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~~G Kachadourian~~

COMPARATIVE ANALYSIS OF DYNAMICS OF FREIGHT AND PASSENGER RAIL VEHICLES

NOVEMBER 1976

SUMMARY REPORT

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16. Abstract

During the course of this contract, analytical studies have been conducted on several different rail vehicles typical of North American railroad operations. Mathematical models, and computer codes for the mechanization of these models, have been generated under this contract to provide a predictive methodology for determining vehicle/track dynamic interaction under a range of conditions. These models have been validated through comparison of results with test data from several of the rail vehicles.

Results of a comparative analysis of nine different rail vehicles were presented in a Final Report (FRA-ORD&D-74-39) dated March 1974. In subsequent contract modifications, the analytical techniques were used to provide support to the Metroliner Ride Improvement Program (DOT-FR-20049). Both the vehicle ride comfort and truck hunting stability were investigated by means of computer simulation. Parameter variation studies were used to determine optimum suspension values for both ride comfort and hunting stability.

This report provides a summary of work conducted under the original contract and five subsequent modifications. Recent modifications to the mathematical models to improve simulation accuracy are discussed. In addition, improvements in the representation of track geometry in power spectral density format based on recent measurements are discussed.

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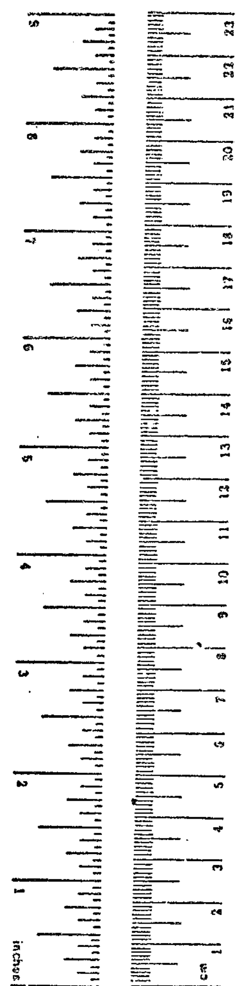
METRIC CONVERSION FACTORS

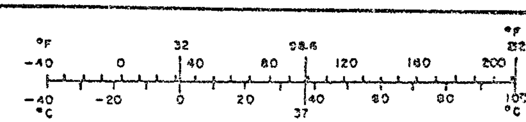
Approximate Conversions to Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
in	inches	2.5	centimeters	cm
ft	feet	30	centimeters	cm
yd	yards	0.9	meters	m
mi	miles	1.6	kilometers	km
		AREA		
in^2	square inches	6.5	square centimeters	cm^2
ft^2	square feet	0.09	square meters	m^2
yd^2	square yards	0.8	square meters	m^2
mi^2	square miles	2.6	square kilometers	km^2
	acres	0.4	hectares	ha
		MASS (weight)		
oz	ounces	28	grams	g
lb	pounds	0.45	kilograms	kg
	short tons (2000 lb)	0.9	tonnes	t
		VOLUME		
tsp	teaspoons	5	milliliters	ml
Tbsp	tablespoons	15	milliliters	ml
fl oz	fluid ounces	30	milliliters	ml
c	cups	0.24	liters	l
pt	pints	0.47	liters	l
qt	quarts	0.95	liters	l
gal	gallons	3.8	liters	l
ft^3	cubic feet	0.03	cubic meters	m^3
yd^3	cubic yards	0.76	cubic meters	m^3
		TEMPERATURE (exact)		
°F	Fahrenheit temperature	5/9 (after subtracting 32)	Celsius temperature	°C



Approximate Conversions from Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
mm	millimeters	0.04	inches	in
cm	centimeters	0.4	inches	in
m	meters	3.3	feet	ft
m	meters	1.1	yards	yd
km	kilometers	0.6	miles	mi
		AREA		
cm^2	square centimeters	0.16	square inches	in^2
m^2	square meters	1.2	square yards	yd^2
km^2	square kilometers	0.4	square miles	mi^2
ha	hectares (10,000 m^2)	2.5	acres	
		MASS (weight)		
g	grams	0.035	ounces	oz
kg	kilograms	2.2	pounds	lb
t	tonnes (1000 kg)	1.1	short tons	
		VOLUME		
ml	milliliters	0.03	fluid ounces	fl oz
l	liters	2.1	pints	pt
l	liters	1.06	quarts	qt
l	liters	0.26	gallons	gal
m^3	cubic meters	35	cubic feet	ft^3
m^3	cubic meters	1.3	cubic yards	yd^3
		TEMPERATURE (exact)		
°C	Celsius temperature	9/5 (then add 32)	Fahrenheit temperature	°F



PREFACE

This report was prepared by Battelle's Columbus Laboratories under Contract DOT-FR-20077 for the Office of Passenger Systems of the Federal Railroad Administration, Washington, D. C. The original objective of the contract was to provide a technical background through analytical studies for modification to the Track Safety Standards in terms of freight and passenger train speed limits. In subsequent modifications to the contract, the objectives were expanded to provide technical support to the Metroliner Ride Improvement Program and other areas of rail vehicle dynamics. This report summarizes the work conducted under the original contract and modifications.

Mr. Richard Scharr was technical monitor during the preparation of this summary, and his cooperation and suggestions are gratefully acknowledged. In addition, Battelle wishes to acknowledge the contributions of ENSCO, Inc., in the form of track geometry and vehicle acceleration power spectral density data.

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1. INTRODUCTION

In 1972 Battelle's Columbus Laboratories were awarded a contract (DOT-FR-20077) for a study entitled "Comparative Analysis of Dynamics of Freight and Passenger Rail Vehicles" to provide technical background in support of the High-Speed Ground Transportation Alternatives Study. The emphasis in this study was on the analysis of vehicle and track interactive dynamics where several types of trains are required to operate on the same tracks at different speeds. Results of this study were summarized in the Final Report of March, 1974(1).

Subsequent modifications to the original contract were awarded to support the Metroliner Ride Improvement Program and the Improved Passenger Train Program through analytical studies in vehicle stability and ride quality. This work is reported in the Interim Report of June, 1975(2).

The purpose of this Summary Report is, therefore, to provide a comprehensive review and summary of the work conducted under the original contract and four subsequent modifications, to review the current status of the analytical models and techniques used, and to recommend areas in need of further research in the analysis of vehicle/track dynamics.

2. SUMMARY

During the course of this contract (Comparative Analysis of Dynamics of Freight and Passenger Rail Vehicles, DOT-FR-20077, 1972-1976), analytical studies have been conducted on a range of different rail vehicles typical of North American railroad operations. Mathematical models, and computer codes for the mechanization of these models, have been generated during this contract period to provide a predictive methodology for determining vehicle/track dynamic interaction under a range of conditions. The following studies were conducted under this contract:

(1) Nine representative rail vehicles in present or proposed use on the nation's railroads were analyzed and rank-ordered on the basis of ride quality, stability, and track forces(1).

(2) Ride quality of the Improved Metroliner with SIG trucks was analyzed, and a suspension parameter variation study was conducted to optimize the ride quality (in the critical 1-10 Hz range) and track forces(2).

(3) Secondary (truck) hunting stability of the Improved Metroliner with SIG trucks was analyzed, and the effects on critical speed of suspension parameter variations were determined(2).

(4) A computer analysis of the wheel/rail forces generated at high speed by the LIM Research Vehicle was performed using track geometry power spectra from measurements at the Transportation Test Center in Pueblo, Colorado(3).

(5) A time-domain, nonlinear computer model of the Improved Metroliner was programmed on Battelle's hybrid computer to study ride quality and wheel/rail forces resulting from transient geometry perturbations in the track and from bolted-rail track with staggered joints(4).

(6) An analysis of forces and L/V ratios generated during curving by 6-axle locomotives was conducted using both AAR and Battelle computer models(2).

The original Work Statement for Contract DOT-FR-20077 suggested a model of "one or two degrees of freedom". Critical evaluation of the work summarized in the Final Report suggested that the fourteen degree of freedom model was too simplistic. Consequently, a vehicle with forty-six degrees of freedom for a complete rail vehicle (two 2-axle trucks) was derived mathematically and programmed in the frequency domain. In addition, a nineteen degree of freedom model of a single truck was developed and programmed. Selected preliminary results showed that the fourteen degree of freedom model compared well with the forty-six degree of freedom model for acceleration and wheel/rail force power spectral density (PSD) outputs.

Since a frequency domain solution is (by definition) linear only, a separate nonlinear time-domain model was developed for the hybrid computer[4]. Nonlinearities such as wheel lift, flanging, hardening springs and suspension stops, damper force limits, and friction were included. This program was used to compare the present Metroliner and the Improved Metroliner design. Superior response in terms of ride quality was predicted by computer runs for the improved truck design, particularly for staggered-joint, bolted-rail track in the 60 mile-per-hour range, and in transient response to track alignment errors. Two problems were noted from the computer runs: first, with the limited travel provided by the air springs of the secondary suspension, contact with suspension stops could be expected under some conditions of maximum track geometry errors and train speeds allowed under the Track Safety Standards. Second, the low roll stiffness of the SIG truck could result in large roll excursions during low-speed car rocking, particularly on cropped rail with staggered joints. The need for an auxiliary roll stiffness (roll bar) was recognized; and the superior ride quality was borne out by measurements during comparative test runs of standard and Improved Metroliner cars.

An extensive evaluation of the truck stability for the Improved Metroliner with SIG trucks was conducted using a linear, seven degree of freedom truck hunting model. Truck parameters were varied over a wide range, and the degree of sensitivity with relation to the hunting critical speed was established for each. The analysis confirmed the stability studies by LTV that the critical speed of the SIG truck configuration was well above 150 mph.

Finally, algebraic equations for steady-state curving were formulated and nomographs constructed to allow quick evaluation of vehicle safety (overturning) on curves as a function of speed, curvature, superelevation, c.g. height, and c.g. lateral shift. Steady-state curving forces (including the lateral and longitudinal creep and flanging forces) and lateral to vertical (L/V) force ratios were investigated for both 2- and 3-axle trucks by use of a linear steady-state curving simulation of a multi-degree of freedom truck[2]. Flanging and wheel slip were handled in this program by a two-regime iterative solution.

3. CONCLUSIONS

The accuracy of analytical results reported in this comparative analysis program is dependent on three fundamental areas:

(1) An adequate vehicle/track model, in terms of simulating the important degrees of freedom for the type of analysis undertaken.

(2) An accurate description of the vehicle parameters, such as suspension damping, kinematic constraints, nonlinearities, mass moments of inertia, etc.

(3) An accurate and realistic representation of track geometry (or "synthesized input").

During the course of this contract, the vehicle/track models have undergone a continual evolution to provide a more accurate and realistic predictive tool. The linear, frequency-domain (random vibrations) model of the vehicle and track has been improved and validated by comparison with field test data generated under similar conditions. This model is valid on "good" track geometry where the linear range of the mathematical model is not exceeded. For transient response to large-amplitude perturbations in track geometry where nonlinear behavior can dominate (wheel lift or hard contact with suspension stops, for example), a nonlinear, time-domain model or quasi-linear (describing function) frequency-domain model must be used. Such phenomena as the nonlinear "jump" resonance noted in freight car rocking can only be analyzed with the nonlinear model.

The truck hunting stability analysis conducted under this contract was performed using a linear model to calculate the eigenvalue/eigenvector solution at a fixed speed, from which the critical speed (point of neutral stability) was estimated. Small-amplitude oscillations about an "operating point", with linear values of wheel/rail contact geometry and creep coefficients, were therefore assumed.

The "working" model of the rail vehicle and track was limited in the number of simulated degrees of freedom to keep computer running times (and costs) reasonable. Central processor (CP) time has been estimated (from

Battelle experience) for the linear-frequency-domain model to increase by the ratio of degrees of freedom to the 2.5 power. The relative importance of the different degrees of freedom depends on the specific desired result: for example, the vertical body bending and truck frame resonances in the Metroliner were important to ride quality. Body bending, on the other hand, is generally unimportant in the wheel/rail load calculation, unless high-amplitude, low-frequency oscillations from an extremely limber body occur. Rigid car body modes (roll, bounce, pitch, etc.) are important both from the ride comfort and track load viewpoint.

An accurate description of the vehicle parameters is vital to the overall accuracy of analytical results. Some vehicles (for example the 100-ton hopper car and the Metroliner) were very well defined from previous test and analytical programs. Damping values were particularly difficult to obtain for most of the representative vehicles, and for the most part were estimated from past experience with rail vehicles. Hydraulic dampers used with rail vehicles typically have a velocity-dependent force limit, a nonlinearity that results in higher force and acceleration levels at the higher frequencies than occur in the field. Mass moments of inertia of the individual components were also seldom known and therefore estimated from weights and approximate distributions.

Finally, the synthesized inputs (estimated track geometry power spectra) can be a major source of error in comparing computer-generated results with actual test data. The "Class 6" track cited in the Final Report[1] was based on an approximation of geometry PSD's generated during the Metroliner Ride Improvement Program (DOT-FR-10035). It should be termed a "worst case" Class 6 PSD, rather than a nominal Class 6, in view of more recent geometry measurements from both the Northeast Corridor track and track in the Pueblo, Colorado area. For example, the effects on vertical wheel load of the simulated Metroliner due to track surface power spectra are shown in Figure 3-1. The NEC "Class 6" is seen to be rougher than the Colorado "Class 4" track.

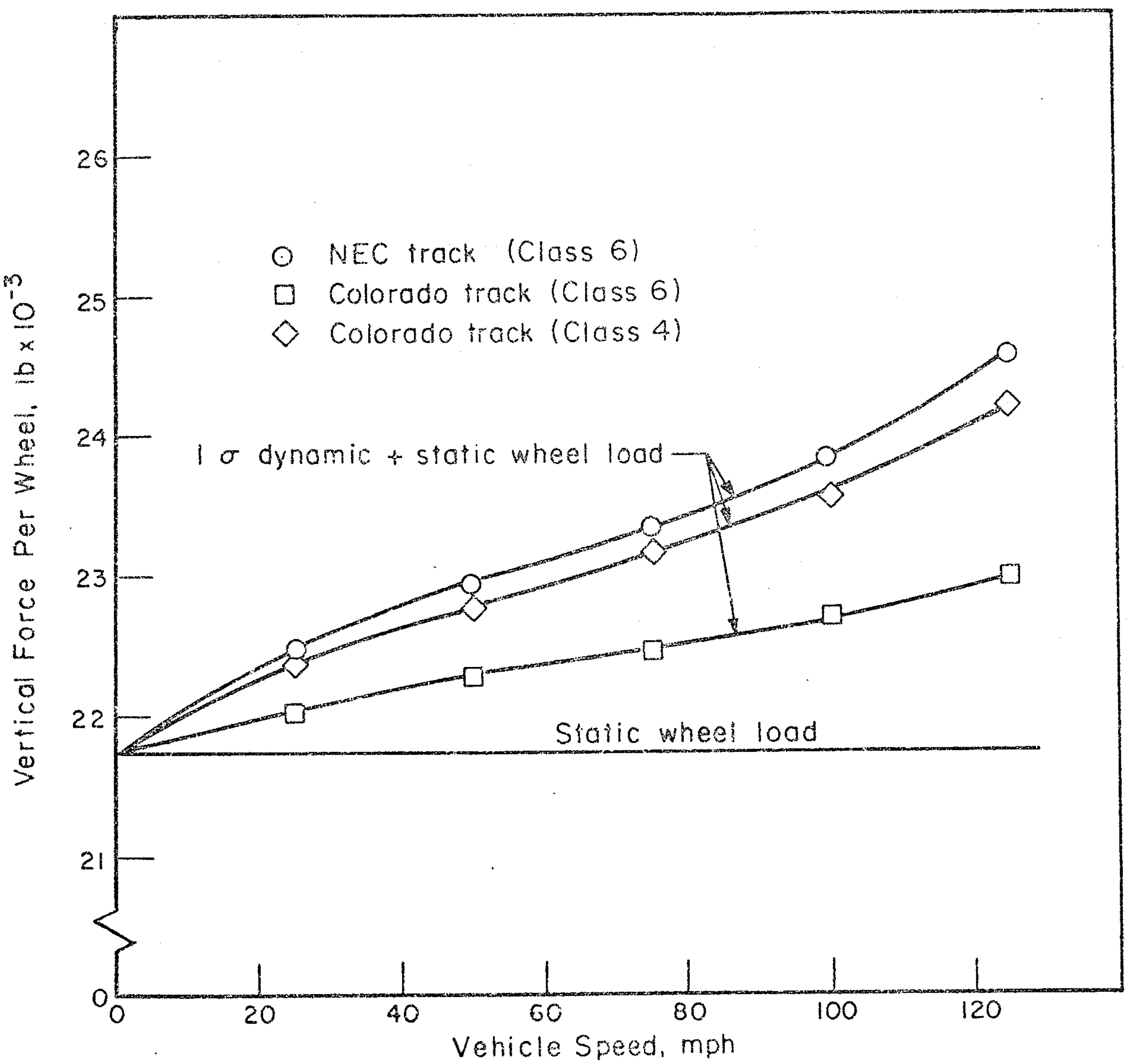

FIG. 3-1. EFFECT OF SPEED ON STATIC PLUS 1σ DYNAMIC VERTICAL WHEEL LOAD FOR STANDARD METROLINER – RANDOM (PSD) SURFACE INPUTS FROM SIMULATED STANDARD CLASS 6 TRACK AND CLASS 4 TRACK

4. RECOMMENDATIONS FOR FUTURE RESEARCH

Based on the results to date accomplished under this comparative analysis program and other related programs sponsored by the Federal Railroad Administration, the following specific recommendations are made for further research in vehicle/track dynamic analysis:

(1) Random vibration models are seen as an important tool in the prediction of vehicle/track dynamic interaction, and these models should be developed more fully. Only a few selected runs have been made to date comparing the large (forty-six degree of freedom) model with the more limited-sized linear models. A systematic check of the importance of specific degrees of freedom must be made, so that the optimum model (in terms of detail versus computation costs) may be determined.

(2) Improved approximations of the important system nonlinearities by describing functions or other techniques should be investigated and incorporated where possible into the random vibrations model.

(3) An updated version of the nonlinear, time-domain vehicle/track model (including the important wheelset and truck frame yaw modes) should be programmed and validated on the hybrid computer facility at the Transportation Systems Center, Cambridge, Massachusetts. A working model of a 6-axle locomotive including the important nonlinearities could, over the past several years, have been invaluable in the investigation and evaluation of locomotive ride and tracking problems experienced by AMTRAK. The nonlinear, time-domain program is particularly useful in predicting the transient response to singular perturbations such as line and cross level errors at joints and switches.

(4) A nonlinear steady-state curving program for 3-axle trucks can be an extremely useful tool for determining curving forces and L/V ratios. More effort is needed to develop a converging algorithm to find the equilibrium configuration of the 3-axle truck.

(5) The statistical approach to defining track geometry needs further research as measurements of different track structures become available. Track geometries at wavelengths greater than 100 ft and shorter than 10 ft are still poorly defined in terms of power spectra.

(6) There are several ongoing large-scale testing programs to verify modeling techniques developed under DOT/FRA sponsorship. These programs should be monitored to determine how the measurements and data formats may best be used to validate and improve the existing vehicle/track models.

(7) Work should continue on the specification of track geometries by track class in power spectral density format (including the spectral peaks due to singular wavelengths). Track geometry PSD's should be developed for different track structures (wood tie versus concrete tie track, for example), as well as the generalized PSD for the track safety classes.

5. TECHNICAL DISCUSSION

5.1 Review of Linear Vehicle/Track Models

The comparative analysis of rail vehicles conducted under this contract has depended primarily on the use of mathematical models of the vehicle/track dynamic system. A mainstay of this analytical modeling has been the linear, lumped-parameter vehicle/track model programmed for digital computer solution. During the course of the past five years, this model has been in a continual evolution to improve simulation accuracy and to provide a wider range of useful output data formats.

The equations of motion for the system of springs, dampers, and masses representing the vehicle/track model were written, then modified by LaPlace transform techniques to a set of algebraic equations in the frequency domain. This transformation of the original differential equations was then set up in matrices of real and imaginary components and programmed for solution on the digital computer by matrix inversion and multiplication by an input (track geometry) matrix. Amplitude response of system variables (accelerations, displacements, forces, etc.) was then calculated versus frequency for the given inputs of track geometry surface, alignment, and cross level.

A brief review of the several stages of evolution of this linear, frequency-domain vehicle/track model is given below.

5.1.1 Phase I (1972-1973)

5.1.1.1 Mathematical Model. A simplified, linear lumped-parameter model was generated to represent several rail vehicles for the first phase of this study. This model consisted of two masses with six degrees of freedom: one representing the unsprung mass of the truck (more accurately, that mass below the secondary suspension) in vertical, lateral, and roll motions; and the other mass representing one-half the car body, also in vertical, lateral, and roll motions. A sketch of this six degree of freedom model is shown in Figure 5-1. This model was soon expanded to a nine degree of freedom model to represent the sprung

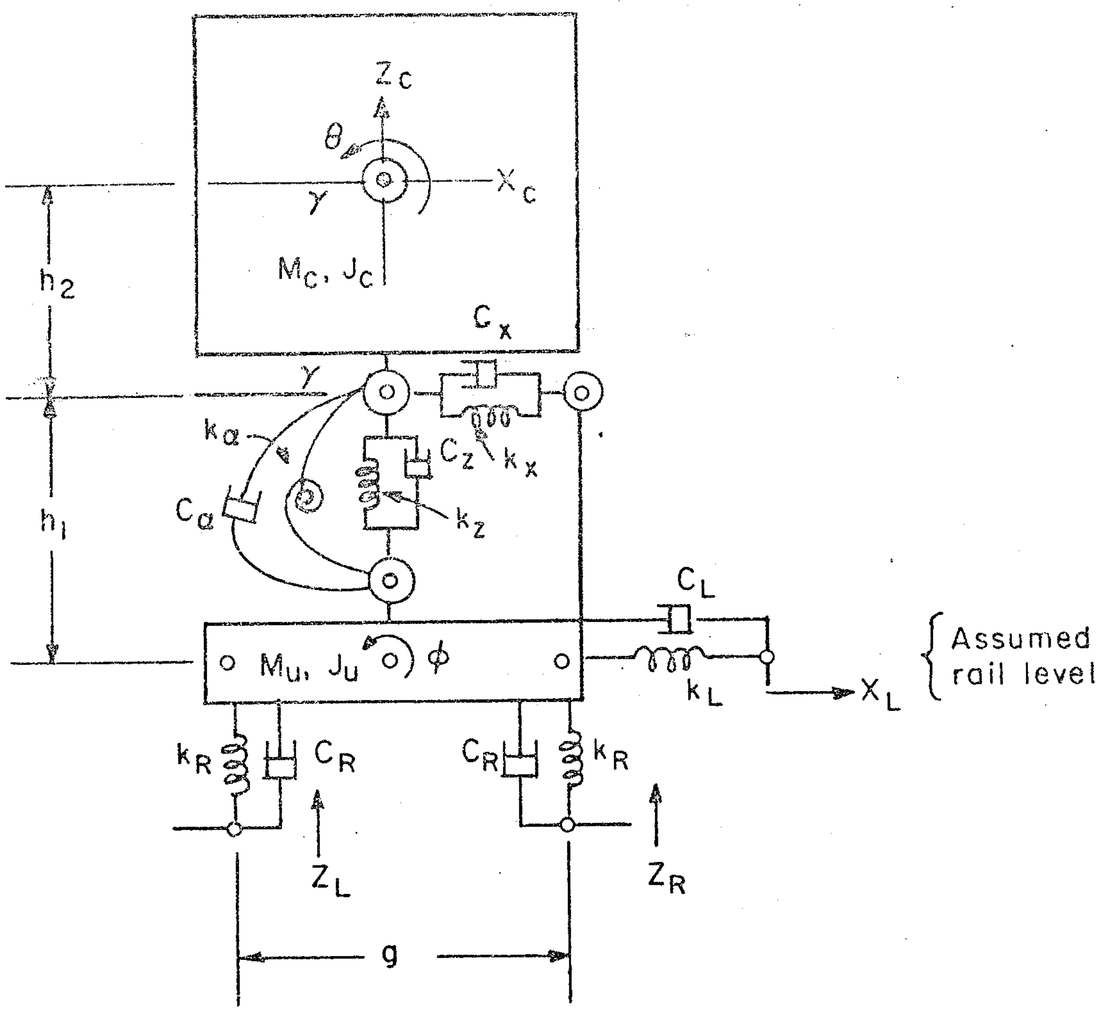

FIG. 5-1. SIMPLIFIED, LINEAR MODEL OF RAIL VEHICLE

masses of the car body and truck frame, and the unsprung mass of the wheelsets in vertical, lateral, and roll motions.

5.1.1.2 Model Limitations and Assumptions. The simplified model used in this first phase was aimed primarily at the examination of low-frequency rigid body modes of vibration: car rocking and bounce. At this point car body bending and torsional vibration modes were ignored, as well as the pitch and yaw modes of the rigid body. Also the more complex vibrations of the truck masses (including truck hunting) were not considered.

A limitation to the simplified model was the linearization of a more-or-less nonlinear dynamic system. Several basic nonlinearities are present to some degree in all vehicle suspension systems; these are typically sliding (Coulomb) friction, hard stops (bumpers), hysteretic damping (elastomers), and variable system geometry (side bearing contact, wheel lift, etc.) These elements were considered as linear approximations or ignored completely. If the system is fairly linear within the normal operating range, the approximation is good (for example, the Metroliner); if, however, the system operates well beyond the linear range (as does the 100-ton hopper car during the car rocking condition), the approximation is less satisfactory.

5.1.1.3 Validation of Model. Results from this program were compared with field test data from the 100-ton hopper car, and good correlation was noted for system natural frequencies and for magnitudes of roll angle and track forces. In July of 1972 a second version of the model was developed which would accept a power spectral density (PSD) representation of the track as an input to the model, and generate outputs in PSD form also. Results were compared with data generated during the Metroliner tests (DOT-FR-10035) conducted by the Budd Company, and the results correlated quite well.

5.1.2 Phase II (1973-1974)

5.1.2.1 Mathematical Model (Program TRKVPSD). To provide a more complete simulation of the rail vehicle, the model was then expanded to a total of fourteen degrees of freedom[1]. Since linear relationships were assumed, the vertical and roll/lateral portions of the model were considered separately as two seven degree of freedom models. In the vertical, Figure 5-2, rigid-body bounce and pitch motions of the car body were modeled, as well as the first body-bending mode (modeled as a free-free beam) and the vertical motion of a suspended mass (transformer). Vertical motions of both the sprung (truck frame, etc.) and unsprung (wheels, axles, etc.) masses of the front truck were modeled. Since the truck frame dynamics may influence the body-bending oscillations of some vehicles, the rear truck frame vertical motion was retained as a separate degree of freedom.

Wheel/rail contact resonances are well above the assumed bandwidth of the model (0.2 to 51 Hz), therefore, the effective mass of the track structure at the front truck was lumped with the truck unsprung mass, and the track itself taken to be a simple spring-damper impedance. Consequently, the track geometry input was applied at the point normally considered to be "ground", rather than the more precise wheel/rail interface. Within the bandwidth of the computer model, wheel/rail forces predicted by this simplified model are the same as for a more complicated model.

The roll/lateral model, shown in Figure 5-3, considered the car rigid-body motions: roll, yaw, and lateral translation. Sprung and unsprung masses at the front truck were modeled, each in roll and lateral translation. Since truck yaw and pitch modes were not considered, a track input attenuation function based on track geometry wavelength and truck axle spacing was used to account for the effective chordal transfer function provided by the typical side-frame type of truck design. Resulting track forces represented the averaged sum per truck, per rail. Truck dynamics at the rear of the car were not considered, based on the car body and secondary suspension acting as a low-pass filter, so that relatively high-frequency truck dynamics at the rear would have negligible effect at the front truck. An effective overall spring rate and damping at the rear truck was calculated from the appropriate set of differential equations by allowing the truck masses and mass moments of inertia to approach a zero value. Track geometry disturbances, properly phase-related to inputs at the front truck, were then applied to the rear of the car through this effective impedance. The secondary, primary-truck, and track structure impedances in series result in

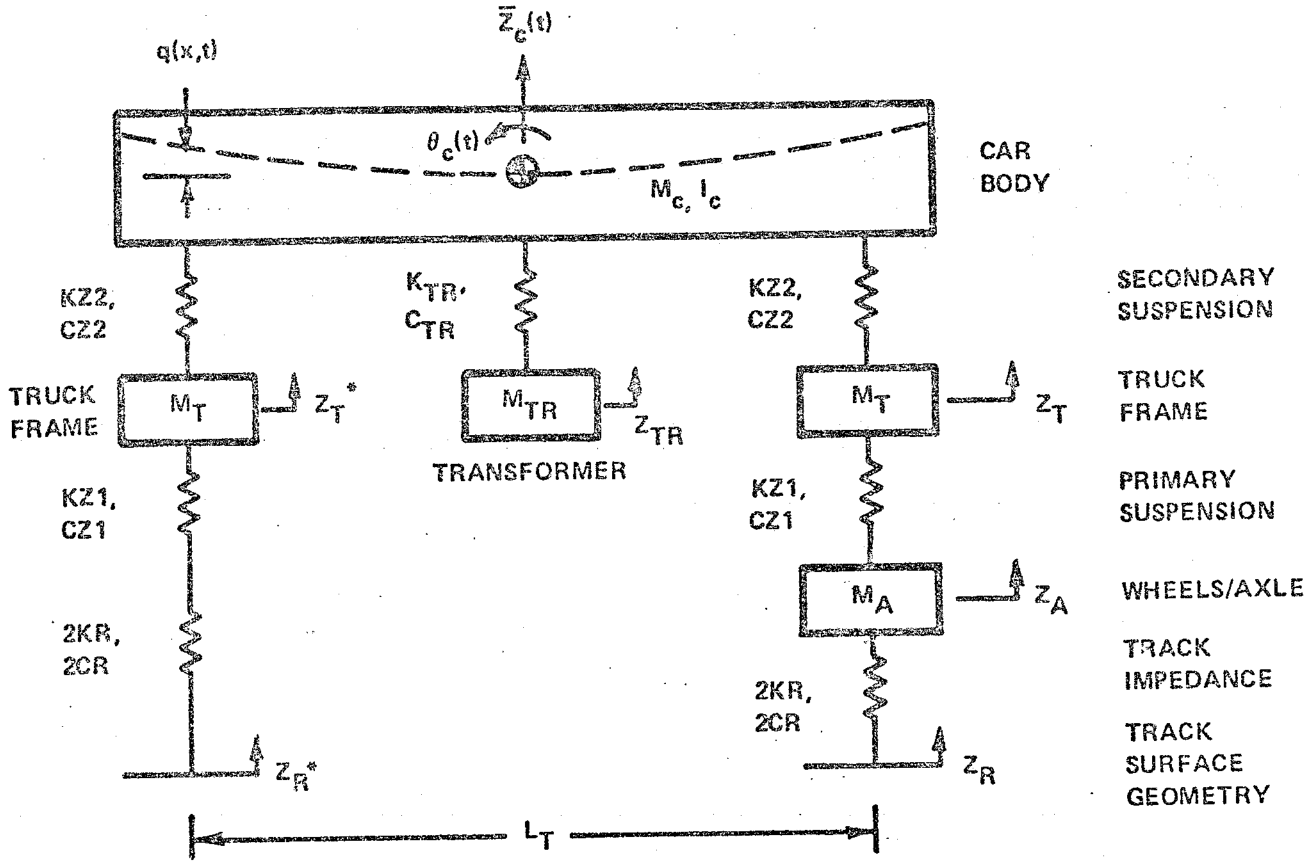

FIGURE 5-2. VERTICAL (PITCH/BOUNCE) MODEL OF VEHICLES AND TRACK STRUCTURE

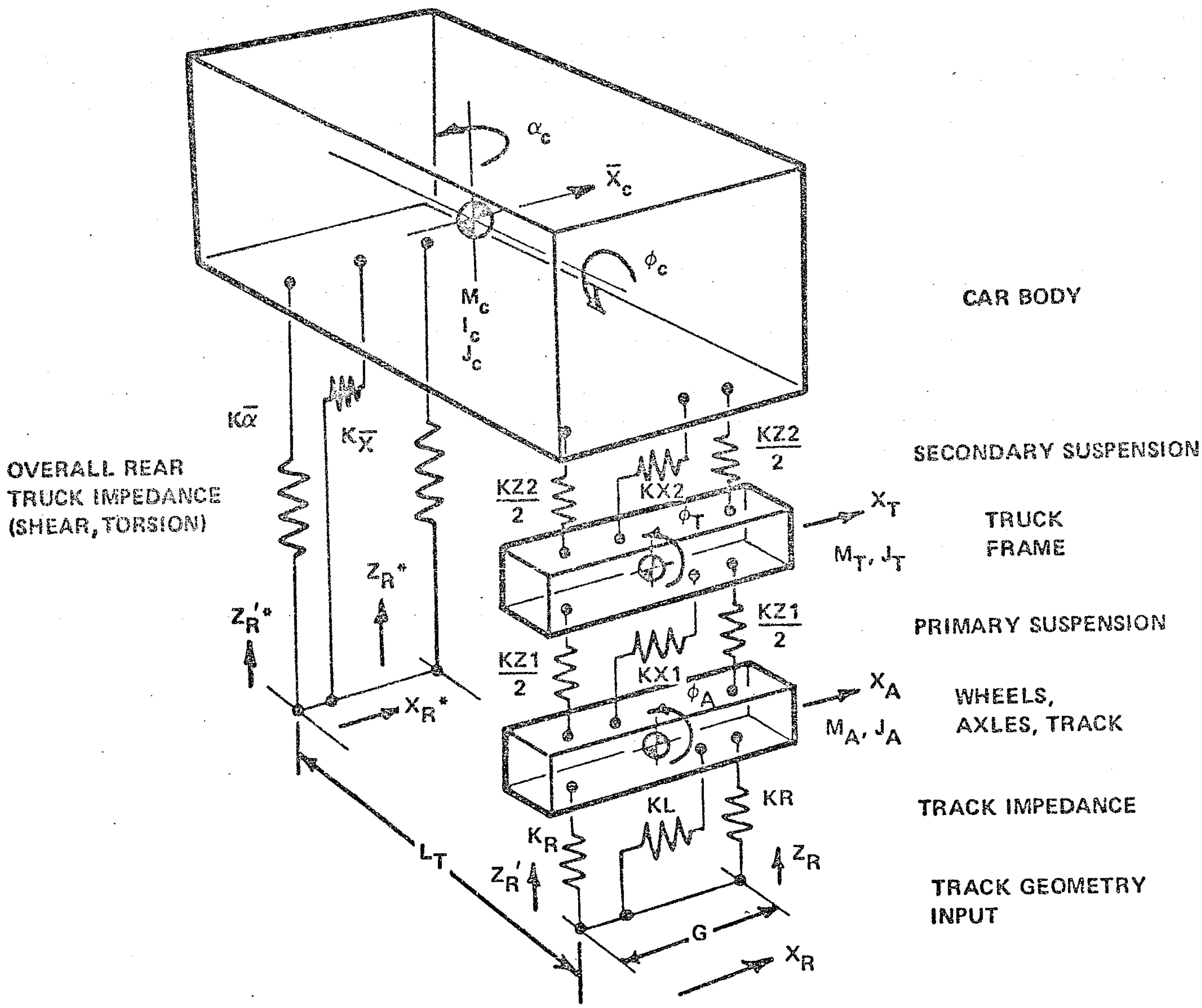

FIGURE 5-3. ROLL/LATERAL/YAW MODEL OF VEHICLE AND TRACK STRUCTURE

frequency-dependent stiffness and loss terms, analogous to an elastomer. A check of these terms showed small percentage changes over the frequency range of interest, however.

Relatively small angular motions were assumed in the model to maintain geometric linearity, and all dynamic effects (springs, dampers) were assumed to be linear. The effect on roll of lateral translation of the car body center of mass was included in the equations of motion as a "negative" torsional spring rate dependent on the car body weight and the height of the center of mass above the secondary suspension. This resulted in reduced secondary torsional stiffnesses of up to 10 percent.

Two of the vehicles studied have swing hanger trucks, an arrangement that interposes a four-bar linkage between the primary and secondary suspensions for improved stability and reduced lateral stiffness. Again assuming relative small motions, the swing hanger was treated as an effective lateral stiffness (and damping), ignoring the relatively small vertical component of motion due to lateral translation. This stiffness may be calculated by treating the car body as a pendulum(5), or determining the apparent increase in stored energy(6).

The Turbo Train suspension system presented a more complex set of equations than the other vehicles studied. Equations of motion for this system were developed for the Sikorsky Aircraft Division by Dr. D. E. Newland(7), representing the car body in vertical, lateral, and roll motions, with the truck frame providing a motion input. The resulting linearized three-by-three matrix of Turbo Train suspension stiffnesses and masses was transferred directly to the appropriate part of Battelle's computer program by adding the truck frame and unsprung masses to the original three degrees of freedom. Comparison with a linear computer model recently developed by the United Aircraft Research Laboratories showed a high degree of similarity in the two models(8).

Since the 10-axle GG1 electric locomotive presented a more complex vehicle to model than the average rail vehicle, a special variation of the computer model was generated to simulate this vehicle. The GG1 locomotive consists of a cab (sprung mass) supported on two main frames, each with three driving axles (with equalizing spring rigging) and a 2-axle guiding truck.

5.1.2.2 Model Limitations and Assumptions. Oscillatory modes in the lateral plane, commonly referred to as "hunting" modes, were not considered in this study. The dynamic forces resulting from hunting can contribute to derailment and excessive wear of both vehicle and track components. In general, high-speed passenger vehicle trucks are purposely designed to operate well below the critical hunting speed. However, current freight car truck designs ("three-piece bogie") have been noted for dynamic instabilities within the normal operating speed range under some conditions, particularly a "parallelograming" oscillation (or "lozenging") that may occur at speeds as low as 40 miles per hour.

Other dynamic modes of truck oscillation such as truck pitching and the vibration of individually-sprung masses (such as gear boxes and traction motors) were not considered in detail, nor were the gyroscopic effects of wheel or traction motor rotation included.

Parameters representing the dynamic characteristics of the vehicles modeled in this study were supplied for the most part by the Federal Railroad Administration or the vehicle manufacturers from previously-generated test or analytical data. Several of the vehicles had undergone extensive prior study, and were generally well defined in terms of dynamic parameters: included in this group were the Metroliner, the DOT Test Car (modified Budd Silverliner), the Turbo Train, and the 100-ton freight car. The two locomotives and the passenger car (representing a typical lightweight coach with outside swinghanger trucks) were somewhat less well defined.

Generally, the suspension stiffnesses of the vehicles and the component weights were known accurately. Damping values were less readily available, particularly those vehicles with "undamped" suspension elements dependent on friction between pivots and slides, or with elastomeric bushings. Some damping characteristics were obviously dependent on operating conditions and loads in other planes of action: high damping under traction or braking, but little damping while coasting, for example. In lieu of more accurate data, damping coefficients equivalent to 15 percent of critical damping based on the supported mass were used in several cases. Mass moments of inertia and heights of the centers of mass, particularly for truck elements, were seldom known and were therefore estimated.

Again, the basic limitation of linearization of a more-or-less nonlinear system was recognized. Nonlinear elements were replaced by linear approximations about a normal operating point, or by linear equivalent values.

5.1.2.3 Validation of Model. Data describing vehicle dynamic action in response to fairly well defined track conditions were available for three vehicles, the 100-ton freight car, the Metroliner, and the DOT Test Car. These data were derived both from field tests and previous analog and digital computer modeling. To validate the computer model used in this study, a comparison of results with data from prior studies was made for these three vehicles, and results are summarized below.

5.1.2.3.1 100-Ton Freight Car. A number of studies have been conducted to define the dynamic response of the 100-ton hopper car to staggered-joint track[9,10,11]. These studies have included extensive field tests on specially-prepared tracks at Hollidaysburg, Pennsylvania, and Frankfort, Kentucky. Detailed computer simulations have also been conducted on both digital[12] and analog[13] computers. Results from the linear, frequency-domain program of car response to a simulated staggered-joint track compared very well with previously calculated data.

Roll Angle. Total roll angle of typical 100-ton hopper cars (without auxiliary damping devices) on staggered-joint track shimmed to a 3/4-in. cross-level error ranges from 5 to 10 degrees in a resonant speed range from 15 to 19 miles per hour. A maximum roll angle of 6.7 degrees total ($\pm 3.35^\circ$) at 19.8 miles per hour was calculated by the linear program for a 3/4-in. crosslevel track. The lower actual resonant speed can be attributed to nonlinear effects, particularly wheel lift and centerplate separation, which are not included in the linear model. Roll angles less than 1 degree total ($\pm 0.5^\circ$) at speeds above 33 miles per hour were calculated, and agreed with analog computer results.

Car Body Lateral Accelerations. Accelerations measured at the car-rocking resonance[11] on 3/4-in. shimmed track were typically 0.6 G peak to peak, while accelerations calculated by linear program for a 3/4-in. crosslevel error were 0.51 G peak to peak.

Car Body Vertical Accelerations. In previous Battelle studies, analog computer results for the 100-ton freight car showed a very strong vertical response to car rocking due to nonlinear coupling of bounce and roll modes, resulting in maximum accelerations up to 0.55 G in the 17 to 21 mph range. Maximum accelerations up to 0.35 G peak to peak have been recorded during analog simulation in the 40 to 70 mph range, but these peak accelerations were primarily higher frequency impulses due to rail joint impacts. The linear model used in this study contained neither the nonlinear coupling effects nor the rail joint impacts responsible for these high-level vertical accelerations. A more valid comparison can be made between computer results and accelerations recorded by Luebke[14]. A 70-ton boxcar used in that C&O/B&O study on track shimmed to a 1/2-in. crosslevel error showed a maximum unfiltered acceleration level of 0.09 G peak to peak at 30 miles per hour. The TRAKVEH program calculated 0.05 G by comparison at 27 miles per hour for comparable track with a 100-ton car.

Track Forces. Analog computer simulation of 100-ton hopper cars under severe car rocking conditions has shown that the wheel/rail vertical forces ranged from 68,000 to 89,000-pounds peak (compared with a 32,500-pounds static wheel load) and that wheel lift occurred during the roll cycle. For a comparable 1-in. crosslevel error the linear program calculated a peak vertical wheel load of 79,000 lb at the resonant speed, with wheel lift implied by a dynamic component far greater than the static. Lateral forces as high as 66,000 pounds transmitted into a side frame during car rocking were recorded by analog simulation, due primarily to impact of the bolster gibs. The linear model calculated a maximum side frame lateral force of 40,300 pounds at the roll resonance. The AAR tests[11] have shown lateral side frame forces generally under 20,000 pounds; however, Peterson[15] has measured net lateral axle forces of 5,000 pounds for a standard deviation on "representative jointed rail track sections", which would imply a three-sigma total level for a side frame on the order of 30,000 pounds. Bandwidth of the force measurement has a significant effect on the recorded peak, of course.

Suspension Travel. As a check on the linear range of solution, maximum suspension deflections were monitored in all runs. At the roll resonance, secondary suspension (spring group) deflections were calculated to be 1.54 in.

per inch of crosslevel. Since the 3-11/16-in. travel springs can compress only approximately 1.3 in. from static to solid height, the computer solution in this particular case had exceeded the linear bounds. In the actual case, therefore, higher accelerations and forces would occur due to spring bottoming.

5.1.2.3.2 Standard Metroliner. Laboratory and road tests were performed on a Metroliner(16) to define the dynamic characteristics. From sinusoidal response tests, the following comparisons were made of resonant frequency and gain (ratio of peak response to peak input, Table 5-1).

Resonant frequencies and gains calculated by the computer program were in response to track profile excitation, and therefore represent total system response. The Budd tests were more specific in nature: the car body, for example, was excited with the primary suspension blocked out; and the truck dynamic tests were run with axles and car body fixed, and a force applied to the truck assembly at appropriate points. This accounts for discrepancies in the comparison, particularly in resonant peak gains.

TABLE 5-1. COMPARISON OF TEST AND MODEL VIBRATIONAL MODES OF THE METROLINER

	Budd Tests		TRAKVEH Program*	
	Frequency, Hz	Gain	Frequency, Hz	Gain
Car body bounce	1.05	9.0	1.06	9.6
Car body pitch	1.32		1.3	
Car body lateral	0.7	1.5	0.65	5.2
	1.4	2.8	1.13	0.7
			1.44(1)	
Truck frame, vertical	9.5	5.4	9.1	5.3
Truck frame, lateral	6.1		5.6	3.8
Transformer, vertical	4.8	13.0	4.9	9.8
Car body bending, vertical	7.4	12.0(2)	8.4	

(1) Response to crosslevel excitation.
(2) Modified by longitudinal resonance at 6.6 Hz.
(*) In response to sinusoidal 0.1 G input at track.

5.1.2.3.3 DOT Test Car. Before 1976, limited test data had been published for the Department of Transportation Test Car, a modified Silverliner car built by The Budd Company. The standard deviations (sigma) of vertical and lateral accelerations at the center of the car were calculated from measurements taken during high-speed runs on the Pennsylvania Railroad[17]. Values at different speeds were compared below with root-mean-square accelerations calculated for the 0.2 to 12.8 Hz frequency band (Table 5-2).

TABLE 5-2. COMPARISON OF MEASURED AND COMPUTER MODEL-CALCULATED ACCELERATION LEVELS, DOT TEST CAR ON CLASS 6 TRACK

	Vertical Accelerations, Car Center (g)		Lateral Accelerations, Car Center (g)	
Speed, mph	Measured, 1-sigma	Calculated, rms	Measured, 1-sigma	Calculated, rms
100	0.028	0.015	0.019	0.021
110	0.030	0.019	0.021	0.022
125	0.035	0.025	0.025	0.025
140	0.045	0.026	0.033	0.027

Calculated values of vertical acceleration were somewhat lower than the measured, possibly due to dynamic stiffening of the air bag suspension, or possibly due to a rougher track surface (especially if bolted-rail track was traversed in the runs). A good correlation of lateral accelerations was seen (the calculated value represents an rms sum of response due to alignment and crosslevel PSD inputs), up to a speed of 125 mph. At higher speeds, a rapid rise in both lateral and vertical measured accelerations could indicate the occurrence of some truck hunting, which has not been included in the model.

5.1.3 Phase III (1974-1976)

5.1.3.1 Mathematical Model. The linear, frequency-domain model (Program TRKVPSD) was further developed after publication of the Final Report[1] to incorporate improvements, particularly in the calculations of wheel/rail forces. The current version of the model shown in Figure 5-2 and 5-3 (TRKVPSD MOD Ia) has been modified to include the inertial effects of the track, with the geometry input applied at the wheel/rail interface. A better simulation of the chordal geometry effects due to the axle spacing has also been included.

A more extensive modification of this basic model (Program TRKVPSD MOD II) has now been programmed. In this version each wheelset has been modeled individually in vertical, lateral, yaw, and roll degrees of freedom, using the basic equations of motion developed by Wickens[18] and others for the kinematics of a wheelset. Inertial effects of the track and the side frames (equalizer beams) have been incorporated, and in addition the yaw degree of freedom of the truck frame (truck sprung mass) has been programmed. While the lateral wheel/rail stiffness is normally calculated from the creep coefficients and gravitational stiffness, both program versions have a flanging/nonflanging option. In the case of assumed flange contact, the rail lateral stiffness (on the order of 300,000 to 500,000 lb/in.) is used.

5.1.3.2 Model Limitations and Assumptions. Although the lateral hunting modes have been added to the simulation, the basic limitation of linearization is still present. Wheelset dynamics are based on linear approximations of wheel/rail geometry and generation of the creep forces. Both can be highly nonlinear, including flange contact and wheel slip (adhesion limit).

To reduce computer solution time, only the front truck has been modeled in detail in the currently-operating programs (solution time increases approximately by the 2.5 power of the ratio of degrees of freedom). This is not a severe limitation since the program can be expanded to both trucks quickly if required. It has been observed in the field that more often than not just one truck of a vehicle will hunt.

5.1.3.3 Validation of the Model. Recent field measurements of wheel/rail loads on Northeast Corridor track[19] have shown the mean vertical wheel load and the mean plus 1-sigma vertical wheel load for the standard Metroliner to be 22,700 lb and 26,200 lb, respectively at a representative point of measurement. Using the TRKVPSD MOD IA program with the most recent track geometry representation of the NEC track (in random PSD form), a static plus 1-sigma vertical load of 26,000 lb was calculated for a given static load of 21,700 lb. This point of correlation shows the analytical model to be giving very realistic predictions.

Car body vertical and lateral acceleration power spectra from the DOT Test Car from runs on NEC track during 1975 (ENSCO RG-145) are compared in Figures 5-4 and 5-5, with spectra generated from the computer simulation using a composite of track geometry spectra for these same runs. Variations in the measured spectra on sections of track varying in length from 4 to 12 miles can be seen. The correlation between measured and predicted spectra is good, except that the "nodes" or antiresonances tend to be more pronounced in the computer-generated spectra. The broadening of measured acceleration power at these frequencies may be due to nonlinear effects or the interaction of degrees of freedom not modeled, particularly the cross coupling between pitch and longitudinal modes.

Relatively little has been done yet in the validation of the TRKVPSD MOD II Program; however, a preliminary comparison has been made between this program version and the previously-used program (TRKVPSD MOD IA), with good correlation in car body accelerations. The modified program does provide better resolution of wheel/rail forces. A comparison of computer-generated forces at the side frame/bearing adapter interface with some available field test data[19,20] is shown in Figure 5-6. Conditions for these two plots are somewhat different, inasmuch as the tests were run on Bessemer and Lake Erie Railroad track on which the track geometry was not available. Therefore, the Northeast Corridor track geometry (see Figures 5-10 through 5-12 below) was used for the computer runs since both the B&LE Erie Branch and the Northeast Corridor tracks are CWR. Note from the comparison that the Northeast Corridor track PSD exhibits stronger long-wavelength spectra--particularly in cross level--than the B&LE data indicate for the Erie Branch track. In spite of these differences in the inputs, the resulting force spectrum is seen to be quite realistic.

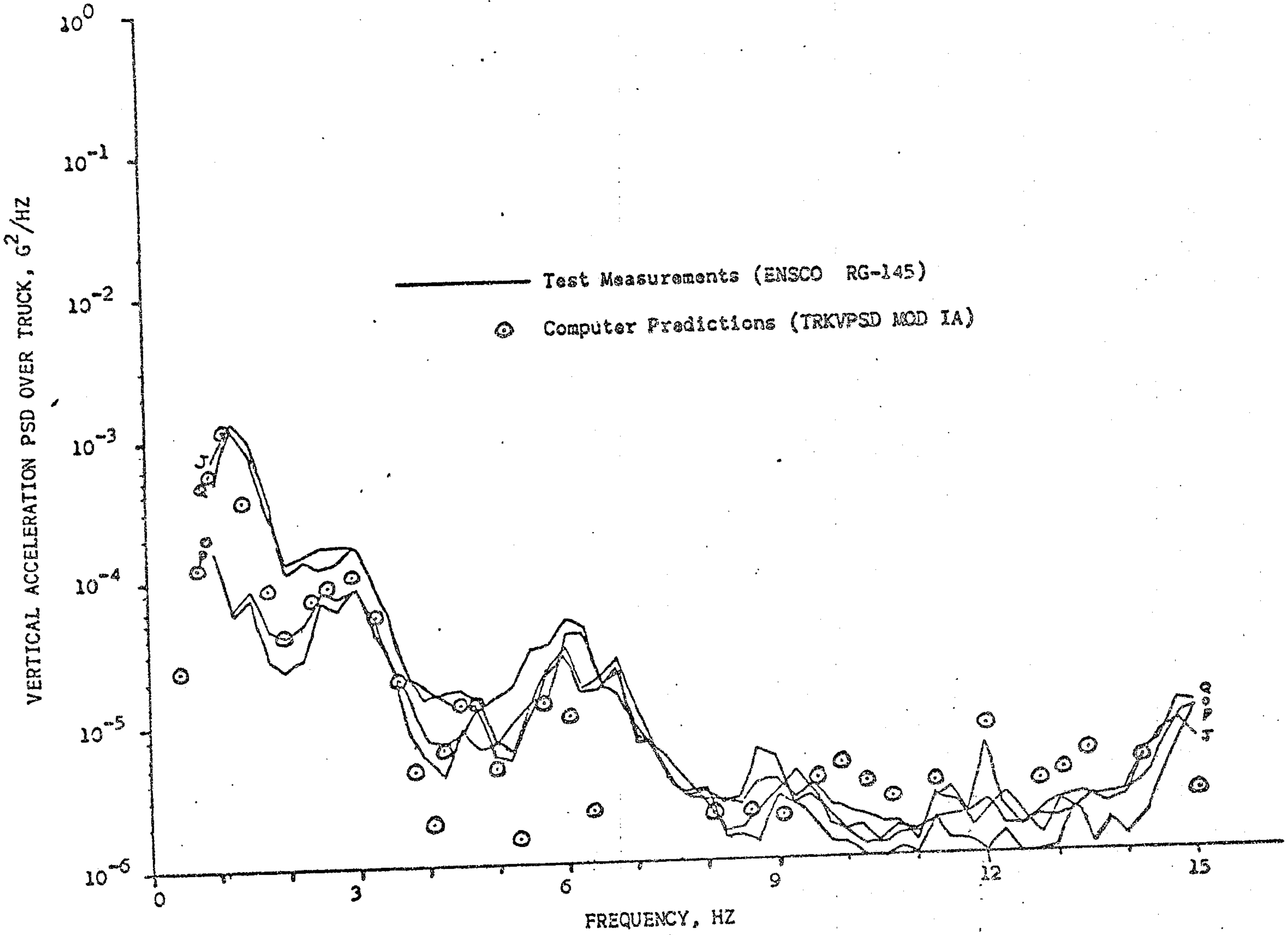

FIGURE 5-4. COMPARISON OF VERTICAL ACCELERATION POWER SPECTRA FROM MEASUREMENT AND COMPUTER PREDICTION, DOT TEST CAR ON NORTHEAST CORRIDOR TRACK, 80 MPH

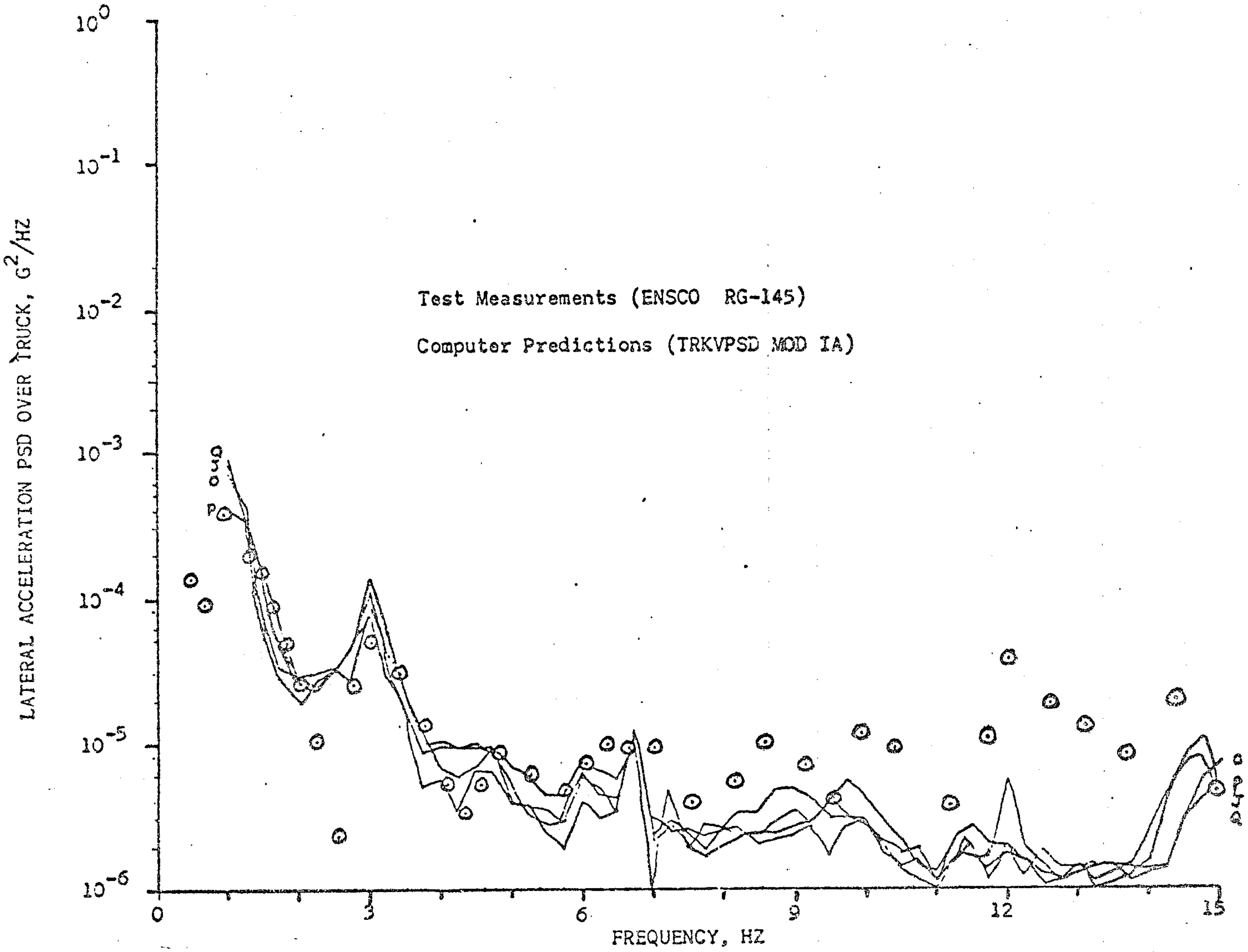

FIGURE 5-5. COMPARISON OF LATERAL ACCELERATION POWER SPECTRA FROM MEASUREMENT AND COMPUTER PREDICTION, DOT TEST CAR ON NORTHEAST CORRIDOR TRACK, 80 MPH

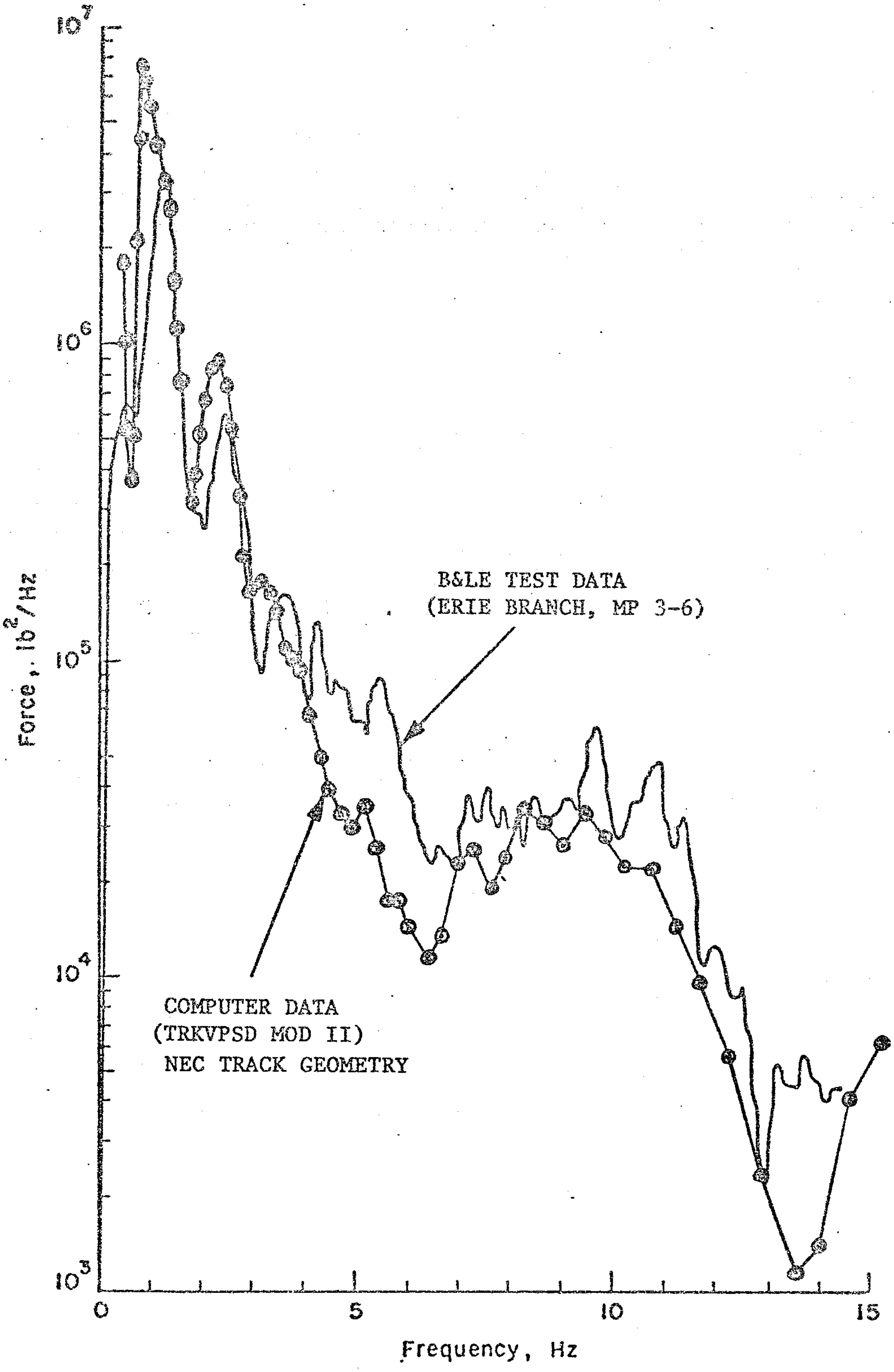

FIGURE 5-6. COMPARISON OF VERTICAL FORCE POWER SPECTRA AT SIDE FRAME/BEARING ADAPTER INTERFACE, 100-TON HOPPER CAR AT 35 MPH, LOAD OF IRON ORE, TEST DATA VERSUS COMPUTER-GENERATED DATA

5.2 Review of Nonlinear Vehicle/Track Models

5.2.1 Phase II (1973-1974)

As part of the comparative analysis program, a time-domain model of the rail vehicle was also developed for vehicle analysis by analog/digital (hybrid) computer techniques[4]. This time-domain model was developed to provide an efficient means for determining vehicle transient response to track geometry errors, to handle known nonlinear parameter characteristics, and to provide a link between the frequency-domain model and time-domain test data from a prototype vehicle.

Similar to the TRKVPSD program, only the front truck of the rail vehicle was modeled in detail, with the rear truck considered an overall complex impedance between the car body and track. The track structure, individual axles, equalizer beams (side frames) and truck frame were modeled on the analog computer (a Beckman 2133). Important nonlinearities such as wheel flange contact, wheel lift, suspension clearances and hard stops, damper force limits, and friction were included in the simulation. Secondary (truck) hunting modes were not considered in the model, however.

Car body vertical and roll/lateral/yaw degrees of freedom were programmed on the digital computer (a PDP-7). Forces and torques developed through the secondary suspension (modeled on the analog computer) were transferred through analog/digital (A/D) converters to the digital portion of the program, which in turn transferred car body velocities and displacements back through D/A lines to the analog computer. Program set-up and checks were processed by the digital computer, and output data were recorded on two 8-channel strip recorders.

5.2.2 Phase III (1974-1976)

Since the TRKVPSD model neglected some degrees of freedom (such as the lateral and torsional bending and--until TRKVPSD MOD II--the wheelset and truck frame yaw modes), a generalized model was developed[2] which is detailed enough to simulate all the important degrees of freedom of a 4-axle rail vehicle. Equations of motion were derived by LaGrange's methods from kinetic and potential energies, the energy dissipation function, and the generalized external forces. The resulting 46 nonlinear equations of motion are outlined in Table 5-3.

TABLE 5-3. DEGREES OF FREEDOM FOR COMPLETE RAIL VEHICLE

Mass	Degrees of Freedom	Total
Car body (rigid)	Yaw, pitch, roll, lateral, vertical	5
Car body (flexible)	Torsional, lateral, vertical (first modes)	3
Truck frames (2)	Yaw, pitch, roll, lateral, vertical	10
Wheelsets (4)	Yaw, roll, lateral, vertical	16
Track (4)[a]	Roll (crosslevel), lateral, vertical	12
Total		46

(a) Track associated with one wheelset.

The equations of motion are strongly coupled in the inertial terms. They are nonlinear not only in the sense of nonlinear suspension elements, but also in that they contain products of angular displacements, velocities, and accelerations. Although a time domain model can be programmed from these equations, frequency domain solutions are not possible because of the nonlinearities. In order to solve the eigenproblem and to find the transfer functions for a random vibration analysis, it is necessary to linearize the equations of motion. The only way to eliminate all of the nonlinearities is to assume small angular displacements, velocities, and accelerations and neglect products of all such terms.

5.2.2.1 Program PSD46. The linearized set of 46 equations of motion result in two sets of uncoupled equations. The first set of 15 are for the pitch plane modes alone. The yaw/roll/lateral set contains the remaining 31 equations. These linear equations have been programmed on a digital computer (PSD46) to generate the eigenvalues, eigenvectors, and transfer functions necessary to solve for PSD's of the degrees of freedom, or the vertical and lateral wheel/rail force PSD's due to random surface, alignment, or crosslevel track geometries. Although the program will accept a cross PSD between alignment and crosslevel, it is believed that such a function has relatively small value. Integrating the PSD's of the above degrees of freedom (displacement or acceleration) or the forces over a specified frequency range yields mean square values of the variables. Such information can be used to establish passenger ride comfort (frequency weighted acceleration of car body), track damage (vertical forces) and track safety (lateral loads/vertical loads).

Although it is a relatively straightforward task to develop a time domain computer code to solve the 46 nonlinear equations of motion, it has not yet been done. The main disadvantage of such a program would be the large cost in computer time. In fact, the frequency domain solution just described is also very expensive to run. For this reason, and also to compare the effects of degrees of freedom at different frequencies, these equations have been further simplified by considering a single 2-axle truck.

5.2.2.2 Program PSD19. A computer program (PSD19) has been developed for a frequency domain solution of the single 2-axle truck model. The degrees of freedom included in the resulting equations are summarized in Table 5-4. As in PSD46, the PSD19 program will solve for both the eigenvalues and eigenvectors, and the appropriate transfer functions to predict PSD and root mean square values of system outputs, such as accelerations and wheel/rail forces.

TABLE 5-4. DEGREES OF FREEDOM FOR SINGLE TRUCK MODEL

Mass	Degrees of Freedom	Total
Truck frame	Yaw, pitch, roll, lateral, vertical	5
Wheelsets (2)	Yaw, roll, lateral, vertical	8
Roadbed (2)	Roll (crosslevel), lateral, vertical	6
Total		19

5.3 Review of Linear Truck Stability (Hunting) Models

5.3.1 Mathematical Model

A linear stability model was formulated to determine the effect of various parameters on truck hunting for the Improved Metroliner with SIG trucks[(2)]. The analytical model included seven degrees of freedom for the truck only: truck frame roll, yaw, and lateral, and two wheelsets yaw and lateral. The truck frame was considered rigid (no lozenging degree of freedom). The suspension included primary lateral, longitudinal, and vertical stiffness and damping elements, and secondary lateral, vertical, yaw, and auxiliary roll (roll bar) stiffness and damping elements. Lateral and yaw gravitational stiffnesses for both wheelsets were taken into account in the equation of motion. Longitudinal and lateral creep forces resulted from wheelset yaw and lateral damping and stiffness terms.

The derivation resulted in seven linear second order coupled differential equations of motion. The coupling was associated with the seven variables and their first derivatives--stiffness and damping terms. There was no coupling in the second derivative terms. Because the system of equations was linear, all suspension parameters, creep coefficients, and wheel/rail profiles were also linear. No nonlinear effects, including flange contact, were simulated.

Assuming a harmonic response, and using an appropriate subroutine to solve for all of the eigenvalues and eigenvectors of the seventh order system, stability information such as the natural frequency and damping ratio for each of seven normal modes of oscillation are generated at several vehicle speeds. This type of analysis assumes that the system is only slightly perturbed from equilibrium, so that it remains linear. If the damping ratio becomes negative at a given speed for any of the normal modes of oscillation, the linear system is said to be unstable. Theoretically, this implies that the harmonic motions of the degrees of freedom will increase without bound. In practice, the degrees of freedom eventually reach a magnitude such that the physical properties of the system change. Such a change may then make the system stable, at which point a limit cycle will develop--that is, the oscillations will not diverge nor will they damp out. For a rail vehicle, this limit cycle consists of a rail-to-rail impact of the wheel flanges, typically at 2 to 4 Hz.

5.3.2 Limitations of Model.

The linear truck hunting analysis only predicts the stability of the vehicle in the neighborhood of its unperturbed equilibrium. It does not predict the vehicle response after the system has moved out of the linear range, particularly after flange contact has occurred. It therefore cannot be used to predict wheel/rail forces, nor can it predict sustained oscillations at speeds somewhat lower than the linear critical speed which results from nonlinear dynamics. However, this type of analysis is important first to predict the small-amplitude stability (critical speed) of a vehicle with a given set of parameters; and second to determine the relative sensitivity of this critical speed to changes in these parameters that might occur over the life of the vehicle.

5.4 Review of Curving Models

5.4.1 Phase I (1972-1973)

In the first part of this comparative analysis program, a simplified program was formulated to study vehicle overturning stability (the AAR "one-third"rule) at high speeds on curves. This program was evolved from the nine degree of freedom (half-car) linear model by applying the centrifugal and gravitational accelerations to the mass centers, and considering the steady-state solution as a function of train speed, track curvature, and superelevation. As a result of the vehicle parameters (c.g. height, suspension stiffnesses, etc.) the vertical and lateral shift in c.g. position, vertical and lateral loads, and L/V ratios were then calculated.

5.4.2 Phase III (1974-1976)

A program for the analysis of lateral forces on each wheel of a 2-axle truck during constant-radius curving was developed under a Northeast Corridor evaluation study. This program (SSCUR2) was based on equations developed by Newland[21] and included two axles in lateral and yaw, and one rigid truck frame in lateral, yaw, and roll motions. To predict displacements and wheel forces, Battelle's computer code includes a secondary (frame to car body) yaw stiffness, constant centerplate torque, flange contact, limited creep force due to wheel slip, and the option to simulate a leading or trailing truck. In addition, the centrifugal forces are assumed to act through the individual mass centers (wheelsets, truck frame, car body), rather than being lumped at the truck center of mass.

As part of the vehicle analytical support function under this contract (DOT-FR-20077), Battelle was asked to consider the problem of steady-state curving of 6-axle locomotives used in passenger service by AMTRAK. The curving program was expanded to include three axles, resulting in a system of nine equations (Program SSCUR3).

In both the 2- and 3-axle truck curving programs, if flange contact is predicted for any wheel by a lateral displacement greater than the flange clearance, the flange force is estimated by using a lateral rail stiffness to limit this lateral displacement. Creep forces on a wheelset are limited to a maximum slip force dictated by the axle load and the coefficient of friction. After each simultaneous solution of the equilibrium equations, the total creep force acting on each wheelset is compared to the maximum slip force, and the possibility of flanging is checked. An iterative solution with equations appropriately modified to satisfy these conditions then generates the final solution.

The FRA was provided access by the Association of American Railroads to a computer program for calculating steady-state forces developed by 3-axle locomotive trucks during curving. This program was developed by Electro-Motive Division of General Motors Corporation under the AAR-FRA-RPI-TRA Track Train Dynamics Program. It determines by iterative solution the location of the "friction center" for the balance of forces and moments on the truck when given the truck "degree of constraint", or wheel flanging configuration. Although good comparison was found between published lateral force data and results with Battelle's SSCUR2 Program for 2-axle vehicles, results from the SSCUR3 Program did not correlate well with results from the AAR program. In general, the lateral forces and L/V ratios calculated from the Battelle program were higher than those calculated by the AAR program or measured during field tests. Further development of the 3-axle truck curving program is therefore required before this program can be used with confidence.

5.5 Track Geometry

5.5.1 Random Geometry Spectra

A realistic input, or "forcing function", is as important in mathematical modeling as an accurate and realistic rail vehicle simulation. For this reason much emphasis has been placed during the course of this comparative analysis study on the development of representative track geometry. Track geometric irregularities--rail surface, alignment, track cross level, gage--are found to have random distributions in amplitude and wavelength that can be described in the power spectral density (PSD) format of "power" (in.^{2}/cycle/ft) versus frequency (cycle/ft). Superimposed on these random geometric variations are discrete spectral components that result from periodic rail joints or rail welds, from joint stagger, or from other constructional peculiarities.

A number of investigators have found that track irregularities, in common with road and runway surfaces (at least over a limit range of wavelengths), exhibit a random variation in amplitude and wavelength of the form,

$$P_i\ (\lambda) = C\lambda^N. \tag{1}$$

By assuming the track geometry to be a stationary random process (at least for a reasonable time period) over a broad frequency range with a Gaussian amplitude distribution, the response spectrum of each output variable may be calculated

$$P_o(f) = \left|H(f)\right|^2 P_i(f) \qquad \text{(for a single input)}, \tag{2}$$

where $P_o(f)$ = output spectrum (G^2/Hz, for example)

$P_i(f)$ = track geometry spectrum (in.^{2}/Hz)

$= P_i(\lambda)/V$

f = frequency, Hz

λ = geometry wavelength, ft

V = vehicle speed, ft/sec

$H(f)$ = vehicle transfer function (input to output).

By the use of a random input, all phase information is lost between the different inputs, unless cross power spectra are also generated. Note, however, that for a rail vehicle the phase relationship of the same input at the different axles must be maintained as a function of wavelength and axle (or truck) spacing. In the comparative analysis study, cross power spectra have not been included, and a simple mean-square addition of the output spectra of a variable due to more than one input has been used for an overall result.

Typical track geometry measurements in the PSD format were analyzed during the comparative analysis program. Measurements from Northeast Corridor (NEC) track[22] were used to estimate "Class 6" spectra for surface, alignment and cross level[1], using a continuous function in the general form of Equation (1). Surface and alignment PSD functions followed a λ^3 relationship over approximately the 10 to 100 ft wavelength range, rolling off to a smoother λ^6 slope at shorter wavelengths; while the cross level spectrum followed a $\lambda^{1.7}$ slope under 100 ft. All three spectra were rolled off to a constant value at very long wavelength (>200 ft).

Measurements of "Class 6" CWR track in Colorado, sponsored by the FRA and conducted by ENSCO with the Track Survey Device (under TG-69), were made available during the Phase III period of this contract. Some startling differences were apparent between the NEC and Colorado "Class 6" spectra, particularly in the cross level. For example, at a wavelength of 10 feet the Colorado track geometry spectrum fell more than one order of magnitude below the comparable NEC spectrum for cross level. Measured spectra and approximate (two-slope) curves for the Colorado track are shown in Figure 5-7 through 5-9.

More recent surveys of the Northeast Corridor track by ENSCO (under RG-125 and RG-145) using the DOT Track Geometry Car have also been analyzed. The range of spectra from six representative sections of NEC track, ranging from 4 to 12 miles in length, from the most recent survey (RG-145, April, 1975) are shown in Figures 5-10 through 5-12, along with the approximate (two-slope) spectra used in Battelle's most recent simulation efforts (see Section 5.1.3.3). It is interesting to note that the NEC track exhibits, in addition to the 39-ft wavelength, a strong spectral peak near 100 ft in wavelength. This has been tentatively associated with the dynamic response of the older (GG1) electric locomotives in hitting some initial perturbation.

Track geometry inputs in PSD form may be mechanized in the computer model several ways. For convenience the two-slope, or bilinear PSD input has

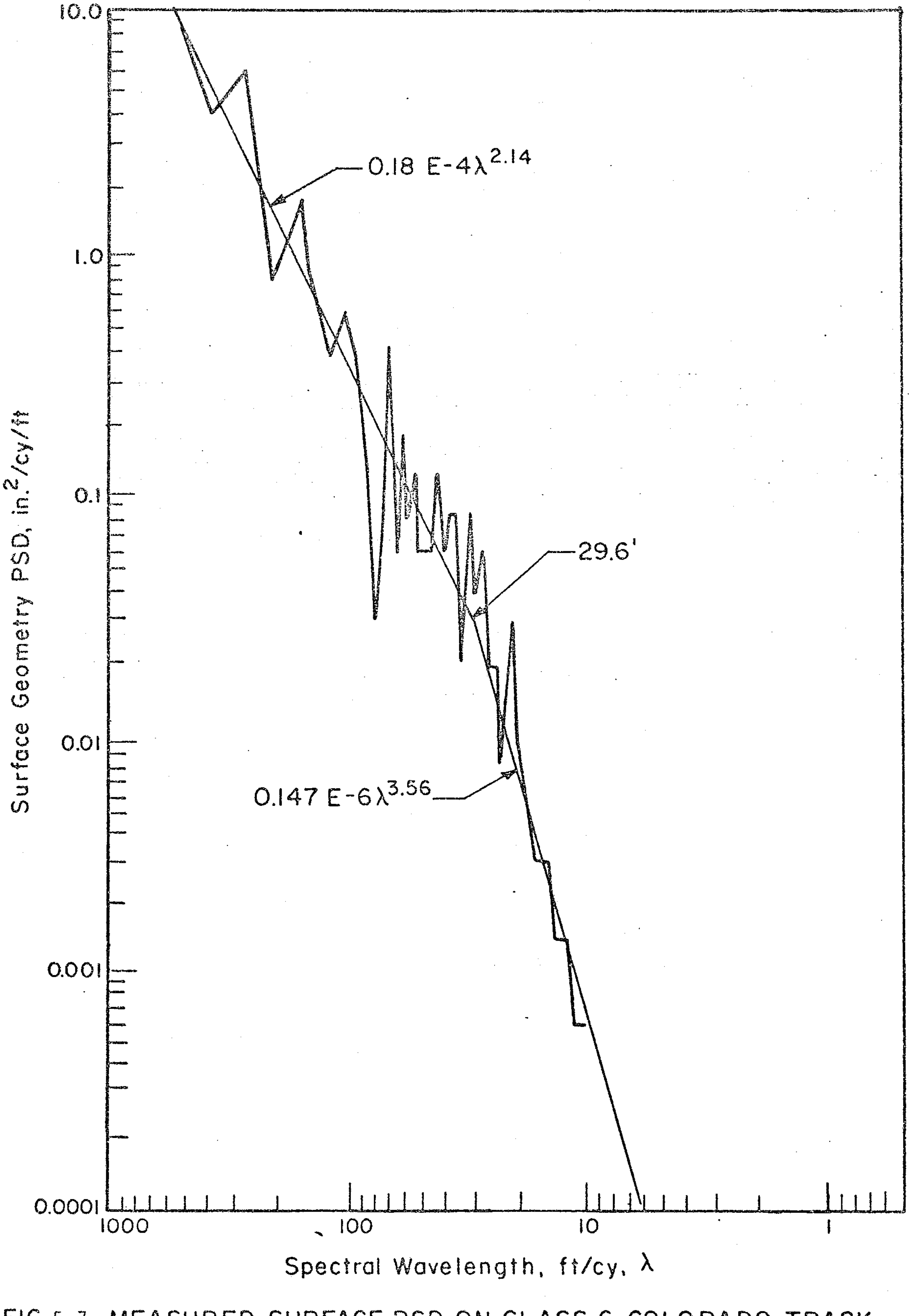

FIG.5-7. MEASURED SURFACE PSD ON CLASS 6 COLORADO TRACK (TEST RUN TG-69, ZONE 440, CWR, ENSCO, INC.)

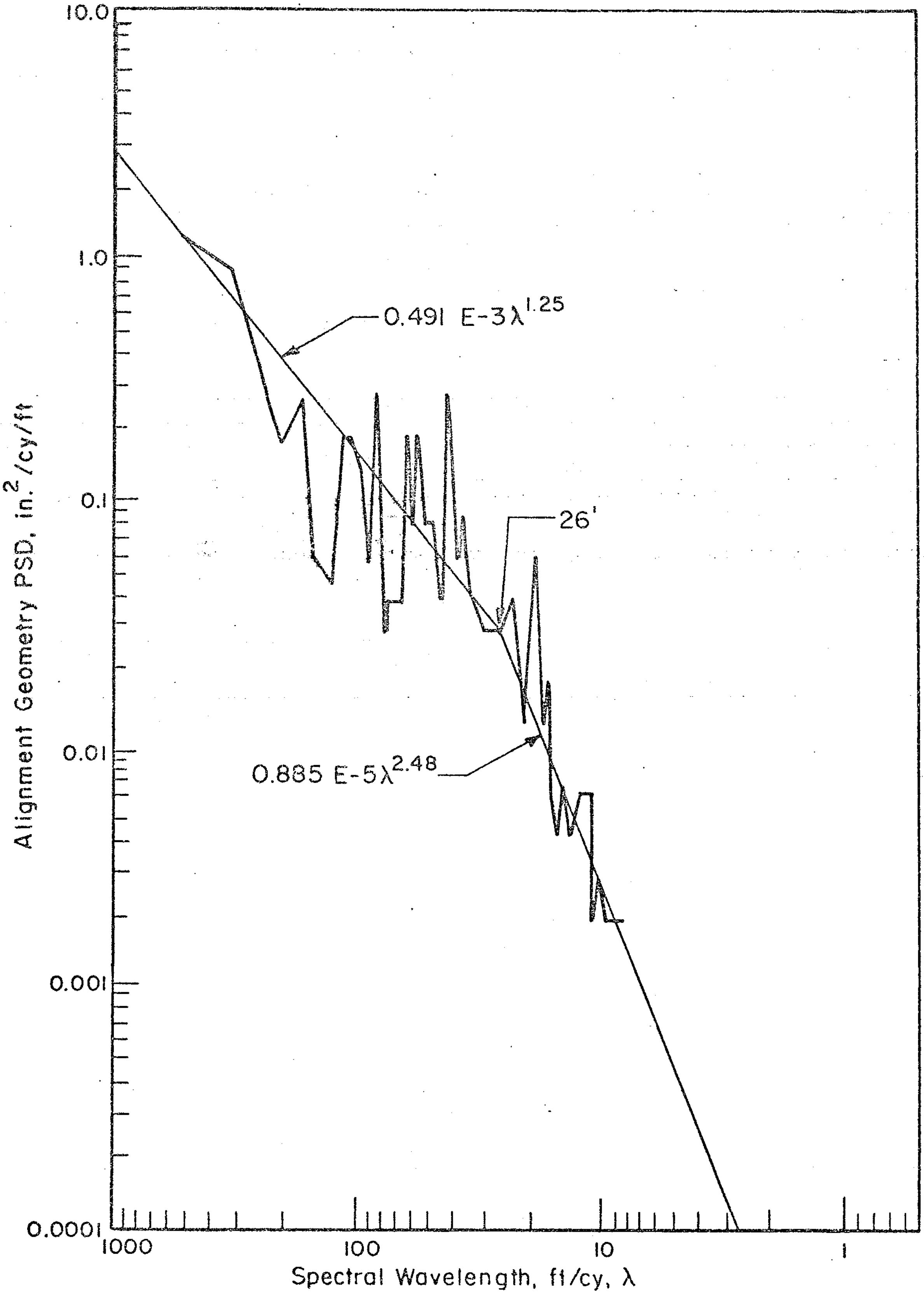

FIG. 5-8. MEASURED ALIGNMENT PSD ON CLASS 6 COLORADO TRACK (TEST RUN TG-69, ZONE 440, CWR, ENSCO, INC.)

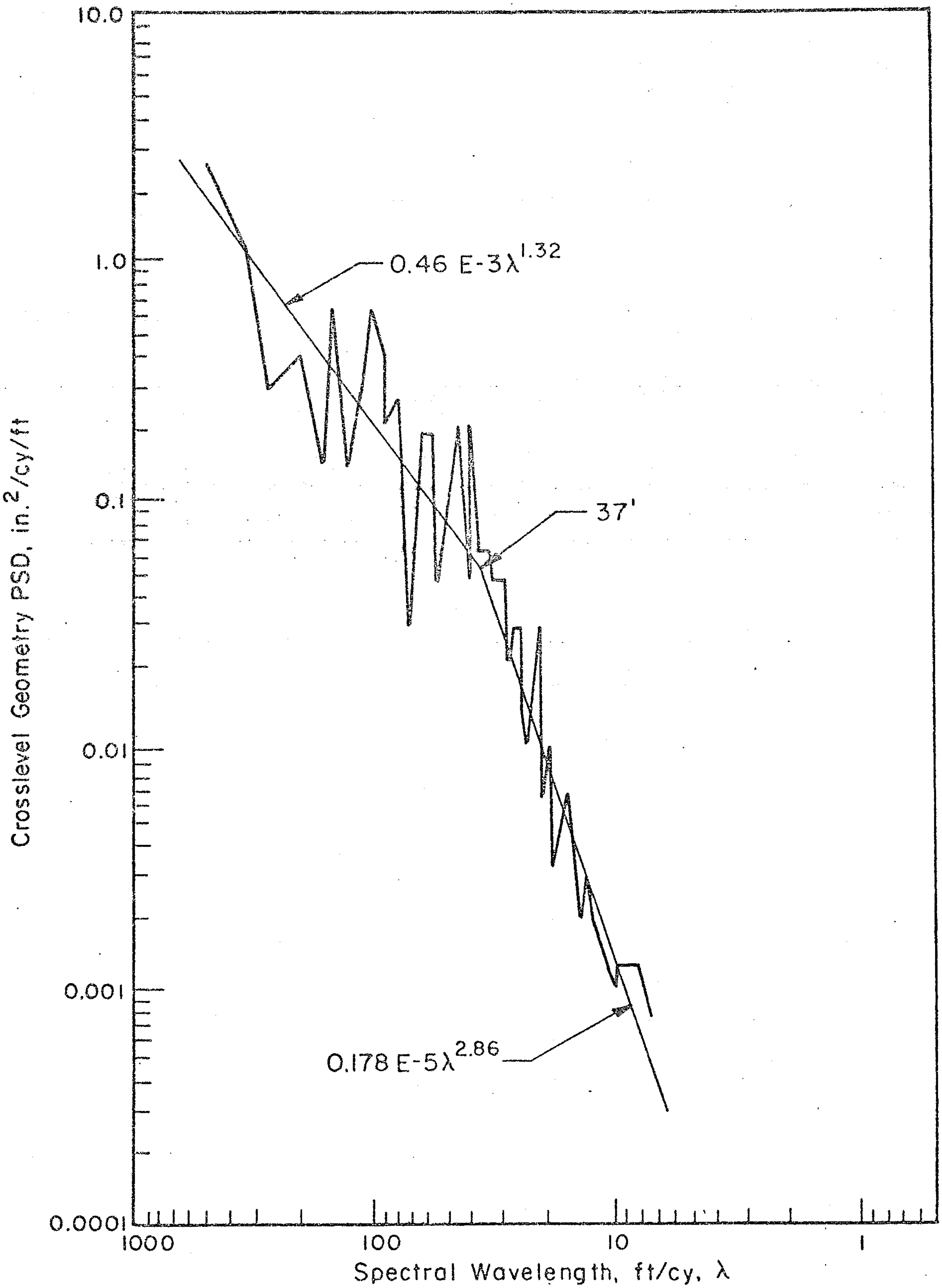

FIG. 5-9. MEASURED CROSSLEVEL PSD ON CLASS 6 COLORADO TRACK (TEST RUN TG-69, ZONE 440, CWR, ENSCO, INC.)

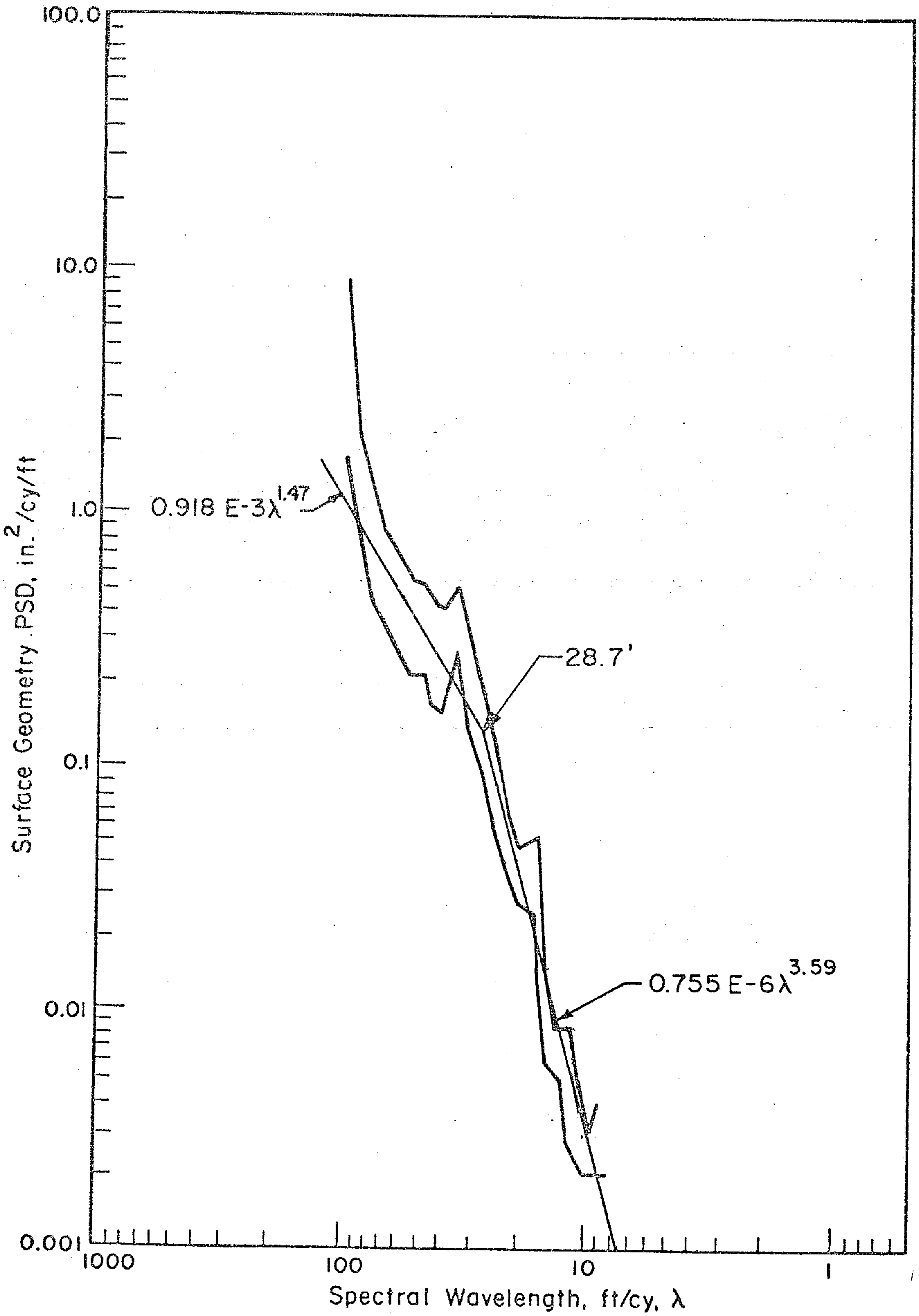

FIG.5-10. MEASURED SURFACE PSD ON CLASS 6 NORTHEAST CORRIDOR TRACK (RIDE QUALITY DATA, TEST RUN RG-145, APRIL, 1975, ENSCO, INC.)

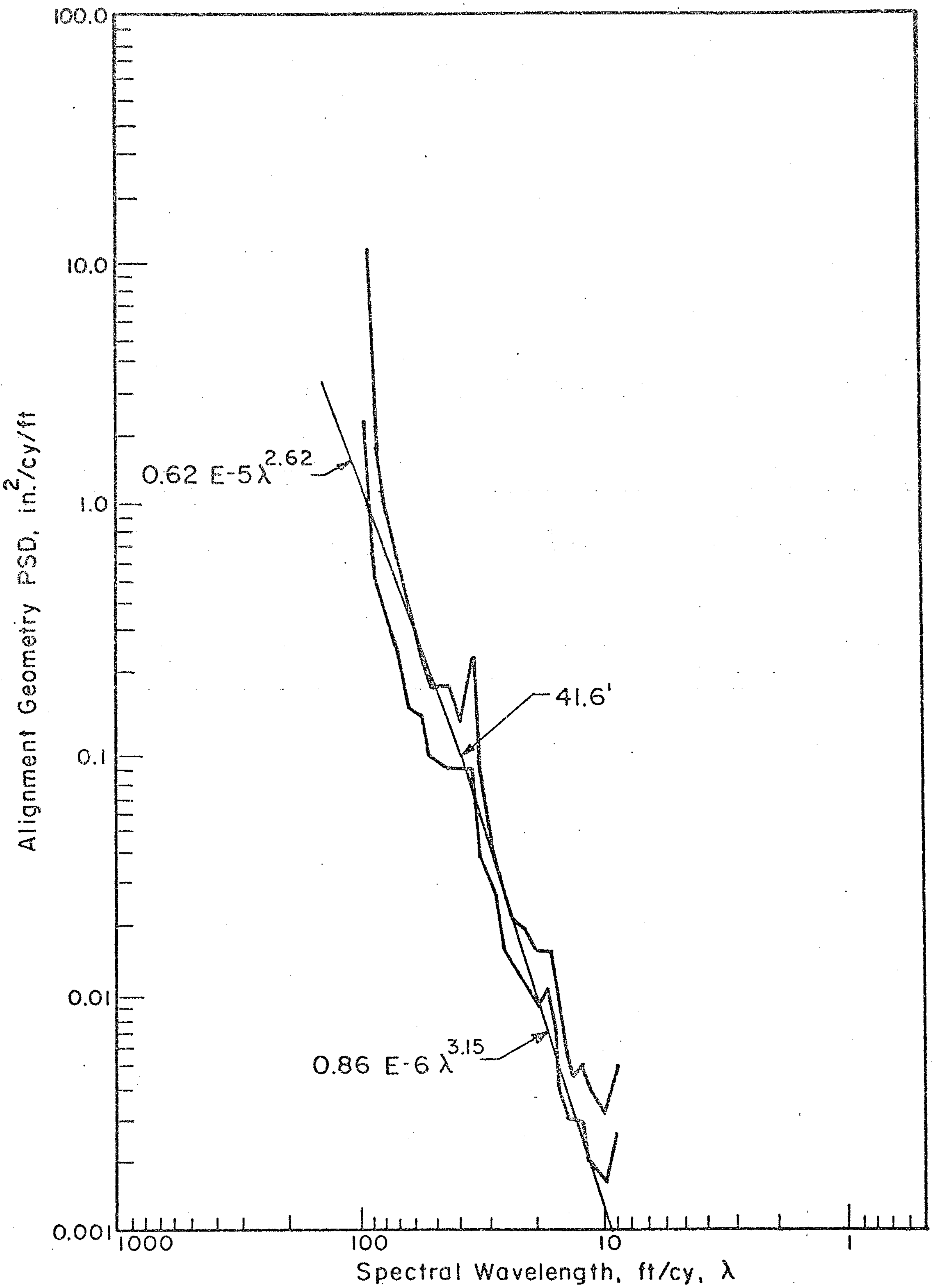

FIG. 5-11. MEASURED ALIGNMENT PSD ON CLASS 6 NORTHEAST CORRIDOR TRACK (RIDE QUALITY DATA, TEST RUN RG-145, APRIL, 1975, ENSCO, INC.)

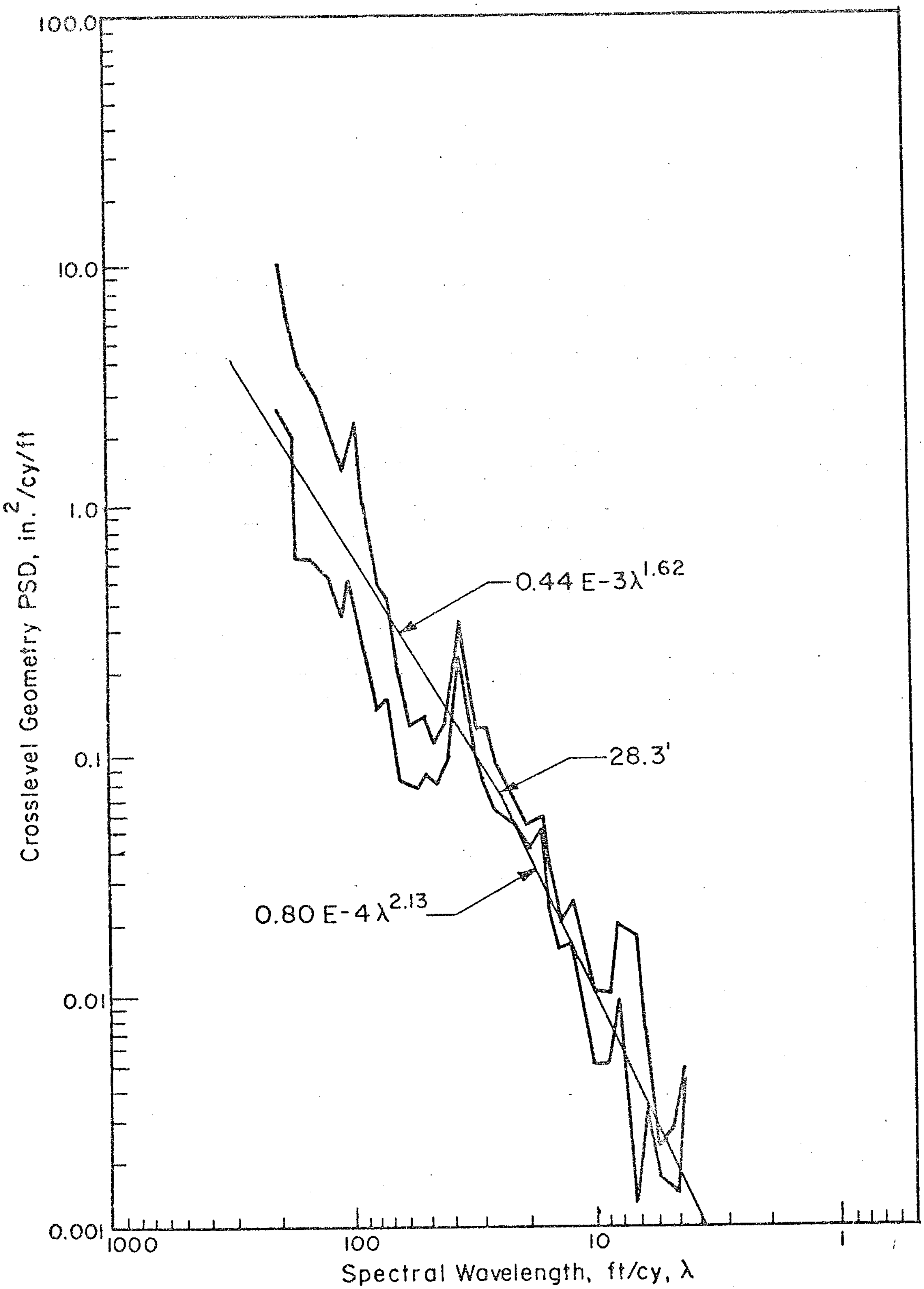

FIG. 5-12. MEASURED CROSSLEVEL PSD ON CLASS 6 NORTHEAST CORRIDOR TRACK (RIDE QUALITY DATA, TEST RUN RG-145, APRIL, 1975, ENSCO, INC.)

been used recently, with or without the addition of spectral peaks at the harmonics of 39 ft. A multilinear PSD input (a look-up table of values) may also be used, and a comparison of wheel/rail vertical forces resulting from these two formats is given in Figure 5-13. The limited accuracy of the measured data hardly warrants this additional complexity, however.

5.5.2 Discrete Geometry Spectra

In the early phases of this study, discrete geometry spectra were used with the linear, frequency-domain model to represent a bolted-rail, half-staggered-joint track. Each rail was considered to have a surface geometry profile in the form of a rectified sinewave, so that spectral harmonic peaks were calculated as

$$E_{zn} = 4e/\pi(4n^2 - 1),\ n = 2, 4, 6 \ldots \text{ surface,} \qquad (3)$$

$$E_{cn} = 8e/\pi(4n^2 - 1),\ n = 1, 3, 5 \ldots \text{ cross level,} \qquad (4)$$

where e = peak geometry error (joint to midspan of rail), in.

These Fourier components were used directly as inputs to the vehicle model to calculate peak amplitude outputs. Response to the first three harmonics in surface and cross level were combined to calculate a root-mean-square overall amplitude. In the linear model the higher harmonics necessary to simulate the joint impact were, of course, deleted.

The accuracy of the rectified sinewave representation of bolted-joint rail (BJR) track with service-bent rail (rather than a specially-shimmed test track) is of interest. The power spectral peak is related to the Fourier component of the rectified sinewave by

$$Q_n = E_n^2/2. \qquad (5)$$

In actuality, there is some broadening expected in the spectral peak, so that if the bandwidth (β_n) of the peak is defined at the half-power point of the actual peak (C_n), then

$$C_n = Q_n/\beta = E_n^2/2\beta. \qquad (6)$$

Estimates of theoretical spectral peaks at the 39-ft wavelength of standard rail are given in Table 5-5. Here a root-mean-square amplitude for the rectified sinewave has been assumed as one-sixth the maximum allowable change in cross level (the peak-to-peak value) within a 62-ft length as set by the Track Safety Standards. The spectral peak for surface represents <u>one</u>

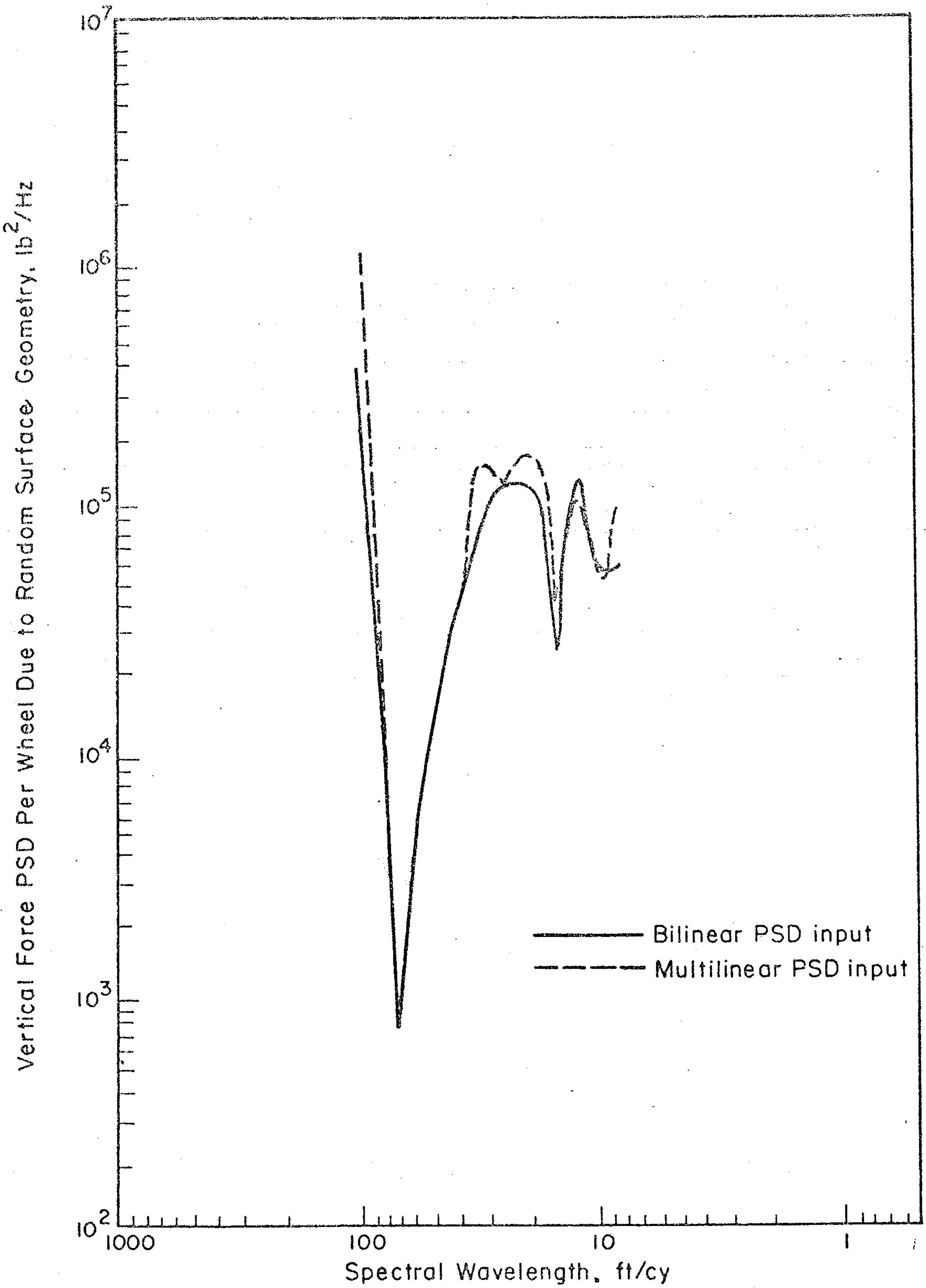

FIG. 5-13. COMPARISON OF VERTICAL WHEEL FORCE PSD FOR BILINEAR AND MULTILINEAR APPROXIMATIONS TO SURFACE PSD ON NEC CLASS 6 TRACK

rail (or adjacent joints), since the odd harmonics for surface with half-staggered joints are ideally zero. Typical values for the bandwidth (β) are taken from the study by Corbin and Kaufman[23] of track geometry measurements in Pueblo, Colorado area. The calculation of spectral peak values is seen to depend critically on the choice of bandwidth.

Typical measured spectral peaks at the 39-ft wavelength from both Colorado and NEC track measurements are given in Table 5-6. It is interesting to note at this point that the surface spectral peak is higher in all but one example than the cross level peak. Table 5-7 shows the range of spectral peak values from recent NEC geometry measurements with the DOT Geometry Car for the first four harmonics of the 39-ft wavelength. Finally, in Table 5-8, the calculated spectral peaks for a "Class 6" rectified sinewave are listed. Here, the spectral contribution from the rectified sinewave (Equations 3,4) is added to the "random background" (see Figures 5-10 and 5-12). These results correlate well with the measured NEC spectral peaks. However, the presence of all harmonics in the PSD (rather than odd harmonics only in cross level, even harmonics only in surface) can mean one of two things; either the track geometry is radically altered at the midspan of the rail due to the joint at the opposite rail (which has been observed in rail surface measurements with inertial transducers), or the geometry PSD curves represent the average of spectra from left and right rails, rather than the spectrum of the average.

TABLE 5-5. THEORETICAL SPECTRAL PEAKS AT 39-FT WAVELENGTH FOR RECTIFIED SINEWAVE

Track Class	e_{rms},[a] in.	E_z,[b] in.	E_c, in.	β cy/ft	C_z,[b] $in.^2$/cy/ft	C_c, $in.^2$/cy/ft
6	.10	.044	.085	.010	.097	.36
5	.17	.071	.14	.012	.21	.82
4	.25	.106	.21	.015	.37	1.5
3	.29	.125	.25	.018	.43	1.7

(a) Maximum allowed change in cross level (peak-to-peak) under Track Safety Standards divided by 6, assumed rms value.

(b) Based on one rail (would be zero for exactly half-staggered joints)

e_{rms} = joint depth, midspan to joint (rms)
E_z = first harmonic, rectified sinewave
E_c = first harmonic cross level error, half-staggered joints
β = bandwidth of spectral peak
C_z = spectral peak, rail surface
C_c = spectral peak, cross level

TABLE 5-6. TYPICAL MEASURED SPECTRAL PEAKS AT 39-FT WAVELENGTH

Estimated Track Class	Test Zone	Type of Track	C_z, $in.^2$/cy/ft	C_c, $in.^2$/cy/ft
6	440[a]	119 lb/yd CWR	.14	.21
5	430.1	136 lb/yd CWR (relay)	.21	.10
4	320	112 lb/yd BJR (relay)	1.4	.45
6	J[b]	140 lb/yd CWR	.34	.24
6	O	ditto	.40	.33
6	P	"	.37	.28
6	Q	"	.40	.25

(a) ENSCO TG-69 (Colorado), Track Survey Device.
(b) ENSCO RG-145 (Northeast Corridor), DOT Geometry Car.

TABLE 5-7. TYPICAL VALUES FOR SPECTRAL PEAKS FROM NORTHEAST CORRIDOR TRACK (ENSCO RG-125, RG-145)

	Spectral Peak, C_i		in.^{2}/cy/ft	
Wavelength, ft	39 ft	19.5 ft	13 ft	9.75 ft
Surface	.28 to .43	.028 to .051	.0054 to .0091	.0021 to .0040
Cross Level	.23 to .33	.048 to .059	.017 to .025	.010 to .020

TABLE 5-8. CALCULATED VALUES FOR SPECTRAL PEAKS FOR "CLASS 6" RECTIFIED SINEWAVE

Wavelength, ft		39	19.5	13	9.75
Random background	Surface	.20	.032	.0075	.0027
	Cross Level	.17	.045	.019	.010
Spectral contribution	Surface	.097	.0039	.0007	.0002
	Cross Level	.36	.014	.0026	.0008
Spectral peak	Surface	.30	.034	.0082	.0029
	Cross Level	.53	.059	.022	.011

5.5.3 Transient Geometry Perturbations

The rectified sinewave was used as an input to the time-domain (hybrid computer) model to simulate response to the staggered-joint, BJR track. In addition, single geometry perturbations were used to determine the transient response to typical disturbances. These disturbances might include track surface errors at grade crossings, cross level errors at a low joint, or alignment errors at buffer rails between CWR strings.

To simulate the transient geometry error, one or a combination of versine functions were used as an input to the program:

$$E_i = e_i(1 - \cos \frac{2\pi x}{L}), \quad 0 \leq x \leq L, \tag{7}$$

where e_i = maximum geometry error for i^{th} mode (surface, alignment, or cross level) allowed by Track Safety Standards.

5.6 Figures of Merit

Criteria for judging the acceptability of vehicle/track dynamic response fall generally into three categories: ride quality, safety, and vehicle/track loads. A comparison between different vehicles must rely on "figures of merit" that summarize the dynamic response in these three different categories. Criteria used in the comparative analysis study are reviewed in the following sections.

5.6.1 Ride Comfort Criteria

Human tolerance to vibration is both frequency and amplitude sensitive in a way analogous to the sensitivity of the human ear to sound levels. There is, therefore, a need to define a subjective scale relating ride quality, track class, and vehicle dynamic response. Much has been published on human sensitivity to vibration, and more specifically, ride comfort[24-28]. There is general agreement that the most sensitive frequency band for people falls between 4 and 8 Hertz, which corresponds to the major resonances in the human body. On the other hand, there is a diversity of opinion on what constitutes good ride quality (in terms of acceleration levels) and how the frequency ranges should be weighted.

Little has been published, however, on human tolerance to random vibration excitation and simultaneous excitation in the three orthogonal axes. Since the output variables from the PSD computer program represent response to random, broad-band excitation, a correspondence must be established between octave-band or broad-band root-mean-square accelerations and the tolerance (or ride index) curves. Two methods were explored to establish ride index values: the first method was to calculate for the various vehicles the ride index numbers based on the British Rail curves*, using the computer-calculated octave-band accelerations and the corresponding center frequencies. The ride index V_r was calculated as follows:

$$V_r = 7.5\ (g_{rms})^{0.3}(f_c/5.9)^{1/3} \quad \text{vertical}, \ f_c \leqq 5.9 \text{ Hz} \tag{8a}$$

$$V_r = 7.5(g_{rms})^{0.3}(5.9/f_c) \quad , \ f_c > 5.9 \text{ Hz} \tag{8b}$$

$$V_r = 8.1\ (g_{rms})^{0.3}(f_c/5.4)^{1/3} \quad \text{lateral}, \ f_c \leqq 5.4 \text{ Hz} \tag{8c}$$

$$V_r = 8.1\ (g_{rms})^{0.3}(5.4/f_c) \quad , \ f_c > 5.4 \text{ Hz} \tag{8d}$$

where

V_r = ride index

g_{rms} = octave-band rms acceleration, g

f_c = octave-band center frequency $= \sqrt{2}\ f_{low}$.

Once the ride indices were calculated for each frequency band, the overall ride index was calculated:

$$V_r = (V_{r1}{}^{10} + V_{r2}{}^{10} + \ldots + V_{r8}{}^{10})^{0.1} . \tag{9}$$

Using the British Rail system of subjective rating(25), the indices are equated with ride quality as follows:

1	very good
1.5	almost very good
2	good
2.5	almost good
3	satisfactory
3.5	just satisfactory
4	tolerable
4.5	not tolerable
5	dangerous.

A second method was investigated, taking the root-mean-square value of weighted octave-band acceleration, with the weighting factors based on the

(*) Reference 25 gives additional references as background to the development and use of these curves. These curves may originate with the "W_z" factors developed by the German Federal Railway.

ISO proposed tolerance curves for vibration exposure time[28]. Since the British Rail ride indices have been specifically formulated for rail vehicle ride quality from empirical data, these criteria are recommended at this time.

5.6.2 Safety Criteria

Safety criteria appropos of vehicle/track dynamics fall into four categories: (1) vehicle overturning stability, (2) track lateral shift, (3) wheel-climb derailment, and (4) rail rollover derailment[29]. In the first, vehicle overturning stability, the AAR "one-third rule" provides a conservative limit. This states that the total force vector (the vector sum of the static weight, lateral acceleration on curves, wind force, etc.) shall not fall outside the middle third of the track.

In the last three categories, the safety criteria are generally based on the ratio of lateral to vertical wheel/rail forces (L/V or Q/P ratios), although modified to some extent by the absolute value of the lateral and vertical forces. Track lateral stability is, of course, dependent on many factors, including the type of ballast, ballast shoulder, ties, fasteners, rail, track consolidation, etc. The ratio of critical lateral load to axle load (or L/P) may range from a low of 0.3 on newly-worked track, rising quickly to 0.7 under traffic compaction (within perhaps 100,000 gross tons), then stabilizing to 0.9 to 1.0 with continued traffic. This can be affected drastically by longitudinal compressive load with high rail temperatures.

Wheel-climb derailment is generally characterized by the instantaneous single-wheel L/V ratio and the time-duration of the lateral force pulse. Wheel/rail angle of attack, flange angle rail head contour, and the friction coefficient also play a role in the wheel-climb event. Laboratory tests with 1/5th and 1/10th scale models conducted by the Japanese National Railways[30] have shown the wheel-climb L/V ratio to range from values greater than 1.7 for negative angles of attach, down to 0.8 for positive angles of attack greater than one degree. This has been confirmed by test data from high-speed passenger cars, so that the Japanese have established a single-wheel L/V ratio of 0.8 as the conservative safe limit for force pulse durations greater than 50 milliseconds. The safe limit for lateral force pulses of shorter time duration may range up to an L/V ratio of 4.0 at 10 milliseconds.

Finally, the rail rollover (or gage spread) L/V criteria are considered. For an unsupported rail, rollover will occur when the force vector falls beyond the outer edge of the rail base. This L/V ratio will range from 0.4 to 0.7, depending on the rail section and the wheel/rail contact point. If some minimal torsional restraint from fasteners and rail is assumed, a conservative total-truck L/V ratio can be established:

$$L/V = 0.5 + 2300/(\text{wheel load,lb}) \quad \text{maximum.} \tag{10}$$

For this comparative analysis study, root-mean-square L/V ratios were developed for comparing the different vehicles, using an assumed worst-case (flanging) condition.

5.6.3 Vehicle/Track Load Criteria

Several criteria for judging the effects of higher speed operation of vehicles in given track classes were considered. The operational force levels given by computer model response data for the 100-ton hopper car were chosen as baseline maxima for judging the effects of other vehicles, on the premise that these force levels are "acceptable" in practice. To be conservative, levels in the normal operating speed range of 15 to 70 miles per hour were used.

One criterion for establishing a consistent level of damage to the track is to limit any vehicle to speeds below which it develops vertical and lateral track forces equal to, or less than, the baseline values for the 100-ton car. The lighter-weight passenger vehicles would, of course, develop lower total vertical track forces; however, the same level of lateral dynamic force might pose a greater potential of track damage because of this lower vertical stabilizing force level.

A second criterion is based on a consistent dynamic stress level, assuming that vehicle component design is proportioned in strength approximately to the static vehicle weight. Based on this rationale, two ratios were chosen as indices: the vertical dynamic force divided by the vertical static force, and the lateral dynamic force divided by the vertical static force. Vehicle speed limits were then established below which both ratios (if possible) were less than the baseline ratios established for the 100-ton hopper car.

5.7 Results of the Study

5.7.1 Phase I (1972-1973)

In addition to the monthly progress reports, results of the comparative analysis study (Contract DOT-FR-20077) were summarized in a preliminary draft report (dated June 14, 1972) and an Interim Report (dated December, 1972). This phase of the study was completed with the publication of a Final Report[1] (dated March, 1974).

The linear, frequency-domain computer model of a rail vehicle (14 degrees of freedom) was used to compare the vehicle/track interactive dynamics of nine different rail vehicles. Output data were generated in both power spectra and root-mean-square formats representing vertical and lateral wheel/rail loads, L/V ratios, and vertical and lateral accelerations. These data were used to assess the response to track geometry errors in the different track classes versus speed, and to rank-order the different vehicles according to the criteria for ride comfort, safety, and vehicle/track forces.

The accuracy of results summarized in the Final Report depended on the mathematical model (as then structured), the vehicle/track parameters, and the track geometry power spectra. All three of these have been in evolution since the report was published as new information from both analytical and measurement programs have been available. Although these published results are now to some extent outdated, the relative comparison between vehicles shown in Table 5-9 is still valid.

5.7.2 Phase II (1973-1974)

During the early part of this time period, a time-domain model of the Metroliner was run on a hybrid (analog/digital) computer to examine the response of both the standard (GSI truck) and improved (SIG truck) configurations. Both transient track profile disturbances in surface, alignment and cross level, and the rectified sinewave representation of BJR track were used with this model. Results were reported in a monthly progress report[4], dated November 30, 1976.

TABLE 5-9. RELATIVE RANKING OF VEHICLES IN RESPONSE TO TRACK GEOMETRY VARIATIONS*

	(1)(2) Vertical Dynamic Track Force	(1) Lateral Dynamic Track Force	(1) Vertical Car Body Acceler-ation	(1) Lateral Car Body Acceler-ation	(3) Cross-Level Sensi-tivity	(4) Derail-ment Index
100-ton freight car	9	8	9	10	10	8
Metroliner as built	5	2	8	5	3	6
Standard passenger car	2	3	2	1	2	1
DOT test car	1	7	5	8	5	3
Turbo train, 2-axle	6	4	3	6	9	4
Turbo train, 1-axle	3	1	10	9	6	10
Improved Metroliner	4	5	1	2	4	5
E-9 locomotive	8	10**	4	3	7	2
SD-45-2 locomotive	10	9	7	7	8	7
GG1 electric locomotive	7	6	6	4	1	9

1 = best, 10 = worst.

(1) Based on response to poor bolted-rail track, good bolted-rail track, and good CWR track (three categories equally weighted).

(2) Static axle load added as fourth category.

(3) Based on roll response to bolted rail (39-ft), staggered-joint track.

(4) Low-frequency derailment quotient at maximum recommended speed, Class 6 track.

(*) See "Cost-Effectiveness: The Economic Evaluation of Engineered Systems", J. Morley English, Ed., John Wiley & Sons, pp 140-144, pp 156-157, for comments on the "pitfalls and fallacies" of ranking and weighting.

(**) Recent track measurements by Battelle have shown this conclusion to be entirely wrong due to an inadequate description of the wheelset-truck frame (primary) lateral stiffness. A ranking of 6th would be more appropriate.

Two problems were noted from these results: first, the limited travel provided by the improved suspension (SIG truck), particularly in the air-spring secondary suspension, could be exceeded by possible combinations of track geometry error amplitudes, train speed, and wavelengths allowed under the Track Safety Standards. And second, the relatively low torsional stiffness of the suspension could result in large amplitude roll oscillations under certain conditions, particularly at low speeds on cropped (33-ft) rails with half-staggered joints. The need for an auxiliary roll stiffness (a roll bar) was noted.

On the whole, however, the computer simulation showed a superior ride quality that was later confirmed by the test results[31] comparing the two configurations. Computer results showed a strong vertical excitation of the 8-Hz body-bending mode, up to .05 g peak-to-peak on "Class 3" track at 60 mph, due to a strong vertical resonance of the truck frame. This mode was essentially eliminated in the improved configuration.

Additional runs were made with the linear, frequency-domain model during this time period to investigate the ride quality of the Improved Metroliner configuration with added roll stiffness. Runs were also made to simulate the quasi-static unbalance on curves with the lateral suspension shifted into the stiffer region (the stops). Curve unbalances of 3 and 4-1/2 in. were checked at a speed of 110 mph. Although the addition of a roll bar of $30(10)^6$ in.-lb/rad to the truck secondary suspension caused some increase in the lateral acceleration on the passenger due to cross level geometry errors, these acceleration levels (particularly on curves) were still substantially lower than levels calculated for the Standard Metroliner, as shown in Table 5-10.

TABLE 5-10. COMPARISON OF CALCULATED RMS LATERAL ACCELERATION FOR STANDARD AND IMPROVED METROLINER CONFIGURATIONS AT 110 MPH

Track	Vehicle	Lateral Acceleration, g* Over Truck	Car Center
Tangent, Class 6	2	.028	.017
	6	.027	.011
	6R	.027	.014
Curve, 3-in. unbalance	2	.048	.023
	6	.035	.013
	6R	.037	.017
Curve, 4.5-in. unbalance	2	.166	.033
	6	.045	.014
	6R	.047	.020

Vehicle 2 = Standard Metroliner (GSI truck)
6 = Improved Metroliner (SIG truck)
6R = Improved Metroliner, roll bar added
(*) RMS Acceleration in 0.2-13 Hz frequency band.

5.7.3 Phase III (1974-1976)

Technical efforts during the period through June, 1975 were summarized in an Interim Report[2], dated June, 1975. Based on FRA's needs at that time, this effort was aimed primarily at support of the Metroliner Ride Improvement Program (DOT-FR-20049), and also at providing technical assistance to the FRA on the curve negotiation of 6-axle locomotives. An extensive evaluation of the Improved Metroliner configuration with the SIG trucks was conducted by computer simulation. Hunting stability and ride quality were both evaluated, and suspension parameters affecting both were optimized as the result of a parameter variation study. Results are summarized below.

5.7.3.1 Improved Metroliner Ride Comfort. The ride comfort levels of the Metroliner equipped with the SIG trucks were examined by computer simulation(2). Track geometry inputs representing the random variations in surface, alignment and cross level of NEC "Class 6" track were used to provide an excitation of the model at speeds to 150 mph. Both acceleration PSD and British Railways ride comfort indices were used to compare the Standard Metroliner (GSI trucks) and Improved Metroliner with several configurations (parameter settings) of the SIG trucks. A suspension parameter variation study was conducted to allow a choice of optimal parameters providing the best ride quality in the critical 1-10 Hz range and minimizing track dynamic (rms) forces.

The parameter study of the Improved Metroliner was made at speeds of 50, 100, and 150 mph. Lateral secondary stiffness, vertical and lateral secondary damping, vertical primary damping, and secondary auxiliary roll stiffness were varied. In terms of the BR ride index, decreased lateral secondary stiffness improved ride quality markedly, while increased auxiliary roll stiffness improved ride quality modestly. Vertical secondary damping was found to be near optimum, with some decrease in vertical and roll ride quality with change in either direction. Increased vertical primary damping showed improvement in ride quality, while decreased lateral secondary damping showed a very marked improvement in ride quality. Lateral secondary damping appeared to have the greatest effect of any parameter on ride quality, with an optimum value lower than the originally-used setting. Increased vertical primary damping was noted to reduce both the vertical and lateral wheel/rail loads, while reducing the secondary lateral damping indicated some increase in lateral wheel/rail loads.

Based on the results of the parameter study, an optimum vehicle suspension configuration was designed to maximize ride comfort and control the track forces of the Improved Metroliner. Recommended parameters are compared in Table 5-11 with the nominal parameter settings.

To better correlate the analytical results with measurements to be taken on the prototype vehicle by LTV, runs were made with the simulated Improved Metroliner and the optimized configuration in which the accelerations were calculated at the car floor over the front truck. A comparison of BR ride indices calculated for this position (9.5 in. below the c.g.) is provided in Table 5-12 for speeds of 150, 100, and 50 mph.

TABLE 5-11. SUSPENSION PARAMETERS FOR NOMINAL AND OPTIMAL CONFIGURATION OF IMPROVED METROLINER (SIG TRUCKS)

Suspension Parameter	Nominal	Optimal
Vertical primary damping - lb-sec/in./truck	325	800
Lateral secondary stiffness - lb/in./truck	4760	3700
Lateral secondary damping - lb-sec/in./truck	297	190
Vertical secondary damping - lb-sec/in./truck	342	342
Secondary roll bar stiffness-lb-in./rad/truck	24×10^6	92.5×10^6

TABLE 5-12. BRITISH RAILWAYS RIDE INDICES FOR THE IMPROVED METROLINER WITH NOMINAL AND OPTIMAL SUSPENSION PARAMETERS

	Nominal			Optimal		
Speed, mph	Vertical Excitation	Lateral Excitation	Roll Excitation	Vertical Excitation	Lateral Excitation	Roll Excitation
150	1.8	2.3	1.8	1.7	2.1	1.4
100	1.6	2.0	1.5	1.6	1.8	1.3
50	1.3	1.5	1.3	1.3	1.4	1.2

where

1	very good
1.5	almost very good
2	good
2.5	almost good
3	satisfactory.

5.7.3.2 Improved Metroliner Hunting Stability. To verify the stability of the Improved Metroliner with SIG trucks, a secondary (truck) hunting analysis was conducted using the linear model described in Section 5.3. A series of computer runs was made based on vehicle parameters provided by LTV, with variations in these parameters over an expected range to determine effects on hunting stability (linear critical speed). As a result of these runs, only two parameters were found to cause a significant decrease in critical speed: secondary yaw stiffness, and primary longitudinal stiffness. Of course, wheel conicity was assumed constant in these runs, and wheel profile due to wear is recognized as one of the most critical parameters affecting truck hunting.

The dramatic reduction in critical speed of hunting is illustrated in Figure 5-14 for a loss in either secondary yaw stiffness or primary longitudinal stiffness. Two curves are shown for each of two primary longitudinal stiffness values, the hunting critical speed at which the (linearized) truck becomes unstable; and second, a 10 percent modal damping curve, providing a factor of safety.

Two other parameters, secondary lateral damping and primary lateral stiffness, showed slight changes in critical speed due to changes in their values. Negligible changes in critical speed occurred for the following parameter ranges:

(1) Wheelset yaw moment of inertia (8500 - 13,000 lb-in.-sec^2)

(2) Secondary auxiliary roll stiffness (0 - 120 x 10^6 lb-in./rad/truck).

(3) Secondary longitudinal damping (0 - 514 lb-sec/in./truck: 0 - 8.6 percent critical)

(4) Primary vertical damping (0 - 800 lb-sec/in./truck: 0 - 54 percent critical).

(5) Secondary lateral stiffness (300 - 6000 lb/in./truck)

(6) Secondary vertical damping (0 - 500 lb-sec/in./truck; 0 - 27 percent critical).

5.7.3.3 Steady-State Curving Analysis. The prediction of quasi-static lateral wheel/rail forces during curving of rail vehicles is of great interest because the resulting lateral to vertical force (L/V) ratios can contribute to

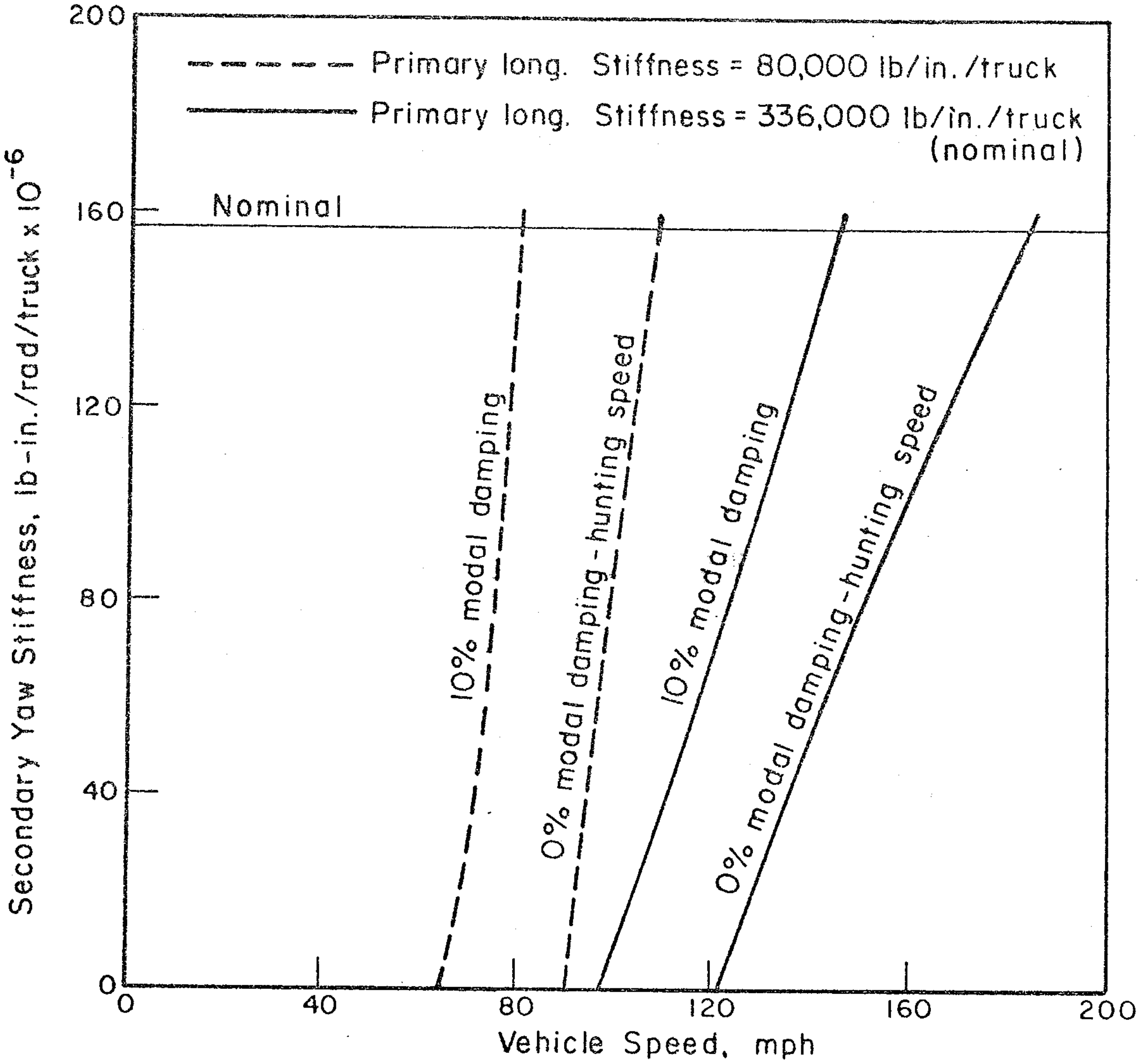

FIG. 5-14. EFFECT OF SECONDARY YAW STIFFNESS ON TRUCK HUNTING OF SIG TRUCK (CONICITY = 0.2)

derailment and rail rollover. As part of this vehicle analytical support program, Battelle was requested to consider the problem of steady-state curving of the 6-axle locomotives recently purchased by AMTRAK. The FRA was provided access by the Association of American Railroads to a computer program for calculating steady-state forces developed by 3-axle locomotive trucks.

A number of combinations of curvature, lateral centerplate force, buff and draft forces were run by the AAR for parameters representing the SDP-40F locomotive with the 3-axle HT-C truck. Net lateral forces at the outer lead wheel versus degree of curvature for two conditions, balance speed and a 7-in. unbalance speed (0.12-g lateral in the plane of the track) are shown in Figure 5-15. A transition point in constraint from all three outer wheels flanging to just the lead and middle outer wheels flanging above four degrees of curvature may be seen in the 7-in. unbalance curve. At balance speed, the lead and middle outer wheels flange throughout the given range. A comparison with published measured data[32] for the 3-axle SD truck is also provided in this figure, showing the median and range of net lateral loads on dry rail, and the range of loads for heavily-sanded rail.

Ratios of lateral to vertical wheel load based on the AAR program results are shown in Figure 5-16 both for the maximum loaded single wheel (the lead outer wheel) and for all three outer wheels. A conservative limit for wheel derailment for steady-state or relatively long duration lateral forces, and a limit for total-truck (rail rollover) lateral forces are shown in the figure.

Maximum L/V ratios for wheel climb range up to 0.53 for the unbalance and curvatures computed, which is well below this conservative limit. Maximum values of L/V for considering rail rollover, based on the summation of the forces on the outer rail under three wheels, range up to 0.17. This is well below a conservative figure of 0.5, based on a rail without added fastener restraint or torsional resistance. Several of the other combinations run on the AAR program are summarized in Table 5-13 for the 4-degree curvature. From the computer results, the buff and draft forces (for the chosen combinations of coupler angles, etc.) appear to make a negligible change in lateral wheel/rail forces and L/V ratios. Reduction in axle/frame lateral clearance at the different axles does have some effect, although not as pronounced as might be expected. With high lateral centerplate force and no clearance on the lead axle, wheel slip apparently

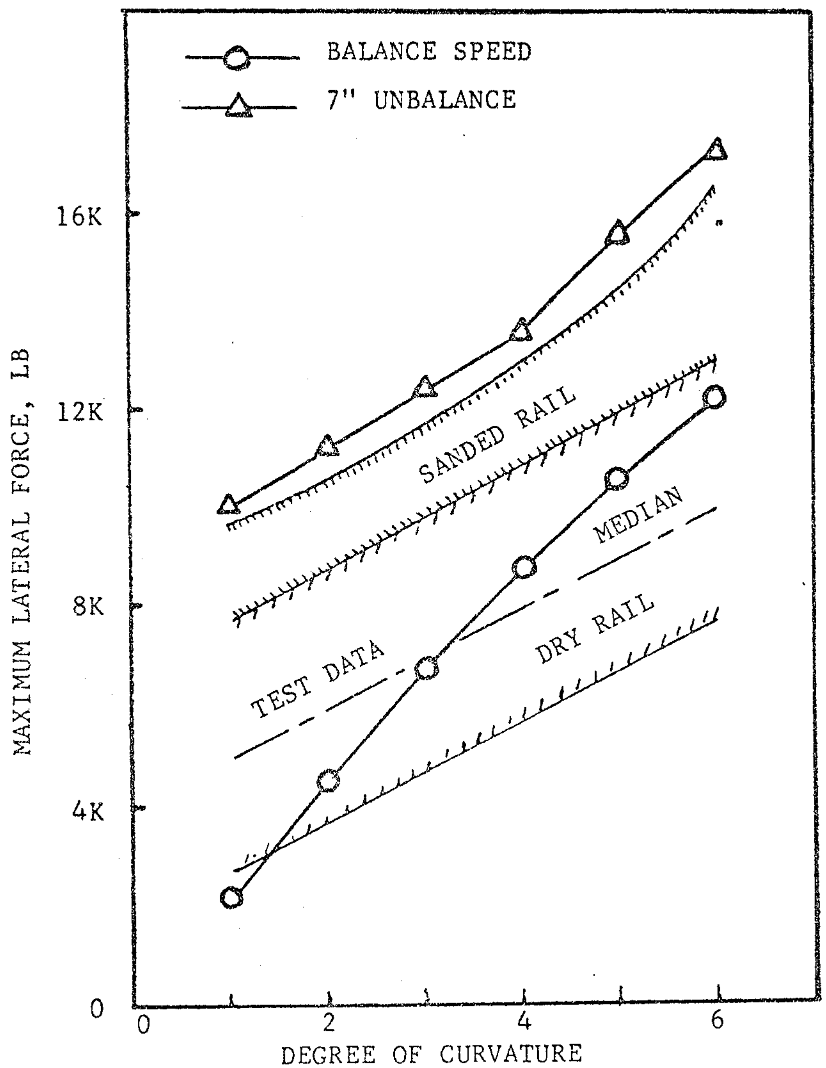

FIGURE 5-15. CALCULATED MAXIMUM LATERAL WHEEL FORCES VERSUS CURVATURE FOR SDP40F TRUCK (RANGE OF TEST DATA FOR SD-45 TRUCK)

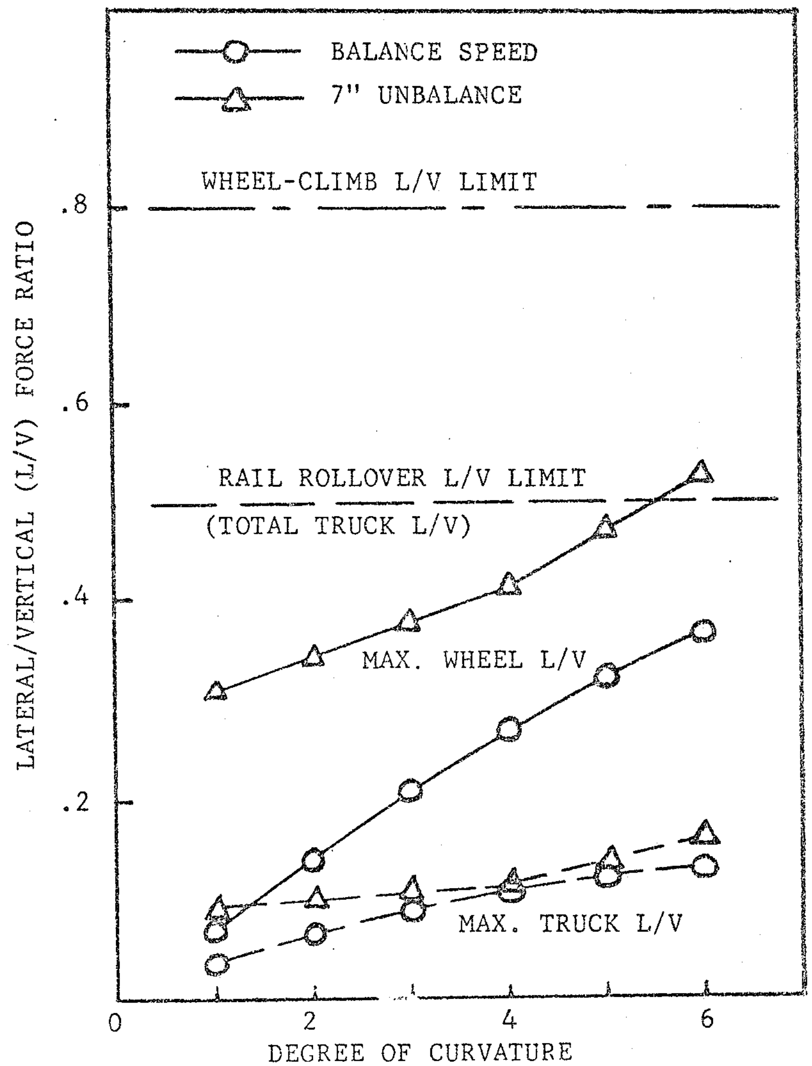

FIGURE 5-16. CALCULATED LATERAL TO VERTICAL FORCE RATIOS FOR LEAD OUTER WHEEL AND TOTAL TRUCK, SDP40F TRUCK.

occurred, and no computer solution was possible. In all runs with the AAR program, the lead truck was found to generate slightly higher lateral forces and L/V ratios than the trailing truck.

TABLE 5-13. MAXIMUM WHEEL/RAIL FORCE, MAXIMUM WHEEL/RAIL L/V RATIO FOR DIFFERENT OPERATING CONDITIONS, LEAD AXLE, LEAD TRUCK OF SDP40F (3-Axle HT-C truck, on 4 degree curve)

Buff-Draft Force, lb	Centerplate Lateral, lb	Lead Outer Wheel W/R Lateral Maximum Force: Flange	Lead Outer Wheel W/R Lateral Maximum Force: Creep	Lead Wheel Net Lateral Force, lb	Maximum L/V Ratio
0	0	15,051	6281	8,770	.270
+15,000	0	15,013	6230	8,783	.270
+30,000	0	14,763	6301	8,637	.266
-15,000	0	15,975	6285	8,690	.267
-30,000	0	14,755	6219	8,536	.263
0	17,000[(a)]	16,935	3358	13,577	.418
+15,000	17,000	16,785	3260	13,525	.416
+30,000	17,000	16,307	3060	13,246	.408
-15,000	17,000	16,729	3345	13,384	.412
-30,000	17,000	16,170	3205	12,965	.399
No Axle Clearance					
A1	0	15,999	5685	9,914	.305
A2	0	15,325	5994	9,330	.287
A3	0	15,051	6281	8,770	.270
A1	17,000	*	* No Solution	*	*
A2	17,000	16,935	3358	13,577	.418
A3	17,000	16,935	3385	13,577	.418
Half Wheel Load, Low Joint					
A1	0	11,737	3120	8,617	.530

(a) 17,000 lb CPL equivalent to 7.2 in. unbalance, 0.12-g in plane of track.

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