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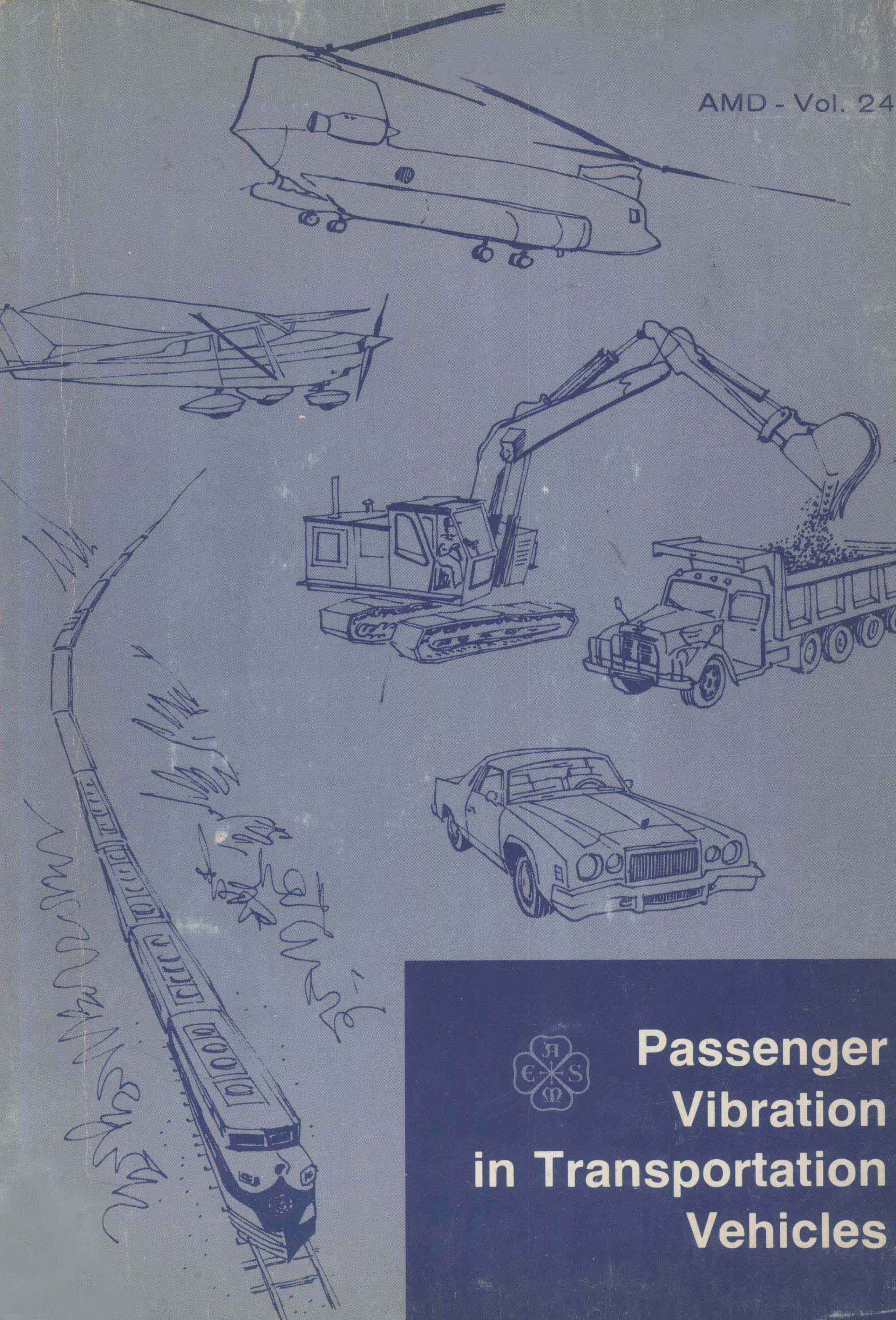

Passenger Vibration in Transportation Vehicles

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FOREWORD

Transportation systems have come to play a large part in the lives of a significant segment of the world's population. From daily trips to the place of employment, to occasional cross country business or vacation trips, to once-in-a-lifetime intercontinental emigration, the human race has achieved a level of mobility which would have been incomprehensible a short time ago.

Within the past several years, the attention of the technical community has become attracted to the vibration environment of those making use of all types of transportation vehicles. For the passengers, discomfort and fatigue due to vibration are major considerations which will determine their ability to perform tasks or enjoy recreation at the end of their journey. For those engaged in operating the vehicle, possible deterioration in efficiency and effectiveness in carrying out their duties and the impact on the safety of the trip are areas of concern.

In organizing this volume, an attempt has been made to sample the major types of transportation vehicles and to present the state-of-the-art related to the understanding of vehicle vibration and its impact on passenger comfort. The causes of vibration, methods of evaluating the vibration environment, and means of reducing the vibration are all treated by the contributing authors.

The paper by Stephens is general and treats vibration standards and comparative vibration environments for a number of vehicle types. Healy discusses automobile ride evaluation and means of measuring and evaluating measured data. The article by Glotzbach, Wentz, and Mehta considers heavy duty trucks, the vibration environment, and methods of improving the vibration characteristics. Sayers and Hedrick discuss the effects of track alignment and rail car suspension on ride quality of railroad passenger cars. In their paper, they illustrate methods of achieving acceptable standards at the lowest possible cost.

Large passenger aircraft are treated by Brumaghim and McKenzie. Their paper covers ride control technology from two distinct approaches or a combination of them. The final two papers treat the vibration problem in helicopters. Gabel presents a comprehensive discussion of the inherent causes and a number of means of reducing the vibration. The paper by Jones describes a specific project in which a rotor isolation system was designed, installed, and evaluated on a helicopter.

The editors are appreciative of the effort expended by the authors and thank them for all the hours of labor and the expertise they contributed to this project.

Alex Berman
Alan J. Hannibal

CONTENTS

DIGITAL PROCESSING OF MEASURED VIBRATION DATA FOR AUTOMOBILE RIDE EVALUATION

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ABSTRACT

This paper is concerned with the measurement and processing of motor vehicle ride vibration data so that comfort can be assessed. Typical vibrations experienced are random, with some periodic components sometimes included. Measures that are of interest here include r.m.s. (root mean square) acceleration in the three linear directions relative to the passengers' orientations, spectral density and one-third octave band r.m.s. acceleration levels. Here, procedures are described using digital analysis techniques. The effects of detrending, and windowing are considered and are shown to have little overall effect in calculating spectral values. Special consideration is given to the coverage here of the low frequency range 0.1 to 40 cycles per second. Typical results are shown for automobiles riding over highways having a wide range of roughness.

PRIMARY NOMENCLATURE

B - bandwidth for one-third octave, Hz.
c_p - cumulative probability (%)
f - general frequency Hz.
f_s - sample frequency Hz.
N - number of data points in record
p - probability density (% per g)
P - discrete acceleration spectral density value
t - time seconds
T - total record time
α - normalized acceleration (g units)
Δf - discretisation frequency Hz.
Δt - sample time, second
σ - root mean square value

INTRODUCTION

The ride quality of ground transportation systems is of significant current interest. With motor vehicles, the rubber tires act both as steering, traction and vertical suspension units. The necessary good adhesion to the road surface implies that roughness also acts as an input to the vehicle.

Many previous studies of automobile and truck ride quality have described the analysis of ride quality. In this case, it boils down to the need to measure and analyze the six degrees of freedom of body vibration in the frequency range 0.1 to 40 Hz. Comparison between vibration measures and acceptability evaluators then yields a figure of ride quality merit.

We are interested in covering the amplitude range 0.01 to 0.2 g for the r.m.s. level of vibration which is generally of a random nature.

In the past, methods of analysis have been primarily analog. Analog accelerometers are used as the basic vibration transducer and their outputs are recorded on analog tape. Conventional spectrum and one-third octave band analyzers have been used. The difficulty here is that the band width capabilities of usual analog equipment do not permit low frequency analysis. The solution has been to employ a variable speed tape recorder so that play back at high speed essentially shifts the frequency range of interest upward to the point at which the equipment can be used.

More recently, hybrid analyzers are available, but these do not allow the degree of flexibility afforded by completely digital analysis methods.

Digital analysis here is based on use of analog transducers, an analog data tape recorder, a digitizer and a computer (mini or otherwise) for separate analysis.

In this paper, considerations are given to the operations and procedures that the author has employed in the measurement and analysis of automobile ride quality. Some results concerning the evaluation of automobile ride quality are also given.

THE BASIC MEASUREMENT SYSTEM

The basic measurement system to be installed in an automobile is illustrated in Figure 1. A 110 volt, 60 Hz A.C. power supply is obtained from an inverter which is driven by the alternator through the 12 volt on-board battery. A 250 VA capability has been found to be sufficient.

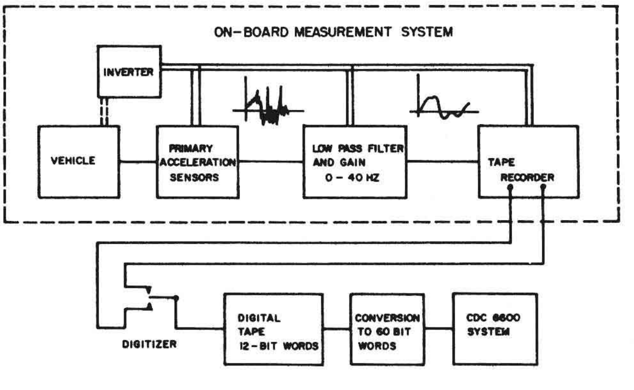

Fig. 1: Schematic Diagram of Measurement and Analysis System

Initial considerations are the frequency range and amplitude levels of interest. Here, 0 to 0.2 g r.m.s. level acceleration need to be measured covering the frequency range 0 to 40 Hz. Pieso-resistive or strain gauge seismic mass accelerometers enable the low frequencies to be measured and usually pass higher frequencies than necessary. Since higher frequencies are present, but not of interest in vehicle ride quality studies, a low-pass filter is extremely desirable as shown in Figure 1. These studies have used a 0-100 Hz bandwidth although a 40 Hz bandwidth filter is better. The reason for the filter is to reduce the high amplitude, high frequency content of the vehicle's actual vibration response so that maximum utilization of the range of the recorder can be made. The oscilloscope is extremely useful in verifying the quality of the signal being recorded. The tape recorder (F.M.) is required also to record signals in the low range of frequency and the gains must be set so that the average signal level occupies about one-third of the scale range. In this way, peak levels may be recorded undistorted. A typical vibration record is illustrated in Figure 2.

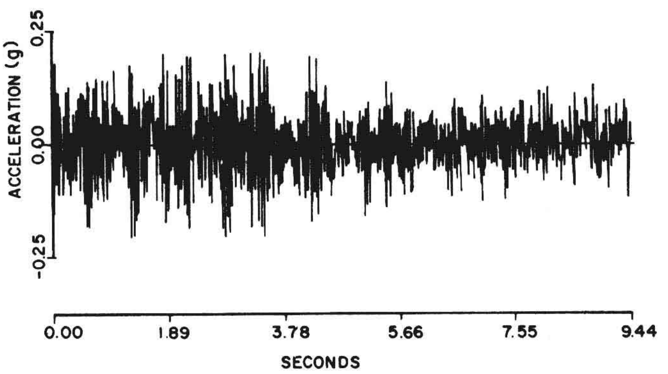

Fig. 2: Typical Vibration Record

DIGITAL ANALYSIS OF RANDOM VIBRATION

Generally, vibrations of interest in ride quality are random in nature. A few transient shocks may be present but these can often be isolated. In this context, it is important to review some of the measures of interest in dealing with random vibration.

As with most time varying signals, the two measures needed for adequate description of the signal relate to amplitude and frequency. With a deterministic signal, quantification at one point in time will determine its future behavior. With a random signal, this is not so, although by averaging over enough samples, at least, an expected behavior can be found.

In dealing with a random vibration, the amplitude probability density is used to determine the expected probability of finding a given level of acceleration. The cumulative probability function relates to the probability of having acceleration levels below a certain peak value.

The frequency content of a random signal is given by its so called power spectral density. The power spectral density value is an averaged measure of the amplitude of its frequency components. This is theoretically obtained from the auto-correlation function of the signal. The auto-correlation function is the average value of the product of two values of the signal separated by a predetermined time interval. This average value is therefore a function of that time interval. And, if the signal is stationary, only the time interval, not starting time, affects the answer. Converting time into frequency via the fourier transform, then yields the average amplitude of the signal's frequency components, i.e. its power spectral density.

In practice here, amplitude probability functions and power spectral density functions were computed digitally, the latter being computed directly from the discrete fourier transform procedure.

Other operations of interest calculate, cross spectral density, coherency functions, one-third octave band and frequency weighted r.m.s. values.

Basic Considerations in Choice of Record Length and Sample Frequency

Practically, some record time T must be chosen - sufficiently long to yield a reasonable estimate of the true spectral density but short enough to restrict the number of data points to a manageable quantity.

The record length is determined by the lowest frequency of resolution required in the analysis. For a resolution of 1 Hz, a record length of at least 1 second will be required. Practically, longer record lengths are required, but may lead to nonstationarity in the data depending on the character of the ride. For automobile rides, 10 seconds for each record length seems reasonable. The accuracy of statistical estimates of the spectral density are increased by ensemble averaging and data smoothing operations.

The sample frequency is based on considering the highest frequency component of interest. Limiting our interest to 40 Hz and allowing two octaves of frequency for attenuation by analog filters to prevent aliasing, gives a Nyquist folding frequency of 160 Hz. Thus sample frequencies should be higher than 320 Hz and the author has used between 400 and 500 Hz for safety.

Finally, in order to utilize the benefits of the fast fourier transform algorithm, the record length must have an integer power of two data points (512, 1024, 2048, 4096, etc.).

Thus a continuous record $\alpha(t)$ is reduced to a time series of some length N.

$$x_0, x_1, x_2, \text{--}\ x_k\ \text{--}\ x_{n-1}$$

If the total record time is T and the sample frequency is f_s (Hz) then

$$N = f_s\, T$$

$(f_s = 409.6;\ T = 10;\ N = 4096)$

Also, the time interval between samples is

$$\Delta t = T/N$$

and the corresponding discretized frequency interval is

$$\Delta f = f_{s/N} = 1/T$$

Auto and Cross Spectra

Using the fast Fourier Transform algorithm, the procedure for calculating the auto spectral density of a vibration record is

- Select 4096 data points
- Detrend and window the data
- Calculate the discrete fourier transform of the data
- Calculate its squared amplitude and multiply by the record length
- Ensemble average the power values from some number of similar records
- Smooth the averaged values over adjacent frequency bands to increase the number of statistical degrees of freedom
- Plot the results

For cross spectra

- Select 4096 simultaneous data points from each channel of data
- Detrend and window each data set
- Calculate the complex discrete fourier transform of each set (X,Y)
- Compute the cross power as $X^*_{(f)} \cdot Y_{(f)} \cdot T$.
- Ensemble average real and imaginary parts of the cross power over a

number of similar sets of record
- Smooth real and imaginary parts separately
- Plot averaged amplitude and phase results

Averaging and Smoothing and Statistical Errors

Since P(msf) are only estimates of the true spectral density of the measured vibration, some averaging is desireable. Each raw spectrum estimate has only two statistical degrees of freedom. This means that the expected standard deviation of error normalized by the true value (the random error) is unity. For q separate records of length T, the number of degrees of freedom n and the random error can be improved and are,

$$n = 2q; \quad \varepsilon_r = \sqrt{\frac{1}{q}}$$

with the resolution frequency is 1/T [3].

Frequency smoothing accomplishes the same objective. For example, if 10 estimates of the power at 1 Hz were required, a record of 10 second length would yield a discretization of 0.1 Hz and the values at 0.5; 0.6; 0.7; 0.8; 0.9; 1.0; 1.1; 1.2; 1.3; 1.4; averaged to produce the smoothed value $\hat{P}(1)$. Since each power value has two statistical degrees of freedom, this smoothing yields 20 degrees of freedom and a random error of $\varepsilon_r = 1/\sqrt{10}$. If 2d is the number of adjacent frequency bands used then,

$$\hat{P}(m) = \frac{1}{2d+1} \sum_{m'=m-d}^{m'=m+d} P(m')$$

giving 2(2d+1) degrees of freedom.

Thus for q ensembles, smoothed over 2d adjacent bands, we have

$$n = 2q(2d+1)$$

The resolution bandwidth for statistically independent estimates is now 2d, Hz, so that the random error is now

$$\varepsilon_r = 1/\sqrt{q(2d+1)}$$

Using a moving averaging scheme where m is indexed at intervals of the discretizing frequency yields statistically dependent estimates but the resolution is now increased to 1/T.

The relationship between expected tolerance of the spectral density value and the number of statistical degrees of freedom for independent estimates is given in Figure 3 for various confidence levels.

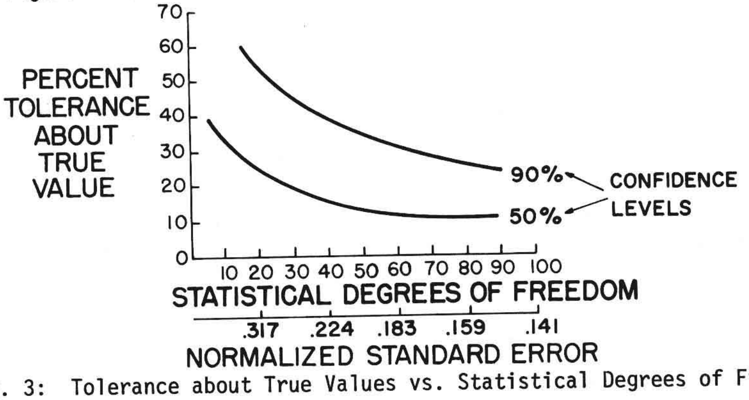

Fig. 3: Tolerance about True Values vs. Statistical Degrees of Freedom

In what follows, examples of spectral density values are given for the automobile case where for simplicity only a few degrees of freedom are used. It has been found useful to vary the number of bands in the frequency smoothing according to the frequency range. Here,

0 - 1 Hz	d = 1	n = 6
1 - 10 Hz	d = 2	n = 10
10 Hz up	d = 4	n = 18

with resolution of 0.1 Hz.

Frequency Weighting

While statistical measures such as mean and variance are important results in assessing statistical quantities, passenger response to random vibration is also affected by the dominant frequency content of the random vibration. Frequency weighting using a passenger comfort transfer function may be employed to emphasize the effects of harmful frequency content in computing root mean square values.

It is apparent from [4] that human passengers are more sensitive to vibration frequencies in the 4-15 Hz range. In [4], see Fig. 4), lines of constant comfort levels of acceleration for a range of frequencies are given for various exposure times. These are boundaries of fatique decreased proficiency. It seems appropriate then to invert a contour of this type for use as a frequency weighting function. Also, since the constant comfort contours in [4] relate to root mean square acceleration, the squared contour must be used in order to correctly weight spectral density values.

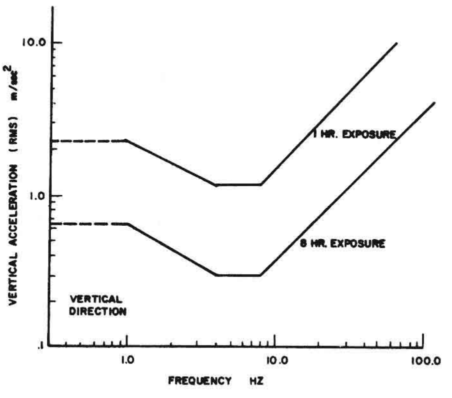

Fig. 4: Fatigue Decreased Proficiency Boundaries for 1 Hour and 8-Hour Exposure from I.S.O. 2631 - Vertical Direction

For the 8-hour reduced comfort exposure limit, the squared magnitude of the weighting filter $|G(f)|^2$ is given for vertical accelerations by

$0 < f < 1$	$\|G(f)\|^2 = 0.25$
$1 < f < 4$	$\|G(f)\|^2 = 0.25f$
$4 < f < 8$	$\|G(f)\|^2 = 1.0$
$8 < f < 100$	$\|G(f)\|^2 = 64/f^2$

For transverse vibrations

$0 < f < 2$	$\|G(f)\|^2 = 1$
$2 < f < 100$	$\|G(f)\|^2 = 4/f^2$

One-Third Octave Bands for Comparison with the I.S.O. Guide

Of particular significance, as well as plain r.m.s. acceleration, is the root mean square value of the original signal contained in one-third octave frequency bandwidths. If B is a one-third octave band centered at $\hat{f}$ = m/T Hz, then the one-third octave root mean square value is given by $\sigma(\hat{f})$ where

$$\sigma^2(\hat{f}) = \frac{1}{T} \sum_{m=(\hat{f}-B/2)T}^{(\hat{f}+B/2)T} P(m)$$

Naturally, care must be taken to ensure that $\hat{f}$ and B/2 are multiples of M/T, and, in cases where this is not possible, provision must be made to account for loss of content of the mean square value.

A one-third octave is defined in terms of the ration of two frequencies

$$\frac{f_2}{f_1} = 2^{1/3} = 1.25992$$

Thus the center frequency of a one-third octave band is

$$\hat{f} = (f_1 + f_2)/2$$

$$f_1 = 2\hat{f} - f_2$$

$$= 2\hat{f} - 2^{1/3} f_1$$

$$\text{So that } f_1 = \frac{2}{(1+2^{1/3})} \hat{f} = \underline{0.88498\hat{f}}$$

$$f_2 = \frac{2^{4/3}}{(1+2^{1/3})} \hat{f} = \underline{1.115\hat{f}}$$

So, the root mean square in a one-third octave band width centered at $\hat{f}$, $\sigma(\hat{f})$ may be computed and the relationships between $\hat{f}$ and B are

$$(f_2 - f_1) = (1.115 - 0.88498)\hat{f}$$

which gives

$$\left.\begin{aligned} B &= 0.23002\hat{f} \\ \text{and} \quad \hat{f} &= f_1/0.88498 \end{aligned}\right\}$$

Thus adjacent one-third octave bandwidths starting with the first frequency at 1.0 Hz are given by

f_1	f_2	B bandwidth	$\hat{f}$ center frequency
1.0	1.259	0.259	1.1299
1.259	1.587	0.337	1.423
1.587	2.0	0.413	1.793
2.0	2.51	0.51	2.251
2.51	3.17	0.66	2.843
3.17	4.0	0.83	3.587
4.0	5.04	1.04	4.52
5.04	6.35	1.31	5.695
6.35	8.0	1.65	7.175
8.0	10.08	2.08	9.04
10.08	12.7	2.62	11.39
12.7	16.0	3.30	14.35
16.0	20.16	4.16	18.08
20.16	25.34	5.18	22.72
25.34	32.0	6.36	28.70
32.00	40.32	8.32	36.16
40.32	50.8	10.48	45.56
50.80	64.0	13.20	57.39
64.00	80.6	16.60	72.29
80.60	107.6	21.00	91.12

These center frequencies will appear in later plots of $\sigma(\hat{f})$ plotted versus $\hat{f}$ for measured vibration records.

RESULTS FOR AUTOMOBILE RIDE QUALITY ASSESSMENT

Based on a series of experiments in the Central Texas area using a 1974 Buick Century sedan automobile, analog vibration records of vertical and lateral vibration were made. The automobile speed was held constant at 50 mph and driven over roadways of varying degrees of roughness. The accelerations were measured on a box fixed beneath the driver's seat between the seat and the body floor panel. While some panel vibration may have been present, this was an attempt to measure the acceleration input to the driver's seat.

The digitizer produces a twelve bit binary word (2^{12} range) covering a range ± 1 volt. Thus the resolution of the digitized signal was about 0.0005 volts or a 0.02% of full-scale resolution leaving the tape recorder as the prime source of inaccuracy for low level signals. The choice of sample frequency relates to the possibility of aliasing errors in the sampled data. To minimize aliasing errors, a 100 Hz bandwidth aliasing filter was employed before sampling and a sample rate about 434 Hz was used. The amplitudes of frequency components at or around $f_{s/2}$ (k.e. 217 Hz) must be small in the data to be sampled; otherwise aliasing errors will be significant. Examination of the data has shown that frequency components at the folding frequency 217 Hz are usually 2 decades lower than other values thus ensuring that aliasing errors be less than 1%.

Utilizing the speed of the Fast Fourier Transform Algorithm [1], and sampling at 434 Hz with a record length of 4096 data points yields a record time of

$$T = \frac{4096}{434} = 9.437 \text{ seconds.}$$

Thus, $\Delta f = 1/T = .106$ Hz.

Detailed analysis was accomplished using a CDC 6600-6400 system. A conversion program is thus required to convert the 12-bit words as digitized into 60-bit words as used in the 6600 system computer.

Figures 5, 6, and 7 show the vertical and lateral acceleration spectra together with one-third octave r.m.s. values compared to the I.S.O. guide for one hour and eight hours fatigue decreased proficiency. The I.S.O. guide has been converted into equivalent spectral values and is also compared with the U.T.A.C.V. [5] spectral envelope. In Figs. 5, 6, and 7, three different roadway test sections have been isolated corresponding to rough, medium and smooth quality.

In assessing ride quality using I.S.O., the penalty by exceeding the guide in any one-third octave band is not known. However, the importance of Figures 5-7 is to show the relationships between ISO and typical sedan rides.

It should be pointed out that the rough, medium and smooth categories refer to rides with which almost everyone is familiar. That is, "rough" corresponds to a 50 m.p.h. ride over a rough secondary road, "medium" refers to a 50 m.p.h. ride over average quality U.S. highway construction and "smooth" refers to U.S. Interstate level roadways.

For vertical vibrations, the energy around 1 Hz corresponds primarily to the combined body modes of heave, roll and pitch, while the peaks around 12-18 Hz are associated with wheel axle bounce and roll modes. In the range of 30 Hz, the vibrational energy is possibly associated flexible body modes and tire nonuniformity. As can be seen, no two spectra are exactly alike, although the general shape is preserved.

For lateral accelerations, the spectra show generally higher values in the 0.1 to 1 Hz range than at higher frequencies. This can be attributed to steering adjustments and long wavelength bank angle changes. For medium and rough sections, there is a definite increase in the 10-30 Hz range content of lateral vibrations.

As can be seen, and as expected, the automobile spectra exhibit one-third octave rms.g. levels which mostly lie below the 8-hour I.S.O. fatigue decreased proficiency guideline. The rough section exceeds the 8-hour I.S.O. guide in both the low and high range. This ride is definitely rough corresponding to

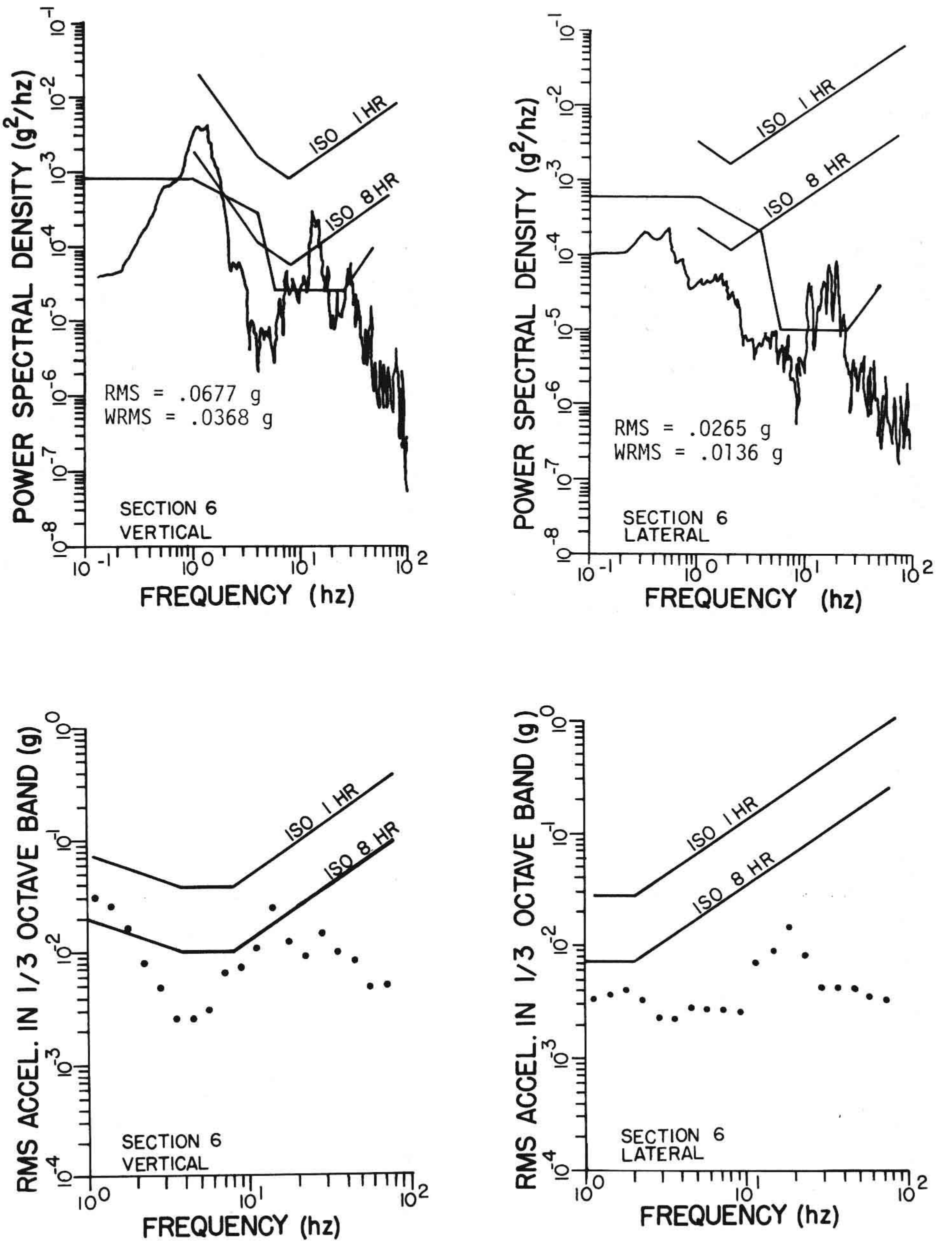

Fig. 5: Vertical and Lateral Acceleration Spectra
for a Buick at 50 m.p.h. - Rough Road

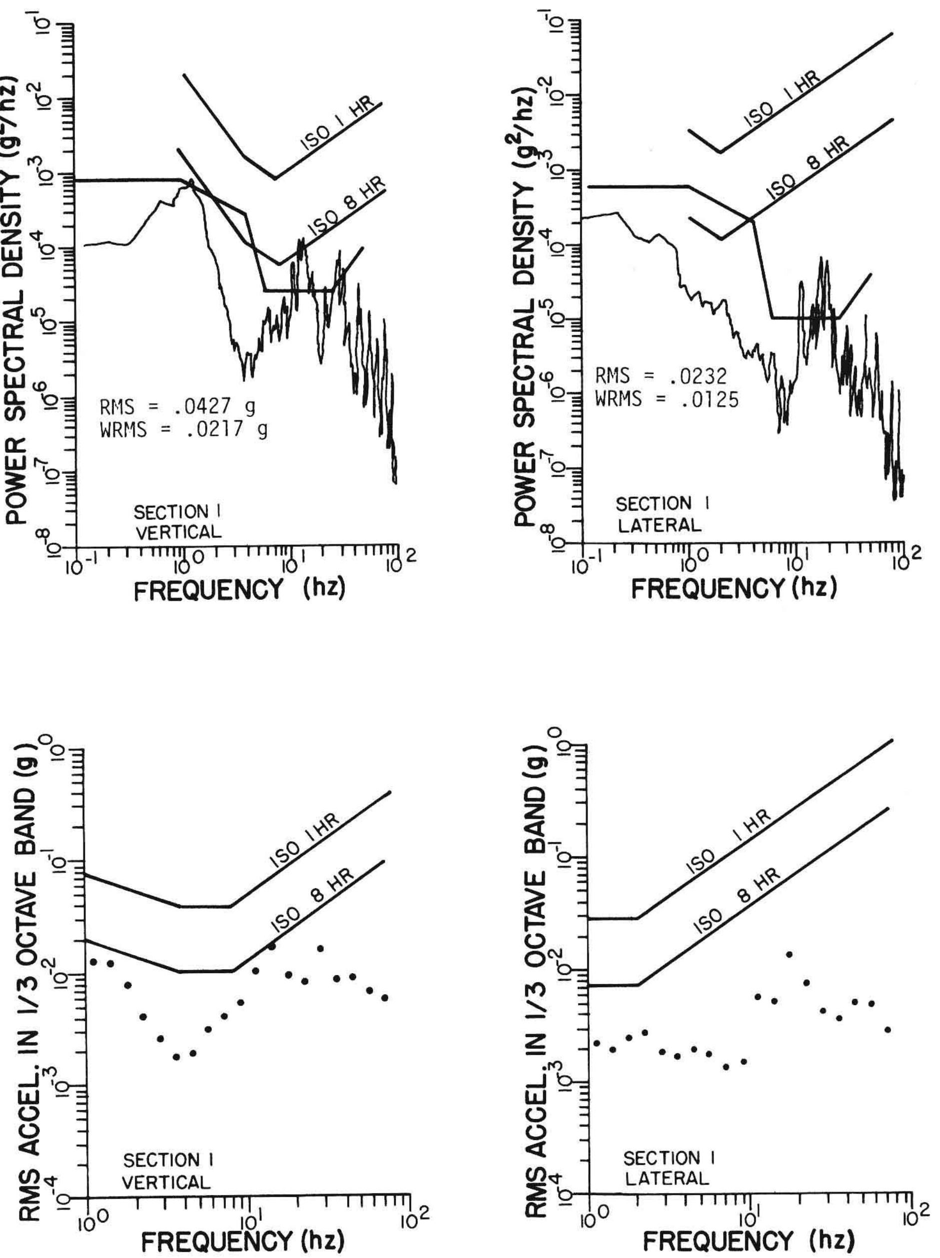

Fig. 6: Vertical and Lateral Acceleration Spectra
for a Buick at 50 m.p.h. - Medium Road

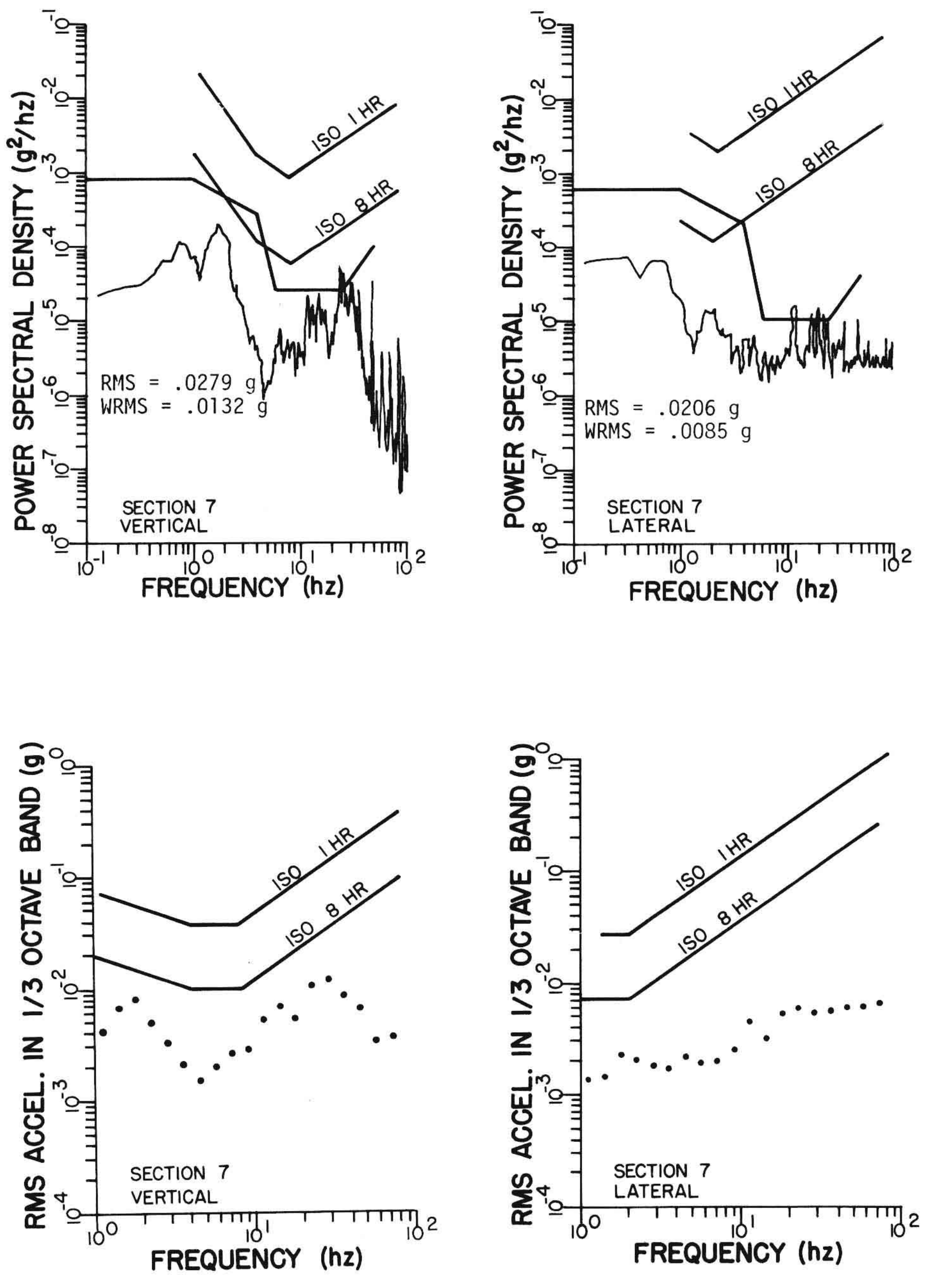

Fig. 7: Vertical and Lateral Acceleration Spectra - Smooth Road

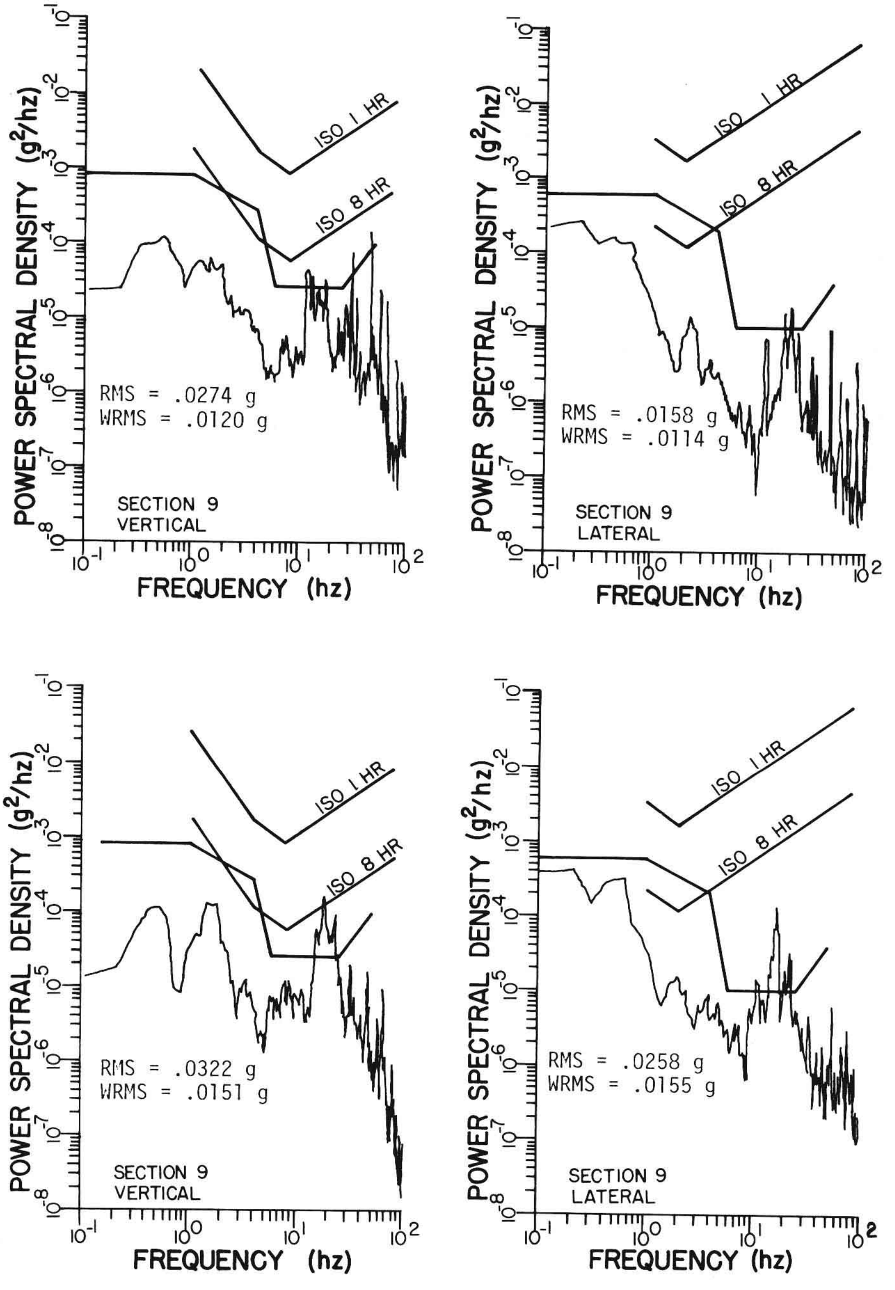

Fig. 8: Vertical and Lateral Acceleration Compared,
Upper Buick, Lower Maverick