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COMPUTER ANALYSIS OF RAILCAR VIBRATIONS

Robert R. Vlaminck

Boeing Vertol Company

SUMMARY

Computer models and techniques for calculating railcar vibrations are discussed along with criteria for vehicle ride optimization. The effect on vibration of carbody structural dynamics, suspension system parameters, vehicle geometry, and wheel and rail excitation are presented. Ride quality vibration data collected on the State-Of-the-Art Car (SOAC) and Standard Light Rail Vehicle (SLRV) is compared to computer predictions. The results show that computer analysis of the vehicle can be performed for relatively low cost in short periods of time. The analysis permits optimization of the design as it progresses and minimizes the possibility of excessive vibration on production vehicles.

INTRODUCTION

Historically, the analysis of railcar vibrations has been limited in scope presumably because of the complexity and time-consuming nature of the calculations required to solve systems with multiple degrees of freedom. Large numbers of simultaneous equations are necessary to accurately and completely describe the dynamic response of a complete vehicle. Detailed carbody dynamic analysis, for example, can best be made by using a finite-element model which may have as many as 3000 degrees of freedom and perhaps an equal number of structural elements.

At the time the bulk of the intraurban and intercity vehicles in use today were analyzed and constructed, the solution of a complex problem of this type was not practical. The high-speed electronic computer and associated programs that have been developed over the past decade now permit a total system approach and provide solutions to railcar ride quality problems on a timely basis; thus analysis is permitted to guide a vehicle design. The technique presented herein involves two steps: a separate solution of the carbody dynamics as a free-free structure and an integrated total system analysis considering the carbody modal solutions along with truck and track dynamics. This approach is recommended since vehicle response occurs over a wide frequency range and results from excitation of rigid body as well as carbody flexible modes. This can best be illustrated in figure 1, a typical railcar ride quality vibration goal, the 3-hr endurance ISO standard, and the predominant modes of response.

The railcars discussed in this paper fall into the main categories of rapid transit and light rail, the latter being the new SLRV now being built by the Boeing Vertol Company for Boston and San Francisco.

It is the intent of this paper to illustrate how computer analyses can be applied to the prediction and optimization of railcar ride quality.

WHEEL/RAIL EXCITATIONS

Dynamic forces which excite the carbody vertically through the primary and secondary suspension systems result from track misalignment and wheel eccentricity. On jointed rail, typically 11.89 m (39 ft) length, the predominant rail-induced excitation frequencies occur at rail joint and twice rail joint frequencies because of geometrical stagger. Wheel excitations occur at the fundamental rotational frequency of the wheel/axle assembly and are significantly higher in frequency than the periodic rail inputs. Since rapid transit and light railcars usually operate below 112 km/hr (70 mph), it is the rail inputs which excite the low frequency (1 Hz to 1.5 Hz) rigid-body suspension modes and wheel eccentricity which predominantly excites the flexible bending modes of the carbody structure. For typical railcars these structural resonances occur above 7 Hz. It should be noted that excitation of the flexible modes also occurs even with concentric wheels when rail joints are impacted. This response is, however, much less severe than at the critical speeds where the wheel rotational frequency coincides with a carbody flexible natural frequency. The variation with vehicle speed of the wheel and rail joint excitation frequencies is shown in figure 2.

CARBODY STRUCTURAL ANALYSIS

One of the first and most important steps in computer modeling of the vehicle is accurately determining the flexible modes of vibration of the carbody structure. This assessment of the dynamic characteristics of the carbody structure is necessary to avoid undesirable vibration under actual operating conditions. Historically, calculations of the natural frequencies and mode shapes of the carbody were performed by representing the structure as a uniform beam or a series of beam elements. This is a very misleading approach since railcar structures are far from being uniform beams because of their many cutouts for doors and windows. Even attempts to represent such a complex structure by a series of beam elements with shear and bending stiffness properties is unlikely to yield correct results, especially when determining higher order bending modes which contribute significantly to vibration at higher vehicle speeds. Effects of local structure such as floor beams, side sills, and attachments of heavy components demand representation of three-dimensional effects, such as section breathing, bulging, or lateral parallelogramming. Traditionally, carbody structural analysis only involved bang tests to determine the fundamental mode with the carbody shell mounted on a simulated suspension system.

A NASTRAN, finite-element, structural representation of the carbody provides a method for accounting for actual details of the structure, including effects of cutouts for doors and windows. This mathematical model of the three-dimensional gridwork of node points, structural elements, coordinates, and mass data that represents the distribution of mass and stiffness in the actual vehicle is used to form mass and stiffness matrices from which natural modes and frequencies are computed. The SLRV carbody NASTRAN structural dynamic model of the SLRV is shown in figure 3.

Dynamic analysis of the SLRV carbody structure was performed to

(1) Detune carbody structure to insure minimal flexible carbody vibration. Those primary sources of excitation in the operating speed range which were avoided by detuning the carbody structure are shown in figure 4.

(2) Optimize any structural changes required to achieve placement of the carbody vertical-bending natural modes above 13-Hz objective with minimum weight penalty. This insures that these modes will not cause amplification of vibration in the important frequency range of 4 to 7 Hz where human sensitivity is greatest.

Since the carbody is suspended on soft secondary springs, the structural natural frequencies can be considered decoupled from the rigid-body suspension frequencies. This allows the carbody to be analyzed as a free-free structure.

Early in the design of the SLRV, calculations using this finite-element model indicated that the first vertical-bending frequency, the primary source of the carbody flexible vibration, was only 8 Hz with the structural members sized on static load considerations. This meant that the frequency of the first harmonic of wheel rotation, a major source of excitation, would coincide with this natural frequency in the operating speed range and would result in high vibration throughout the car. Examination of the modal deflection data from the finite-element analysis indicated that the two large cutouts required for the center doors contributed most to this problem. Several structural modifications were evaluated on the computer.

The lightest and simplest design which met the 13-Hz frequency objective involved the designing of a truss at the rear of the longitudinal equipment enclosure compartments mounted on both sides of the car underframe. This arrangement provided two longitudinal beams approximately 0.6096 m (24 in) deep running almost from the forward bolster to the articulation bolster. The analytical results were available rapidly and the necessary structural design changes incorporated to ensure that the required natural frequency goal for optimum ride quality was achieved.

RIDE QUALITY VIBRATION MODELS

Once the carbody natural frequencies, mode shapes, and modal masses are determined, a ride quality computer model similar to that shown in figure 5 can be used to optimize suspension system parameters and predict vehicle vibration levels at any desired carbody location. This model was developed for the prediction of vibrations which affect ride quality and for the prediction of primary and secondary suspension dynamic loads. It considers the dynamics of the entire vehicle (truck and carbody) which permits a total systems analysis. Historically, truck and carbody parameters evolved independently, and consideration of the dynamic characteristics of the total vehicle was neglected. To accurately represent the important modes of vibration, appropriate car and truck geometry along with the following degrees of freedom should be included as a minimum:

(1) Carbody: Vertical, pitch, and roll

(2) Flexibly mounted body component: Vertical, pitch

(3) Truck: Vertical, roll, independent side frame pitch

(4) At least two carbody flexible modes.

As main line vehicle speeds increase to 241 km/hr (150 mph) wheel excitation frequencies will occur in the frequency range above 15 Hz. This implies that higher order bending modes will be significant contributors to carbody vibration and must be considered in the analysis. The carbody flexible modes can be described to the model from either finite-element analyses such as NASTRAN or shake test data. The flexibly mounted body components are included to analyze the effect on ride quality of massive sprung components such as the 1587.6 kg (3500 lb) motor alternator on the SOAC or energy storing flywheels on the Advanced Concept Train (ACT). Independent truck side frame pitch, coupled through a torsional spring, is necessary to model trucks which equalize by mechanical pivots or truck frame flexibility.

Elastomer springs and dampers should be made nonlinear by specifying an appropriate table hookup for each element. This is important when analyzing suspension configurations employing elements which can be deflected through large amplitudes or are made highly nonlinear after small initial linear deflections. A common example of this type of suspension arrangement is shown in figure 6. In addition to the features described above, the capability to excite the vehicle at each wheel/rail interface with phased displacement inputs is required. These inputs should be sinusoidal excitations to determine the vehicle acceleration transfer functions and track dynamic profiles to simulate actual running conditions.

The equations of motion for the figure 5 model described above were derived using LaGranges' method and have been programmed at Boeing Vertol on an IBM Continuous System Modeling Program (CSMP) and on a Xerox Sigma 9 machine using the SL1 language.

RESULTS OF PARAMETRIC STUDIES

To illustrate the importance of accurately simulating not only the amplitude and phase of rail excitation, but also the vehicle geometry and suspension characteristics, selected ride quality computer predictions of carbody vertical acceleration over bolster are presented in figures 7 to 11. These results are for a 22.86 m, 31752 kg (75 ft, 70000 lb) rapid transit car having a first vertical bending frequency of 5.88 Hz, concentric wheels, traveling 128.72 km/hr (80 mph) on 11.89 m (39 ft) jointed rail. The computer model used was similar to that shown in figure 5. The intent is to illustrate, for example, the integrated approach to evaluate the effect of staggered rail joints. This requires consideration of carbody and truck modes. For each two seconds of analytical data shown, the dynamic track profile was phased to all eight wheels based on vehicle speed, truck wheelbase, and truck spacing.

Figure 7 compares over bolster vertical acceleration levels calculated by exciting the model with measured right and left dynamic track profile excitation to acceleration levels determined where the left rail profile is assumed similar to the right rail profile. From these data several significant conclusions can be determined. Although the left rail profile looks significantly different than the right, the wavelength content (11.89 m (39 ft)) and overall amplitude levels are similar. The large spike at 1.75 seconds occurs because of an anomaly in the right rail characterized by a dip between rail ends. The mixed frequency of the waveform is composed of 3-Hz and 6-Hz vibration corresponding to rail joint and twice rail joint excitation frequencies at 128.72 km/hr (80 mph). (See figure 2.)

The effect of the 5.9 m (19.5 ft) stagger between rail joints can be seen in figure 8. A comparison is made between acceleration levels calculated using two identical profiles; one having the right rail mathematically displaced 5.9 m (19.5 ft) to shift the right to left input phasing. These data show that the predicted amplitude with the rail excitation not staggered is in error by as much as 100% and does not have twice rail joint frequency content.

Figure 9 presents a comparison of vibration levels calculated using a measured track dynamic profile to predictions using an "idealized" jointed rail profile. Both the amplitude and frequency content of the waveform are similar except for the previously discussed spike at 1.75 seconds. The importance of considering rail input phasing is well illustrated in figure 10 where the effect of staggered versus no stagger idealized rail is shown. In this case the amplitude and frequency content is significantly in error where input phasing is neglected.

An example of the effect of truck geometry can be seen in figure 11. Comparison is made between a truck having an 208 cm (82 in) wheelbase and an identical vehicle modeled with the wheelbase assumed equal to zero. Over bolster acceleration levels for the zero wheelbase case are approximately twice those calculated for the 208 cm (82 in) wheelbase. This attentuation in response occurs because the resultant input amplitude to the secondary suspension springs is reduced as each wheelset encounters a rail joint.

COMPUTER ANALYSIS OF SOAC RIDE VIBRATIONS

Background

The SOAC was developed to demonstrate the state-of-the-art and was assembled from available carbody structure and truck components using analyses common to the industry at that time. The car has demonstrated good riding qualities in testing at Pueblo over several combinations of jointed and welded rail and received favorable comment in public service at NYCTA, MBTA, CTS, CTA, and SEPTA. The ride demonstrates the advantage of an application of modern state-of-the-art in rubber chevron primary suspension and airspring secondary suspension. Analyses of the SOAC ride, however, indicate further improvements are possible by tuning the carbody and trucks for compatibility.

Ride quality, carbody shake tests, and wheel concentricity tests were performed on the State-Of-the-Art Car (SOAC) at the High Speed Ground Test Center at Pueblo, Colorado. These ride quality tests conducted on the welded fail sections of the 128.72 km/hr (80 mph) UMTA test oval indicated that there were noticeable vertical car floor vibrations near 72.4 km/hr and 128.72 km/hr (45 mph and 80 mph). Test data at empty car weight show that near 128.72 km/hr (80 mph) the floor vibrations are predominantly 15 Hz and that near 72.4 km/hr (45 mph) the carbody vibrations are predominantly in the 7.8 to 8.2 Hz frequency range.

Shake testing of the SOAC indicated that there are two vertical carbody flexible modes of interest occurring at frequencies below 20 Hz, the first vertical bending made at 8.1 Hz, which is a characteristic of the primary structure of the car, and a 15.2 Hz higher order mode involving vertical bending of the underfloor lateral motor alternator support beams and the primary side sill structure in the area of the rear door cutouts.

The 1587.6 kg (3500 lb) motor alternator, located at mid car, is flexibly mounted on elastomers giving an uncoupled vertical frequency of 15 Hz. This counting causes the motor alternator to act as a highly damped dynamic absorber, attenuating response from the second bending mode at 15.2 Hz. This was confirmed by mechanically "locking out" the elastomer mounts during the shake test. Figure 12 compares frequency response curves with the motor alternator flexibly and rigidly mounted to the underfloor structure.

Near 72.4 km/hr (45 mph) a resonant condition exists where the wheel rotational frequency coincides with the first vertical bending frequency at 8.1 Hz. Acceleration data shows that vertical motion at the forward end of the car is out of phase with vertical motion at the mid car location. This is expected since these two positions are located on opposite sides of the node of the first mode. Near 128.72 km/hr (80 mph) the wheel excitation frequency coincides with the 15 Hz higher order body bending mode resulting in carbody vibration throughout the car.

Data at speeds slightly higher and lower than 128.72 km/hr (80 mph) and 72.4 km/hr (45 mph) show that vibration levels are reduced, and this is expected since the wheel excitation frequency is then separated from the carbody bending frequencies. From these data, it could be concluded that wheel excitations resulting from wheel eccentricity provide significant harmonic inputs in the frequency range of the two carbody flexible modes. Figure 13 shows the frequency spectrum for the SOAC vehicle.

Wheel concentricity tests indicated that wheel out-of-round was typically 0.018 cm (0.007 in) with a maximum of 0.025 cm (0.010 in).

Test Results and Analytical Predictions

SOAC ride quality data collected at 72.4 km/hr (45 mph) and 128.72 km/hr (80 mph) is shown in figure 14 and compared with analytical predictions. Using the ride quality computer model shown in figure 5, calculations were made at 128.72 km/hr (80 mph) using welded rail dynamic track profile excitation superimposed with 0.025 cm (0.010 in) wheel out-of-round excitation at the wheel rotational frequency. The predicted vertical acceleration levels agree well with the test data, both amplitude and frequency content. If the carbody second flexible mode had not been included in the model, only the low-frequency response at the rigid-body frequencies would have been predicted. These data clearly show the importance of analytically representing the carbody dynamics including flexible and rigid-body modes on the suspension system, truck dynamics, and rail plus wheel excitations.

It should be noted that the State-Of-the-Art Car was developed from an existing carbody structure and truck components and was not optimized by extensive computer analysis.

COMPUTER ANALYSIS OF SLRV RIDE VIBRATIONS

Background

The SLRV is a 32659 kg (72000 lb), three truck, articulated car having a maximum speed of 96 km/hr (60 mph). Vertical ride quality data was collected on this vehicle at the Boeing Vertol jointed rail test track in the 32.2 km/hr (20 mph) to 64.4 km/hr (40 mph) speed range. Previous NASTRAN dynamic analysis of the carbody structure indicated that the first vertical bending frequency at 13 Hz was well above the wheel rotational frequency throughout the operating speed range; thus minimum flexible response is insured.

Test Results and Analytical Predictions

SLRV ride quality data, mean vertical acceleration at station 55 on the car centerline as a function of carspeed, is shown in figure 15 and compared to analytical predictions. Calculations were made at 8 km/hr (5 mph) increments from 32.2 km/hr (20 mph) to 64.4 km/hr (40 mph) using the analytical model shown in figure 16 and jointed rail dynamic track profile excitation. This model includes the three rigid-body vertical/pitch modes associated with an articulated car. Predicted vertical acceleration levels agree well with the test data over the speed range investigated and showed that vibrations at the rigid-body suspension frequencies dominated the response and were maximum near 56.3 km/hr (35 mph). At this speed the rail excitation at joint frequency is close to the car out-of-phase rigid-body pitch frequency on the secondary suspension; the amplitude of response being limited by the orifice damping provided by the airsprings.

Figure 17 compares ride quality vibration predictions on welded rail to measurements collected during tests conducted at Boston. These data are presented against the SLRV ride quality goal and show the low vibration levels throughout the entire operating speed range. This results predominately from the low-frequency secondary suspension and the detuned carbody structure.

CONCLUDING REMARKS

The computer technology which has been developed over the years primarily for aerospace applications provides the ability to solve many railcar ride quality problems that only a decade ago were treated with oversimplified analyses. It is now a reality that these computations can be performed accurately, for relatively low cost, in short periods of time prior to the detail design of the carbody structure and suspension components. Experience in applying these new analytical tools is still a prerequisite for success but the computer models described in this paper permit analysis to impact a design, reduce costs, and lower the possibility of problems on the production vehicle.

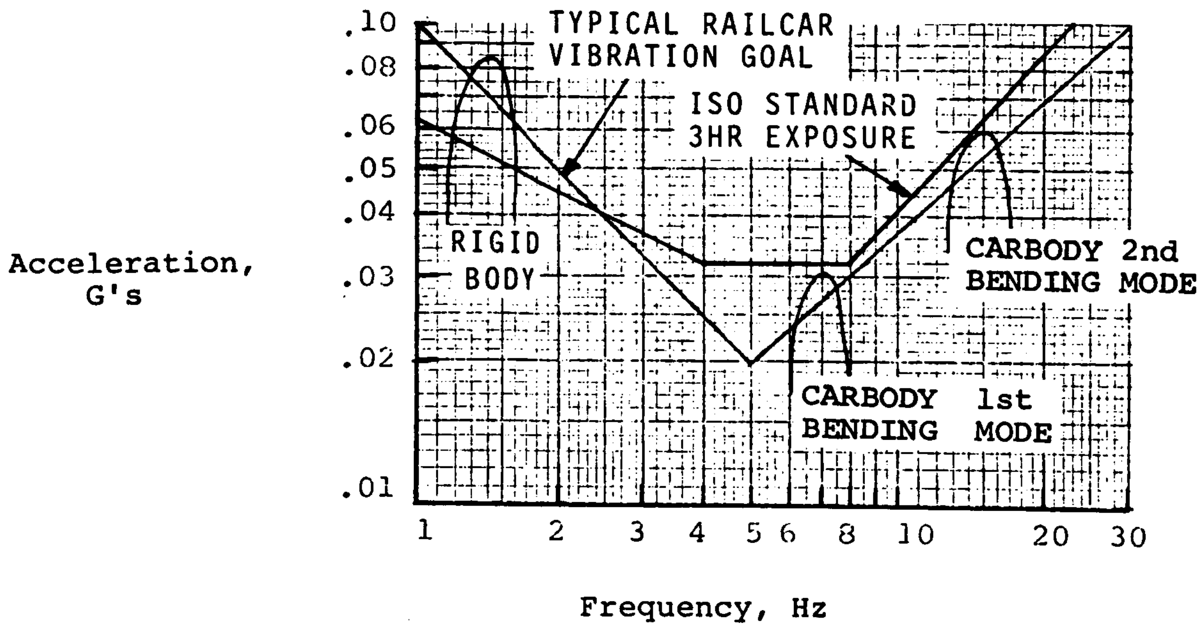

Figure 1.- Typical ride quality vibration goal and modal responses.

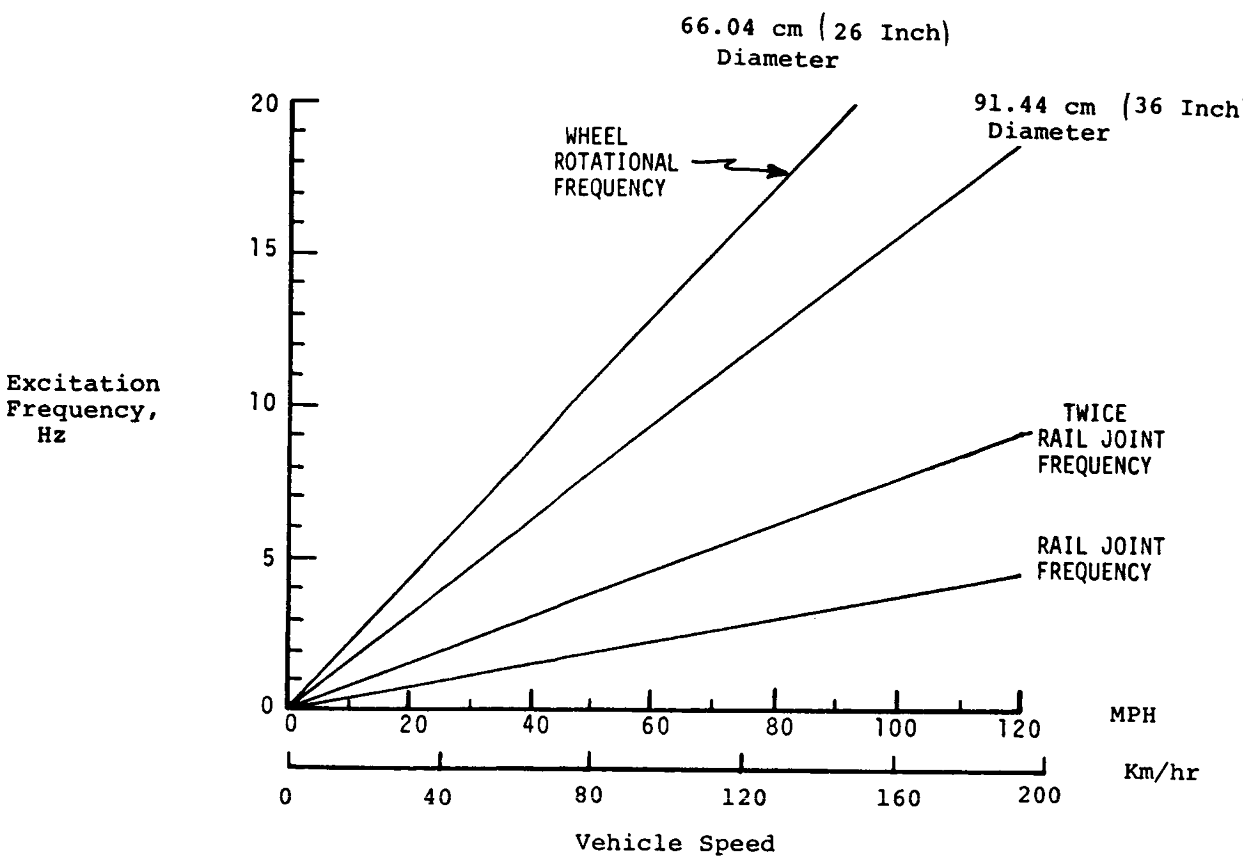

Figure 2.- Wheel/rail excitation spectrum.

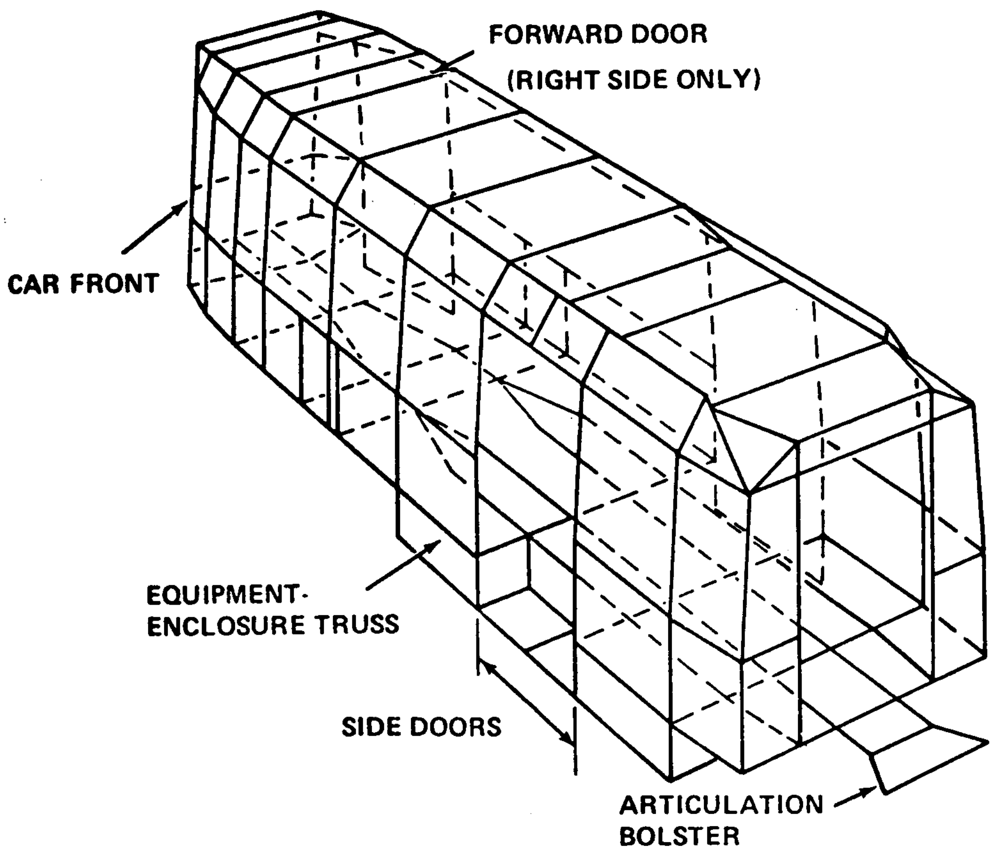

Figure 3.- NASTRAN idealization of SLRV car body.

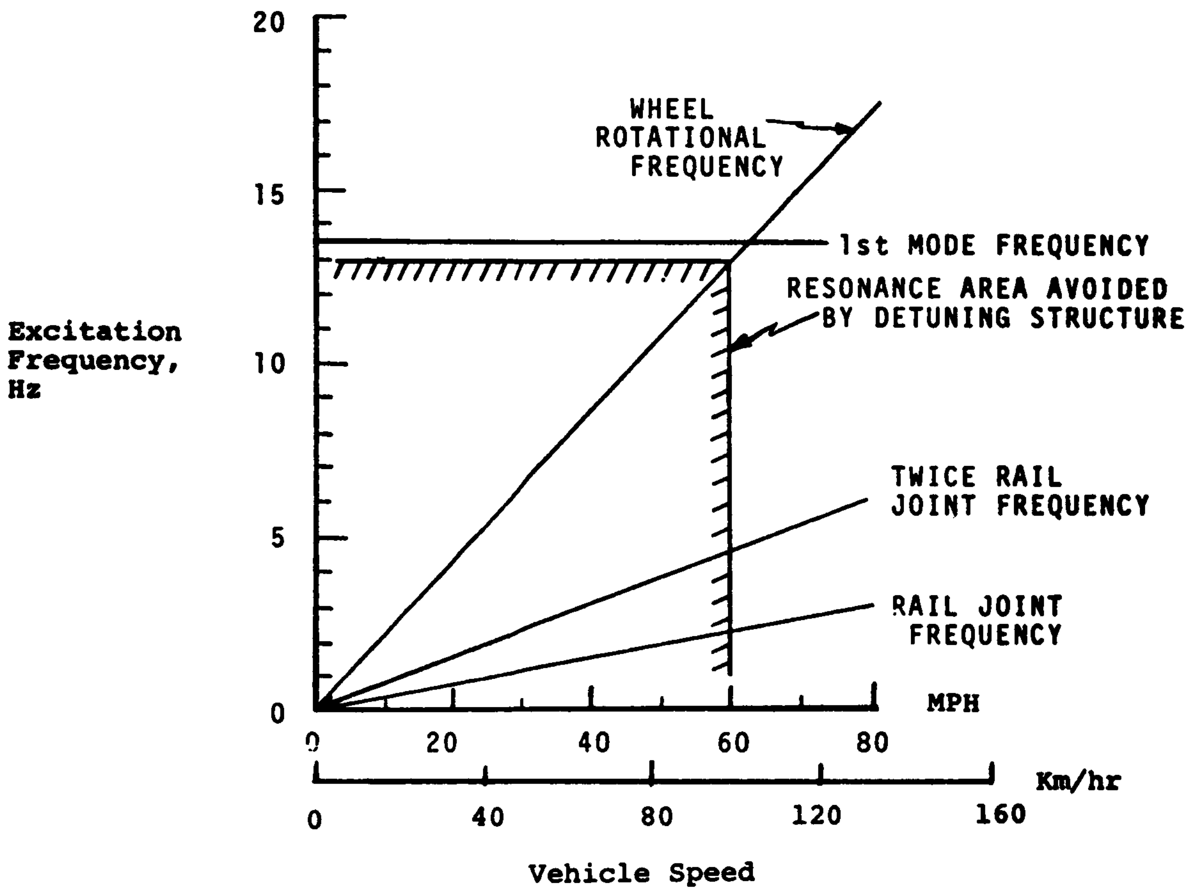

Figure 4.- SLRV wheel/rail excitation spectrum.

- 19 DEGREES OF FREEDOM
- TWO CARBODY FLEXIBLE BENDING MODES
- NONLINEAR SUSPENSION ELEMENTS
- FREQUENCY SENSITIVE SECONDARY DAMPING
- TRUCK FRAME FLEXIBILITY
- PHASED TRACK PROFILE AT EACH WHEEL

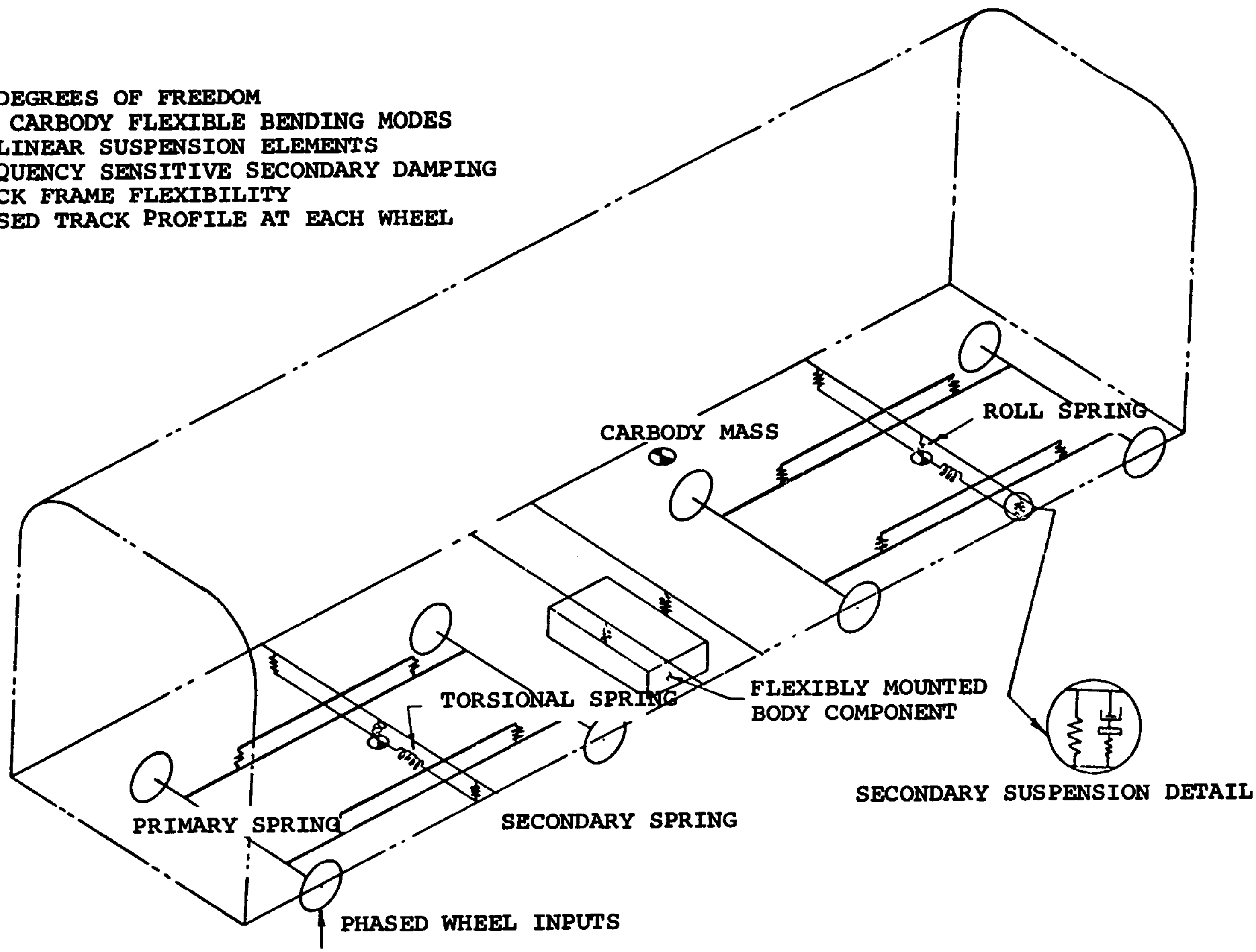

Figure 5.- Railcar ride quality vibration model.

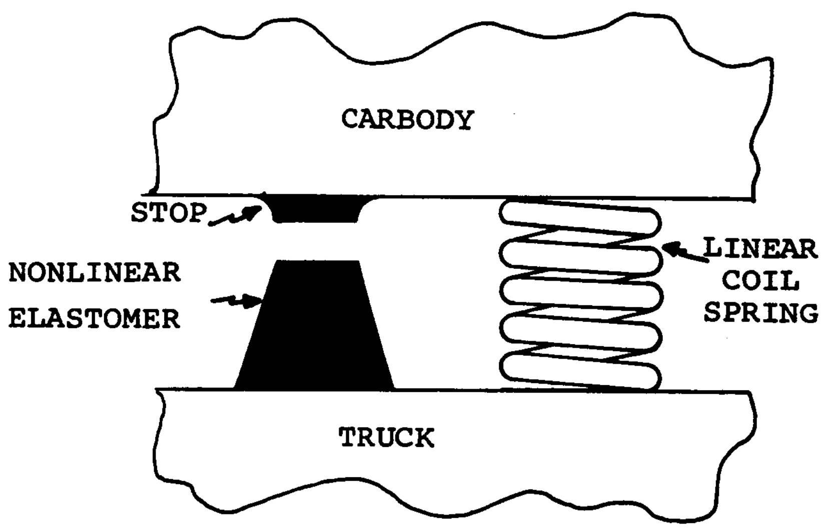

Figure 6.- Typical nonlinear secondary suspension system.

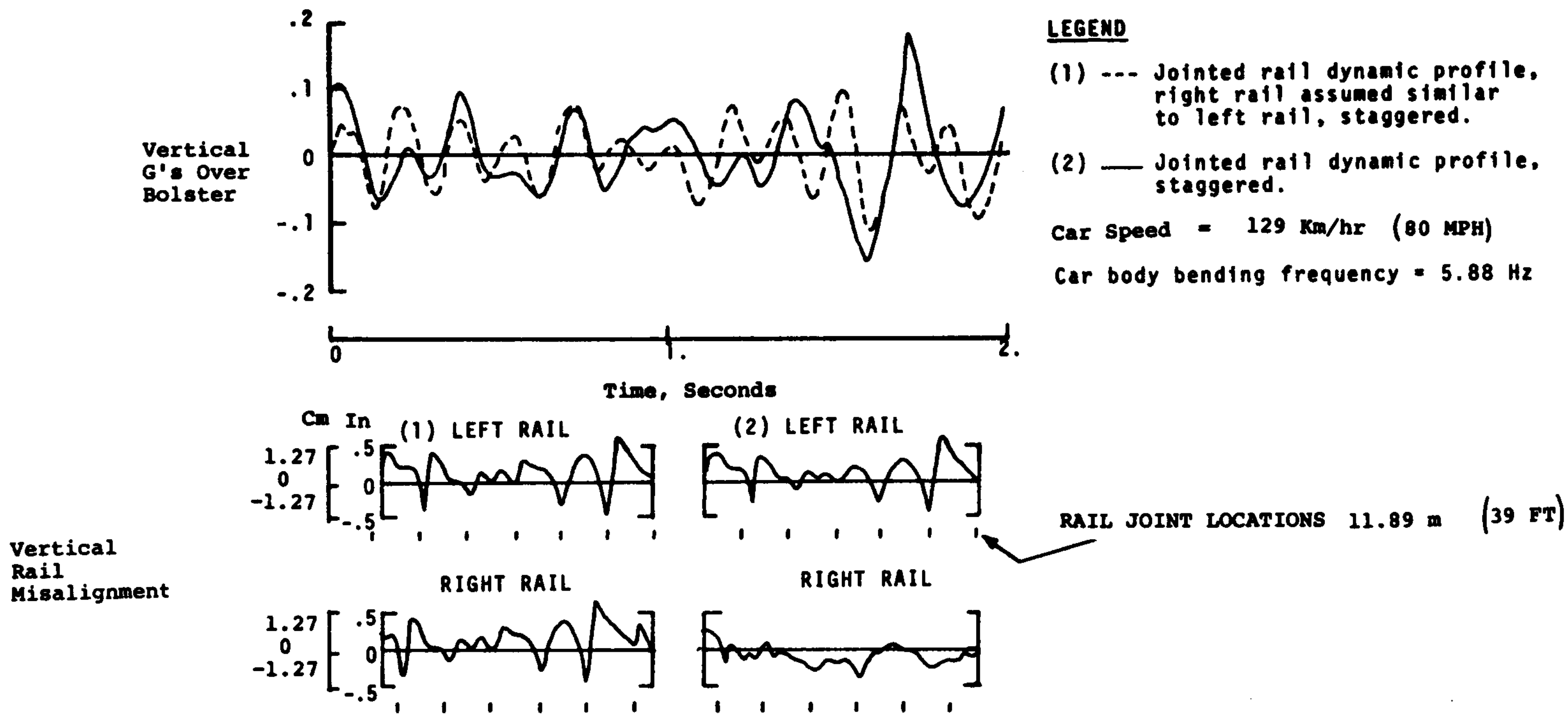

Figure 7.- Predicted car body vertical acceleration over bolster. Measured profiles.

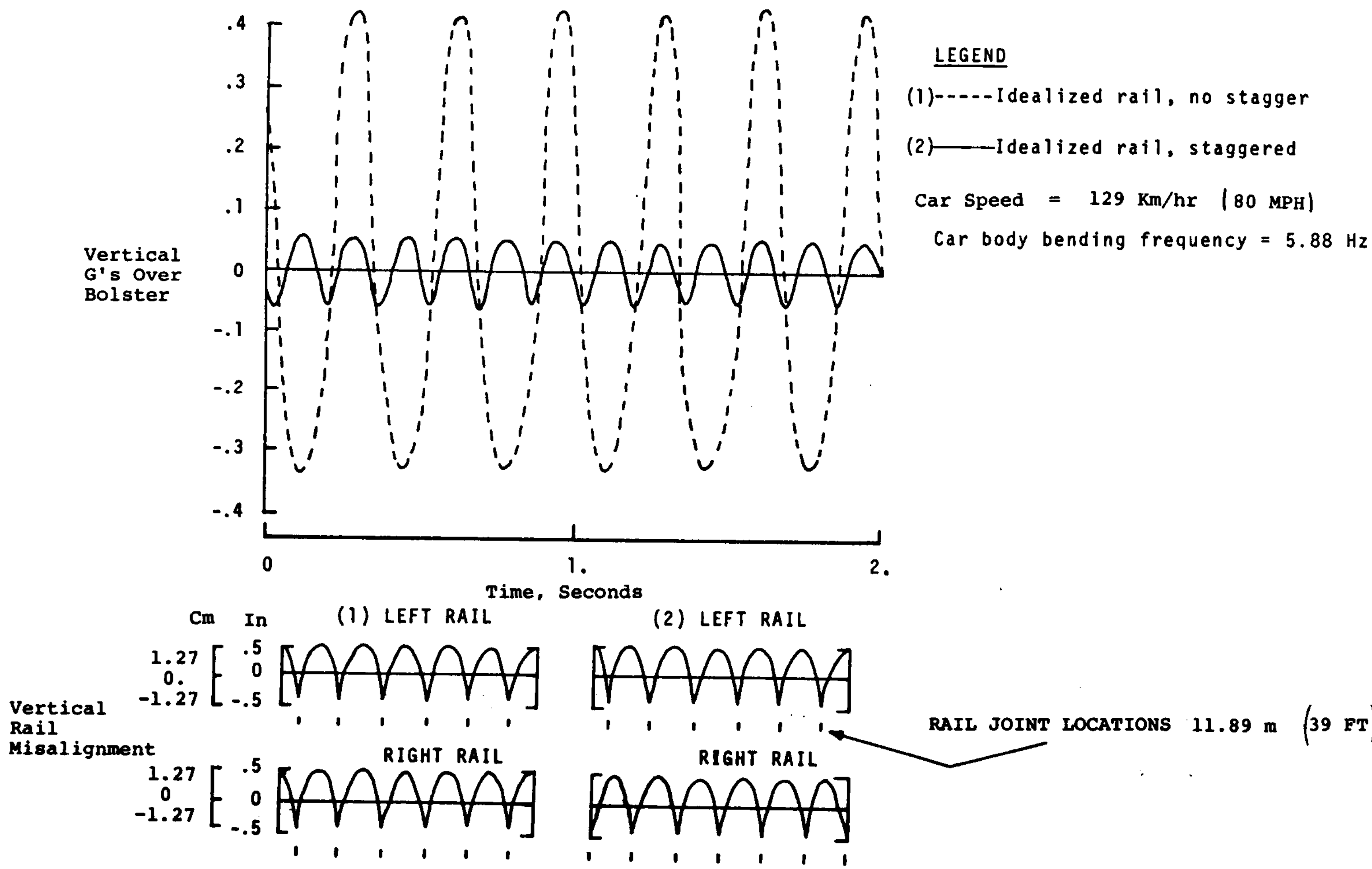

Figure 8.- Predicted car body vertical acceleration over bolster. Stagger.

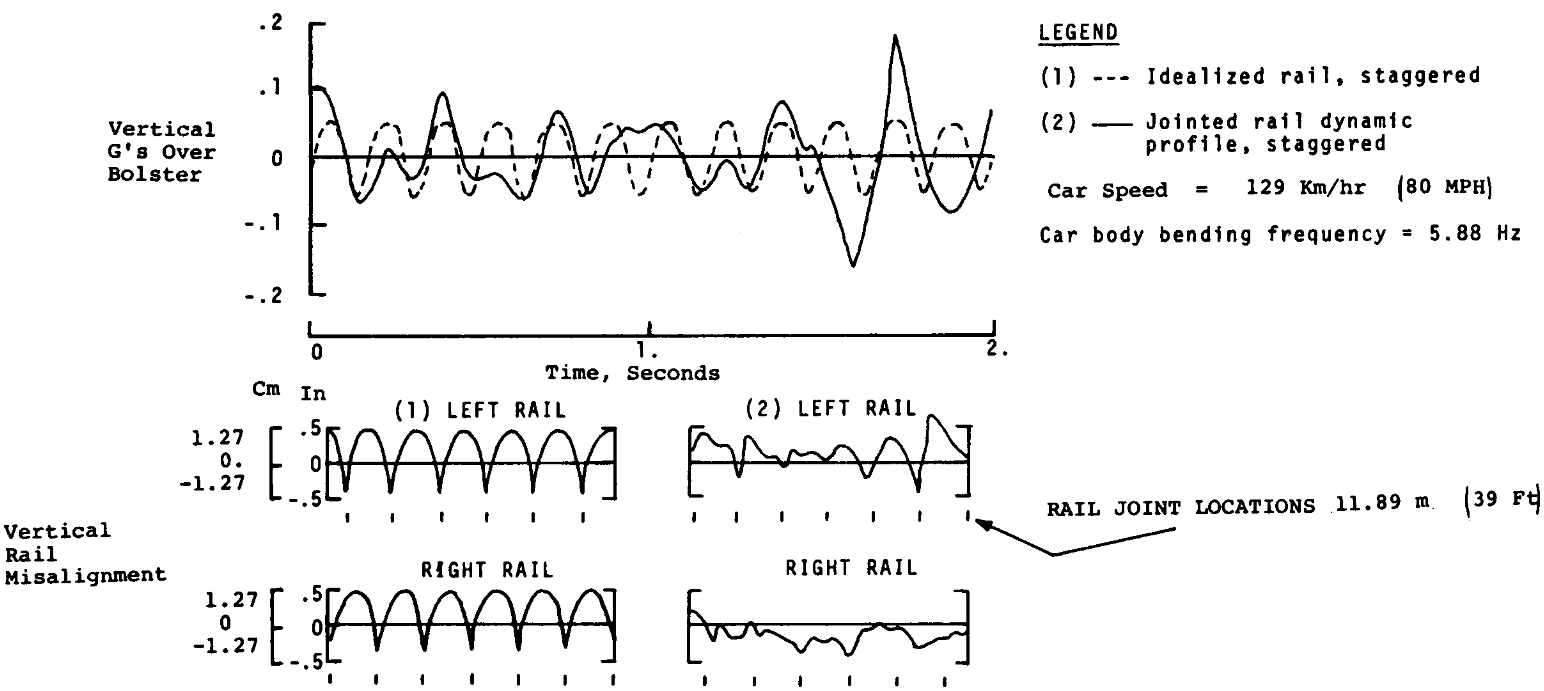

Figure 9.- Predicted car body vertical acceleration over bolster. Measured values compared with those of ideal profile.

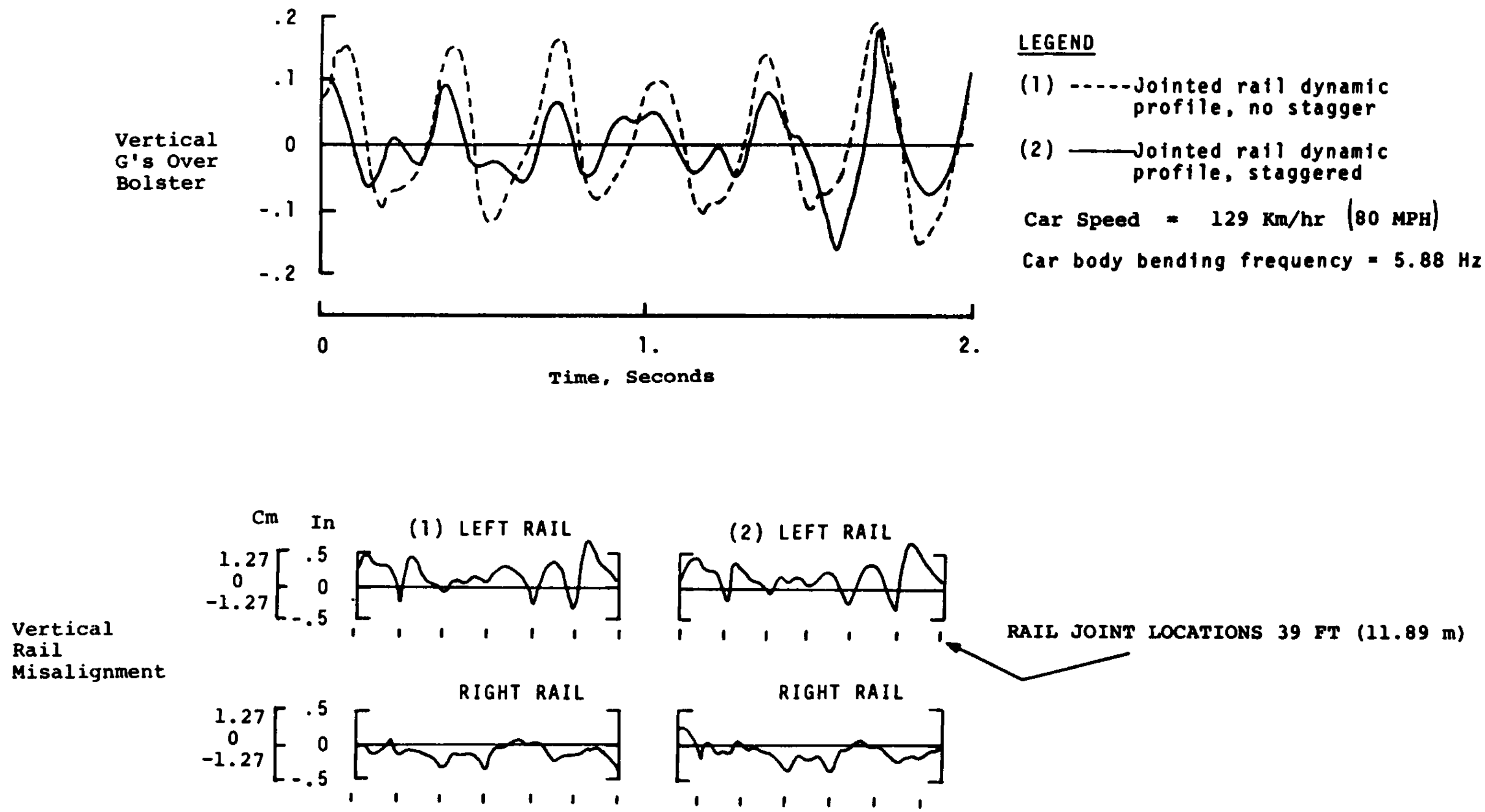

Figure 10.- Predicted car body vertical acceleration over bolster. Stagger.

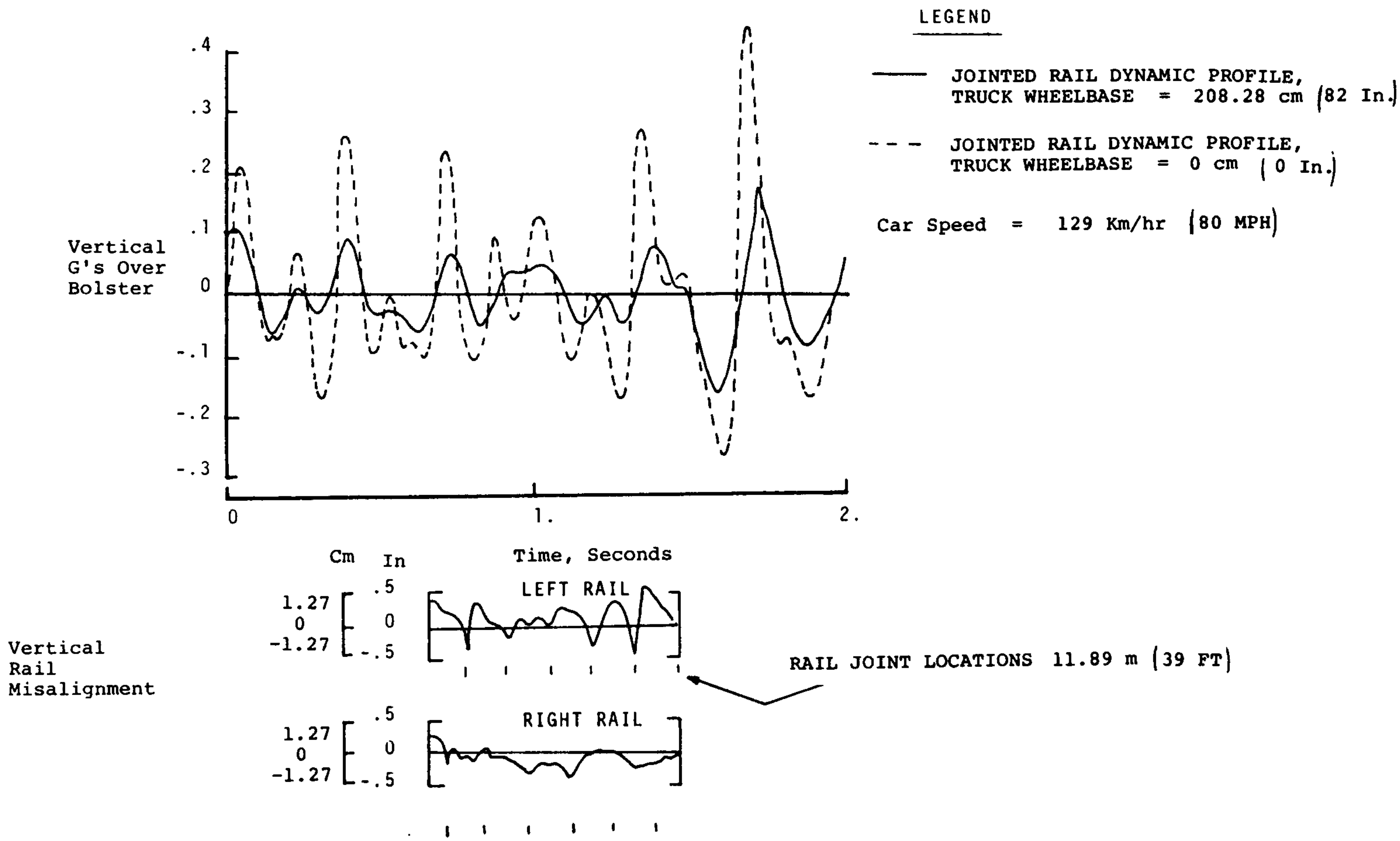

Figure 11.- Effect of truck wheelbase on car body vertical acceleration over bolster.

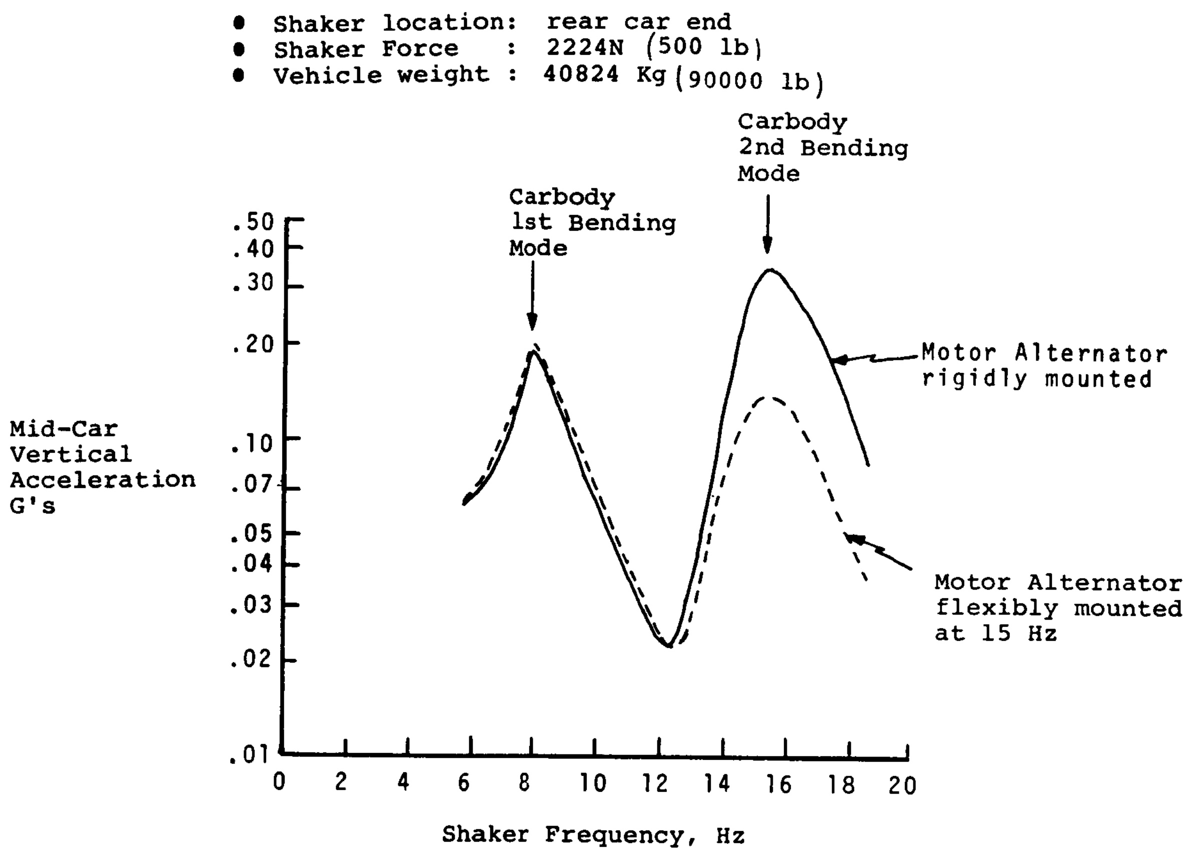

Figure 12.- SOAC shake test frequency response curves.

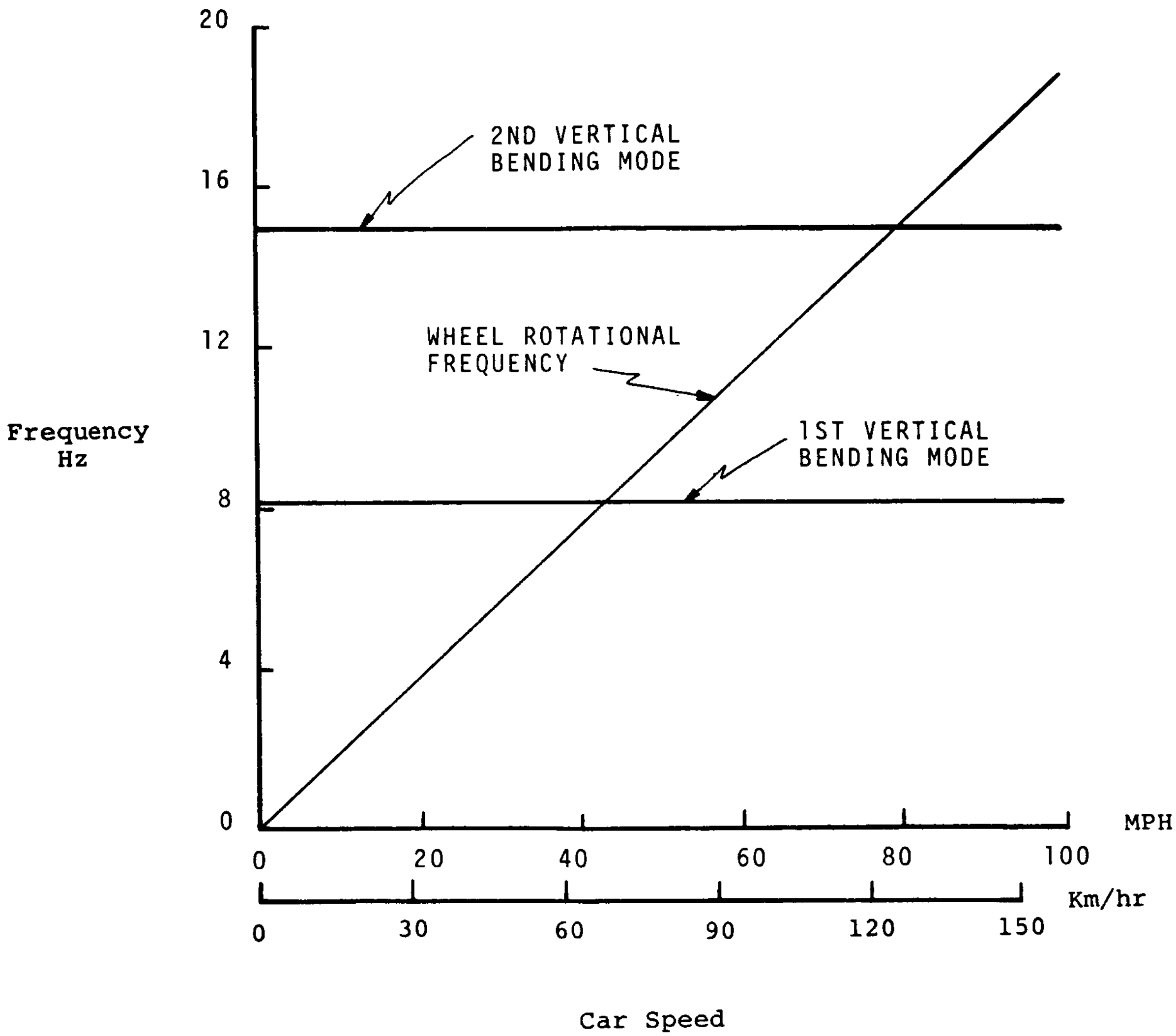

Figure 13.- SOAC frequency spectrum.

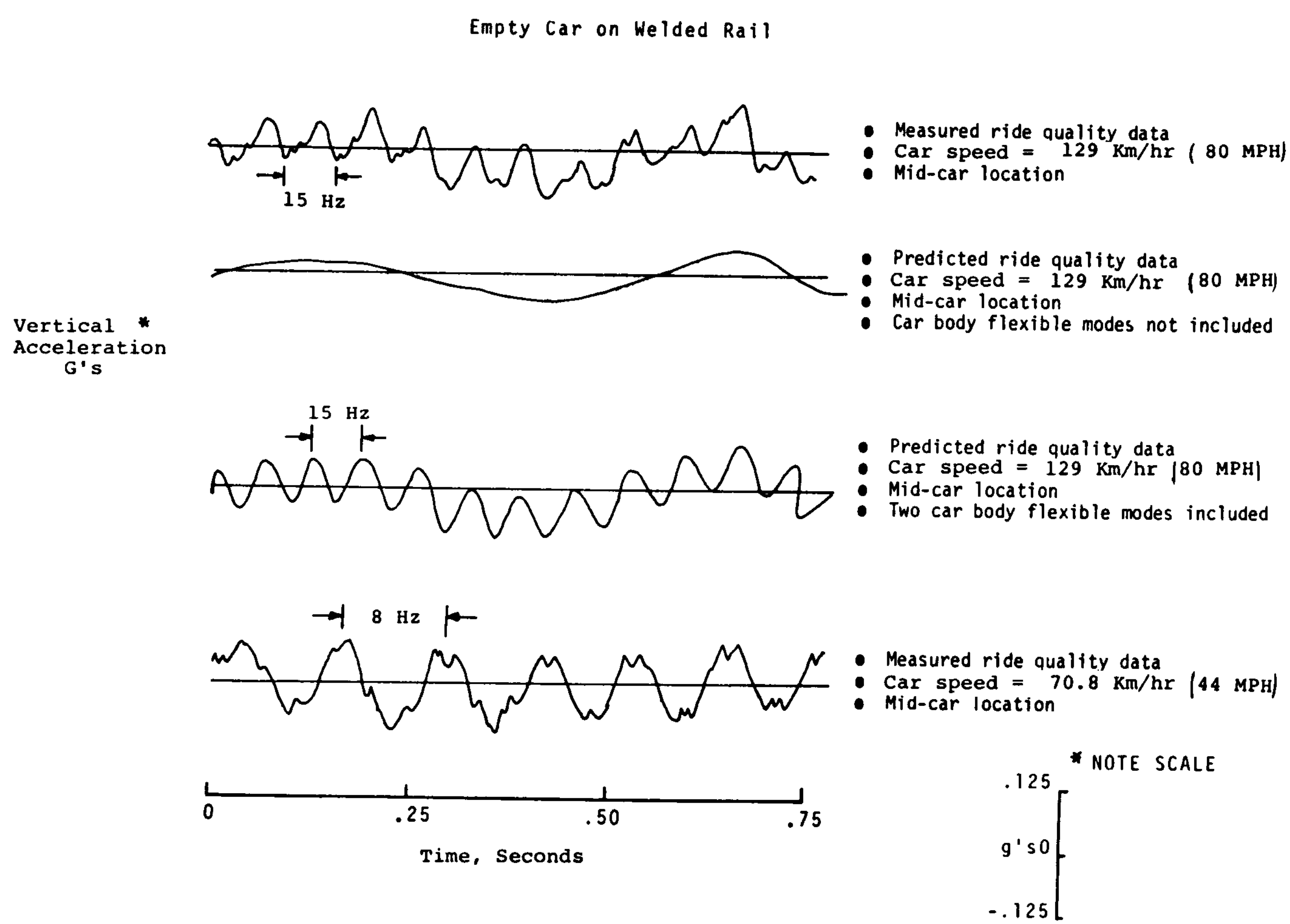

Figure 14.- SOAC ride quality vibration data.
Test and analytical predictions.

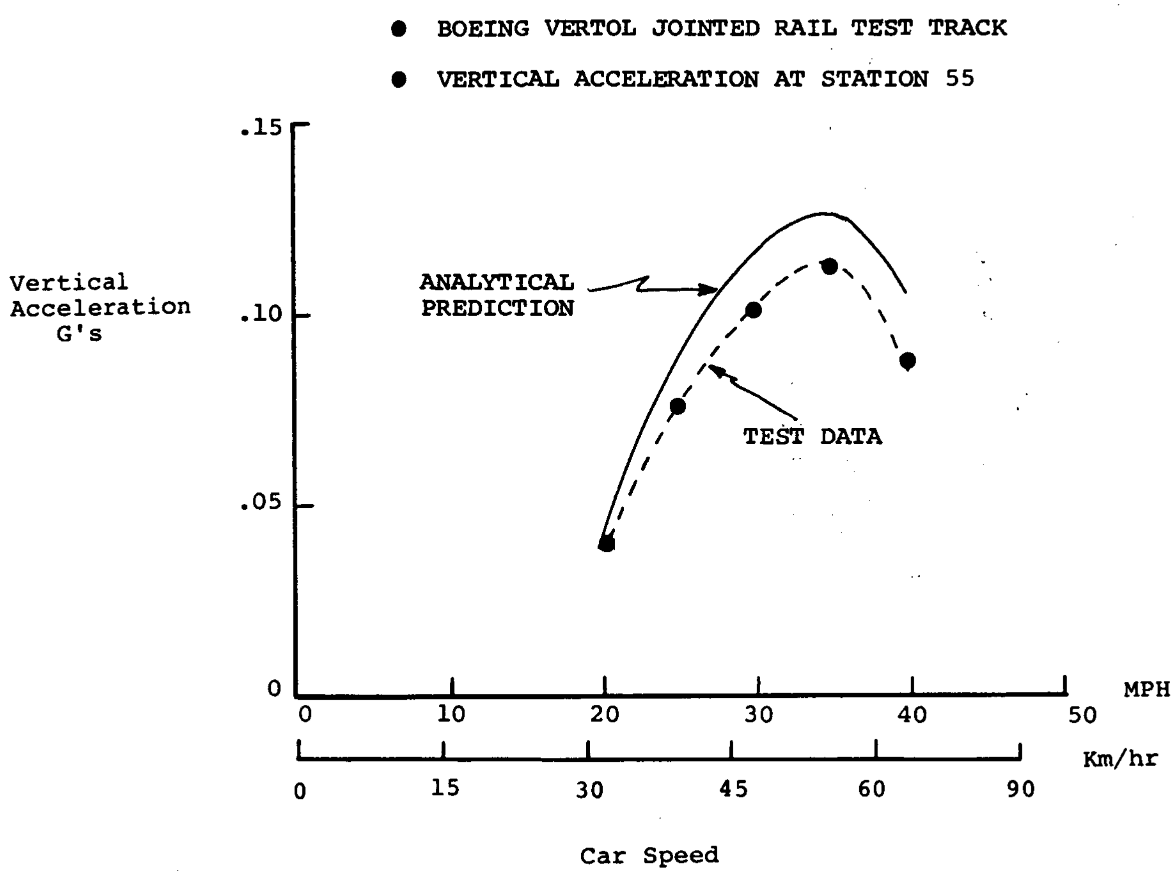

Figure 15.- SLRV ride quality vibration data. Test and analytical predictions.

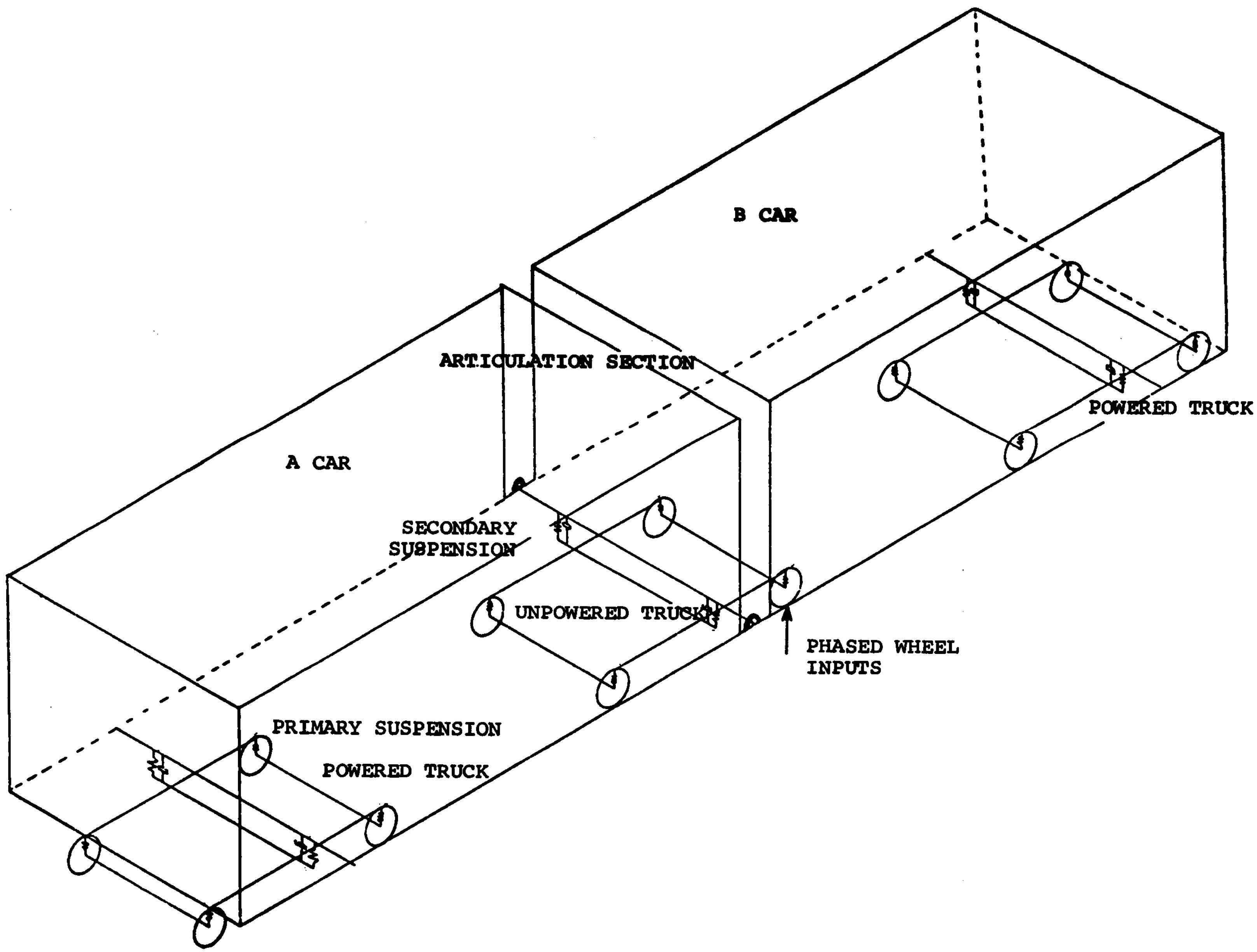

Figure 16.- SLRV ride quality vibration model.

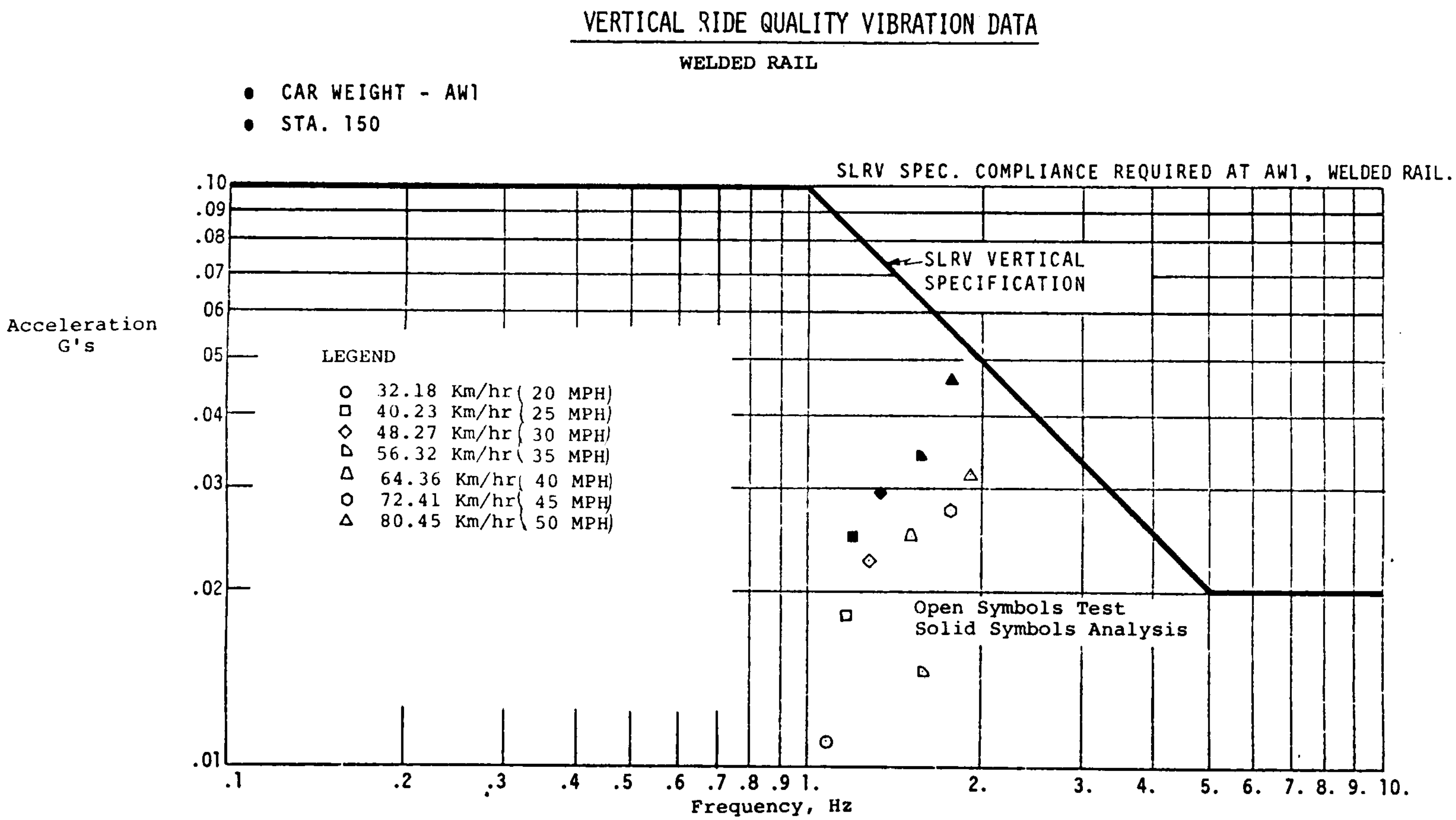

Figure 17.- SLRV ride quality vibration data.
Test and analysis.