, and train action loads. Track shift and buckling can be accelerated under severe loading conditions.
Ride Quality	Instability → flanging → severe lateral and yaw motions.	Requires nonflanging curve negotiation without excessive yaw or roll.	Requires mitigation of track roughness, train action loads, aero loads from wind, passing trains, and tunnel entrance.
Structural Integrity	Unstable operation increases wear and likelihood of component failures.	Curving at high speed with uncompensated superelevation creates high wheel/rail loads and leads to premature failure of critical components.	Forces from adjacent cars, track roughness, and wind, combined with high speed and uncompensated superelevation, accelerates component failures.

Table 2-2. Derailment governing factors and avoidance criteria.

Derailment Mode	Governing Factors	Derailment Avoidance Criteria
Wheel Climb	Wheel/rail loads Wheel/rail contact geometry Coefficient of friction Primary suspension parameters Wheelset angle of attack Load time duration Wheelset effective mass Wheel lateral impact velocity	Single wheel L/V Single wheel L/V with time duration Axle L/V Wheel vertical load
Rail Rollover	Wheel loads: L,V, moment Rail size, bending stiffness Fastener & tie strength & stiffness Wheel tread & flange geometry Adjacent wheel loads Wheel/rail friction	Truckside L/V Wheel/rail force levels
Gauge Widening	Gauge-spreading loads Fastener & tie strength & stiffness Existing wide gauge Adjacent wheel loads	Gage Reserve Index GRI = 59-(G+Δg) Δg = Change in gauge under load
Track Panel Shift	Wheelset net lateral load Wheelset L/V Tie/ballast lateral resistance Longitudinal loads (thermal, braking, train action) Adjacent wheel loads Track geometry, misalignments, curvature Track dynamic response	LatAxleLoad = f(Pvert) Static vertical wheel load limits Truck frame lateral acceleration
Overspeed on Curves	Sum of forces on vehicle - gravitational, centrifugal, dynamic, crosswind Track radius and superelevation Train speed, c.g. height Suspension characteristics	Vmax = f(Superelevation & Curvature)

2.1.2.1 HSR Ride Quality. Ride quality is generally considered a comfort issue (e.g., a good ride is one which isn't felt or heard, and a bad ride subjects the passenger to discomforting jostling, vibration, and noise). Ride quality becomes a safety issue when the acceleration responses of the passengers and other contents of the cars are sufficiently high to cause injury (e.g., baggage shifting, loss of footing) or extreme discomfort (nausea, disorientation, etc.). Flange-to-flange wheelset hunting, suspension bottoming due to guideway defects, flanging and excessive roll and yaw during curve negotiation, forces transmitted by adjacent cars, and sudden aerodynamic loads from passing cars or wind gusts are typical sources of poor ride quality.

Since subjective responses vary greatly with individual passengers and their attitudes, there are well-defined criteria for defining acceptable limits for control of low-frequency acceleration, vibration, and noise. While there is no well-defined difference between low-frequency acceleration and vibration, there is a definite perceived difference between accelerations that jostle the body and those that merely cause irritation and fatigue.

The frequently used and generally accepted measures of ride quality include:

- Limits to the lateral and vertical acceleration peaks measured within low frequency ranges at the passenger seat

- Limits to the vibratory accelerations experienced at the passenger seat, assessed in terms of frequency spectra showing average levels within prescribed bands, such as third-octave, and compared against an accepted standard, such as the ISO standard.

- Noise levels, often measured in service or test runs and reduced to frequency spectra, often using third-octave bandwidths.

Table 2-3 summarizes the governing factors and assessment criteria most often applied to determine acceptable ride quality.

Table 2-3. HSR ride quality - governing factors and assessment criteria.

Ride Disturbance	Governing Factors	Assessment Criteria
Excessive vibration	Suspension stiffness and damping Track roughness Wheel surface conditions Vehicle overspeed in curves Flanging guidance	Accelerations allowed in specified frequency bandwidths, e.g. ISO standards, lateral and vertical
Jerk	Suspension stiffness, damping Suspension element travel Track roughness Vehicle overspeed in curves Wheel surface conditions	Permissible rate of change in acceleration, lateral, vertical and longitudinal
Excessive noise	Suspension stiffness and damping Track roughness Wheel surface conditions Propulsion, braking Isolation of car body through sound deadening	Permissible levels of perceived noise (dB) in the passenger cabin

2.1.2.2 ***HSR Structural Integrity.*** The structural integrity of vehicle and track ensures safe operation over the intervals between planned inspections and/or maintenance. Maximum service loads and fatigue loading environments on critical components are of concern. Unstable operation accelerates wear and increases the probability of component failure. Curving at high speed, with uncompensated superelevation and track roughness exceeding class limits, can create very high loads at the wheel/rail interface and in the primary suspension system. These can lead to premature failure of critical components. Forces from adjacent cars, track obstructions, and wind loads, when combined with other critical conditions including high speeds, uncompensated superelevation and track roughness, can create unsafe conditions from accelerated car body and component wear and failures. Table 2-4 lists the major causes of structural failure and the recommended assessment criteria.

Table 2-4. Structural integrity - governing factors and assessment criteria.

Failure Mode	Governing Factors	Assessment Criteria
Excessive component wear	Track roughness Lack of isolation Wheel surface conditions Vehicle overspeed in curves Unstable or flanging guidance	Truck frame, equipment accelerations Physical evidence of wear
Component failure	Track roughness Lack of isolation Wheel surface conditions Vehicle overspeed in curves Unstable or flanging guidance	Truck frame, equipment loads and stresses (levels and frequencies of occurrence)

2.1.3 Implications for Modeling HSR Systems

Analytical studies of HSR systems to assess derailment safety, ride quality, and structural integrity for HSR systems should investigate:

1. The track standards that must be maintained to assure:
   - Safety against derailment
   - Ride quality
   - A loading environment that does not accelerate wear or threaten failure of vehicle and track components.

2. The ability of the system to negotiate maximum planned speeds on tangent and curved track with performance at the wheel/rail interface which does not threaten derailment or accelerate wear beyond controllable limits.

3. The critical speed of hunting instability for conditions of variable wheel/rail contour, wheel/rail surface condition, suspension characteristics and car body loads.

4. The wheel/rail forces and wheelset positions, over the vehicle's speed range, that include curves, uncompensated superelevation, track misalignments, forces from adjacent cars under emergency conditions, propulsion, and wind.

5. The actions of trains subjected to emergency situations, such as track defects, critical component failure, and obstacles on the track.

6. Car body motions, over the entire speed range, in response to track contours and irregularities, superelevation deficiencies, external forces from adjacent cars, cars on adjacent tracks, and wind.

7. The vibration environment of the car body in response to track irregularities, wheel and rail defects, and equipment vibration through the important structural vibration transmission paths.

8. The noise environment of the car body in response to track irregularities; noise from propulsion, air conditioning, and other on-board equipment; and noise from external sources including passing trains.

9. The peak and fatigue loading environments of all vehicle and track components which are critical to safe operation of the system.

10. The dynamic behavior of vehicles on elevated guideways.

The following features maybe needed in modeling tools to support these assessments:

▪	Solution methods	Time and frequency domain solutions
▪	Car bodies	Up to 6 rigid body degrees of freedom for basic vertical, lateral, and torsional bending modes
▪	Other rigid masses	Bolsters, truck frames, wheel sets, and traction motors, where applicable
▪	Special wheelsets	Independently rotating, interconnected
▪	Suspension elements	Primary, secondary, nonlinear, passive, and active
▪	Wheel/rail interaction	The wheel/rail contour, vertical and lateral track stiffness, creep force algorithm, representation of flanging, wheel/rail climb, wheel lift
▪	Track structure	Vertical and lateral stiffnesses, vertical and lateral beam bending of elevated spans, variable support for spans, track shift (nonlinear ballast force-displacement)
▪	Track contour	Spiral transitions into and out of superelevated curves

▪	Track roughness	Discrete and stochastic track irregularities
▪	External forces	Aerodynamic: lateral force and yawing moments varying with time, sudden gusts (step loads), train-action forces
▪	Propulsion	Nonlinear draft gear connectors exerting lateral and fore-aft displacement constraints between car bodies; applied moments on wheel sets

Section 4.1 provides a detailed definition of modeling requirements for HSR vehicles.

2.2 Safety-Related Dynamic Performance of Maglev Systems

In this section we describe the critical issues associated with the safety-related dynamic performance of Maglev vehicle/guideway systems.

2.2.1 Performance Objectives for Maglev Systems

Assurance of safe guidance, ride quality, and structural integrity of Maglev vehicles will require examination of performance as the Maglev vehicle encounters events challenging stability, ride quality, and structural integrity. Table 2-5 describes the problems expected in meeting the performance objectives when the vehicle is subjected to challenging events.

2.2.1.1 Maglev Safety Against Loss of Guidance. Combinations of high speed, complex interactions between vehicle and guideway, and significant aerodynamic effects create concern over the possibility of losing guidance, which might occur through any of the following mechanisms:

- Electrical power is lost. The lifting, lateral guidance, and propulsive forces are lost; support and guidance of the vehicle revert to backup systems
- Obstructions on the guideway cause loss of containment structure and lift
- Aerodynamic forces exert large lateral loads on the vehicle at a critical moment such as in a curve or at a tunnel entrance
- The control system(s) regulating lift, guidance, and/or propulsion fail.

Table 2-6 summarizes the factors governing safe guidance and its essential assessment criteria.

2.2.1.2 Maglev Ride Quality. Ride quality concerns are dominated by requirements for secondary suspension systems to mitigate the effects of high primary stiffnesses, inadequately controlled pitch and yaw, and aerodynamic effects. Table 2-7 defines the principal factors governing ride quality assurance and the criteria to be applied in its assessment.

Table 2-5. Performance objectives for Maglev vehicles vs. governing modes of dynamic response.

Performance Issues	How Responses Affect the Objectives		
	Stability	**Curving**	**Forced Response**
Safe Guidance	Lift and guidance control is a central issue—gap maintenance in each mode Loss-of-power "landing" loads can be severe	Curving affects: ▪ Lift and guidance gap control ▪ Lateral loads to be balanced by guidance control	Danger is increased when unbalanced speed is combined with traction/braking, wind loads, guideway irregularities, and train action loads.
Ride Quality	Instability → loss of gap control → severe lateral and yaw motions.	Requires adjustment of suspension and guidance parameters, affecting car body vibration	Requires mitigation of guideway roughness, train action loads, aero loads from wind, passing trains and tunnel entrance.
Structural Integrity	Unstable operation increases wear and likelihood of component failures.	Curving at high speed with uncompensated superelevation creates lift and guidance gap control problems	Forces from adjacent cars, guideway roughness, and wind, combined with high speed and uncompensated superelevation, accelerate component failures.

Table 2-6. Maglev safe guidance — governing factors and assessment criteria.

Loss-of-Guidance Mode	Governing Factors	Assessment Criteria
Guideway liftoff Gap bottoming	Poor control of lifting, guidance, and propulsion forces Aerodynamic forces Guideway roughness and contour Inadequate restraint	Maximum suspension component loads Allowable carbody and accelerations
Loss of power	Loss of lift, guidance, and propulsion Landing gear damaged Inadequate restraint	Maximum loads sustainable by the vehicle during the recovery Maximum safe carbody displacements and accelerations

Table 2-7. Maglev ride quality — governing factors and assessment criteria.

Ride Disturbance	Governing Factors	Assessment Criteria
Excessive vibration	Suspension stiffness and damping Guideway misalignments Vehicle overspeed in curves	Accelerations allowed in specified frequency bandwidths, e.g. ISO standards, lateral and vertical
Jerk	Suspension stiffness, damping Suspension element travel Guideway roughness Vehicle overspeed in curves Contact guidance	Permissible rate of change of acceleration
Excessive noise	Suspension stiffness and damping Track roughness Propulsion, braking Isolation of car body through sound deadening	Permissible levels of perceived noise (dB) in the passenger cabin

*2.2.1.3 **Maglev Structural Integrity***. "Landing loads" occurring when a system has lost power will dominate the need for structural integrity determinations. The vehicle must be brought to a safe condition on backup support and guidance systems and be capable of reaching a station under emergency conditions. In addition, the primary, electromagnetic suspension/lift system can present a very harsh environment, and its effects may not be well understood or anticipated. Table 2-8 defines the factors governing structural integrity and the criteria to be applied in its assessment.

Table 2-8. Structural integrity — governing factors and assessment criteria.

Failure Mode	Governing Factors	Assessment Criteria
Excessive component wear	Guideway roughness Lack of isolation Vehicle overspeed in curves Unstable guidance Suspension bottoming guidance	Truck frame, equipment accelerations Physical evidence of wear
Component failure	Guideway roughness Lack of isolation Vehicle overspeed in curves Unstable guidance Suspension bottoming guidance	Component loads and stresses (levels and frequencies of occurrence)

2.2.2 Implications for Modeling Maglev Systems

Analytical studies of Maglev systems to assure loss-of-guidance safety, ride quality, and structural integrity should investigate:

1. The guideway standards that must be maintained to assure:

    - Safety against loss-of-guidance
    - Ride quality
    - A loading environment which does not accelerate wear or threaten failure of vehicle and guideway components.

2. The ability of the system to negotiate maximum planned speeds on tangent and curved guideway with performance at the vehicle/guideway interface which does not threaten loss of guidance or accelerate wear beyond controllable limits.

3. The critical speed of lift/propulsion/guidance instability for conditions of variable vehicle/guidance surface conditions, suspension characteristics, and car body loads.

4. The vehicle/lift, vehicle/guidance positions, over the vehicle's speed range, for maneuvers that include curves, uncompensated superelevation, guideway misalignments, forces from adjacent cars under emergency conditions, propulsion, and wind.

5. The response of Maglev trains subjected to emergency situations, such as guideway defects, critical component failure and obstacles on the track.

6. Car body motions, over the entire speed range, in response to guideway contours and irregularities, superelevation deficiencies, external forces from adjacent cars, cars on adjacent guideways, and wind.

7. The vibration environment of the car body in response to guideway irregularities, vehicle and guideway defects, and equipment vibration through the important structural vibration transmisson paths.

8. The noise environment of the car body in response to guideway irregularities; noise from propulsion, air conditioning, and other on-board equipment; and noise from external sources including passing trains.

9. The peak and fatigue loading environments of all vehicle and guideway components which are critical to safe operation of the system.

10. The dynamic behavior of vehicles on elevated guideway.

The features needed in modeling tools to support the assurance of safe guidance, ride quality, and structural integrity include:

▪	Solution methods	Time and frequency domain solutions
▪	Car bodies	Up to 6 rigid body degrees of freedom for basic vertical, lateral, and torsional bending modes
▪	Other rigid masses	Bolsters, truck frames, magnets or magnet pads
▪	Suspension elements	Primary, secondary, nonlinear, passive, and active
▪	Vehicle/guideway interaction	The guideway/magnet contour, vertical and lateral guideway stiffness, guideway tolerances.
▪	Guideway structure	Vertical and lateral stiffnesses, vertical and lateral beam bending of elevated spans, variable support for spans, guideway shift
▪	Guideway contour	Spiral transitions into and out of superelevated curves
▪	Guideway roughness	Discrete and stochastic guideway irregularities
▪	External forces	Aerodynamic: lateral force and yawing moments varying with time, sudden gusts (step loads), train-action forces

- Propulsion — Nonlinear draft gear connectors exerting lateral and fore-aft displacement constraints between car bodies; applied moments on lift and guidance components.

Section 4.2 defines detailed modeling requirements for Maglev systems.

3.0 Systems Overviews and Descriptions

This section provides a discussion of those aspects of HSR and Maglev vehicle/guideway system designs which are significant from the standpoint of modeling and simulation of safety-related dynamic performance. As indicated in this section, there exists a wide range of configurations and features among these systems, which must be considered in developing analytical models.

3.1 HSR Vehicle/Track Systems

3.1.1 HSR Vehicles

A brief review of those design and operating characteristics of candidate HSR vehicles that are related to safety-related dynamic performance is provided in this section. Of the following six HSR vehicles considered in this review, two are tilt-body trains (X2000 and ETR 450), one is an articulated train (TGV family), two are conventional trains (ICE and ETR 500), and one is a multiple-unit train (Shinkansen family). A physical overview of these systems is summarized in Table 3-1.

Table 3-1. Main operating characteristics of candidate HSR systems.

Parameters	X2000	Shinkansen Series 300	TGV - A	ICE	ETR 450	ETR 500
Train Consist *	M+4T+ C	10M+6T	M+10T+M	M+12T +M	8M+T	M+11T +M
Train Weight (t)	343	720	444	784	470	640
Train Length (m)	340	393	237	357	230	328
Seats	267	1323	485	627	416	563
Power (kW)	3260	12000	8800	9600	6250	8800
Max. Speed (km/h)	200	275	300	280	250	300
Motor Axleload (t)	17.5	11.3	17.0	19.5	13.5	17.0
Trailer Axleload (t)	13.6	11.3	16.1	13.0	13.0	11.5
Remarks	Tilt-body	Multiple Units	Articulated	Conven-tional Pressure Sealed	Tilt-body, 4 coupled pairs	Conven-tional Pressure Sealed

These reviews are based largely on information contained in the open literature, which has been used to identify many of the salient features that should be included in analytical models of the dynamic behavior of these vehicles. A summary of these features is provided in Table 3-2.

3.1.1.1 Wheelset and Wheel/Rail Interface. Several wheelset design features can influence dynamic performance significantly, and should be considered as part of an HSR model, including:

- Wheel/rail profile geometry
- Wheelset torsional and bending stiffness
- Wheelset mass
- Wheel diameter.

As indicated below, there are wide variations in wheelset designs among existing HSR systems.

Wheel Profiles. The equivalent conicity is one of the most important parameters associated with the stability and curving performance of trucks. It is defined as half the difference in rolling radii for a unit lateral displacement of the wheelset[3-1].* The SNCF (TGV-A) uses a low equivalent conicity, ensuring very high critical speeds for trucks (up to 700 km/h). The SNCF wheels have a 1/40 conical profile and the rails have a 5 percent inward cant[3-1]. This gives an equivalent conicity of 0.025 for new wheels and it increases by 0.05 per million km of service[3-2].

The DB (German Rail) uses a standard European wheel profile on rails with a 2.5 percent inward cant. Although this minimizes wheel wear, the resulting higher equivalent conicity results in lower critical speeds for trucks than those attained by the SNCF.

Flange Clearance. Another factor that can impact the equivalent conicity is the flange clearance. The nominal gauge is 1435 mm and the distance between the flange faces can vary between 1410 and 1425 mm. A flange clearance of 10 to 15 mm is typical. A smaller value of flange clearance is associated with a higher value of effective conicity.

Wheel Shape. Wheel diameter varies by system and is generally greater for power units than for trailers. On the TGV-A, wheel diameters vary from 920 to 1100 mm for motive units and from 860 to 920 mm for trailers. The Japanese are studying wheels that have thinner webs in complex 3D shapes in an effort to develop lighter wheels that have the same stiffness, fatigue, and thermal resistance as the more conventional wheels[3-3].

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* References appear at the end of this section.

Table 3-2. HSR vehicle features associated with safety-related dynamic performance.

Feature	Relationship to Safety-Related Dynamic Performance
Wheelset and wheel/rail interface	Wheel profile, rail profile, and rail cant determine the equivalent conicity of the wheelset, and the evolution of this conicity with wheel and rail wear. Also, sensitivity to changes in wheel/rail contact geometry and adhesion limits at HSR speeds may be greater than at conventional speeds.
Truck frame and primary suspension composite materials trucks	Impact of primary suspension and wheelbase on truck stability at high speed. Use of soft longitudinal suspension in steerable trucks. Low modulus and orthotropic properties of composite materials may result in significant truck frame dynamics (e.g., flexural response), which in turn can influence safety-related dynamic performance of vehicle.
Power collection system	Ability to maintain continuous power to vehicle depends on pantograph and overhead line dynamics; high-speed operation in the U.S. may require pantograph modifications or redesign of catenaries.
Traction motor/drivetrain	Vehicle stability/curving performance influenced by motor/drive dynamics (unsprung mass effects, sprung mass dynamics, etc.). Ability to control wheel/rail adhesion influences tendency for wheel slip, flanging, high L/V forces.
Braking and wheel slip control systems	Braking at high speeds could result in coupled wheelset/drivetrain/ brake system dynamics. Wheelset dynamics may be affected strongly by modulation of torque to wheelsets in response to sensors that indicate slip conditions.
Secondary suspension, including car body tilt systems and their controls	Influences ride quality; tilt system malfunction could result in unsafe condition; tilt system response to severe loading (e.g., high crosswinds) may be inadequate.
Car body and aerodynamic design	Aerodynamic forces at HSR speeds may have significant influence on stability and derailment tendencies during curving, train crossing, and tunnel entry/exit
Car to car connections, including articulated trains	A unique configuration that is not easily modeled with traditional single car-body models.
Torsionally flexible wheelsets, wheelsets with direct interconnection, force-steered wheelsets and independently-rotating wheels	Candidate HSR vehicles may have one or more of these features; each feature has a potentially strong influence on vehicle dynamic performance, and modeling requires the addition of vehicle degrees-of-freedom, truck components and/or kinematic constraints to those typically used for "conventional" vehicles.
Measurement schemes to indicate unsafe or uncomfortable operating conditions	Reliability/effectiveness of using measurement variables such as truck or wheelset lateral accelerations to indicate derailment tendency (and possibly ride comfort) is uncertain.

3.1.1.2 Truck Frame and Primary Suspension. One of the main objectives in designing a truck is to have a critical speed well above the maximum operating speed (a stiff truck) while at the same time providing good curving performance (a flexible truck). Another important objective is to reduce as much as possible the lateral and vertical forces at the wheel/rail interface during running on tangent and curved track. Thus, it is critical that the HSR truck is represented in sufficient detail so that its safety-related dynamic performance can be predicted with good accuracy.

Typical truck layouts of HSR systems are shown in detail in the "Requirements for Safety-Related Analysis and Simulation of High Speed, Guided, Ground Transportation Systems" submitted earlier on this contract. Trucks may take several different configurations but are basically a series of beams to constrain the axles and running gear. Side beams are necessary and most trucks also have end beams and even both center and end beams (ICE, ETR 500, X2000). By eliminating the end beams, trucks can become significantly lighter (SNCF, Shinkansen). The SNCF is developing a light alloy truck which will be 500 kg lighter than the present Y237 truck. However, if the truck exhibits strong frame dynamics (flexural response), the critical speed could be significantly lowered[3-4]. The interconnections of these truck beam elements is critical to obtain the desired high longitudinal and lateral stiffness. Pivot rods, coil springs, and/or elastomeric pads are used in conjuction with hydraulic dampers and friction dampers to provide the desired charactoristics on the TGV-A trucks[3-5]. Trucks such as the X2000 truck employ soft longitudinal suspension elements to enhance the self-steering tendency of the wheelsets, where the axles tend to align themselves radially in curves. A downside is a reduction in the maximum critical speed of the truck.

3.1.1.3 Power Collection System. Safe operation of HSR systems depends in part on uninterrupted power to the vehicle. Interruptions in power can result in erratic and uncontrolled dynamic behavior of the vehicle. From the standpoint of safety-related dynamic performance, it may be important to include the dynamic behavior of the power collection system in a modeling and simulation tool.

The high power needs of HSR systems require high voltage AC catenaries (15 to 25 kV). High currents necessitate the use of multiple pantographs for collection, and multiple pantographs will interact dynamically with each other and the catenary. At high train operating speeds, a higher wire tension typically is needed along with simpler, lighter designs. For example, in the SNCF design, the contact strips have their own suspension, and their unsprung masses are minimized.

3.1.1.4 Traction Motor/Drivetrain. The design of high speed motive units must conform to the same constraints of high truck stability and safety as in design of coaches. Two of these constraints, low axle load and low unsprung mass, dictate that the traction motors and drive trains be partially (if not fully) suspended under the car body. The traction motor/drivetrain system also may provide a source of dynamic loading to the vehicle, through both rotating unbalance forces and independant ridgid body response of the motor/drivetrain which can cause high dynamic stresses and possible fatigue failures in adjacent components. Consequently, this part of the vehicle system may be an important element of an HSR model.

The X2000, TGV, and ETR 500 locomotive have axle loads close to 17 t. Their traction motors are entirely suspended under the car body, except for the X2000, which has its motors suspended from its truck frame. The ICE has an axle load of 19.5 t, and the weight of its drive unit (traction motor, gear drive, and integrated disk brake) is spread 65 percent on the car body and 35 percent on the truck by means of pendulum rods[3-9]. The new Shinkansen Series 300 "Nozomi" has brought its axle load down to 11.5 t and has its traction motors suspended on the truck frame. The correct weight distribution must be represented in any model to assure correct representation of the vehicle by the model.

The remaining unsprung mass is being lowered by innovative wheelset designs, such as hollow axles (ICE, ETR 500, X2000) and new wheel profiles. The Nozomi has achieved an unsprung mass of 1660 kg per axle, which is significantly less than the 2048 kg of the TGV-A driving axle[3-10, 3-4].

To obtain a high tractive effort it is important to maximize the use of the available adhesion. Most high speed locomotives have now implemented some sort of wheel-slip detection and control and realized an increase from approximately 5 percent adhesion value for the earlier locomotives to up to 8 percent adhesion up to their top operating speeds (300 km/h).

3.1.1.5 Braking Systems. HSR response during braking is an important safety consideration for several reasons. These include:

- Uneven braking applied to a wheelset could excite wheelset torsional modes.
- The dynamic interaction of the trucks with slip control devices could result in degraded or unsafe vehicle response, and may require detailed modeling of the slip/skid control system.
- Even for operation without braking forces, the accurate and credible modeling of an HSR vehicle generally requires that the mass and inertias associated with braking equipment are included.
- The HSR vehicle response during braking depends on the manner in which the brakes are applied. Thus, the brake application sequence must be simulated accurately so that the predicted vehicle response represents that of the physical system.

Locomotive Braking. The braking subsystem of locomotives uses first the dynamic (regenerative) brakes from top speed down to 20 - 50 km/h. . The mechanical brakes take over at lower speeds as the dynamic brakes fade out at low speed. Various configurations of disk brakes are used on the HSR vehicles studied.

Trailer Car Braking. Most trailer cars are also equipped with disk brakes. On the ICE, the trailer cars have track eddy-current brakes in addition to mechanical brakes, while on the Shinkansen the trailers also have rotational eddy-current brakes[3-11].

Wheel Slip Control. A wheel slip/skip control subsystem is usually part of the computer-controlled braking system. The speed and acceleration of each axle is measured and the fastest axle (in braking) or the slowest axle (in motoring) is chosen as the reference. The braking forces or the traction effort is then modulated according to algorithms specific to each HSR system. In Japan, the application of fuzzy logic is being investigated for the anti-skid algorithm which will further complicate modeling of braking performance.

3.1.1.6 Secondary Suspension. The main purpose of the secondary suspension is to decouple as much as possible the movement of the truck from the car body movement in the vertical and lateral directions, which in turn provides good ride quality and low stresses in the car body. It is important that the secondary suspension system is modeled accurately, as the car body and truck response are strongly influenced by the suspension stiffnesses, damping characteristics, active control schemes (if used) and nonlinearities (e.g., hardening or softening spring effects, friction, hysteresis, and mechanical stops).

Longitudinal Coupling. In the longitudinal direction, a strong coupling must exist to transmit traction and braking forces from the truck to the car body. On motor trucks, this is often done by means of a traction bar (ICE, Shinkansen, ETR500). On trailer trucks, pins or Z-links are used for that purpose. On the TGV-A, an articulated pin mounted on resilient bearings ensures the longitudinal coupling to the truck. Air bags are often used for the vertical suspension and to some degree for the lateral suspension. They have advantages in that they lend themselves to load levelling schemes and their stiffness can be controlled to some extent. Depending on the design of their skirts, a certain amount of lateral and longitudinal movement is also possible.

Damping Schemes. On conventional trains, damping of movements is done between trucks and car bodies, and is usually parallel to the corresponding suspension elements. Some also have anti-yaw dampers between trucks and car bodies. On an articulated train like the TGV, damping is done directly between car bodies via four longitudinal dampers, each at a corner of the car body's end, and one anti-roll damper, across the roof tops. The only car body-truck connections are two anti-yaw dampers.

Tilt Systems. On tilt-body trains (X2000, ETR450), the tilting linkage is between a bolster and the car body. This linkage allows the car body to tilt $\pm 6.5°$ (X2000) or $\pm 10°$ (ETR 450) with a center of rotation slightly above its center of gravity. The secondary suspension is between the truck and the bolster, and includes both vertical and lateral dampers. The tilting hydraulic actuators are between the bolster and the body[3-12]. In addition, the ETR 450 has an active lateral suspension between the truck and the bolster to keep lateral movements of the car body within the prescribed gauge[3-13]. Control of the tilting actions is based on readings of accelerometers and gyroscopes on the trucks and is designed to keep residual lateral acceleration, residual lateral jerk, and car body roll speed within defined comfort norms. In the X2000 and the newer versions of the ETR450, the accelerometer is at the head of the train and the tilt signals are processed in the locomotive and sent with delays to each car[3-12].

3.1.1.7 Car Body and Aerodynamic Design. The car body design can influence the safety-related dynamic response strongly.

Car Body Structural Design. The challenge in the design of an HSR car body is to achieve sufficient rigidity, great longitudinal crush resistance, and a high first mode frequency (10 Hz or higher), while minimizing the car body mass. For good energy efficiency and overall cost-effectiveness, a high strength-to-weight ratio for the car body is desirable. It is important that the flexural stiffness of the car body is sufficiently high so that bending and torsional modes are not excited during operation. If this does occur, it can lead to high stresses in the car body structural and eventual fatigue failures, as well as poor ride quality. Thus, it may be important to model car body flexibility in some HSR modeling tools.

Car Body Shape. The shape of the car body may have a strong influence on safety-related dynamic performance, particularly at the high speeds associated with HSR operation, because it influences the aerodynamic forces and wind loading on the vehicle. Thus, the car body shape should be included in the model through appropriate aerodynamic terms when performing certain types of analysis.

3.1.1.8 Articulated Trains. Traditionally, most aspects of the safety-related performance of trains have been analyzed using single vehicle models. However, articulated trains may require the modeling of more than one vehicle for the accurate prediction of dynamic response. The kinematics of the articulation, and the associated car-to-car and car-to-truck connection stiffnesses, damping characteristics and nonlinearities should be represented in sufficient detail in HSR articulated train models. This may require the development of a separate set of HSR models to handle this class of vehicles.

The TGV is a prime example of an HSR vehicle with articulation. The principle of the articulated car connection of the TGV is shown in Figure 3-1. The carrier ring is mounted on the end face of one body and is supported from the truck pneumatic suspension by carrier tanks mounted on the ring itself.

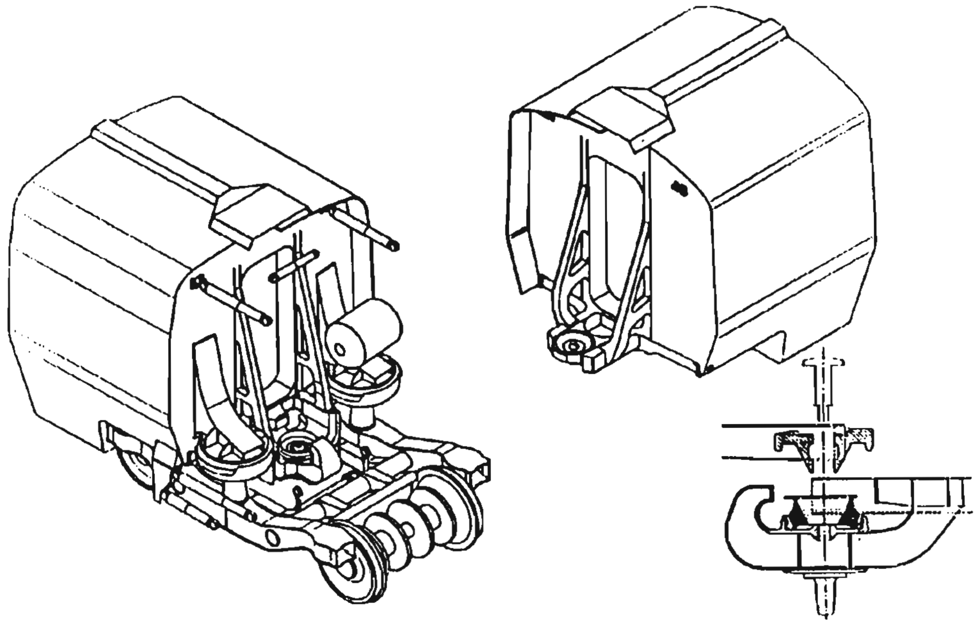

Figure 3-1. TGV Articulated Train Connection

The other body has a welded supporting ring, which is coupled to the carrier ring through a ball joint with resilient bearings. Through the ball joint, a pin extends to the truck providing the path for transmitting longitudinal forces from the truck to the body during braking[3-14].

3.1.1.9 Special Truck Designs. Several "special" truck designs are under development throughout the world. These have some unique features that must be included in an HSR modeling tool for the accurate prediction of dynamic performance. Their aim is to allow good curving (no flange contact) on curves with small radiuses, while still maintaining good stability (no hunting) at high speed. Wheel and rail wear are also greatly reduced. The price of these desirable characteristics has usually been an increase in mechanical complexity, and therefore in acquisition and maintenance costs. These complexities hamper the valid implementation of simple modeling methods.

Asymmetric trucks. Asymmetric trucks are currently being investigated for possible application in HSR[3-15]. Three configurations have been tried:

1. Trucks with asymmetric suspension—the leading wheelset has a soft primary longitudinal suspension while the trailing wheelset has a much stiffer longitudinal suspension.

2. Symmetric primary suspension—trailing wheelset with independently rotating wheels.

3. Both asymmetric primary suspension and trailing wheelset with independently rotating wheels.

Of these three configurations, only the first one presented good curving ability while retaining stability at high speed. The dynamic characteristics of this truck were simulated using the A'GEM Rail Vehicle Dynamics Package.

Independently-Rotating Wheels. Trucks with independently-rotating wheels are being studied in Japan for a proposed new transit vehicle, called the Flip Flop Linear Motor car, with an operating speed of 300 to 400 km/h[3-16]. This type of truck, which has wheels with a cylindrical profile, is not able to steer without some sort of lateral guidance. Steering is provided by a guide-rail in the track center, on each side of which guiding wheels are pressed from both the front and rear of the truck.

3.1.1.10 Performance Measurement Schemes. Several HSR systems use measurement schemes to monitor safety-related dynamic performance in real time during operation. It may be important to model these measurement systems so that their effectiveness can be evaluated under simulated operation near or at the vehicle's safety limits. Examples of these measurement schemes follow.

The TGV-PSE. Despite their inherently high stability, the trucks of the first TGV-PSE are equipped with lateral accelerometers that are continuously monitored from the cab[3-4]. For purposes of track maintenance, a revenue service train, fitted with accelerometers on its axle boxes, and with a system for identifying the exact location of any defects, is run on the main lines once a week. Measurements of accelerations on trucks of axle boxes, however, give only a filtered, indirect indication of the L/V forces at the wheel/rail interfaces. A device for continuous and direct measurement of the wheel/rail forces (L, V, and L/V) has been developed in Japan[3-17]. Such measurement techniques have been already been used in Germany and in other countries; however they are limited to capturing force fluctuations up to 30 Hz.

Shinkansen. On the Shinkansen lines, at high speeds, oscillatory wheel load fluctuations phenomena occur at frequencies of 50 to 70 Hz. These result in derailment quotients that sometimes exceed the limit value of 0.8. To capture this, a new method was devised and a new device was developed which is capable of measuring vertical, lateral, and longitudinal forces acting at the wheel/rail contact point continuously up to 100 Hz.

3.1.2 HSR Guideway Configurations

The HSR vehicle and guideway is a coupled system. It is critical that the guideway and its interaction with the vehicle is modeled with sufficient accuracy so that safety-related dynamic performance can be predicted. In this section, we discuss several important aspects of HSR guideways that influence modeling requirements.

3.1.2.1 General HSR Track Design Features. The following examples indicate the range of track design features used with existing HSR systems:

- The experience of HSR service (250 km/h and above) in the last 20 years has shown conclusively that the classical ballasted track is adequate for routine 300 km/h operations, and has even been found safe in tests that have reached 515 km/h.

- On the TGV lines, the rails are laid on 9 mm thick rubber pad, with a 5 percent inward cant. The flange clearance is kept ideally at 10 to 15 mm. The TGV track rests on heavy twin-blocks concrete ties (245 kg), held down by the spring-loaded NABLA fasteners. The ballast is of high hardness and quite thick (30 35 cm under the ties). This results in a high stiffness from the top of the tie down. Consequently, most of the vertical movement is absorbed by the rubber pad. A high lateral stiffness is another property of this track design. A 170 kN axle load results in a lateral stiffness of 130 kN for compacted ballast[3-19].

- The standard for new ICE lines is a ballasted track with heavy monobloc concrete ties (B70) [3-20].

- In tunnels, the DB uses a concrete slab base.

- In Japan, a "resilient tie track" has been developed for the newest Shinkansen lines (Tohoku and Joetsu)[3-21]. These are concrete slabs which can be laid directly on the ground with cement asphalt injected beneath the slabs to provide for elasticity or on concrete viaduct. Ties are then laid in corresponding slab indentures, on filling concrete. Near the rail joints and between concrete slabs, several ties are embedded in resin mortar for better isolation.

3.1.2.2 Track Geometry Standards. The track geometry standards used by the different systems for lines with operating speeds in the 250 to 300 km/h range are summarized in Table 3-3.

For tilt-body trainsets such as the X2000 and the ETR 450, there are no minimum radius standards, since these systems were designed to run on existing older, curvy lines. Their speed is limited only by a maximum unbalanced superelevation of about 300 mm. This unbalanced superelevation corresponds to a lateral acceleration of somewhat more than 2 m/s^2 at the track level. This is still well below the Prud'homme limit for track shift. Inside the car body, the tilting mechanism reduces this lateral acceleration to below 1 m/s^2, which is well within the accepted norms of comfort. This greater unbalanced superelevation, compared to that of 100 to 120 mm allowable for non-tilting trains, enables a speed limit about 30 percent higher through curves. Modeling of the tilt mechanisms and controls is an important part of predicting the performance of these vehicles.

3.1.2.3 Special Trackwork. Special trackwork (switches, turnouts, etc.) can provide a source of significant transient excitation to a HSR vehicle, and is typically a speed-limiting factor for safe vehicle

Table 3-3. Track geometry standards for HSR system.

	Shinkansen		TGV	ICE	ETR-450
Parameters	**Tokaido**	**New Lines**	**Atlantique**	**New Lines**	**Diretissima**
Max. Operating Speed, km/h	270	275	300	300	300
Max, Grade, percent	2.0	1.5	2.5	4.0	2.1
Min. Vert. Radius - Hill, m	10000	15000	14000		
Min. Vert. Radius - Vale, m	10000	15000	12000		
Minimum Radius, m	2500	4000	4000	3500	5450
Max. Superelevation, mm			180	180	105
Max. Unbal. Super., mm			100		
Formation Width, m	10.7	11.6	13.6	13.7	13.6
Track Center to Center, m	4.2	4.3	4.2	4.7	5.0

operation. Differences in stiffness and mass of various special trackwork components may effect vehicle response and should be modeled appropiately. Examples of special trackwork used in existing HSR systems are described below:

- For high speed turnouts, the SNCF has chosen the UIC60/A61 technology with low symmetrical switch blades and movable point spring frogs. There are two version of this turnout. Type 1/46 allows turnout speeds of 160 km/h, and type 1/65 allows the higher speed of 230 km/h[3-18]. In the future, on the TGV Nord line, the whole turnout will rest on a concrete slab.

- In the Texas Franchise Application of the ICE, specifications for all of the main line switches call for movable point spring frogs. They are the UIC 60 or AREA 136 - R = 2500 m - 1:26.5 for 129 km/h and the UIC 60 or AREA 136 - R = 7000/6000 m - 1:42 for 201 km/h.

3.1.2.4 Elevated Guideways. The dynamic response of an HSR vehicle on an elevated guideway may differ strongly from that on at-grade track, depending on how strongly the vehicle and guideway dynamics are coupled. Further, the transition from elevated to at-grade track may provide a significant excitation to the vehicle because of the change in guideway stiffness and inertial characteristics. Thus, knowledge of the elevated guideway dynamic characteristics is essential to the development of an accurate HSR simulation.

3.1.2.5 Tunnels. The tunnel's diameter can be a key factor to maximum speed and passenger comfort if the cars are not pressure-sealed. Pressure waves are created by the train as its front enters the tunnel and again as its tail enters. These pressure waves move at the speed of sound through the tunnel and a backwave is produced when they reach the other end of the tunnel. The tail-end-forward wave then interacts with the front-end-backwave and again with the train itself, causing resistance and passenger discomfort. While the air resistance in the tunnel is reduced by nose and tail design and by the design of the trainset to minimize drag (flush door mounts, sealed coaches, etc.), passenger discomfort due to abrupt pressure changes is not.

3.2 Maglev Vehicle/Guideway Systems

A discussion of the features of Maglev systems that are important to safety-related dynamic performance is presented below. The modeling of Maglev vehicles is significantly different from HSR systems at the vehicle/guideway interface. The difference is due to the much tighter coupling between the vehicle and the guideway that is inherent in Maglev systems. The two components are coupled through the control laws and parameters specific to each system. Two fundamentally different modes of magnetic levitation are being considered:

- Electromagnetic suspension system (EMS)
- Electrodynamic suspension system (EDS).

In terms of the dynamic performance, the systems have several unique characteristics, which are described in the following sections.

3.2.1 Vehicle

Maglev vehicle configurations may comprise ethior single car body or multi-car body consists. In developing a vehicle model, each car body can be represented by a lumped-parameter or finite degree-of-freedom (DOF) model. The model must approximate the rigid body motions (e.g., translational DOFs accounting for longitudinal, lateral, and vertical motions, and rotational DOFs reflecting yaw, pitch, and roll angles) as well as flexible modes (e.g., bending and torsional modes). Depending on the purpose of the model, such as to study stability, curving behavior, and/or ride comfort, the models may be formulated with a subset of the rigid and flexible body mode DOFs.

For example, a longitudinal DOF may be neglected in a model if vehicle acceleration/deceleration is not to be studied; a first vertical bending mode and/or a first torsional mode may be included in a ride quality investigation if their associated natural frequencies fall within the range of human body sensitivity; lateral bending modes may be neglected if they are excited at frequencies above those of interest; since some vibration and noise problems are caused by elastic deformations, car body flexibility may need to be part of the vehicle model.

Decisions related to the number of DOFs to be represented in a model are system configuration dependent. These decisions are ultimately related to the purpose and required accuracy of the model. The number of required DOF depends on the vehicle type and the dynamic problem under consideration. The individual bodies of the vehicle model may be interconnected by joints and linkages constraining their relative motion as well as by compliant elements resulting in kinematic or dynamic coupling between contiguous bodies. These systems are called multi-body systems. These bodies may be structurally flexible e adding flexible (or distributed) body problems to the vehicle modeling activity.

Vehicle suspensions are of great importance since they determine the dynamic and vibrational behavior of the vehicle through the constraints and interaction forces they provide. Typical models for coupling elements include springs and viscous dampers arranged in parallel or in series. The describing force laws may be rather complex, described by nonlinear characteristics (hardening spring, Coulomb friction, stiction) and described by algebraic or differential equations. Electromagnetic suspensions exhibit nonlinear stiffness and damping characteristics, including the possibility of negative stiffness/damping regimes.

A distinction is made between the primary suspension that provides levitation or guidance along the guideway and the secondary suspension that provides cushioning for the vehicle bodies. Possibilities for primary suspensions are magnetic levitation based on attractive electromagnets or repulsive superconducting magnets. Maglev vehicles with these types of primary suspension systems employ active feedback control, consisting of sensors, feedback control laws, and actuators, for stabilization. Through active components, qualities can be achieved that would not be obtainable via passive systems.

A Maglev vehicle's suspension system is required to maintain the primary suspension air gap while minimizing passenger compartment vibrations in the presence of guideway irregularities and aerodynamic disturbances. It must meet these requirements while minimizing

- The size of the required air gap so that reasonable magnet forces can be achieved
- The stroke length of the secondary suspension to accommodate physically implementable components
- The size, weight, and required power of active suspension elements.

A challenge of Maglev vehicle design is that these goals conflict with the desire to increase the allowable guideway roughness (to reduce guideway construction costs) and minimize the influence of aerodynamic disturbances (such as crosswinds). Active control offers great potential to improve suspension performance. Further research and modeling is needed to determine the optimum Maglev vehicle suspension.

A Maglev vehicle's dynamic response is influenced by the characteristics of the guideway. For example, as the guideway becomes more flexible, the magnitude of the gap variations tends to increase and the magnitude of the car body accelerations tends to increase. Guideway considerations are discussed in Section 3.2.2.

3.2.1.1 Primary Suspension. A Maglev vehicle has vertical and lateral suspension requirements in a simalar fachion to conventional HSR Systems. These forces are provided by the primary suspension, whose role is to transmit the loads for supporting the vehicle (i.e., vertical suspension) and for guiding the vehicle (i.e., lateral suspension) to cause the vehicle to follow the prescribed route alignment.

Traditionally, ground-based vehicles have used a primary suspension with a relatively high natural frequency (5 to 10 Hz) and low damping (0 to 5 percent of critical damping) to closely follow the guideway, and a secondary suspension with a relatively low natural frequency (~1 Hz) and relatively high damping (30 to 50 percent of critical damping) to isolate the passengers.

In Maglev systems, the interaction between the vehicle and guideway is fully described by forces (for lift and guidance), i.e., no kinematic relations need be considered as in the case of rail/wheel contact. These forces are provided by the primary suspension by magnetic levitation. It should be noted that the guidance problem is similar to the levitation (lift) problem except that, in general, the forces are smaller and more variable. When traveling in a straight line only a small lateral force is required; when negotiating a turn or in strong crosswinds the lateral force may reach half of the levitation force. While some Maglev vehicles operate with separate lift and guidance magnets, other concepts are based on have considered one line of magnet arrays that provide combined lift and guidance. In both cases, if the control law is designed properly, the levitation and guidance behavior can be stable and a nominal operating gap can be achieved between the vehicle and guideway. In ehtior case the modeling of the vehicle must include the control system.

There are two major types of Maglev technology based on the primary suspension concepts. One concept is based on the attractive force between an electromagnet and a piece of ferromagnetic material and is called an electromagnetic suspension (EMS). In EMS systems, electronically controlled electromagnets are suspended below and attracted to a steel rail, and position feedback is used to achieve stability. The other concept is based on the repulsive force between a magnet and electric currents (eddy currents) induced in a conductor moving relative to the magnet and is referred to in the literature as electrodynamic suspension (EDS). In EDS systems, magnets move above conducting media in such a way that induced currents repel the moving magnet. These schemes can be turned upside down to have an array of magnets in the guideway that repel a conductor on the lower side of the vehicle.

*3.2.1.2 **Electromagnetic Suspension (EMS)***. Vehicles with EMS systems employ a magnetic suspension that generally wraps around active guideway elements, achieving levitation by attraction upward toward the guideway surface.

EMS systems operate with small air gaps and limited range of air gap movement. The suspension has a characteristically high stiffness (and, generally, a secondary suspension is needed to ensure acceptable ride quality). EMS systems are often designed to distribute the levitation forces over the full length of the vehicle, providing higher levitation effectiveness than a scheme employing concentrated magnetic forces and complicating the modeling of the vehicle system..

An EMS system is inherently unstable. That is, there is no natural restoring force which acts to automatically restore the vehicle to its equilibrium position once disturbed. Since EMS systems are inherently unstable, active control is required to ensure a constant gap between the vehicle's magnets and the rails on the guideway. Typically, the stability is achieved by using position feedback that automatically adjusts the coil current to achieve a nearly constant air gap. The continuous control requirement of EMS systems is made stringent by the small size of the air gap used in such systems. The small air gap provides very little freedom for fluctuations that will result from guideway misalignments, wind gusts, and debris on the guideway.

*3.2.1.3 **Electrodynamic Suspension (EDS)***. EDS technology achieves levitation through repulsive eddy currents generated in guideway-mounted coils. Some relatively high speed (150 km/h or more) is required for sufficient induction to levitate. In order to produce enough force for practical implementation, superconductors are required. Superconducting magnets can generate intense magnetic fields to create large air gaps (~10 cm) between the levitated vehicle and the guideway. They weigh less and require less power to operate than equivalent iron-core electromagnets. EDS levitation is inherently stable and characterized by low stiffness.

There are a variety of variations of the EDS concept. In its simplest form, superconducting magnet coils used for vehicle levitation are placed on the vehicle in such a way that as the vehicle moves, the magnetic fields of the vehicle coils induce eddy currents in conducting strips on the guideway. The interaction between the eddy currents and the magnetic fields produced by the vehicle-borne coils results in a vertical repulsive force. At rest, there is no repulsive force. As the speed of the vehicle increases over the guideway conductor strips, the repulsive force increases until it just balances the vehicle weight, whereupon the vehicle becomes levitated.

Wheelsets in the form of retractable landing gears are required for low speed suspension and for vehicle maneuvering when the power is shut off. As the vehicle speed is increased from zero, the levitation force gradually increases until it equals the vehicle weight and lift-off is achieved. Thereafter, the landing gear can be retracted to reduce aerodynamic drag and the air gap can be adjusted as required. The interaction also produces electrodynamic drag forces, which must be overcome by the propulsion system. The magnetic drag force, due to the electrical dissipation of induced currents in the guideway conductor, increases with speed, reaches a peak at a fairly low speed, and then decreases with speed.

EDS systems are inherently stable. That is, restoring forces are present that automatically tend to restore the vehicle to its equilibrium position when it is disturbed by some perturbing force such as a wind gust, guideway discontinuity, passenger movement, etc. For example, when the vehicle is pushed closer to the rail, the repulsive force increases, tending to push the vehicle back to its original position. This feature eliminates the need for continuous monitoring of the air gap and continuous adjustment of the field strength. Although the EDS system is dynamically stable, additional passive damping in the primary suspension and a secondary suspension are generally required to achieve satisfactory ride quality.

3.2.1.4 EMS vs EDS. The limited range of magnet motion in an EMS system translates into a very stiff suspension system that is incompatible with the more compliant suspension required to meet ride quality criteria. With its larger clearance, an EDS system is less stiff. Both EMS and EDS Maglev suspension systems generally employ secondary suspension systems to be compatible with ride quality criteria. In contrast to the speed dependence of the levitation repulsive force of the EDS system, the attractive force in the EMS system exists whether or not the vehicle is in motion. However, speed-dependent electromagnetic drag forces still arise in the EMS system. Due to the differences in the two systems the modeling of the systems will be significantly different.

3.2.1.5 Secondary Suspension. As noted previously, suspension systems are commonly divided in at least two stages, a primary and a secondary suspension. The primary suspension directly interfaces with the guideway to support and guide the vehicle using magnetic forces. The function of the secondary suspension system on a Maglev vehicle is to provide good ride quality for the passengers while preventing vehicle contact on the guideway and keeping the secondary suspension stroke within practical limitations. As such, the secondary suspension system provides additional isolation of the vehicle body from the guideway to provide acceptable ride quality.

The secondary suspension system can contain both passive and active elements. A passive secondary suspension may consist of air springs, hydraulic shock absorbers, and pendula. An active secondary suspension includes actively controlled elements (hydraulic or electro-mechanical actuators) that exert forces between the car body and the magnet frame. The main disadvantages with active suspensions are their added weight, cost, and reliability penalties relative to the passive system.

The secondary suspension components present restoring forces to translational and rotational motion offsets. The secondary suspension design may preferentially accommodate vertical and roll motions, since these may most significantly affect the car body ride quality, and the implementation may be coupled. For example, the stiffness in roll will depend on the vertical stiffness components and the lateral distance between them. To increase roll stiffness passively, a swaybar can be added. Alternatively, active elements can be added to influence the secondary suspension stiffness and damping characteristics.

It should be noted that there is a trade-off between the secondary suspension stiffness and the effects of the crosswind and guideway disturbances. A stiffer suspension reduces the roll and yaw angles due to crosswind gusts, but again also increases the transmission of guideway irregularities to the passengers.

3.2.1.6 Primary vs. Secondary Suspension. A design trade-off exists between the primary and secondary systems in terms of the distribution of suspension stiffnesses. There is generally a conflict between ride quality and guideway tracking in the choice of the primary suspension stiffnesses, since a stiffer primary suspension generally provides better tracking of the guideway, but at the expense of a larger transmission of guideway disturbances through to the passenger compartment. This is also a factor in the choice of the secondary suspension stiffness.

3.2.1.7 Damping. There is very limited damping in magnetic suspension systems. Damping can be accomplished, at least in part, by passive shock absorbers between the vehicle coils and the load carrying compartment. Alternatively, active damping can be employed to achieve a desired ride quality over a guideway too rough to be used with passive damping alone. There may be an advantage to using an active system in conjunction with a passive damping system. The use of passive damping is highly desirable even if other types of damping are also employed, since it provides a reliable backup system to ensure safety in the event an active damping system fails.

An active suspension system provides continuous or discrete variation in effective stiffness and damping, according to a software-determined (rather than hardware-determined) control law. In a semi-active system, the damping in the suspension is controlled while the effective spring stiffness of the magnetic suspension is not controlled.

3.2.1.8 Aerodynamics. The aerodynamic effects on high-speed vehicles may be significant in comparison to those on slower vehicles since the higher speed results in a larger dynamic pressure. A high-speed Maglev vehicle system operating at 300 mph (480 km/h), corresponding to Mach 0.4 at sea level, will experience aerodynamic loading, such as longitudinal drag and lateral wind effects. It is expected that vertical wind variations (updrafts and downdrafts) on the vehicle will not be nearly as strong as crosswinds.

To calculate the aerodynamic loads and pressure distributions on the vehicle, a full 3D Navier-Stokes analysis, including the effects of viscosity, compressibility, and turbulence, can be carried out using finite-element codes. Although models can be developed to account explicitly for the dynamic effects of fluid/structure interaction, it is usually reasonable to decouple the fluid and structure models and view the aerodynamic effects as producing external loads (forces and moments) that act on the vehicle.

There may be potential benefits of actively controlled aerodynamic surfaces implemented in conjunction with the conventional secondary suspension. The aerodynamic control surfaces can be considered as winglets that exert forces directly on the vehicle body, which, due to high vehicle operating speeds, can produce reasonably large forces even when modestly sized. Aerodynamic control surfaces have the advantage of exerting forces directly on the vehicle without reaction forces on the bogies. Active aerosurfaces mounted on a Maglev vehicle provide for additional control authority. Winglets mounted on the sides of the car body produce vertical forces at the centers of pressure; winglets mounted on the top of the car body (in "rudder-like" arrangements) provide lateral forces.

Two of the Maglev system concept definition teams (Bechtel and Magneplane) came up with horizontal aerodynamic surfaces mounted on the passenger compartment for producing controllable vertical forces. Active control of aerodynamic surfaces is an option, although unsteady air flow may complicate its implementation. In order to evaluate the feasibility and benefits of aerodynamic control surfaces, a dynamics and control analysis can be conducted.

3.2.1.9 Ride Quality. The most significant factor affecting passenger comfort appears to be passenger acceleration. The forces that accelerate the passenger are caused by motion of the vehicle and the vehicle-guideway interactions. The passenger's impression of the ride quality or comfort is also affected by jerk, or the rate of change of acceleration, usually measured in g's/sec. Large magnitude jerk is a fact of motion that a passenger readily senses and finds objectionable since it induces motion sickness. It is a measure of the rapidity with which forces applied to the body are changing. Jerk occurs during both nonoscillatory and oscillatory type movement. Examples of the first type would be abrupt starts, stops, entering curves, and passing over sharp bumps.

In ride quality studies, acceleration and jerk should be evaluated for a range of speeds. It is best to consider the front and rear of the passenger compartment, since the peak accelerations occur at the ends of the vehicle. Generally, ride quality degrades and gap variations increase with increasing vehicle speed.

3.2.2 Guideway

The Maglev guideway constitutes the stationary structure whose principal function is to bear the supporting and guiding loads of the vehicle. It also contains electronically active elements required for Maglev vehicle propulsion and speed control, including starting and stopping functions. Since the vehicle is confined to move along the guideway, there must be provisions to allow for branching out and merging together of the various routes through guideway switching mechanisms.

The guideway consists of a sequence of spans that are generally elevated and supported at the ends, and sometimes in the middle. The spans can be modeled as distributed flexible beams. Boundary conditions are specified depending on the physical configuration. For example, if the spans are constrained in translation but free in rotation, a pinned-pinned model is appropriate. If the spans are resting on elastomeric pads at the ends, a free-free beam model with a vertical lumped stiffness may be a more realistic representation.

The mathematical models generally assume homogeneous beams with prismatic geometry. This may be an acceptable approximation assuming the inertia and cross-sectional geometry do not change along the length of the span. If the geometry and/or make-up do change (such as spans with thinner flared sections at the center and/or spans with embedded devices at discrete locations along their length), then it may be necessary to incorporate a more complicated beam description. If the guideway is not elevated but "at-grade" as in a tunnel, then the beam model may be replaced with a distributed lateral/vertical stiffness model, such as a beam-on-elastic-foundation model similar to that developed for railroad track. Furthermore, the mathematical models generally assume straight (tangent) guideway spans. Since tangent as well as curved, banked spans may be encountered by a Maglev vehicle, both situations need to be modeled. Curved, banked beam models have been developed, but are mathematically more cumbersome.

The guideway is subjected to various loads, such as distributed or concentrated (discrete) forces, depending on the magnet module configuration of the Maglev vehicle. These vehicle primary suspension loads, which act as inputs to the guideway model, are generally a function of the vehicle dynamics (as described above). One can distinguish between two vehicle input conditions:

a. The vehicle assumes a certain position at the guideway and is not traveling ($V=0$) but "hovering"

b. The vehicle is traveling with a certain speed along the guideway.

Situation (a) is the more critical case for Maglev vehicles, since if the vehicle is hovering at one spot instabilities associated with the vehicle-guideway interaction can build up.

It is also possible to identify two theoretical models of guideway dynamic response:

1. A discrete model that treats the column pier supports and suspension cables as discrete elastic elements. This model is appropriate model for elevated guideways.

2. A continuous model that uniformly distributes the elastic restoring forces and moments caused by continuous ground support. This model is appropriate for representing at-grade guideways.

Both guideway models can be developed to predict the dynamic behavior (including resonant frequencies) for guideways subjected to moving vehicles. The vertical, lateral, and torsional deflections of the guideway can be considered in both models. (Although generally not considered, longitudinal dynamics of the guideway could be included to permit investigations of braking and accelerating phases of Maglev vehicles.)

3.2.2.1 Elevated Guideway. The dynamic deflections and stresses of elevated guideways caused by moving transit vehicles differ from those experienced by the guideway with stationary vehicles.. This is because the loading and unloading of a structure in a short time induces transient vibrations and because vehicle dynamics can create loads that are applied to the guideway in a time-varying fashion. The degree to which the dynamics affect the guideway deflections and stresses depends on the vehicle characteristics and speed and the guideway span length, stiffness, mass, and other characteristics. These must all be modeled when considering a Maglev model.

In elevated vehicle-guideway systems the majority of effort has been devoted to characterizing the vertical plane interactions in which the guideway is excited by vehicle weight and inertial forces and the vehicle is excited by the vertical guideway profile. It is also possible to consider lateral plane as well as rotational (e.g., roll) motions. Fundamentally different formulations of a model may be necessary to consider these additional DOF's.

To handle many sequential spans, it is possible to model only two sequential guideway spans (corresponding to odd and even numbered spans, respectively). Logic in the computer program can determine which span a given component of the vehicle is over and select the appropriate set of guideway modes. When the vehicle has completely passed over a span the set of modes for that span is initialized to become the modes for the next span the vehicle will encounter (i.e., two spans ahead).

3.2.2.2 At-Grade Guideway. Beam-like guideways resting on the ground have been represented in analyses as beams on elastic or visco-elastic foundations. Following the classical work of Timoshenko and others, some have developed solutions for the beam on an elastic foundation with a traveling vehicle load.

3.2.2.3 Guideway Irregularities. Irregularities occur in a guideway as a result of construction practice, settlement, dead weight loads and environmental conditions. The guideway static irregularity profile may be represented as the summation of a number of effects:

- Span vertical offset resulting from differential span height (i.e., discontinuous alignment of adjacent guideway spans)
- Span angular misalignment resulting from differential pier height
- Span camber resulting from dead weight loading (sagging due to creep and long term loading), intentional precamber, or thermal effects (temperature gradients)
- Surface roughness caused by local surface variation.

Because these irregularities result from a wide variety of effects, including construction tolerances and environmental conditions, they are often represented as random irregularities. In some studies it is assumed that the guideway profile is a realization of a random process that can be described by a power spectral density (PSD) function. The profile is modeled as a stationary Gaussian random process, that can be generated by an inverse Fourier transform of the PSD function. Measurements of guideway profiles provide guidelines with respect to the amplitude probability distributions and amplitude spectral densities for various irregularity types. Guideways have irregularity spectra that typically consist of large amplitudes at long wavelengths and small amplitudes at short wavelengths. Long wavelengths typically represent route alignment while short wavelengths are typically due to surface roughness and assembly tolerances.

Guideway profile deviations such as curves, grades, geometric irregularities, misalignments, and elastic deformations have adverse effects on the dynamic stability, ride comfort, and safety of high-speed Maglev vehicles. On the other hand, accommodating irregularities and elastic deformations, especially for elevated guideways, is desirable in order to reduce guideway construction and maintenance costs. The guideway designer may be required to limit the span deflections and other irregularities as a way of limiting vehicle motion and providing passenger comfort.

3.2.2.4 Aerodynamics. Elevated guideway spans may be subject to aerodynamic loads, that may induce flutter under strong wind loading. The sources of these forces are ambient wind and vehicle reaction forces such as drag, downwash, etc. A detailed analysis would involve determination of vortex shedding off the guideway, lift on the guideway span due to ambient wind and other environmental effects.

3.2.2.5 Switching. For a guideway to diverge to two or more paths, a mechanism is required to switch a moving vehicle smoothly from one path to another. One approach is to accomplish the switching operation by having a section of the guideway bend to direct a vehicle to one of two paths. Electromechanical or hydraulic actuators can be employed at movable spans to bend the guideway. The movable spans are supported on a transverse support frame with wheels to allow for lateral movement of the guideway. Since the switch is a movable mechanism of the guideway it has charactoristics that are different than the nominal guideway and may require different guideway modeling charactoristics.

3.2.3 Control

The performance of a Maglev vehicle under a wide range of maneuvers such as elevation changes (vertical curves) and coordinated turns (horizontal curves) along with disturbances acting on the vehicle such as guideway irregularities and wind gusts, is directly dependent on the design characteristics of the levitation and lateral guidance control systems. The control system requirements include levitating and guiding the vehicle, and simultaneously providing a comfortable ride quality for the passengers.

The controller and associated hardware serve the role of primary (and sometimes secondary) suspension components, in that they provide "effective" stiffness, damping, and inertia components between the vehicle and guideway. There are many modeling and design decisions related to the controller, including specification of configuration (such as linear vs. nonlinear control logic, measurement of feedback signals, selection of gain values.) Since the magnet characteristics are generally nonlinear, a linear controller may not be successful over a range of gap errors. It may be necessary to include a gain-scheduled controller (i.e., a controller whose gain values change as a function of gap error or another variable) or to develop a fully nonlinear controller.

In the selection of the controller the following specifications maybe important:

- Required speed of response
- Robustness (or the ability to accommodate disturbances)
- Optimal effective stiffness/damping or the "operating point" stiffness/damping
- Power consumption
- "Fail-safe" (or "fail-soft") design requirements.

The controller architecture may be distributed, i.e., a high level controller may monitor the gap error and its derivative(s), and may invoke different lower level controllers accordingly. Clearly, different controllers would be employed for EMS and EDS systems, and for levitation, propulsion, and guidance.

To investigate the improvement of the dynamic response and ride comfort of Maglev systems, different control designs (active and semiactive) should be examined. For control-law synthesis, it is desirable to work with linear dynamic models of low order and increase the model complexity as added design fidelity is obtained. The aim is to generate a robust and flexible controller. Most likely, it will be necessary to extend the controller strategy to incorporate adaptive methods and nonlinear control as well as fault monitoring capabilities. Further, preview control methods should be explored.

Uncontrolled vehicle contact with the guideway is considered unacceptable. The vehicle levitation and guidance functions can not be lost for any combination of system failures, and the vehicle must maintain its own suspension until it is brought to a stop by either high-level control or its own internal (low-level) control system. Some aspects of control can and should be done by passive means in order to provide a "fail soft" mode, as well as possible. The active/passive tradeoffs and distribution should be explored in the controller model.

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4.0 Modeling and Simulation Requirements

In this section, we describe techniques for modeling HSR and Maglev systems, including solution techniques, modeling requirements and criteria for selecting models for safety-related dynamic performance assessments.

The technical issues involved in the dynamic modeling of HSGGT vehicles and guideways are multifaceted. The dynamicist (modeler) must have a broad knowledge of mathematical modeling techniques and computer programming experience, and must also have an understanding of vehicle dynamics and a working knowledge of vehicle and guideway configurations. HSGGT systems "push the envelope," to speeds at which there is relatively little operating and maintenance experience.

Predicting and understanding the effects of both vehicle and guideway parameter variations due to aging and wear on high-speed stability, curving performance, and operating safety are critical to the success of an HSGGT system. For Maglev, little experience has been accrued in the dynamic loads generated between the vehicle and guideway, in fault tolerance, and in high-speed operating safety. For both systems, aerodynamic effects are important in terms of operating costs, comfort, and safety. For these and many other reasons, the development of credible, accurate HSGGT vehicle-guideway models has become a vital goal.

Modeling requirements are predicated on the need to satisfy a number of different types of uses. These are, for example:

- HSGGT vehicle or guideway structure designers, who need to focus on specific parameter characteristics as part of their detailed design objectives
- HSGGT system designers, who need to verify the design parameters, capabilities, and compatibility of both vehicle (trainsets) and guideway structures
- Theoreticians, who require an academically defensible model to investigate trends and effects, and to conduct parameter variation studies
- Regulatory authorities, who need to verify safe operation as part of the system certification process
- "Forensic engineers," who need a quick response capability in their troubleshooting role investigating operating problems or accidents.

Specific modeling requirements depend on the focus and goals of a particular user. The choice of a specific model or modeling approach must be tailored to satisfy these user goals. "All-purpose" models generally cannot satisfy the complete range of needs of all user groups. Therefore, a modeler's "toolbox" approach has distinct merit in providing a wider range of choices and capabilities to satisfy the matrix presented in Figure 1-1 and Table 2-1.

Some of the basic requirements of vehicle and vehicle-track interaction modeling are addressed in the following sections.

4.1 Modeling of HSR Systems

Several major factors must be addressed in the modeling of a high-speed rail (HSR) vehicle and guideway. For the vehicle, these factors include lateral stability, curving performance (including curve entry and exit dynamics), and forced response due to guideway geometry or environmental influences such as wind gusts. For the vehicle and guideway as a total system, the interactive dynamic response can be a major influence on safety and ride quality, particularly with the more flexible elevated structures. Vehicle-guideway simulation models must be able to address these several factors in order to provide an adequate prediction of system safety, component loading and fatigue life, and vehicle ride comfort.

4.1.1 Regimes of Dynamic Response

Three regimes of safety-related dynamic response have been considered: stability, curving, and dynamic forced response. Overviews of these regimes are provided below.

4.1.1.1 Stability. Lateral stability of high-speed vehicles is of primary importance to assure safe operations. Because of the complex wheel-rail contact and creep relationships, the wheelset differential equations of lateral and yaw motions contain forward velocity-dependent and "negative" damping (positive feedback) terms. This results in complex speed-dependent zones of stability and instability in a phenomenon similar to the aircraft wing flutter problem. Coupled lateral and yaw wheelset oscillations with a rail vehicle running above its "critical" speed can grow in amplitude to a stable limit cycle with hard flange contact with the rail. If severe enough, wheel climb derailment can occur.

The lateral stability problem was studied extensively in the 1970s and early 1980s by researchers such as Wickens[4-1], Law and Brand[4-2], Hadden and Law[4-3], Hull and Cooperrider[4-4], Hannebrink et al.[4-

[5], and Horak and Wormley[4-6].* These studies highlighted the importance of the wheel-rail contact geometries, the influence of truck component and suspension characteristics, and the effects of inherent system nonlinearities. Vehicle lateral stability studies have in the past relied on linear models to predict critical speeds and oscillatory mode shapes through eigenvalue/eigenvector solutions of sets of differential equations. These solution techniques can, at best, use describing function approximations of the crucial nonlinearities.

The rail vehicle stability problem is to determine the minimum vehicle forward velocity (critical speed) at which a perturbation to the dynamic system will cause a divergent mode of oscillation[4-8].

With modern computers, time-domain solution of a comprehensive nonlinear model is possible to investigate lateral stability and the resulting dynamic response. This allows use of the full nonlinear wheel-rail contact equations and can account for flange contact with lateral motions exceeding the wheel flange-to-rail clearance. These nonlinear characteristics are addressed in the next section.

4.1.1.2 Curving Behavior. Curving performance is generally at odds with lateral stability. The wheel-rail contact geometry and truck frame rigidity that is optimum for stability will result in non-optimum curving ability. In addition to the basic centrifugal effects on equipment and passenger comfort in curves, wheelsets in curving can generate track gauge spreading forces that can cause wheel and rail wear, and in the extreme, rail rollover or wheel climb derailment. Curve entry and exit through transition curves (spirals) create transient dynamics that can cause track safety-related forces as well as passenger ride discomfort.

Wheelset curving phenomena have also been studied extensively in the past 25 years by a number of authors. Pivotal papers by Newland[4-9], Elkins and Gostling[4-10], Elkins and Eickhoff[4-11], and Wormley, Hedrick and Nagurka[4-12], among others, have established mathematical relationships at the wheel-rail interface. Models for studying curving phenomena have ranged from relatively simple steady-state curving solutions to comprehensive nonlinear vehicle and guideway models to study the dynamics of curve entry or turnout negotiation. These models have included some of the more complex vehicle configurations such as the UTDC steering truck[4-13]. The complex wheel-rail contact characteristics are most important in the modeling of both dynamic and steady-state curving response and the accurate prediction of loads on the guideway.

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* References appear at the end of this section.

The system equations are solved by first determining the normal and tangential loads on each wheel in the plane of the track. The load normal to the contact patch is calculated from coordinate transformation. After determining the forces and moments at the contact patch, a reverse coordinate transformation is done to reestablish the forces in the plane of the track. At some point in this series of calculations, it must be determined whether one or two point wheel-rail contact exists: tread and/or flange support of the loads. Different approaches and computational algorithms have been used to model flange contact, the apportionment of loads between tread and flange, and the readjustment to a force equilibrium condition.

There are several viable formulations for calculating nonlinear creep forces. Some of these are approximate or ad hoc methods to account for the creep forces and moments limited by the wheel-rail adhesion level. A rigorous treatment of nonlinear creep forces is due to Kalker[4-7]. Here the creep forces and moments are nonlinear functions of the lateral, longitudinal, and spin creepages, which are nonlinear functions of the state variables and the wheel-rail geometric constraint functions. In addition, creep forces are also nonlinear functions of the normal forces acting between wheel and rail at the contact patch, as well as the radii of curvature of the wheel and rail profiles at this contact patch. Excellent agreement between Kalker's full nonlinear creep model and experiments has been reported for uncontaminated surfaces. However, reasonably large discrepancies have been noted for contaminated rail[4-15], which is the more usual case in revenue service.

Typically, corrections are introduced into the forces by, for example, reducing the linear small-creep coefficients by 50 percent to account for contamination, or by reducing the coefficients proportional to an assumed coefficient of friction.

By the mid-1980s, rapid advances in computers made it practical to include these effects in a nonlinear model of the dynamic response of a rail vehicle. The nonlinear creep forces and wheel-rail constraints are impossible to describe in concise, closed-form equations. Rather they are calculated by algorithms implemented in subroutines and function subprograms within digital simulations of the vehicle dynamics. The trade-offs in doing this have always been fidelity (and complexity) of modeling versus computational efficiency and subsequent analysis effort.

To aid in computational efficiency, different formulations such as those in FASTSIM (based on Kalker's simplified nonlinear theory of creep[4-16]) and a heuristic model for creep based on Vermeulen and Johnson[4-17] have been used. These formulations have been shown to give fairly accurate results over the entire range of the force curve, at the same time reducing computation time significantly. The AAR's NUCARS program, for example, uses a four-dimensional look-up table of FASTSIM values as a function of wheel-rail relative lateral position, with linear extrapolation between points, to calculate creep forces. CMRI's DYNCUR and Battelle's VEHDYN3 use the heuristic creep force model.

In modeling vehicle curving dynamics, one must account for the yaw and roll accelerations induced by changes in curvature and superelevation (track cant angle) in the "spiral", the transition track geometry between tangent and fully curved track. Ideally, Euler angles and cross-product terms would be involved in a full 6 degree-of-freedom model of each body. Most rail vehicle models, however, make small-angle assumptions and use the "local" track frame of reference for each wheelset. Supported bodies -- truck frames, bolsters, car body -- must then be referenced to the wheelsets at different points in the spiral and curve, resulting in appropriate position offsets. Centrifugal and gravitational accelerations on each body are determined from the local radius of track curvature and superelevation.

4.1.1.3 Dynamic Forced Response. The forced response of vehicle and guideway by geometry errors, both random and repetitive, and by variations in guideway compliance has been the subject of many studies. Forced dynamic response may be transient in nature (a turnout or grade crossing, for example) or may be repeated guideway features such as rail joints and welds. Forced response to the spectral components of staggered-joint bolted rails, for example, includes the high c.g. freight car "rock 'n' roll" problem (16 to 19 mph) and a higher-speed yaw-lateral response with passenger cars (40 to 50 mph). In both of these examples, repetitive track geometry errors excited vehicle response at a primary resonance. In the first case, the heavy freight car was excited at its low-center roll resonance. Investigating this phenomenon required a computer model that could account for significant nonlinearities, including side bearing and "gib" clearances, centerplate and wheel "lift-off", and spring group compression limits. The roll resonance produced the classical nonlinear "jump" phenomenon, where the peak roll response was much more severe when slowing down through the critical speed than when accelerating through this same speed range. A linear representation of the same vehicle would miss this critical aspect of dynamic forced response.

The repetitive variation in flexibility of elevated guideways has been addressed in studies of transit and high-speed trains[4-18], and more recently Maglev vehicles[4-19]. The "forced" excitation in this case can consist simply of deflections of the guideway under the quasistatic loading of each passing wheelset or magnet pod. These disturbances excite the elevated guideway or bridge beams into bending vibrations in at least several modes. These modal vibrations can be reinforced if train speed and the geometries of spacing, beam length, and support conditions are sympathetic.

Forced response can cause fatigue damage to both guideway and vehicle components and, in extreme cases, failure and derailment. An example of this was noted in the failures of truck frames and traction motor support bolts due to severe track conditions on the New York City transit lines in the early 1980s. Relatively rough track geometry on the Northeast Corridor (NEC) in the 1970s caused fatigue failure of the pantograph and smoothing reactor mounts on at least one European locomotive tested by Amtrak.

Modeling requirements for studying forced response may range from linear or quasilinear models using frequency-domain techniques and power spectral density (PSD) representations of guideway geometry errors to complex nonlinear time-domain solutions with transient or repetitive space curve representations of guideway geometry. Variations in elevated guideway compliance and the resulting dynamic response of the structure may be handled as separate degrees of freedom in a mode summation approach. This technique has been used successfully to study not only the effects of vehicle body bending modes, but also to evaluate the effects of dynamic loads on guideway elevated spans and bridges.

4.1.2 Solution Techniques

Modeling of HSR vehicles and guideways may be approached by frequency-domain techniques (for example, random, statistically-based inputs and results) or by time-domain techniques (transient, deterministic inputs and responses). Frequency-domain models are better tailored for ride quality and fatigue-related load evaluations. The weakness in this technique is the need for linearization, where system nonlinearities must be handled by approximations, such as the describing functions. Time-domain models handle the nonlinearities in a straight-forward manner and can simulate limit cycles, even chaotic behavior. These models can provide peak loads and accelerations on vehicle and guideway components, and can be configured to predict wheel climb and other safety-related limiting events.

4.1.2.1 Frequency Domain Solutions. Frequency domain models are of necessity linear or "quasilinear" in structure. The equations of motion are set up in a matrix format, and solved [4-20] using a Laplace transform method. The equations of motion may be developed using Lagrangian or Newtonian mechanics, and the complex variable elements are separated into the real and imaginary parts of the matrices. The complex matrix then is inverted using standard matrix inversion subroutines (a Gaussian substitution technique, for example), multiplying the result by the input column matrix. Track geometry inputs matrix are handled at trailing axles by phase-shifting the input at the leading axle.

For a frequency domain model to provide useful results, realistic inputs must be used. Track geometry irregularities tend to exhibit a random variation in amplitude and wavelength. In addition to random geometry errors, track will display particular spectral peaks related to spatially repetitive events such as rail joints or welds. These may be added to the random geometry power spectrum through separate closed-form functions.

By assuming the track geometry to be a stationary random function over a broad frequency range with a Gaussian amplitude distribution, the response spectrum for each output variable may be calculated from the linear model. Solutions are obtained for the individual random geometries (surface, crosslevel, etc.) and an overall root-mean-square (rms) response calculated.
These rms responses can represent vehicle body accelerations and forces, and can be used to estimate various ride quality indices such as the NASA, W_z, Peplar, and ISO criteria.

As stated previously, the weakness of the frequency domain approach is the need to linearize all elements in the simulation, including the fundamentally nonlinear wheel-rail contact elements. Describing function methods are commonly used to approximate nonlinearities such as hardening springs and Coulomb friction[4-22]. Typically, the first term of an infinite series representation of the nonlinear function is used.

4.1.2.2 Time Domain Solutions. Time domain solutions, while computationally less efficient than frequency domain solutions, have the advantage of representing nonlinearities explicitly.

The time domain simulation model takes a more direct approach to the solution of a set of nonlinear differential equations representing the vehicle and guideway. Again, these equations may be developed using Lagrangian or Newtonian mechanics. The vehicle and guideway may include rigid-body (lumped-parameter) representations, or flexible-body (distributed-parameter) modes. Body degrees of freedom may be set up as first-order state variables or as second-order equations. In either form, a time integration routine is used to predict body accelerations and motions based on inter-body forces and torques. These may range from the simple Euler approximation to more mathematically complex routines. The most commonly used method is the 4th-order Runge-Kutta algorithm. The use of an appropriate time step is a critical compromise between computational accuracy and solution time and cost. Some time domain programs include a variable time step feature to improve solution efficiency.

Inputs to a time domain model may include track geometry variations (rail surface, alignment, gauge, and crosslevel; track curvature and superelevation; and wheel-rail contact parameters), track modulus or guideway beam support stiffness variations, wheel tread geometry variations (wheel flats or runout), inter-car forces (buff and draft loads due to train action), traction or braking torques, or externally applied forces such as wind gust loads. These inputs may be transient, repetitive, or steady state.

Geometry variations represent rail position and rate-of-change of position (velocity) as a function of time (distance/velocity along the track). In the vertical direction, the surface geometry errors are "pulled through" the nonlinear Hertzian wheel-rail contact stiffness to generate time-variable vertical dynamic forces. Changes in relative wheel-rail lateral position and velocity due to line and gauge errors cause variations in creep forces. These are weakly coupled to the wheelset up to the point of flange contact, at which point the wheel and rail are strongly coupled dynamically.

Outputs from a time domain model can include time histories of body accelerations, absolute and relative velocities and displacements, forces (for example, at suspension elements and at the wheel-rail interface), wheel-rail lateral-to-vertical force (L/V) ratios, and guideway bending moments. Maximum/minimum peak values, total energy dissipation, and wheel-rail wear indices may also be computed as part of the solution. Time history outputs may be post-processed by Fast Fourier Transform (FFT) algorithms to provide estimates of their frequency response. High-speed stability (hunting) can be investigated by inducing small perturbation inputs as train speed is incrementally increased and checking for divergent response. The disadvantage of using time history solution methods to perform frequency analysis is the extensive computational requirements needed to perform comprehensive investigations. Days or weeks of model runs may be necessary to generate sufficient time history outputs for a parametric study. Similar time may be necessary to complete analysis of the time histories.

4.1.2.3 Steady-State Solutions. Steady-state solutions represent limiting cases of frequency or time domain solutions where system inputs are uniform or unchanging. This is particularly true of vehicle curve negotiation where the specific interest is, for example, car body accelerations and deflections in the context of ride quality and overturning safety, or average wheel-rail forces in the context of wheel and rail wear. While steady-state solutions provide limited information, there is a substantial savings in computational time and cost.

The earlier rail vehicle curving models were based on steady-state solution of a set of algebraic equations[4-23,4-24,4-25]. To handle the important nonlinearities such as wheel-rail flange contact, creep saturation, and suspension stops, an iterative solution of this set of equations was necessary. All different possible wheel flanging configurations were tried in a program do-loop until force and torque equilibrium was achieved. Program outputs included individual wheel creep and flanging forces, wheelset, truck frame, and car body displacements, and estimates of wheel-rail flange wear.

Steady-state solution of curve overturning safety has also been programmed[4-26] to explore the effects of suspension stiffness and stop clearance parameters, car body c.g. height, and vehicle weights versus curvature, speed and superelevation. This type of program, however, does not provide an estimate of dynamic response due to curve entry and exit geometries.

4.2 Considerations for Modeling Maglev Systems

Several major factors must be addressed in the modeling of a Maglev vehicle and guideway, and these are similar to those mentioned in Section 4.1 for HSR systems. For example, the dynamic response of Maglev vehicles can be categorized in the three basic regimes of stability, curving performance (including curve entry and exit dynamics), and forced response due to guideway geometry or environmental influences such as wind gusts. Further, the vehicle-guideway dynamic interaction can be a major influence on safety and ride quality, particularly with the more flexible elevated structures.

To investigate the safety and performance of a high-speed Maglev vehicle-guideway system subjected to various perturbations, such as wind gusts, centrifugal forces in cornering, and guideway irregularities, analytical models can be formulated and computer simulation studies can be conducted. Typically, a time-domain analysis (in contrast to a frequency-domain analysis) would be conducted, since it can account for nonlinear and nonperiodic effects. The inputs that may be included in a time-domain dynamic analysis are: suspension forces such as lift and guidance forces and (passive and active) damping forces; "virtual" forces resulting from accelerations caused by grades and curves; wind gusts and turbulence, and loads due to guideway geometry (misalignments and roughness) and flexibility. The result of analytic modeling and computer simulation would be predictions of the dynamic motions and forces resulting from vehicle-guideway dynamic interactions. If the modeling and simulation have been successful (i.e., the model is faithful and the algorithm has been implemented correctly), these predictions relate to actual passenger safety and comfort, as well as mechanical demands (e.g., stresses which correspond to fatigue and wear) of vehicle and guideway components. The predictions can also be used to specify guideway structural requirements and vehicle operating practice.

In a time-domain analysis, the vehicle-guideway dynamic equations can be expressed as a set of coupled second-order differential equations of motion representing the vehicle and guideway and their interaction. These equations are nonlinear if finite (sizable), multi-dimensional motions, nonlinear suspension components, and/or flexible modes are included in the analysis. The vehicle and guideway equations are dynamically coupled due to the suspension forces that act (and react) simultaneously on the vehicle and guideway. The vehicle and guideway equations may be strongly coupled dynamically if suspension dynamics exert a significant influence upon guideway deflection or may be weakly coupled if dynamic suspension forces are small (compared to the static vehicle force). In the latter case, the suspension forces are essentially constant and independent of the vehicle dynamics.

When the vehicle-guideway equations are strongly coupled (i.e., dynamic suspension forces are an appreciable fraction of the weight), the vehicle and guideway forces must be solved simultaneously. The principle tool for solving the generally nonlinear equations of motion is direct numerical integration. A candidate integration routine would be a fourth-order Runge-Kutta algorithm (possibly with error control and variable step-size), used to time-step through the coupled guideway and vehicle motions as a function of time. When the equations are weakly coupled, the vehicle suspension forces may be approximated as constants, and the guideway equations may be solved independently of the vehicle dynamic equations. In this case, the guideway deflection can be predicted first, and the vehicle time history response then determined by numerical integration of the vehicle equations. In modeling high-speed Maglev systems, it will generally be necessary to consider the guideway and vehicle as a dynamically coupled system.

A significant number of parametric studies have been conducted in which a moving vehicle is represented by a traveling constant value discrete or distributed load. The assumption of a moving constant force neglects the inertial effects of the vehicle. This is justifiable only if the carbody (suspended or secondary mass) moves without vertical deflection and the primary mass (unsuspended mass in contact with guideway) can be neglected. The other extreme is the case of a moving mass, where the main mass of the vehicle follows the guideway motion exactly (very stiff suspension). A Maglev vehicle can approach both extremes: the first (zero inertial effects case) if tight level control is applied (to maintain the horizontal position of the vehicle independent of the guideway) and the second if tight gap control (to maintain a constant gap between the vehicle and guideway) is applied.

Modeling and simulation always involves simplifying assumptions made in order to simplify the derivation of the equations of motion and reduce the numerical effort. This means that there are always features not studied in a given model. For example, a Maglev vehicle-guideway dynamics study may not consider magnetic field magnitudes in the passenger compartment and surrounding areas. If it is critical, it can be incorporated in the dynamics model, or, alternatively, it can be introduced as a function fitted to the results of a separate program that calculates the magnetic forces.

4.3 Hierarchies of Models

Vehicle/guideway modeling is an abstraction of the considerably more complex real-world physical problem. An effective approach to modeling frequently applied is to use only as many degrees of freedom as appropriate to focus on phenomenon of interest. This becomes something of an art, to truncate and streamline a model, but not to omit esential components of the problem or invalidate the results. Conversely, the inclusion of unnecessary effects increase the need for input data, increases solution and analysis efforts, and increases the potential for invalid results. All-encompassing simulation models, on the other hand, tend to be very difficult to provide input for, difficult to execute, and then they obscure the "trees with the forest" since so much output is generated.. The typical approach is to have available a substantial library of vehicle-guideway models to explore a wide range of different problems, from highly specific problems such as the rail corrugation phenomenon or the fatigue failure of traction motor support bolts, to more general problems such as the response of a Maglev vehicle on a flexible elevated guideway to a wind gust.

The frequency bandwidth of interest, which depends on the specific problem at hand, is an important criterion in the development of vehicle-guideway models. If stability, curving, and ride quality are of primary interest, a frequency bandwidth of about 100 Hz may be sufficient. This effectively limits the number of degrees of freedom and distributed (bending) modes necessary in the model. Such problems as wheel-rail impact loads and rail corrugations require a frequency bandwidth in excess of 1000 Hz. Modeling these phenomena requires additional degrees of freedom such as axle and rail bending modes, and at the same time may neglect degrees-of-freedom associated with some of the lower frequency modes, such as the car body mass on the secondary suspension. These tradeoffs are essential in modeling to achieve a practical program size and reasonable computer solution efficiencies.

The complexity of a vehicle model can be bounded by some prior knowledge of the level of dynamic response. For example, it is usually expeditious to separate the vertical (bounce/pitch) and lateral (roll/yaw/lateral) modes of motion into separate models when relatively small motions and minimal "crosstalk" could be assumed. This is possible, for example, in modeling a rail passenger vehicle response to random (PSD) geometry variations of Class 6 track. It would not be true, however, for a freight car's response to Class 2 BJR track at the roll resonant speed. Because of strong cross-axis interactions, a full three-dimensional model becomes necessary for the freight model. Examples of this are the rail corrugation model, where the creep forces are strongly coupled to the vertical contact force, and the curve entry models, where centrifugal effects and a changing frame of reference must be addressed.

4.4 Model Selection Criteria for Evaluating Safety-Related Performance of HSGGT Systems

Tables 4-1 and 4-2 describe model selection criteria for HSR and Maglev vehicles, respectively, by identifying the aspects of loss of guidance, ride quality and structural integrity, that could be evaluated by models developed for each dynamic performance regime. These tables represent a summary of the discussions provided in the previous sections of this report, and provide a basis for the identification of specific simulation codes for the HSGGT analytical toolkit in Section 6. The tables are a further expansion of Table 2-1. The safety related dynamic performance issue is presented in the left hand column. A number of specific different dynamic issues are presented as column headings across the page. Specific models are identified in the matrix positions of the Table. TD#1 refers to Time Domain Model Number 1 as presented in Table 6-1. In this way a specific model or class of model can be identified as necessary to perform the analysis associated with a set of safety and performance criteria. Since many of these models are written in Fortran they are readily chanageable to meet specific configurations and needs.

Table 4-1. General HSR modeling requirements for the HSGGT analytical toolkit.

			DYNAMIC PERFORMANCE REGIME					
SAFETY-RELATED DYNAMIC PERFORMANCE ISSUE	SAFETY-CRITICAL EVENT	PRIMARY RESPONSE VARIABLES	Vehicle/Guideway Interaction	Dynamic Curving	Longitudinal Dynamics	Steady-State Curving	Frequency Response	Stability and Modal Response
Loss of Guidance	Vehicle Derailme nt (Wheel Climb & Vehicle Overturn)	Critical Speeds	TD#1 - Transient Response to Track Inputs, Wind Loads					FD#2 - EV
		Limit Cycle Conditions	TD#1a - Transient Response to Track Inputs, Wind Loads					FD#3 - QL
		W/R Forces and Motions	TD#1 - Forced Response to Special Trackwork and Discrete Anomalies and Wind Loads	TD#6 - Curve Entry and Exit Dynamics	TD#8 - Response to Hard/Emergency Braking in Curves	SSC#1- NL Constant Radius Curving on Smooth Track	FD#1-Response to Periodic and Random Track Inputs	
	Guideway Failure (Rail Rollover, Gauge Widening, and Track Panel Shift)	W/R Forces, Rail Displacements	TD#2 - Forced Response to Special Trackwork, Track Misalignments, Track Irregularities, Wind Loads	TD#7 - Curve Entry and Exit Dynamics	TD#9 - Buff/Draft Forces During Curving with Braking and Traction Forces	SSC#2- NL Constant Radius Curving on Smooth Track		
Ride Quality	Excessive Passenger Acceleration & Vibration Levels	Car Body Acceleration Response, Ride Quality Indices	TD#3 - Forced Response to Special Trackwork, Track Geometry Irregularities	TD#6 - Curve Entry and Exit Dynamics	TD#9 - Buff/Draft Forces During Curving with Braking and Traction Forces	TD#9 - Buff/Draft Forces During Curving with Braking and Traction Forces	FD#1 - Response to Periodic and Random Track Inputs to Determine Spectral Content of Forces, Moments	
Structural Integrity	Component Wear, Fatigue and Fracture	Component Forces, Moments, Deflections	TD#4 - Forced Response to Special Trackwork, Track Geometry Irregularities TD#5 - Impact Response to Wheel and Rail Surface Anomalies	TD#7 - Curve Entry and Exit Dynamics	TD#9 - Buff/Draft Forces During Curving with Braking and Traction Forces	TD#10 - Wear Due to Repeated Flanging in Curves (Especially Tilt-Body Trains)	FD#1 - Response to Periodic and Random Track Inputs	FD#2 - EV to Identify Resonant Modes

Legend: W/R= Wheel/Rail
EV = Eigenvalue/Eigenvector Solution
QL = Quasilinearization
HSR= High-Speed Rail
TD = Time Domain Solution
SSC= Steady-State Curving
DF = Describing Function
DOF= Degree of Freedom
FD = Frequency Domain Solution
NL = Nonlinear
MBS= Multibody System

Table 4-2. General Maglev modeling requirements for the HSGGT analytical toolkit.

SAFETY-RELATED DYNAMIC PERFORMANCE ISSUE	SAFETY-CRITICAL EVENT	PRIMARY RESPONSE VARIABLES	DYNAMIC PERFORMANCE REGIME					
			Vehicle/Guideway Interaction	Dynamic Curving	Longitudinal Dynamics	Steady-State Curving	Frequency Response	Stability and Modal Response
Loss of Guidance	Vehicle Contact with Guideway	Limit Cycle Conditions and Self-Excited Oscillations	TD#1 - Transient Response to Guideway Inputs, Wind Loads					FD#2 - EV FD#3 - QL
		Vehicle and Gap Displacements; Vehicle/Guideway Forces & Moments	TD#1 - Forced Response to Special Guideway Features,Discrete Guideway Anomalies and Wind Loads	TD#2 - Curve Entry and Exit Dynamics	TD#3 - Vehicle Longitudinal Response During Acceleration/Deceleration		FD#1-Response to Periodic and Random Guideway Inputs	
	Vehicle Departure from Guideway	Vehicle Lateral/Roll Response	TD#1 - Forced Response to Special Guideway Features, Discrete Anomalies and Wind Loads	TD#2 - Curve Entry and Exit Dynamics	TD#3 - Vehicle Longitudinal Response During Acceleration/Deceleration	SSC#1 - NL Constant Radius Curving on Smooth Guideway		
	Guideway Failure	Vehicle/Guideway Forces; Guideway Moments and Displacments	TD#1a -Forced Response to Special Guideway Features, Discrete Anomalies and Wind Loads	TD#2a - Curve Entry and Exit Dynamics	TD#3 - Vehicle Longitudinal Response During Acceleration/Deceleration			
Ride Quality	Excessive Passenger Acceleration Levels	Car Body Acceleration Response	TD#1 - Forced Response to Special Guideway Features, Guideway Geometry Irregularities	TD#2 - Curve Entry and Exit Dynamics	TD#3 - Vehicle Longitudinal Response During Acceleration/Deceleration		FD#1 - Response to Periodic and Random Guideway Inputs to Determine Spectral Content of Forces, Moments	
Structural Integrity	Component Wear, Fatigue and Fracture	Component Forces, Moments, Deflections	TD#1a - Forced Response to Special Guideway Features, Guideway Geometry Irregularities, Wind Loading	TD#2a - Curve Entry and Exit Dynamics	TD#3 - Vehicle Longitudinal Response During During Curving	SSC#1 - NL Constant Radius Curving on Smooth Guideway	FD#1 - Response to Periodic and Random Guideway Inputs	FD#2 - EV to Identify Resonant Modes

Legend: TD = Time Domain Solution
EV = Eigenvalue/Eigenvector Solution
NL = Nonlinear
DF = Describing Function
FD = Frequency Domain Solution
SSC = Steady-State Curving
QL = Quasilinearization/Describing Function Technique

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5.0 Model Validation

A prerequisite for using a dynamic model of an HSGGT system to predict safety-related dynamic performance is that it has been validated successfully at some level. A model's validity, i.e., the extent to which it predicts the behavior of the physical system adequately, must be established and understood before the model can be used with confidence. In this section, we discuss the issues and considerations associated with model validation for HSGGT systems.

5.1 Validation Methods

A model can be validated at several levels. These are described in ascending order of quality.

- ***Qualitative Validation***. This is generally considered the lowest level of validation, at which the general qualitative behavior of the model is determined to be similar to that observed for the actual vehicle. Examples of this are lateral stability models that predict linear critical speeds in the range of those observed in service, and time-domain models that predict "rock and roll" resonances in response to bolted joint track at frequencies similar to observed values. One generally is restricted to this level of validation when no measurements or other, previously-validated models are available for comparison. This level of validation at best ensures the reasonableness of the model, but the level of confidence with which quantitative predictions may be used is quite low.

- ***Indirect Quantitative Validation***. A higher level of validation is comparison with another model that has been validated previously against experimental data (or perhaps against yet another previously-validated model). A feature of this approach is a high degree of control of the simulation conditions, which ensures an "apples to apples" comparison of predicted behavior. A disadvantage may be the potential for increasing inaccuracy of each subsequent model.

- ***Direct Quantitative Validation***. The highest level of model validation is achieved by successfully comparing model predictions against measurements of vehicle response. This is also the most difficult level of validation to achieve. Field tests are costly, time-consuming and difficult to control. Very few field tests have been run for the primary purpose of model validation; thus, in many cases an insufficient number of measurements are available for a comprehensive model validation. Further, difficulties in controlling the tests (e.g., vehicle speed, initial conditions into a test zone), measurement system limitations, and in some cases in adequately characterizing the vehicle/guideway system (e.g., uncertainties in the wheel/rail interface geometry, track stiffness) have resulted in many cases in inconclusive agreement between measurements and predictions.

5.2 Factors Affecting Successful Validation

Generally, the validity of a model is established when it has been demonstrated to predict adequately the behavior of the corresponding physical system. The criteria for the *adequacy* with which the model predicts behavior can vary widely, depending on many factors. These include:

- *The aspects of dynamic performance that the model will be used to evaluate (e.g., stability analyses, wheel climb derailment analysis, ride comfort studies).* This affects the selection of the primary response variables that form the basis for validation (e.g., lateral and vertical wheel/rail forces for derailment analyses, car body accelerations for ride comfort studies), as well as the requirements for model accuracy.

- *The range of vehicle and guideway configurations for which the model will be used.* The successful validation of a model for one vehicle/guideway configuration does not necessarily imply that the model is valid for other vehicle/guideway configurations. The degree to which the model could be used in evaluating other configurations depends on the similarities between configurations. A determination of the range of validity of the model must be based on an in-depth understanding of the dynamics of the physical systems with respect to how the model was developed.

- *The range of operating and loading conditions over which the model will be used.* The model should be validated over a sufficiently wide range of operating and loading conditions. There are practical limitations to the ranges over which tests can be conducted to provide validation data (e.g., it probably would be cost-prohibitive and unsafe to run tests that result in derailments in order to provide validation data for models that predict derailments). Again, an in-depth understanding of the dynamics of the physical system and the development of the model is required so that the range of validity of the model can be determined.

- *The accuracy and completeness with which the vehicle/guideway systems have been characterized.* In many cases, the ability to validate a model is limited because sufficient data on the vehicle/guideway dynamic characteristics are not available. Consequently, several model parameters must be estimated, and the accuracy of these estimates will affect strongly the agreement between model predictions and measurements. A potential pitfall that should be avoided in this situation is excessive and undefensible adjustment of one or more estimated model parameters to achieve good agreement with measured data. Ultimately, the final model parameter values must be substantiated to ensure that the model was not "validated" for an artificial vehicle configuration.

- *The degree of experimental control and quality of measurements made in field validation tests.* Good experimental control is critical to providing measurements with which a model can be validated. Lack of sufficient experimental control can be manifested in measurements that differ significantly from run to run under seemingly identical test conditions. It is also critical that the bandwidth and accuracy of the measurement system are consistent with the model validation requirements, and that transducers are installed in locations at which the model can predict their response.

A summary of general validation requirements for using HSGGT models are provided in Tables 5-1 and 5-2, for HSR and Maglev systems, respectively.

Model validation is a critical step in the development and maintenance of HSGGT models. Part of the effort on VA3204 is to plan and perform validation activities and to review and critique previous validation activities. Without this step, the credibility of an HSGGT model cannot be established.

Table 5-1. Validation requirements for HSGGT vehicles.

Aspect of Dynamic Performance	Key Comparison Variables	Operating Conditions
Lateral Stability	Wheelset, truck and car body relative displacements and accelerations	Response to guideway perturbations over the operating speed range.
Curving	Lateral and vertical wheel/rail forces; lateral & yaw displacements of truck and wheelsets; lateral, yaw and roll displacements	Curve entry and steady curving at speeds up to incipient wheel climb on range of curves expected in service.
Forced response to tangent track irregularities	Lateral and vertical wheel/rail forces; vertical and lateral accelerations of wheelsets, trucks and car body	Worst-case allowable track geometry and track stiffness; wood and concrete tie track; speeds up to maximum operating speed
Forced response to special trackwork	Lateral and vertical wheel/rail forces; vertical and lateral accelerations of wheelsets, trucks and car body	All special trackwork expected in service at speeds up to maximum operating speed (including tunnels and elevated guideways)

Table 5-2. Validation requirements for Maglev vehicles.

Aspect of Dynamic Performance	Key Comparison Variables	Operating Conditions
Lateral Stability	Relative displacements and accelerations of the car body and the primary suspension system	Response to guideway perturbations over the operating speed range.
Curving	Lateral and vertical primary suspension forces; lateral, pitch, yaw and roll displacements of the car body	Curve entry and steady curving at speeds up to guidance bottoming on range of curves expected in service.
Forced response to tangent track irregularities	Lateral and vertical primary suspension forces; vertical and lateral accelerations of the car body and the primary suspension system	Worst-case allowable track geometry and track stiffness for elevated and at-grade guideways; speeds up to maximum operating speed
Forced response to special guideway features such as turnouts	Lateral and vertical primary suspension forces; lateral and vertical accelerations of car body and primary suspension system	All special guideway features expected in service at speeds up to maximum operating speed (including tunnels and elevated guideways)

6.0 Review of Existing Models of HSGGT Systems

In this section a review of candidate existing models for an HSGGT analytical toolkit is presented. These include specialized simulation codes and commercial general-purpose dynamic simulation codes.

6.1 Background

Over the past 15 years, a number of complex, nonlinear, multi-degree-of-freedom computer models have been developed to study the dynamic response of rail vehicles. These range from simple wheelset equation solvers to comprehensive programs to study curving and stability of complete rail vehicles. Most of these models have been developed by deriving system equations, converting these equations to an applicable computer code, and solving the equations by time integration or matrix inversion techniques.

These equations were initially solved on analog computers. As digital computers replaced the analog computers in the early 1970's better methods for implementing the coding of the system dynamic equations were developed in the form of system simulation languages. These language extensions were developed to provide easier coding and to provide some of the modeling characteristics of the analog computer (ease of model development using block diagrams, for example)[6-1] * when using the digital computer. A example of this type of modeling aid still in use is the Advanced Continuous Simulation Language (ACSL) developed circa 1975. In recent years, some of these complex models have been formalized into multibody system (MBS) methods and software, which provide automated development of equations and computer models[6-2]. With these MBS codes, the user defines mass, inertia, and coordinate properties of the vehicle bodies, the interconnections between bodies (joints, force elements), model inputs (forces or geometries), and the desired model outputs.

A recent technical review paper by Sharp[6-3] provides an excellent overview of the strengths and weaknesses of multibody computer simulation. In this paper, he describes computer modeling in general as a four-part process: 1) an idealization of the system under study, 2) the mathematical description of this idealized system, 3) a more-or-less exact solution of the mathematical model, and 4) an interpretation of the idealized solution with recognition of the real (non-idealized) problem.

References appear at the end of this section.

The model-building problem has been simplified in the past few years. Computer power on the desk and well-developed numerical methods are now readily available. The problem, therefore, is not one of *principle,* but rather one of *detail and implementation.* Models potentially contain many small terms which contribute greatly to solution time, but little to accuracy. The *implementation details* determine the number of terms, cross terms, and effects that are included in any given model.. Computer model development falls into two basic categories:

- Manually prepared models
- Automatically derived models.

Manually prepared models tend to be focused and efficient since only the acknowledged major terms are carried along in the equation derivation process. These models rather naturally set standards for simulation run times, since there are deliberate procedures to reduce complexity during development without introducing unacceptable precision.

Automated model building falls into two basic types which are outlined below:

- Numerical multibody codes (DADS, ADAMS, MEDYNA, NUCARS, etc.)
- Symbolic codes (AUTOLEV, AUTOSIM, MESA-VERDE, NEWEUL, etc.).

Numerical Codes

- Formulation of equations and solution by numerical integration closely interconnected
- Highly developed, "as-is" modeling with fixed multi-body dynamics formalism
- Rapid development of a given simulation
- Tend to run extremely slowly
- Geared to display outputs as time histories or by animation.

Symbolic Codes

- Provide the equations themselves
- Difficult to obtain equations, but once obtained for a given system, need not be generated again
- Used the same way as manually prepared equations in numerical solution
- Geared to requirements of a range of "solvers", e.g., numerical integration, frequency response, eigenvalue/eigenvector, control system analysis, etc.
- Good choice of coordinates and reference frames
- "Development requires substantial dynamic skills, typically requires skills beyond the design engineer."

Model development with both automation schemes requires an expert dynamicist who knows the methods and limitations of dynamic systems, and more importantly, understands the particular system (rail vehicle, for example) in great detail. The development of a system model requires substantial effort and numerous iterations of equation development, implementation, and refinement before a dynamic model reaches useable status. Even after the completion of the model development effort the results from exercising any model will only be as good as the input data and the output data presentation and interpretation. Based on this discussion and the discussions in Section 4, the salient advantages and disadvantages of these modeling approaches are summarized in Table 6-1

6.2 Review of Existing HSR Models

Computer models of HSGGT vehicle and guideway dynamics have proliferated during recent years as digital computer capabilities have expanded and costs have dropped. Every academic institute, research organization, government agency, and railroad property who has dealt with vehicle dynamics problems seems to have one or more in-house computer codes. These range from simple "home-made" model to some rather complex commercially-available multi-body system (MBS) codes.

Table 6-1. Comparison of attributes of VDS codes.

Manually Derived Codes (DYNCUR, VEHDYN3, etc.)	Symbolic Codes (AUTOSIM, NEWEUL, etc.)	Multi-Body Numerical Codes (NUCARS, MEDYNA, etc.)
	--- Advantages ---	
▪ Variety of solution techniques possible ▪ Tend to focus on specific need (problem) ▪ Model changes can be made (source code is available) ▪ Model restrictions and limits are more likely to be known and recognized by the developer/user ▪ Usually more efficient model and code ▪ Less expensive, even than public domain codes	▪ Variety of solution techniques possible ▪ Gives the equations themselves (same as hand generated) ▪ More choice of coordinates, reference frames ▪ Moderate development time and effort ▪ Developer/user knows model restrictions and limitations ▪ Well-developed user's manual available	▪ Most rapid development of code for given model ▪ User does not need a complete knowledge of equations and solutions ▪ Output formats range from adequate to elegant (time histories, animation, etc.) ▪ Widely accepted by the railroad industry ▪ Well-developed user's manual available
	--- Disadvantages ---	
▪ Development time and effort is significant ▪ New model may require new program/code ▪ Output formats usually range from crude to modestly adequate ▪ User's manual usually crude or not available ▪ Support or help for users difficult to obtain	▪ No existing code geared to wheel-rail systems* ▪ Commercial codes with associated costs ▪ Support or help for users difficult to obtain	▪ Fixed multi-body dynamics formalism ▪ "As-is" modeling with "black box" elements ▪ Tend to run extremely slow (unless math "shortcuts" are taken) ▪ Commercial codes with associated costs ▪ Support or help for users varies with code from good to marginal

6.2.1 Available Commercial MBS Codes

Bechtel recently conducted an evaluation of Vehicle Dynamic Simulation (VDS) programs for the Korean High Speed Rail Construction Authority, who were interested in acquiring a VDS package to design high-speed turnouts. Although there are a number of packages available, the list was narrowed to the four most promising VDS programs, worldwide, with wheel-rail simulation capabilities. These programs have all been under development over a ten to twenty-year period, and have been exercised on a variety of problems. Each package contains input and output processors, calculation and graphical output modules. The packages reviewed are:

- **A'GEM** (Automatic Generation of Equations of Motion) developed by Queens University, Kingston, Ontario, Canada by Dr. R. J. Anderson. This package is a MBS solver with pre- and post processors that are interactive with *AutoCad*. The *AutoCad* interface is used to help develop the model, visualize its construction, and then present and/or visualize the output results.

- **MEDYNA** (MEhrkörper DYNAmik or Multi-Body Dynamics) developed by a German joint venture group headed by Dr. W. Kortüm including DLR, MAN, the Technical University of Berlin, and the German Ministry of Research and Technology.

- **NUCARS** (New and Untried Car Analytical Regime Simulation) developed by the Association of American Railroad's (AAR) Transportation Test Center at Pueblo, Colorado. This package is based upon work at British Rail Research Centre, Derby, U.K., carried out under Dr. Alan Wickens by Dr. John Elkins. Later contributions were made by Dr. Fred Blader, Peter Klauser, Stephen Handal, and Nicholas Wilson.

- **VAMPIRE** (Vehicle DynAmics Modeling Package In a Railway Environment) has been under development for 20 years by British Rail Research. Dr. Alan Wickens was instrumental in initiating this work by pioneering many of the analytical methods used. Wheel-rail simulation is based on work by A. O. Gilchrist, J. A. Elkins, R. J. Gostling, B. M. Eickoff, and others of British Rail Research.

Other automated rail vehicle dynamic simulation programs exist, but they lack availability and/or documentation in English at this time. These include AUTODYN/ROBOTRAN (Université Catholique de Louvain), SIDIVE (Construcciones y Auxiliar de Ferrocarriles, S.A.), and VOCO (INRETS-LTN). A recent update of this review has shown that the only major change is the addition of ADAMS/Rail to this list of MBS codes. This package has been released as Version 1.0 and is still undergoing implementation.

6.2.2 Comparison of Commercial Code Technical Features

The major technical features of these VSD packages are compared in Table 6-2 . A comparison of the overall features, including purchase and leasing costs of each simulation package, has been compiled in Table 6-3 . Each program is relatively user-friendly* in the input and control of the analysis activities, providing automatic equation generation, and static and graphical or animated display of a wide variety of output variables as detailed in Table 6-2. The technical comparison of programs highlighted the following:

A'GEM was developed as a general purpose rail and road vehicle simulation program[6-4,6-5]. It can generate virtually any type of vehicle, including non-conventional steering-axle bogies and control device feedback in any speed regime. It does not offer as many trackwork geometry options as, say, the NUCARS package, nor does it have coupled vehicle-guideway dynamics capabilities built in. It offers an AutoCad interface, which allows the user to more easily input data and plot output. It will handle all common tasks such as eigenvalue solution, wheel unloading calculations, nonlinear curve entry, and tangent track stability. User definable modules are permissible and source code is available.

MEDYNA is a comprehensive package originally developed to support high speed rail and Maglev projects in Germany[6-6,6-7,6-8]. It integrates all options for multi-body analysis into one package and can simulate fully nonlinear elements with bodies connected either open or closed loop. A large program, it is capable of handling large problems including vehicle-guideway interactions. However, the downside is that it must be run on a powerful work station or mainframe computer, which adds to the cost of operation. It is complex to implement and has little US user support.

NUCARS has grown out of a need to investigate vehicle-track interactions on U.S. freight railroads[6-9,6-10,6-11]. It is basically a time-integration type analyzer with a good selection of track geometries, wheelset profiles, and nonlinear interconnections (suspension elements). It has the capability for simulating any type of vehicle, even though applications have been concentrated on freight cars. It has been used to investigate safe speeds through both American and European designs of turnouts. To date, the track structure characteristics are limited to spring-dashpot elements. The package lacks the completeness of output and analysis capabilities of MEDYNA and VAMPIRE, but is a useful tool for investigating time-varying response of nonlinear vehicle systems.

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* The authors have hands-on experience only with NUCARS and cannot attest to the user-friendliness of other packages.

VAMPIRE is the product of 20 years development at British Rail Research and contains a varied array of tools for rail vehicle analysis[6-12,6-13]. Applications include British Rail high speed rail evaluation, and freight derailment problems. It offers an extensive combination of modeling and analysis features and will run on IBM-compatible PCs.

Table 6-2. Technical comparison of commercial MBS rail vehicle model codes.

ITEM	A'GEM	MEDYNA	NUCARS	VAMPIRE
Analysis Performed				
Static Analysis	√	√	√	√
Time Integration (Nonlinear Eqs.)	√	√	√	√
Steady State Response	√	√	√	√
Frequency Response	√	√	No	√
Stochastic Analysis	√	√	No	√
Eigenvalue/Eigenvector	√	√	No	√
Currently Available Model Elements				
6 DOF Rigid Bodies	√	√	√	√
Flexible Bodies	No	√	√	√
Translation/Rotation Constraints	√	√	No	√
Coordinates: Absolute or Relative	A,R	A	R	R
Coupling elements -				
▪ Series Spring/Damper	√	√	√	√
▪ Parallel Spring/Damper	√	√	√	√
▪ Hysteretic	No	√	√	No
▪ Coulomb Friction	√	√	√	√
▪ Friction Wedge	No	√	√	√
▪ Pinned Link	√	√	No	√
▪ Closed Loop/User Eqs.	√	√	No	√
▪ Bushing, 3-D	√	√	√	√
Wheelset Models				
Rigid Wheelset	√	√	√	√
Independent Wheelset	√	√	√	√
Torsional Coupling	No	√	√	No
Asymmetric Wheel Profile Geometry	√	√	√	√

Table 6-2. (Continued) Technical comparison of commercial MBS rail vehicle model codes.

ITEM	A'GEM	MEDYNA	NUCARS	VAMPIRE
Wheel/Rail Contact Model				
Two-Point (Tread and Flange)	√	√	√	√
Flangeback Contact	No	No	√	Unknown
2D Geometry + 3D Approximation	No	Unknown	√	√
Full Nonlinear Geometry	√	√	√	√
Wheel-Rail Connection	No	√	√	√
Track Model				
Parallel Spring/Damper	No	√	√	√
Point Mass	No	Unknown	No	No
Continuous Beam	No	Unknown	No	No
Flexible Guideway Beam Effects	No	√	No	No
Creep Force Models				
Kalker Coeff. Lookup Table	√	No	√	No
Kalker's Nonlinear Theory	√	√	√	√
Linear Law	√	√	No	√
Square Root Law	No	No	No	√
Variable Tread and Flange Friction	√	√	√	√
Excitation Sources				
Forces/Moments on Any Body	√	√	√	√
Displacements from Measurement or Analysis	√	√	√	√
Track Spirals, Curves, Superelev., Kinks	√	√	√	√
Track Geometry Irregularities	√	√	√	√
Track Geometry Spatial PSDs	√	√	No	√
Wheel-Rail Contact Geom. Variation	No	No	√	No

Table 6-3. Overall features of selected vehicle dynamic simulation models.

ITEM	A'GEM	MEDYNA	NUCARS	VAMPIRE
Supplier	RJA Eng. Analysis	DLR (Germany)	AAR - TTC	B.R. Research
Applications				
■ Road	Yes	Yes	No	No
■ Rail	Spiral curves entry, tangents, ride, load transfer	Curves, twisty track, tangents	Measur. track inputs, long/short wavelength geom.	Curves, twisty track, tangents, Measur. Inputs
Hardware Required				
■ Hard Disk	120 MB	30 MB	> 30 MB	N.A.
■ Platform	486 PC Super VGA	Workstation or 486 PC	286-486 PC Apollo, VAX IBM-RISC HP7400 Plot	DEC VAX, SUN Workstation 386 PC, HPGL Plotter
Program				
■ RAM	8 MB	8 MB	640 kB	4 MB
■ Size	N.A.	3 MB	1 MB	15 MB
■ Op. System	DOS	UNIX, AEGIS, MVS, VM,PRIMOS, DOS	DOS, UNIX	VMS, SPAR, DOS
Price - $US (Including installation and training)				
■ Buy	41 k 9/94 $17.5 k w/o AutoCad	106 k	38 k 9/94 $15k	Not an Option
■ Lease	Not an Option	110 k / 3 yr	43 k / 3 yr	54 k / 3 yr
Comments	Interactive uses of AutoCAD interfaces for input for animation outputs	Batch or interactive modes, Library of coupling elements. Flexible guideway/car interactions predicted	Batch and interactivemode processing. Used in wide variety of car simulations and for special trackwork studies. Roots in common with Vampire.	Batch or inter. modes, libs of example input files incl. gen. vehicle types, elastic/rigid bod., libs of gen. coupling elements, 2 wheelset models, Nonlin. contact gemetry
	Japan, India, Canada, US (M.K)	Germ: 24, UK: 3, Fra: 2, Swit: 3, Aus: 2, Neth: 3, Can: 1, China: 1, Italy: 2, Belg: 1, Czech: 1, Finl: 3	Approx 50-80 Users	UK: 7, Europe: 6, Far East: 3, Man ufacturers and Operations

6.2.3 Cost Comparison of Commercial MBS Codes

The cost of these four commercial multi-body simulation codes is given in Table 6-3. These costs reflect mid-1993 prices in U.S. dollars with mid-94 updates for AGEM and NUCARS. Costs of packages have been compared for two options, purchasing and leasing. Not all packages offer both options. For example, VAMPIRE cannot be bought, and A'GEM cannot be leased.

In the case of MEDYNA, the cost of a work station has been added, while for A'GEM the cost of additional software (AutoCad) was added in the 1993 prices. It is shown in Table 6-3 that MEDYNA is the most expensive to buy, with A'GEM and NUCARS about equal and less than half the cost of MEDYNA. MEDYNA is also the most the most expensive to lease with VAMPIRE second and NUCARS third. Price updates in mid-1994 indicate that the prices of these packages have dropped over the last year. The software price market is constantly changing and latest known prices are shown in the Table 6-3.

6.2.4 Non-commercial ("Specialized") VSD Codes

Most of the modeling and simulation tools developed over the past 25 years can be categorized as specialized codes, i.e., those developed for predicting the dynamic response of a specific class of vehicle/guideway configurations (e.g., rail passenger vehicles, freight cars with three-piece trucks), and to address a specific issue (e.g., an observed hunting problem, a derailment). These codes typically have been developed by government and university researchers, government contractors and railroad organizations. Many of these codes have been used and maintained primarily (and sometimes solely) by the developers; thus, in many cases, the documentation and "user-friendliness" of the codes is limited to that necessary for the developers to use them. Consequently, a significant challenge in including specialized codes is limited to that necessary for the developers to use them. Consequently, a significant challenge in including specialized codes in the HSGGT analytical toolkit is to provide sufficient validation, documentation maintenance of the codes so that they can be used by others. The Battelle VA3204 project team has been prolific in developing a wide range and Maglev modeling and simulation tools. A list of over 50 of these codes is provided in Tables 6-4 and 6-5. The brief descriptions contained in the tables demonstrate that the capabilities of these specialized codes cover many of the modeling requirements defined in Section 4 of this report. This collection of codes represent a solid foundation upon which an analytical toolkit can be established.

Table 6-4. Selected HSGGT modeling tools developed by Battelle.

Program	Rev	DOF	Emphasis	Description
CURVIC	3-93	7	Rail Vehicle	Nonlinear steady-state curving of 2-axle truck (bogie) to establish curving initial conditions (see RAILCORR).
GWMGLV2X, GWMGLV3, GWMGLV6, GWMGLV7	4-92	24, 60, 32, 60	Mag Lev	Vertical modes of EDS- or EMS-type Maglev vehicle on elevated guideway (2 cars, 2 or continuous spans). Includes car body and span bending modes. (Time integration, modal summation method)
GWMGLV4, GWMGLV8	3-92	24, 32	Mag Lev	Lateral modes of EDS and EMS-type maglev vehicle on elevated guideway (2 cars, 2 spans). Incl. torsional, lateral bending modes. (Time integration, modal summation method)
IMPWHLQ	3-93	25	Rail Veh. and Track	Impact of railroad wheel on rail due to wheel flats or rail surface defects. Includes axle and tie transverse bending modes. (Time integration, modal summation method)
INTRMDLV, INTRMDLH	7-92	17, 24	Rail and Maglev Vehicles	Vertical, lateral modes of EMS-type maglev vehicle riding on railroad flatcar. Track geometry PSD input; accel. PSD, ride quality indices as outputs. (Frequency domain)
MCLLAT, MCLVERT	3-84	21, 14	Rail Transit	Nonlinear model of UTDC intermediate capacity transit car with steering truck. Lateral, vertical modes. Inputs: track geometry, cubic spiral and curve, turnouts. Outputs: loads, accelerations. (Time integration, modal summation method)
PERTRK	2-84	17	Rail Vehicle	Nonlinear model of rail vehicle with two 2-axle trucks. Inputs: track line and cross level errors. Outputs: accelerations and loads on vehicle and track. (Time integration, modal summation method)
RAILBRK	6-88	na	Elevated Guideway	Calculation of fastener loads and rail break gap side under thermal tensile loads (CWR), with effects of column and bearing stiffnesses, restraint from other rail(s), multiple spans. (iterative solution)
RAILCORR	3-93	29	Rail Vehicle, Track	Nonlinear model of curving wheelset, rail, and track components to investigate rail corrugation phenomenon. Uses CURVIC as subroutine for initial conditions. (Time integration, modal summation method)
SEPTACAR	3-92	19	Rail Transit Vehicle	Model of transit car with detailed truck, gearbox and traction motor. Inputs: rail line and cross level geometry, gearbox mounting eccentricity ("wobble"). (Time integration, modal summation method) Outputs: loads, accelerations.
SEPTAVRT	12-91	14	Rail Transit Vehicle	Model of transit car with detailed truck, gearbox and traction motor. Inputs: rail surface geometry (joints, etc.). Outputs: motor support bolt and other component loads, accelerations. (Time integration, modal summation method)
SSCRV2	2-93	7	Rail Vehicle	Nonlinear SS curving of 2-axle truck, freight or passenger. Inputs: speed, curve; Outputs: wheel loads, car and truck comp. motions.
SYS21CRn	8-91	11	Transit Vehicle	Nonlinear simulation of "System 21" transit vehicle on tangent track, AREA spiral and curve. Suspension hydropneumatic units. (Time integration, modal summation method)
SYS21OUT	11-89	5	Transit Vehicle	Simulation of "System 21" pivoted car body (lateral, yaw), two simplified outrigger trucks. Inputs: standard AREA spiral and curve, discrete rail geometries. Outputs: loads and accelerations. (Time integration, modal summation method)
TRKTHRM1	6-88	na	Elevated Guideway	Calculation of fastener loads under thermal expansion or contraction load of girders and rails, considering effects of column and bearing stiffnesses. (interative solution)
TRKVHNT	6-91	19	Rail Vehicle	Eigenvalue/eigenvector routine for determining stability of rail vehicle car body and trucks on smooth tangent track. Frequency, damping and oscillatory modes.
TRKVPSDV, TRKVPSDH	6-91	13, 15	Rail Vehicle	Generalized rail vehicle with cargo (trailer, etc.) in vertical, lateral response to track surface geometry PSD inputs. Output: accel. PSDs, 1/3rd octave, octave band, ride quality indices.(Frequency Domain)
METROBRK	8-88	6	Articulated Transit Bus	Nonlinear simulation of two-car articulated transit bus traction motor, driveline, axle/wheels, and tire characteristics to investigate wheel hop problem during braking. (Time integration, modal summation method)

Table 6-5. Selected HSGGT modeling tools developed by VA3204 subcontracts.

Program	Type*	Description
		Clemson University
NLWRGCON	CF	Uses actual profiles to develop asymmetric wheel-rail geometry and characteristics necessary for dynamic analyses.
LCRPC	CF	Computes lateral, longitudinal, lateral-spin, and spin creep coefficients using Kalker's linear theory.
NLCRPFS, NLCRPFE	CF,I	Calculates nonlinear creep forces based on Kalker's simplified and exact theories, respectively.
9CFC	LS	Conventional freight car with 2 standard 3-piece trucks. 9 DOF (lateral, yaw, and roll warp)
17AFT, 19FC	LS	Freight car with advanced freight trucks. Trucks have primary suspension elements and wheelset interconnections. 17 DOFs: body lateral, yaw, and roll displacements; truck (x 2) lateral, yaw, warp displacements; wheelset (x 4) lateral and yaw displacements. 19FBC has additional car body bending and torsion model.
23FFCIW	LS	23 DOF Flexible body freight car with independently rotating wheels. Similar to 19FBC but with 4 additional DOFs representing wheelset torsional flexibility.
17PC	LS	Passenger car model. 2 trucks having primary suspension elements. DOFs: car lateral, yaw, and roll displacements; truck frame (x 2) lateral, yaw, and roll displacements; wheelset (x 4) lateral and yaw displacements.
21LOCO6X	LS	6 axle locomotive. 21 DOFs: car lateral, yaw, and roll displacements; truck (x 2) lateral, yaw, and roll displacements; wheelset (x 6) lateral and yaw displacements.
21FC3X	LS	Freight car with 3-axle freight trucks. 21 DOFs: car lateral, yaw, and roll displacements; truck (x 2) lateral, yaw, and warp displacements; wheelset (x 6) lateral and yaw displacements.
24ART	LS	Articulated intermodal vehicle with 5 platforms and 6 conventional freight trucks. 24 DOFs: lateral displacement of 6 car body bolster centerplate stubs; truck (x 6) lateral, yaw, and warp displacements.
27ART	LS	Articulated 2-platform vehicle with 3 advanced freight trucks having primary suspensions and wheelset interconnections. 27 DOFs: car (x 2) lateral, yaw, and roll displacements; truck frame (x 3) lateral, yaw, and warp displacements; wheelset (x 6) lateral and yaw displacements.
9PSDFC	FD	Similar model as 9CFC. Inputs are track lateral alignment and cross level. Outputs are PSDs of the lateral response variables (displacements, accelerations, etc.) and mean values of the wear indices and energy losses in the suspension and across the wheel-rail interfaces.
17PSDPC	FD	Similar model as 17PC. The input is track lateral alignment. Outputs are PSDs of lateral response variables (displacements, accelerations, etc.).
11PSDPC	FD	Passenger car model. The input is track vertical alignment. 11 DOFs: car body heave pitch, and first bending; truck (x 2) heave and pitch; wheelset (x 4) heave. Outputs are PSDs of the displacements, accelerations, etc.
9QLFC	QL, FD	Similar freight car model as 9PSDFC, with nonlinear wheel-rail geometry and Coulomb friction in the suspension. Nonlinearities are represented by describing functions and equations are solved iteratively. Outputs are PSDs and rms values of accelerations, displacements, etc.
5NLHFC	TI	Half-freihgt car model. 5 DOFs: car body lateral; and truck lateral, yaw, and warp. Nonlinear wheel-rail geometry creep force saturation, coulomb friction at the centerplate. Inputs are from random irregularities in lateral centerline.
9NLCE	TI	Nonlinear curve entry. 9 DOFs: lateral and roll of car body; truck frame lateral, yaw, and roll, wheelset (x 2) lateral and yaw. Nonlinear wheel-rail geometry, creep law, and suspension. Outputs are time histories and peak values of response variables including wheel-rail forces. Tangent, spiral, and constant radii curve inputs.

Table 6-5. (Continued) Selected HSGGT modeling tools developed by VA3204 subcontracts.

Program	Type*	Description
27NLCE	TI	Nonlinear curve entry of locomotive. 27 DOFs: car body lateral, yaw, and roll displacements, truck frame (x 2) lateral, yaw, and roll displacements; wheelset (x 6) lateral and yaw displacements and spin velocity. Nonlinear wheel-rail geometry, creep law, and suspension. Outputs are time histories and peak values of response variables including wheel-rail forces.
NLSCLOCO	SSC	Nonlinear steady state curving of 6 axle locomotive. 11 DOFs: truck frame lateral and yaw displacements; wheelset (x 3) lateral and yaw displacements and spin velocity. Kalker's nonlinear creep law, nonlinear wheel-rail geometry, weight shifts due to traction and cant deficiency, single or two point contact on any wheel, nonlinear primary stiffnesses, different coefficients of adhesion for flange and tread. Traction or braking torque inputs. Outputs include vehicle displacements, wheel-rail forces, power losses, and wear indices.
NLSC4X	SSC	Nonlinear steady state curving of 4 axle rail (passenger of freight) vehicle. 8 DOFs: truck frame lateral and yaw displacements and warp if freight truck); wheelset (x 2) lateral and yaw displacements and spin velocity. Nonlinear creep law, nonlinear wheel-rail geometry, weight shift due to cant deficiency, single or 2 point contact, nonlinear primary stiffnesses. Traction or braking torque inputs. Outputs include vehicle displacements, wheel-rail forces, power losses, and wear indices.
33ROCK	TI	Nonlinear rocking of freight car with 2, 3 axle trucks. 33 DOFs: car heave, pitch, and roll displacements; truck bolster (x 2) lateral, heave, roll displacements; sideframe (x 4) vertical displacement of middle, front, rear of "hinged", 2 piece unit; wheelset (x 6) vertical and roll. Nonlinear suspension, wheel lift, rail flexibility, centerplate lift-off. Input is time history of cross level. Outputs include car response variables, wheel-rail forces.
26ROCK	TI	Nonlinear car rocking a 2-platform articulated car. 26 DOFs: 3 vertical displacements of front, middle, and rear of "hinged" platforms, roll of each platform; wheelsets (x 6) heave and roll displacements; bolsters (x 3) heave, lateral, and roll displacements. Nonlinear suspension, wheel lift, rail flexibility, centerplate lift-off. Input is time history of cross level. Outputs include car response variables, wheel-rail forces.
		CMRI
DYCURV	TI	Dynamic curving of rail passenger vehcile with 2-axle trucks. Conventional and steered trucks. Nonlinear suspension wheel/rail interaction, including 2-point contact.
SSCURV	SSC	Steady-state curving of ral vehicle with radial freight tracks. Two-point contact (flanging) of lead outer wheel. Nonlinear suspension and wheel/rail interaction. Lateral rail flexibility.

* Solution Type:
LS — Lateral Stability; CF — Closed-form; I — Iterative; FD — Frequency Domain; QL Quasi-linear; Ti — Time Integraton; SSC — Steady-state curving; MS — Modal summation

6.3 Review of Existing Maglev Models

During the 1970s, there was a substantial effort in modeling both the vehicle-elevated guideway dynamic interactions and magnetically levitated (Maglev) vehicles[6-14,6-15]. The program described in Reference 17 (called MAGDYN) was developed by the University of Toronto and is probably typical of the "hand-developed" programs of this time period.

With reduced Federal research funding, interest in Maglev vehicles waned in the United States in the 1980s. Germany and Japan, however, actively pursued both the analytical and experimental work on demonstration Maglev systems. In Germany, the MEDYNA multibody computer program was used to investigate a variety of Maglev vehicle dynamics cases[6-16].

The recent burst of research activity in the United States has been fueled by the National Maglev Initiative (NMI) program. A number of individual research projects were pursued under the NMI in which many aspects of Maglev systems were examined. A number of Maglev vehicle and vehicle-guideway models resulted from these projects[6-17,6-18,6-19,6-20]. For the most part, these models were simplistic in approach to the vehicle and guideway dynamics, concentrating instead on other aspects of the problem, such as magnet force and gap control, and the aerodynamic effects on the vehicle. Again, model equations are hand-developed (the authors in Reference 20 used a software package called TUTSIM). In Reference 21, the model was developed as a matrix Lyapunov equation.

As a part of one NMI project[6-21], Battelle developed a more complex time-domain simulation model of a moving two-car Maglev train passing over a series of stationary elevated guideway beams. The vehicle portion of the model included rigid and first-bending body modes, magnet pod frames, and magnet pods, with versions for both electromagnetic (EMS) and electrodynamic (EDS) configurations. The guideway beams were simulated as free-free beams, using the mode summation method, which allowed finite compliance of connections (support pads) at the columns. Magnet-guideway interaction forces were calculated from relative motions due to both rigid-body and bending modes, using spatially-changing modal influence coefficients. By "daisy-chaining", a continuous series of spans was simulated in a model of up to 52 degrees of freedom. Bounce-pitch and roll-yaw-lateral models of vehicle and guideway were programmed separately to limit the number of DOFs.

Somewhat similar vehicle-guideway models have been developed recently by Argonne National Laboratory[6-22] and by the Japanese Railway Technical research Institute (RTRI)[6-23]. Equations and computer codes for these three models were developed "by hand" rather than with the aid of automated MBS computer codes.

In another NMI project[6-24], Battelle developed a frequency-domain model of a rail flatcar with a single Maglev vehicle riding piggyback, using stochastic track geometry representations (PSDs) as inputs to the railcar. This model uses matrix inversion techniques to calculate acceleration PSD response at different locations on the vehicles. From these responses, ride quality indices were calculated, including the NASA, ISO 2631, W_z, and Peplar indices.

A number of other "in-house" models of Maglev vehicles of varying degrees of complexity exist, including models developed by the VNTSC. Battelle and CMRI are currently in final development of a nonlinear time-domain model of the Grumman EMS Maglev vehicle configuration. With canted magnet pairs, this model required a full 6 DOF approach to account for both cross coupling and large angle motions. This model incorporates vehicle dynamics, guideway dynamics, and controller characteristics in a closed-loop system, with "noise" inputs from guideway irregularities or from external forces such as wind gusts.

6.4 Available Modeling Resources

In the previous sections a number of different available models have been presented. These models make up a currently available capability for modeling HSGGT vehicles for safety related evaluations and analysis. The available resources contain a commercial MBS code as well as other, special-purpose codes for addressing particular needs such as track structural response and Maglev controller analysis.

Specific model requirements (degrees-of-freedom, model features, solution methods, etc.) are provided for HSR and Maglev systems in Tables 6-6 and 6-7, respectively, along with existing codes that meet some or all of the requirements. These codes are considered the most appropriate starting point for developing and maintaining the toolkit.

Table 6-6. Recommended HSR models for first-generation HSGGT analytical toolkit.

		Salient Degrees of Freedom$^+$ and Features$^\prime$				Candidate Modeling and Simulation Tools			
Code Type/General Capability (from Table 4-1)	Applications	Car Body	Truck	Wheelsets	Track	Specialized	Commerical/ MBS	Commercial/ Symbolic	Model Development Requirements
TD#1 - Transient Response to Track Inputs, Wind Loads	Loss of Guidance Vehicle Derailment Critical Speeds and Limit Cycle Oscillations	y,ϕ,ψ C1,C3 Q2	y,ϕ,ψ S1,S2 F1,F2 P1	y,ψ W1,W2,W3, W4,W5	T5	VEHDYN3 DYNCUR	NUCARS VAMPIRE MEDYNA A'GEM	AUTOSIM MAPLE	Wheel climb mechanisms, derailment criteria
TD#1a - Forced Response to Special Trackwork and Discrete Anomalies and Wind Loads	Loss of Guidance Vehicle Derailment W/R Forces and Motions	y,z,Θ,ϕ,ψ C1,C3	y,ϕ,ψ S1,S2 F1,F2 B1,B2 P1	y,z,ϕ,ψ W1,W2,W3, W4,W5	y,z,ϕ T1,T2,T5 G1,G2,G3	VEHDYN3 DYNCUR	NUCARS VAMPIRE MEDYNA A'GEM	AUTOSIM MAPLE	Wheel climb mechanisms, derailment criteria
TD#2 - Forced Response to Special Trackwork, Track Misalignnments, Track Irregularities, Wind Loads	Loss of Guidance Track-Based Derailment W/R Forces, Rail Displacements	y,z,Θ,ϕ,ψ C1,C3	y,z,Θ,ϕ,ψ S1,S2 F1,F2 B1,B2 P1	y,z,Θ,ϕ,ψ W1,W2,W3, W4,W5,W6, W7	y,z,ϕ,ψ T1,T2,T5 G1,G2,G3	VTRKDYN VEHDYN3	NUCARS VAMPIRE MEDYNA A'GEM	AUTOSIM MAPLE	Nonlinear rail force/deflection parameters
TD#3 - Forced Response to Special Trackwork, Track Geometry Irregularities, Wind Loads	Ride Quality Excessive Passenger Acceleration & Vibration Levels Car Body Acceleration Response	y,z,Θ,ϕ,ψ C1,C2,C3,C4 Q1	y,z,Θ,ϕ,ψ S1,S2 F1,F2 B1 P1	y,z,ϕ,ψ W1,W2,W3, W4,W5	y,z,ϕ T1,T2,T3, T4,T5 G1,G2	VEHDYNV VEHDYN3	NUCARS VAMPIRE MEDYNA A'GEM	AUTOSIM MAPLE	Ride quality indices for transient response
TD#4 - Forced Response to Special Trackwork, Track Geometry Irregularities, Wind Loads	Structural Integrity Component Wear, Fatigue and Fracture Component Forces, Moments, Deflections	y,z,Θ,ϕ,ψ C1,C2,C3,C4	y,z,Θ,ϕ,ψ S1,S2 F1,F2 B1,B2 P1	y,z,ϕ,ψ W1,W2,W3, W4,W5,W6	y,z,ϕ T1,T2,T3, T4,T5 G1,G2,G3	SEPTACAR MCLLAT MCLVRT	NUCARS VAMPIRE MEDYNA A'GEM	AUTOSIM MAPLE	Wear, fatigue, fracture criteria and models Pantograph dynamics models
TD#5 - Impact Response to Wheel and Rail Surface Anomalies	Structural Integrity Component Wear, Fatigue and Fracture Component Forces, Moments, Deflections	z,ϕ	z,Θ,ϕ F1 P1	z,ϕ W2,W6	z,ϕ T1,T2,T3, T5,G1,G2	IMPWHLQ	VAMPIRE	AUTOSIM MAPLE	Rail as Timoshenko beam on discrete supports

Table 6-6. (Continued) Recommended HSR models for first-generation HSGGT analytical toolkit.

		Salient Degrees of Freedom+ and Features#				Candidate Modeling and Simulation Tools			
Code Type/General Capability (from Table 4-1)	Applications	Car Body	Truck	Wheelsets	Track	Specialized	Commerical/ MBS	Commercial/ Symbolic	Model Development Requirements
TD#6 - Curve Entry and Exit Dynamics	Loss of Guidance Vehicle Derailment W/R Forces Ride Quality Excessive Passenger Acceleration & Vibration Levels Car Body Acceleration Response, Ride Quality Indices	y,ϕ,ψ C1,C4 Q1	y,ϕ,ψ S1,S2 F1,F2 P1	y,ϕ,ψ W1,W2,W3, W4,W5	y,ϕ T1,T2,G1, G2,	VEHDYN DYNCUR	NUCARS VAMPIRE MEDYNA A'GEM	AUTOSIM MAPLE	Wheel climb mechanisms,derailment criteria
TD#7 - Curve Entry and Exit Dynamics	Loss of Guidance Guideway Failure W/R Forces and Rail Displacements Structural Integrity Component Wear, Fatigue and Fracture Component Forces, Moments, Deflections	y,z,ϕ,ψ C1,C3,C4	y,z,ϕ,ψ S1,S2 F1,F2 P1	y,z,ϕ,ψ W1,W2,W3, W4,W5	y,z,ϕ T1,T2 G1,G2,G3	VEHDYN3 DYNCUR	NUCARS VAMPIRE MEDYNA A'GEM	AUTOSIM MAPLE	Nonlinear rail force/deflection characteristics Pantograph dynamics models
TD#8 - Longitudinal Dynamic Response During Braking/Traction in Curves	Loss of Guidance Vehicle Derailment W/R Forces, Buff/Draft Forces	x,y,θ,ϕ,ψ C1,C4	x,y,θ,ϕ,ψ S1,S2 F1,F2 B1,B2 P1	x,y,ψ W1,W5			NUCARS VAMPIRE MEDYNA A'GEM	AUTOSIM MAPLE	Realistic 3-D draft gear representation

Table 6-6. (Continued) Recommended HSR models for first-generation HSGGT analytical toolkit.

		Salient Degrees of Freedom+ and Features'				Candidate Modeling and Simulation Tools			
Code Type/General Capability (from Table 4-1)	Applications	Car Body	Truck	Wheelsets	Track	Specialized	Commerical/ MBS	Commercial/ Symbolic	Model Development Requirements
TD#9 - Longitudinal Dynamic Response During Braking/Traction in Curves	Loss of Guidance Guideway Failure W/R Forces, Buff/Draft Forces Ride Quality Excessive Passenger Acceleration & Vibration Levels Car Body Acceleration Response, Ride Quality Indices Structural Integrity Component Wear, Fatigue and Fracture Component Forces, Moments, Deflections	x,y,z,Θ,ϕ,ψ C1,C4 Q1	x,y,z,Θ,ϕ,ψ S1,S2 F1,F2 B1,B2 P1	y,z,ϕ,ψ W1,W2,W3, W4,W5	y,z,ϕ T1,T2 G1,G2		NUCARS VAMPIRE MEDYNA A'GEM	AUTOSIM MAPLE	Realistic 3-D draft gear representation
TD#10 - Wear Due to Repeated Flanging in Curves	Structural Integrity Component Wear, Fatigue and Fracture Component Forces, Moments, Deflections	y,z,ϕ,ψ C1,C3,C4	y,z,ϕ,ψ S1,S2 F1,F2 P1	y,z,ϕ,ψ W1,W2,W3, W4,W5,W8	y,z,ϕ T1,T2 G1,G2	VEHDYN3 DYNCUR	NUCARS VAMPIRE MEDYNA A'GEM	AUTOSIM MAPLE	Nonlinear rail force/deflection characteristics
SSC#1- NL Constant Radius Curving on Smooth Track	Loss of Guidance Vehicle Derailment W/R Forces	y,ϕ,ψ C1,C3	y,ϕ,ψ F1	y,ψ W1,W2,W3, W5	T1 G1	VEHDYN3 DYNCUR SSCURV	NUCARS VAMPIRE MEDYNA A'GEM	AUTOSIM MAPLE	
SSC#2- NL Constant Radius Curving on Smooth, Compliant Track	Loss of Guidance Guideway Failure W/R Forces, Rail Displacements	y,ϕ,ψ C1,C3	y,ϕ,ψ F1	y,ψ W1,W2,W3, W5	y,z,ϕ T1,T2 G1,G2	VEHDYN3 DYNCUR SSCURV	NUCARS VAMPIRE MEDYNA A'GEM	AUTOSIM MAPLE	

Table 6-6. (Continued) Recommended HSR models for first-generation HSGGT analytical toolkit.

Code Type/General Capability (from Table 4-1)	Applications	Salient Degrees of Freedom⁺ and Features′: Car Body	Truck	Wheelsets	Track	Candidate Modeling and Simulation Tools: Specialized	Commerical/ MBS	Commercial/ Symbolic	Model Development Requirements
FD#1-Response to Periodic and Random Track Input, and Rotating Unbalance Forces from Wheels Traction Motors, etc.	Loss of Control Vehicle Derailment W/R Forces and Motions Ride Quality Excessive Passenger Acceleration & Vibration Levels Car Body Acceleration Response, Ride Quality Indices Structural Integrity/Component Wear, Fatigue and Fracture/Component Forces, Moments, Deflections	y,z,θ,ϕ,ψ C1,C2,C3,C4 Q1,Q2	y,z,ϕ,ψ S2 F1,F2	y,z,ϕ,ψ W1,W4,W5	y,z,ϕ T1,T2,T4 G1,G2	TRKVPSDV TRKVPSDH ARTFQ 17PSDPC	VAMPIRE MEDYNA A'GEM	AUTOSIM MAPLE	
FD#2 - EV	Loss of Guidance Vehicle Derailment Critical Speed Prediction Structural Integrity Component Wear, Fatigue and Fracture Component Forces, Moments, Deflections, and Resonant Modes	y,ϕ,ψ C1 Q2	y,ϕ,ψ F1,F2	y,ψ W1,W4,W5		TRKVHNT 21LOCO6X ARTSTAB 9QLFC	VAMPIRE MEDYNA A'GEM	AUTOSIM MAPLE	
FD#3 - QL	Loss of Guidance Vehicle Derailment Limit Cycle Conditions	y,z,ϕ,ψ C1 Q2	y,ϕ,ψ F1,F2	y,ψ W1,W4,W5, W7		TRKVHNT 21LOCO6X ARTSTAB 9QLFC	VAMPIRE MEDYNA A'GEM	AUTOSIM MAPLE	Limit Cycle Indices

Table 6-6. (Continued) Recommended HSR models for first-generation HSGGT analytical toolkit.

* x=longitudinal, y=lateral, z=vertical, Θ=pitch, ϕ=roll, ψ=yaw

 # Special HSR Modeling Capabilities

•	C1	Multiple Car Bodies/Articulated Trains		C2	Car Body Flexibility
	C3	Aerodynamic Forces on Car Body	•	C4	Tilt Control System Dynamics
	S1	Secondary Suspension Nonlinearities	•	S2	Active Suspension Control
	F1	Truck Frame Flexibility (Lozenging, Racking, etc.)		F2	Motor/Drivetrain Dynamics
	P1	Primary Suspension Nonlinearities			
•	W1	Wheelset Interconnection and Forced-Steering Linkages		W2	Nonlinear Wheel/Rail Profile Geometry
	W3	Nonlinear Creep Laws with Flanging		W4	Torsionally Flexible Wheelsets
•	W5	Independently-Rotating Wheels		W6	Wheelset Bending Flexibility
	W7	Quasilinear W/R Interaction (Describing Functions)		W8	Wheel/Rail Wear Models
	T1	Track Lateral/Vertical Flexibility		T2	Track Dynamics (Mass/Inertia Effects)
	T3	Tie Flexural Modes		T4	Random Track Geometry Characteristics
	T5	Discrete Track Geometry Characteristics			
	G1	Guideway Flexibility		G2	Guideway Dynamics (Mass/Inertia Effects)
	G3	Pantograph/Catenary Dynamics			
	B1	Brake System Dynamics	•	B2	Anti-Skid Control System Dynamics
	Q1	Ride Quality Criteria		Q2	PSDs and/or Modal Response Calculations from Time-Histories

•Specific vehicle design feature

Table 6-7. (Continued) Recommended Maglev models for first-generation HSGGT analytical toolkit.

		Salient Degrees of Freedom[+] and Features[']			Candidate Modeling and Simulation Tools			
Code Type/General Capability (from Table 4-2)	Application	Car Body	Bogie	Guideway	Specialized	Commerical/ MBS	Commercial/ Symbolic	Model Development Requirements
TD#2 - Curve Entry and Exit Dynamics	Loss of Guidance Vehicle Contact with Guideway Vehicle and Gap Displacements; Vehicle-Guideway Forces and Moments Loss of Guidance Vehicle Departure from Guideway; Vehicle Lateral-Roll Response Ride Quality Excessive Passenger Acceleration & Vibration Levels Car Body Acceleration Response	y,ϕ,ψ C1,C4 Q1	y,ϕ,ψ S1,S2 F1,F2 P1	y,ϕ T1,T2,G1, G2		MEDYNA A'GEM DADS	AUTOSIM MAPLE	Large Motions Robust Controller; Nonlinear Magnet Models
TD#2a- Curve Entry and Exit Dynamics	Loss of Guidance Guideway Failure Vehicle-Guideway Forces; Guideway Moments and Displacements Structural Integrity Component Wear, Fatigue and Fracture Component Forces, Moments, Deflections	y,z,Θ,ϕ,ψ C1,C2,C3,C4	y,z,Θ,ϕ,ψ S1,S2 F1,F2 B1 P1	y,z,ϕ T1,T2 G1,G2,G3		MEDYNA A'GEM DADS	AUTOSIM MAPLE	Large Motions Robust Controller; Nonlinear Magnet Models

Table 6-7. (Continued) Recommended Maglev models for first-generation HSGGT analytical toolkit.

		Salient Degrees of Freedom[+] and Features[']			Candidate Modeling and Simulation Tools			
Code Type/General Capability (from Table 4-2)	**Application**	**Car Body**	**Bogie**	**Guideway**	**Specialized**	**Commerical/ MBS**	**Commercial/ Symbolic**	**Model Development Requirements**
TD#3 - Vehicle Longitudinal Response to Traction/Braking Forces Forces During Curving	Loss of Guidance Vehicle Contact with Guideway Vehicle and Gap Displacements; Vehicle-Guideway Forces and Moments Loss of Guidance Vehicle Departure from Guideway; Vehicle Lateral-Roll Response Loss of Guidance Guideway Failure Vehicle-Guideway Forces; Guideway Moments and Displacements Ride Quality Excessive Passenger Acceleration & Vibration Levels Car Body Acceleration Response Structural Integrity Component Wear, Fatigue and Fracture Component Forces, Moments, Deflections	x,y,z,Θ,ϕ,ψ C1,C4 Q1	x,y,z,Θ,ϕ,ψ C1,C4 B1 P1	$x,y,z,\Theta,\phi,$ ψ S1,S2 F1,F2		MEDYNA A'GEM DADS	AUTOSIM MAPLE	Robust Controller; Large Motions
SSC#1 - NL Constant Radius Curving on Smooth Guideway	Loss of Guidance Vehicle Departure from Guideway; Vehicle Lateral-Roll Response Structural Integrity Component Wear, Fatigue and Fracture Component Forces, Moments, Deflections	y,ϕ,ψ C1,C3	y,ϕ,ψ F1,F2	T1 G1		MEDYNA A'GEM DADS	AUTOSIM MAPLE	Large Motions

Table 6-7. (Continued) Recommended Maglev models for first-generation HSGGT analytical toolkit.

Code Type/General Capability (from Table 4-2)	Application	Salient Degrees of Freedom⁺ and Features#			Candidate Modeling and Simulation Tools			Model Development Requirements
		Car Body	Bogie	Guideway	Specialized	Commerical/ MBS	Commercial/ Symbolic	
FD#1 - Response to Periodic and Random Guideway Inputs	Loss of Guidance Vehicle Contact with Guideway Vehicle and Gap Displacements; Vehicle-Guideway Forces and Moments Ride Quality Excessive Passenger Acceleration & Vibration Levels Car Body Acceleration Response Structural Integrity Component Wear, Fatigue and Fracture/Component Forces, Moments, Deflections	y,z,Θ,ϕ,ψ C1,C2,C3,C4 Q1	y,z,ϕ,ψ F1,F2 B1 P1	y,z,ϕ,ψ T1,T2,T3, T4,T5 G1,G2,G3	GWMLMODL@	MEDYNA A'GEM	AUTOSIM MAPLE	
FD#2 - EV	Loss of Guidance Vehicle Contact with Guideway Stability Boundaries Structural Integrity Component Wear, Fatigue and Fracture/Component Forces, Moments, Deflections	y,ϕ,ψ C1	y,ϕ,ψ F1,F2			MEDYNA A'GEM	AUTOSIM MAPLE	

⁺x=longitudinal, y=lateral, z=vertical, Θ=pitch, ϕ=roll, ψ=yaw

 # Special Maglev Modeling Capabilities

C1	Multiple Car Bodies	C2	Car Body Flexibility
C3	Aerodynamic Forces on Car Body	C4	Tilt Control System Dynamics
S1	Secondary Suspension Nonlinearities	S2	Active Suspension Control
F1	Bogie Flexibility	F2	Propulsion System Dynamics
P1	Primary Suspension Nonlinearities		
B1	Brake System Dynamics		
G1	Guideway Flexibility	G2	Guideway Dynamics (Mass/Inertia Effects)
G3	Random Guideway Geometry Characteristics	G4	Discrete Guideway Geometry Characteristics
Q1	Ride Quality Criteria	Q2	PSDs and/or Modal Response Calculations from Time-Histories
@	Under development in VA3204		

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7.0 Logistical Considerations for Modeling

There are a number of logistical issues that must be addressed when modeling HSR and Maglev systems. These issues often tend to get neglected as mundane, but the success or failure of a modeling endeavor often can be traced to the implementation and concern with the logistical issues. These issues will be described in this section along with approaches to their resolution.

7.1 Essential Elements of HSGGT Modeling

The essential elements of a modeling effort are presented in Figure 7-1. These as basically the same as those denoted by Sharp[6-3] in section 6-1. The four major steps to modeling are:

Step 1. Define the vehicle, component, or system to be modeled and define the modeling goals

Step 2. Prepare a model by idealizing the vehicle, component, or system and creating a mathematical description of the vehicle, component, or system

Step 3. Execute a mathematical algorithm, or series of algorithms to "solve" or exercise the mathematical description of the vehicle, component, or system

Step 4. Interpret the results of the algorithmic execution to relate them to the physical vehicle, component, or system.

Each of these steps involves a number of logistical issues that must be addressed to successfully develop, and use a model. The issues associated with each step will be discussed in the following sections.

7.1.1 Definition of the Object (Vehicle or System) to be Modeled and Modeling Goals

The most important step in modeling is to determine the resultant goal for the analysis. This initial goal setting will serve as the guideline to determine the complexity and effort necessary to accomplish the goal for the model. The determination of the performance issue to be addressed will dictate the model to be used, if an existing model is available to address the goal. Tables 6-6 and 6-7 define the general goals (applications) that may be addressed by the range of currently available models. If a model does not already exist then the definition of the goals will serve to help construct a new model to accomplish the goals.

The definition of the object to be modeled includes the definition of the specific parameters needed to describe the object mathematically. These parameters may be obtained in a number of different ways, and take a number of different forms. The simplest parameters are the basic physical parameters necessary to describe the object to be modeled. There is a continuous set of increasingly difficult parameters to obtain as one moves from modeling existing hardware such as the TGV to the modeling of concepts and vague designs as is needed when attempting to model Maglev systems. An abbreviated list of commonly needed parameters and a method to get them follows to illustrate the issue:

•	Dimensions	Obtain from Drawings, Estimate or scale from pictures Physically measure actual hardware
•	Masses	Obtain from engineering specifications Estimate from drawings and pictures, Obtain from physical measurement and estimation
•	Configuration Definition (Interconnections, etc)	Define from Drawings, Define from Pictures, Define from Observance of Actual Hardware Define based on Engineering Judgement
•	Spring Rates and Damping Values	Define from Engineering Specifications Define from Contact with Vendors, based upon Specifications or Hardware Observance Measurement or Test
•	Etc	Etc
•	Etc	Etc
•	Etc	Etc
•	Lift Magnet Controller Gains	Define from Engineering Specifications Make Engineering Estimates Develop Values from Design Calculations

In most cases the parameters of interest needed to model an object must be obtained, in order of preference, from the manufacturer or their documentation, from direct measurement of observance of the system, or from engineering estimates. The typical scenario is a combination of the three. These parameter determination efforts may be very time consuming and directly affect the validity of the modeling results. Experiences of the VA3204 team with obtaining good engineering values for exercising model evaluations have only served to reinforce the difficulty of meeting this objective. Published parameters used for various models runs with NUCARS early in the program were found to be in error when a detailed parameter validation was attempted. This validation was initiated as a result of poor validation of the model output. This illustrates the iterative nature of many modeling efforts.

The best way to obtain good parameters for modeling is through a combination of close cooperation with the system manufacturer and direct physical measurement of the system of interest. Unfortunately both efforts can be very time consuming. An additional problem is that very often the manufacturer does not have the information needed for accurate modeling. This is very true for older equipment unless it has been modeled at a previous date. For example, a simple parameter such as the mass moment of inertia of a truck frame with accessories is not typically calculated. An assembly drawing along with component drawings is needed in order to calculate an estimate for this parameter. Even if the manufacturer has the value they may consider it proprietary or business sensitive and limit access to the data.

There is better methodology for calculating these physical parameters that is beginning to be available. Many of the properties associated with the definition of a model are readily available if the design and assembly drawings for the item to be modeled have been created in a CAD system. The ability to directly calculate parameters from simple distances to complex 3D moments of inertia is resident in many of the more powerful CAD packages. Fortunately the rail industry is rapidly converting over to CAD. The ability to gain access to such CAD files on a vehicle or system to be modeled points again to the close cooperation that is needed between modeler and manufacturer.

7.1.2 Model Preparation

The second step of the modeling process is to create the mathematical representation of the vehicle, component, or system to be modeled. This abstraction process must be based upon an accurate description of the system as developed in step 1. These two steps are very closely related and interactive. An accurate understanding of the system configuration must be obtained before an accurate representation in the form of a model can be constructed. After the initial physical descriptions of the system are obtained the system must then be described in equation form. The derivation of the system of equations that actually constitute the model may be done in one of three methods:

- Manual derivation of the equations of motion and manual programming of original, usually specialized computer programs (most often written in FORTRAN, but not always) that focus on more-or-less specific analytical problems and solutions. Numerous assumptions are usually made to make the problem manageable, typically including small angle assumptions.

- Automatic derivation of the equations of motion through the use of multi-body numerical platform codes such as MEDYNA, NUCARS, ADAMS/Rail, and VAMPIRE. The codes provide a dynamic model based upon using the system description developed in step 1 without having to manually derive equations of motion. The system equations derived in this manner tend to get very complicated and somewhat convoluted. Multi-body numerical methods are usually not restricted to small angle assumptions, but rail models usually make this assumption to reduce complexity and decrease solution time. Due to the multibody formulations and problem size the systems of equations typically are expansive and require considerable computational capability.

- Automatic derivation of the equations of motion through the use symbolic platform codes, among which the most applicable is AUTOSIM. The codes provide a dynamic model based upon using the system description developed in step 1 without having to fully manually derive equations of motion. These codes develop symbolic models for compilation as MATLAB or FORTRAN codes that may be further modified or manipulated. These codes represent an approach half-way between the previous two methods.

At this point detailed mathematical descriptions of the system under study have been created. The third step of the process is to actually exercise the model. Very many different levels of complexity and fidelity may be implemented with similar results at this stage of modeling. A very detailed model that includes a number of terms and effects that have no bearing on the problem will provide the same results as a simpler model which includes only the terms of interest. The real skill comes in separating the two.

7.1.3 Solution of the Model

Once the mathematical representation of the system is constructed it is necessary to "solve" the equations. The form of the solver will typically fall into one of three different types of operations. The solution will be performed using:

- time domain integration techniques
- frequency domain matrix solutions
- matrix solvers and iterative techniques.

It is not unusual for several of these techniques to be combined or multiple analysis to be performed using each method.

Time domain techniques have the advantage of being the most versatile because a wide variety of non-linearities may be implemented in the model. Matrix solvers and iterative techniques may be imbedded into each time step of a solution for such items as detailed calculations of the wheel/rail interface forces. The major disadvantage of time integration methods is the long runs necessary to implement this type of solution. If the full nonlinear effects of the wheel/rail interface are to be implemented there is no other alternative. Frequency domain analysis may be accomplished by performing FFT's on the output time histories generated by a time domain model.

Frequency domain techniques are best used when performing analysis such as ride quality or when searching for various system frequency sensitivities. Linearized forms of the equations are used to complete the needed analysis. Some inaccuracies may be encountered due to the linearization but these are more than offset by the short run times and computational efficiencies.

Matrix techniques and iterative solutions have specialized uses such as determining a steady state system configuration during curve negotiation as well as problem sub-sets such as the wheel/rail interface interactions.

Each of these solution techniques may have multiple implementations and configurations. One major issue faced by all is the numerical stability of the problem and its formulation. The systems of equations developed to represent typical HSR systems are some of the harder systems to solve in a stable fashion due to their wide frequency content. This marginal stability of the numerical formulations leads to an issue of robustness of the model for all formulations. A model may provide a solution for one case but prove numerically unstable for a slightly different case. This presents the modeler with a case of logistical considerations such as choosing the integration technique. Several methods are considered valid with differing pros and cons. In many cases the method chosen is the result of what is available within the software package used as the "solver".

Another set of logistical issues surround the "solver" with regard to the input and output of the systems of equations. It is seldom a straight forward exercise to get the system equations into a form that may be "solved". This is one of the driving forces for the MBS packages such as NUCARS or A'GEM where the development of the system equations and their solutions are performed with the same software codes. The ability to interface with other codes is also important due to the need to present and analyze the results from any model.

7.1.4 Output Analysis and Presentation

The solution of a particular problem typically leads to the generation of a large output data set for further analysis. In the current day of the desktop PC the analysis and presentation of data is usually performed using other packages at the desktop. It is a matter of logistics as to how this transfer of data from the "solver" to the output analysis and data presentation stage of the process occurs. The solutions in place range from completely integrated output such as NUCARS to a simple tabular data file that is the usual output format from a custom hand derived model. The logistical issue to be addressed is usually one of standardized data formats. The main concern when evaluating the needs for modeling output is to determine the types of data output and presentation required. Typical formats in increasing complexity are:

- Binary output files
- ASCII or DOS output files
- Graphical output formats and data files
- WINDOWS compatible formats and files
- Animation output with simple representations
- Animation output with pictorial type representations

Currently available modeling packages offer each of these levels of output. Unfortunately, most packages offer only a partial subset of these output types. A'GEM has excellent data and graphical outputs since it uses AutoCad as its graphics engine, relying on the inherent capabilities of that package to provide the graphics capability. With the proper interfaces A'GEM and AutoCad can provide output from simple tables to full animation. While NUCARS has an integrated graphing package it is severely limited in its capabilities. Hardcopy outputs are also available but limited due to the imbedded nature of the graphing subsystem and minimal support and driver updates.

7.2 General Considerations for Modeling and Simulation

Many factors affect the selection, implementation, and maintenance of modeling tools. A balance must be reached between the various items that affect the choices of modeling implementation. ***A successful, useful model is the blend of engineering and computer sciences for a particular set of hardware and software platforms that is constrained by user and cost limitations.*** This balance is illustrated in Figure 7-2. Engineering and computer science overlap in the science of numerical methods while the world of hardware and software meet in the selection of an operating system. The heart of a modeling effort and the overall goal is driven by the engineering needs in most cases. The engineering must be performed in a comprehensive and appropriate manner for the problem at hand. The computer sciences produce software to provide user interfaces in both text and graphical formats, again in a manner appropriate for the need at hand. Hardware platforms are chosen by the engineer mainly on the basis of speed, accuracy, and robustness of operation. These qualities of the model are the result of the combination of numerical methods and an operating systems ability to provide robust error recovery from limiting mathematical operations. Choices in these central four issues, two scientific and two facility related, are limited in their selections by logistical elements of cost, user training, and available technical support. Only when all elements are joined into a functional model is success realized. To ignore any element and make poor selections will doom a model to failure or marginal acceptance.

The computer industry is changing so fast that special considerations must be applied to computer modeling. It is necessary to continually reconsider the current positioning of a product or computational tool to prevent obsolescence. Try and find a card reader to execute an old, but valid model or a on-line IMSL library to implement a subroutine call. Models must be continually updated. Three out of the four elements of modeling are subject to this rapid change. The engineering element has been relatively con stant over the same time period and incurs change at a much slower rate. The computer industry is very different from the rail industry that has been relatively static for decades. Discussion on the major elements follow with approaches for addressing the concerns.

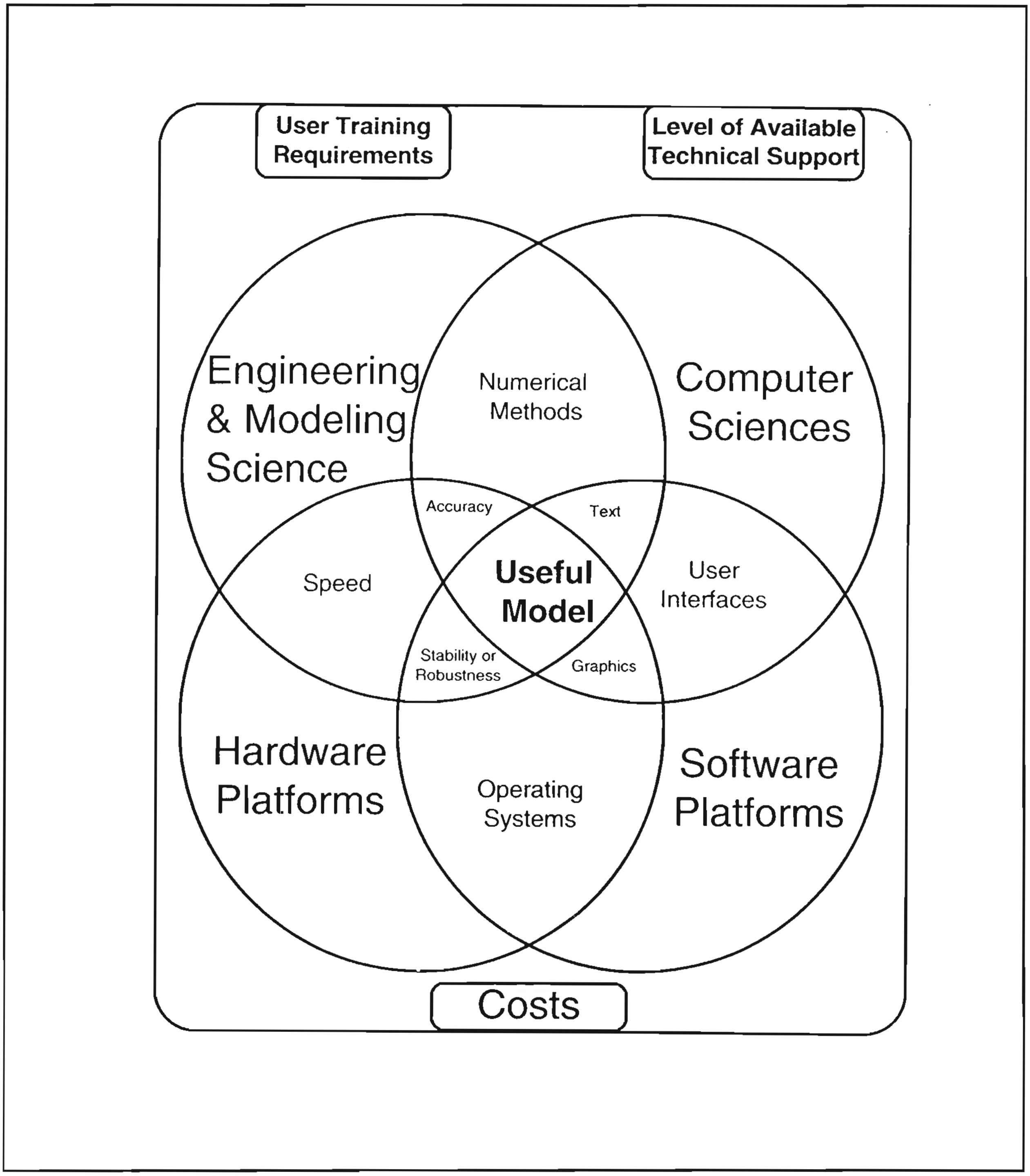

Figure 7-2 Logistical Elements of the Modeling Process

7.2.1 Hardware Platform Considerations

There are three major hardware platforms for consideration to chose between:

- Mainframes
- Engineering Workstations
- Desktop PC Systems

Each of these platforms must be evaluated in terms of its speed, cost, availability, accuracy, peripherals, and ability to solve the problem under consideration.

The usefulness of the mainframe computer as a modeling platform is all but past for the typical modeling effort. They may be used for accounting and servers in large corporations but otherwise they are not found in engineering use today. The exceptions are supercomputers that are used for specialized engineering and scientific investigations. These systems may be available through shared resources and national labs but they tend to be difficult and expensive to use. None-the-less numerous modeling codes were written for mainframe use and many codes still have remnants of structure from running on the mainframes. These systems range from slow to very fast and run operating systems of all types. Unix or a similar derivative is the OS preference for the newer supercomputers while Fortran is still the preferred engineering language.

Engineering workstations have capabilities that exceed the those of mainframe systems from five to ten years ago. These systems are considerably less costly than the mainframes but are still an order on magnitude more costly than typical desktop systems ($5k to $150k). These systems typically have graphical user interfaces with extensive graphics and numerical capabilities. Unix or a derivative is the OS of choice. These systems are typically networked but may also operate standalone. Software costs are much higher than desktop systems with a comparative limited availability of software from which to chose. These systems feature full 32 or 64 bit processing capability with multitasking capability. Many of the larger modeling codes such as ADAMS and DADS require a workstation to run effectively. Computational times are typically measured in minutes or tens of minutes for most problems. For engineering uses the workstations have reached a point where they are cost effective for most activities. Multi processor systems are beginning to be available that greatly speed up the solution of many numerically or graphically intensive applications.

The VA3404 team uses SparcStations as their engineering workstation of choice. These systems are well supported and most large commercial codes are available for them. These systems provide good interconnectivity for engineering uses across the team and also across the engineering community. These machines provide for limited animation and graphics presentation. Like most Unix based workstations they do require a knowledgeable system administrator to set them up and configure the systems to obtain optimum efficiency. Levels of user friendly interfaces and graphical operating systems are being introduced to make the use of these system more widespread.

The IBM compatible computer has become the most widely used machine for basic engineering work. The capabilities of the current machines rival those of workstations just a few years old and of mainframes from a decade ago. They very often are still running in a 16 bit mode of operation due to software limitations, but this will be changing with the next generation of OS. The interconnectivity of PC systems is improving steadily but problems still occur due to a lack of standardization. The engineering system of choice is the IBM compatible but several serious and very effective applications for engineering use run on Macintosh systems. One of the biggest issues when running on a PC type platform is the issue of support. Another issue is that many applications are running on a PC type platform because they are relatively inexpensive. Often the capability is mis-applied for this reason. It is not unusual for engineering problems to be set up that run into hours of computational time. These runs might be made on a engineering workstation in one tenth the time but at a cost that is five times higher. The overall purpose of a code and level of usage should be considered when selecting a platform for implementation. It may be more cost effective to run an application on a workstation just to assure timely results. The VA 3204 team uses the latest array of Intel based machines to run the current suite of available models.

7.2.2 Software Platform Considerations

The issues surrounding selection of software standards for modeling are similar to the hardware issues. Software platforms must be evaluated in terms of speed, cost, availability, accuracy, supported devices and platforms, and ability to solve the problem under consideration. Additionally it becoming very important to chose a software standard and then adhere to the standard. The choice of a software platform and a hardware platform are linked through the issue of standards. Most systems are becoming more standardized to improve efficiency and portability of codes and interfaces. The major issue with software still seems to be cost and functionality. Software is very expensive to write and maintain. Engineering workstation codes are particularly expensive because the OS are not yet fully standardized and various "ports" must be made to move one code from one platform to another. The low volume of engineering applications for workstations also helps keep the cost of ownership high. Several of the MBS codes considered for VA3204 run only on a workstation. Usually this is warranted, NUCARS, which was designed as a streamlined code benefits from the speed available on a workstation. Typical problem solution times run hours. These long solution times become cost prohibitive due to the wasted staff time and calender time that is associated with such run times.

There is an undefined issue of the suitability of government funded model code development being run on a workstation that requires substantial yearly software licences and updates. If codes that are used on the program must be available to the government the allowable cost of such codes must be defined. Fortran and UNIX are still the main scientific applications language and OS of choice for the workstation environment. A large number of older, but useful codes can still be run on these newer platforms with little code conversion.

The software limitations of the earlier DOS based desktop systems are about to be overcome with the introduction of the new Windows operating system in late 1995. This update should provide the general desktop user with the robustness and operating system features of Unix and the ease of use of a single user system. This enhancement will overcome the existing 16 bit limitations of DOS and move to a visual system interface. Unfortunately this move is also liable to obsolete much software written in the last ten years for the desktop.

The basic paradigm for user interfaces is currently moving from the text based interfaces of past years to a visual, menu based interface with full on-screen prompts and help. Older codes need to begin moving toward this newer paradigm to improve their usability. Considerable effort will need to be expended to convert existing codes from the older interfaces to new visual based, prompted, object oriented programming methods and portable codes. This conversion will also help with stretching the lives of these older codes as the embedded calls and procedures from past software is removed and the code is modularized.

7.3 Future Directions

The direction of modeling is determined mainly by the state-of-the-arts in computer science. The pace of change in the computer sciences is expected to continue at the high rate in the near future. For many areas of the technology it will seem like the rate of change will be increasing as each segment of computer capability catches up with ever increasing changes and advancements. This will make the coming years most challenging to many segments of the user community.

The differentiation between workstations and desktop systems is disappearing in terms of cost, computing speed, and operating interfaces. In fact most workstations are desktop units today, albeit expensive ones. The price of a given computational capability is continually dropping. The result of this drop is an ability to tackle more complex problems and solve them in a more expeditious fashion. While the hardware costs are dropping, SYSTEM costs are moving much more slowly due to increased problem complexity, increased software and system complexity, and maintenance/support requirements. The move toward standardized, supported platforms for both hardware and software is well underway with many shakeouts occurring in the industry. This shakeout and the resulting consolidation of computing alternatives will help manage the changes that will be occurring. Inter-package data exchange is becoming a more critical capability for model development in the future as programs are used to calculated inputs and analyze outputs from other programs. Such interfaces as a 3D CAD to Dynamic Model interfaces are beginning to play more important roles in the development of models. This trend is certain to continue.

8.0 Conclusions

This examination of critical issues, HSR and Maglev systems, model requirements, validation methods, modeling approaches, and modeling logistics has led to the following conclusions and recommendations.

In conclusion, it is the primary purpose of this report to recommend an approach to modeling HSGGT vehicles. In order to do this we must address the primary needs and reasons for these models. Models in the context of Task VA 3204 are focused on safety issues and must be able to predict:

- Operating safety (running stability and loss of guidance in its several modes),
- Ride comfort (both stochastic and transient), and,
- System (vehicle-guideway) integrity and maintenance.

Under operating safety, wheel climb derailment has to be considered in modeling capabilities, although "flange climbing very seldom occurs at high speed without a serious prior fault in vehicle or track or both" (B. M. Eickoff)[1]. Current models predict "derailment" in different ways: for example, "wheel lift", lateral relative displacement of wheel to rail beyond the profile look-up table (e.g., NUCARS, DYNCUR), or loss of tread contact for more than a specified period of time ΔT (e.g., HSRDYN). The adequacy of these model indicators is a topic of worthy of additional investigation as there is no universally accepted standard criteria for predicting derailment. The validity of the various models have has been explored in the Report, Tech Memo 2: Wheel/rail Interaction Model Comparisons.

Other modes of derailment such as track panel shift require that accurate *track* forces are predicted by a given model. The accuracy of track forces is a function of the frequency bandwidth of interest. If frequency content beyond perhaps 20 Hz is not of interest, a track modeled by simple parallel spring-damper element may be quite sufficient. Most of the current commercially available railway-oriented MBS codes take this approach. There is increasing interest, however, in modeling the vehicle and track as a unified system so that the safety-related and integrity-related events above 20 Hz can be modeled[2].

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1 Eickoff, B. M. "Vehicle Track Interaction Issues for Hifgh Speed Conventional Railways", Paper No. A4-2-(3), The International Conference on Speedup technology for Railway and Maglev Vehicles, Vol 2, Nov 1993, Yokohama, Japan.

2 The Fourth International Conference on Contact Mechanics and Wear of Wheel/Rail Systems at the University of British Columbia, Vancouver, B.C. (July 1994) and the IAVSD Herbertov Workshop in the Czech Republic (September 1994) will have a number of papers addressing vehicle-track interaction modeling.

Events in frequencies from 20 to perhaps 1500 Hz need to be investigated primarily in the context of track damage. None of the commercial MBS codes can at this time handle these problems.

Ride comfort is perhaps best addressed by frequency-domain modeling using stochastic inputs. Filtered results can be translated into accepted ride comfort indices such as the NASA and ISO parameters. Ride comfort by transient response analysis has relatively little analytical or experimental basis at this time. Therefore, any recommended commercial code would need this capability as well as the ability to perform eigenvalue/eigenvector analysis to check for running stability. Stability may also be checked by time-domain models, but perhaps not as efficiently.

Therefore, Battelle has recommended a "Toolkit" approach for the baseline establishment of a HSGGT modeling "facility". The baseline toolkit should contain the validated codes resident within the VA3204 team as denoted in Tables 6-4, 6-5, 6-6, and 6-7. It should also include the most widely used North American commercial Rail MBS code, NUCARS, as well as other, special-purpose codes for addressing particular needs such as track structural response and Maglev controller analysis as required by the FRA. We do, however, recommend that commercial codes, NUCARS in particular, *be used with caution* until the internal algorithms (connection force elements and, particularly, the wheel-rail model) are defined explicitly in a technical users manual and are validated. Recent validation efforts have cast doubts on the fidelity of this model. Other commercial codes do not currently contain the level of modeling details deemed necessary to model the safety directed processes of this contract.

Another use for vehicle and system models is in system design studies (suspension design, magnet controllers, etc.). This role is typically needed by manufactures and operators of the systems and is therefore outside the scope of this program. It should be noted that at this time there is little effort to bring together the modeling efforts of the system developers and the models developed for the FRA. In fact the opposite is often true as the regulatory agencies charged with evaluating safety are at odds with the users of design type models. Better distribution and use of the various models has the potential to standardize the modeling used during the design of a vehicle or system with the model used for safety evaluations. This is the role that has been set out for the New and Untried Car Analytic Regime Simulation, NUCARS, by the AAR. As its name implies this model is designed for evaluation of new designs, primarily in the freight industry. This model is relatively focused and therefore has a defined set of users with defined needs.

8.1 Conclusions

The modeling/analysis of HSGGT systems is a complex process that requires a high level of expertise and credible modeling techniques. There are no shortcuts in this process. The modeling of the complete vehicle/rail system is still one of the more complex simulation problems tackled to date. The formation of an HSGGT Modeling and Simulation Center consisting of a team of experts with a "toolkit" of available vehicle dynamics models under this VNTSC Contract is a valid approach to meeting the modeling and analysis needs and requirements.

We have shown in the review of HSGGT systems, Section 3, a wide variety of truck designs, suspension systems, and car bodies (tilt versus non-tilt, for example) are currently used in high-speed rail (HSR) applications. Variety also exists in track structures, ranging from wood crossties on ballast to direct-fixation track on slab structure. An HSGGT Modeling and Simulation Center must be prepared to support the following analysis and simulation activities for this wide range of vehicles and track:

- Vehicle/guideway dynamic response studies for both the typical installed U.S. track environment and for newer track structures
- Parameter studies of vehicle sensitivity to the US operating environment
- Quick-response forensic engineering to support accident investigations
- Vehicle/System certification and validation studies.

A similar variety of Maglev vehicle/guideway combinations have been identified, falling into two categories: the electromagnetic system (EMS), which is based on magnetic attraction between vehicle and guideway; and the electrodynamic system (EDS), which is based on eddy current-based magnetic repulsion. With Maglev designs, the vehicle and guideway are even more intimately coupled as a total system and must be analyzed as such.

The modeling requirements discussion of Section 4 clearly indicates that time domain, frequency domain, and eigenvalue/eigenvector solutions of models of varying complexities will be required to support both HSR and Maglev system evaluations. Each of these solution techniques provides unique strengths and efficiencies that can be used to advantage in a dynamic analyses.

The models and the modeling approach must be tailored to the problem at hand. Models used for HSR vehicle response are much different from the models used for Maglev vehicle evaluations. There are three general modeling approaches that are used in current practice:

- Original, usually specialized programs (most often written in FORTRAN, but not always) that focus on more-or-less specific analytical problems and solutions
- Multi-body numerical platform codes such as ADAMS/Rail, MEDYNA, NUCARS, and VAMPIRE. The codes provide a dynamic model without having to manually derive equations of motion.
- Symbolic platform codes, such as AUTOSIM (which has been used primarily for automotive vehicle modeling). These codes develop symbolic models for

compilation as MATLAB or FORTRAN codes that may be further modified or manipulated.

No single computer program currently provides an adequate modeling platform for all aspects of either HSR or Maglev systems. The multi-use "all purpose" multi-body codes reviewed in Section 6 have many strengths, but all also have some weaknesses, ranging from limited solution capabilities to size (computer system requirements) and cost. Most of the multi-body codes have certain "black box" aspects that make a detailed evaluation or validation of the code difficult. Several currently available modeling platform codes can prove useful for certain applications and should be included in an analysis "toolkit". The specialized codes already developed by the team represent a baseline set of effective tools that meets many of the modeling requirements for HSGGT systems.

There are a number of logistical issues that must be addressed when modeling HSR and Maglev systems. These issues often tend to get neglected as mundane, but the success or failure of a modeling endeavor often can be traced to the implementation and concern with the logistical issues. Many factors affect the selection, implementation, and maintenance of modeling tools. A balance must be reached between the various items that affect the choices of modeling implementation. ***A successful, useful model is the blend of engineering and computer sciences for a particular set of hardware and software platforms that is constrained by user and cost limitations***. Choices in these central four issues, two scientific and two facility related, are limited in their selections by logistical elements of cost, user training, and available technical support. Only when all elements are joined into a functional model is success realized. To ignore any element and make poor selections will doom a model to failure or marginal acceptance.

The state of HSR and Maglev system modeling in North America reflects the lack of emphasis placed on our HSR rail systems. Many of the existing analysis codes are old and were written for previous generations of computers and operating systems. Many of these codes are in need of rewrites and conversion to modern programming methods and I/O methodologies. The rapidly ongoing conversion of the computer industry to visual GUI (Graphical User Interface) based operating systems will be placing added emphasis on this conversion in the coming years. There are several commercial rail modeling packages including NUCARS, A'GEM, and, and ADAMS/Rail. NUCARS is in need of a major rework due to the limitations imposed during its original derivation. In its current configuration it is also a "bad" computing corporate citizen. It must be run stand-alone due to hardware interactions that prevent its being run without excessive computer reboot and resets. A'GEM is not widely accepted and needs added fidelity of the wheel/rail interface before it can be considered for the safety related analysis required by the Volpe Center. ADAMS/Rail has promise as a complete modern implementation of a MBS type code. Unfortunately this package is in its early stages of commercial introduction. This package was developed in conjunction with the Dutch Railways and the European Partner of ADAMS (Mechanical Dynamics). Therefore most of the technical expertise for this package resides in Germany, making use and support difficult.

The necessary components for modeling HSGGT systems are presented and summarized in the diagram of Figure 8-1. The figure shows the esential components and their relationship to both the users of the HSGGT Models and Simulations and the manufacturers of rail equipment that might be considered for evaluation. The various components are interrelated and synergistic. If the dynamic modeling of HSGGT (and HSGGT in general) is to be successful, an integrated cooperative approach to the effort should be undertaken.

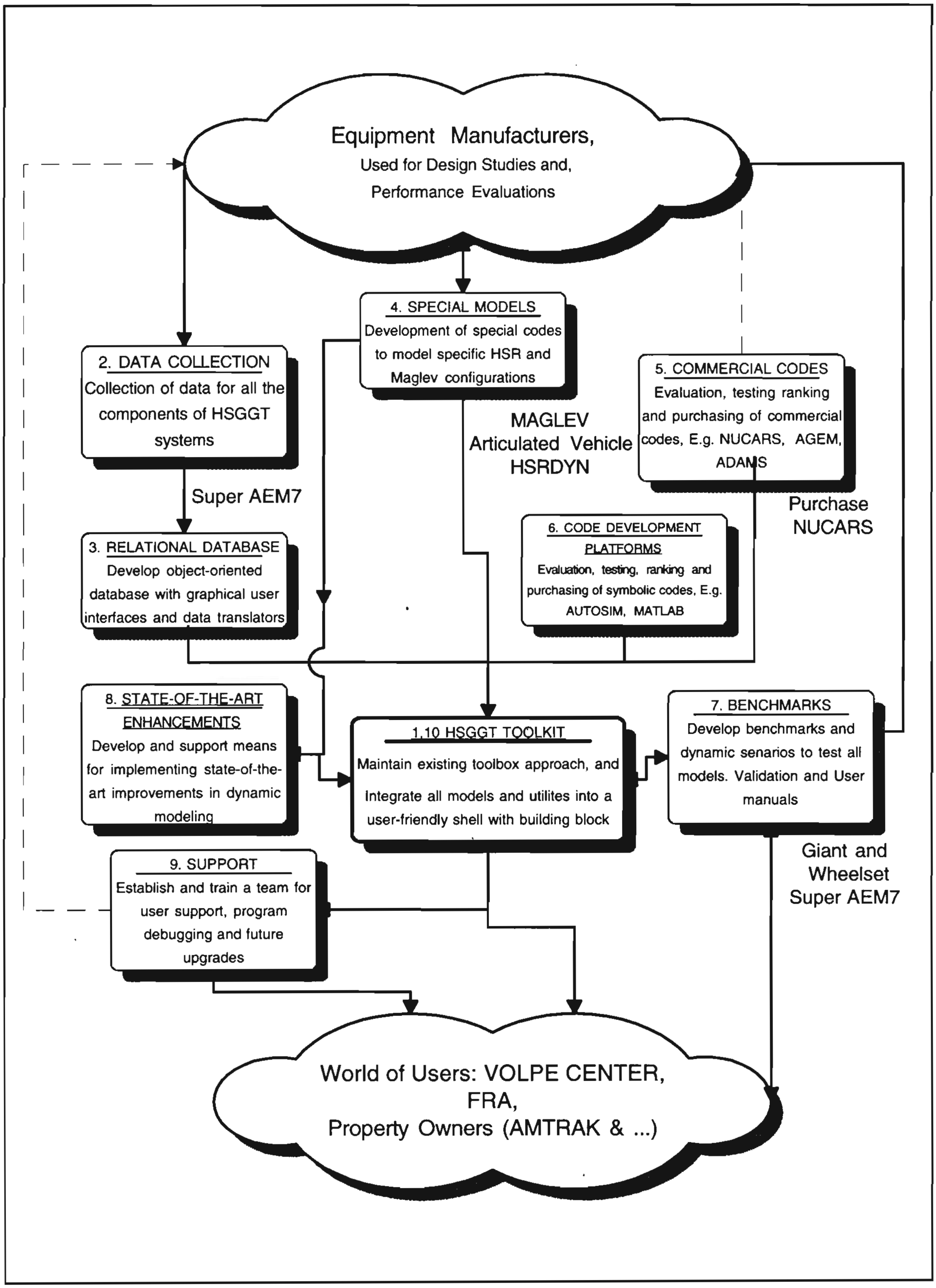


Figure 8-1. Recommended strategy for establishing an HSGGT analytical toolkit.

Appendix A

HSR Vehicle Configurations

Appendix A

HSR Vehicle Configurations

A.1 ETR450 Tilting Trains

The Italian ETR450 trainsets, built by Fiat and ABB, are descendants of the "Pendolino" (ETR401) train, which began operation in 1975. The ETR401 had a passive tilt-body feature, where the car body roll center is above the vehicle center of gravity. Complaints about travel sickness on routes with high curves forced the FS to reduce the tilt to 3-4° and was a motivating factor for the development of the active tilting train ETR450. This new train has been in service since 1989 at a maximum operating speed of 250 km/h. It can tilt up to 10° and can negotiate curves with up to 1.8 m/s^2 of unbalanced lateral acceleration, without subjecting passengers to more than 1 m/s^2 of lateral acceleration. This allows it to travel 25-30 percent through curves than a conventional train[A-1,A-2,A-3][1].

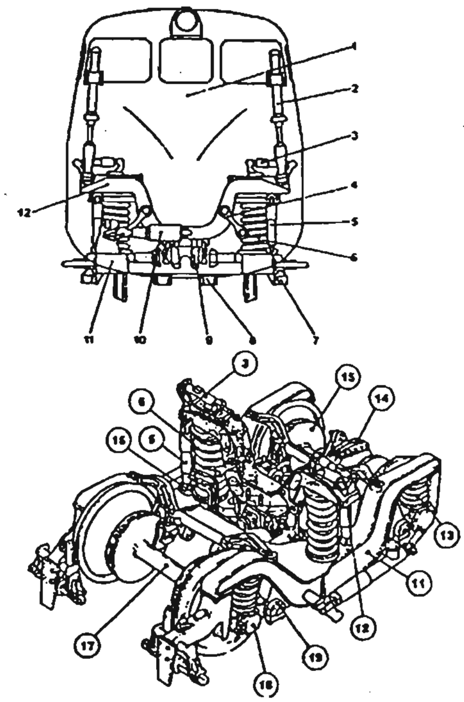
The ETR450 trainset consists of eight power cars (four coupled pairs) and a trailer car. The total train weight is about 440 tonnes (969,160 lb) and its length is 237 m. (778 ft.). There are two traction motors per car, each suspended under the car body and driving one axle per truck. Total train continuous power is 6,250 kW.

A.1.1 Trucks and Suspension

The truck of the ETR 450 is shown in Figure A-1. It has low unsprung masses and very low axleloads (11.5 t. to 13.5 t.). The truck frame is articulated which allows it to adapt to track twist irregularities without wheel-rail dynamic load variations. The primary suspension elements include coil springs oriented vertically between the axle boxes and truck frame. The axleboxes are constrained longitudinally by an axlebox guide yoke and have a stiff lateral and longitudinal elastomeric primary suspension.

The secondary suspension consists of vertically oriented coil springs with parallel dampers, located between the truck frame and the bolster. The car body is connected to the bolster by guiding levers which allow it to tilt around a point above its gravity center. This ensures inherent stability.

[1] References appear at the end of this appendix.



1 - Car body
2 - Tilting cylinders
3 - Secondary lateral damper
4 - Tilting guide lever
5 - Secondary vertical damper
6 - Secondary suspension
7 - Accelerometer
8 - Gyroscope
9 - Lateral bumpstop
10 - Lateral active suspension
11 - Bogie frame
12 - Bolster
13 - Primary suspension
14 - Axle gear box
15 - Traction axle
16 - Longitudinal bumpstop
17 - Trailing axle
18 - Axle box
19 - Axle guide

Figure A-1. ETR 450 motor truck.

An active lateral suspension between the truck and the bolster forces the car body to remain within the prescribed dynamic gauge, no matter what tilting angle, and a lateral bumpstop is included as a failsafe limit to lateral movements. Lateral dampers are also included between the carbody and the bolster.

In the tilt control system, signals from a rate gyroscope and an accelerometer mounted on each truck are used to drive a pair of hydraulic tilt cylinders on each truck. Each cylinder is connected between the bolster and a point relatively high on the car body. The control system starts the tilting action as soon as the truck enters a transition, and the tilting angle builds up so as to minimize the residual lateral acceleration and jerk and the car body roll speed. The active lateral suspension applies a force between the truck and the bolster that is proportional to the tilt angle and therefore opposes exactly the centrifugal force.

Because of the high unbalanced lateral acceleration possible with tilt body trains, track forces were extensively measured. The results showed that derailment quotients (Y/Q) stayed well below the

0.8 limit and the lateral forces stayed below the Prud'Homme limit. The FS uses a coefficient of 1 instead of 0.85 for passengers trains in the Prud'Homme limit formula, i.e., $Y = (10 + P/3)$. The wheels, after 350,000 km of service showed a small amount of flange wear. These low forces and wheel wear are due largely to the very low axleloads.

A.1.2 Power Transmission and Braking

Each DC traction motor drives the inner truck axles through Cardan shafts and a reduction gear. The longitudinal traction and braking forces are transferred to the car body through the secondary suspension until the truck displacement reaches the longitudinal bumpstop. The ETR450 uses disc-type air brakes and regenerative braking, rated at 7800 kW. The power supply on the Florence-Rome line (direttissima), where the ETR450 are in service, is 3 kV DC.

A.1.3 Special Features

The trucks have anti-skid devices. Each trainset has two pantographs (one on each of two cars), which are carried on a truck-supported frame so that the pantograph position is independent of car body tilt. The tilting mechanism and the pantograph frame limit the car body's interior to a narrow corridor above each truck, which results in a loss of about 12 seats.

A.2 ETR 500 Train

The ETR500 is Italy's new, non-tilting train, which aims for high speeds, good passenger comfort, and low track wear[A-4 through A-7]. It has more powerful traction equipment, simplified component designs, improved diagnostic systems and an improved driver's cab design, compared to the ETR450. It will run on the new high speed Turin-Milan-Naples line by the end of the century, at speeds of up to 300 km/h. In size and performance, it will be similar to the ICE train. It has two 68 t. locomotives at each end and 11 coaches for a train weight of 640 t. and a length of 328 m. An order of 30 trainsets has been placed in 1992 with the TREVI Consortium, and delivery will start in 1995. The first 25 will be for the Italian network at 3 kV DC, and the last 5 will be multi-current (25 kV 50 Hz) for international service.

A.2.1 Trucks and Suspension

The motor truck of the ETR 500 is shown in Figure A-2. The frame of the truck is a rectangle with the center of the long sides depressed and connected by a central beam. The wheelsets have hollow axles to lighten the unspung mass and are connected to the truck frame by two longitudinal rods (Z-links). The resulting wheelbase is 3 m. Each axlebox has three coil springs and dampers for primary suspension.

The secondary suspension consists of two pairs of flexicoil springs and an antiroll bar. Hydraulic dampers are provided in the vertical, lateral, and yaw directions. In addition, a pneumatic active lateral suspension keeps the car body centered over the truck. The control signal is provided by a lateral accelerometer. In this way, the lateral suspension characteristic in curve remains nearly the same as on straight track.

The traction motor and transmission gear are suspended under the car body by four rods supporting the weight and a central rubber pad allowing some freedom in yaw and lateral displacement. A patented twin hollow transmission shaft allows relatively large movements in lateral and longitudinal direction between the motor and the wheelset. On tight curves, two pre-charged, lateral acting spring cylinders, which connect the motor in yaw to the car body, are bypassed and the motor gets linked to the truck movements. A traction rod is mounted in low position between the central beam of the truck frame and the car body's end. The wheels are of a new monoblock lightweight type.

The trailer truck is shown in Figure A-3. It is a light and simple frame formed by two longitudinal beams, depressed at their center, and connected by two tubular traverses. Each axle has three steel disk brakes, a primary triple coil suspension, a vertical damper, and the axlebox is connected to the truck by a Z-link scheme. The truck is also connected longitudinally to the car body by a Z-links with bumpstops. The secondary suspension consists of two pairs of short coil springs. Damping is provided in the vertical, lateral and yaw directions.

Tests have been performed at speeds of up to 316 km/h with new and worn wheel profiles. The trucks have remained stable with good safety margins. Even without anti-yaw devices, the trucks remained stable at speeds of 250 km/h. The locomotive had a behavior similar to the trailers. The ride index with quite good, especially in the lateral plane. With worn wheels, the ride quality decreases somewhat. The lateral and vertical wheel dynamic forces stayed with 60 - 80 percent of their allowable limit. The L/V ratios did not exceed 0.6.

A.2.2 Power Transmission and Braking13

Each locomotive axle is powered by an asynchronous traction motor and each motor pair is powered by an inverter composed of oil-cooled GTO thyristors. The locomotive can develop 4,400 kW of traction at the wheel rim (continuous rating). For dynamic braking, the locomotive has

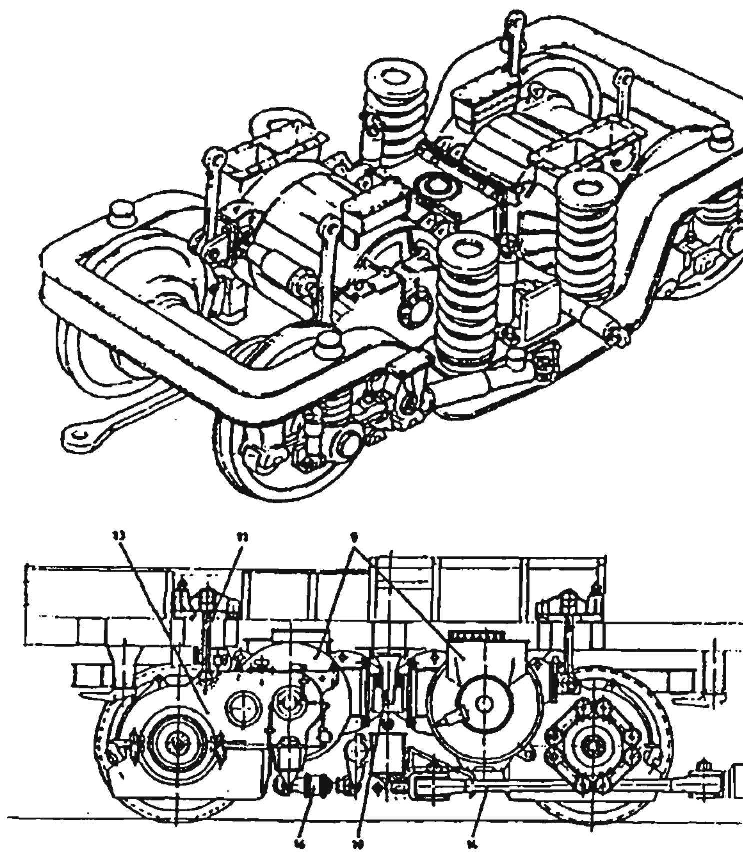


Figure A-2. ETR 500 motor truck.

resistors that can dissipate up to 3,400 kW, and this can be done even if catenary power is down. The maximum startup tractive effort is 200 kN, and the maximum dynamic braking force is 120 kN. The traction invertors are reversible and are able to regenerate energy during braking, up to a maximum of 4,400 kW.

Braking is provided by blending electro-pneumatic disk brakes with dynamic braking. At 300 km/h, emergency braking can stop the ETR 500 in 3,400 m, or an average deceleration of about 1 m/s^2.

A.3 ICE Train

The German InterCity Express (ICE) has been in joint development for over a decade by the railway industry and the German Federal Railways (Deutsches Bundesbahnen)[A-8 through A-18]. In this development program, two basic objectives were met: test operations at speeds up to 350 km/h

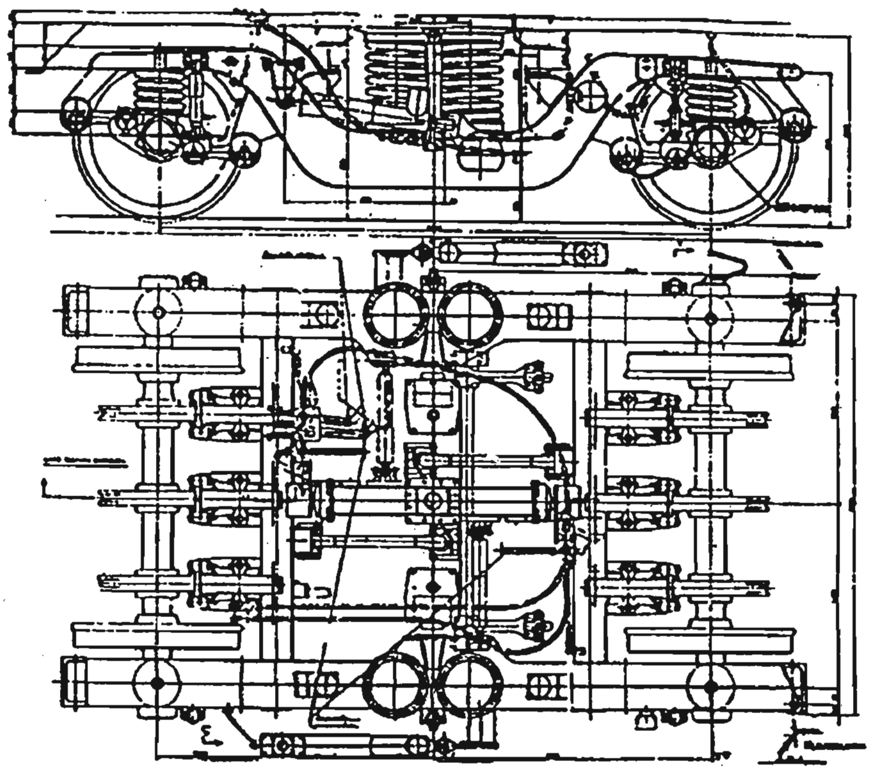


Figure A-3. ETR 500 trailer truck.

(218 mph), and revenue operations at speeds up to 300 km/h (186 mph). These objectives were met by analytical studies (computer simulations of train stability at high speeds, for example), laboratory roller rig experiments, and extensive high-speed running on specially-prepared test tracks. A maximum speed of 407 km/h (252 mph) was actually achieved in tests in 1988.

The ICE train is now in service at an authorized speed limit of 250 km/h between Hanover and Würzburg allowing start-to-stop schedules between certain cities of 180 km/h or higher. This has been made possible by construction of new high-speed lines (Neubaustrecke, NBS) and upgraded automatic train control equipment on some existing lines.

The ICE train consists of two power cars of 78 t. each and up to 12 (14 maximum) trailer cars in the standard two-bogie, four-axle configuration for all cars. The total train weight is 784 tonnes (2 power, 1 service, 1 restaurant, and 10 trailer cars) at a length of 357 meters (1171 ft). Current service speed is listed at 250 km/h (155 mph) with a maximum speed limit of 280 km/h (174 mph).

A modified ICE train was tested and demonstrated on Amtrak's NEC trackage during the summer of 1993, reaching speeds of 261 km/h (162 mph) during these tests. Revenue service demonstrations were run as a Washington to New York Metroliner train. The ICE train is expected to be offered by the consortium of Siemens Transportation Systems, Electro-Motive Division (GM) and AEG Transportation Systems as one group in Amtrak's upcoming high-speed train procurement

The German railway industry currently is developing an ICE-M version for European-wide use. The design speed of the ICE-M is up to 350 km/h. Full commercial service is expected by 1998.

A.3.1 Trucks and Suspension

For the trailer cars, the wheel diameter in new and worn condition is 920 mm and 870 mm, respectively and the profiles conform to UIC/ORES 1002, the European wheel. On the power car, the wheelsets have hollow axles and monobloc wheel discs. The wheel diameter in the new and worn condition is 1040 and 950 mm, respectively. The contact geometry on rails with an inward cant of 1:40 results in the equivalent conicity falling in the 0.1 - 0.2 range. There are two steel brake discs per wheelset.

The passenger coaches on the production train have Type MD530 trucks with 2.5 m wheelbase and a 13 t. axleload. Coil springs are used in the primary and secondary suspension, in parallel with vertical and lateral hydraulic dampers and in combination with friction surfaces. The damping force of the hydraulic lateral ("transverse") bolster dampers is quoted at 4000 N. Radius rods connect the bolster longitudinally to the truck sideframes.

The power car has two 3 m wheelbase trucks with welded frames and an axle load of 19.5 t. Like the MD530 trucks, the primary and secondary suspension stages include coil springs and vertical primary dampers. Longitudinal "radius" rods connect the wheelset to the truck frame The secondary coil springs are hinged to the car body with "push-pull rods". Figure A-4 shows a top and side view of the power truck. The traction motors are connected between the trucks and car body so that about two-thirds the weight of the motors and intermediate-gear drives are sprung to the car body (at a pivot with rubber elements), and one-third by the "bogie head support" (through vertical swing links that hang below the truck frame). The longitudinal load is transmitted between the car body and the rigid, rectangular truck frames via draw bars. The bogie frame is bolsterless and is sprung to the car body via vertical coil springs.

A.3.2 Traction and Braking

Power is transmitted to the wheelsets from the motor through the transmission, brake hollow-shaft star, cardanic rods, fork star and cardanic rods.

The braking system of the ICE train has been described in several publications[A-9,A-10,A-13] and reports[A-12]. Details of the braking system on the ICE train for NEC demonstration was provided by Knorr-Bremse AG through Siemens Transportation Systems, Inc.[A-16]. The salient features of the

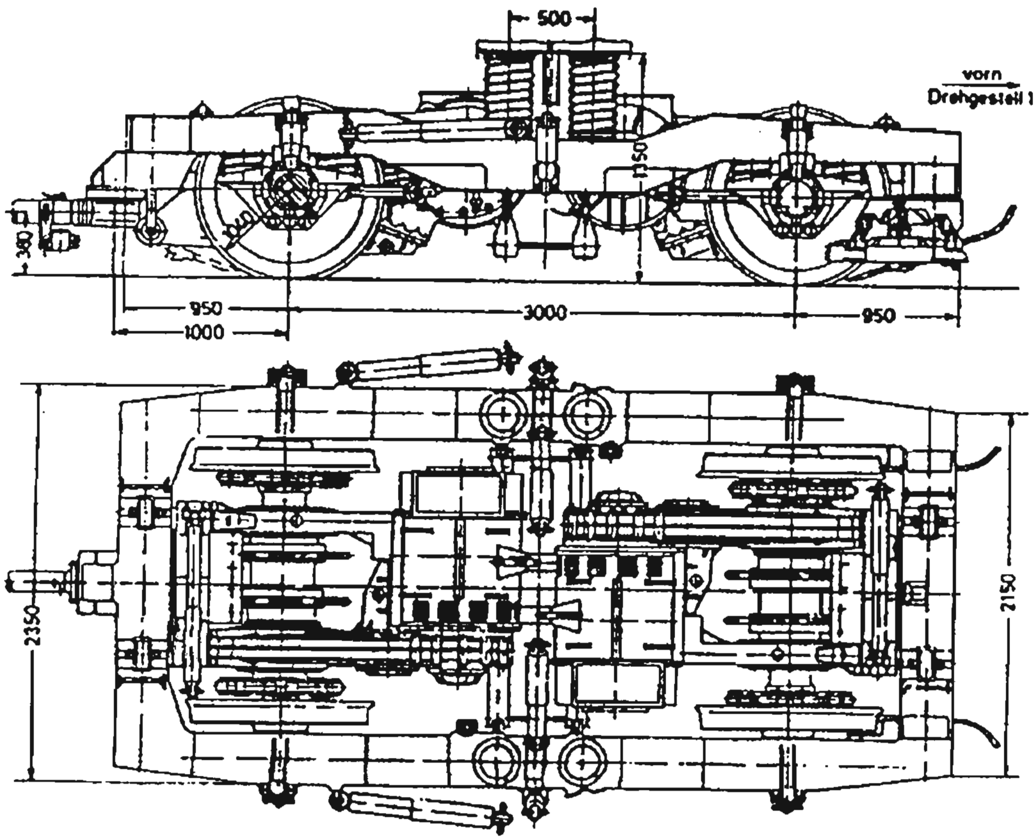


Figure A-4. ICE power truck.

brake system include:

- Dynamic/regenerative braking, returning kinetic energy to the power grid,
- Computer-controlled priority allocation of braking between dynamic and friction brakes,
- Computer-assisted fault monitoring, diagnostics, and automatic brake test,
- Communications via a fiber optic waveguide train data bus.

A.3.3 Special Features

The anti-skid device on the power car interfaces with the traction control as well as the dynamic and friction braking systems to assure optimum use of adhesion during both power and braking modes. To improve wheel/rail adhesion in adverse conditions, power cars are also equipped with sanders to sand the rail ahead of the leading wheelsets of the two bogies of the leading power car. Sand is metered according to train speed, above or below 140 km/h (87 mph), to get roughly the same sand per distance traveled.

A.4 Shinkansen Trains

The Shinkansen trains began operation in 1960, and several versions have evolved since[A-19,A-20,A-21]. In 1989 alone, 236 million passengers took the Shinkansen and generated 66 billion passenger-km. The prototype Series 961 train was evaluated by the Unified Industries team (which included Battelle and CMRI) under the Improved Passenger Equipment Evaluation Program (IPEEP). These analyses included computer simulation runs to evaluate the dynamic performance of the Series 961.

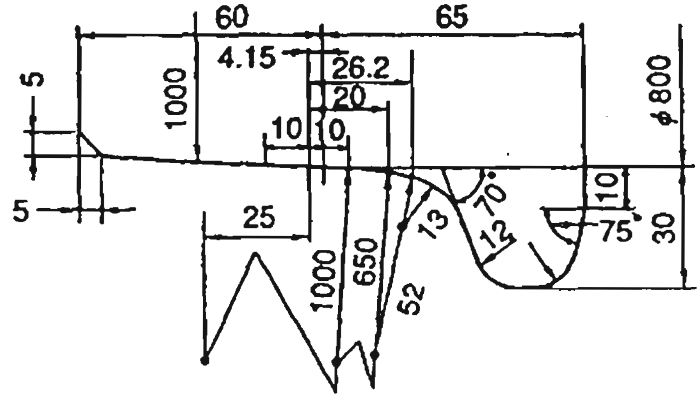
Since 1985 three new series have been introduced: the 100 Series, 200 Series and the latest, the 300 Series - Nozomi, introduced in march 1993. Major differences in the 961, 100, 200, and 300 Series include changes in the traction motors and control equipment, decreased vehicle weight (achieved, e.g., by changing from steel to aluminum car bodies), car body shape changes for improved aerodynamic performance, and changes in the truck design (e.g., changing to a bolsterless truck). All of these are multiple-units, non-tilting trains. For the latest 300 Series, the total weight is 40.6 tons per car, and the acceleration capability 1.6 km/h/s (0.96 mph/s). Top speed is rated at 270 km/h (162 mph).

Several new trains are in testing and development for future revenue speed of up to 350 km/h: the STAR21 and the WIN350. Car weights are lowered and aerodynamic shapes are refined in an effort to lower the noise impact, which is becoming the main obstacle to higher operating speed in Japan. Discussion below will focus on the *Series 300 Shinkansen:* the "Nozomi".

A.4.1 Trucks and Suspension for the Nozomi

To decrease noise and vibration, special attention has been focused on wheel mass balance. The brake disks which are mounted on each side of each wheel are the main source of unbalance. Balancing these disks and fully machining the wheels has improved the ride quality. With 1.6 Nm of unbalance, ride quality is in the "Very Poor" area, whereas below 0.4 Nm it is in the "Excellent" category. Wheel diameter is 0.86 m, and the tread has an arched profile to minimize wheel wear. This profile has been in use since the Series 100 trains. On the original series 961 trains, the wheels had a conical profile and had to be reprofiled every 70,000 km to maintain riding quality. An example of a wheel with arched profile in shown in Figure A-5.

The Series 300 truck, shown in Figure A-6, has been considerably lightened from the earlier Series 100 trucks. The truck mass has been brought down from 10 t. to 7.7 t. by dropping the



Arched Profile - Dim (mm)

Figure A-5. Profile of Shinkansen wheel.

bolster, removing the end beams of the frame, and replacing the DC motors by AC traction motors. The unsprung mass has also been lowered from 4.6 t. to 3.5 t. by hollowing the axles and by using aluminum alloy on the axle-box and gear casing. The wheelbase has been kept at 2.5 m.

The primary suspension consists of a coil spring, a laminated cylindrical rubber spring and an axle spring vertical damper. The combination of leaf springs and coil springs in the earlier Skinkansen Series has been abandoned. To reach a good compromise between high running stability and good curve negotiating performance, the optimum longitudinal and lateral stiffness as well as the damping characteristics of the rubber bushing were selected through simulation and confirmed by testing. These are, per axle, 18.8 kN/mm for longitudinal and 9.6 kN/mm for lateral stiffness. This represents about half and a quarter of the earlier Series trucks longitudinal and lateral stiffnesses, respectively. Yaw dampers with very low free play were also used to raise the critical speed. These trucks were tested on a roller rig with a 450 km/h top speed. They remained stable at speeds up to 400 km/h and were even stable at 300 km/h with the two anti-yaw dampers removed.

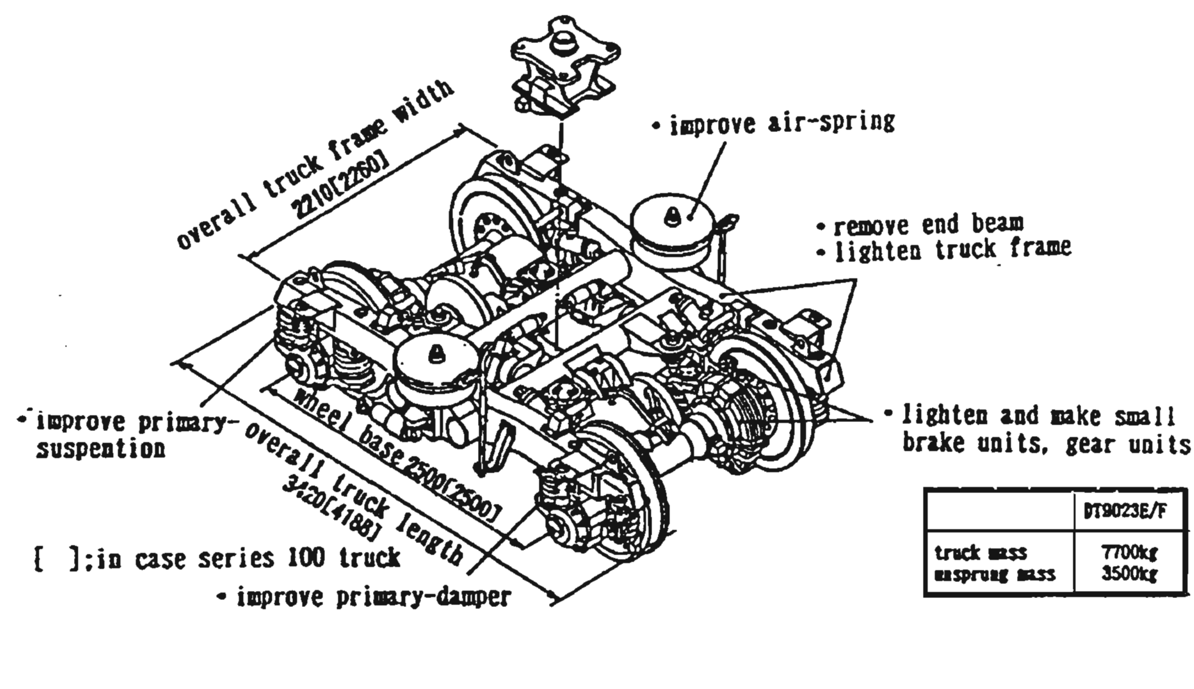
The truck is linked to the car body by a resilient pin, and air bags are used for the secondary suspension. The air spring effective diameter and variation rate of the effective area under pressure are reduced, leading to improved riding quality. Tests on the Osaka - Tokyo line showed that the maximum peak to peak amplitude of vertical and lateral vibrations in the car body stayed around 1 m/s^2 over a large range of speeds.

Improvements to these existing trucks are being investigated. In particular reduction of the unsprung mass by suspending partially or fully the traction motors and the disk brakes to the car body are studied. The motors would have the disk brakes on the rotor shaft and the transmission would consist of a right angle cardan gear with a coupling allowing large lateral displacements.

A.4.2 Power Transmission and Braking

A Nozomi trainset consists of five groups of three cars (two motive units and a trailer in-between) plus a head unit (10M+6T). Each motive unit has four, 300 kW, ventilated, squirrel cage, AC induction motors, which are powered by a Variable Speed Variable Frequency system. Also no heat is transferred to the rails. A special continuous torque monitoring and control system ensures best use of available adhesion. The power conditioning system allows regenerative as well as dynamic braking.

The motor trucks are equipped with electrically controlled air brakes. Trailer cars have mechanical disk brakes as well as eddy-current steel disk brakes with a car load sensitive control. The eddy-current steel disks absorb the heat from the braking and do not experience any mechanical wear.



	DT9023E/F
truck mass unsprung mass	7700kg 3500kg

Figure A-6. Shinkansen Series 300 motor truck.

A.4.3 Special Features

The 300 Series uses two pantographs per trainset, which were specially designed for small size and have a primary and secondary suspension. The roof has pantograph cowlings to improve air flow and reduce noise from the pantographs. Car bodies have been designed with a lower height, improved aerodynamic shape, and very smooth skin to reduce drag and noise generation. The car center of gravity has also been lowered by placing the HVAC units and other equipment bays, under the car body.

A.5 TGV Trains

The French high-speed rail technology is embodied in the Train à Grande Vitesse (TGV) system[A-22 through A-28]. The first TGV line, the TGV Sud-Est (TGV-SE), has been operating since 1981 at a maximum speed of 270 km/h (168 mph). Planning for the TGV Atlantique (TGV-A), began in 1978, construction started in 1985, and revenue operations in 1989. For the TGV-A the maximum authorized speed is 300 km/h (186 mph). In mid 1993, a third high speed line opened to serve the north of France, and soon, Bruxelles and London through the Euro-Tunnel. The trainsets on that line are TGV-N which are derived from the TGV-A but have only 8 trailers and are pressure-sealed. For the international traffic, the Eurostar trainset has been developed. It has 2 power cars and 18 trailers and can seat 794. It will conform to the norms of the SNCF, SNCB, BR, and Euro-tunnel, especially regarding clearance, voltage and signalling.

A third generation of TGV has been ordered, the TGV-2N, a bi-level train which will start revenue service in 1996. Even though this new trainset has almost 50 percent more seats than the TGV-PSE, and has a greater gauge (taller cars), its axleload will stay under 17 t. (same train length and weight). This illustrates well the innovations in design and materials achieved by the SNCF.

To date, the French National Railways (SNCF) has transported over 160 million people on the two operating TGV lines with an impressive safety record. No passenger fatalities have resulted from TGV operations on the dedicated high-speed lines.[2] It should be noted that SNCF builds new track alignments for high-speed operation using premium components. On the international market, the TGV train concept has been selected for the Korean High Speed Rail (Seoul to Pusan) and for the Texas Franchise (Houston Dallas/Fort Worth San Antonio and Austin).

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[2]A TGV operator and one passenger were killed when a TGV-SE train, traveling on a non-high-speed line, struck a highway truck carrying a 59 t (65 ton) press. The truck crossed the track at an unapproved location.

A.5.1 Trucks and Suspension

The "SNCF wheel" is monoblock for reasons of lightness, diffusion of heat during braking and for the elimination of the risk of loosening of the shrink-fit under traction. It has a common plane of symmetry for the hub, the web, and the rim to avoid the risk of buckling. The nominal new wheel diameter is 0.92 m (3.0 ft) and the axle-load is kept under 17 t. The wheels have a conical 1:40 profile and run on the UIC 60 rails which are laid with an inward cant of 1:20. This results in a low initial equivalent conicity of 0.025 which increases only very slowly with wear, because of the absence of tread brakes. On the TGV-A, wheels are reprofiled approximately every 400,000 km, before they reach an equivalent conicity of 0.05.

The TGV power trucks are designed to be lightweight and stiff so they will be stable throughout the operating speed regime. During the 1990 high speed tests, which reached a record speed of 515 km/h, the critical truck speed has been estimated at 700 km/h. The truck has two side frames with a central transom. The primary suspension uses both metal-rubber and helical coil springs. Vertical motion is absorbed by the helical coil springs, as well as by the resilient components. The lateral motion is absorbed by resilient components, and the assembly is equipped with an anti-pitch damper. The motorized truck, shown in Figure A-7, is equipped with one transverse, two vertical, and two anti-hunting dampers.

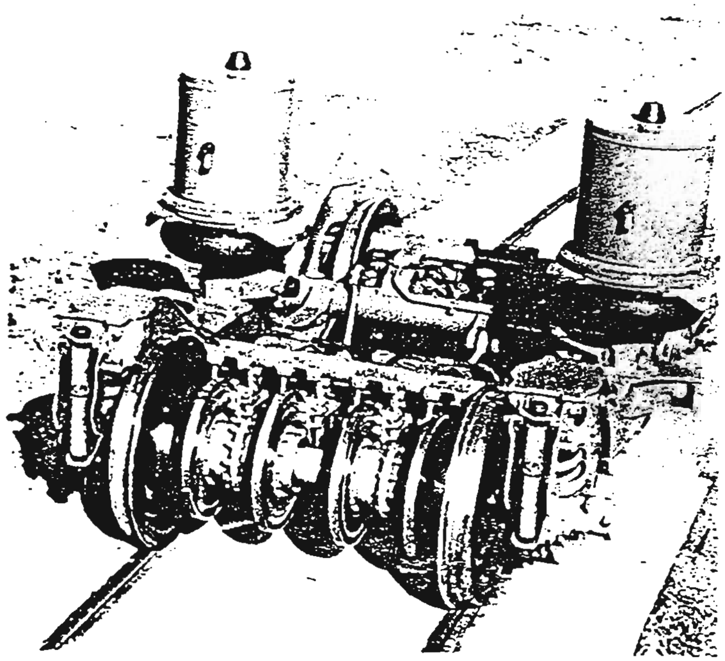


Figure A-8. TGV-A carrying truck.

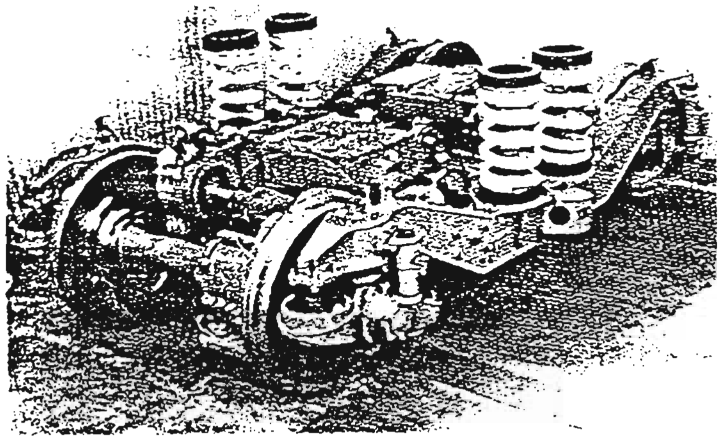


Figure A-7. TGV-A power truck.

The non-motorized TGV truck has two side frames and two transoms which support the disc brakes (Figure A-8). The primary suspension is a link arm type which allows decoupling of the vertical and lateral or guidance functions. Dampers are provided between the linkage and the truck frame. Anti-yaw dampers between the truck and the car body contribute to the truck's high speed stability. In the primary suspension of the TGV-A truck, the axlebox is constrained longitudinally and laterally, by the primary suspension pivot rod, while a coil spring provides the vertical suspension. As a result it has been possible to optimize the lateral and longitudinal suspension independently of the vertical suspension, according to criteria specific to each.

The unsprung mass per wheelset is 2048 kg on the motor truck and 2003 kg on the trailer. The motor truck sprung mass is about 2400 kg, and the trailer sprung mass is 3100 kg. This results in a total weight of 6500 kg for the motor truck and 7100 kg for the trailer truck.

The car body is attached to the truck trough with a double-hinged vertical pin mounted in resilient bearings. The traction motors are mounted on the car body to reduce the unsprung mass and provide high-speed stability. In the motorized trucks, the secondary suspension consists of coil springs in series with elastomeric pads. In the non-motorized trucks, the secondary suspension is provided by two air bags and associated reservoirs. The air bags provide also a low lateral stiffness which decouples the body frame hunting movement and the truck hunting movement perfectly, yielding excellent behavior on curves negotiated at medium speeds on conventional lines. Secondary damping is done directly between car bodies which further contributes to the elimination of parasite transmitters of truck-to-body frame vibrations.

Each truck has accelerometers that can detect the onset of hunting behavior. The operator would be notified of the condition and would then slow down until the hunting stopped and presumably would call for inspection and maintenance of the truck assembly at the earliest opportunity.

A.5.2 Power Transmission and Braking

Each TGV-A power car is fitted with two pantographs, one for 25 kV, 50 Hz AC power collection and the other for 1500 VDC power collection. The TGV-A locomotives have four, 1100 kW, light-weight, inverter-driven, AC synchronous traction motors. The inverters, based on GTO thyristor circuits, include a power factor control system which helps lower the line currents and allows reductions in the size of wayside power and distribution components. These motors deliver a very substantial starting effort that has made it possible to raise the maximum value of grades on the TGV-A line from 1.5 percent to 2.5 percent thereby yielding considerable savings in civil engineering work while at the same time allowing for the possibility of re-starting on a grade with one truck inoperative.

The drive system consists of a motor gearbox, sliding cardan shaft and axle-mounted bevel gearboxes. The traction motors, which are forced-ventilated, are hung from the car body frame and connected to the gear boxes through cardan shafts, thus reducing the unsprung mass of the truck.

The braking system on the TGV-A consists of dynamic, pneumatic and electro-pneumatic braking components. In addition to dynamic braking, there is one sintered metal shoe per wheel. In the trailer cars, braking is accomplished with four steel double disks, unventilated, with sintered metal linings. Each truck's braking elements contain independent controls with redundancy of vital automatic controls, such as blending the action of the different brakes, and providing anti-skid control. The dynamic brakes can operate even if catenary power is lost, because storage batteries on the locomotive can energize the inverters.

The benefits of the high performance of the disk brakes together with microprocessor-based wheel slip prevention (to manage adhesion available for each axle continuously in real time), have been exploited to simplify the unpowered truck brake equipment by eliminating brake shoes. The emergency braking distance from 300 km/h on tangent level track, with low adhesion and loss of power has been brought under 3300 m.

A.5.3 Special Features

The TGV family of trains is based on an articulated consists with trucks shared by two consecutive trailers except at the ends of the trainset. The articulated arrangement (shown in Figure 3-2) employs a unique "articulating ring" with a ball-joint suspension design for support and stability of the two trailers. The design results in reduced number of trucks, decreasing the overall cost of the trainset (both in construction and maintenance), its weight, its aerodynamic drag, and some of its external noise impact.

More important to the crash management of the TGV are the intercar connections, especially the articulated car attachments. SNCF stated that these are designed to deform during severe collisions, reducing the peak longitudinal accelerations. The attachments do not allow override. This is critical in collision energy management, as the mass of the entire train must be considered, not just the individual cars. In addition, the attachment is designed to allow the longitudinal forces to be transmitted through the underframes and side sills, which are designed for large loads. The soundness of this design has been dramatically illustrated by a TGV Nord trainset which derailed recently at high speed, when the track sub-structure collapsed after high rains, and yet landed upright and intact, without any harm to its passengers.

Another advantage of articulated trainsets is the ease with which the latest trainsets, the TGV-N, could be made air-tight without impacting the circulation between cars, and the fact that the new bi-level cars could be designed with an unbroken second level circulation. An pressure-sealed trainset shelters the passengers from shock waves experienced in entering tunnels and has lower internal noise levels.

A.6 X2000 Tilting Train

The X2000 trainset is one of the technologies which has been demonstrated in the U.S. and may be procured by Amtrak for high speed operation on the Northeast Corridor[A-29 through A-33]. The system developer, Asea Brown Boveri Traction AB (ABB) recently teamed with American partners Raytheon, General Dynamics, and GE Transportation Systems to pre-qualify for the Amtrak bid of 26 trainsets starting in April, 1994.

The X2000 tilting-body train has been under development in Sweden for over ten years, in cooperation with the Swedish National Railways (SJ). Revenue service operations with the train began on SJ's Stockholm to Gōteburg line in September 1990. A limited number of trainsets are currently in revenue service on three other SJ lines. The trainset order is scheduled for completion in 1994. Two additional proposed X2000 services include Oslo to Bergen in Norway, and Helsinki to Tampere in Finland.

The X2000 trainset currently in service consists of a 70 t. locomotive, four trailer cars, 54.5 t. each, and a cab control car, 55 t. at the train's other end. Overall, the nominal train weight is 343 tonnes with a length of 340 m (459 ft). The train is designed for a maximum speed of 210 km/h (130 mph), with a revenue service speed of 200 km/h (124 mph). With a power car at each end, a maximum of 12 trailer cars may be accommodated in one train.

The power car has no tilt mechanism so as not to require special pantograph supports. All trucks are self-steering with soft primary longitudinal stiffness and have low unsprung masses, except the cab end truck which is ballasted with 5 - 6 t. to ensure good performance in extreme weather conditions in the "driving trailer ahead" configuration. The driving trailer car also tilts.

A.6.1 Trucks and Suspension

The wheelsets are solid with a conical profile and have diameters of 1.1 m (43.3 inches) on the power car and 0.88 m (34.6 inches) on the trailer car. The power car truck has a rubber chevron primary suspension. Radial steering is achieved by a relatively soft primary, allowing the wheelsets to be positioned in a curve closer to the actual curve radial line by longitudinal creep forces. Stability is then attained by using primary hydraulic dampers located between the wheelset and the truck frame in an orientation that provides lateral and vertical damping forces. Sketches of these trucks are shown in Figures A-9 and A-10.

Load transfer between the trucks and car bodies is achieved in up to three manners: secondary suspension elements, traction force reaction elements, and the tilt mechanisms. Secondary suspension is provided by air springs. Secondary dampers between the bolster beam and truck frame, including hydraulic yaw dampers, control car body ride quality and kinematic hunting stability. Traction forces are transferred from the powered truck to the car body by a traction bar, which is located below the truck frame for vehicle dynamics considerations.

A.6.2 Power Transmission and Braking

Each power car has four AC, 3-phase asynchronous traction motors, which provide a maximum tractive effort of 160 kN and total continuous power rating of 3260 kW. Each pair of motors is powered by a variable voltage, variable frequency power supply with full regenerative braking, and power factor correction capabilities. The power supply units use oil-cooled GTO thyristor bridges. The motors are forced-ventilated, of squirrel-cage type construction. The motor

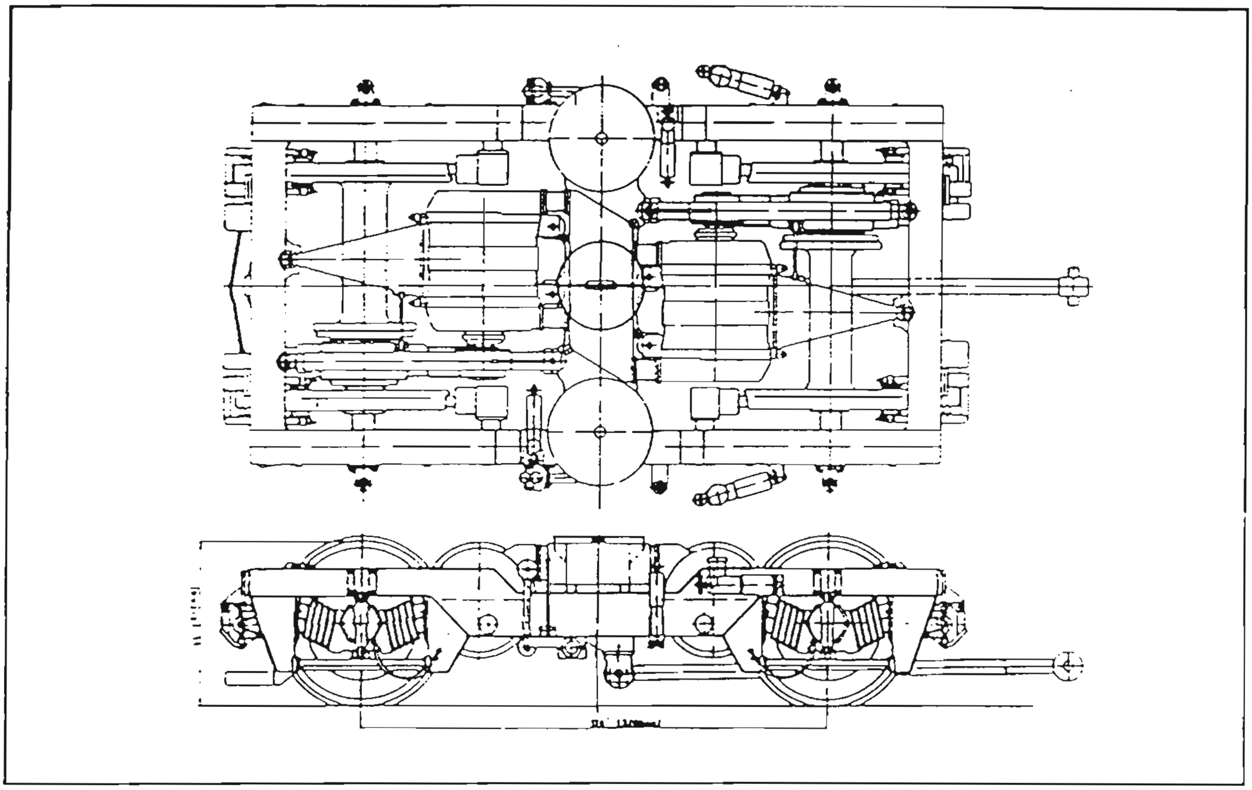
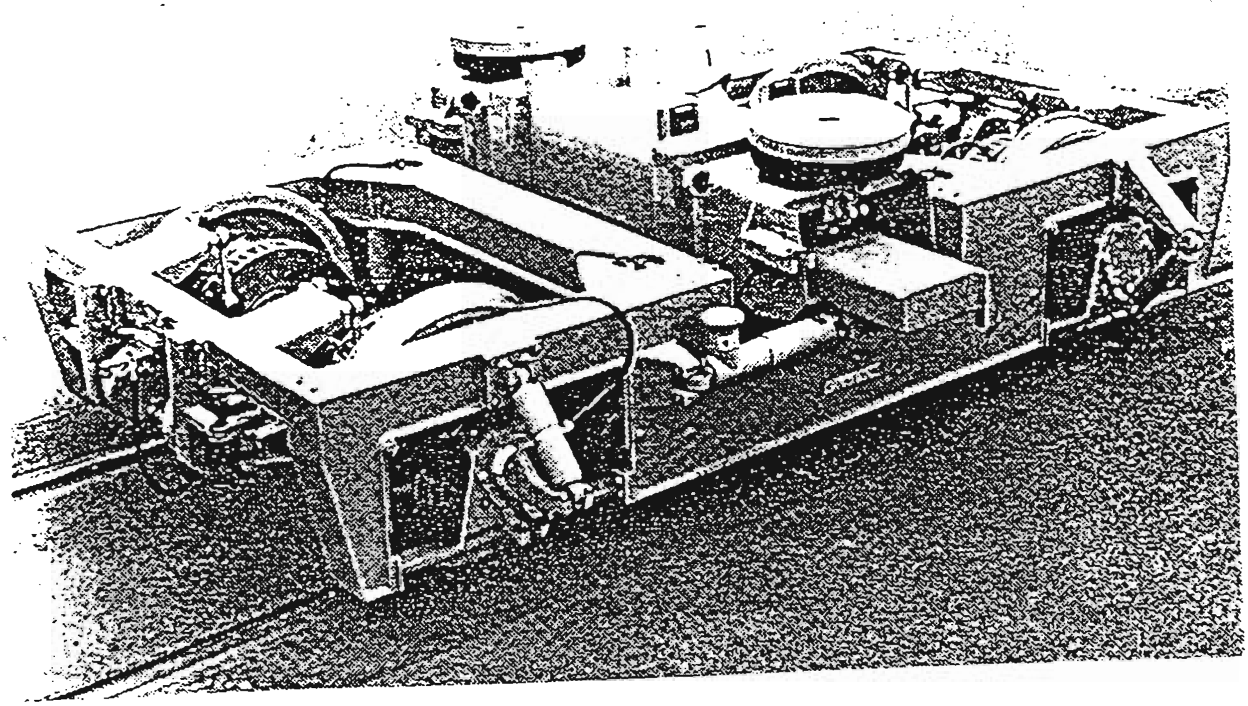


Figure A-9. X2000 power truck.



(Source: International Railway Journal, April 1990)

Figure A-10. X2000 trailer truck.

and gearbox are mounted on the truck frame via rubber elements. This, together with the use of hollow axles help reduce the unsprung masses.

The gearbox is connected to the wheelset by a quill tube. The tube is attached to one of the wheels by rubber bushings and to the gearbox by a tooth coupling. This arrangement allows for both angular alignment and lateral movement of the wheelset.

The brake system includes electric regenerative dynamic braking in the power car, compressed air operated disk brakes on all axles of the train, magnetic track brakes on the trailer cars and tread brakes on the wheels of the power car. The operator can select either regenerative or air-plus-regenerative braking. If regenerative braking is not available, the air brake systems automatically compensates. The magnetic track brakes are only used in emergency braking. All brakes are provided with an anti-skid system that reduces the risk of wheel flats.

A.6.3 Special Features

The coach and driving trailer cars have active tilt control system. The primary advantage of active-tilt technology is its ability to start the rotation as soon as the car enters the transition, and to program the rotation speed so as to minimize lateral acceleration, jerk and rotation speed that must be endured by the passengers. The tilting mechanism using hydraulic actuators, is controlled by accelerometers on the leading and trailing trucks, a speed sensor, and differential transformers on each tilting truck. A maximum tilt of 8° between bolster and car body can be reached; the effective tilt however is 6.5°. The tilt system is inactive at speeds below about 70 km/h (43 mph). Maximum tilt rate is about 4 degrees per second; this limit was established for ride comfort (travel sickness) considerations.

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Appendix B

Maglev Configurations

Appendix B
Maglev Configurations

B.1 DaeWoo Maglev System

DaeWoo Heavy Industries commenced a research and development program in 1989 leading to the DaeWoo Maglev System (DMV = DaeWoo Magnetically Levitated Vehicle) designed for intercity commuter service. In October 1992 three full size Maglev vehicles were produced and tested on a DaeWoo special track. A cross-sectional view of the DMV system is shown in Figure B-1.

The vehicle dimensions are 15 m x 3 m x 3 m, vehicle weight is 18 ton, the seating capacity is 40 passengers, and maximum design speed is 110 km/hr. The design employs an EMS system with an 11 mm air gap. Propulsion is accomplished using single sided linear induction motors.

The secondary suspension system consists of four pneumatic dampers per module providing support in the vertical direction. Lateral movement of the carbody is accommodated by a slide table and linear bearing to which the dampers are connected. Lateral movement of the module end of the damper is restricted by a spring equalizer. The slide table is connected to the module frame by a thrust rod enabling the transfer of the thrust force from the linear induction motor to the carbody.

The levitation system consists of six modules, each of which have four gap sensors, four accelerometers, two magnet drivers and two levitation controllers.

B.2 Japanese Railway MLU-002

The MLU-002 was a second generation prototype test vehicle of the Japanese (National) Railways (JR) that began operation in 1987. Over 40,000 km were logged on the Miyazaki test track before the vehicle was destroyed in an accidental fire. The design was an EDS system that used superconductivity for levitation and a long stator linear synchronous motor for propulsion. As the car accelerated, the superconductors induced eddy currents in the guideway coils that caused the vehicle to raise off the guideway (about 0.1 m above 100 km/h). Below lift-off speed the vehicle was supported by retractable rubber wheels. The power for the superconducting magnets as well as the cryogenic cooling system were obtained from on-board batteries. Mutual attraction and repulsion

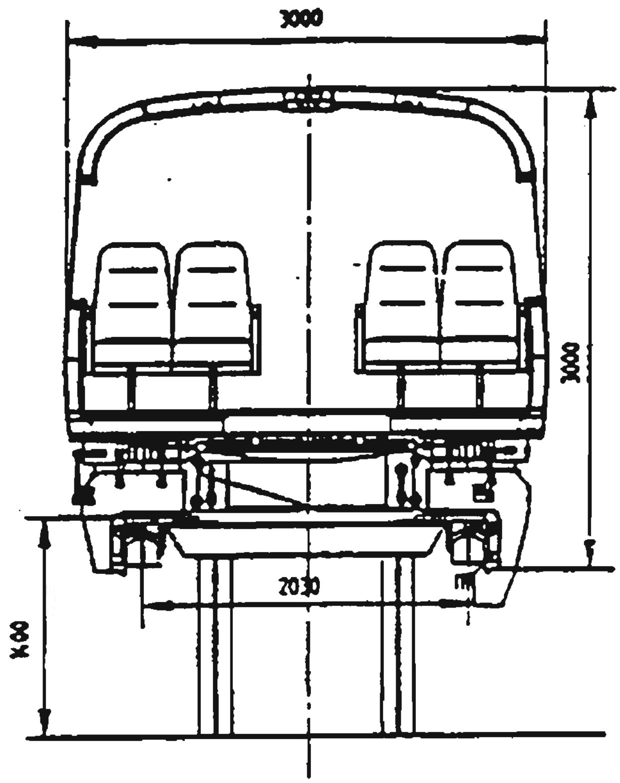

Figure B-1. Cross-Sectional view of the DaeWoo Maglev vehicle system.

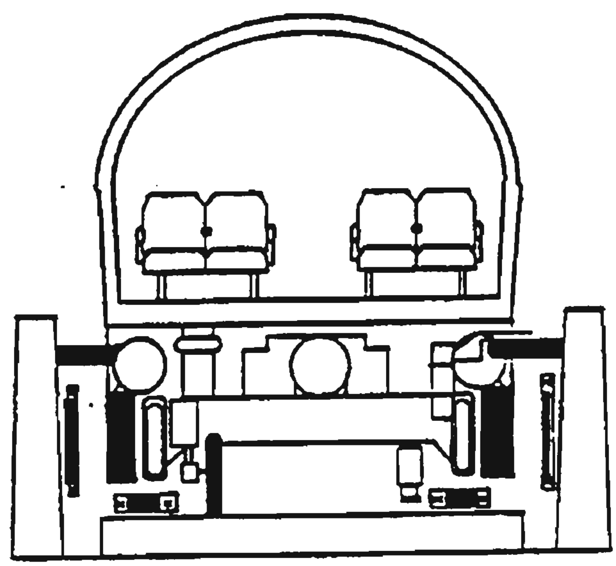

Figure B-2. Cross-sectional schematic of the JR MLU-002.

between the superconducting magnets and the propulsion coils in the U-shaped guideway centered the vehicle and restored it from lateral deviation. A cross-sectional schematic of the MLU-002 is shown in Figure B-2.

The principal features of the system can be summarized as follows: (i) EDS suspension, (ii) Null-flux magnetic guidance, (iii) Active track air core linear synchronous motor propulsion, and (iv) Multi-function superconducting magnets on vehicle which performs three different functions: levitation, guidance and propulsion. (In the "null-flux" concept, a passive coil configuration is set up in the guideway that intercepts zero net magnetic flux and generates no net guidance or drag forces when the vehicle is centered. Both the guidance and drag forces increase as the vehicle departs from its centered position. With this concept, during high speed operations, the electrodynamic drag forces are very small compared to the aerodynamic drag forces.)

The MLU-002 had two bogies with four niobium-titanium alloy superconducting coils located in the corners of each of the vehicles. These locations tended to minimize the magnetic exposure of the passengers. It had the disadvantage of reducing the control freedom of the vehicle, and making it more difficult to produce an acceptable ride quality.

In January 1993, the MLU-002 successor, the MLU-002N, was introduced on the Miyazaki Test Track. It was designed taking lessons from the fire accident (of October 3, 1991) by incorporating non-combustible materials in the body, new aluminum wheels without using magnesium alloy, fire-resistant hydraulic oil with high-temperature flashing points, fire extinguishing appliances, etc. Several components were redesigned, including the superconducting magnet which is now in a concentric arrangement. (It was reported that the superconducting magnet of MLU-002 was not sufficiently reliable, being subject to frequent quenching problems.) The MLU-002N also has two pairs of aerodynamic brake devices on the superstructure of the vehicle added as an emergency brake system. The MLU-002N has the same size and external appearance as the MLU-002.

The MLU-002N weights approximately 20 tons, has a 420 km/h maximum speed, and has dimensions of 22.0 m (L) x 3.7 m (H) x 3.0 m (W). The MLU-002 had 44 passenger seats; the MLU-002N has 12 (for now).

The conventional bogie incorporated in the MLU-002 consisted of cross-beams and side-beams rigidly connected with superconducting magnets. (See Figure B-3.) To improve ride comfort, a more elastic design has been introduced in the MLU-002N (as shown in Figure B-4). This "Double Bogie Frame" uses two different frames. One of them (bottom side) is the superconducting magnet supporting frame connecting another bogie frame with 4 air suspensions, and the other is called the "Equipment Frame" connecting the body with 2 air-springs.

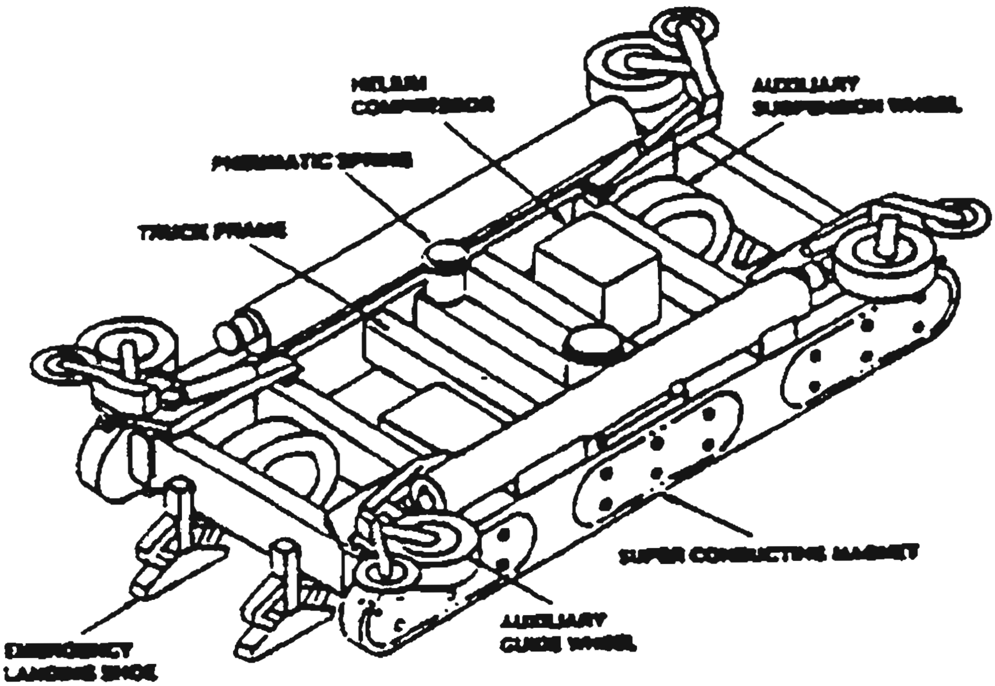

Figure B-3. Conventional bogie of the MLU-002.

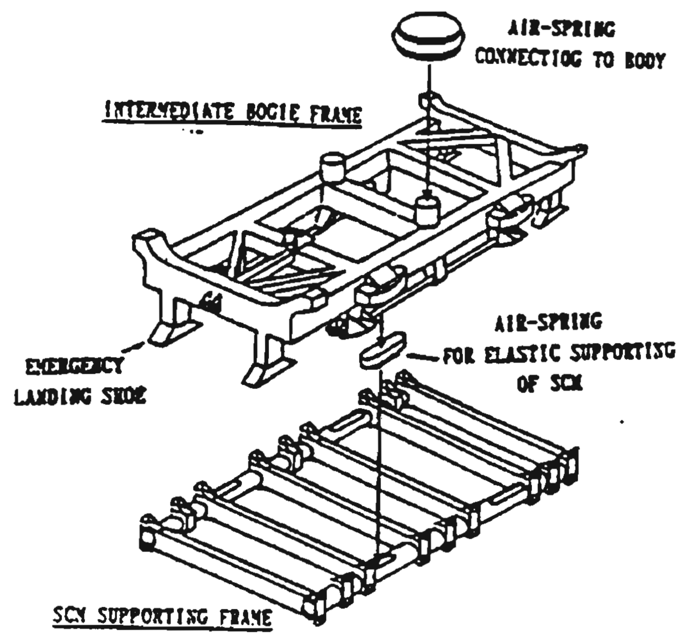

Figure B-4. Elastic bogie of the MLU-002N.

Although the primary (main service) brake is a regenerative brake capable of fine deceleration control, the MLU-002N incorporates two pairs of aerodynamic brakes as an auxiliary mechanical brake augmenting emergency landing shoes.

B.3 Japanese Air Lines HSST

A series of electromagnetic levitation vehicle systems, driven by linear induction traction motors and employing EMS for support and guidance, has been developed by Japanese Air Lines (JAL). These include HSST-03 (July 1984), HSST-04 (Jan 1988), HSST-05 (Jan 1989), and HSST-100 (May 1991). The HSST-05 consists of a two-car train, with 8 magnet modules and a 9 mm nominal gap; the HSST-100 employs 6 modules and has an 8 mm gap.

The HSST-05 carried about 1.3 million paying customers at the Yokohama Exposition held in 1989. The JAL system uses EMS for levitation and a short stator linear induction motor (LIM) for propulsion. The motor and the on-board auxiliaries use power pickup from wayside rails. The use of the short stator significantly reduces the cost of the guideway, but increases the weight of the vehicle and also results in a less efficient propulsion system (and consequently a higher operational cost).

The HSST-05 two-car train has a length of 36.5 m, weighs 43.5 tons empty and 59.4 tons loaded, and can carry up to 160 passengers. Eight suspension magnets are employed and 32 air springs are used for secondary support of the cars. The suspension magnets are also utilized for lateral stability. The HSST-05 braking system utilizes eight mechanical brakes per train as well as reversal of the motors and regeneration.

HSST utilizes either 12 or 16 m girders elevated to around 4.5 m on single beams. A ferromagnetic rail is attached to the girder to provide attraction for the suspension magnets on board the vehicle. Compensation mechanisms were included as part of the column design to enable adjustment to the guideway height should settling occur.

An early premise of the HSST system was that it be built on single track guideways, thus precluding the need for switches. Subsequent evolution has, however, caused HSST to develop a hydraulically-powered switch for dual guideway use.

The HSST-100 test train consists of a unit of two 8.5 m long vehicles and weighs 1 ton or less per meter of length of the module on each side. The vehicle is equipped with six modules, three on each side. Each module is equipped with four electromagnets for levitation and guidance, a primary coil of the linear motor, and a hydraulic brake unit.

Each module is equipped with two pairs of integrated levitation/guidance magnets (pair magnet), each of which is controlled to produce a constant levitation lift (8 mm for model 100) by controlling the magnet current in response to the displacement signal from the gap sensor and the acceleration signal from the acceleration sensor.

A schematic of the HSST-300 configuration is shown in Figure B-5.

B.4 Transrapid TR-07

The Transrapid TR-07 is an EMS Maglev system designed for operating speeds up to 500 km/h in revenue service. The TR-07 uses separate sets of conventional iron-core magnets to generate vehicle lift and guidance. The non-tilting vehicle wraps around a T-shaped guideway. Propulsion is by a long-stator linear synchronous motor. Attraction to edge-mounted guideway rails provides guidance; attraction to the stator-pack beneath the guideway generates lift. Control systems regulate levitation and guidance forces to maintain small (8-10 mm) air gaps. A concern with this design, given the small nominal air gap, is a magnet striking the guideway.

Figures B-6 and B-7 show a sketch of the Transrapid Maglev system from which one can see the support and guidance system. The Transrapid vehicle uses a suspension system that wraps around the guideway in a manner that effectively captures the guideway. A design feature is the uniform distribution of suspension and guidance magnets over the length of the vehicle. This produces an even loading of the guideway with potentially less stress in the guideway girder.

The TR-07 has been extensively tested in long term operation amounting to more than 150,000 km at the Transrapid Test Facility in Emsland. The dimensions of each TR-07 section are 27.0 m (L) x 3.7 m (W) x 4.1 m (H) for an end section and 24.8 m in length for an interior section. Each section can carry between 72 and 100 people depending on the cabin configuration. The maximum number of sections per train during revenue operation will be limited to 10. The maximum operational speed of 500 km/h requires a horizontal radius of 434 m when the guideway is banked at 12 degrees. The system is also capable of climbing 10% grades.

The Transrapid vehicle utilizes sixteen suspension and twelve guidance electromagnets. The suspension magnets also act as the synchronous magnets of the motor, and the distribution of these magnets along the entire length of the vehicle minimizes the force per cross-sectional area seen by the guideway, although the force is transmitted through the bolts holding the stator packs to the guideway. A feedback control system monitors and maintains the 8-10 mm air gap between the

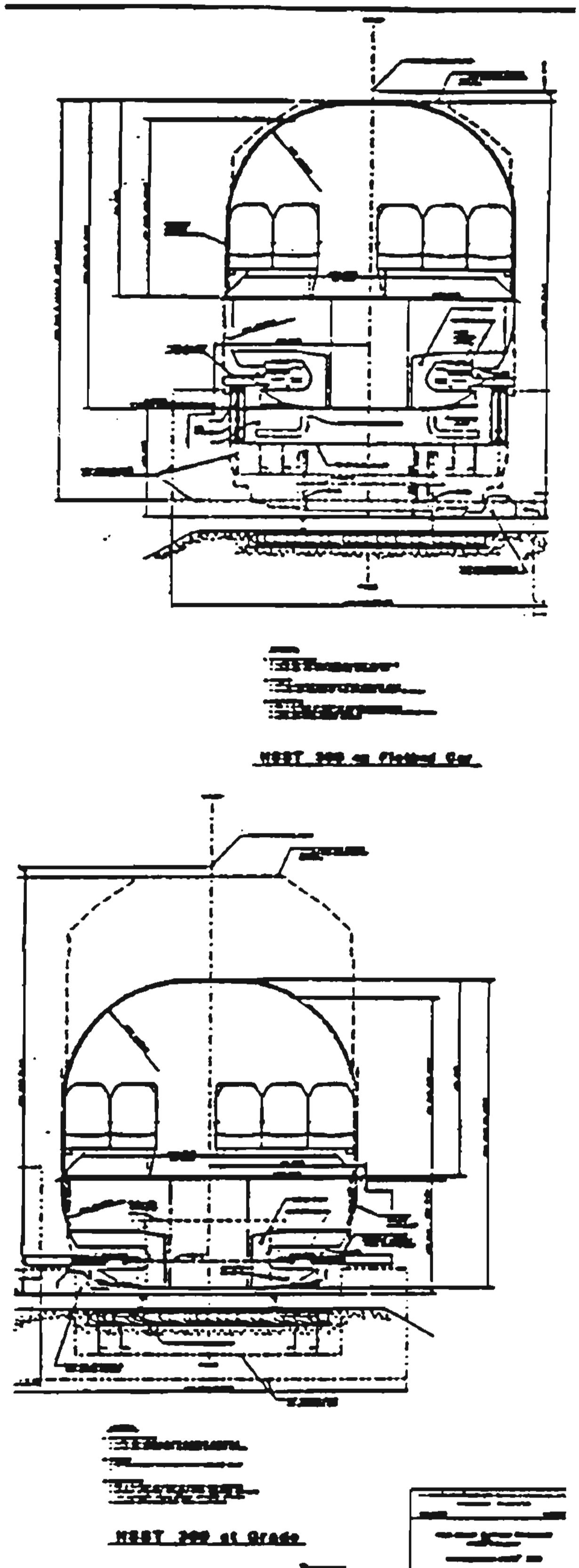

Figure B-5. Schematic of the HSST-300 configuration.

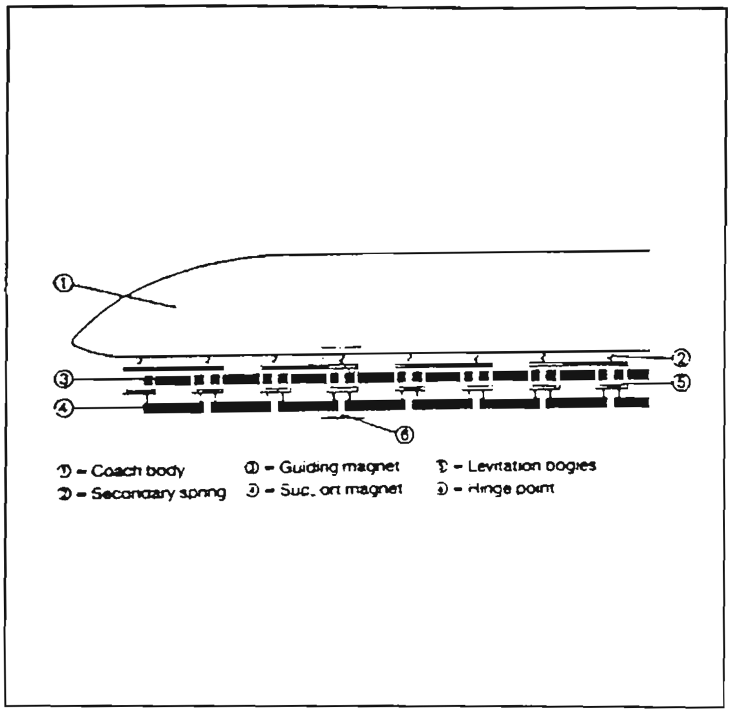

Figure B-6. Structure of the support and guidance of the TR-07.

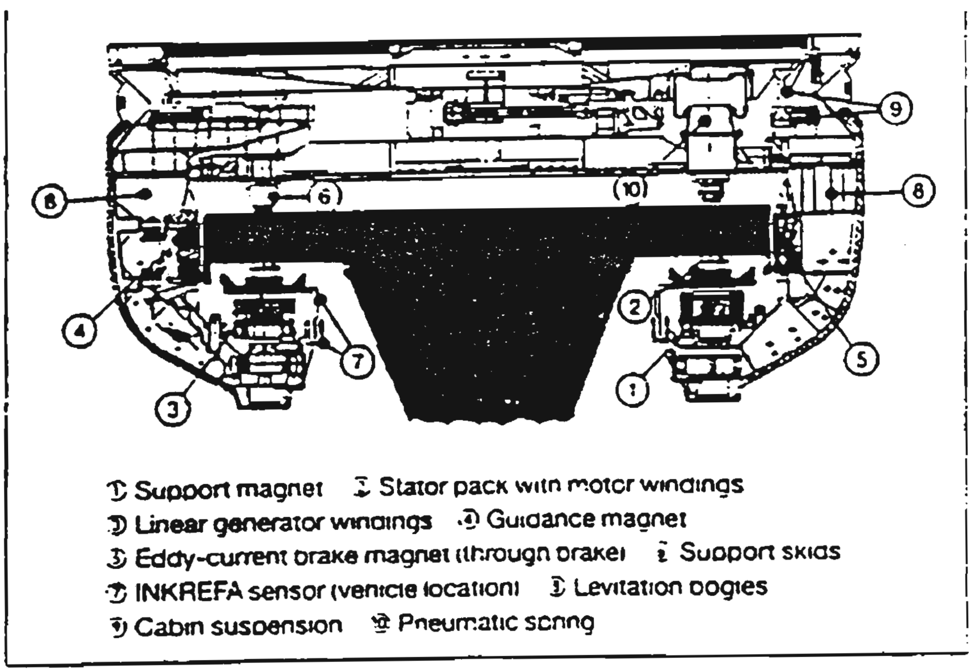

Figure B-7. Cross-section of the support and guidance of the TR-07.

vehicle's electromagnets and the guideway stator packs by modifying the current sent to the electromagnets. The vehicle shell utilizes aluminum and fiberglass to attain its high stiffness and low aerodynamic drag.

To follow the lateral and vertical irregularities on the guideway, the magnets along the length of the vehicle are connected together to form a chain-type arrangement. Each magnet is 3 m long, with 30 support magnets and 24 guidance magnets over the two vehicle sections. The support and guidance magnets are mounted on the bow of the levitation frame and are arranged to pivot relative to each other to form hinge points. The support magnets slide on lateral guides and are sprung laterally on the levitation frame, while the guidance magnets slide on vertical guides and are sprung vertically.

Ride quality is achieved via two separate systems. The stiff primary suspension system is provided by the eight support magnets located on both sides of the vehicle, while the soft secondary suspension is supplied by sixteen pneumatic springs mounted between the levitation bogie and the vehicle.

The Transrapid system uses three separate methods for braking the vehicle. The primary braking system involves reversing the current fed to the linear motor, thus producing a reverse thrust. The second braking method makes use of the electronic drag generated by inducing eddy currents in the guide rails via the guidance magnets. This method is only effective above speeds of approximately 50 km/h. During emergencies the train is slowed to near 50 km/h through the use of eddy currents and aerodynamic drag, at which time the power to the suspension magnets is removed and the vehicle settles onto the low-friction skids, which brings the train to a stop.

The guideway is comprised of T-shaped steel beams supported by concrete columns. Attached to these beams are several functional components, including a long stator motor, guidance rails, and low-friction skids. To minimize the cost of guideway construction and achieve the necessary very tight tolerances, a computer integrated manufacturing process has been implemented whereby the measurements taken at the construction site are input directly to the beam fabrication equipment.

Transfer from one guideway to another is accomplished using a bendable steel beam switch.

B.5 Bechtel

The Bechtel concept is a proprietary "flux canceling" EDS system. The vehicle contains six sets of eight superconducting magnets per side and straddles a concrete box-beam guideway. Interaction between these magnets and an aluminum ladder on each sidewall generates lift; similar interaction with null-flux coils provides guidance. Propulsion is by a sidewall-mounted LSM. The single-car vehicle has an inner tilting shell and uses aerodynamic control surfaces to improve ride quality. To avoid magnetic interactions, the upper portion of the guideway contains non-magnetic, fiber-reinforced plastic reinforcing rods. The switch is a bendable beam constructed entirely of fiber-reinforced plastic. A concern with the Bechtel concept is the possible interaction between the vehicle roll behavior and the torsional flexibility of the guideway. The guideway has a narrow cross-section, and consequently is relatively flexible in torsion. However, active suspension control (including active roll control) can potentially allow acceptable ride quality over such a guideway.

Features of the Bechtel concept include:

- A high efficiency EDS system that can suspend the vehicle down to very low speeds and reduce power consumption
- A box-beam guideway that reduces structural cost and environmental impact while providing a high degree of safety and longevity
- A linear motor propulsion system that provides high acceleration and braking and can operate at reduced speed in the presence of many types of failure
- An automated and fault tolerant control system that allows highly reliable fail-safe operation with short headway and high availability
- Use of air bearings for low speed stop/start in lieu of wheels, for emergency situations.

The baseline vehicle and guideway are shown in Figure B-8. The vehicle resembles the passenger compartment of a Boeing 737 but with more doors and larger aisles to facilitate rapid loading and unloading. Each vehicle can carry 106 passengers (or 120, depending upon the reference), is 36.1 m long, 4.1 m wide, 5.08 m high, and has a mass between 48.5 and 63.3 Mg depending on load. In normal operation, the vehicle can negotiate a 400-meter turn and operates in a unidirectional model.

Bechtel claims that their flux canceling EDS system design produces less magnetic drag than other EDS systems, and has the ability to provide full magnetic levitation and guidance down to 10 m/s. The guidance is provided by figure-eight coils on the guideway which are cross-connected to provide no guidance force when the vehicle is centered, but a strong restoring force if the vehicle deviates from the symmetrical position. This suspension and guidance system is totally passive so that as long as the vehicle is above the takeoff speed it is suspended and guided independent of the successful operation of any power source or active control system.

The vehicle uses an actively controlled secondary suspension system that creates forces between

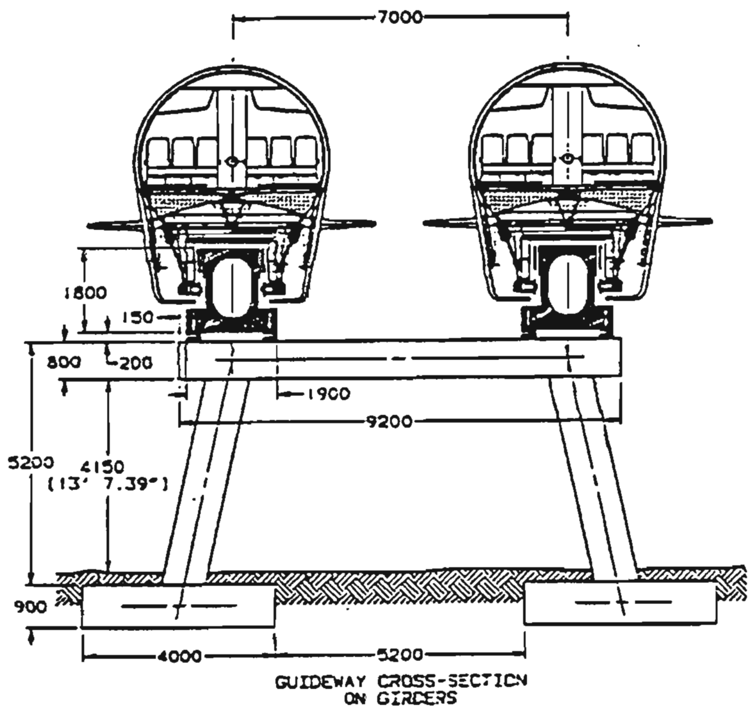

Figure B-8. Cross-sectional schematic of the Bechtel concept.

the magnetic suspension and the passenger-carrying part of the vehicle body. Additional control is provided by small winglets at the bow and stern. These surfaces are actively controlled to provide additional improvements in ride quality with only modest increase in aerodynamic drag. The secondary suspension mechanism also allows the vehicle to tilt up to 15 degrees relative to the guideway. Since the guideway itself may also be banked up to 15 degrees, a total vehicle bank angle of 30 degrees is possible in order to minimize lateral accelerations and the amount of speed change required to negotiate turns.

B.6 Foster-Miller

The Foster-Miller team designed an EDS system that uses superconducting magnets and sidewall-mounted null-flux coils in a configuration similar to that of the MLU-002. A "tilt body" car is proposed, as shown in Figure B-9. The proposed linear synchronous motor is locally commutated, applying power only in the vicinity of the vehicle rather than to a complete block. This system is called a "locally commutated linear synchronous motor" (LCLSM). The innovative LCLSM sequentially energizes individual propulsion coils in sync with the vehicle. Propulsion, and the primary guidance, are provided by a single set of coils, which are connected across the guideway and powered in parallel from the wayside.

The vehicles are configured as a consist of at least two cars, one of which houses the major equipment and operator and the other transports the passengers. The consist can be expanded to handle larger numbers of passengers. The vehicle consists of passenger modules with end bogies containing four magnets per side.

The baseline 150-passenger, 73-metric-ton, 2 car train is levitated on three bogies. Each bogie contacts eight magnets and must levitate 24.3 metric tons.

The U-shaped guideway has two parallel, post-tensioned concrete beams joined transversely by precast concrete diaphragms. To avoid magnetic interactions, the upper post-tensioning rods are fiber-reinforced plastic. The prefabricated guideway is designed to be open on the bottom to avoid the accumulation of debris, snow, and ice. The high-speed switch uses switched null-flux coils to guide the vehicle through a vertical turn-out; it requires no moving structural members. A concern with the Foster-Miller concept relates to the ride quality, and the ability to produce guideway geometry necessary to achieve acceptable ride quality.

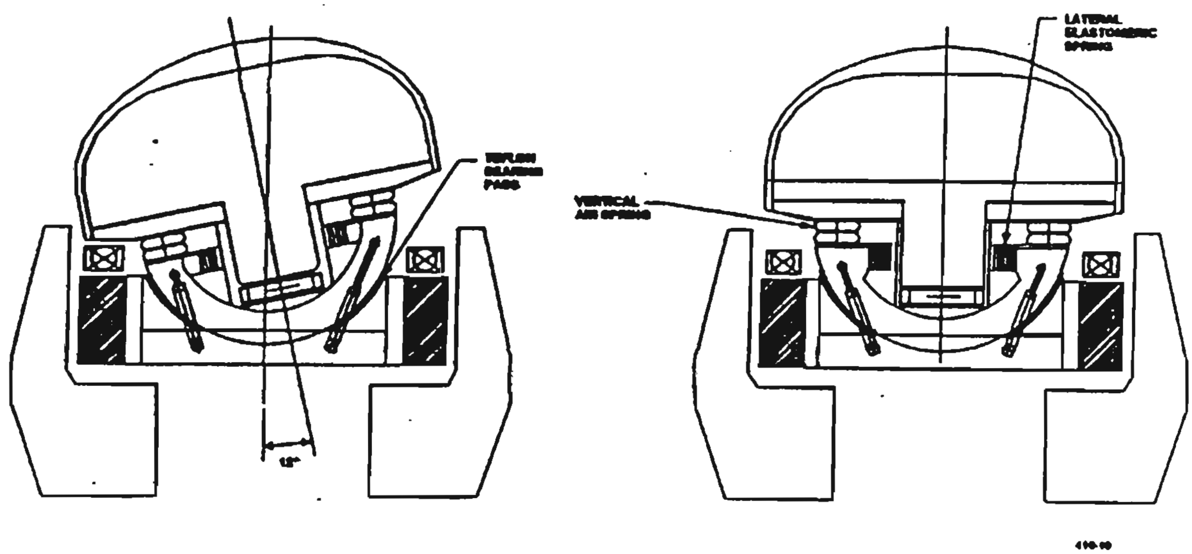

Figure B-9. Tilting mechanism of the Foster-Miller design.

B.7 Grumman

The Grumman Maglev design is an EMS system using constant-current superconducting magnets with similarities to the TR-07 design. However, Grumman's vehicle wraps around a Y-shaped guideway and uses magnet modules that are canted along the vehicle to provide simultaneous guidance and levitation. Propulsion is by a conventional linear synchronous motor.

Figure B-10 illustrates the Grumman Maglev concept. The figure shows a cross section of the vehicle with the iron core magnets and guideway rail (identified in black). The laminated magnets and iron rail are oriented in an inverted "V" configuration with the attractive forces between the magnets and rail acting through the vehicle's center of gravity. Vertical control forces are generated by sensing the gap clearance on the left and right side of the vehicle and adjusting the currents in the control coils to maintain a relatively large 4 cm gap between the iron rail and the magnet face. Lateral control is achieved by differential measurements of the gap clearance between the left and right sides of the vehicle magnets. The corresponding magnet control coil currents are differentially driven for lateral guidance control. There are 48 magnets, 24 on each side of a 100 passenger vehicle. In this manner control of the vehicle relative to the rail can be achieved in the vertical, lateral, pitch, and yaw directions. Vehicle roll control is achieved by offsetting each magnet in a two magnet module by 2 cm to the left and ride side of a 20 cm wide rail. Control is achieved by sensing the vehicle's roll position relative to the guideway and differentially driving the offset control coils to correct for roll errors. The total number of independent control loops required for a complete 100 passenger vehicle control is 26 (1 for each of 24 magnet modules and 2 for roll control).

Vehicles may be single- or multi-car consists. The Grumman design has provided the capability of tilting the vehicle passenger compartment by ± 9 deg relative to the guideway. The design will allow for coordinated turns up to ± 24 degree, banking (± 15 deg in the guideway and ± 9 deg in the vehicle). This capability allows for coordinated turns to be performed at the appropriate tilt angle independent of the speed that the vehicle is traversing the turn, as well as allowing for high-speed off-line switching.

An innovative spine girder supports two Y-shaped guideway sections. Switching is with a TR-07-style bending guideway beam.

The Grumman concept requires a single set of magnets to provide both lift and guidance. These are two separate functions that in general have different control response characteristics. A concern is the force-capability of the suspension. The suspension travel must be adequate for the range of guideway perturbations that the vehicle may encounter.

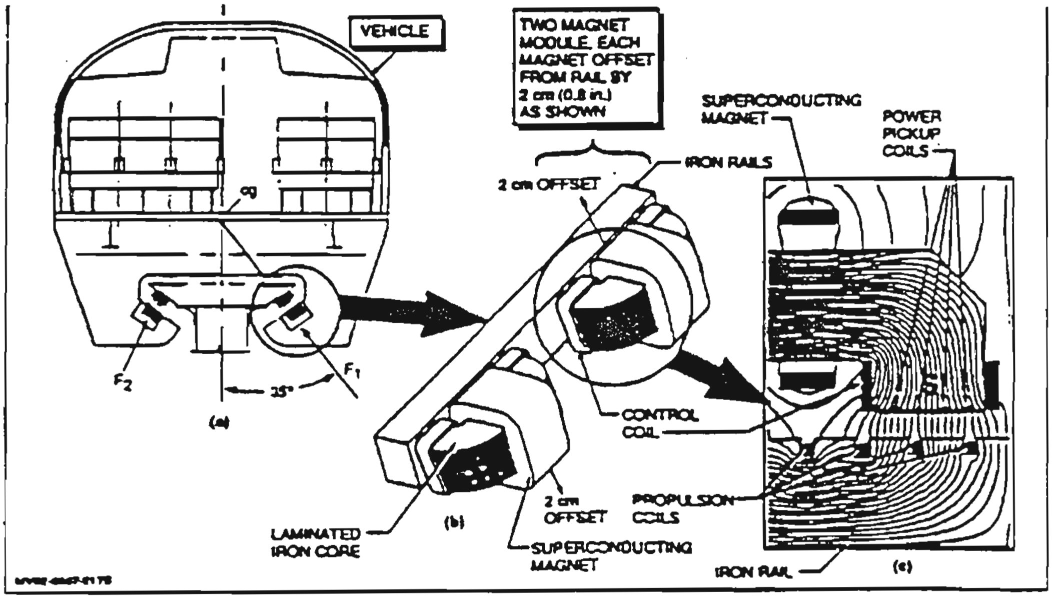

Figure B-10. **Grumman vehicle-guideway design showing magnet configuration.**

B.8 Magneplane

The Magneplane concept is a single-vehicle EDS using a trough-shaped aluminum guideway supporting a vehicle with two saddle shaped arrays of levitation and propulsion magnets. The vehicle and guideway cross-section is shown in Figure B-11. The levitation magnets are tilted 35 degrees with respect to the horizontal plane providing a center of lift in the vehicle above the center of mass. This design provides for a naturally stable configuration. Centrifugal forces tilt the "magplane" into coordinated turns. Front and rear bogies contain superconducting levitation and propulsion magnets. Centerline magnets interact with linear synchronous motor windings for propulsion and also generate electromagnetic guidance forces. Side magnets react against the aluminum guideway sheets to provide levitation. Magneplane uses aerodynamic control surfaces and linear synchronous motor phase control to dampen vehicle motions. The guideway sheets form the tops of two structural aluminum box beams supported directly on piers. The high-speed switch uses switched null-flux coils to guide the vehicle through a turn-out; it requires no moving structural members.

The Magneplane system uses a 0.15 m levitation gap lowering the natural frequencies for pitch and heave motion to less than 2 Hz. This suspension has very low natural damping and must be