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INFLUENCE OF TIRE/WHEEL NONUNIFORMITIES ON HEAVY TRUCK RIDE QUALITY

MVMA Project #1163

T.D. Gillespie

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16. Abstract

Nonuniformities in the tire/wheel assemblies of heavy trucks, such as mass imbalance or geometric runout, add to the ride vibrations on the road at the rotational frequency of the wheel and harmonics thereof. On-the-road measurements of the ride degradation attributable to specific nonuniformity conditions are time-consuming and difficult because of the inability to control the individual nonuniformity conditions. Accordingly, an experiment was performed in which a cab-over-engine tractor was mounted on a hydraulic road simulator and exposed to a typical road roughness input, together with simulated tire/wheel nonuniformity inputs in the vertical direction. Subjective ratings of the ride were obtained from a 10-person jury for 100 test conditions covering different amplitudes, frequencies, wheel positions, and combinations of nonuniformities.

The jury ratings exhibited the greatest sensitivity to second harmonic inputs at the rear axle, whereas their sensitivity to first and third harmonics were nominally equal, and somewhat lower. For inputs at the front axle, a sensitivity to various nonuniformity combinations (i.e., different harmonics or wheel positions) was sometimes dependent on phasing; but, as a rule, the ride degradation at the most critical phase condition was nominally equivalent to the sum of the degradations associated with the individual nonuniformity components.

These findings were used to predict the ride degradation arising from the radial nonuniformities in the wheels of the test tractor. The combined effect of first- through fourth-harmonic radial nonuniformities totaled approximately one point on a 0-10 ride scale, when first harmonics were minimized by balancing and match mounting. Under these conditions, the first through fourth harmonics, respectively, contribute 28, 37, 23, and 12 percent each to the total ride decrement.

It appears that the hydraulic road simulator is an efficient and valid method for evaluating truck sensitivity to radial nonuniformity inputs. The development of a similar methodology for studying the effects of nonuniformity excitation in other force directions is recommended.

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1. INTRODUCTION

1.1 Background

Heavy trucks and tractor-trailers used for transporting goods must be designed for efficiency and durability. Meeting these goals constrains designers in their efforts to provide a good ride environment for the truck driver [1]. As a consequence, the U.S. truck manufacturers are constantly seeking means to improve truck ride quality consistent with the truck's primary mission. Vehicle vibrations, the primary ingredient in ride quality, are caused by the combination of road roughness and vibration sources on-board the vehicle. Of the on-board sources, nonuniformities (imbalances, runouts, etc.) in the rotating tire/wheel assemblies are an important source, causing excitation to the vehicle at their rotational frequencies and multiples thereof (harmonics). Especially on smooth roads, the tire/wheel excitations may become more noticeable and perceptible as a cause of ride degradation.

In 1979, the Motor Vehicle Manufacturers Association (MVMA), representing the common interests of the U.S. truck manufacturers and tire manufacturers, initiated a research program at the Highway Safety Research Institute (HSRI) at The University of Michigan to investigate the truck ride effects resulting from tire/wheel inputs. The research program, entitled the "Truck Tire/Wheel Systems Research Program," was organized into two concurrent phases.

-Phase I is an experimental investigation of the cyclic force variations produced by truck tire/wheel assemblies on a tire uniformity test machine to relate them to specific nonuniformities in each of the rotating components. This information is intended to provide direction for the manufacturers to make coordinated improvements in the individual components to reduce the force variations of the rolling wheel.

-Phase II looks at how force variations on the wheels of a heavy truck cause degradations in the ride in order to identify which force variations and harmonics are most critical.

This report documents the findings obtained in the Phase II project.

1.2 Problem Statement

The project purpose as originally defined was to "...conduct a Pilot Program with one test vehicle to develop and validate methodology for quantifying the ride degradation resulting from cyclic force input at each wheel location by obtaining jury evaluation of the test vehicle's ride characteristics on a smooth road under closely controlled conditions of wheel nonuniformity input."

The ride significance of force variation in a truck tire/wheel assembly is difficult to assess in practice because the variations are always present in complex combinations adding to an existing vibration environment attributable to the road. Their significance is further dependent on:

- The vehicle's responsiveness in transmitting that excitation to the cab (which varies with wheel position, force direction, amplitude and frequency).
- The sensitivity of the rider to specific amplitudes, frequencies, and directions of vibrations.
- The confounding effects of multiple excitations from the road, wheels, and other sources, on both the vehicle responsiveness and rider sensitivity.

Any systematic attempt to measure and characterize such ride degradation effects confronts two problem areas. First, the performance measure (ride), which ultimately must be related back through the vehicle to individual wheel nonuniformities, is subjective. Subjective measures are notoriously imprecise due to the large magnitudes of random error that occur. In order to extract a cause-effect relationship from data which are so imprecise, large numbers of tests are required to obtain a statistically significant sample. Second, actual nonuniformity conditions on a test vehicle are difficult to control with much precision. "Perfect" tire/wheel assemblies are not available as a base from which to start, therefore "real" assemblies must be used. In that case, the nonuniformity excitations present on a truck will vary with time in an uncontrollable fashion as the wheels phase relative to each other while on the road. Especially in the case where investigation into the higher harmonics is desirable, this latter problem severely complicates road test methods.

Therefore, the first priority in this study is to establish a methodology by which the influence of individual nonuniformities on a ride rating can be assessed. The second priority is to obtain appropriate measurements on a typical vehicle, both to validate the methodology and as a start toward building a data base for characterizing typical truck sensitivity to specific nonuniformity inputs.

1.3 Approach and Method

In view of the above-mentioned problems, it became clear that an innovative method would be necessary to obtain a precise cause-effect relationship between tire/wheel nonuniformity inputs and subjective ride rating. In addition to on-road testing, a research method based on ride degradation measurements performed on a hydraulic road simulator was proposed and used. On the assumption that the smooth road ride conditions could be validly replicated on a road simulator, the simulator was seen as a viable means by which to superimpose large numbers of nonuniformity conditions in any arbitrary pattern with precise control of amplitudes, frequencies, wheel positions, phasings, or combinations thereof. Ride conditions of interest can be created in an efficient manner on the simulator, so that the raters can be exposed to a large number of conditions within a time frame that will not produce confounding effects of learning or fatigue. Further, the raters will be free of bias associated with knowledge of the actual conditions under test.

The research approach was therefore structured around the following tasks:

1. Baseline On-Road Tests - In order to establish a baseline reference condition for the laboratory tests performed on a hydraulic road simulator, on-road tests were first conducted. The vehicle was instrumented with accelerometers to record the actual vibration experienced when driven over a smooth road section. The vehicle was rated under the same conditions for its ride characteristics by a panel of 10 experienced engineers from industry.

2. Hydraulic-Road-Simulator Tests - The profile of the smooth road test section was measured and recorded on magnetic tape for input as the background excitation of the test truck on the hydraulic road simulator. Sine waves were selectively added to the road profile to replicate tire/wheel radial nonuniformities on the simulator. One hundred test conditions representing different nonuniformity amplitudes, frequencies, wheel positions, phasings, and combinations were generated. The rating jury were seated in the vehicle while on the simulator, and asked to evaluate the ride under each of the 100 conditions. The overall changes in the ratings were then analyzed to determine the thresholds at which ride degradation began with each type of nonuniformity, and the rate of degradation with increasing nonuniformity amplitudes.

3. Inertance/Impedance Tests - In order to relate the sine wave nonuniformity inputs on the simulator to the equivalent force variations on a tire/wheel assembly, the dynamic properties of the tire, wheel, and vehicle must be known. Inertance/impedance tests were conducted in an effort to determine the dynamic sensitivities of the test vehicle, so that the ride degradation effects could be related to nonuniformity force directions other than vertical.

1.4 Report Organization

This report describes the results of the research program outlined above. Emphasis is given in the report to an assessment of the meaning and utility of the test methodology used in the research as a basis for planning comparable tests on other vehicles in the future. Chapter 2 documents the test methods used in each of the three research tasks. Chapter 3 presents a technical explanation of truck ride vibration and how tire/wheel nonuniformities contribute to that vibration using engineering models to characterize that relationship. The findings relating ride degradation to actual levels of tire/wheel excitation are presented in Chapter 4. Conclusions and recommendations from the project are presented in Chapter 5. The detailed results from the hydraulic road simulator tests are included in Appendix A.

2. TESTING METHODOLOGY

2.1 Technical Introduction

The research objective of measuring truck ride decrement due to tire/wheel nonuniformities is not entirely without precedent. Within the vehicle and tire manufacturing industries, acceptance criteria for mass imbalance and various tire force properties have been based on experimental tests using subjective ratings as the performance measure [2]. Typically, test vehicles will be set up with various known levels of a given test condition and driven on-road for the evaluation. In the case of tire balance, for example, the vehicle can be set up with one wheel at a known imbalance and all others balanced. Thence the vehicle will be driven at various speeds over different road courses to complete the evaluation.

In this study, a more precise look at the effects of tire/wheel nonuniformity is required. The scope of the Tire/Wheel Systems Research Program includes the full breadth of tire and wheel nonuniformities, including the different forcing directions, as well as higher harmonic effects. In general, the magnitude of the first harmonic of the nonuniform force produced by tire/wheel assemblies can be controlled arbitrarily by screening, match mounting of the tire and wheel, balancing or even tire grinding. However, technology is not available to readily control higher order nonuniformities. The design of a research methodology whereby a rater can be given controlled exposure to higher order harmonics in an actual on-road test setting does not exist. To meet this need, a test method employing a hydraulic road simulator was conceived. Modern simulators are capable of replicating road profile inputs to frequencies well over 30 Hz. They are limited, however, by their inability to replicate (1) tire envelopment phenomena and (2) the stiffness changes in the rolling tire. The radial (vertical) excitation equivalent to runout of the tire/wheel assembly can be readily duplicated on the simulator by the addition of sine waves to the profile, the frequency of which determines the harmonic. However, in this setting, rating ride vibrations is peculiarly different than on the road. In particular, the rater experiences a different noise environment and lacks the distraction normally associated with the driving task. Hence, it would be expected that subjective ratings in the laboratory will have a bias error when compared to the road. Inasmuch as the research objective is to measure "ride degradation," a bias error does not invalidate the simulator method. Rather, one might expect the baseline measures in the laboratory to be offset from that obtained on the road, but the thresholds at which degradation begins and the rate at which it proceeds are not as likely to differ from that which would be obtained from considered judgments on the road.

In order to answer the research objectives, road simulator tests alone are not sufficient. The major shortcoming is that only radial (vertical) nonuniformity effects are duplicated directly (although significant vibration levels in other directions result from just the vertical input). To correct this deficiency, separate tests were conducted to investigate vehicle sensitivity to longitudinal excitation at the wheels. These tests, i.e., the inertance/impedance tests, are capable of quantifying the wheel-input/acceleration-output relationship; but the subsequent relationship to ride degradation must be obtained through supplementary analysis. Additional motivation existed for conducting the inertance/impedance tests out of the interest in illustrating the vehicle sensitivity to wheel nonuniformity inputs directly. The measured inertance, reflecting the wheel-force-input to seat-acceleration-output, provides a very visual picture of that sensitivity.

Finally, on-road tests were performed to establish reference conditions for the laboratory tests. As a comparison point for the subjective ratings gathered in the laboratory, subjective ratings of the vehicle were obtained on the road using basically the same jury. In addition, the vibration conditions on the road were measured for comparison to those achieved on the road simulator as a means of verifying that the simulator conditions reasonably approximate the situation on the road.

2.2 Test Vehicle Description

Because the first priority in the project was the development of methodology, the test vehicle was selected more for convenience in testing, rather than on the basis that it represents the mean of the road tractor population. "Convenience in testing" translates into three requirements:

1) Two-axle tractor - The choice of a two-axle tractor simplifies the test method because only four input points are required. With the popular three-axle tractor, six input points exist, which would add to the test and data reduction effort.

2) Sleeper cab - A sleeper cab facilitates installation of ride measurement instrumentation in a protected environment.

3) Axles - In order to measure and adjust the nonuniformity levels at each wheel position, the Rockwell FF931 front axle and Rockwell R-170 rear axle were required. The tire uniformity test machine being used in the Phase I study accepts these axle end configurations. Therefore, by specifying these axles, the entire tire/wheel assembly mounted on the test vehicle could be installed on the uniformity test machine to set up and measure the nonuniformity levels.

A 1980 GMC cab-over-engine (COE) tractor was provided by General Motors Truck and Coach on a loan basis for the program. Similarly, a 45-foot van trailer was loaned by the Fruehauf Corporation. A photograph of the test vehicle is provided in Figure 1. The tractor is 142 inches in wheelbase, with an 86-inch aluminum sleeper cab. The front axle has a 10,860-lb gross axle weight rating with 54-inch taper leaf springs and shock absorbers. The rear axle is rated at 19,040 lbs and has a 51-inch flat leaf suspension with auxiliary springs. The tires are of bias-ply construction, size 10.00x20, load range F. The air-suspended driver's seat in the tractor is a Cush-N-Aire Exec 95 manufactured by the National Seating Company of Mansfield, Ohio.

The 45-foot Fruehauf van semitrailer has a tandem axle incorporating three 40-inch taper leafs at each wheel position with equalization between axles.

The sliding fifth wheel of the tractor was placed six inches forward of the center of the rear suspension. The combination vehicle was loaded with steel ballast for the testing. The load was concentrated over the king-pin and trailer tandem to yield the following axle load conditions:

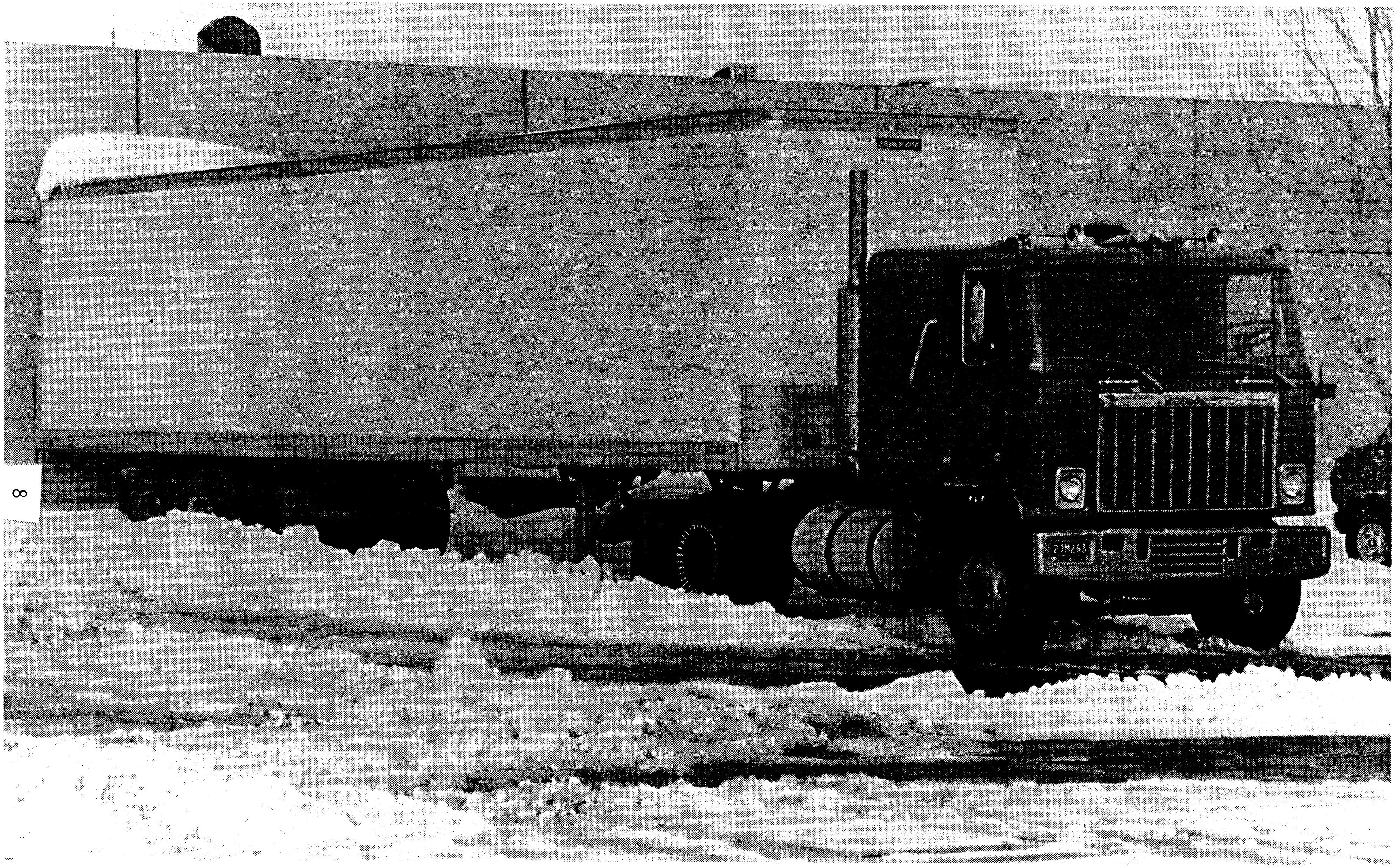


Figure 1. GMC cab-over-engine test tractor with Fruehauf van trailer.

Axle	Load
Tractor front	9,200 lb
Tractor rear	17,300 lb
Trailer tandem	26,500 lb

(This vehicle was also used in a truck ride test program conducted for the Federal Highway Administration in the project entitled "Measurement of Truck Ride Quality" [3]. In that program, the truck vibration levels were related to road roughness levels on ten different road sites. The report is available through NTIS to the interested reader.)

2.3 On-Road Test Program Description

The purpose of the on-road testing was twofold—to obtain vibration measurements characterizing the truck in the smooth-road ride condition, and to obtain a jury evaluation of the vehicle's ride under the same conditions. Two series of road tests were conducted. In the first series, a controlled nonuniformity condition was set up on the left-front wheel while the other three wheels were dressed to minimize imbalance and first harmonic radial force variations. Comprehensive vibration measurements were made at five speeds on the smooth road section, and subjective ratings by a jury of ten engineers were obtained at the speed of 55 mph. The companion tests for the FHWA were also performed in the same time frame. At MVMA's request, a second series of tests on the same road section were conducted toward the end of the project to develop maps of the ride vibration spectrum experienced on this tractor. To produce these "spectral maps," tests were performed at 11 speeds over the range of 30 to 55 mph with controlled nonuniformity conditions introduced on the left-front and left-rear wheels.

In the first series of tests, the tractor was instrumented with accelerometers on the cab, chassis, and axles, together with necessary signal conditioning and recording equipment. The accelerometer locations are illustrated in Figure 2.

Five accelerometers were placed in the cab to measure seat vertical and fore/aft vibrations, cab floor vertical at the forward and aft positions, and cab floor fore/aft. The seat accelerometers were ± 5 g piezoresistive units mounted in an SAE J1013 seat interface pad, borrowed from the FHWA.

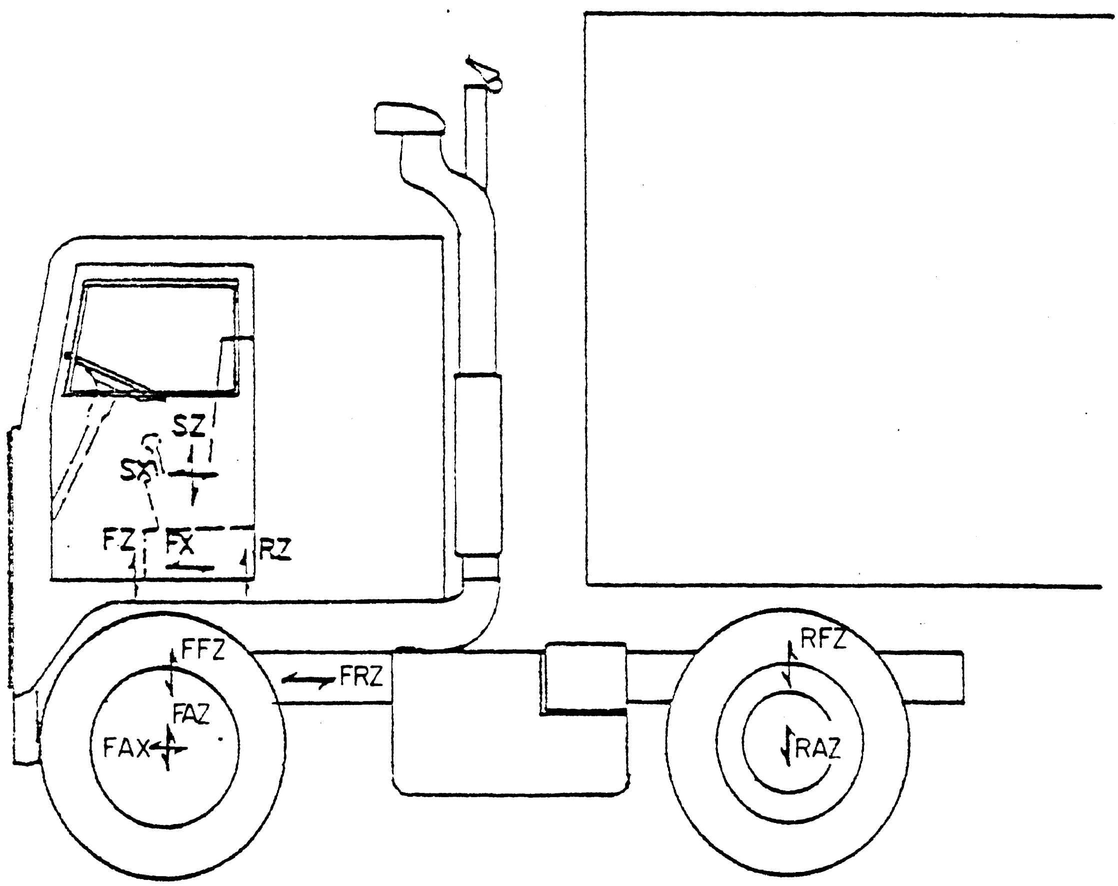


Figure 2. Accelerometer locations on the test tractor.

The cab floor accelerometers were Entran piezoresistive strain-gauge types ($\pm$10 g range, 150 Hz bandwidth) obtained from the STI Ride Quality Instrumentation Package [4].

Three accelerometers were placed on the tractor frame to measure frame vertical vibrations above the front and rear axles, and the frame fore/aft vibrations. Likewise, accelerometers were placed on the axles to measure the vertical and fore/aft vibrations of the front axle and the vertical vibrations on the rear axle. Each of these accelerometers was a Schaevitz $\pm$10 g servo type.

In the second series of tests, a reduced instrumentation package was used. The three accelerometers mounted on the cab floor were replaced with two Schaevitz 10-g servo accelerometers to measure vertical and fore/aft vibrations, respectively. The accelerometers mounted on the frame and axle were deleted.

In all tests, the accelerometer outputs were connected to an FM magnetic tape recorder through an HSRI general-purpose amplifier/controller. The amplifier/controller allows convenient rescaling of the accelerometer signals for different test conditions, and provides a control signal indicating test status, along with calibration reference voltages. The signals were recorded in analog format on a Honeywell 5600C FM magnetic tape recorder using the IRIG Intermediate Band at 1-7/8 inches per second (3.375k Hz center frequency) which provides 0-625 Hz bandwidth with better than 40 db dynamic range. Test identification was entered on the voice track of the tape recorder.

The recorded data were processed in the HSRI laboratory using either the University's computer system or a Hewlett-Packard Model 3582A spectrum analyzer to obtain the acceleration spectra shown throughout this report.

In order to obtain the most repeatable test conditions possible, each wheel of the test tractor was set up to a controlled nonuniformity input condition. For the first test series, the objective was to minimize the nonuniformity conditions at all wheels, except the left-front wheel which was to be left at a typical condition. In this way, the vehicle would have a tire/wheel nonuniformity present in its ride spectrum, but with this input coming from only one wheel, it would be less variable due to the random phasing normally obtained with a multiple-wheel input. Each of the wheels of the tractor were removed at the bearings and the entire wheel assembly was installed on the MTS Tire Uniformity Test Machine being used in the Phase I study. The individual wheels were set to the nonuniformity conditions summarized in Table 1.

Table 1. Summary of Tractor Wheel Nonuniformity Conditions for the First Series of Tests.

Nonuniformity Type	Left Front	Right Front	Left Rear	Right Rear
Balance	<7 in-oz	<2 in-oz	<33 in-oz	<7 in-oz
Radial 1st Harmonic*	184.5 lb	44.3 lb	30.4 lb	63.3 lb
Radial 2nd Harmonic	35.9	27.2	36.8	56.4
Radial 3rd Harmonic	36.7	20.4	29.7	8.5
Radial 4th Harmonic	12.9	30.1	22.3	19.6
Radial 5th Harmonic	13.7	11.1	15.3	10.0

*Radial harmonic magnitudes measured on Tire Uniformity Test Machine at 5 mph

The smooth-road test site was selected to be the section of US Route I94 running eastbound just south of Ann Arbor, Michigan between mileposts 180 and 182. The site is a bituminous asphalt surface which had been identified in previous HSRI research on road roughness [5]. By special arrangement, the elevation profiles of the road section were measured by the latest digital version of the GMR-type inertial profilometer built by K.J. Law Engineers, Inc. of Farmington, Michigan. The profilometer is a van-mounted instrumentation system that runs over the highway section at normal traffic speeds measuring the vertical elevation of the road surface in the left and right wheel tracks. The "profile" that is recorded is an accurate representation of the road over the range of wavelengths which influence vehicle ride. Figure 3 shows the profile elevation spectrum obtained for the I94 test site. The recorded profiles were processed by HSRI to prepare left and eight, front and rear, road elevation input signals for the hydraulic road simulator at International Harvester Company.

For the baseline ride evaluation of the GMC tractor, a jury of ten experienced ride engineers was selected. These ten were comprised of eight engineers from the MVMA member company truck manufacturers, plus a representative from the MVMA, and the author. The subjective rating form shown in Figure 4 was used. The form was designed with the assistance of HSRI's Human Factors Division staff experienced in the subjective rating process. In the design, the subjective rating form developed by Human Factors Research (HFR) for the MVMA Truck Ride Quality Demonstration [6] was considered, but deemed inappropriate. The major changes reflected in the HSRI choice of a rating form were:

1) A subjective rating of "acceptability" is a more familiar and appropriate variable for use by ride engineers, rather than "tolerance" as used by HFR with typical truck drivers.

2) An 11-point, rather than an 11-interval, scale is used.

3) The scale is balanced about a midpoint boundary between acceptable and unacceptable ratings with a series of verbal tags selected to provide a nominally linear scale between well-anchored end points.

In the on-road tests, the subjective rating form was used to rate seat vertical, seat fore/aft, and steering-wheel vibrations for operation on the smooth road at 55 mph. The same form was used for the subjective ratings on the IHC hydraulic road simulator with the addition of ratings for seat lateral vibration and "cab shake" modes.

2.4 Hydraulic Road Simulator Test

The International Harvester Company Truck Engineering Center in Fort Wayne, Indiana maintains a hydraulic road simulator facility accommodating trucks with up to six wheel positions. The simulator was manufactured by MTS Systems, Inc. and is comprised of a hydraulic power supply, six

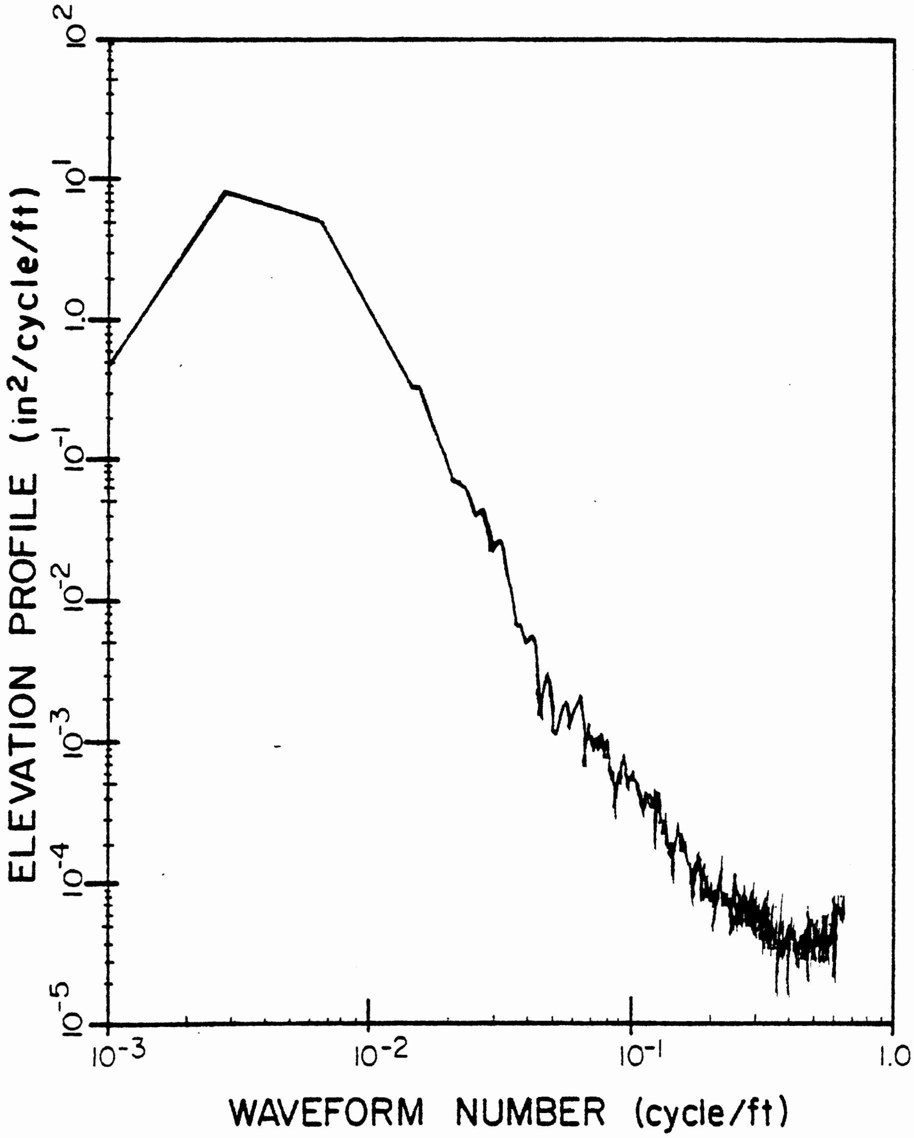


Figure 3. Elevation spectrum for the smooth-road test site.

SUBJECTIVE RATING FORM
FOR
TRUCK RIDE VIBRATIONS

Object: The object of this rating is to obtain your opinion on the acceptability of the truck ride vibrations experienced for the test period designated by the experimenter. Base the opinion on the acceptability assuming the vibrations to be a full time occupational exposure to you as a truck driver.

Procedure:

1. At the end of the test period, rate each of the following vibration modes separately:

   - -Seat vibration - vertical
   - -Seat vibration - fore/aft
   - -Steering wheel vibration

2. Designate your rating by a numerical value on the Vibration Rating Scale shown below. Fractional values may be used if you consider them appropriate.

3. Rate only the current vibration experience. Try not to be influenced by opinions developed from earlier test conditions.

ON-ROAD TRUCK RATING FORM

MVMA Project #1163

Driver ______________________

Test Engineer ______________________

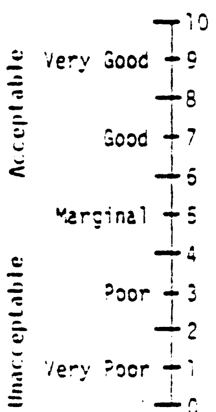
Date ____________ Time ____________

Vehicle: GMC Tractor with Fruehauf Trailer

Test Site ______________________

Speed ______________________

Rating Scale



Numerical Rating

Seat Vertical: ________

Seat Fore/Aft: ________

Steering Wheel: ________

Figure 4. Subjective ride ratings instruction sheet and forms.

servo-controlled actuators for placement under the individual wheels of the vehicle, and a servo command/control system. The actuators are capable of strokes up to six inches with frequency response well above 30 Hz. The International Harvester Company has further added provisions to load a tractor emplaced on the facility by means of a tanker trailer, the load of which is varied by the addition of water ballast.

The GMC test tractor was installed on this facility and loaded to the axle load conditions used in the on-road testing. HSRI prepared a magnetic tape with four signals representing the road elevation input to the actuators for each tractor wheel position. The left and right inputs were separate elevation signals as measured by the GMR-type road profilometer on the smooth-road site, and the rear wheel signals were the same profiles time-delayed in accordance with the 142-inch tractor wheelbase, 55 mph speed condition. The "road" input signals were played back directly from a Honeywell 5600c tape recorder into the actuator command hardware for testing. Additional tracks on the tape recorder contained sine waves of 7.3, 14.6, 21.9, and 29.2 Hz which could be added to any of the wheel input signals to represent the first through fourth harmonics of wheel nonuniformity. Precision ten-turn potentiometers controlled the amplitude of the "added" sine wave component. Phase-shifted versions of each sine wave were also provided on the tape so that controlled combinations could be achieved. By providing the sine waves directly on the same recording as used for the profile inputs, harmonic phasings and their relationship to the road input were always controlled. Sine waves for frequency intervals below 7.3 Hz (specifically, 6.6, 5.97, 5.31, 4.64, 3.98, and 3.32 Hz) were produced by a sine wave generator. These lower frequencies were used to explore first harmonic effects which occur at lower operating speeds. It should be noted, however, that the road was held at a 55-mph speed in these tests in order to avoid confounding effects due to a change in road input conditions.

After installation of the tractor on the road simulator, each of the actuators was calibrated to ensure that the road signal was being replicated at its true amplitude.

The ride jury members were scheduled to come to Fort Wayne for half-day test sessions. Each member was seated in the truck and exposed to

100 "ride" tests. Each test was 60 seconds in duration, during which the rater was exposed to the smooth-road input along with some nonuniformity condition unknown to the rater. At the completion of each test, the rater was given 30 seconds to record his rating for

- -Seat vertical vibrations
- -Seat fore/aft vibrations
- -Seat lateral vibrations
- -Steering-wheel vibrations
- -Cab shake vibrations

using the HSRI Acceptability Rating Scale. At the end of the 30-second break, a new test condition was begun. The 100 tests represented 90 unique nonuniformity conditions distinguished by frequency, amplitude, wheel position, and combinations (multiple wheel positions or multiple harmonics on one wheel position). The remaining ten tests included replications and null conditions. The test order was randomized separately for each of the ten raters to counteract fatigue or learning effects. The total test sequence was presented to the rater in a period of approximately three hours.

In addition to the jury rating data acquired by HSRI, the IHC personnel made seat vibration measurements on a sampling of test conditions with each rater, and reduced those measurements to some standard numerics commonly used by the IHC engineering staff to quantify ride vibrations. These numerics included ISO-weighted rms accelerations, histograms, and absorbed power values [7].

While the tractor was on the simulator, dynamic measurements were made utilizing the capabilities on hand for characterizing the vertical dynamics of the tractor. On the simulator, it is convenient to expose the vehicle to both random-road and sinusoidal inputs, selectively at each wheel position, while concurrently measuring ground plane motion, ground plane force, axle accelerations, and seat accelerations. The data were compiled as force or acceleration spectra and transfer functions. (Note: the transfer function data was ultimately found to be marginally useful because the high degree of nonlinear behavior in the truck resulted in poor coherence between the input and output variables.)

2.5 Inertance/Impedance Tests

In the HSRI laboratory, the tractor was set up to measure its sensitivity to wheel inputs in the longitudinal direction. Though the original project plan envisioned measurements for force inputs in both the longitudinal (tractive force) and lateral directions, only the longitudinal measurements were made. The lateral tests were dropped, in part, to compensate for the additional on-road (spectral mapping) tests, and, in part, due to the growing recognition that the lateral vibrations of the tractor were of second-order importance in the overall vibration picture.

The test setup is illustrated in Figure 5 which shows a servo-controlled hydraulic exciter acting on the centerline of the front or rear wheel. The force input from the exciter was measured by means of a strain-gauge-type load cell. A random excitation generated by a Hewlett-Packard spectrum analyzer controlled the servo displacement input. Concurrent measurements of acceleration on the cab floor and seat were obtained and recorded on magnetic tape for later processing. The force and acceleration data were then reduced to transmissibility plots by means of a Hewlett-Packard spectrum analyzer. (These plots will be shown and discussed in a subsequent portion of this report.)

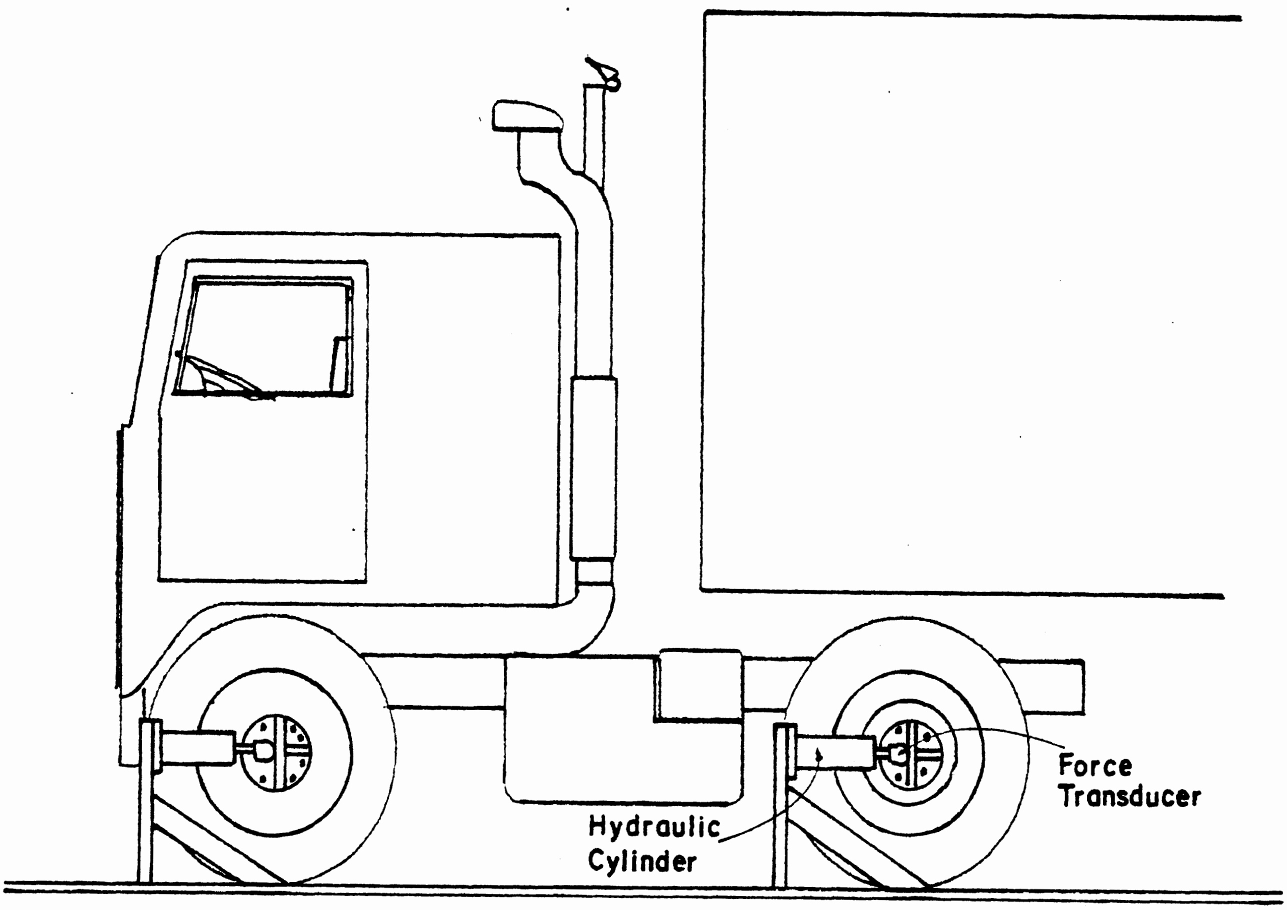


Figure 5. Setup for the inertance/impedance tests.

3. MECHANICS OF TRUCK RIDE VIBRATIONS

3.1 Introduction

The objective of this chapter is to present a technical explanation of the mechanics of truck vibration as background for understanding how tire/wheel nonuniformities contribute to ride vibrations.

Truck response to road roughness is considered, first, in order to characterize the background vibration environment to which the truck driver is exposed. The vibrations arising from tire/wheel nonuniformities can be considered as an addition to this background environment. To understand the cause/effect relationship of the tire/wheel-induced vibrations, engineering models of the dynamic system are formulated and compared to experimental measurements on the test vehicle.

3.2 Truck Ride Response to Excitation from the Roadway

As is obvious to any road user, roughness in the road is a major source of ride vibrations. Road roughness is largely random in nature, yet all roads share certain characteristic qualities. When viewed either as an elevation (displacement) or an acceleration input to the vehicle [8] at a constant travel speed, typical roads have the amplitude-frequency characteristics shown in Figure 6.

At the most basic level, the suspension system acts to attenuate the input like a two-degree-of-freedom system similar to that shown in Figure 7. The diminishing transmissibility of the vehicle at high frequency is the mechanism which isolates the rider from the ever increasing acceleration inputs of the road at high frequency. In practice, the gross nonlinearities in truck suspension systems cause the response to be sensitive to road roughness level [9]. The hysteresis arising from inter-leaf friction on truck suspensions tends to increase the effective spring rate and diminish the damping on smooth roads. Thus the vehicle exhibits more pronounced resonances on smooth roads. For example, Figure 8 illustrates the way in which the response differs on rough and smooth roads as a result of hysteresis in the suspension. Characteristically, the two major resonances

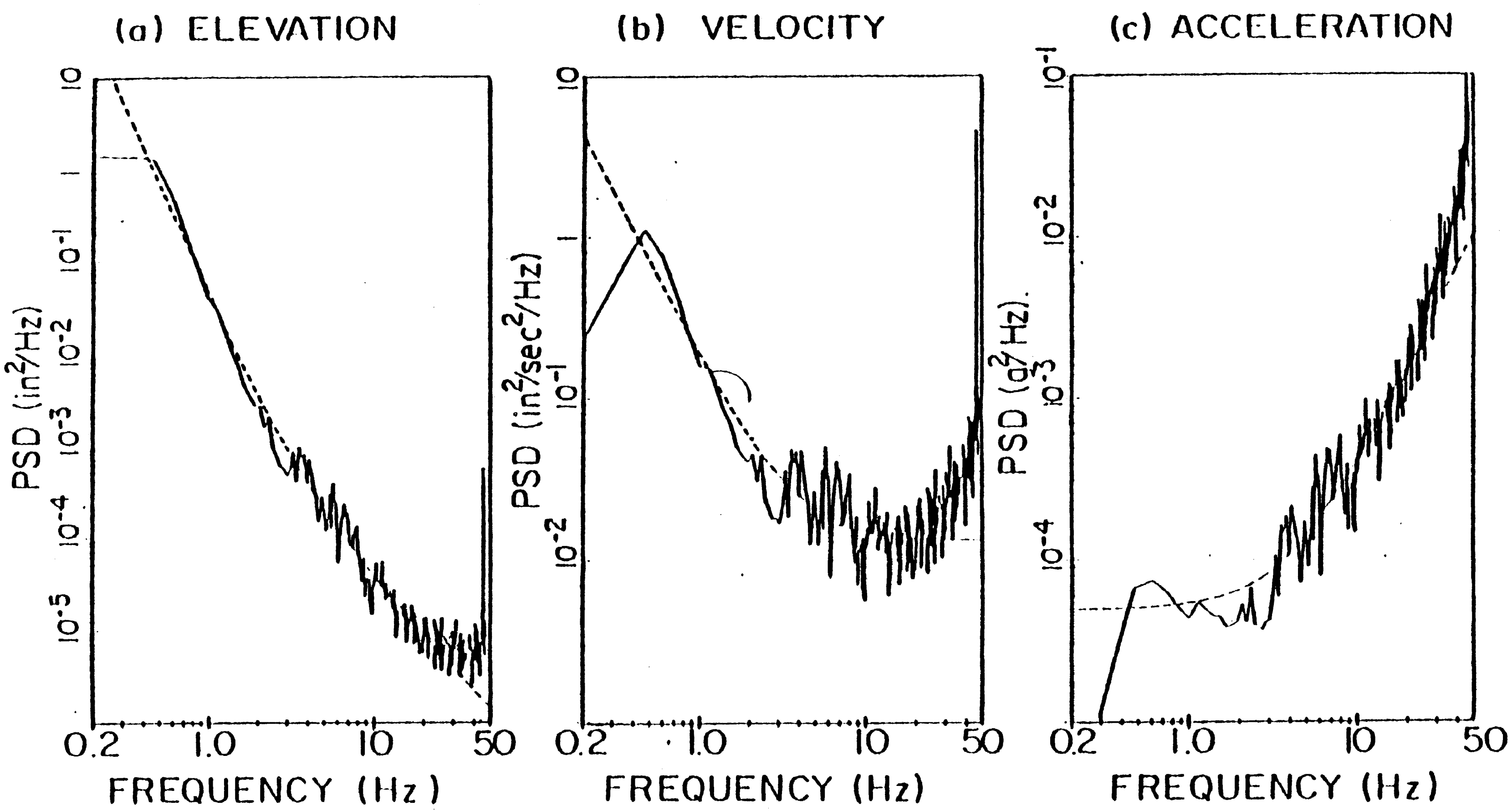


Figure 6. Elevation, velocity, and acceleration PSD's of the road roughness input to a vehicle traveling at 50 mph on the real and model roads

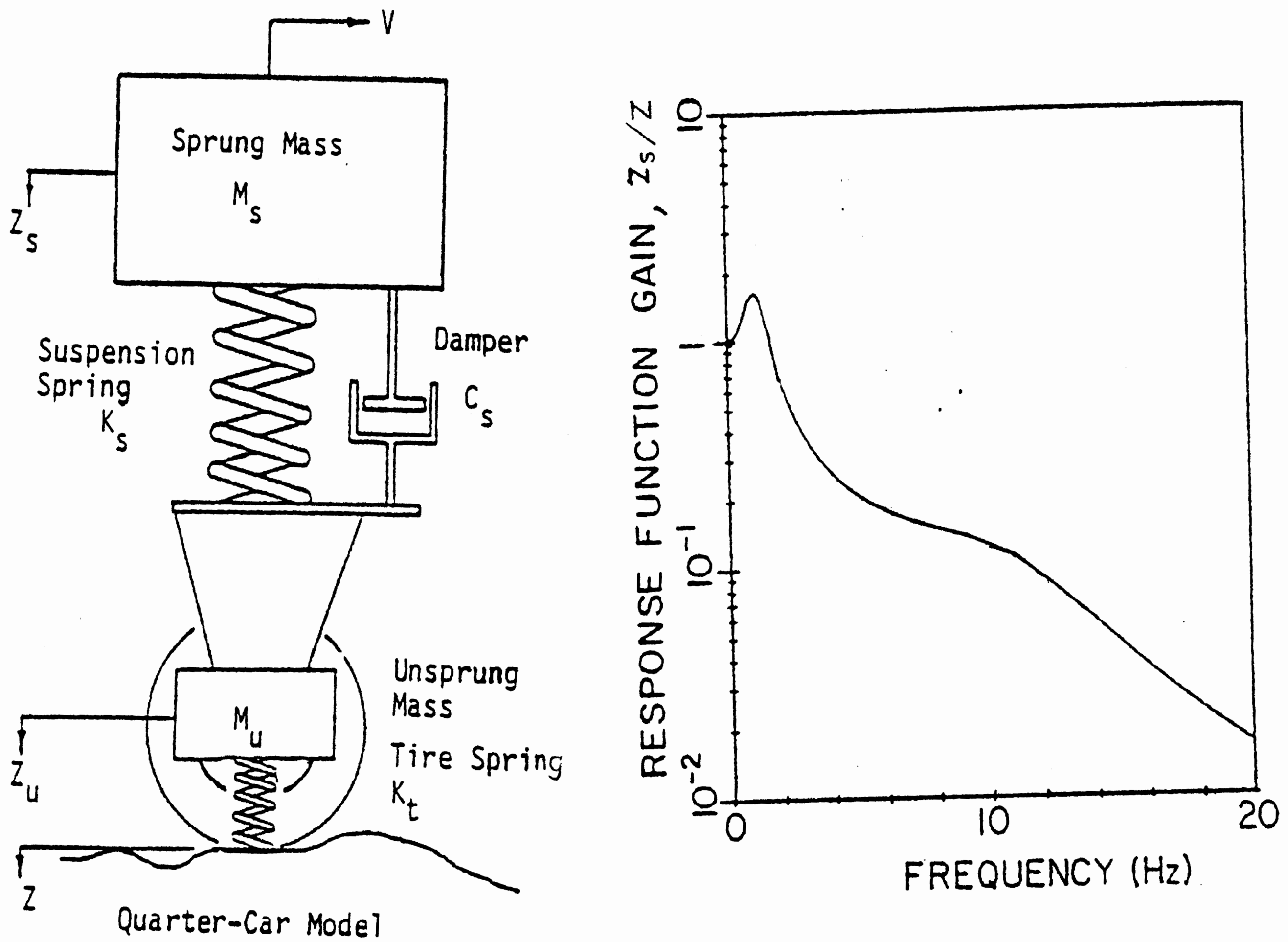
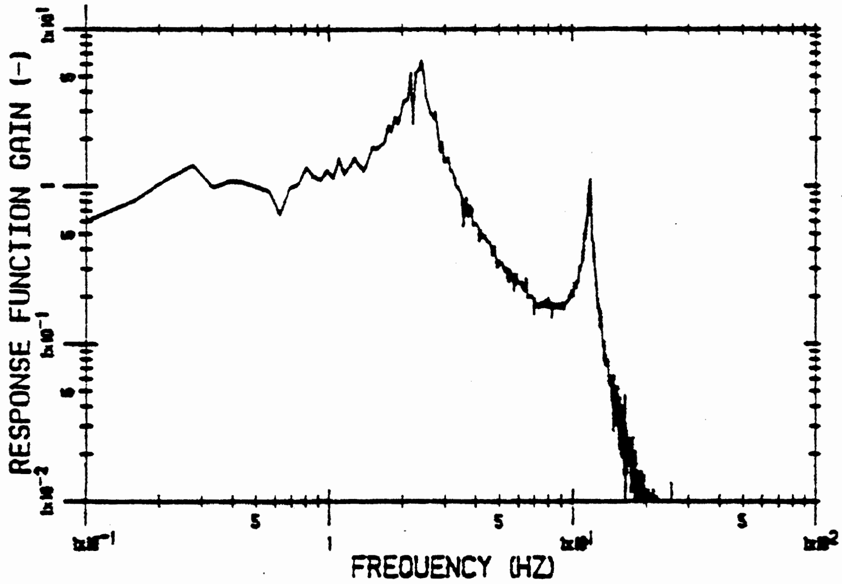


Figure 7. Illustration of the "ride isolation" behavior of a two-degree-of-freedom vehicle model.

Smooth Road



Rough Road

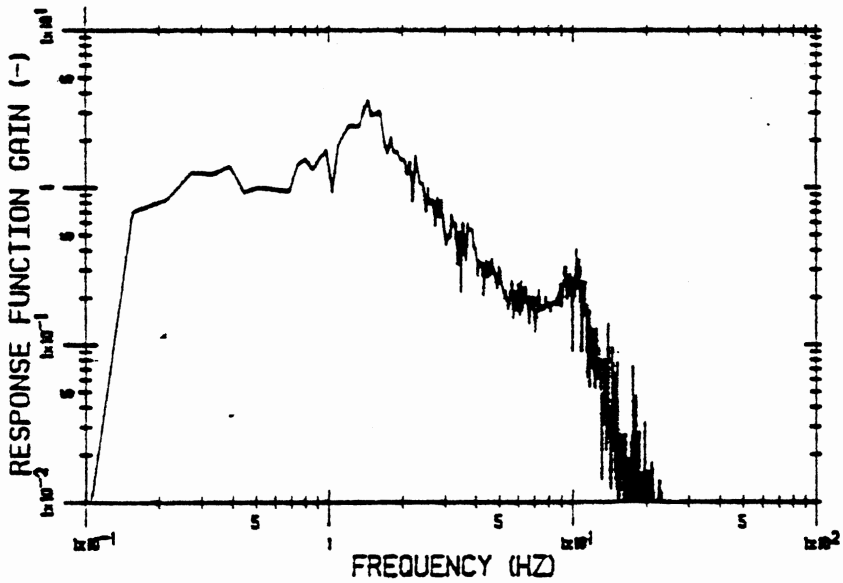


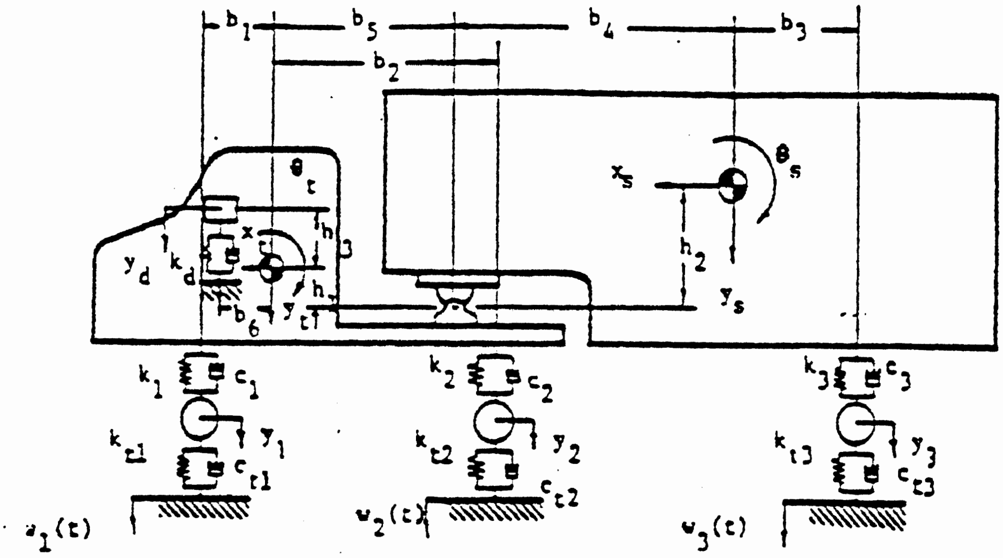
Figure 8. Response function gain of a two-mass model with a hysteretic spring on rough and smooth roads.

associated with sprung-mass-bounce and axle-hop, tend to increase in response amplitude and frequency as the road becomes smoother. This phenomena will be seen to be rather important when examining the truck's response to tire/wheel input in the later sections of this chapter.

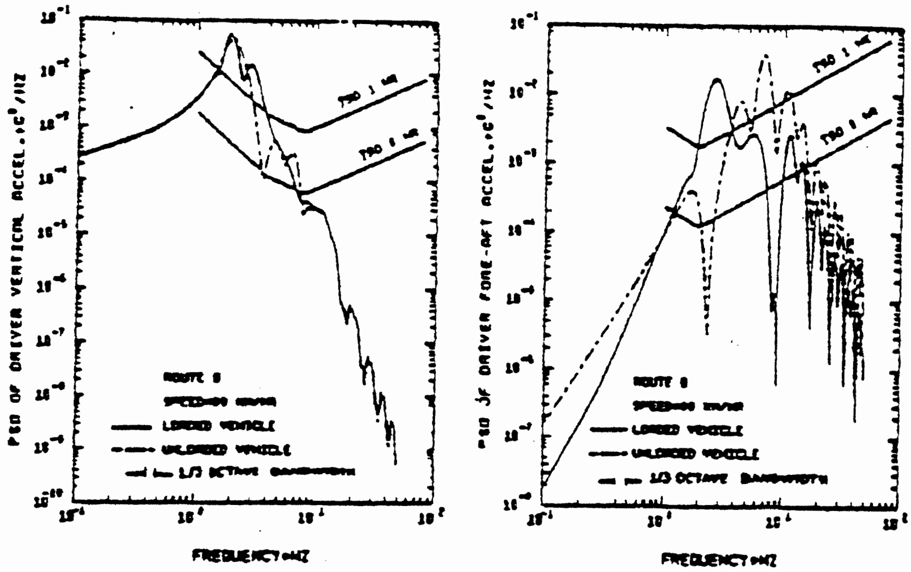
The actual ride process of a vehicle with two or more axles is more complicated than indicated in Figure 8 since the same road input acts at several positions on the vehicle, delayed only in time. This situation leads to a "wheelbase filtering" phenomena [8,10] which causes the vehicle to exhibit characteristic nulls in its bounce and pitch responses. This effect is seen in theoretical calculations of truck ride behavior such as are plotted in Figure 9. Because all trucks are subject to these same mechanisms of vibration generation, all trucks exhibit a similar overall spectrum of ride vibration. The differences between trucks become primarily differences in the overall amplitudes of vibration, and the relative importance of specific frequencies or natural modes of vibration. Similarly, only minor variations in the character of the vibration occur on different roads. The general form of the spectra are shown in Figure 10, which is data acquired on the GMC test vehicle when tested on a sample of roads which vary in their roughness quality.

In this figure, the spectra are shown with a linear vertical scale for acceleration and a linear horizontal scale for frequency. The linear-linear format is an advantageous way to look at the spectra (in contrast to the log-log format used in Figure 9) because the area under the plot equates with mean square acceleration. Although there is a lack of agreement as to how humans perceive and judge vibrations, there is no doubt that the best first estimate of the severity of a vibration is the root mean square (RMS) or the related mean square value. Hence the area under the plot provides a visual picture of the overall severity of the vibration, as well as which frequencies make the biggest contribution to the RMS value.

The curves in Figure 10 are examples of the baseline vibration spectrum that will be present on a truck under all on-road conditions. On smooth roads, the excitation level will be, perhaps, an order of magnitude lower than that of a rough road, but will always be of the same general spectral form.



Tractor-semitrailer model.



Vertical and fore and aft acceleration spectra – rough road.

Figure 9. Calculated acceleration response of a multi-degrees-of-freedom tractor-semitrailer model to random road input.

Note: The ISO curves shown in the bottom figures are for the "reduced comfort" boundary.

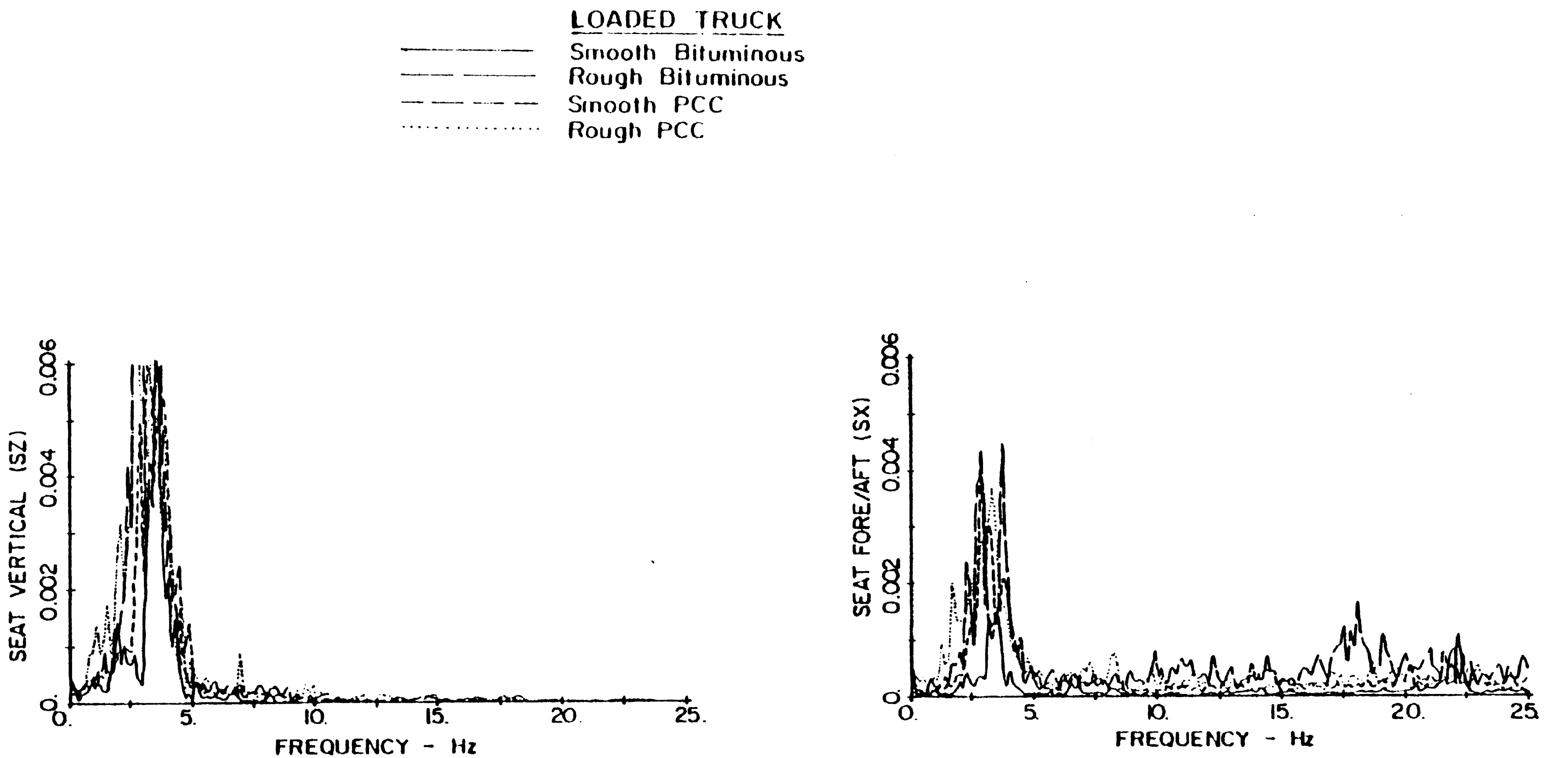


Figure 10. Seat acceleration spectra for a loaded tractor plotted on linear scale to illustrate frequency distribution of the mean square value.

3.3 Ride Excitation by Tire/Wheel Nonuniformities

Nonuniformities in the nature of mass imbalances, runouts, and stiffness variations in a tire/wheel assembly cause cyclic force variations to be produced when the wheel rotates, which variations constitute a periodic excitation to the vehicle. For example, a mass imbalance imposes a rotating force vector on a wheel which creates a force and moment excitation in both the vertical and longitudinal directions. From the standpoint of the wheel, those excitations may be categorized as radial (constituting vertical input to the truck), tractive force (constituting longitudinal excitation to the truck), and lateral (constituting lateral or roll excitation to the truck). The overturning moment and aligning moment produce moments on the truck which are negligible in their direct effect on vibration. However, both these moments, and the translational forces, have the potential for inducing vibrations in the steering system—vibrations which may constitute another objectionable form of truck vibration.

The periodic excitations that arise from tire/wheel nonuniformities can be represented as the superposition of a series of sine waves (a Fourier Series) having a fundamental frequency which is the rotational frequency of the wheel, and harmonics at integer multiples of that frequency. Each sine wave component differs in its absolute amplitude and its phase relative to the fundamental. In the case of a linear system, each harmonic can be examined separately for its individual effect on the dynamic system and the combined effects of multiple harmonics are the sum of the individual effects. However, a truck is not a linear system, nor is a driver's response (perception of ride) necessarily a linear function of the ride environment. In effect, this means that the significance of one harmonic may be dependent on the presence or absence of certain others. Whether this mechanism plays an important role, precluding the simple addition of individual harmonic effects to predict overall ride degradation, is an important area of the investigation and findings in this project.

The manner in which tire/wheel nonuniformities affect truck ride vibrations can perhaps best be seen by examining a "spectral map," as obtained for a typical truck. The spectral map is simply a collection of ride vibration spectra measured on the truck at increments of speed and plotted to show the spectral changes with speed.

Figure 11 shows spectral maps produced by the GMC tractor when driven over a smooth road. Four maps are presented, one each for the accelerations measured at the cab floor (vertical and fore/aft) and at the seat (vertical and fore/aft). The maps illustrate several important points about truck ride vibrations:

1) The spectral peak present at 3.5 Hz (due to road excitation of bounce/pitch modes) is consistently present over the speed range.

2) Spectral excitation attributable to wheel inputs is obvious at harmonics as high as the seventh harmonic. Note, however, that the excitation from the driveline (corresponding to a 3.70 rear axle ratio) falls near the fourth harmonic and predominates over that harmonic.

3) Multiple vehicle resonances (sensitivities) in the 10-25 Hz range are evident in the floor acceleration spectra and may be excited by tire/wheel harmonics at certain speeds.

4) The transmission of vertical acceleration from the floor to the seat is amplified at the 3.5 Hz resonance, but gradually attenuated above 5 Hz by the air suspension system of the seat.

5) The seat produces little isolation of fore/aft accelerations and even tends to amplify the seventh harmonic wheel effect. (Note: the seat fore/aft isolator, often called a "chugger snubber," was disengaged during these tests.)

Given that the area under the spectral curves shown in Figure 11 is a visual picture of the mean square vibration, it is clear that the ride environment is dominated by a low frequency resonance (road-excited bounce and/or pitch motions on the tires), along with multiple higher frequency resonances excited by tire/wheel nonuniformities. The vibrations excited by tire/wheel nonuniformities may be accentuated if they occur at frequencies coincident with truck resonant modes. Given that ride vibrations would already be present on the vehicle even if the tire/wheel assemblies produced no excitation, the key question becomes—What is (are) the engineering

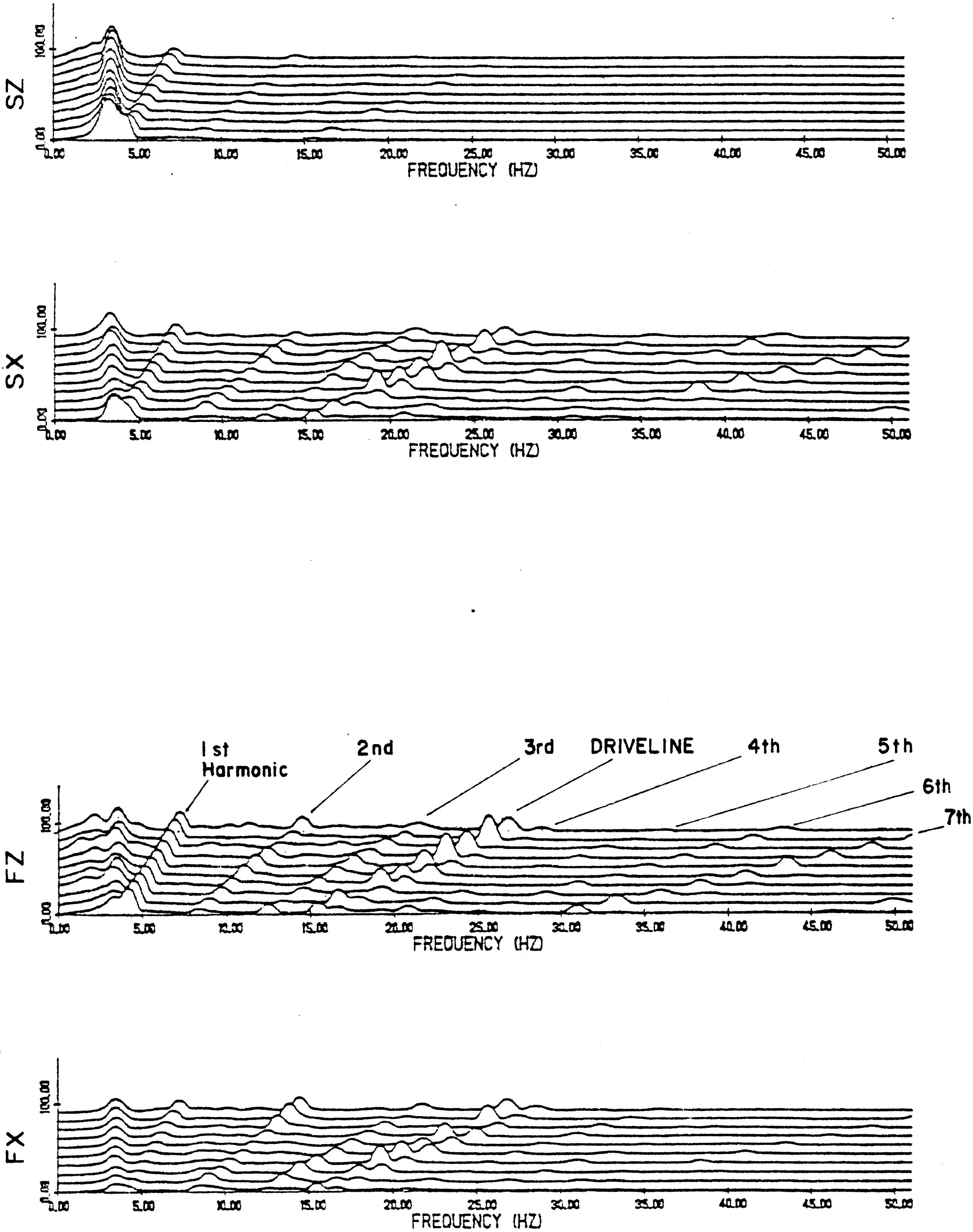


Figure 11. Spectral maps for the loaded GMC tractor on a smooth road: speeds 30-55 mph.

model(s) by which we can predict the vibration response that will appear in the spectrum for particular nonuniformity magnitudes, harmonics, and directions?

3.4 Relationship of Tire/Wheel Nonuniformities to Truck Ride Vibrations

Though the peaks in the spectra seen in Figure 11 are obviously excited by tire/wheel input, the characterization of this phenomena can be very complex. As an illustration, consider the fore/aft vibration direction. Tractive force variations at either the front or rear wheels have a potential influence on fore/aft vibrations in the cab by their excitation of longitudinal vibrations of the vehicle. Similarly, the vertical excitation caused by radial force variations at the front and/or rear wheels induce pitch mode vibrations involving a fore/aft acceleration component in the cab.

If the vehicle (system) is linear, it can be modeled analytically using transfer function methods [11]. The relevant equations are:

$$A_x = \sum_{k=1}^{n} H_{xxk} \times F_{xk} + \sum_{k=1}^{n} H_{yxk} \times F_{yk} + \sum_{k=1}^{n} H_{zxk} \times F_{zk}$$

$$A_y = \sum_{k=1}^{n} H_{xyk} \times F_{xk} + \sum_{k=1}^{n} H_{yyk} \times F_{yk} + \sum_{k=1}^{n} H_{zyk} \times F_{zk}$$

$$A_z = \sum_{k=1}^{n} H_{xzk} \times F_{xk} + \sum_{k=1}^{n} H_{yzk} \times F_{yk} + \sum_{k=1}^{n} H_{zzk} \times F_{zk}$$

where

A_x = fore/aft acceleration spectrum

A_y = lateral acceleration spectrum

A_z = vertical acceleration spectrum

H_{ijk} = transfer function characterizing the action of force in the "i" direction on acceleration in the "j" direction from wheel "k"

F_{ik} = nonuniformity excitation force in the "i" direction from wheel "k"

Based on the transfer function modeling method represented by the above equations, it is seen that to characterize the relationship of tire/wheel nonuniformities to truck ride vibrations, two types of properties must be quantified:

1) The nonuniformity must be described in terms of the excitation force it produces at the wheel.

2) The transfer functions, H_{ijk}, must be determined. Even considering only the pitch plane motions (vertical and fore/aft), 16 transfer functions are required to relate vertical and fore/aft vibrations to radial and tractive force variations at four wheel positions.

It should be further noted that some of these excitation forces may be correlated; that is, when one is present, the other will also be present and related in a fixed way. As a case in point, mass imbalance causes both radial and tractive force excitation, with the two related by a 90-degree phase shift. Its effect on the individual acceleration spectra then depends on how the two correlated inputs transfer through the vehicle and combine in the spectrum. On the one hand, they may add together, intensifying a given acceleration, or on the other hand, act in opposition (out of phase) to diminish the acceleration. Thus, in the precise modeling of correlated excitation forces, such as from mass imbalance, the phase relationship of the multiple inputs must be preserved. On the other hand, multiple inputs at the different wheels, though generally at the same frequency, are not correlated but shift in their phase relationship in a random manner. With a linear system, their effects are superimposed directly. Though all possible combinations might be considered, the worst case in which they add together is the case of primary interest.

As has been mentioned, actual truck properties tend to be rather nonlinear. In that case, frequency is not well preserved between an input and output (i.e., input at one frequency may cause output response at another). Thus the transfer function when actually measured becomes almost meaningless. As an alternative, the "transmissibility," which is simply the ratio of the output-to-input amplitude at each frequency, may be used. In the on-going discussion, the term "transmissibility" will thus often be used in lieu of transfer function.

3.4.1 Nonuniformity Excitation Force. Consider, first, the vehicle excitation force caused by a tire/wheel nonuniformity. The apparent amplitude of the force imposed on a vehicle depends on how the force is defined and the conditions under which it is measured. Oftentimes, the nonuniformity has been characterized in terms of the "force produced on the axle," when in actuality that force may be different than the "excitation force" attributable to tire/wheel nonuniformity. It is therefore convenient to turn to an engineering model of the tire/wheel system as a basis for precisely defining forces. A dynamic analysis of the tire/wheel system [12] clarifies the understanding of the various forces present when a tire/wheel assembly is installed on a dynamic system (whether a vehicle or a uniformity test machine), and how they are related. The force generated by a nonuniformity can be represented dynamically by an equivalent external force applied at the center of the wheel.* The equivalent model is then a uniform tire/wheel assembly with a cyclic force, F_w, applied at the wheel center, as shown in Figure 12. The magnitude and frequency of the cyclic force quantifies the excitation force associated with a nonuniformity.

When the tire/wheel assembly is installed on a spindle (or axle), the magnitude of the force imposed on the spindle depends on its dynamic properties. If the spindle is extremely rigid, the full magnitude of the nonuniformity excitation force is felt on the spindle. If, however, the spindle is compliant such that it will deflect under load, only part of the force, F_w, goes into the spindle, while the other part is dissipated in deflection of the tire. The relationship defining the force that would be imposed on the spindle, F_s, is given by:

*This representation appears to be accurate for frequencies below the tire resonances (nominally up to 20 Hz), based on testing conducted in the Phase I project. When higher frequencies are considered, however, its validity will have to be established by experimental tests on the uniformity test machine.

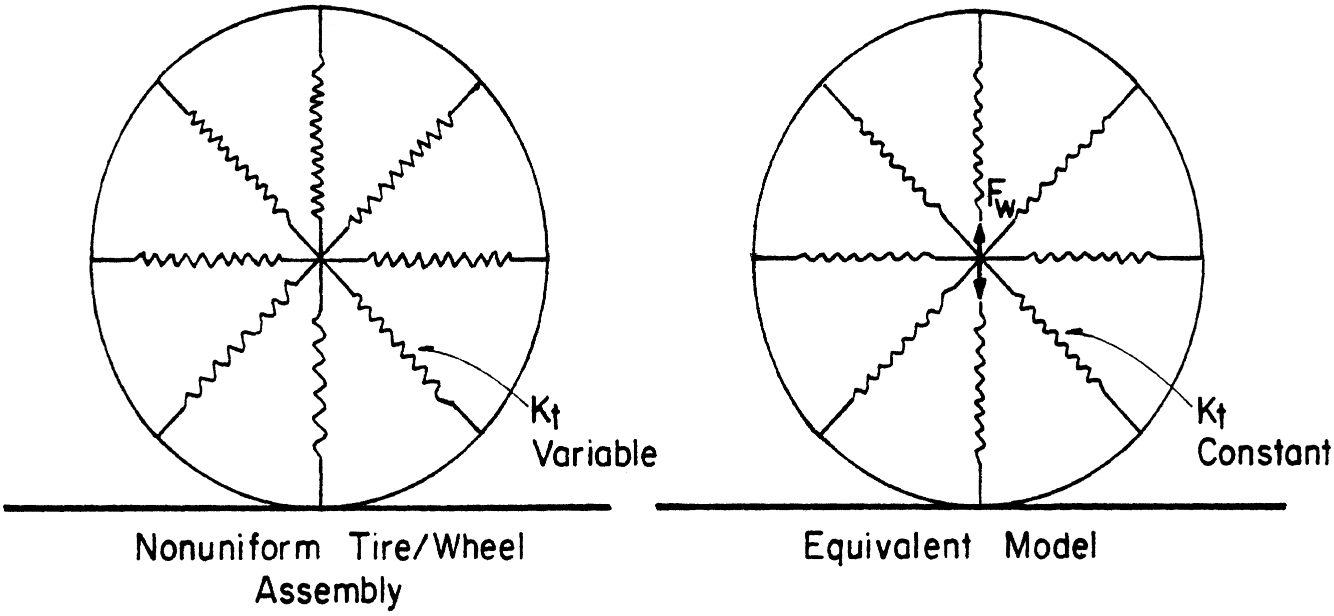


Figure 12. Dynamic model of a nonuniform tire/wheel assembly.

$$F_s = F_w \times \frac{K_s}{K_t + K_s}$$

where

F_s = the dynamic force imposed on a spindle (or axle)

F_w = the nonuniformity excitation force

K_s = the dynamic stiffness of the spindle

K_t = the dynamic stiffness of the tire/wheel assembly

This relationship has great import on the behavior on a tire uniformity test machine, as well as on a vehicle, i.e., on a tire uniformity test machine where the spindle stiffness is on the same order of magnitude as the tire stiffness, substantial errors will result. With the spindle and tire/wheel assembly having the same stiffness (i.e., $K_s = K_t$), the measured force will be only half of the true force arising from the nonuniformity. In general, accurate measurement is obtained only when the machine spindle stiffness is several orders of magnitude greater than that of the tire/wheel assembly. Lacking that condition, the measurement of force variations arising from tire or wheel nonuniformities will be dependent on the machine and tire conditions; and measurements of the simplest effects, such as the influence of tire inflation pressure (which changes the tire stiffness) will produce misleading results.

The dynamic coupling of a tire/wheel assembly to a vehicle is characterized by the same equation, differing only in the dynamic parameters. On a vehicle, the spindle force, F_s, is the force imposed on the axle and the spindle dynamic stiffness is the stiffness of the axle, i.e.,

$$F_a = F_w \times \frac{K_a}{K_t + K_a}$$

where

F_a = the dynamic force on the axle (i.e., carried by the wheel bearings)

K_a = the dynamic stiffness of the axle

Its relationship to the wheel excitation force, F_w, is illustrated in Figure 13 for vertical response of the simple two-mass analytical model used in Figure 7. At very low frequency, the dynamic stiffness of the axle is negligible because the axle carries only the static load. With any small change in force on the axle it no longer balances the static load, and, at least theoretically, an infinitive displacement will result. Thus, "K_a" is effectively zero, and the above equation indicates that a tire/wheel excitation force, F_w, will produce no significant dynamic (cyclic) force on the axle. In other words, slowly applying a vertical force to the spindle at the front wheel of loaded truck does not change the load on the axle, it simply changes the load carried by the tire.

As the excitation frequency approaches the 1-Hz sprung mass resonance frequency, the force ratio increases. However, because the suspension stiffness is much less than the tire stiffness, much of the excitation force is absorbed by the tire and does not get through to the sprung mass. Above this point, the axle force continues to increase because of the increasing axle motion as the excitation approaches the 10-Hz axle resonance frequency. At this point, the sprung mass is nearly stationary, while the axle resonates. Though the suspension deflection amplitude is not substantially different than that seen at 1 Hz, the much higher frequency yields much larger suspension forces (primarily through the action of the shock absorber). Finally, at yet higher frequencies, the force diminishes to a value between zero and one, depending on whether most of the unsprung mass is in the wheel or the axle.

3.4.2 Transmissibility Properties. The second property needed to relate forces generated by tire/wheel nonuniformities to truck vibrations is the transfer function between the force and the vehicle accelerations of interest. That property, related to the so-called inertance,* is illustrated in Figure 14 for a simple quarter-vehicle model. The figure shows

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*The term "inertance" is often used in reference to this force-to-acceleration relationship, presumably implying the mass parameter relating force-to-acceleration in the Newton's Second Law equation, $F = M \times A$. However, inertance is only formally defined as an acoustic term. In the case at hand, the output acceleration-to-input force of interest is the inverse of the inertance which has no recognized nomenclature. Hence, though the term inertance is used in this report, the reader is cautioned to be aware of this nomenclature problem.

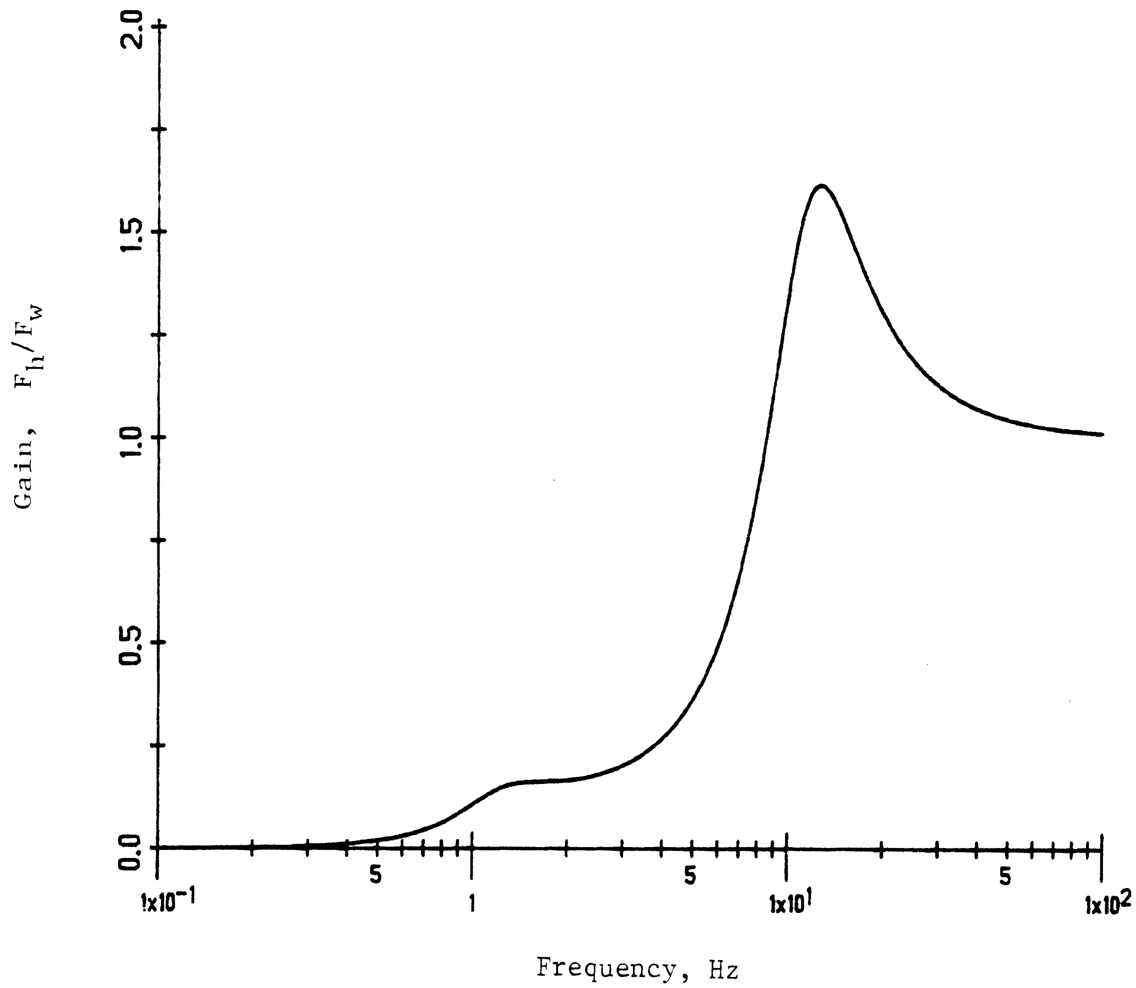


Figure 13. Plot of the ratio of axle force to nonuniformity excitation force for a quarter-vehicle model.

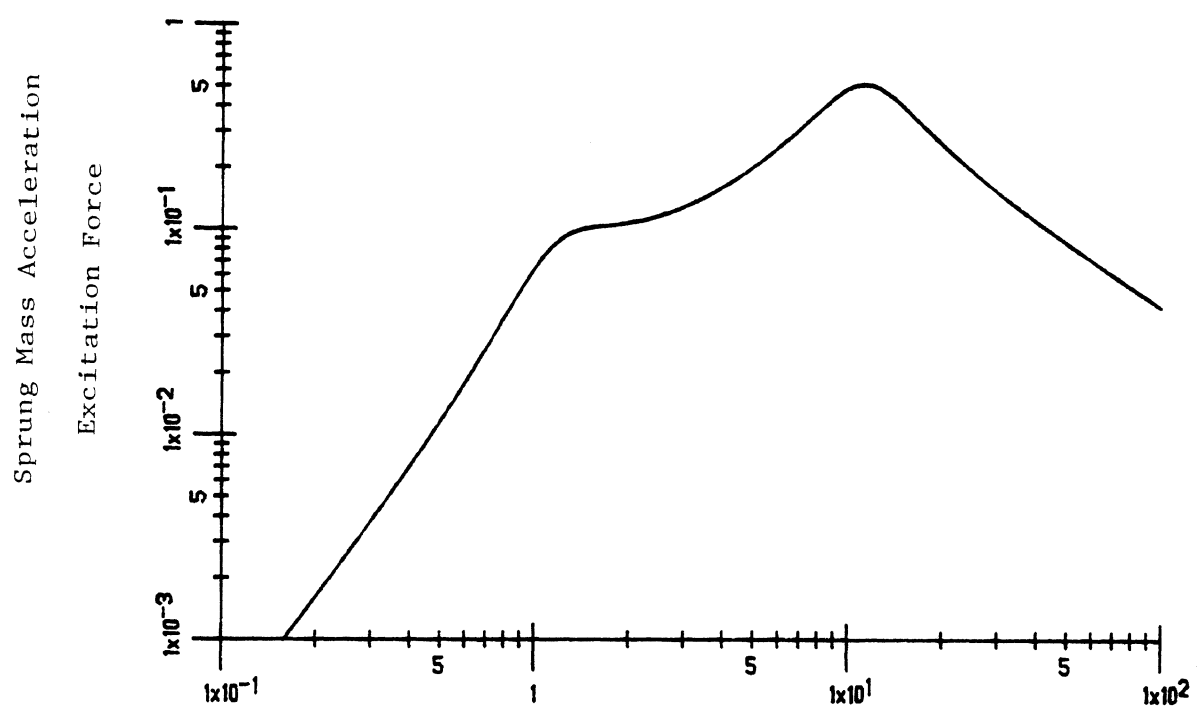


Figure 14. Acceleration of the sprung mass casued by nonuniformity excitation for a quarter-vehicle model.

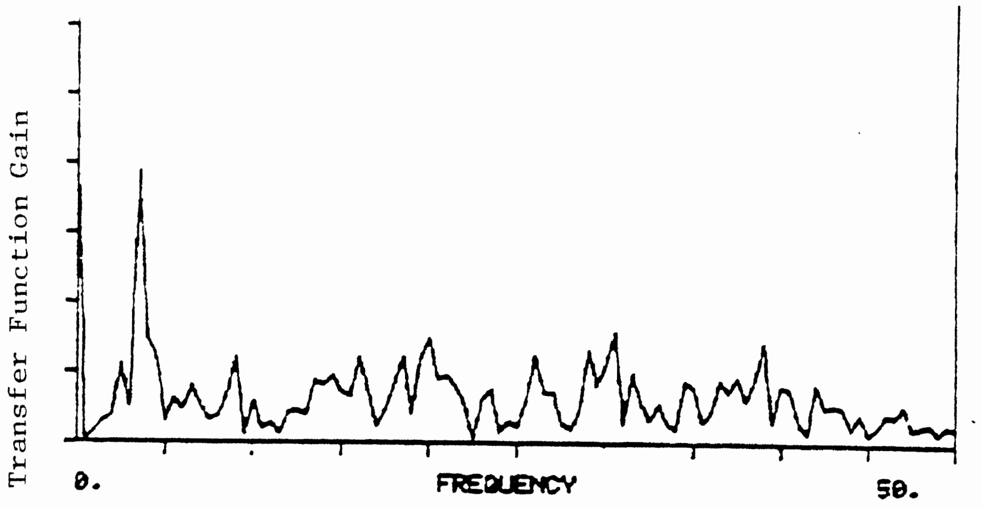
a plot of sprung mass acceleration per unit of vertical force input at the vehicle wheel. Though a theoretical result, in the sense that it is derived from an analytical model, it yields an important observation; namely, that the vehicle has the potential for being most sensitive to nonuniformity excitation in the frequency range of the axle resonance!

3.5 Experimental Measurements of Truck Transmissibility Properties

As discussed earlier, a minimum of 16 transmissibility functions must be determined just to characterize a two-axle vehicle for its pitch plane response to tire/wheel nonuniformity. When expanded to cover the lateral direction as well, 36 functions are required. Within the scope of this project, that full characterization was not attempted. Only certain of the more important vertical and longitudinal functions were measured as needed to characterize the vehicle and test the methodology.

When attempting to measure the transmissibility properties of a truck experimentally, the shortcomings of linear transfer function methods quickly become obvious. At any given frequency, the transfer function describes the amplitude and phase relationship of an input and output. If the system contains nonlinearities, such as in truck leaf spring suspensions, the output at one frequency may, in part, derive from input at another (i.e., the input and output are not coherent). The transfer function only represents that portion of the input/output relationship that is coherent. The problem is illustrated in Figure 15, which shows the transfer function of seat vertical acceleration output for left front wheel axle vertical displacement input. When the coherence is above 0.9, a good transfer function is obtained. However, the transfer function magnitude ratio is decreased proportionately when the coherence is low. Thus, with the poor coherence shown in this figure, the transfer function magnitude is not at all representative of the output-to-input ratio.

Comparable methodology for characterizing nonlinear systems is not well developed. At best, the ratio of the output-to-input spectra can be determined to describe the transmission through the system. In that case, the phase information is lost as it is not possible to associate output at a given frequency to input at that, or any other, frequency. In addition, measurements of the transmissibility are sensitive to the input wave form,



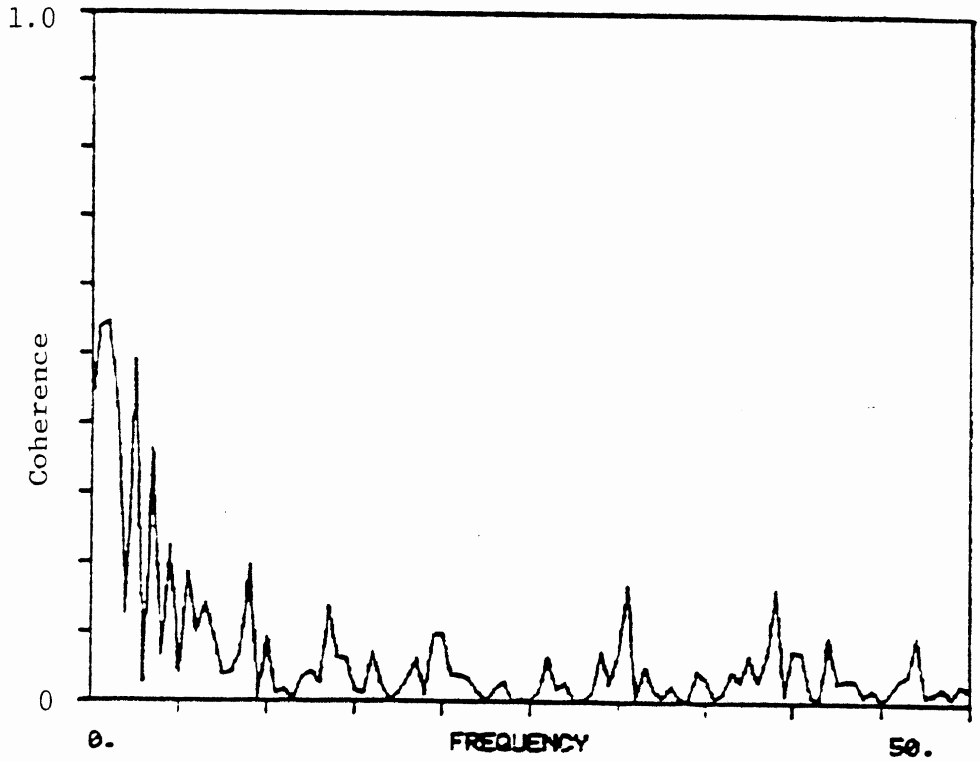


Figure 15. Transfer function and coherence for seat vertical acceleration output to left front axle displacement input.

both in its amplitude and spectral content. Therefore, it can be argued that representative measurements are obtained only when representative input wave forms are used.

3.5.1 Transmissibility of Vertical Inputs. Measurements of vertical transmissibility can be obtained in a number of ways. The most straightforward is by application of a sine wave excitation to each of the wheels separately. This clearly separates each wheel and provides the cleanest measurement, although resonant frequencies and peak amplitudes may be somewhat in error due to the absence of a fully representative input needed to exercise nonlinear components in their typical operating range.

This method was employed on the IHC road simulator using a sine wave ground displacement under the individual wheels in lieu of a force at the spindle. (Note: It can be shown that a ground displacement input produces the same sprung and unsprung mass motions as a force input at the spindle. The equivalence between the force and displacement is the dynamic stiffness of the tire, which for low frequencies is effectively the tire spring rate. The only dynamic difference arising from the two types of excitation is the tire contact force at the ground.) The output of interest is the vertical and fore/aft accelerations produced on the driver's seat. The measurement process involves a slow sweep of the sine wave frequency while the output spectra are averaged. The seat vertical and fore/aft response to pure vertical input at each wheel position is shown in Figure 16a-d.

All data are plotted on the same vertical scale of milli-g's (1/1000th of a g acceleration) per pound of excitation force to allow direct comparisons. A tire force amplitude of 125 lbs was assumed in scaling the plots, based on the sine wave amplitude of 0.025 inches acting against a tire stiffness of 5000 lb/in.

Despite the vertical input, the seat vertical response is less than the worst case fore/aft response. In part, this may be attributed to the vertical isolation provided by the seat suspension. Also note that the seat vertical accelerations in the axle resonant frequency range (15-20 Hz) are not as dominant as would be expected from Figure 14, this result again being attributable to the seat vertical isolation. The plots further indicate a

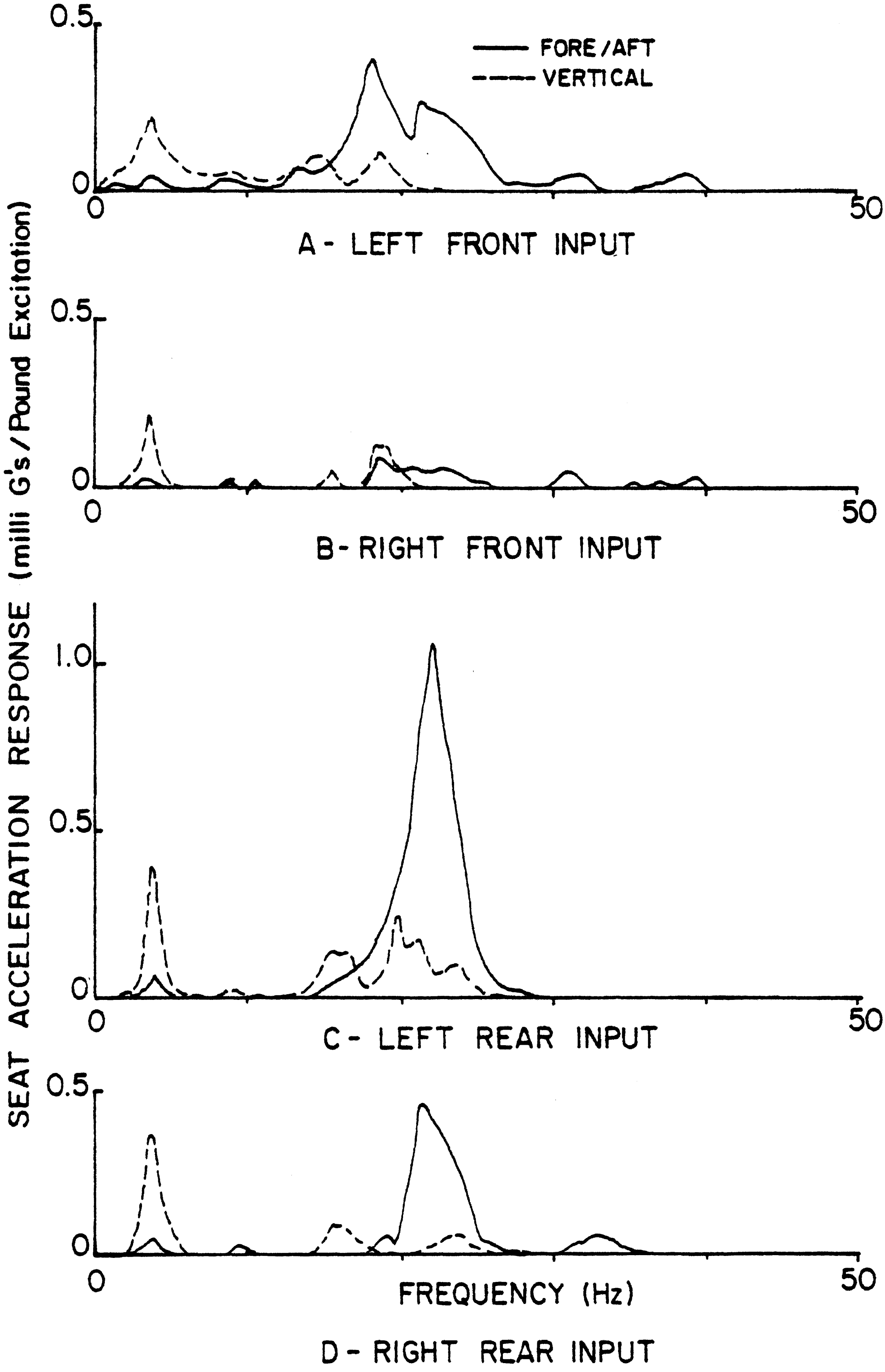


Figure 16. Seat acceleration response to radial excitation at each wheel of the GMC tractor.

strong resonant mode in the 20-25 Hz range affecting seat fore/aft vibrations. The vibration modes responsible are not known.

A comparison of these responses to the vehicle resonances evident in the spectral maps of Figure 11 generally shows good correspondence of the resonance frequencies.

3.5.2 Transmissibility of Fore/Aft Inputs. Response to fore/aft (tractive force) inputs could not be measured on the hydraulic road simulator, but rather was performed in the HSRI laboratory. The input was a random displacement in the fore/aft direction applied slightly outboard of the wheel center. In this case, force was measured directly as the input with the seat vertical and fore/aft accelerations as the outputs. Figure 17 shows the transmissibility functions for both the left front and left rear input positions plotted on the same scale as used in Figure 16.

Again, the fore/aft sensitivities at the seat are dominant, and quite high in the upper frequency range. Inasmuch as there is no discrete suspension isolation provided in the fore/aft direction, there is no pronounced low frequency resonant mode, but rather the response appears characteristic of a multi-resonant system.

3.6 Chapter Summary

The mechanisms of truck ride vibration have been discussed in accordance with the current state of knowledge. The road vibrations transmitted to the driver through the seat are predominantly in the 0-5 Hz range, and reflect rigid-body bounce and pitch modes that are not well isolated by the seat.

The tire/wheel assemblies are seen to influence the vibration spectrum by acting as vibration sources at the rotating frequency and multiples thereof. The vibration amplitude associated with each wheel harmonic depends on the direction of the force excitation, the wheel position, and the frequency. Harmonic frequencies near the truck resonant modes produce the greatest vibration amplitudes. The truck's sensivitiy in this respect can be quantified from the data acquired, and can be summarized as follows:

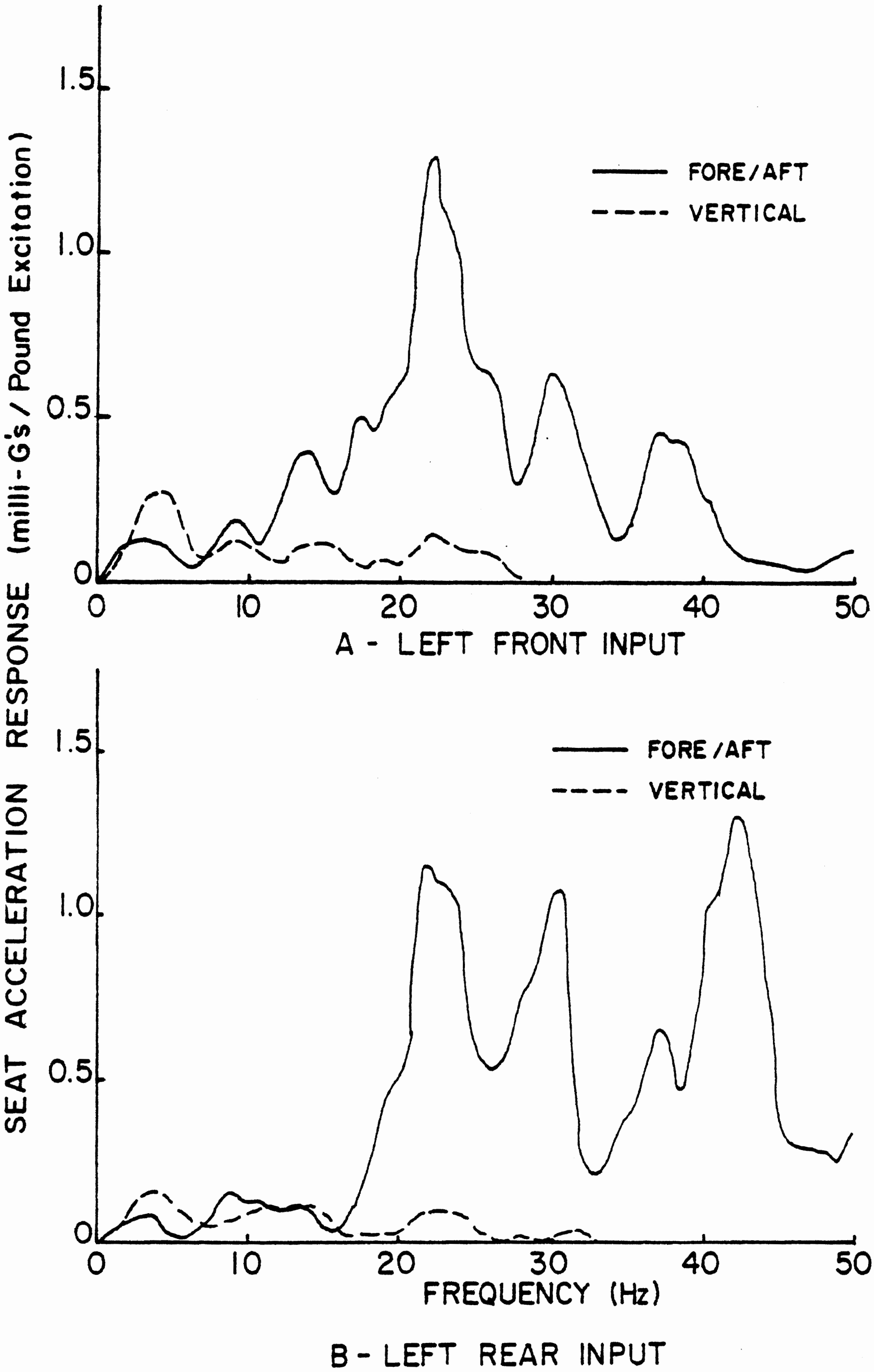


Figure 17. Seat acceleration response to fore/aft excitation at left front and left rear wheels of the GMC tractor.

1) The broadest sensitivity occurs with seat fore/aft vibrations excited by tractive force variations at the left rear wheel. Multiple truck resonance modes exist in the range of 20 to 50 Hz with a sensitivity factor in the range of 1.2 milli-g's per pound of nonuniformity excitation.

2) Next most important is seat fore/aft vibration excited by tractive force variations at the left front wheel, which exhibits a sensitivity of approximately 1.4 milli-g's per pound near a 22-Hz resonance mode.

3) Seat fore/aft sensitivity to vertical (radial) force variations at the left rear wheel exhibits a relatively high sensitivity of 1.1 milli-g's per pound, but is confined to the fairly narrow band of 20-25 Hz.

4) In general, excitations on the left side of the vehicle produce more driver's seat vibrations than excitations on the right side.

5) Seat vertical vibrations are well attenuated by the seat isolation system at higher frequencies. Thus, seat vertical sensitivity is greatest at the low frequency (3 Hz) rigid-body modes. The maximum sensitivity is in the range of 0.4 milli-g's per pound and occurs with vertical (radial) excitation at the rear wheels.

4. RIDE DEGRADATIONS FROM TIRE/WHEEL NONUNIFORMITIES

4.1 Introduction

The additional vibrations produced on a truck as a result of wheel excitation, under some circumstances, are perceptible to the driver. Driving on smooth roads is one of the most critical cases; whereas, on rough roads, the wheel inputs are masked by the greater overall vibration level. When drivers can recognize wheel-induced vibrations, it may become a source of dissatisfaction with the truck. Thus the ultimate criterion for judging the significance of a nonuniformity is the dissatisfaction that would be experienced by drivers.

"Dissatisfaction" is a subjective measure, and will hence vary in degree with individual drivers. The automotive manufacturers routinely try to measure customer satisfaction with many aspects of vehicle design or performance. The most formalized method is to conduct "jury evaluations" in which some aspect of vehicle design or performance is rated on a numerical scale. The numerical ratings obtained from a representative jury can then be treated mathematically to characterize individual and population attitudes toward the factor of interest.

In order to measure the influences of tire/wheel nonuniformities on the ride of the GMC tractor under study in this project, "jury ratings" were obtained under a large number of conditions of tire/wheel nonuniformity excitation as duplicated on a hydraulic road simulator. A smooth-road vibration spectrum was produced on the simulator as a baseline for all tests, to which were added a matrix of wheel excitation effects. Jury ratings of the truck's ride behavior were obtained on a scale of 0 to 10 over a range of wheel excitation conditions in order to map the effects of wheel excitation on ride rating. The tests explored the effects of excitation amplitude, frequency, wheel position, and combinations of the above. The results of those tests are presented in this chapter.

4.2 Comparison of Simulator to On-Road

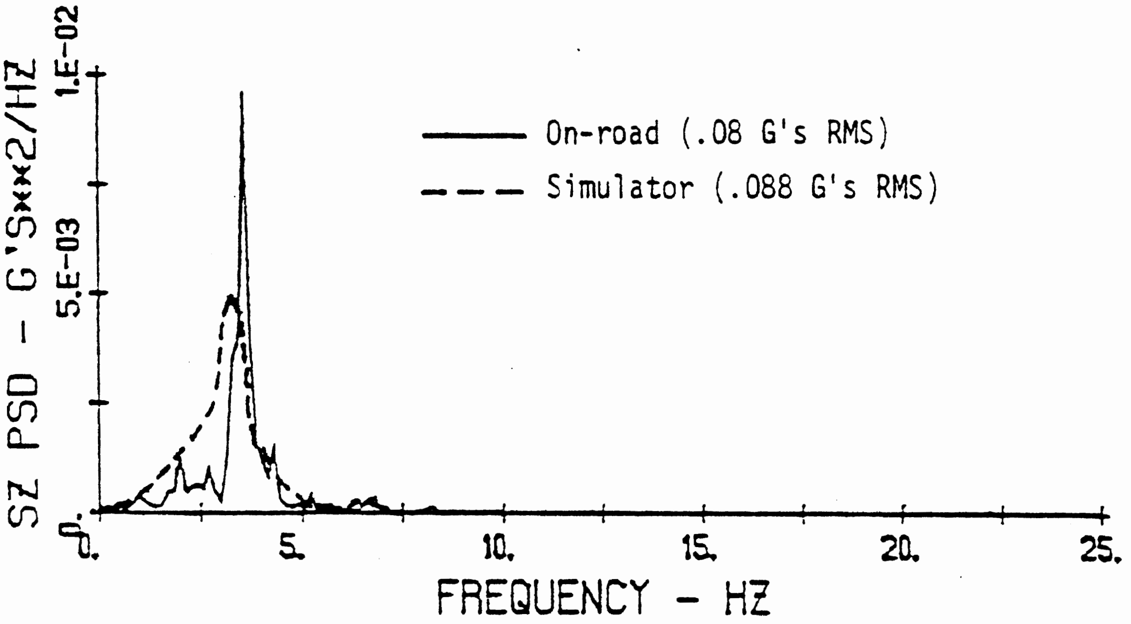
The laboratory environment of a road simulator differs in many ways from the on-road situation that may potentially influence ride ratings. Of first concern is the ability to reasonably replicate the road vibration spectrum on the hydraulic simulator. Vibration measurements were made on the seat of the truck, both on the road and on the simulator, in order to evaluate the "realism" of the simulator vibration conditions. A comparison of the vertical and fore/aft spectra under both conditions is shown in Figure 18. The vertical spectrum is well replicated on the simulator. The RMS g level in both cases is within 10 percent, and its location within the spectrum is essentially identical in both cases. The differences between the two vertical spectra seen in the figure are, in part, attributable to different spectral processing methods used in each case.

The fore/aft vibrations are not replicated as faithfully. Though the 3-Hz rigid-body mode vibrations are essentially equivalent, more high frequency content is evident on the simulator. The source of this difference is not known, but could involve the difference in loading trailers, or differences in the dynamics of the seat acceleration measuring instrumentation. Overall, the fore/aft acceleration spectrum on the simulator is approximately 50 percent greater than on the road.

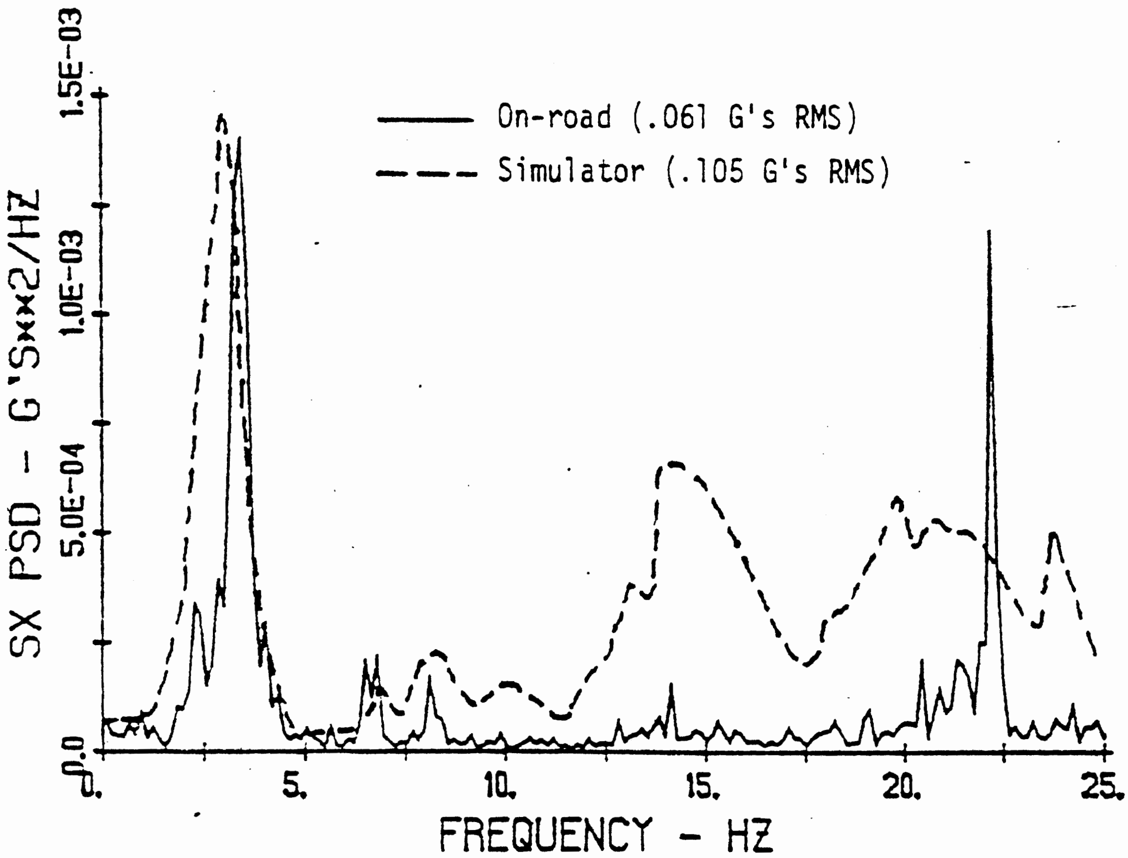
Subjective ratings were also obtained on the road and on the simulator for these ride conditions. Eight of the 10 raters evaluated both conditions. The common ratings covered seat vertical, seat fore/aft, and steering-wheel vibrations. The average ratings of these factors on the road and simulator are as follows:

Vibration Factor	On-Road	On-Simulator
Seat Vertical	7.75	6.22
Seat Fore/Aft	7.0	6.0
Steering Wheel	8.75	6.75

In the laboratory environment, ride ratings thus decreased 1 to 2 points. The increased fore/aft vibrations on the simulator may account for some of the difference, although the absence of the noise and distractions of driving



Comparison of Seat Vertical Accelerations for the GMC Tractor
On the Smooth Road at 55 MPH, and
On the Hydraulic Road Simulator



Comparison of Seat Fore/Aft Accelerations for the GMC Tractor
On the Smooth Road at 55 MPH, and
On the Hydraulic Road Simulator

Figure 18

may also contribute to more considered and critical judgments in the laboratory.

The above differences, per se, do not seriously compromise the laboratory tests. The results that are to be shown provide a good picture of the influences of wheel excitation on ride rating. However, in drawing conclusions from those results, it should be remembered that the laboratory ratings are likely to be at least as critical, or slightly more critical, than on-road ratings.

4.3 Ratings of Single Nonuniformities

A number of test conditions explored the influences of singular nonuniformities. These included first, second, third, and fourth harmonic effects at each of the four wheel positions, at the equivalent of a 55-mph road speed condition. Because of the clearly evident rigid-body modes at approximtely 3.3 Hz, lower frequency conditions were also covered, representing first harmonic excitation at lower operating speeds. In order to avoid the confounding effects of changing road input at lower speeds, the road was held at 55 mph equivalent input during these tests.

The nonuniformity effects were replicated by the addition of sine waves to the road input spectrum. The sine wave amplitudes are directly related to nonuniformity effects expressed in terms of loaded radial runout. The sine wave amplitudes were produced at the discrete levels of 0.025, 0.050, 0.075, 0.100, and 0.125 inches. The higher amplitudes were used to ensure coverage of conditions certain to reflect the worst case combinations of tire and wheel effects, and to ensure that rating trends could be discerned among the expected statistical scatter in the data.

Although three rating factors were used on the road, once the vehicle was emplaced on the simulator, it became immediately clear that additional rating factors were needed to provide the rater with an appropriate "slot" in which to record objectionable vibrations. The most pronounced of these was need for a "cab shake" rating factor. Under certain conditions, objectionable vibrations were being observed on the overall vehicle, but were not perceived strongly on the seat. Thus the need for a cab shake rating factor. In total, five rating factors were used. These were as follows:

Seat vertical

Seat fore/aft

Seat lateral

Steering wheel

Cab shake

The rating results are best illustrated by means of plots showing the mean rating of the jury as a function of the nonuniformity amplitude. A more detailed discussion rationalizing the choice of this summary ride measure is provided in Appendix A. Figure 19a-d shows the ratings as a function of nonuniformity amplitude for the first through fourth harmonic, in each of the four wheel positions. The separate plots cover the five individual rating factors, along with the average of the five factors. The vertical axis in each plot is the average value from the 10 raters, and the horizontal scale is the sine wave excitation amplitude in inches.

The data show rating effects that vary from "no effect" in some cases to rather strong degradation rates in others. The trends, or slopes of the lines, undoubtedly include a certain amount of random variation, which, even when averaged over 10 raters as done here, is still significant enough to distort the individual data points. A second influence that may play a role in the variability is the choice of factor in which the rater penalizes the rating. For example, when a first harmonic excitation is applied at a low level, the rater may at first penalize seat vertical rating, but as it grows larger, he may perceive it more as a cab shake vibration, dropping that rating while leaving the seat vertical rating unchanged, or perhaps even allowing it to increase again. To what extent this may be a rater effect or a consequence of the truck dynamics is not certain. An examination of the truck vibration spectra produced under conditions of increasing nonuniformity amplitude does show some confounding effects. Figure 20 shows the seat vertical spectrum for several ampltiudes of first harmonic excitation at the left front wheel. With the lowest level of excitation (0.025 inches), the major alteration in the spectrum is the emergence of a spectral peak at the first harmonic frequency of 7.3 Hz. However, as the excitation amplitude increases, the 3-Hz spectral peak associated with the road input diminishes.

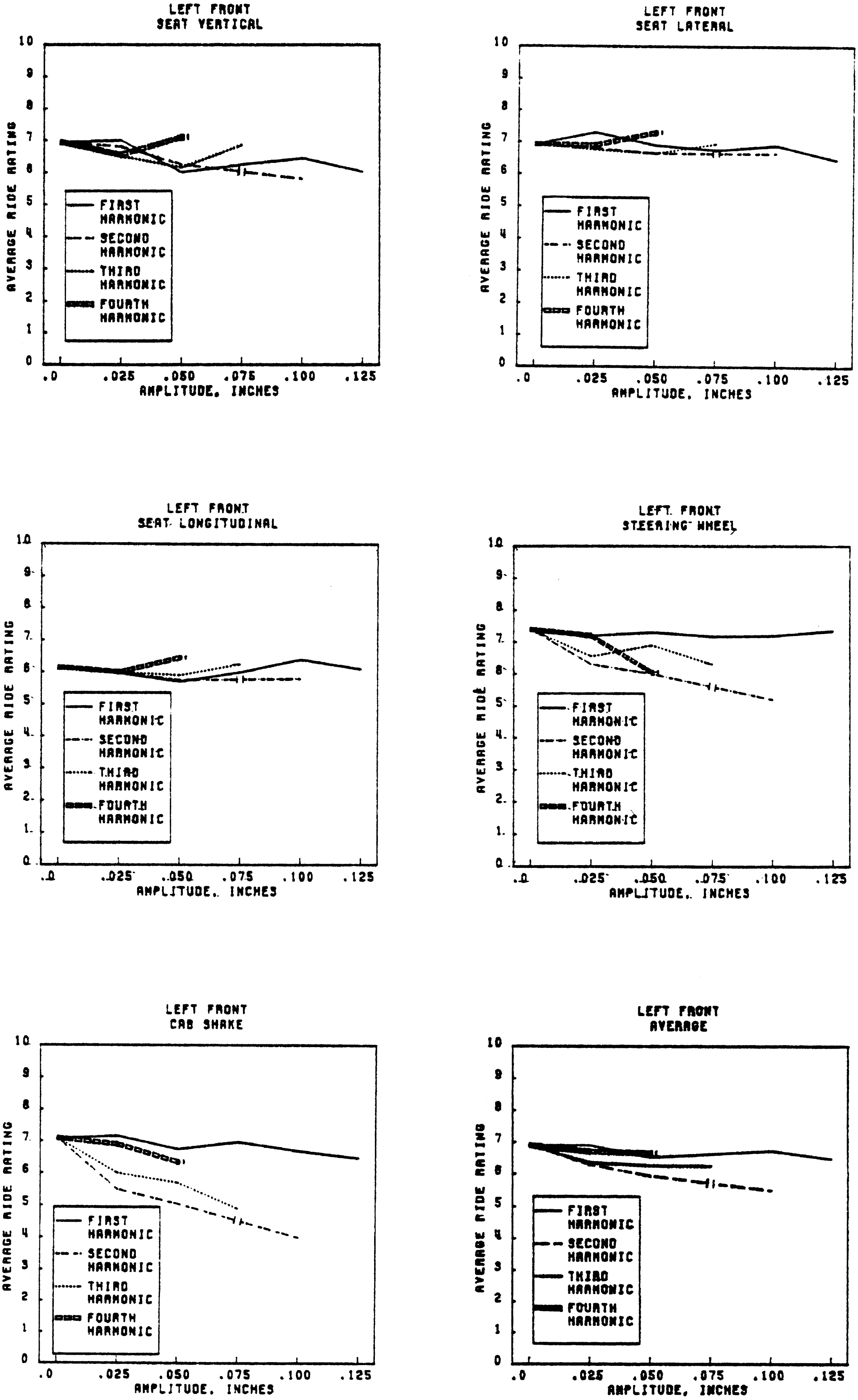


Figure 19a. Rating effects of left front nonuniformity excitation.

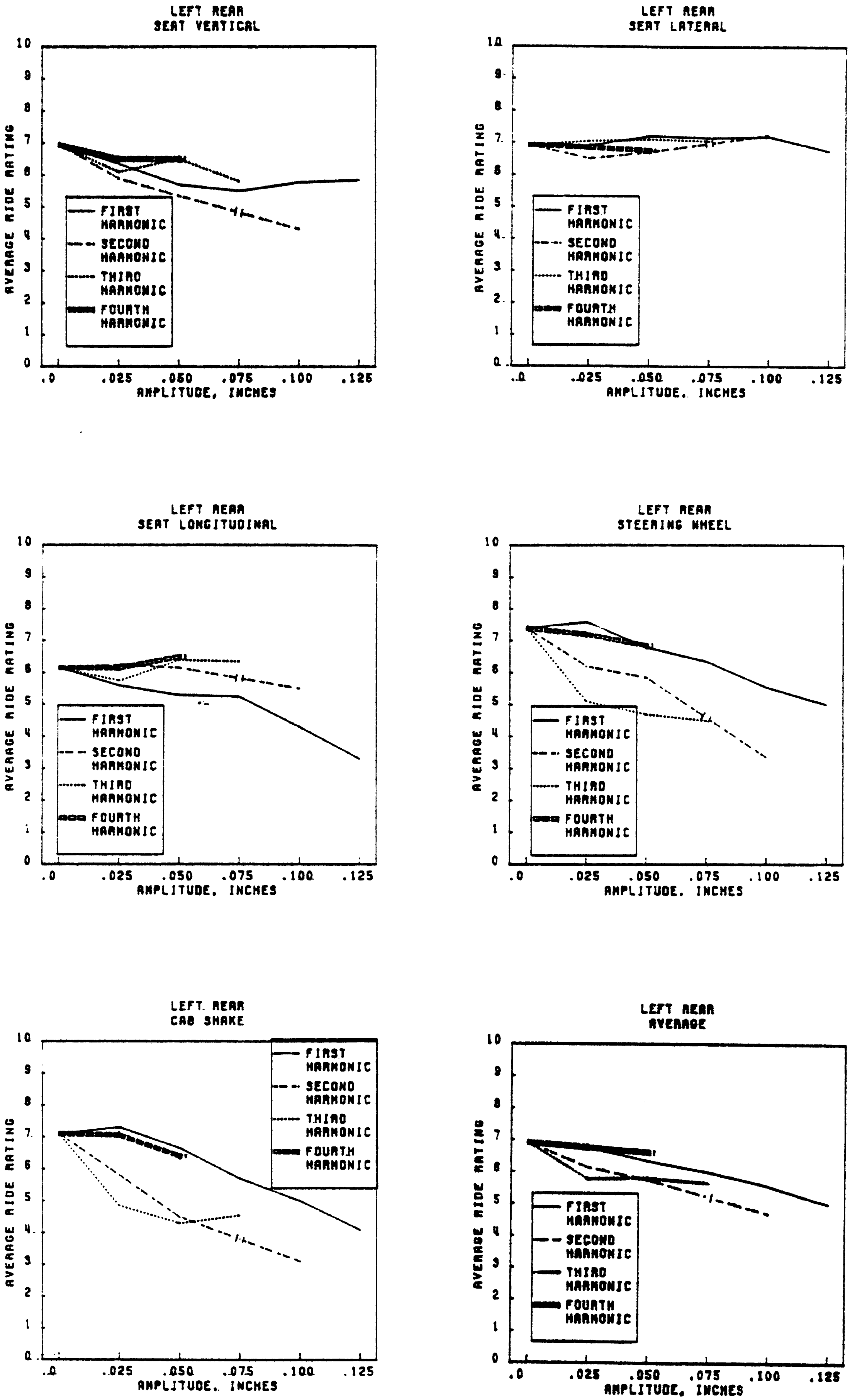


Figure 19b. Rating effects of left rear nonuniformity excitation.

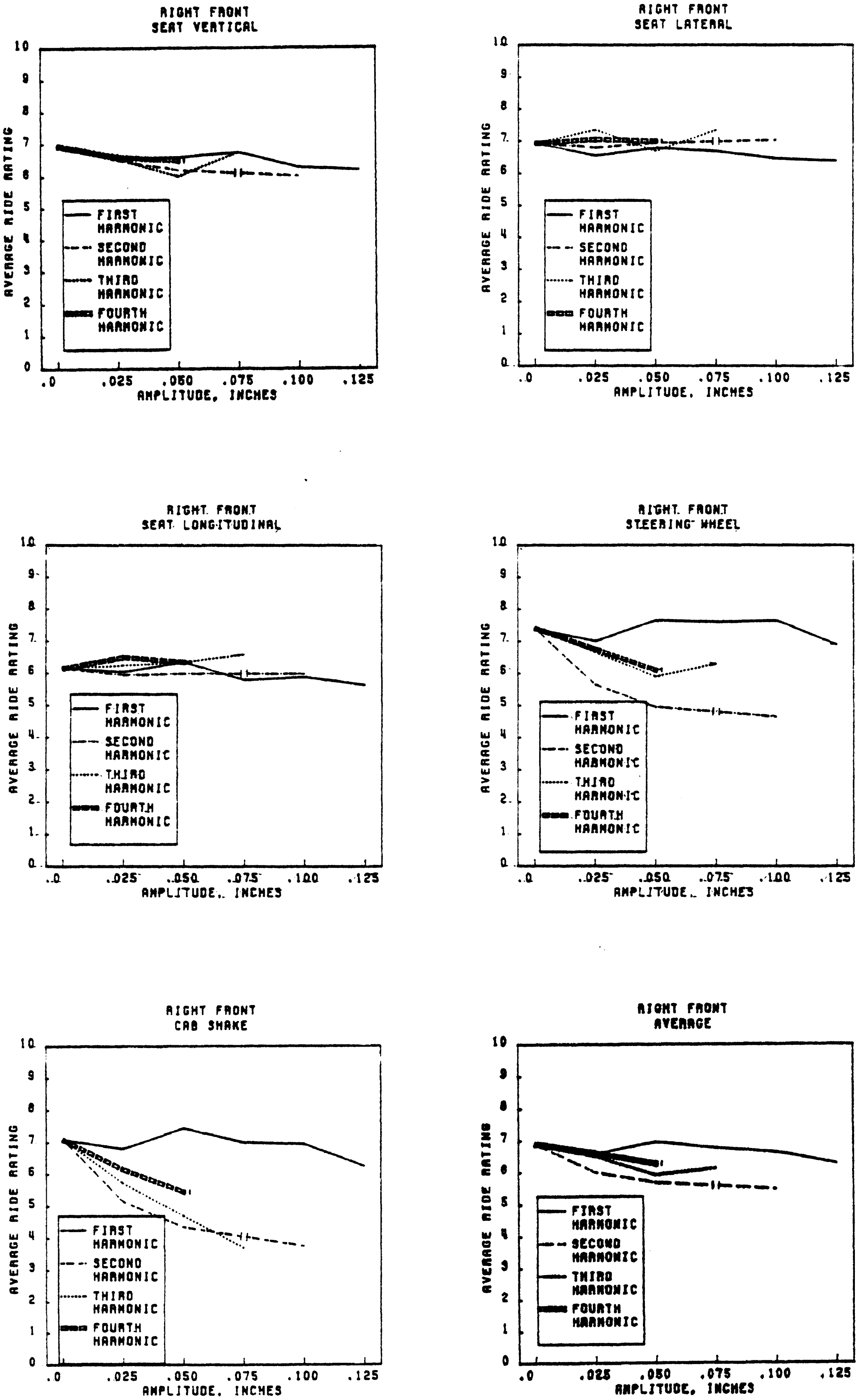


Figure 19c. Rating effects of right front nonuniformity excitation.

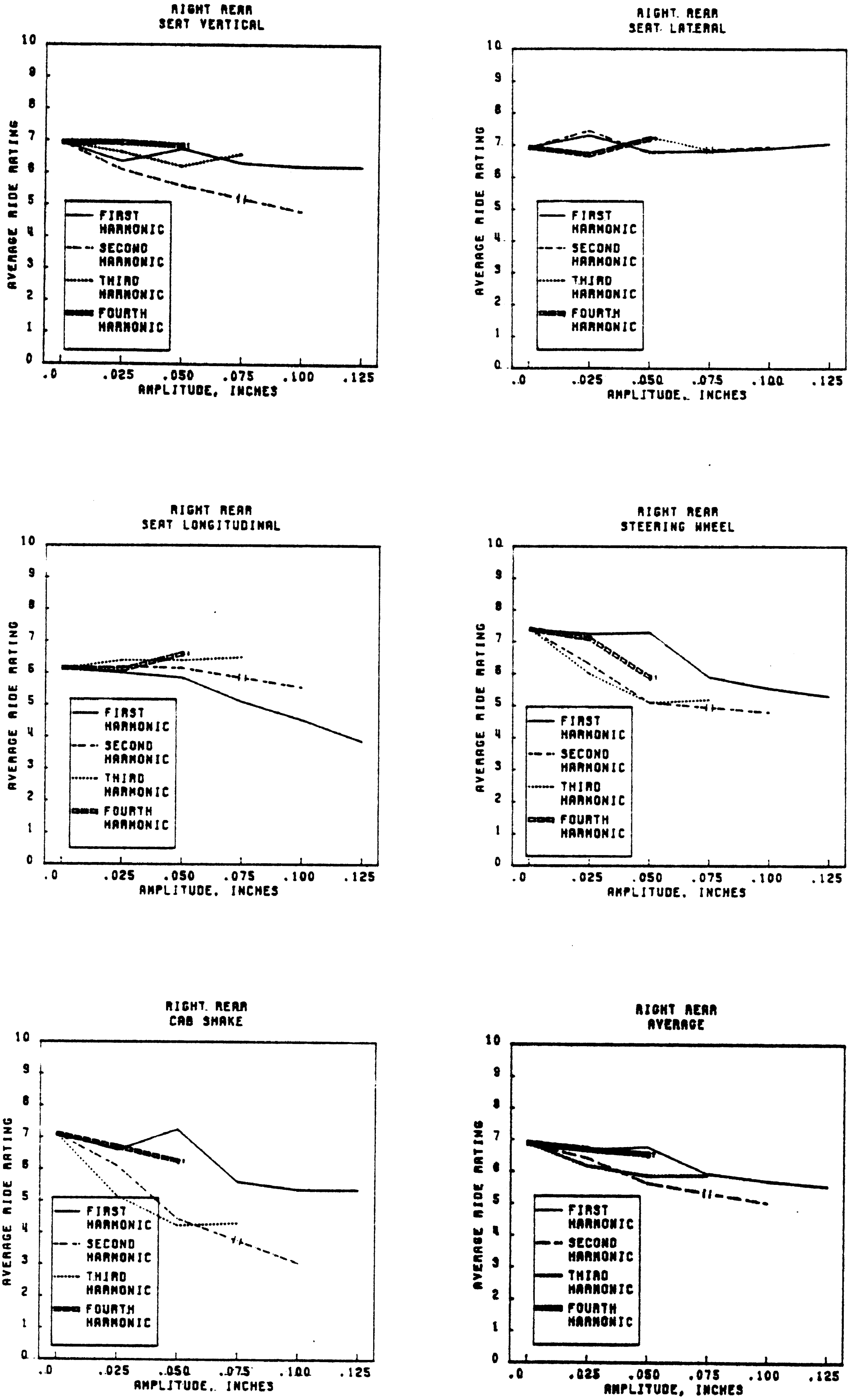


Figure 19d. Rating effects of right rear nonuniformity excitation.

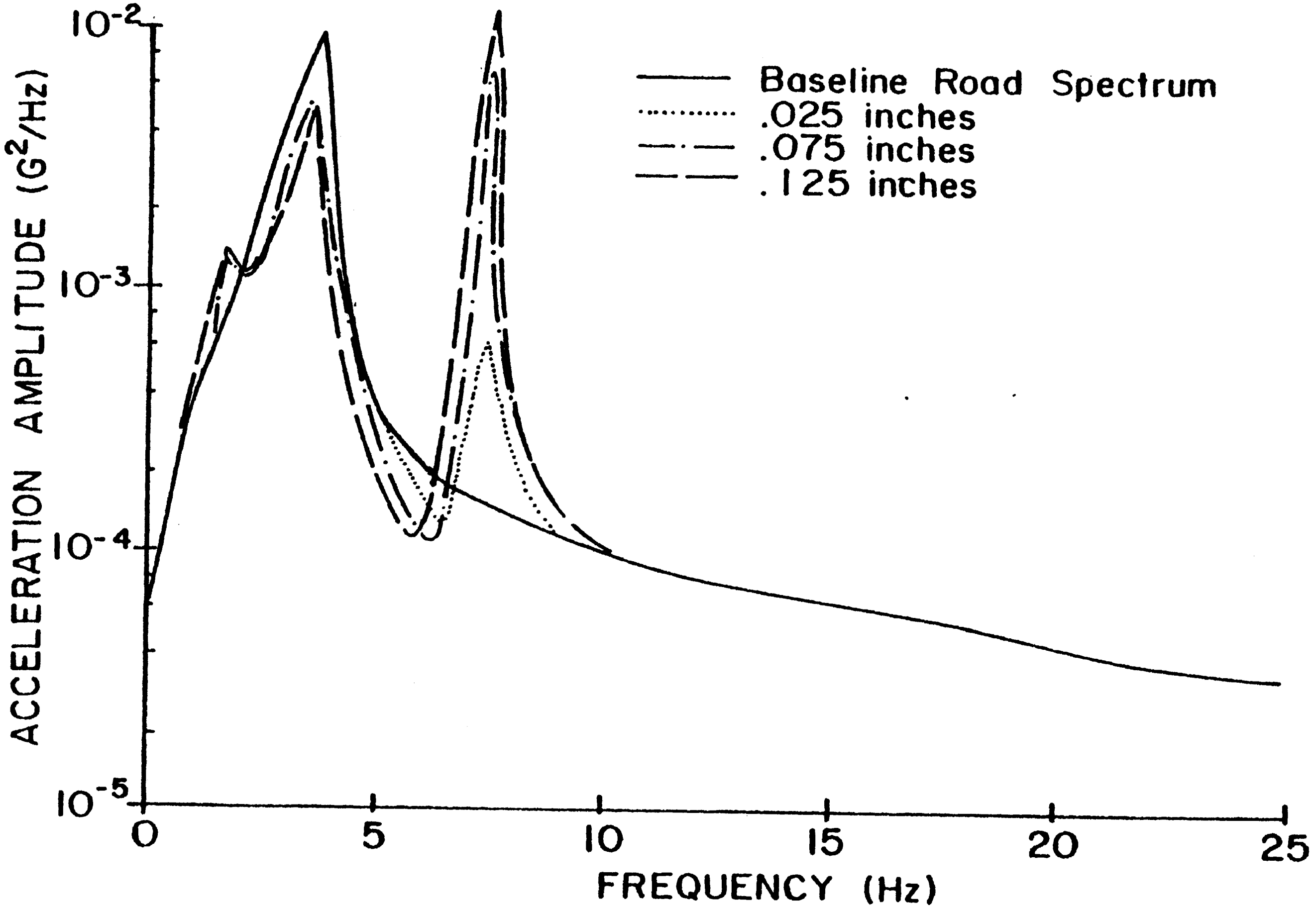


Figure 20. Seat vertical vibration spectra for first harmonic excitation at the left front wheel position.

This is most probably due to the increase of hysteretic damping in the suspension system with the additional deflection caused by the nonuniformity excitation. This effect was not unexpected, but had been predicted in earlier analysis [9].

The overall effect of each harmonic is best seen in the Average Ratings. Consider, for example, the left front case shown in Figure 19a. The effect of a first harmonic nonuniformity on this wheel is to decrease the ride rating the most quickly up to an amplitude of 0.05 inches, and thereafter decrease the rating at a lower rate. A similar trend is indicated for the first harmonic on the right front wheel (Fig. 19c) as well, although it only holds up to the first level of 0.025 inches. More consistent first harmonic influences are seen at the rear wheel positions, where the trends continue out to the largest amplitudes tested. Because of the modest sensitivity to first harmonic excitation in every case, it is difficult to pinpoint any one of the five rating factors as the source of the degradation. At best, one can conclude that the first harmonic has very little effect on the seat lateral, but it appears to show up to some extent in all the other four rating factors.

By and large, the second harmonic has a much more pronounced effect. The degradation rate of the average ride rating is quite similar for all wheel positions. Although this rate is higher, it should be kept in mind by the reader that second harmonic magnitudes in tire/wheel assemblies are generally much lower than first harmonic effects. A more careful examination of the individual rating plots in Figure 19 reveals that second harmonic effects are perceived primarily in the cab shake and steering-wheel vibration categories, although it is also evident in the seat vertical rating.

A pronounced sensitivity to third harmonic excitation is also indicated by the data. The third harmonic degradation rates are less linear than the second, but at low levels have a higher rate at the rear wheel positions. Again, the degradation effects are seen with some lesser influence in the seat vertical factor.

The fourth harmonic, in general, is seen to have a rather minimal effect on ride rating. Though it has a degradation rate comparable to the first harmonic, fourth harmonic magnitudes are generally only a fraction of the level of first harmonic values. Additionally, fourth harmonics are limited to that present in the tire only, as a fourth harmonic present in a wheel is masked out by the tire.

There are several options for reducing the data in these plots to a useable form. Since the data are nonlinear in some cases, the most critical method is to determine the greatest slope (ride degradation rate) among the lower amplitudes representing that to be the nominal degradation rate associated with a nonuniformity condition. While the lower amplitudes are expected to be more representative of actual excitation levels on real vehicles, the gradients obtained can be somewhat erratic due to the random error in the subjective rating process. A second option is to apply statistical methods, namely, a linear regression. The slope of the linear regression equation is then the degradation rate of interest. Table 2 summarizes the results for both methods.

As evident in the above summary, evaluation at low amplitudes yields some very high sensitivity factors. At this juncture, it is difficult to argue that one is necessarily more valid than the other. Nevertheless, in the analysis of combination effects, the linear regression coefficients will be taken as the preferred choice to quantify nonuniformity influences.

Among the single-wheel input tests, a frequency sweep down to 3.3 Hz was conducted to ensure that the ratings covered the low speed condition of rigid-body bounce/pitch as excited by tire/wheel inputs. The tests were limited to left front and left rear wheel input at an amplitude of 0.050 inches. Figure 21 shows the results of those tests plotted separately for each rating factor and for the average.

For both front and rear wheel input, the average rating does not change profoundly with frequency, even down to the point of the 3.3 Hz bounce/pitch range. The ratings of the individual factors, however, do show some sensitivity which is washed out in the averaging process. Not surprisingly, the seat vertical rating diminishes with either input, while the seat fore/aft (longitudinal) is most noticeably degraded by rear wheel input. The explanation of why the average is unaffected is not apparent. The answer may, perhaps, lie in the way in which nonuniformity excitation changes the suspension damping and hence the overall spectrum.

This pseudo-speed effect is not seen as an important influence on truck ride vibrations at this stage in the research. Though a slight change in

Table 2. Tire/Wheel Nonuniformity Ride Degradation Rates (Units of Ride Rating/Inch of Excitation[1])

	Wheel Position			
	Left Front	Right Front	Left Rear	Right Rear
1st Harmonic*	- 7.5/.050	-12.6/.025	-15.2/.125	-10.8/.125
**	- 2.93	- 3.59	-15.3	-12.1
2nd Harmonic*	-24.4/.025	-35.8/.025	-31.0/.025	-25.3/.050
**	-13.57	-12.87	-21.35	-19.1
3rd Harmonic*	-21.8/.025	-19.5/.050	-45.4/.025	-28.6/.025
**	- 8.26	-11.5	-14.94	-13.46
4th Harmonic*	- 5.1/.050	-12.5/.050	- 6.1/.050	- 6.9/.050
**	- 5.1	-12.3	- 6.1	- 6.9

[1]Excitation is equivalent to inches of loaded radial runout.

*Gradient for the range of zero amplitude to/(amplitude shown).

**Slope of linear regression line for all data points.

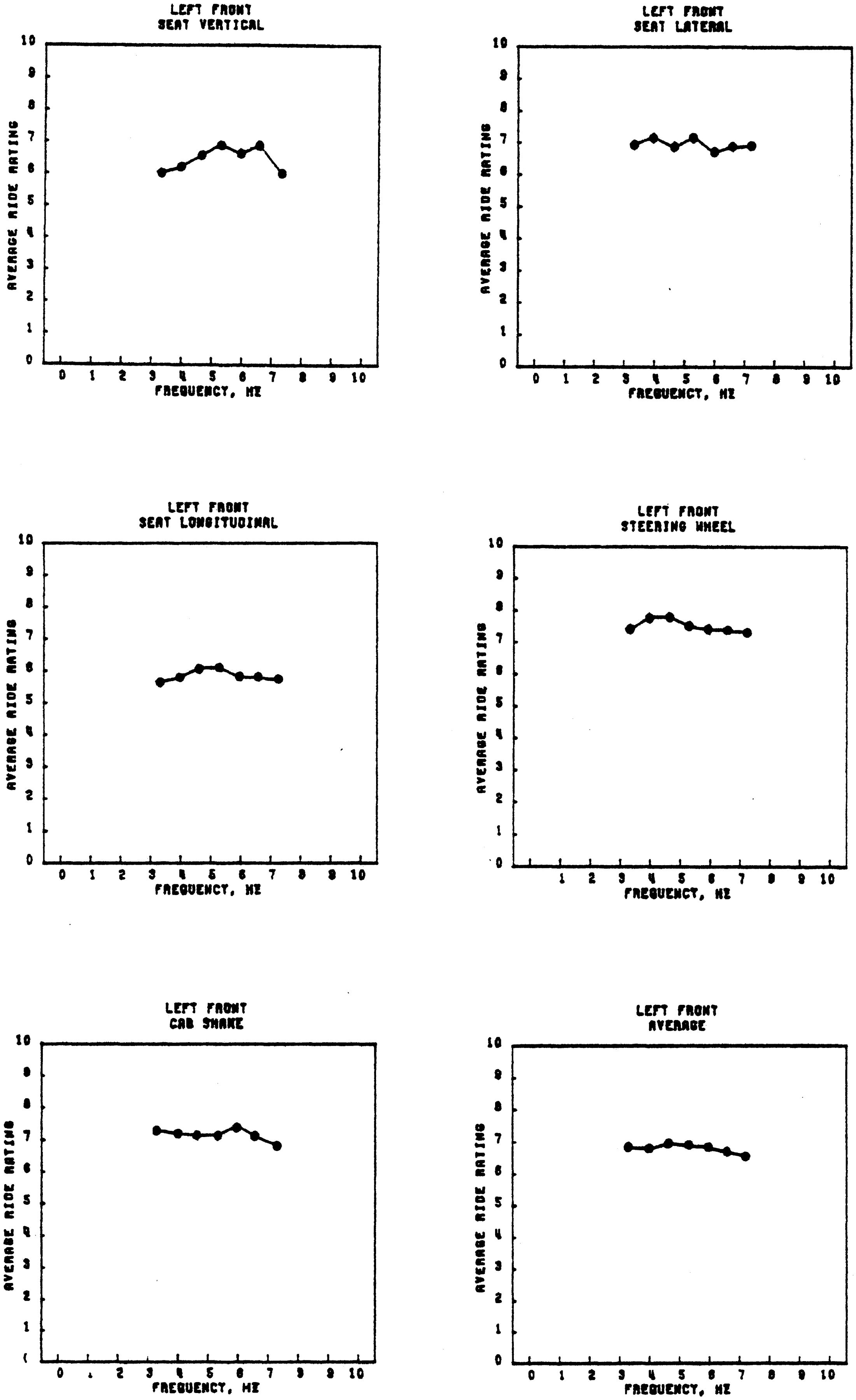


Figure 21a. Ride ratings at low frequencies for 0.05 inch excitation at the left front.

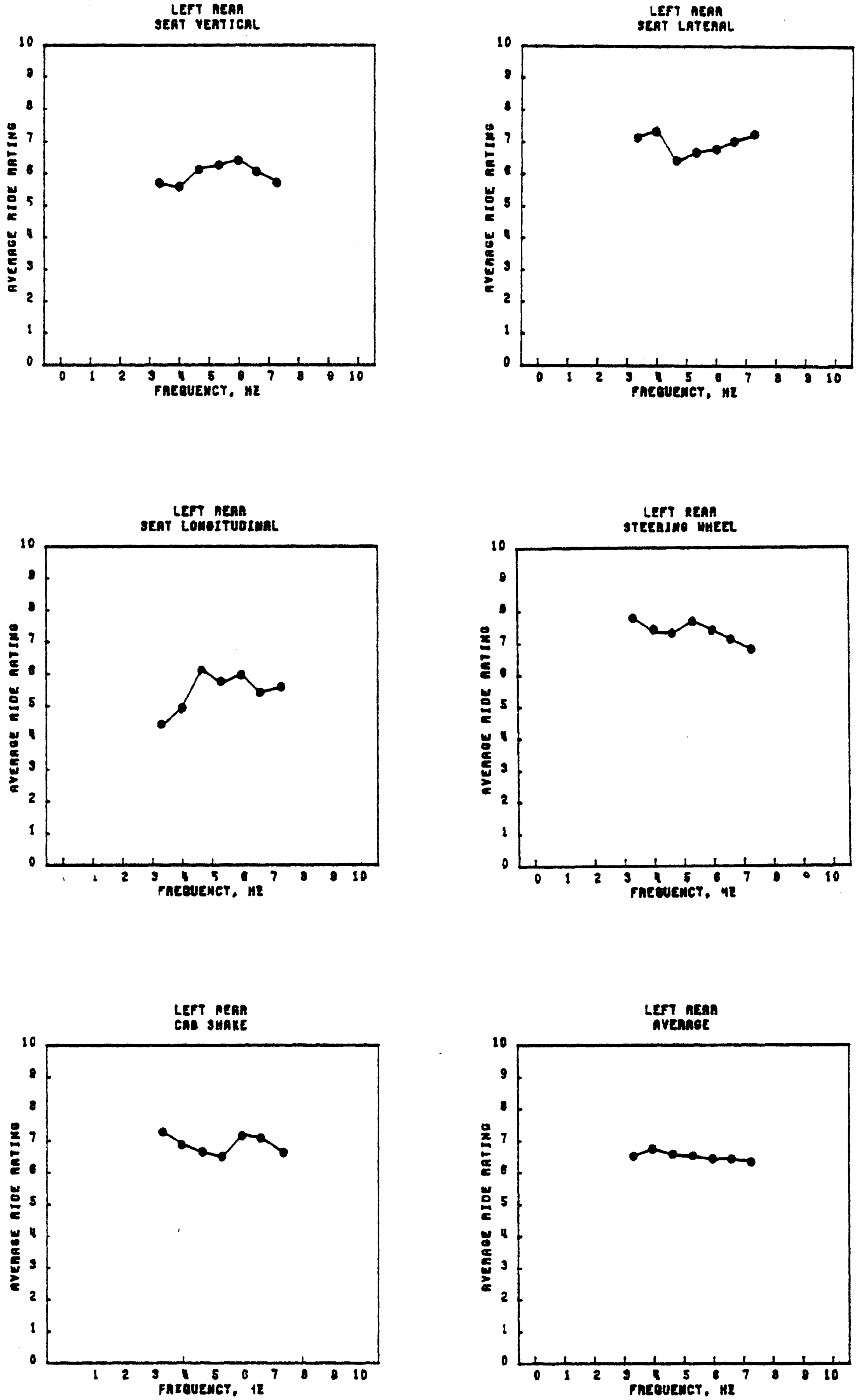


Figure 21b. Ride ratings at low frequency for 0.05 inch excitation at the left rear.

sensitivity to the first harmonic may be indicated, the accompanying changes of road input at lower speed is likely to overwhelm any direct wheel effects.

4.4 Ratings of Nonuniformity Combinations

Among the 100 tests, some were designed to explore the effects of combinations of nonuniformity inputs, inasmuch as the overall vehicle ride is never the result of only one nonuniformity, but the combinations of a number of inputs at every wheel. Each dimension of combination was explored separately.

4.4.1 Combinations on One Axle. Nonuniformity inputs on both the left and right side of an axle were tested for both the front and rear axles. The tests were confined to the first harmonic condition only, but covered the in-phase, out-of-phase conditions. Five amplitudes (including the null condition) were tested. The degradation rates were determined from linear regression analysis. Table 3 summarizes the degradation rates for the in-phase and out-of-phase conditions, and includes, for comparison, the appropriate rates for single-wheel excitation.

Table 3. Tire/Wheel Nonuniformity Degradation Rates for First Harmonic Inputs at the Left and Right Wheels of an Axle.

	Input			
	Left Only	Right Only	Left-Right Bounce (In Phase)	Left-Right Roll (Out of Phase)
Front	- 2.93	- 3.59	- 8.26	-10.28
Rear	-15.32	-12.14	-25.74	- 0.12

The total degrdation rate that would be expected from the front axle would be -6.52 if the left and right sensitivities were purely additive. Though an underestimate of the actual effect, the error is less than 40 percent when predicting the worst case roll input.condition. Considering the measurement error associated with determination of each of the individual rates, this error percentage is not very significant. If this level of accuracy in the prediction can be accepted, then it can be concluded that:

> "The combined effect of concurrent left and right non-uniformity inputs on the front axle is (by first-order estimate) the sum of the individual sensitivities of both the left and right wheels. Further, the combined effect is independent of wheel phasing."

On a similar basis, the rear axle data would indicate a conclusion that:

> "The combined effect of concurrent left and right non-uniformity inputs on the rear axle is the sum of the individual sensitivities of both the left and right wheels when the inputs are in phase. When the left and right inputs are out of phase, they do not produce a ride degradation."

For the rear axle, the total of the left and right (-27.46) is much closer to the sensitivity found in the in-phase combination. The fact that the roll combination appears to cancel at the rear axle does not confound the use of this rule. The overall vibration on the road from the rear axle will wax and wane as the wheel phasing changes, being noticeable to the rider during the worst case (in-phase) combination. Whether the rider degradation will be judged on the basis of worst case or average effects is not known at this time, but is a good topic for future research. Unfortunately, the test program did not allow latitude for exploring these same effects for second and higher harmonics, hence it can only be assumed that these observations hold for the other harmonics. However, as will be seen, that assumption is appropriate for a number of other conditions involving higher harmonics, and therefore represents a best first guess as to how to treat higher harmonic combinations on one axle.

4.4.2 <u>Combinations on Two Axles</u>. The front/rear combinations of first harmonic were explored in the left front/left rear wheel positions. Out of concern that phasing might be important, four tests were conducted covering the phase angles of 0, 90, 180, and 270 degrees. The results are summarized in Table 4 below.

Table 4. Tire/Wheel Nonuniformity Ride Degradation Rates for First Harmonic Combinations on Front and Rear Wheels.

	Phase Angle	Degradation Rate	Average Rate
Front Only	--	- 2.93	-18.22*
Rear Only	--	-15.3	
Both	0	-17.0	-16.0
Both	90	-14.7	
Both	180	-17.5	
Both	270	-14.7	

*Average value of 9.11 times two wheels.

With allowances for the measurement variability, it would appear that the phase influence, if anything, is a second-order effect. The average value of -16.0 for both front and rear inputs compares closely to the individual left front and left rear total of -18.22. Therefore, it may be concluded that:

> "The combined effect of concurrent front and rear nonuniformity inputs is the sum of the individual sensitivities of the front and rear axles, regardless of phasing."

The applicability of the above conclusion for harmonics other than the first is demonstrated in the test results for front/rear combinations involving different harmonics. In general, the relative amplitudes of each harmonic in a combination can differ, in which case the method of summing the gradients should not be used. Rather, the decrement must be computed (using the gradient × harmonic amplitude) for each component, and then the decrements for all components are added. However, for these tests an amplitude of 0.075 was used for all harmonic inputs, so that they can be compared by simple summation of the gradients. The test matrix for this combination consisted of a left front first harmonic input along with left rear inputs of second or third harmonic at differing phase angles. The results are summarized in Table 5 in which the actual rate is shown along with the rate predicted by additive gradients.

Table 5. Tire/Wheel Nonuniformity Ride Degradation Rates for Higher Harmonic Combinations on Front and Rear Wheels.

Test Condition	Phase Angle	Actual Rate	Predicted Rate
LF 1st Harmonic		-25.0	-24.2
LR 2nd Harmonic	0		
LF 1st Harmonic		-27.0	-24.2
LR 2nd Harmonic	90		
LF 1st Harmonic		-13.3	-17.8
LR 3rd Harmonic	0		
LF 1st Harmonic		-20.5	-17.8
LR 3rd Harmonic	90		
LF 1st Harmonic		-19.3	-17.8
LR 3rd Harmonic	180		

Looking first at the combination with the second harmonic on the rear axle, it is evident that there is little effect of phasing. Though the phase angles used produced different wave form combinations, their effect on rating is rather insignificant. In the case of the third harmonic the data would suggest potential for a small phasing effect; but again it is not well established by the data, and is not of first-order significance. Thus the previous conclusion regarding the ride degradation of front axe/rear axle combinations appears to be applicable.

Lastly, it should be noted that several tests in the matrix were dedicated to investigating whether the arbitrary choice of the phasing of wheel nonuniformity effects with respect to the road input was consequential to the results. When a first harmonic input was lagged at 90 and 180 degrees with respect to its usual registration on the road, no significant change in ratings was obtained.

4.5 Predicting Ride Degradation Arising from Radial Nonuniformities

From the findings seen in the preceding section, the prediction of ride degradation for the test tractor at 55 mph is reasonably straightforward. The prospect of being able to add together a number of individual nonuniformity contributions greatly simplifies the method. At this point, an example is helpful to demonstrate application of the method, and perhaps at the same time, to add confidence in its validity. For that example, the method will be applied to the on-road ride test of the GMC tractor to predict the portion of the ratings attributable to the radial nonuniformities in its tire/wheel assemblies.

To the extent possible, the first harmonic radial nonuniformities on all wheels of the test tractor were minimized by match mounting, except for the left front wheel. In addition, all wheels were carefully balanced. Second and higher harmonics were not adjusted. The first through fourth harmonic radial force variations on all wheels were measured. Because the ride degradation rates are described in terms of ride rating per inch of excitation amplitude, those force variations must be converted to equivalent loaded radial runout displacement. This is done by dividing the force variation amplitude by the tire radial stiffness. A stiffness of 5000 lb/in is assumed in the calculations. The associated runout for each harmonic and wheel position is then multiplied by the appropriate ride degradation rate, after which the contributions from all wheels and harmonics are summed. Table 6 shows these calculations and totals.

As indicated, the total ride decrement due to the first through fourth harmonic radial nonuniformities is -0.95 ride points. The average on-road rating for the vehicle (average of three rating factors over eight raters) was 7.83 on the 0-10 rating scale. Had all the tire/wheel assemblies been perfect with respect to radial force variations, the method would then predict that an average rating of 8.78 would have been obtained. That is, of the total decrement of 2.17 ride points (10 - 7.83) arising from excitation by the road, tire/wheel inputs, and other sources, 44 percent is due to the tire/wheel radial excitation. Based on the author's experience, this constitutes a very reasonable result. On the one hand, it is unreasonable to expect that the total effect of all radial excitation would be significantly less than a ride point considering that the mean square ride vibrations

Table 6. Ride Degradation Calculations for Radial Nonuniformities on the GMC Test Tractor.

Wheel Position	Harmonic	LRR Amplitude		Ride Degradation Factor		Ride Rating Decrement
LF	1	.0369 in	×	- 2.93 RR/in	=	-.108
	2	.00712	×	-13.57	=	-.0966
	3	.01446	×	- 8.26	=	-.119
	4	.00258	×	- 5.1	=	-.013
RF	1	.00886	×	- 3.59	=	-.0318
	2	.00544	×	-12.87	=	-.0700
	3	.00408	×	-11.5	=	-.0469
	4	.00602	×	-12.3	=	-.0740
LR	1	.00304	×	-15.3	=	-.0465
	2	.00368	×	-21.35	=	-.0786
	3	.00297	×	-14.94	=	-.0444
	4	.00223	×	- 6.1	=	-.0136
RR	1	.00633	×	-12.1	=	-.0766
	2	.00564	×	-19.1	=	-.1077
	3	.00085	×	-13.46	=	-.0114
	4	.00196	×	- 6.9	=	-.0135
					Total	-.9516

attributable to wheel excitation, seen in the spectral maps of Figure 11, are a significant portion of the total. On the other hand, it would also be unreasonable for the method to predict significantly more than a ride point because of the implication that with only the road and driveline inputs the ride rating would be above 9.

As a matter of interest, these data can be used to estimate the ride degradation that would be expected on the test vehicle with the wheels set up at different conditions. Inasmuch as it is only practical to adjust first harmonic levels (by balancing, match mounting, etc.), three cases representing different levels of first harmonic are considered. As a "worst case" it might be assumed that all four wheels had a first harmonic nonuniformity equivalent to 0.04 inches of loaded radial runout. This may be compared against the vehicle "as tested" with all but the left front wheel balanced and match mounted to minimize first harmonic. Then the best case would be with the left front wheel dressed to the level achieved on the right front wheel—this representing the best case possible by the practice of match-mounting and balancing all wheels on a vehicle.

Table 7 shows the calculated total ride decrement associated with each harmonic for these three cases. In the worst case, an overall ride decrement of 2 ride points would be expected, with 66 percent of that due to first harmonic nonuniformities. At the other extreme, with balancing and match mounting to minimize first harmonics on all wheels, the overall ride decrement attributable to tire/wheel nonuniformities can be reduced to 0.87 ride points, leaving the second and third harmonics as the major contributing elements.

As a second case of interest, one might consider the implications of these findings as they would predict ride degradation resulting from the radial excitation by mass imbalance in the tire/wheel assembly. A 100 in-oz imbalance on a truck wheel at 55 mph equates to a 34-lb excitation force, based on the equation:

$$\text{Force} = \text{Mass} \times \text{Radius} \times \text{Rotation Speed Squared}$$

Table 7. Example Cases of Ride Rating Degradation with Different Wheel Treatments.

	Assuming 0.04" Loaded Radial Runout at All Wheels		3 Wheels Match Mounted LF Wheel Unchanged		Balancing and Match Mounting for 1st Harmonic	
	Decrement	Percent	Decrement	Percent	Decrement	Percent
1st Har.	1.3568	66.3	.2629	27.6	.1867	21.3
2nd Har.	.3529	17.3	.3529	37.1	.3529	40.3
3rd Har.	.2217	10.8	.2217	23.3	.2217	25.3
4th Har.	.1141	5.6	.1141	12.0	.1141	13.1
	2.0455	100.0	.9516	100.0	.8754	100.0

Translating this to an equivalent first harmonic loaded radial runout (again assuming a tire stiffness of 5000 lb/in), it is equivalent to 0.0068 inches. Multiplying by the front wheel ride degradation factors for the GMC tractor leads to the conclusion that 100 in-oz on both front wheels will yield a ride degradation of 0.044 ride points. Because of the higher ride rate sensitivity on the rear wheels, 100 in-oz on both rear wheels equivocates to a degradation of 0.093 ride points. For a typical truck that had production balance specifications of 200 in-oz on front wheels and 500 in-oz on rear dual wheels, a total ride decrement of more than a half point could result from imbalance alone.

4.6 Applicability to Other Frequencies and Force Directions

At the inception of the project, a very mechanistic approach was taken toward relating ride degradation to tire/wheel excitation. If systematic relationships between ride ratings and specific features in the seat acceleration spectra could be determined, then the knowledge of the truck response to other frequencies and force directions (as characterized in Figs. 16 and 17) would allow associating ride decrements with the spectra predicted for other untested conditions. By that means, the subjective ratings acquired over a necessarily limited range of nonuniformity test conditions could be applied to predict results for the nearly infinite number of other possible conditions. The observations from this research, however, suggest at least three major impediments to any such systematic approach.

1) The ride ratings are clearly not products of only the seat accelerations. As seen in Appendix A, very poor correlation exists between the average jury ride rating and any of seven summary measures of seat acceleration. In essence, this implies that no measure of seat acceleration exists that is a good predictor of the effects on the subjective rating of seat vibration. Further, jury members would undoubtedly agree that the seat input is not the only perceived source of objectionable vibration, but clearly includes a generalized "cab shake" phenomenon, as well as hand and feet inputs.

2) The measured seat accelerations varied significantly from jury member to jury member. In cases where measures of identical test conditions were obtained with several jury members, some of the RMS seat vibration levels differed by more than 30 percent.

3) The spectral changes in seat acceleration with amplitude (see Fig. 20) preclude any systematic association of ride decrements with singular spectral features arising from a particular nonuniformity input. Complex spectral changes in a multi-directional response defy such simple associations.

Under these circumstances, the mechanistic approach cannot be used with a hope of accurate predictions. Instead, the nonuniformity conditions of interest must be explored on an individual basis in the subjective testing. That is, the ride degradation deriving from fore/aft (longitudinal) excitation must be measured directly by a jury evaluation with the truck exposed to fore/aft excitation. Further, those tests should explore both the coherent and incoherent combinations of vertical and fore/aft excitations; the coherent representing the action of mass imbalance with its constant phase relationship between the two directions.

On the positive side, it can be stated that with the current level of understanding of the tire/wheel excitation phenomenon, it is anticipated that the fore/aft direction embodies fewer nonuniformities of serious concern. In the first harmonic range, the truck response sensitivity to fore/aft is comparable to that of the vertical (see Figs. 16 and 17) and is thus rather low. Accordingly, the rider sensitivity to the vertical is also low. The preliminary results from the Phase I testing indicate that first harmonic tractive force variations from typical truck tire/wheel assemblies (exclusive of that caused by mass imbalance) generally do not exceed 40 pounds, and are only about 25 percent of the radial magnitudes. Therefore, first harmonic tractive force excitation is likely to present a much less serious influence than that from radial force excitation.

On the other hand, a mass imbalance generates equivalent excitation in both the radial and tractive force directions nominally equal to 34 pounds for every 100 in-oz of imbalance. Hence it is anticipated that the primary interest in tractive force excitation will focus on the effects of mass imbalance.

At this juncture, little has been learned about higher harmonic tractive force variations in the Phase I work because of the frequency limitations of the tire test machine. The assessment of the importance of higher harmonic tractive force variations therefore cannot be made at this time.

5. CONCLUSIONS AND RECOMMENDATIONS

5.1 Introduction

The research performed in this project has proved very rich in findings that enhance the understanding of truck ride vibrations and their influence from tire/wheel nonuniformities. The overall findings that relate to the methodology tested are summarized in the General Conclusions that follow. Those relating specifically to the sensitivities of the tractor tested in the project are presented as a separate section of conclusions. Finally, the findings are utilized to propose a series of recommendations providing guidance as to how to improve the methodology for testing, as well as what further testing would be suggested in order to better characterize truck sensitivity to tire/wheel nonuniformity input.

5.2 General Conclusions

1) It is possible to replicate smooth-road truck vibrations along with radial tire/wheel nonuniformity effects on a hydraulic road simulator in a way that allows meaningful rating of the vibrations by a jury panel. Though the absolute ratings differ from their on-road equivalent (due to the differences between the laboratory and on-road environments), the rating decrements arising from nonuniformity inputs appear to be reasonable. The simulator method provides a very quick and efficient means for obtaining subjective judgments of ride degradation arising from nonuniformities of different amplitudes, frequencies, wheel positions, and combinations thereof.

2) Based on results obtained from the one truck tested, the combined effect of multiple nonuniformities (different harmonics and/or different wheel positions) can be approximated by using superposition. That is, the ride decrement associated with a combination of nonuniformity inputs is nominally equivalent to the sum of the decrements associated with the individual inputs making up the combination. Though the phase relationship of nonuniformity components within a combination may have an effect on ratings in some cases, the "worst case" phase combination of primary concern seems to fit the above method of predicting ride degradation for combinations.

3) No means were found for predicting subjective ride ratings from measures of seat acceleration. The primary reason is that "cab shake" and "hand/foot" vibrations are found to be important elements of the truck vibration environment affecting rider judgment. In addition, through their effect on the nonlinear suspension system, tire/wheel nonuniformity inputs alter the truck's vibration response to road roughness inputs.

4) As a consequence of the preceding conclusion, ride degradation due to nonuniformity inputs cannot be predicted from engineering models of a truck's vibration response. Therefore, to assess rider sensitivity to nonuniformity inputs of different frequencies, directions, or wheel positions, subjective rating tests must be conducted under these different conditions.

5.3 Conclusions - Sensitivity of the GMC Tractor to Tire/Wheel Nonuniformities

For the two-axle GMC cab-over-engine tractor tested in this project, the sensitivity to tire/wheel nonuniformity inputs (as detailed in Table 2) may be summarized as follows:

1) The vehicle is most sensitive to radial nonuniformity inputs at the rear axle. The sensitivity (ride degradation rate) is greatest for the second and third harmonics.

2) First harmonic radial nonuniformities may account for ride rating losses of anywhere from 0.2 to 1.4 ride points (on a 0-10 point scale), depending on wheel setup.

4) Typical wheel imbalances may account for ride point losses in the range of 0.1 to 0.5 points.

5) In the absence of radial tire/wheel nonuniformity inputs, the basic truck would exhibit a smooth road ride rating in the range of 8.5 to 9.0 points. With typical nonuniformities, that rating could diminish to the range of a 6 to 7 ride rating.

5.4 Recommendations

The findings from this research program have shown tire/wheel nonuniformities to be a significant source of smooth-road ride degradation. Further research is recommended to better understand the influence of nonuniformity of other types, and on other vehicles. Specific research recommendations are as follows:

1) Not only the first harmonic, but also higher harmonics are important sources of ride degradation. The higher harmonics therefore merit critical examination at both the truck and tire/wheel testing stages.

-In the phase I investigation of cyclic force variations emanating from tire/wheel assemblies, test capability to at least 50 Hz is recommended.

-In future measurements of truck sensitivity, examination of the fifth, and perhaps sixth, harmonics should be considered on at least one truck to ensure that vital sensitivities are not being overlooked.

2) The hydraulic road simulator method is a very efficient and effective means for subjective measurements of truck sensitivity to tire/wheel excitation. It offers a versatility in measurement that makes it practical to separate the ride effects associated with individual nonuniformity inputs in a subjective rating exercise. In future research, equipment should be assembled to allow exposure of the trucks to tractive force, lateral force, and aligning moment excitations, so that sensitivities to these excitation sources can be assessed.

3) Additional research should be instituted to document the ride sensitivity of additional tractors. The popular three-axle tractor of the conventional cab and cab-over-engine types should be examined. More complete information on the ride sensitivity of tractors is needed to help identify the types of tire/wheel nonuniformities to be given emphasis in the research on tire and wheel components.

NOTE: In the past, drum roller and on-road test methods have often been proposed as means to evaluate truck sensitivity to nonuniform excitations. The drum roller test method facilitates inputs of actual nonuniformity conditions, but is very limited in the degree to which test conditions can be varied. At best, it is only suited to studies of first harmonic effects for which the amplitudes can be varied by the addition of wheel balance weights.

On-road test methods have the advantages of being more realistic and incorporating the concurrent inputs, but are even more limited in the capability to cover multiple test conditions.

The road simulator method is the most versatile and could be expanded to include the other desired features by the additions of multi-directional exciters to the wheels. The approach involves a higher initial investment, but substantially lower expense for actual testing. At this stage, the knowledge of the truck as a dynamic system is sufficient to allow design of such a system. Therefore, it is recommended that the development of methodology in this direction be pursued.

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APPENDIX A

DESCRIPTION OF THE SUBJECTIVE RIDE RATING TESTS ON THE HYDRAULIC ROAD SIMULATOR

Test Procedures

As described in the main sections of the report, the 1981 GMC test tractor was installed on the four-position hydraulic road simulator at the International Harvester Company Engineering Center in Fort Wayne, Indiana. The tractor was loaded at the fifth wheel via a ballast trailer. Hardware controlling the simulator was configured to allow repeated playback of a smooth-road profile into each of the vertical actuators under the tractor wheels. The playback corresponded to a 55-mph road speed, with separate left and right profile inputs, and time lags between the front and rear wheel inputs. The hardware provided for the addition of sine waves to the road profile signal, replicating arbitrary nonuniformity conditions. These conditions were varied in amplitude, frequency, wheel position, combinations, and the phasing of combinations.

A 100-test sequence was devised covering 90 nonuniformity conditions of interest, plus 10 repeats. The nonuniformity conditions represented in each of the tests are listed in Table A-1. The first 21 tests were used to investigate the influence of first harmonic nonuniformity inputs at the individual wheel positions. Six amplitudes were used—the null condition, and five amplitudes from 0.025 to 0.125 inches of sine wave amplitude. In tests 22 through 53, the effects of second, third, and fourth harmonics were tested at the individual wheel positions in a lower range of amplitudes. Tests 54 through 79 were dedicated to exploring the effects of combined inputs, either on both wheels of one axle or on wheels of different axles. These tests explored the following:

- -Tests 54-57 - Effects of first harmonic phasing under front and rear wheel input
- -Tests 58-62 - Effects of first harmonic on the front axle with second or third harmonic on the rear axle at different phase conditions

TABLE A-1
Nonuniformity Test Conditions

Test	Left Front Amp. In.	Left Front Freq. Hz	Left Front Deg.	Right Front Amp. In.	Right Front Freq. Hz	Right Front Deg.	Left Rear Amp. In.	Left Rear Freq. Hz	Left Rear Deg.	Right Rear Amp. In.	Right Rear Freq. Hz	Right Rear Deg.
1												
2	.025	7.3	0°									
3	.050	7.3	0°									
4	.075	7.3	0°									
5	.100	7.3	0°									
6	.125	7.3	0°									
7				.025	7.3	0°						
8				.050	7.3	0°						
9				.075	7.3	0°						
10				.100	7.3	0°						
11				.125	7.3	0°						
12							.025	7.3	0°			
13							.050	7.3	0°			
14							.075	7.3	0°			
15							.100	7.3	0°			
16							.125	7.3	0°			
17										.025	7.3	0°
18										.050	7.3	0°
19										.075	7.3	0°
20										.100	7.3	0°
21										.125	7.3	0°
22	.025	14.6	0°									
23	.050	14.6	0°									
24	.100	14.6	0°									
25	.025	21.9	0°									
26	.050	21.9	0°									
27	.075	21.9	0°									
28	.025	29.2	0°									
29	.050	29.2	0°									
30				.025	14.6	0°						
31				.050	14.6	0°						
32				.100	14.6	0°						
33				.025	21.9	0°						
34				.050	21.9	0°						
35				.075	21.9	0°						
36				.025	29.2	0°						
37				.050	29.2	0°						

TABLE A-1 (Continued)

Test	WHEEL LOCATION											
	Left Front			Right Front			Left Rear			Right Rear		
	Amp. In.	Freq. Hz	Deg.	Amp. In.	Freq. Hz	Deg.	Amp. In.	Freq. Hz	Deg.	Amp. In.	Freq. Hz	Deg.
38							.025	14.6	0°			
39							.050	14.6	0°			
40							.100	14.6	0°			
41							.025	21.9	0°			
42							.050	21.9	0°			
43							.075	21.9	0°			
44							.025	29.2	0°			
45							.050	29.2	0°			
46										.025	14.6	0°
47										.050	14.6	0°
48										.100	14.6	0°
49										.025	21.9	0°
50										.050	21.9	0°
51										.075	21.9	0°
52										.025	29.2	0°
53										.050	29.2	0°
54	.075	7.3	0°				.075	7.3	0°			
55	.075	7.3	0°				.075	7.3	90°			
56	.075	7.3	0°				.075	7.3	180°			
57	.075	7.3	0°				.075	7.3	270°			
58	.075	7.3	0°				.075	14.6	0°			
59	.075	7.3	0°				.075	14.6	90°			
60	.075	7.3	0°				.075	21.9	0°			
61	.075	7.3	90°				.075	21.9	0°			
62	.075	7.3	0°				.075	21.9	180°			
63	.075	7.3	0°									
64	.075	7.3	90°									
65	.075	7.3	180°									
66	.025	7.3	0°	.025	7.3	180°						
67	.050	7.3	0°	.050	7.3	180°						
68	.075	7.3	0°	.075	7.3	180°						
69	.100	7.3	0°	.100	7.3	180°						
70	.025	7.3	0°	.025	7.3	0°						
71	.050	7.3	0°	.050	7.3	0°						
72	.075	7.3	0°	.075	7.3	0°						
73							.025	7.3	0°	.025	7.3	0°
74							.050	7.3	0°	.050	7.3	0°
75							.075	7.3	0°	.075	7.3	0°
76							.025	7.3	0°	.025	7.3	180°
77							.050	7.3	0°	.050	7.3	180°
78							.075	7.3	0°	.075	7.3	180°
79							.100	7.3	0°	.100	7.3	180°

TABLE A-1 (Continued)

Test	WHEEL LOCATION											
	Left Front			Right Front			Left Rear			Right Rear		
	Amp. In.	Freq. Hz	Deg.	Amp. In.	Freq. Hz	Deg.	Amp. In.	Freq. Hz	Deg.	Amp. In.	Freq. Hz	Deg.
80	.050	6.6	0°									
81	.050	5.97	0°									
82	.050	5.31	0°									
83	.050	4.64	0°									
84	.050	3.98	0°									
85	.050	3.32	0°									
86							.050	6.6	0°			
87							.050	5.97	0°			
88							.050	5.31	0°			
89							.050	4.64	0°			
90							.050	3.98	0°			
91							.050	3.32	0°			
92	.075	7.3	0°									
93	.125	7.3	0°									
94				.100	7.3	0°						
95							.100	7.3	0°			
96										.100	7.3	0°
97	.075	7.3	0°				.075	7.3	0°			
98	.075	7.3	0°				.075	14.6	0°			
99	.075	7.3	0°	.075	7.3	0°						
100												

-Tests 63-65 - Effects of first harmonic phase relationship to the road input

-Tests 66-69 - Effects of first harmonic roll (out of phase) input on the front axle

-Tests 70-72 - Effects of combined (in phase) first harmonic input on both wheels of the front axle

-Tests 73-75 - Effects of combined (in phase) first harmonic input on both wheels of the rear axle

-Tests 76-79 - Effects of first harmonic roll (out of phase) input on the rear axle

Tests 80 through 91 looked at the effect of first harmonic frequency on the front and rear axles individually. The frequency range goes from 7.3 Hz, representing a 55-mph road speed, down to 3.3 Hz, the equivalent of a 25-mph road speed. That lower limit was selected to ensure that first harmonic sensitivity was determined at the tractor's bounce/pitch resonant mode. During this frequency sweep, the road input was held constant at the 55-mph equivalent condition.

The remaining tests, 91-100, were a selected series of repeats (the same repeats were used with all raters) to provide data from which to quantify test-to-test variance.

The sequence was randomized into a separate test order for each rater. Ten raters were selected to comprise the subjective rating jury. The members were as follows:

-Terry Baughn - International Harvester

-Marty Ekonen - Ford Motor Company

-Tom Gillespie - HSRI, University of Michigan

-Ed Leiss - General Motors Corporation

-Nick Mehta - International Harvester

-Steve Mittelstadt - General Motors Corporation

-Trevor Norsworthy - Kenworth

-Gary Rossow - MVMA

-Don Schumaker - Mack Truck

-Don Stephens - PACCAR

Each rater was scheduled to be present at the IHC Engineering Center for a one-half-day period. Prior to test, the rater was given instruction on the test procedures and the rating process. The rater was seated in the truck-driver seat position with instruction to take a posture as if driving the vehicle. The simulator was started, exposing the vehicle to the desired test condition for approximately one minute of duration. Start and stop transients were softened by ramping the gain over a two-second period at the beginning and end of test. When the test ended, the rater was given approximately 30 seconds (or more time, if requested) to record his ratings of the ride conditions, during which time the experimenter reset the control hardware for the next test condition. When both the experimenter and rater were ready, the next test was begun. With this procedure, the full 100-test sequence could be completed in a period of about three hours.

The rater was asked to provide a ride rating value for each of five rating factors:

-Seat vertical vibration

-Seat longitudinal vibration

-Seat lateral vibration

-Steering-wheel vibration

-Cab shake vibration

The rating was made on a scale reflecting the "acceptability of that vibration as a full-time occupational exposure to you as a truck driver." The acceptability scale is shown in Figure 4 of the main report. The familiar 0-10 scale is used with verbal tags chosen to present a nominally linear scale balanced around a midpoint representing the boundary between acceptable and unacceptable.

Analysis of Differences Among Raters

Substantial differences in the subjective ride ratings were observed among the ten raters. The seat vertical rating of the null condition received ratings ranging from 5.5 to 8.5, while the cab shake rating at the null condition ranged from 5.5 to 9.0. Null condition ratings are illustrated in Figure A-1. Raters also differed somewhat in the spread, or standard deviation, of their ratings, as illustrated in Figure A-2. Some raters' assessments span a larger range than others.

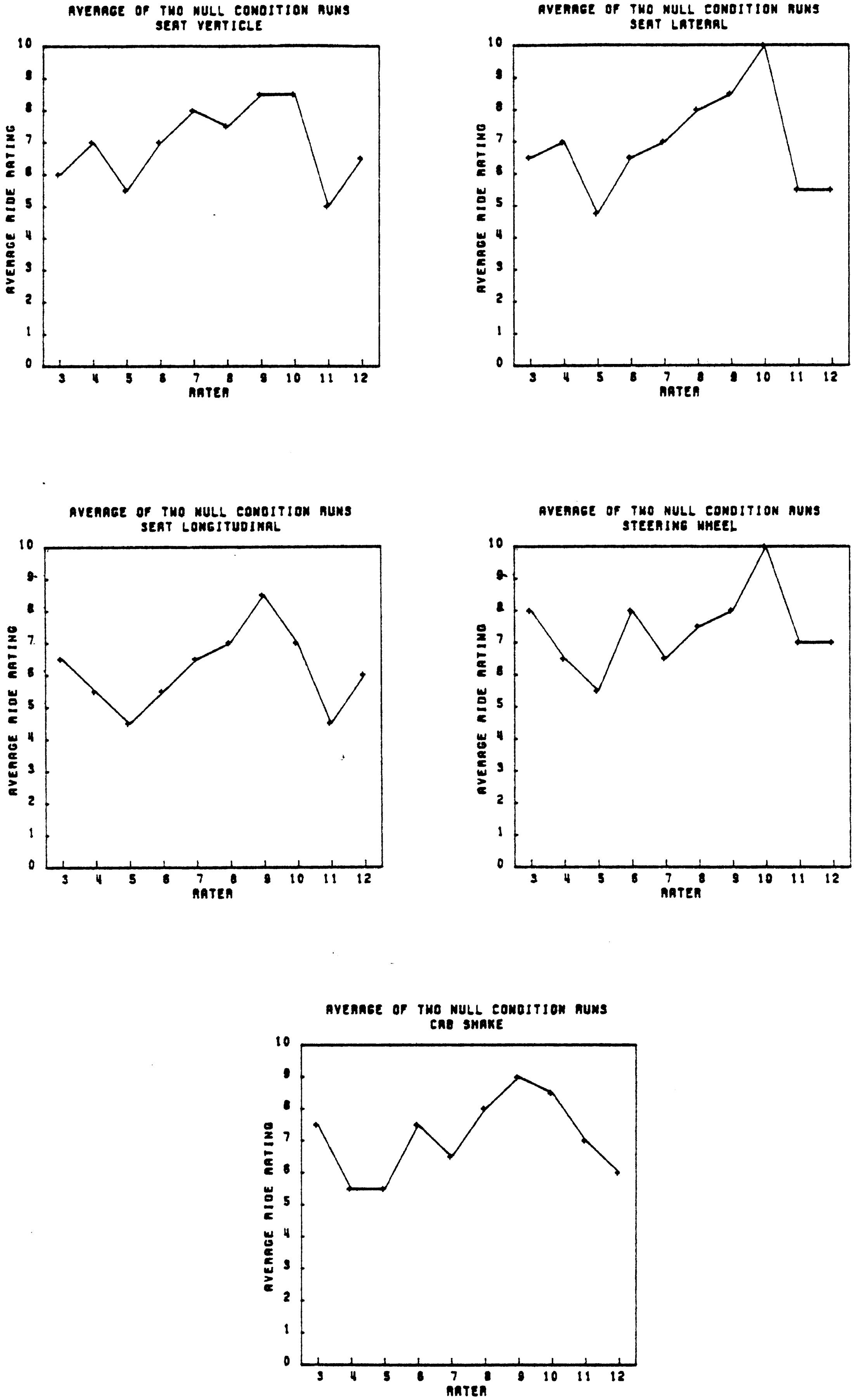


Figure A-1. Comparison of null condition ratings by the ten raters.

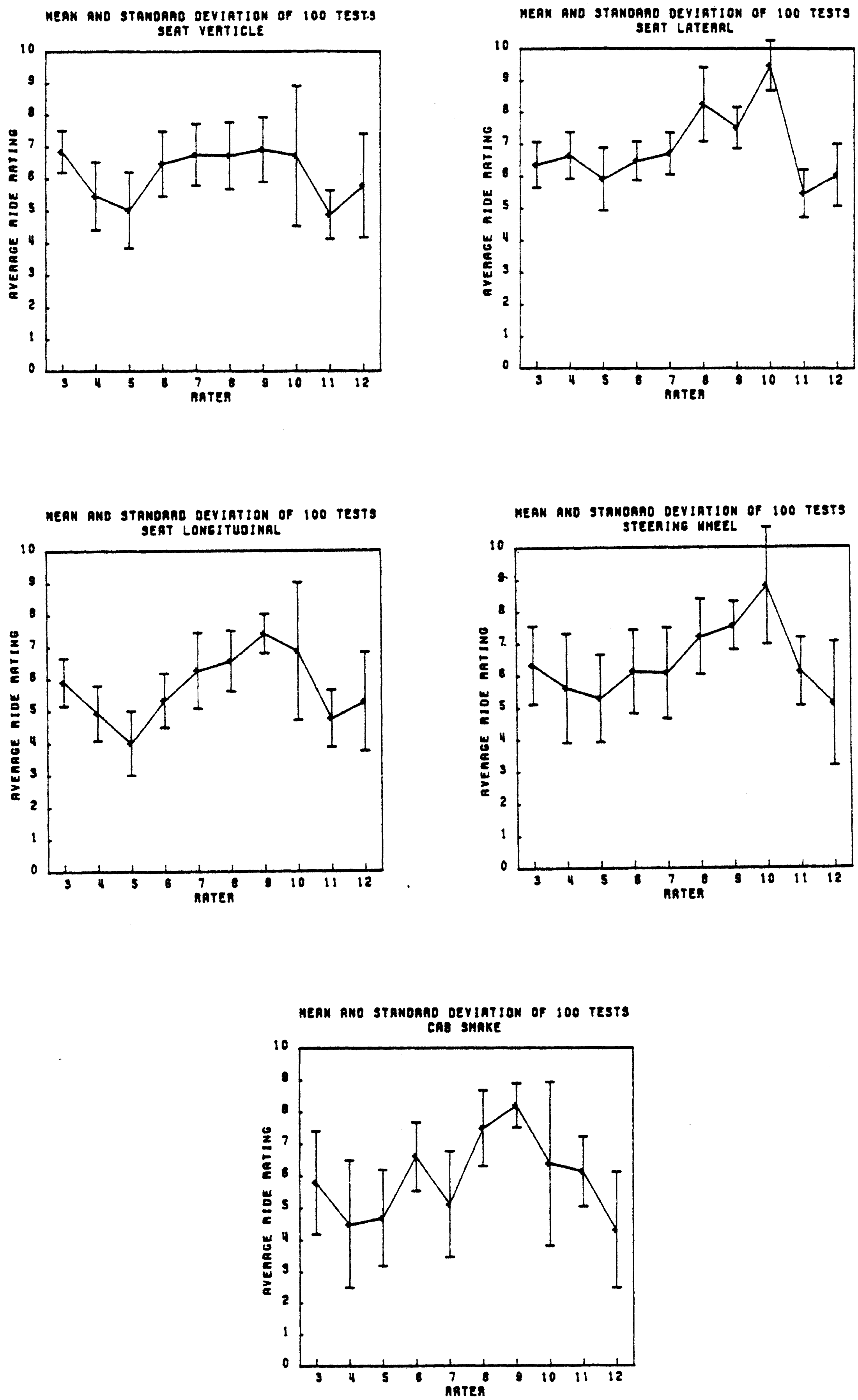


Figure A-2. Comparisons of the means and standard deviations for the ten raters.

The issue of combining the ratings of the ten individuals was given some consideration. The most rigorous approach would be to carry out the entire analysis separately for each rater, and only combine across raters if the trends were sufficiently similar. Following this approach through to conclusion would have been tedious and produced an unmanageable volume of paper. However, selected series of tests were examined separately for each rater. Although the absolute value of the ratings did differ by the magnitudes that are reflected in the differences at the null condition (Fig. A-1), the general trends were exhibited in each rater's scores. Raters did differ in the magnitude of the decrement in subjective ride rating assessed for particular test conditions, some decreasing the rating by as much as two points and others by a half point or less.

Alternative methods of combining scores across raters were considered. Various methods of adjusting the ratings were tried before being rejected. Among these was a full adjustment in which each rater's scores were scaled so as to have the same overall mean and the same standard deviation for the full set of 100 test conditions. Each of the five responses (seat vertical, seat longitudinal, seat lateral, steering wheel, and cab shake) were scaled separately. This adjustment was successful in the sense that confidence intervals were 30 percent to 50 percent smaller. Unadjusted 95-percent confidence intervals for the mean of ten raters typically are plus and minus one point on the subjective ride scale, while the confidence intervals of the adjusted ratings were typically plus and minus one-half to two-thirds of a point.

Results presented in this report are based on unadjusted ratings. Adjustments to minimize rater differences were not used for several reasons. First, the reduced confidence intervals of the adjusted ratings do not dramatically increase the number of comparisons that are statistically significant. Second, detection of statistically significant differences is not a primary objective in an exploratory analysis such as this. Third, rater differences are felt to be an important result of this exploratory analysis that should be included or reflected in the presentation of the data. Consequently, the results presented are an average of ratings recorded by the ten subjects. Ninety-five percent confidence intervals for the mean were also computed for each test condition. The confidence interval reflects the observed variation in the recorded ratings. If the experiment (subjecting ten raters to 100 test conditions) were repeated over and over, one would expect the "true" average to be within the computed interval 19 of every 20 times the experiment was repeated. Selection of a different group of raters is likely to further alter the results.

The complete set of test results (five rating factors for 100 test conditions) representing the unadjusted average over ten raters is presented in Table A-2.

Sources of Variation

The objective of any experimental design is to help ensure that the inferences made are accurate. If ratings are observed to decrease with increasing amplitude, one would like to infer that the amplitude of the vibratory input is the cause of the reduction and not the noise level in the cab or the fatigue of the subject. The possible sources of variation considered in the design of this experiment are listed here.

1. The ability of the human to perceive vibration.
2. The influence of preceding vibratory inputs.
3. Time trends:
   a. Learning by the subject
   b. Fatigue of the subject
4. The influence of uncontrolled factors.
5. Differences in rater judgment.

Each of these will be briefly discussed.

Some variation would be expected as a reflection of the human subject's ability to discriminate one level of vibration from another. This variation is inherent to the use of subjective ratings. However, the subjective ratings may also be influenced by other factors that the experimenter would like to eliminate or minimize. Ratings may be influenced by the noise level or temperature in the cab since they contribute to the general comfort of the subject, or the adjusted position of the seat. Previous vibratory inputs may influence following assessments. A moderate level of vibration may be

Table A-2

Subjective Ride Ratings
(Average of Ten Raters)

Test Cond.	Seat Vert.	Seat Long.	Seat Lat.	Steering Wheel	Cab Shake
1.	6.80	6.00	6.65	7.60	7.30
2.	7.00	5.95	7.30	7.20	7.17
3.	6.00	5.70	6.90	7.30	6.75
4.	6.00	5.90	6.70	7.15	6.95
5.	6.45	6.40	6.90	7.20	6.70
6.	6.60	6.15	6.65	7.10	6.45
7.	6.55	6.05	6.55	7.00	6.80
8.	6.60	6.35	6.80	7.65	7.45
9.	6.75	5.80	6.70	7.60	7.00
10.	6.20	5.60	6.40	7.50	6.61
11.	6.20	5.65	6.40	6.90	6.25
12.	6.35	5.60	6.90	7.60	7.30
13.	5.70	5.30	7.20	6.80	6.65
14.	5.50	5.25	7.15	6.35	5.70
15.	5.80	4.60	7.45	5.85	5.40
16.	5.85	3.30	6.75	5.00	4.10
17.	6.35	6.00	7.30	7.25	6.60
18.	6.75	5.85	6.80	7.30	7.25
19.	6.30	5.10	6.80	5.90	5.60
20.	6.35	4.70	7.00	5.40	5.30
21.	6.20	3.85	7.05	5.30	5.35
22.	6.80	6.00	6.80	6.30	5.50
23.	6.25	5.75	6.65	6.00	5.05
24.	5.80	5.80	6.65	5.20	4.00
25.	6.50	6.00	6.75	6.55	6.00
26.	6.15	5.90	6.65	6.90	5.70
27.	6.85	6.25	6.95	6.30	4.90
28.	6.55	6.00	6.90	7.20	6.90
29.	7.10	6.45	7.30	6.05	6.35
30.	6.50	5.95	6.80	5.65	5.15
31.	6.20	6.00	6.95	4.95	4.35
32.	6.00	6.00	7.05	4.65	3.75
33.	6.55	6.25	7.35	6.65	5.72
34.	6.00	6.35	6.70	5.90	4.70
35.	6.75	6.60	7.35	6.30	3.70
36.	6.60	6.50	7.05	6.75	6.15
37.	6.50	6.35	7.00	6.10	5.45
38.	5.90	6.25	6.50	6.20	5.80
39.	5.35	6.15	6.70	5.85	4.50
40.	4.30	5.50	7.25	3.35	3.10
41.	6.10	5.75	7.05	5.10	4.85
42.	6.50	6.40	7.10	4.70	4.30
43.	5.80	6.35	7.05	4.50	4.55
44.	6.50	6.15	6.85	7.20	7.05
45.	6.50	6.50	6.75	6.85	6.40
46.	6.10	6.20	7.45	6.30	6.10
47.	5.60	6.15	6.75	5.10	4.44
48.	4.80	5.55	6.95	4.80	3.05
49.	6.65	6.40	6.75	6.00	5.15
50.	6.20	6.40	7.25	5.10	4.22

Table A-2 (Cont.)

Subjective Ride Ratings
(Average of Ten Raters)

Test Cond.	Seat Vert.	Seat Long.	Seat Lat.	Steering Wheel	Cab Shake
51.	6.60	6.50	6.85	5.20	4.30
52.	6.95	6.10	6.70	7.10	6.70
53.	6.85	6.60	7.20	5.90	6.25
54.	6.05	4.20	6.90	5.85	5.05
55.	5.60	4.85	7.30	5.95	5.30
56.	4.45	5.05	6.85	5.75	5.85
57.	5.75	4.95	6.55	5.90	5.85
58.	3.95	5.10	6.80	4.60	4.55
59.	4.10	4.95	6.50	4.75	4.10
60.	6.45	6.40	7.05	5.20	4.45
61.	5.35	6.10	6.80	4.15	4.45
62.	5.75	6.30	6.95	4.30	4.00
63.	6.60	5.85	7.30	7.50	7.11
64.	6.45	5.50	6.85	7.05	6.78
65.	6.05	6.05	6.80	7.35	6.65
66.	6.70	6.25	7.10	7.45	7.15
67.	6.80	6.40	6.80	7.30	6.95
68.	6.15	6.05	6.20	6.85	6.50
69.	6.15	5.90	4.85	6.75	5.90
70.	6.95	5.85	7.10	7.40	7.00
71.	6.45	6.40	7.10	7.30	7.00
72.	5.30	5.55	6.85	7.25	6.67
73.	6.80	5.15	6.90	6.70	6.50
74.	6.15	4.30	7.40	6.25	5.85
75.	5.75	3.10	6.95	4.40	4.30
76.	6.35	5.90	6.95	7.35	6.85
77.	6.25	5.70	6.70	7.25	7.15
78.	6.45	6.10	6.75	7.25	6.85
79.	7.15	6.45	7.30	7.20	6.50
80.	6.90	5.80	6.90	7.40	7.10
81.	6.60	5.80	6.70	7.40	7.45
82.	6.90	6.10	7.20	7.50	7.15
83.	6.55	6.10	6.85	7.80	7.15
84.	6.20	5.80	7.20	7.80	7.20
85.	6.00	5.65	6.95	7.40	7.30
86.	6.05	5.40	7.05	7.10	7.10
87.	6.45	6.00	6.80	7.40	7.20
88.	6.25	5.70	6.70	7.70	6.50
89.	6.15	6.15	6.40	7.30	6.65
90.	5.55	4.95	7.40	7.35	6.90
91.	5.70	4.38	7.15	7.80	7.30
92.	6.50	6.10	6.80	7.20	7.00
93.	5.45	6.05	6.25	7.60	6.50
94.	6.35	6.20	6.55	7.80	7.25
95.	5.75	4.00	6.95	5.25	4.60
96.	6.05	4.35	6.80	5.70	5.40
97.	6.20	4.10	7.10	5.50	5.30
98.	4.20	5.45	6.95	4.80	3.90
99.	4.95	5.75	6.40	7.10	6.30
100.	7.10	6.30	7.20	7.20	6.90

perceived differently if it follows several tests at a minimal level than it will be if it follows a severe test. Also, a rater's judgment may change over the course of the experiment as he becomes more experienced in discriminating vibratory inputs. In the later stages, fatigue may influence the ratings. Both of these influences may be generally characterized as time trends. Finally, raters may simply differ in their judgments of ride quality. The scale was deliberately designed to be an absolute scale in the sense that descriptors such as "acceptable/unacceptable" were attached to specific levels of the scale. Differences in the raters' assessments of these absolute characteristics will be reflected in the rating. Differences in age, height, and weight may also influence the ratings.

Each was asked to rate the same 100 test conditions. Unwanted sources of variation were controlled as much as possible. The experimental design required that the runs be conducted in random order for each subject. The objective of the randomization is to try to prevent any remaining uncontrolled sources of variation from being associated in a systematic manner with any of the dependent variables. As a result, possible systematic errors become random errors, and the possibility of false conclusions is reduced. For example, if fatigue influences the ratings, and if the same test conditions were run last, the influence of fatigue is likely to be falsely associated with the last few test conditions.

Rater bias is not likely to influence the general trends observed, since each rater was exposed to the same 100 test conditions (but not in the same order). Rater bias does, however, inflate the observed error. The replicate runs included in the experimental design were used to examine the contribution of rater bias to the observed error. An analysis of variance indicated that the variance associated with rater bias was greater than the variance associated with replicate runs for an individual rater in nearly all of the conditions examined (several different test conditions and all five responses). F ratios (the ratio of the variance associated with rater bias to the variance associated with replicate runs for an individual rater) ranged from 3 to 8 with a typical value of 5 or 6. These ratios were all highly statistically significant for the given sample sizes.* As discussed earlier, the differences among raters were examined and found to be responsible for perhaps 30 to 50 percent of the observed error (i.e., width of the confidence interval).

In general, the relative contributions of the remaining sources of variation cannot be assessed. The data were viewed in the sequence in which the tests were actually conducted, and no evidence of time trends was observed. Other sources of error contributing to the variability associated with replicate runs, are errors associated with the human's ability to perceive vibration, the influence of preceding tests, and any other uncontrolled factors. It is worthwhile to note that the combination of these factors was found to make a much smaller contribution to the observed variance than the rater bias. Because the unadjusted ratings were used, the confidence intervals reflect the influence of all of the sources of variation discussed, including differences among raters.

As a final note, the replicate runs were simply combined for presentation as if there were 20 raters instead of ten. Accordingly, the confidence intervals for these test conditions are somewhat smaller. Table A-3 lists the 95 percent confidence range for the results listed in Table A-2. The 95 percent confidence interval for any result is obtained by taking the average value from Table A-2, plus and minus the range shown in Table A-3.

Comparisons of Subjective Ratings with Accelerometer-Derived Vibration Measures

Raters sat on a seat pad containing accelerometers that were oriented in the vertical and longitudinal (fore/aft) directions. Several measures of ride vibration were derived from the resulting accelerometer data, as listed below.

1. The percent of time the weighted histogram was within ± 0.125 g.

*The analysis of variance assumes the variables are measured on an interval scale. This is not necessarily the case for subjective ratings.

TABLE A-3
Twice the Standard Error for Average Ride Ratings
(Average of Ten Raters)

Test Cond.	Seat Vert.	Seat Long.	Seat Lat.	Steering Wheel	Cab Shake
1.	.7775	.7889	1.0756	.7424	.7917
2.	.7150	1.1781	.7334	.4989	.7417
3.	.9429	.7334	.8667	.8970	1.0248
4.	.7303	1.0089	.9911	.7609	.6404
5.	.5260	.8538	.8138	.7775	.9911
6.	.8001	.8699	.9895	.8667	.8750
7.	.4819	.7811	.8227	.7303	1.1471
8.	.8407	1.0960	1.0242	.8951	.9244
9.	.6192	.7775	.8845	.8001	.7747
10.	1.0242	.9044	.8001	.8028	.9973
11.	.7181	.7311	.6634	.9166	.9340
12.	.9667	.7424	.8667	.8001	.8056
13.	1.0023	.7334	.5812	.8327	1.0117
14.	.8564	.5822	.6334	.9895	1.0350
15.	1.0133	.7273	.9939	.9196	1.1625
16.	1.3829	1.0350	1.0462	1.5777	1.5407
17.	.7609	1.0750	1.1176	.7188	.8001
18.	.7188	.9196	.7775	.6000	.4014
19.	.4269	.8138	.9334	.9166	1.0521
20.	.7000	1.1076	.9889	1.2720	1.1945
21.	.4989	1.0334	.6047	1.2038	1.0756
22.	.4000	.6667	.6532	1.0023	1.0125
23.	.5000	.8334	1.0960	1.1156	1.3035
24.	.5812	.6532	.8172	.9799	1.3251
25.	.6147	.9310	.9099	1.0797	1.0329
26.	1.1553	.8001	.8699	.7572	1.3014
27.	.8951	.8199	.9244	.7334	.9978
28.	.7372	.8433	.9639	.7775	.8667
29.	.5334	.9244	1.0350	.9244	1.2298
30.	.6832	.7372	.8327	.6675	1.0960
31.	.9911	.9310	1.1398	1.1001	1.4147
32.	.8433	.9068	1.1201	1.1161	1.4241
33.	.9244	1.0673	.7000	.8699	1.1950
34.	.7889	.7896	.8460	1.2093	1.1945
35.	.7188	.8001	.8699	.6700	1.2580
36.	.5121	.9662	.9244	.9340	1.0756
37.	.8433	1.0756	.9889	.9753	.9001
38.	1.1334	.9340	.9546	.8718	1.0133
39.	1.2298	.9667	1.3014	1.3504	1.5493
40.	1.1471	1.1255	.8851	1.1743	1.6181
41.	.6290	.7782	.9482	.7718	1.1553
42.	.7454	.9044	1.0935	.8460	1.5507
43.	.8845	.7609	.8750	.8166	1.0589
44.	.6832	.8951	.8440	.7775	.7064
45.	.7454	.7454	1.3925	1.2829	1.4361
46.	1.0521	.7631	.9001	1.0350	1.2093
47.	.9166	.8172	.9340	1.0414	1.3825
48.	1.4159	1.4334	.5667	1.3921	1.6631
49.	.7609	.8538	.9099	1.1833	1.4611
50.	.6532	.9044	1.0248	1.2455	1.5421

TABLE A-3, cont.
Twice the Standard Error for Average Ride Ratings
(Average of Ten Raters)

Test Cond.	Seat Vert.	Seat Long.	Seat Lat.	Steering Wheel	Cab Shake
51.	.9639	.8945	1.0334	.8970	1.4315
52.	.5260	.7718	.6000	.6961	.8970
53.	.6334	.9044	1.1852	.5538	1.0028
54.	.8492	.9452	.6290	1.3504	1.2689
55.	.9405	.9667	.9452	1.7157	1.4923
56.	.7064	1.0377	.9895	1.2584	1.3504
57.	.9575	1.1201	.9482	1.2455	1.3989
58.	.9244	.8667	.6532	.6799	1.2334
59.	1.2455	.9001	.8433	.9804	1.5622
60.	.8492	.6111	.8492	.9799	1.2153
61.	1.2478	.7572	.7775	.7000	1.2689
62.	.8334	.7917	1.0589	.7334	1.3417
63.	.8138	.7609	1.1176	.9546	.9719
64.	.8492	.8564	.5973	.4334	.6913
65.	.6404	.8750	.9334	.7609	.6334
66.	.9911	.8334	1.0089	.7064	.7896
67.	.6532	.8001	.9334	.8970	.7064
68.	.6675	.9001	.9334	.7896	.6147
69.	.6675	.8138	.3958	.9099	1.1720
70.	.7952	.8440	.6961	.8407	.9310
71.	.9939	1.1625	1.2455	1.0771	1.0329
72.	1.0350	1.0589	.7311	.5427	1.0489
73.	.7775	.9196	.8138	1.0350	.9068
74.	1.3001	.9799	.8001	1.1667	1.0547
75.	1.3438	1.1720	.7064	1.6385	1.7141
76.	.8699	.7718	.9244	.7896	1.0334
77.	.8851	.6864	.7334	.8596	.6675
78.	.8750	.9978	.9575	.9340	.7896
79.	.8440	.9244	.8970	.6532	.9068
80.	.6290	.8193	.8667	.8001	.5538
81.	.5121	.7631	.8590	.6799	.8227
82.	.7572	.9753	.7775	.8028	.6675
83.	.9244	.7572	.7896	.5812	.7896
84.	.9799	1.0667	1.1471	.7775	.7775
85.	.7454	.8951	.9244	.7424	.7334
86.	.8227	.9166	1.0161	1.0521	.8667
87.	.5860	.7454	.9799	.8001	.7775
88.	.8063	.9911	.9911	.6700	.6147
89.	.8172	.7311	1.2001	.7334	.7311
90.	.9482	.8227	.8538	.8699	1.0521
91.	.9911	1.0198	.7000	.6532	.6700
92.	.8166	.8795	.9334	.8845	.8433
93.	.8227	.7064	.7782	.7424	.9546
94.	.5973	.6532	.9001	.7181	1.0879
95.	.7188	.8564	.9244	1.2226	1.5479
96.	.7064	.9895	.8845	1.2668	1.3318
97.	.5207	.8274	.7572	1.3664	1.3921
98.	1.2580	1.4942	1.1398	1.1076	1.5334
99.	.9244	.8334	.8001	.5538	1.0350
100.	.8667	1.0771	.9334	.9799	.8667

2. The percent of time the unweighted histogram was within ±0.125 g.

3. The weighted RMS acceleration in g's.

4. The unweighted RMS acceleration in g's.

5. The absorbed power in watts.

These measures were computed for both the vertical and longitudinal directions. The raw data are shown in Table A-4. In addition, the total absorbed power (vertical and longitudinal combined) and the ISO exposure time in the 4 Hz band (ISO Standard 2631-1978(E), "Guide for the Evaluation of Human Exposure to Whole-Body Vibration," International Standards Organization, 1978) were computed. Accelerometer measurements were recorded for about 10 of the 100 test conditions for each rater, for a total of 104 tests. Different tests were selected for each rater so that accelerometer measurements were obtained for 52 of the 100 test conditions.

Only a brief examination of the accelerometer measurements has been made. In general, the accelerometer measurements were only weakly associated with the subjective ratings. Scatter plots of the 12 accelerometer measurements versus the appropriate subjective rating are presented in Figures A-3 through A-5. Correlation coefficients for the same pairs of variables are shown in Table A-5. Most of the correlation coefficients are not significantly different from zero for the given sample size. Correlation coefficients are somewhat sensitive to "outliers," and assume the variables are each coded on an interval scale.

Table A-4

Seat Pad Accelerometer Measurements in the Vertical Direction

Rater	Test Cond.	Wtd.%	Unwtd.%	Wtd. RMS	Unwtd. RMS	Absorbed Power
4.	1.	53.30	53.56	.07405	.08775	.9669
10.	1.	62.35	54.95	.06828	.08030	.8272
3.	2.	60.47	52.85	.06851	.08275	.8218
5.	2.	55.21	49.96	.07657	.08981	1.0410
4.	4.	61.95	56.39	.07305	.08422	.7925
9.	4.	63.15	57.49	.06811	.08107	.6051
7.	5.	58.88	52.99	.07540	.08853	.8266
8.	5.	55.92	57.06	.07221	.08311	.5958
10.	5.	56.34	52.94	.07507	.08463	.7679
11.	5.	59.49	54.01	.07322	.08421	.7913
5.	6.	56.31	52.26	.07567	.08603	.7440
8.	6.	47.41	44.71	.08835	.10080	.8973
11.	8.	56.23	51.30	.07252	.08441	.9053
5.	9.	55.94	51.50	.07703	.08947	.9685
8.	9.	61.69	55.92	.07582	.09099	.8173
9.	10.	60.80	55.80	.06992	.08085	.6476
10.	10.	54.59	50.40	.08088	.09076	.9551
3.	11.	54.56	51.69	.07860	.09025	.8616
9.	11.	52.13	48.88	.08199	.09348	.8238
12.	11.	50.74	47.25	.08454	.09592	.9940
5.	12.	53.90	48.87	.07910	.09211	1.1040
7.	13.	54.25	49.11	.08129	.09467	1.0650
11.	13.	52.46	47.88	.08265	.09456	1.1150
3.	16.	50.61	42.56	.08816	.10580	1.0880
11.	16.	44.89	40.82	.10010	.11320	1.3660
12.	16.	42.05	38.70	.10430	.11730	1.3940
7.	17.	55.00	51.03	.07953	.09341	1.1270
9.	17.	71.83	62.10	.05787	.07158	.5237
9.	18.	67.25	59.25	.06371	.07621	.6150
7.	19.	58.04	52.87	.07574	.08911	1.0030
12.	19.	61.00	54.57	.07410	.08703	.8744
5.	20.	58.59	52.41	.07876	.09345	.9395
6.	20.	56.86	49.88	.07904	.09367	.9222
8.	20.	46.55	39.95	.09435	.10830	1.0860
12.	20.	54.12	50.33	.08288	.09478	.9322
4.	21.	48.57	44.49	.09112	.10450	1.1100
9.	21.	40.70	38.37	.09653	.11030	.9706
12.	21.	56.08	49.97	.08330	.09684	1.0080
3.	24.	71.60	65.58	.05681	.07238	.5453
5.	24.	70.55	58.51	.05742	.07234	.5615
6.	24.	72.49	61.42	.05990	.07587	.6087
9.	24.	82.30	61.75	.05032	.06917	.3827
10.	24.	77.72	65.44	.05379	.06909	.4886
4.	26.	63.68	55.90	.06326	.07481	.7548
5.	26.	55.51	48.70	.07309	.09070	.9415
8.	27.	72.83	43.96	.06217	.09871	.5858
11.	27.	66.65	59.02	.06285	.07880	.6705
12.	27.	69.24	58.38	.06408	.07993	.7084
5.	28.	50.19	50.98	.07252	.08307	.9699
3.	30.	62.10	54.92	.07080	.08469	.8623
7.	30.	65.79	54.54	.06615	.07948	.7580
8.	30.	74.01	59.21	.06367	.08047	.6282

Table A-4 (Cont.)

Seat Pad Accelerometer Measurements in the Vertical Direction

Rater	Test Cond.	Wtd.%	Unwtd.%	Wtd. RMS	Unwtd. RMS	Absorbed Power
11.	32.	59.29	41.62	.07022	.09950	.7586
12.	32.	67.03	46.67	.06543	.09355	.6327
3.	34.	58.08	52.86	.06958	.08470	.8585
7.	36.	56.71	53.21	.07495	.08909	.9988
3.	37.	64.48	57.21	.07007	.08401	.8525
8.	40.	42.72	23.82	.08927	.15360	.9267
9.	40.	58.73	35.63	.06995	.11320	.6280
8.	41.	71.00	58.34	.06622	.08508	.6969
10.	43.	51.51	43.76	.07568	.09973	1.0370
4.	49.	65.09	56.00	.07108	.08524	.8903
5.	49.	56.50	50.86	.07604	.09135	1.0430
8.	51.	76.67	63.68	.05603	.07304	.4925
11.	51.	57.88	51.59	.07172	.08859	.9308
12.	51.	64.11	50.76	.07296	.09549	.9325
4.	52.	54.57	49.38	.07514	.08759	1.0260
11.	52.	54.04	50.03	.07673	.08934	1.0610
3.	53.	57.39	51.59	.07554	.09059	1.0040
6.	53.	58.13	52.32	.07468	.09002	.9858
6.	54.	62.90	53.88	.07212	.08860	.8840
4.	62.	59.81	53.51	.07758	.09195	.9082
10.	63.	54.52	50.47	.07859	.08856	.9235
11.	63.	59.29	54.37	.07388	.08426	.8686
8.	64.	65.88	60.16	.07070	.08133	.6733
5.	65.	57.41	52.14	.07691	.08871	.9469
10.	65.	56.82	52.77	.07795	.08821	.9442
4.	67.	58.40	57.74	.07071	.08378	.8824
4.	76.	61.98	55.04	.07271	.08592	.9288
5.	80.	56.57	48.77	.07659	.08878	1.0133
7.	80.	59.12	49.37	.07492	.08731	.9492
9.	80.	64.32	58.60	.06723	.07711	.7057
3.	81.	62.96	56.08	.07022	.08394	.8565
6.	81.	64.86	58.26	.06602	.07897	.7641
6.	82.	58.43	50.91	.07285	.08537	.9479
3.	84.	47.37	49.02	.07417	.08304	1.0030
6.	88.	57.29	51.24	.07604	.09104	1.0350
7.	89.	52.49	50.79	.07736	.09018	1.0780
6.	90.	42.59	44.64	.08683	.09852	1.3710
10.	90.	50.82	46.94	.08581	.09654	1.3430
3.	93.	68.54	61.55	.06353	.07554	.5674
6.	93.	55.51	49.94	.08051	.09381	.9041
7.	93.	52.60	47.09	.08710	.09953	1.1160
9.	93.	59.71	53.79	.06988	.08279	.5810
10.	93.	49.68	47.65	.08484	.09345	.9292
11.	93.	57.43	53.36	.07513	.08598	.7973
12.	93.	57.77	56.65	.06983	.08047	.5854
4.	94.	54.05	50.77	.07946	.08970	.8618
6.	95.	48.50	43.27	.09085	.10480	1.1240
12.	97.	56.53	50.94	.07840	.09162	1.0000
4.	100.	64.69	57.20	.07491	.08820	.9764
7.	100.	60.02	53.81	.07669	.09089	1.0280
11.	100.	59.20	52.76	.07319	.08633	.9480
12.	100.	59.75	52.90	.07572	.08854	1.0250

Table A-4 (Cont.)

Seat Pad Accelerometer Measurements in the Longitudinal Direction

Rater	Test Cond.	Wtd.%	Unwtd.%	Wtd. RMS	Unwtd. RMS	Absorbed Power
4.	1.	93.36	63.36	.03748	.10170	.6535
10.	1.	95.95	66.59	.02992	.09693	.4098
3.	2.	-0.	-0.	-0.	-0.	-0.
5.	2.	95.44	68.38	.03084	.08494	.4286
4.	4.	90.58	57.02	.04007	.10820	.7058
9.	4.	92.86	60.70	.03703	.11180	.6137
7.	5.	93.15	60.24	.03701	.10340	.6067
8.	5.	95.82	59.87	.02892	.09503	.3705
10.	5.	94.51	63.83	.03641	.09569	.5658
11.	5.	94.42	65.75	.03340	.09371	.5127
5.	6.	93.15	64.53	.03648	.09025	.5762
8.	6.	93.40	54.91	.03782	.10000	.6426
11.	8.	94.47	62.81	.03377	.10600	.5236
5.	9.	94.22	71.09	.03586	.08710	.5987
8.	9.	94.21	61.35	.03525	.10060	.5608
9.	10.	94.81	59.83	.03352	.10760	.4738
10.	10.	95.60	65.06	.03214	.09053	.4507
3.	11.	98.22	48.46	.02636	.07443	.3012
9.	11.	94.33	57.56	.03378	.11110	.4895
12.	11.	93.48	53.88	.03630	.11100	.5682
5.	12.	95.04	71.04	.03353	.08207	.4862
7.	13.	94.63	57.90	.03361	.10360	.5078
11.	13.	94.73	59.10	.03317	.10710	.4768
3.	16.	99.46	81.66	.02109	.04808	.2383
11.	16.	90.58	32.49	.04124	.16540	.6051
12.	16.	78.97	25.99	.04961	.19000	.7970
7.	17.	93.14	64.98	.03723	.09567	.6407
9.	17.	93.99	60.06	.03559	.10650	.5566
9.	18.	94.63	53.45	.03212	.11130	.4251
7.	19.	93.88	64.42	.03578	.09478	.5761
12.	19.	93.25	45.47	.03686	.13410	.5442
5.	20.	93.75	48.09	.03896	.11940	.6294
6.	20.	92.91	39.47	.03819	.13930	.5854
8.	20.	92.19	36.02	.04089	.13210	.6857
12.	20.	92.70	37.37	.03842	.15060	.5455
4.	21.	-0.	-0.	-0.	-0.	-0.
9.	21.	87.68	33.07	.04359	.16390	.6885
12.	21.	88.69	29.30	.04312	.16360	.5720
3.	24.	98.67	84.15	.02338	.04871	.3296
5.	24.	93.88	50.73	.03584	.10100	.5333
6.	24.	93.29	38.75	.03514	.13810	.5164
9.	24.	94.62	30.32	.03378	.16180	.4279
10.	24.	-0.	34.63	.03562	.14110	.5117
4.	26.	95.96	45.82	.02879	.10710	.3346
5.	26.	95.00	32.40	.03252	.13240	.4271
8.	27.	96.18	26.84	.02788	.15910	.2763
11.	27.	94.61	58.81	.03377	.10850	.4856
12.	27.	94.87	58.40	.03292	.15780	.4438
5.	28.	98.98	72.33	.02051	.06311	.1756
3.	30.	99.61	88.86	.01875	.04091	.2051
7.	30.	94.27	58.78	.03303	.10230	.4886
8.	30.	96.00	42.79	.02961	.11160	.3746

Table A-4 (Cont.)

Seat Pad Accelerometer Measurements in the Longitudinal Direction

Rater	Test Cond.	Wtd.%	Unwtd.%	Wtd. RMS	Unwtd. RMS	Absorbed Power
11.	32.	94.77	50.56	.03229	.11710	.4630
12.	32.	93.84	38.44	.03463	.14310	.5065
3.	34.	-0.	-0.	-0.	-0.	-0.
7.	36.	94.02	60.60	.03549	.10140	.5861
3.	37.	99.49	84.95	.01928	.04391	.2171
8.	40.	89.59	28.31	.04421	.17140	.8531
9.	40.	95.89	39.17	.02974	.12850	.3493
8.	41.	94.69	41.72	.03399	.11830	.5023
10.	43.	95.61	25.33	.02424	.18580	.4725
4.	49.	-0.	-0.	-0.	-0.	-0.
5.	49.	93.07	48.13	.03789	.10870	.0060
8.	51.	96.85	31.06	.02714	.14730	.2684
11.	51.	95.46	26.84	.03259	.17520	.4026
12.	51.	94.15	21.91	.03524	.19650	.4647
4.	52.	93.42	65.96	.03596	.09050	.5664
11.	52.	95.35	61.78	.03177	.10470	.4611
3.	53.	-0.	-0.	-0.	-0.	-0.
6.	53.	93.53	49.70	.03492	.11590	.5625
6.	54.	90.67	36.31	.04141	.15500	.6746
4.	62.	88.48	23.41	.04305	.19270	.7631
10.	63.	94.31	68.07	.03527	.08815	.5511
11.	63.	94.47	64.19	.03337	.09758	.5214
8.	64.	96.99	59.15	.02700	.09101	.3103
5.	65.	94.76	67.01	.03331	.08841	.4699
10.	65.	94.70	65.14	.03464	.09396	.5367
4.	67.	91.74	59.62	.03841	.10410	.6502
4.	76.	99.17	88.45	.02210	.04063	.2739
5.	80.	95.04	67.30	.03378	.08767	.4995
7.	80.	94.04	60.27	.03359	.10050	.4842
9.	80.	96.45	60.04	.02735	.10090	.3152
3.	81.	99.65	89.80	.02070	.03855	.2400
6.	81.	96.26	66.28	.02782	.09469	.3161
6.	82.	93.77	62.82	.03454	.10650	.5355
3.	84.	-0.	-0.	-0.	-0.	-0.
6.	88.	93.04	61.09	.03553	.11260	.5538
7.	89.	95.39	65.74	.03167	.09074	.4466
6.	90.	92.94	53.97	.03592	.13500	.5290
10.	90.	94.53	55.88	.03423	.13170	.4786
3.	93.	-0.	-0.	-0.	-0.	-0.
6.	93.	91.91	57.71	.03942	.11610	.6938
7.	93.	92.10	58.99	.03846	.10050	.6561
9.	93.	92.81	56.69	.03629	.11840	.5862
10.	93.	94.47	60.39	.03544	.09170	.5440
11.	93.	94.05	60.04	.03321	.10390	.4920
12.	93.	93.62	53.47	.03603	.11690	.5722
4.	94.	-0.	-0.	-0.	-0.	-0.
6.	95.	90.58	32.91	.04219	.15330	.6735
12.	97.	91.90	55.30	.03930	.11450	.6953
4.	100.	91.91	59.59	.04121	.10940	.7429
7.	100.	93.57	65.84	.03512	.10200	.5625
11.	100.	94.32	66.52	.03364	.09921	.5237
12.	100.	93.80	55.36	.03589	.11500	.5843

Table A-4 (Cont.)

Combined Seat Pad Accelerometer Measurements (Vertical and Longitudinal)

Rater	Test Cond.	ISO Time	Combined Power
4.	1.	3.357	1.1670
10.	1.	3.742	.9230
3.	2.	3.781	.8510
5.	2.	3.063	1.1260
4.	4.	4.299	1.0590
9.	4.	5.354	.8610
7.	5.	4.425	1.0250
8.	5.	4.197	.7010
10.	5.	4.620	.5530
11.	5.	4.282	.9420
5.	6.	4.325	.9410
8.	6.	2.832	1.1030
11.	8.	3.512	1.0450
5.	9.	3.382	1.1380
8.	9.	4.925	.9910
9.	10.	5.184	.8020
10.	10.	3.811	1.0600
3.	11.	4.191	.9120
9.	11.	3.546	.9580
12.	11.	3.913	1.1440
5.	12.	2.937	1.2060
7.	13.	3.208	1.1800
11.	13.	2.973	1.2120
3.	16.	3.874	1.1140
11.	16.	2.876	1.4940
12.	16.	2.420	1.6060
7.	17.	2.813	1.2970
9.	17.	6.110	.7640
9.	18.	5.650	.7470
7.	19.	3.282	1.1570
12.	19.	4.136	1.0300
5.	20.	4.222	1.1300
6.	20.	4.191	1.0920
8.	20.	3.032	1.2850
12.	20.	4.173	1.0800
4.	21.	3.621	1.1340
9.	21.	2.082	1.1900
12.	21.	4.146	1.1590
3.	24.	5.280	.6370
5.	24.	4.962	.7740
6.	24.	4.540	.7980
9.	24.	7.104	.5740
10.	24.	5.346	.7070
4.	26.	3.714	.7980
5.	26.	3.312	1.0330
8.	27.	5.854	.6470
11.	27.	4.557	.8270
12.	27.	4.408	.8360
5.	28.	2.942	.9850
3.	30.	3.721	.8860
7.	30.	3.941	.9010
8.	30.	5.535	.7310

Table A-4 (Cont.)

Combined Seat Pad Accelerometer Measurements (Vertical and Longitudinal)

Rater	Test Cond.	ISO Time	Combined Power
11.	32.	4.121	.8880
12.	32.	5.044	.8100
3.	34.	3.497	.8870
7.	36.	3.157	1.1580
3.	37.	3.968	.8790
8.	40.	2.907	1.2590
9.	40.	5.186	.7180
8.	41.	5.131	.8590
10.	43.	2.858	1.1350
4.	49.	3.654	.9230
5.	49.	3.028	1.0430
8.	51.	6.397	.5600
11.	51.	3.407	1.0140
12.	51.	3.606	1.0420
4.	52.	2.953	1.1720
11.	52.	2.887	1.1570
3.	53.	3.412	1.0320
6.	53.	3.140	1.1350
6.	54.	3.503	1.1120
4.	62.	3.971	1.1260
10.	63.	3.813	1.0750
11.	63.	3.732	1.0130
8.	64.	5.653	.7410
5.	65.	3.581	1.0570
10.	65.	3.471	1.0860
4.	67.	3.532	1.0960
4.	76.	3.451	.9680
5.	80.	3.245	1.1300
7.	80.	3.591	1.0650
9.	80.	5.328	.7730
3.	81.	3.943	.8890
6.	81.	3.956	.8260
6.	82.	3.610	1.0880
3.	84.	2.948	1.0190
6.	88.	3.063	1.1730
7.	89.	3.059	1.1670
6.	90.	2.216	1.4700
10.	90.	2.238	1.4260
3.	93.	6.056	.6380
6.	93.	4.059	1.1390
7.	93.	3.328	1.2950
9.	93.	4.548	.8250
10.	93.	3.418	1.0770
11.	93.	4.229	.9360
12.	93.	4.458	.8180
4.	94.	4.269	.9030
6.	95.	3.805	1.3100
12.	97.	3.333	1.2170
4.	100.	3.645	1.2300
7.	100.	3.206	1.1720
11.	100.	3.339	1.0830
12.	100.	3.072	1.1800

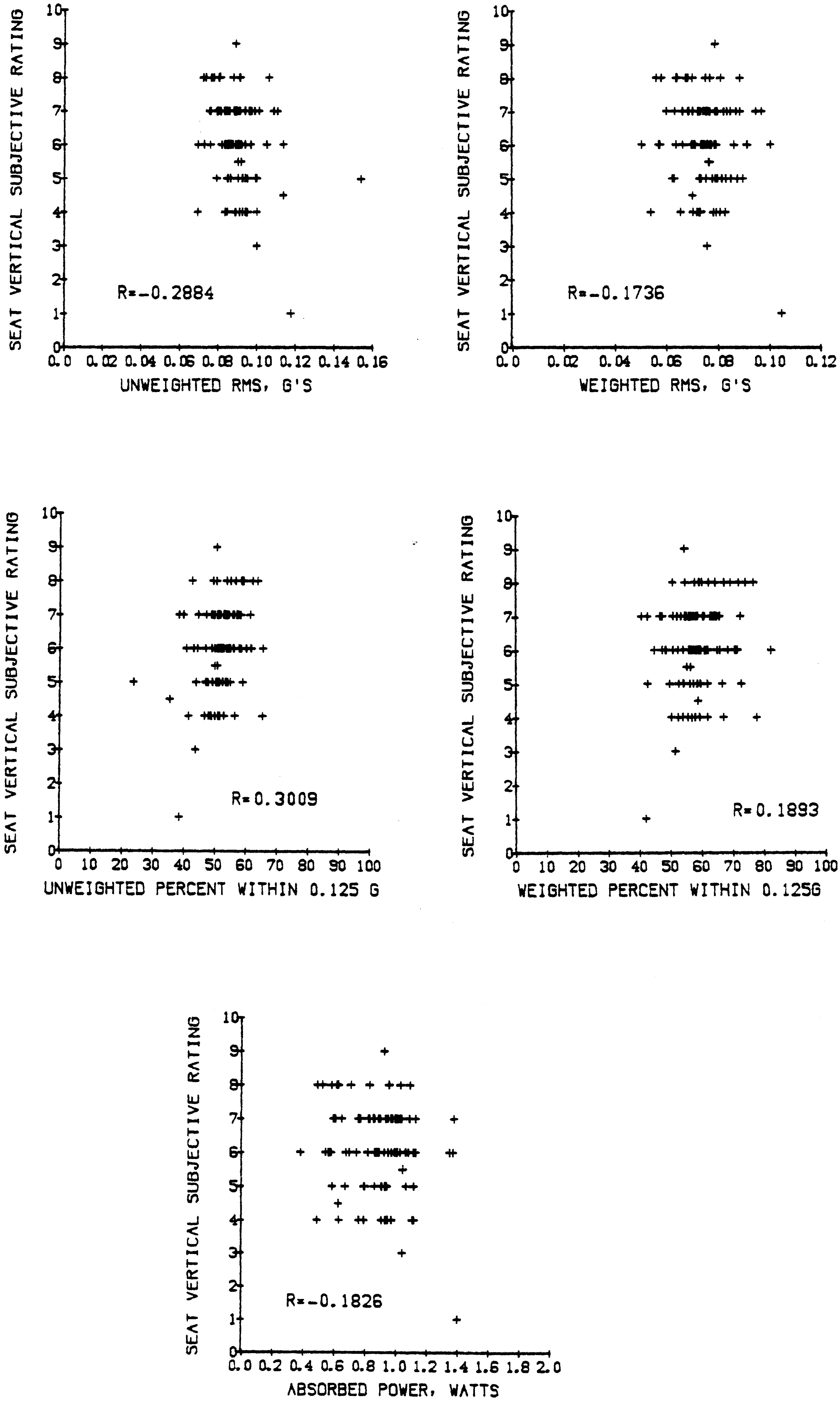


Figure A-3. Comparison of subjective ratings with objective measures of seat acceleration for the vertical direction.

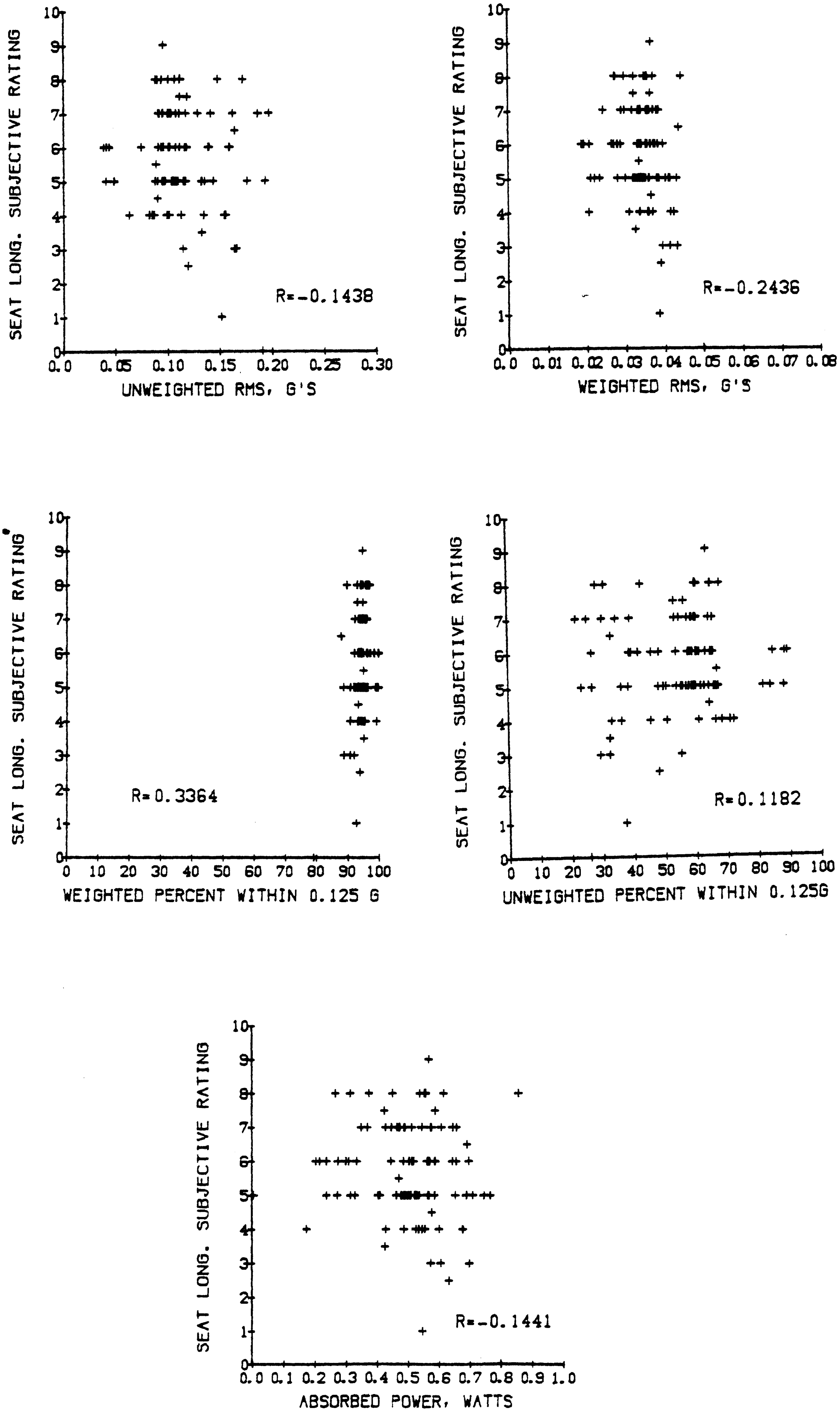
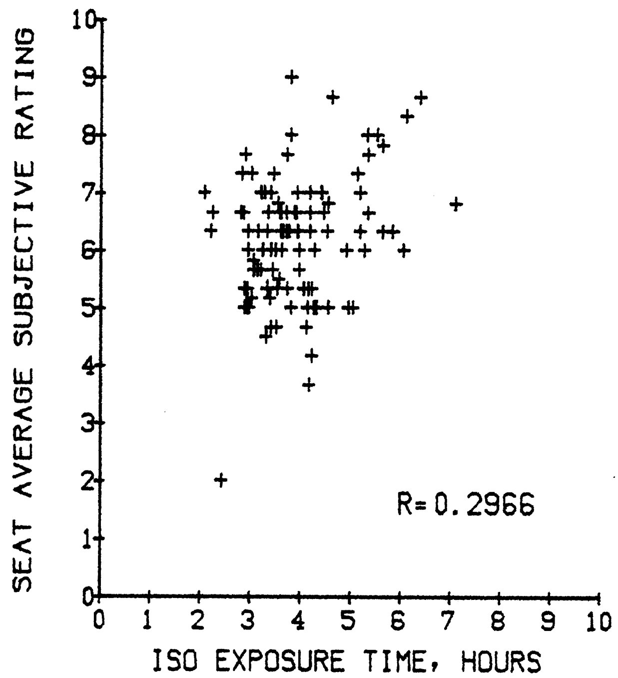


Figure A-4. Comparison of subjective ratings with objective measures of seat acceleration for the longitudinal direction.



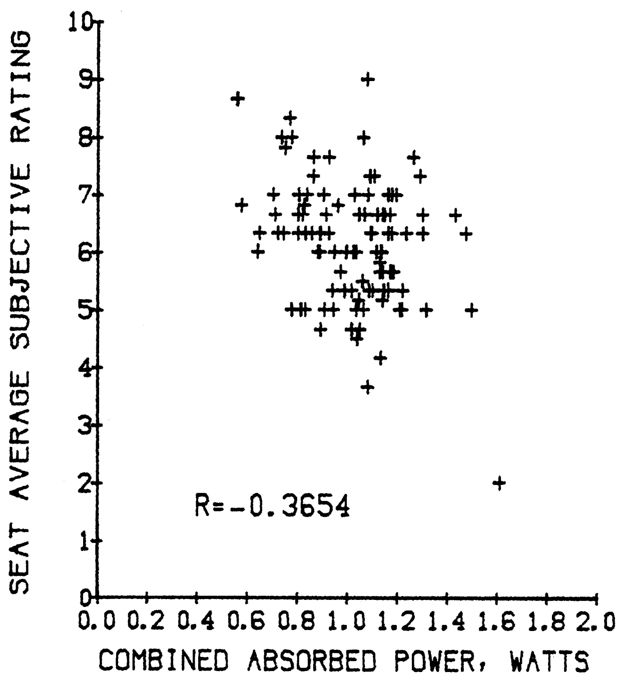


Figure A-5. Comparisons of seat average ratings with the ISO exposure limit and absorbed power.

TABLE A-5
Correlation of Seat Pad Accelerometer Measurements
with Subjective Ride Ratings

Accelerometer Measure	Correlation Coefficient
Correlations of Vertical Accelerometer Measures with Subjective Seat Vertical Ratings (N=103)*	
Weighted Percent Within ±0.125 G	0.1893
Unweighted Percent Within ±0.125 G	0.3009
Weighted RMS	-0.1736
Unweighted RMS	-0.2884
Absorbed Power	-0.1826
Correlations of Horizontal Accelerometer Measures with Subjective Seat Longitudinal Ratings (N=95)*	
Weighted Percent Within ±0.125 G	0.3364
Unweighted Percent Within ±0.125 G	0.1182
Weighted RMS	-0.2436
Unweighted RMS	-0.1438
Absorbed Power	-0.1441
Correlations of Combined Accelerometer Measures (Vertical and Fore-Aft) with the Average Subjective Seat Rating (N=103)*	
Combined Absorbed Power	-0.3654
ISO Exposure Time	0.2966

*For sample sizes of 103 and 95, correlation coefficients with absolute value greater than 0.1937 and 0.2017 respectively are significantly different than zero at the 5% level.