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INTRODUCTION

Many different investigators have attempted to quantify vibration severity by subjecting people to different vibration environments and recording their reaction. The major difficulty encountered in attempting to quantify vibration is that one is not quantifying a physical phenomenon but human reaction to an external stimulus. This reaction must take the form of a qualitative assessment or it must be relative to some other stimulus. The problems associated with both of these methods of evaluation will be discussed.

There have been many attempts to attach subjective ratings to a vibration severity. Referring to Figure 1, which is a compilation of data from Goldman, Chaney, and Magid & Ziegenauecher, it is noted that the intolerable curve from Goldman is only slightly above Chaney's "perceptible" and considerably below his "mildly annoying." Magid & Ziegenauecher's one, two, and three-minute curves are also shown to show their relationship to the other curves. There is one thing that is evident from this conglomeration of data, i.e., there is no agreement on what is "annoying", "tolerable", etc. The reason for this is this type of a description cannot be attached to a vibration without defining the environment. In other words, one cannot say that a vibration is uncomfortable without first defining the environment the individual is in. A vibration in an automobile may be annoying, uncomfortable, or even intolerable, but the same vibration in a truck may be termed "comfortable." A person could ride for hours in his automobile, but if this same vibration were induced in his living room at home, it would be intolerable. Consequently, it is not

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possible to assign this type of a description to a vibration severity.

The other method used to evaluate a vibration is to rate one vibration or sensual input relative to another. This is called cross-modality and one either relates light or sound intensity to a vibration, or one vibration is rated relative to another. This method of evaluating the severity of a vibration is at best a very controversial procedure.

Whole-body vibration is a completely different phenomenon from a highly localized, or a single sensory, input. In whole-body vibrations, the sensations that occur in the 4 to 7 Hertz range are entirely different from the sensations that occur in the 8 to 15 Hertz range. In the 4 to 7 Hertz range, the primary objection to the vibration is the resonating, or relative motion of heart, lungs and other organs located in the thorax. It is believed that this is caused by the mass above the diaphragm resonating, with the diaphragm acting as a spring. In the 8 to 15 Hertz, the primary objection is movement of the head. This is also the range of frequencies where loss of visual acuity begins to become pronounced. It is believed that this loss of acuity is caused by a resonating condition in the spinal column. The major objection in the low frequency range (below 5 Hertz and assuming less than 1g peak) is the relative motion of individual and environment.

It should be understood that the previously mentioned sensations, and an undetermined amount of others, occur simultaneously at all frequencies, but become more pronounced in the frequency range mentioned. Consequently, unless cross-modality measurements made for different sensations give the relation between the measured sensations then they cannot be used to evaluate whole-body vibration at different frequencies.

There is some indication that the evaluation of ride may be possible using cross-modality at a single frequency. It is commonly accepted that ride is proportional to some power of acceleration at a single frequency, i.e.,

$$R \propto A^n \qquad (1)$$

or

$$R = KA^n \qquad (2)$$

Using cross-modality, it may be possible to solve for the exponent n, and determine if it is frequency sensitive, but at the present it is believed that there are too many unanswered questions to evaluate the frequency dependence of K using cross-modality.

ABSORBED POWER

The original formulation of absorbed power as a means to measure vibration severity occurred by observing many subjects being vibrated in a ride simulator and from personally spending many hours being subjected to different vibrations. From this experience, two observations were made:

a. The more relative the motion occurring between various parts of the body, the more severe the vibration sensation.

b. Doubling the amplitude of the acceleration more than doubles the severity.

From these two observations a theory was postulated: "The severity of a vibration is proportional to the rate at which the body is absorbing energy." From this statement an equation can be written that expresses it in mathematical terms:

$$P_{ave} = \lim_{T \to \infty} \frac{1}{T} \int_0^T F(t)V(t)\,dt \qquad (3)$$

This calculates the average power absorbed by a human when F(t) is the input force and V(t) is the input velocity. Note, for a solid mass, the average is zero.

Several important observations can be made from this method of evaluating vibration severity.

a. Absorbed power has a physical significance and interpretation. Its variation with different subjects can be measured.

b. It does not rely on sinusoidal boundary limits to determine comfort limits. There is no concern for having clean acceleration sine waves to determine severity.

c. It gives a single numeric value for a vibration.

d. It can be used for periodic, aperiodic, and random vibration.

By setting absorbed power to a constant and solving for the acceleration at each frequency, a constant comfort or tolerance limit curve is obtained. Figure 1 has a six watt curve for vertical vibration superimposed with the other data to show how it compares with this experimental data. Note, although this 6 watt curve was derived, its correlation with experimental results is very good, giving additional credence that absorbed power accurately measures vibration severity. If a constant power level for a particular vibration environment is determined, then this yields the sinusoidal tolerance limit curve. From past experience 6 watts is about the limit for cross country operation and .2 to .3 watts for automobiles in the vertical direction.

To validate absorbed power as a means of determining vibration severity, a force-measuring platform was built and several volunteers were subjected to a series of vibration tests. Absorbed power was measured and subjective response was recorded. Results of this test were published in a previous paper (1)*. Absorbed power correlated extremely well and it was determined that it accurately measured vibration severity. At this point the shape of the sinusoidal tolerance curve was not known but vibration severity was being determined.

The equipment required to measure absorbed power was fairly elaborate and it could only realistically be done in the laboratory. The force-measuring platform was large and bulky and required considerable instrumentation for the dynamic measurement of force. What was required was a means to obtain force from an accelerometer. To accomplish this a transfer function was derived that relates force to acceleration for the average of 31 subjects. This was done for the three linear motions; vertical, side to side, and fore and aft.

This greatly simplified the calculation of absorbed power and made it possible to calculate it entirely on a computer. However, there was still a problem in the low frequency end of the spectrum. The calculation of absorbed power for low frequency was very dependent upon the phase between force and velocity. Consequently, high grade electronic components were required to calculate power from an acceleration signal or considerable error would be introduced. This problem was eliminated by deriving a frequency dependent weighting function.

*Underlined numbers in parenthesis designate reference at end of paper.

ELECTRONIC WEIGHTING FUNCTIONS

To design a portable electronic instrument to measure absorbed power, it was necessary to derive frequency and amplitude weighting functions. This derivation proceeds as follows:

Writing the force as a sum of sine waves:

$$F(t) = \sum_{i=o}^{n} F_i \operatorname{Sin}(W_i t + \varphi_i) \qquad (4)$$

and velocity as:

$$V(t) = \sum_{i=o}^{n} V_i \operatorname{Cos} W_i t \qquad (5)$$

Inserting these expressions into equation 3, the relationship for absorbed power becomes:

$$P = \sum_{i=o}^{n} F_i V_i \operatorname{Sin} \varphi_i \left[\lim_{T \to \infty} \frac{1}{T} \left(\frac{T}{2} + \frac{\operatorname{Sin} 2W_i T}{4W_i} \right) \right] \qquad (6)$$

The intermediate steps in this and the following computational processes are given in reference (2). Taking the limits, equation 6 becomes:

$$P = \frac{1}{2} \sum_{i=o}^{n} F_i V_i \operatorname{Sin} \varphi_i \qquad (7)$$

The transfer function relating force to acceleration is:

$$G(jW) = \frac{F(jW)}{A(jW)} \tag{8}$$

Using this expression and noting the appropriate substitutions for sinusoidal waves, equation 7 is rewritten as:

$$P = \sum_{i=0}^{n} \left(\frac{\left| (jW_i) \right|^2 A^2(W_i)}{W_i} \right) \mathrm{Sin} W_i \tag{9}$$

Where, $A^2(W_i)$ is the mean squared acceleration at W_i.

This can then be written as:

$$P = \sum_{i=0}^{n} K(W_i) A^2(W_i) \tag{10}$$

Where:

$$K(W_i) = \frac{\left| (jW_i) \right| \mathrm{Sin}\, \phi_i}{W_i} \tag{11}$$

The values for $K(W_i)$ have been derived for vertical, longitudinal, and transverse motions for the seated subject (3).

In order to develop an electronic circuit to compute absorbed power, two conditions must be met. The output from the circuit must vary as the square of acceleration and have appropriate frequency characteristics to agree with the calculation of absorbed power. In this way, a weighting function circuit can be synthesized with the frequency respons* such that its output, when squared, will yi^d absorbed power.

If an acceleration signal is inputed to a filter circuit, the output frequency W_i is given by:

$$H(jW_i)A(W_i) \tag{12}$$

where $H(jW_i)$ is the weighting function for the filter circuit. Squaring this quantity and equating it to absorbed power at W_i:

$$H(jW_i)\ H\ (-jW_i)\ A^2\ (W_i) = K(W_i)\ A^2\ (W_i) \tag{13}$$

or:

$$\left|H(jW_i)\right|^2 A^2\ (W_i) = K(W_i)\ A^2\ (W_i) \tag{14}$$

Thus:

$$\left|H(jW_i)\right| = \sqrt{K\ (W_i)} \tag{15}$$

Because of the squaring process after filtering, we are interested in only the magnitude of $H(jW_i)$.

The entire absorbed power computing process is shown in Figure 2 for the three directions of motion. The frequency response was plotted for each motion and polynomials in terms of the complex variable S were fitted to these curves. The weighting function circuits were then synthesized from the derived transfer functions. The frequency response, transfer function in terms of S and the circuit diagram for the weighting functions are given in Figures 3-5 for vertical, longitudinal, and transverse directions of motion respectively. The units on the axis of the frequency response curves is watts/ft/sec^2. A more detailed description of the development of these functions and circuits is given in reference (4).

FIELD TESTS

Several portable absorbed power measuring devices have been constructed for use in field tests. These have consisted of both single directional (vertical) and three directional devices. Both types consist essentially of three units - a small electronic computer containing the weighting circuits, a power supply, and an accelerometer.

These instruments have been used extensively in determining the effect the dynamic characteristics of a vehicle have on the driver and passengers during cross-country operation. Some of the field tests in which the vibration measuring instruments have been used include:

a. Validation tests of the ride dynamics portion of the AMC Mobility Model.

b. Ride studies to determine the dynamic characteristics of various commercial trucks for use in the WHEELS Study.

c. Ride dynamic tests of the standard M60A1 and the M60A3 with tube-over-bar suspension.

These tests have been conducted at various locations including Ft. Knox, Ft. Sill, Ft. Hood, Yuma Proving Ground and Houghton, Michigan both during summer and winter seasons. Throughout this wide range of environmental conditions the instruments have yielded excellent results.

As an example, results of the ride dynamic tests of the M60A1 and M60A3 conducted at Ft. Knox are presented. These tests were conducted over selected courses at constant speeds (5). Absorbed power measurements were taken at different speeds and averaged for the test run. These levels were then plotted versus vehicle speed for each test condition and a comparison made of the ride dynamic characteristics of each vehicle. The results of the Ft. Knox tests are shown in Figure 6. It is obvious that from this set of curves, the tube-over-bar suspension system is superior to the standard torsion bar suspension. Note that the tolerance limit was not achieved during these tests.

Another example is one where the use of the three directions in computing absorbed power has led to the conclusion that in some cases of cross-country operation, the longitudinal and transverse motions significantly affect and sometimes dominate the vibrational effect on the driver. The case in point is a test conducted at Ft. Sill during validation studies of the ride dynamics portion of the

AMC Mobility Model. In these tests only the vertical absorbed power readings were being made. Although very low readings were being measured, the subjective response of the drivers indicated that they were at or near the tolerance limits. When the longitudinal and transverse readings were taken, it was discovered that these levels were at the tolerance limits. These ride measurements are shown in Figure 7. This particular case was a 4 MPH run with an M35A2.

This condition of higher vibration levels in the longitudinal and transverse directions occurs mainly in wheeled vehicles at lower speeds. As the speed of the vehicle is increased, the vertical vibration levels become the significant ride limiting motions.

Other uses of the instrument include verification of laboratory vehicle simulation tests. For example, if a suspension system design change is required to reduce the severity of a cross-country vehicle, engineers can conduct simulation tests in the laboratory on a wide variety of suspension system modifications. When the best suspension system configuration is determined, it is possible to evaluate the system under actual field conditions, using the data obtained with the vibration measuring instrument to validate the laboratory test results.

CONCLUSIONS

1. To set tolerance or comfort limits to a vibration one must also define the vibration environment.

2. Absorbed power has a physical significance and its variation with individuals and amplitude can be measured.

3. The constant absorbed power curve correlates very well with experimental tolerance data.

4. The application of sinusoidal boundary curves to a random type vibration is ambiguous.

5. Absorbed power can be calculated in both the time and frequency domain with no introduction of ambiguity or error.

6. Vibration levels can in some cases of cross-country operation, be higher in the longitudinal and transverse directions than in the vertical direction.

7. To determine a tolerance limit for a vibrational environment, all three directions must be taken into account.

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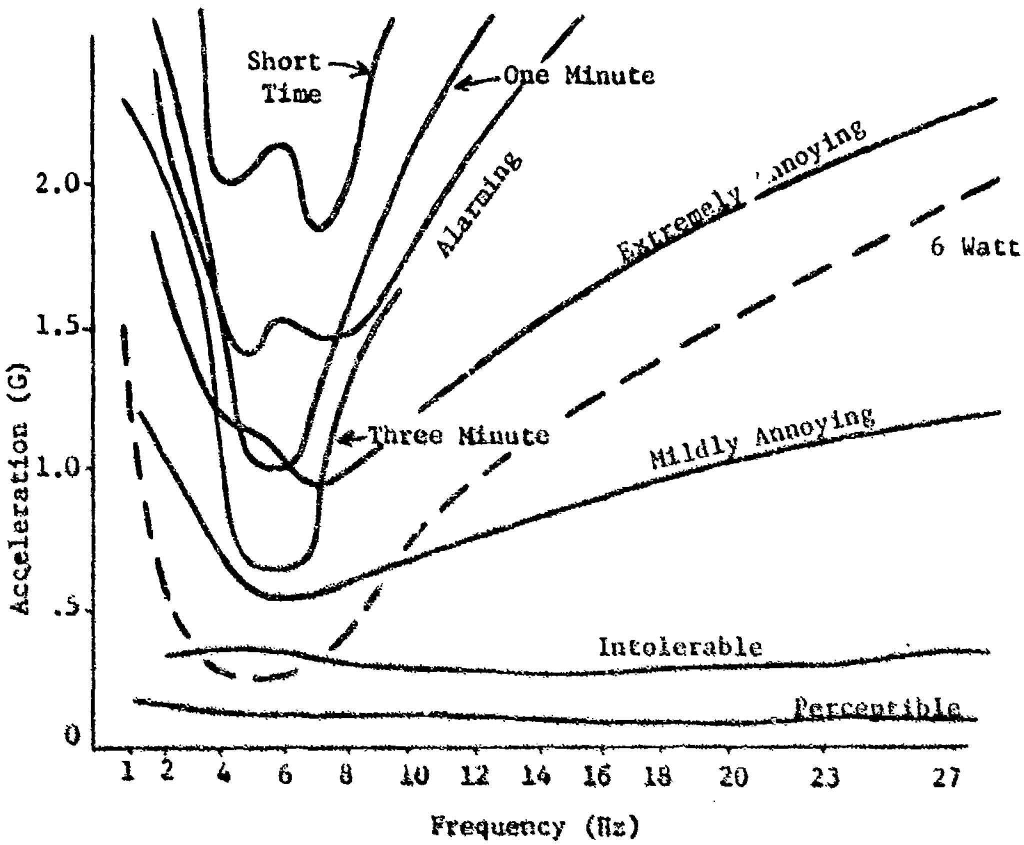


Fig. 1 Comparative Human Vibration Response Curves

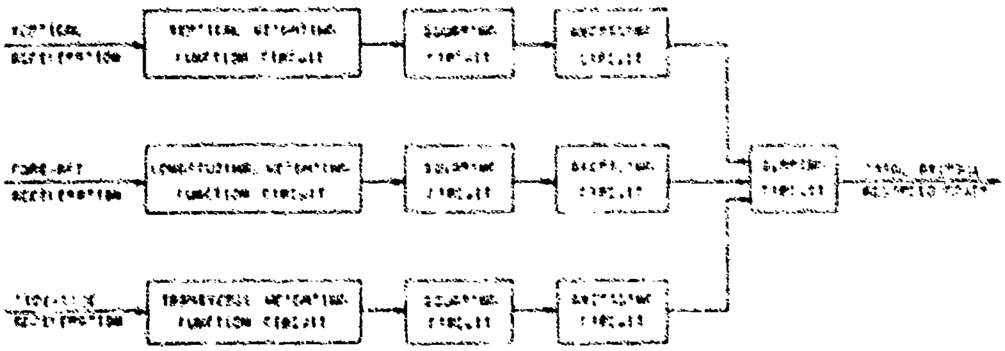
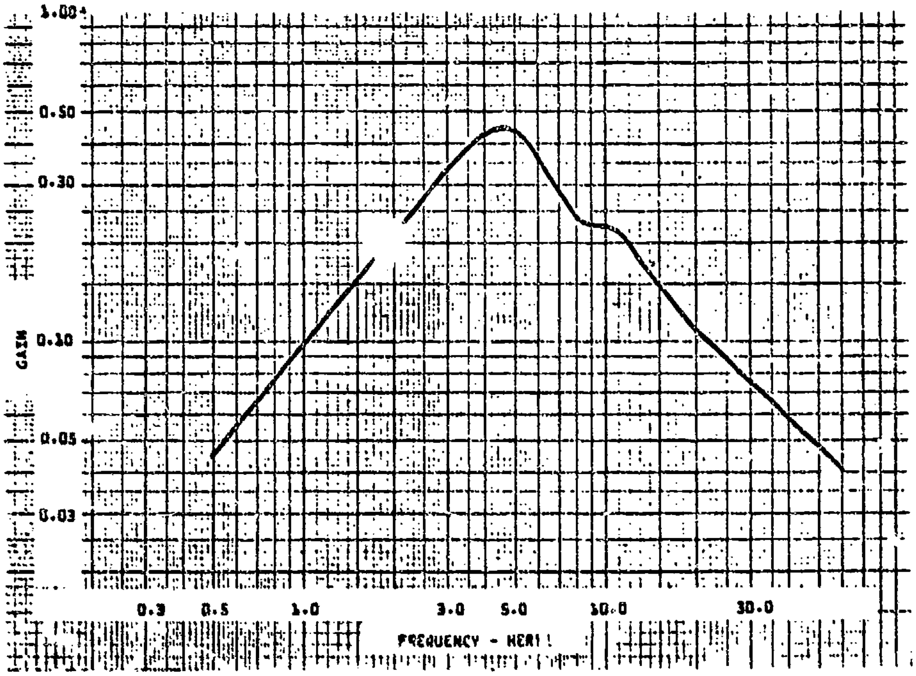


Fig. 2 Total Average Absorbed Power Diagram



$$\frac{15.453\,S\,(S+5.0)\,(S^2+28.3\,S+2800)\,(S^2+105.0\,S+7570)}{(S+6.0)\,(S^2+29.8\,S+1000)\,(S^2+39.1\,S+3800)\,(S^2+125.0\,S+5160)}$$

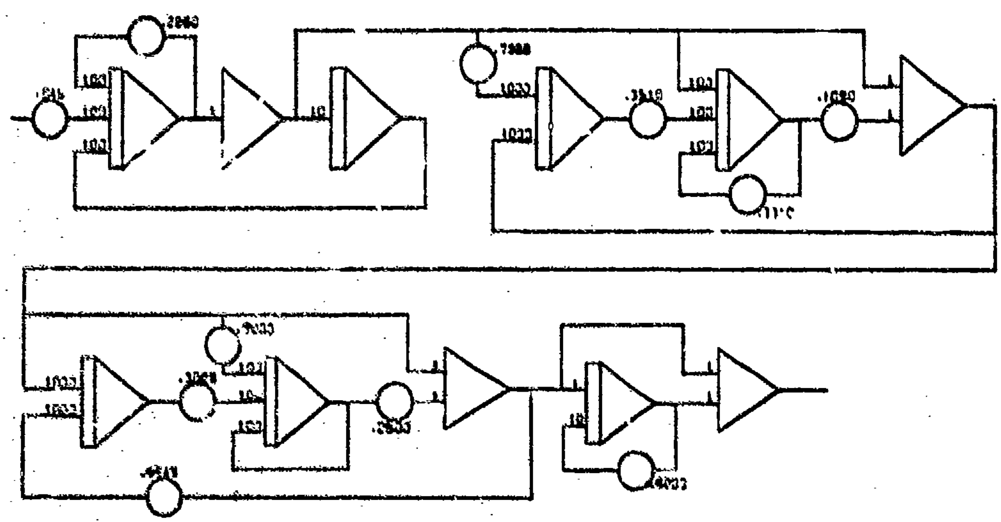
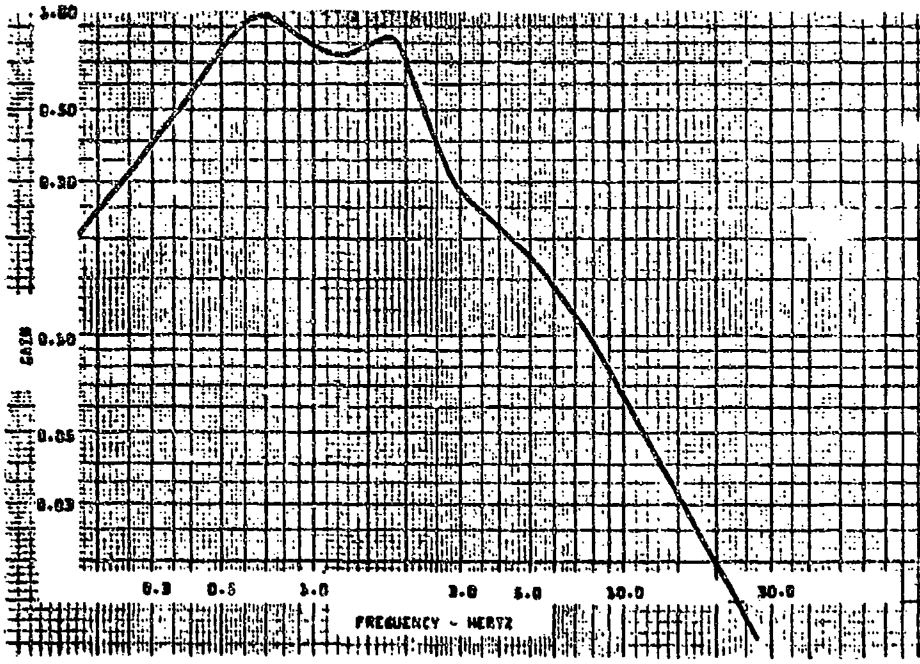


Fig. 3 Vertical Weighting Function and Circuit



$$\frac{478.0\,S\,(S+130.0)\,(S^2+11.2\,S+60.0)\,(S^2+14.2\,S+260.0)}{(S^2+3.33\,S+17.0)\,(S^2+5.5\,S+140.0)\,(S^2+44.0\,S+900.0)\,(S^2+255.0\,S+2500)}$$

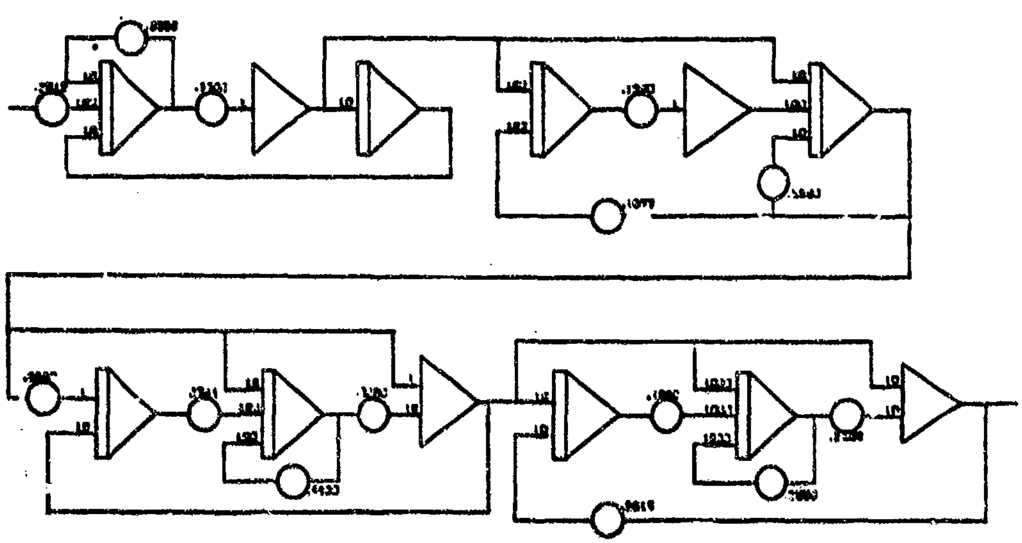
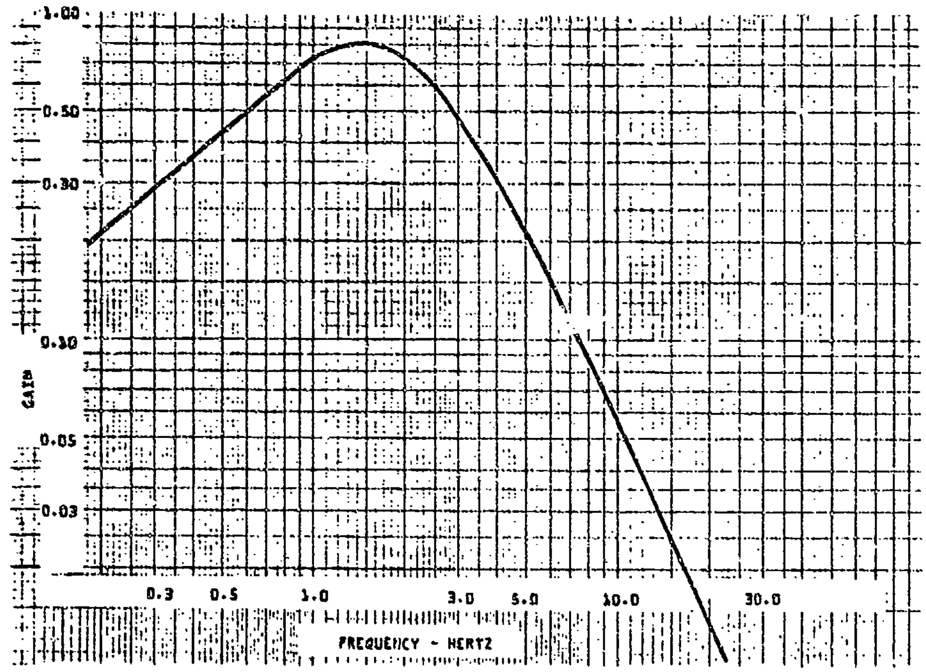


Fig. 4 Longitudinal Weighting Function and Circuit



$$\frac{209.0\,S\,(S + 110.0)}{(S^2 + 17.6\,S + 125.0)\,(S^2 + 110.0\,S + 1300)}$$

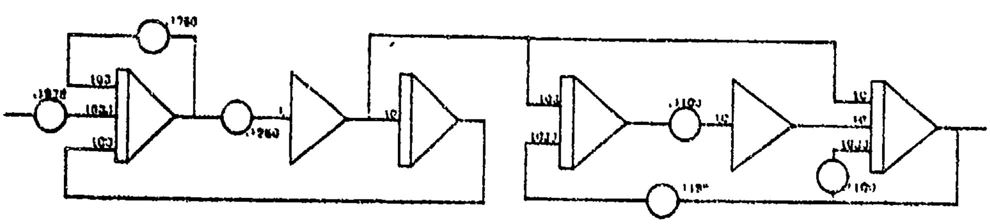


Fig. 5 Transverse Weighting Function and Circuit

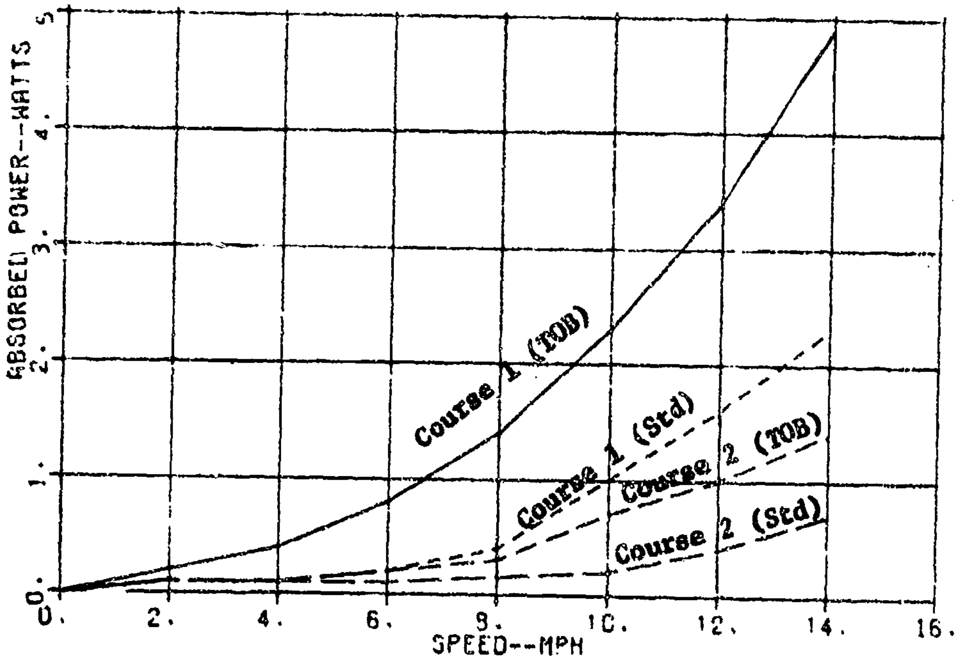


Fig. 6 M60 Suspension Comparison

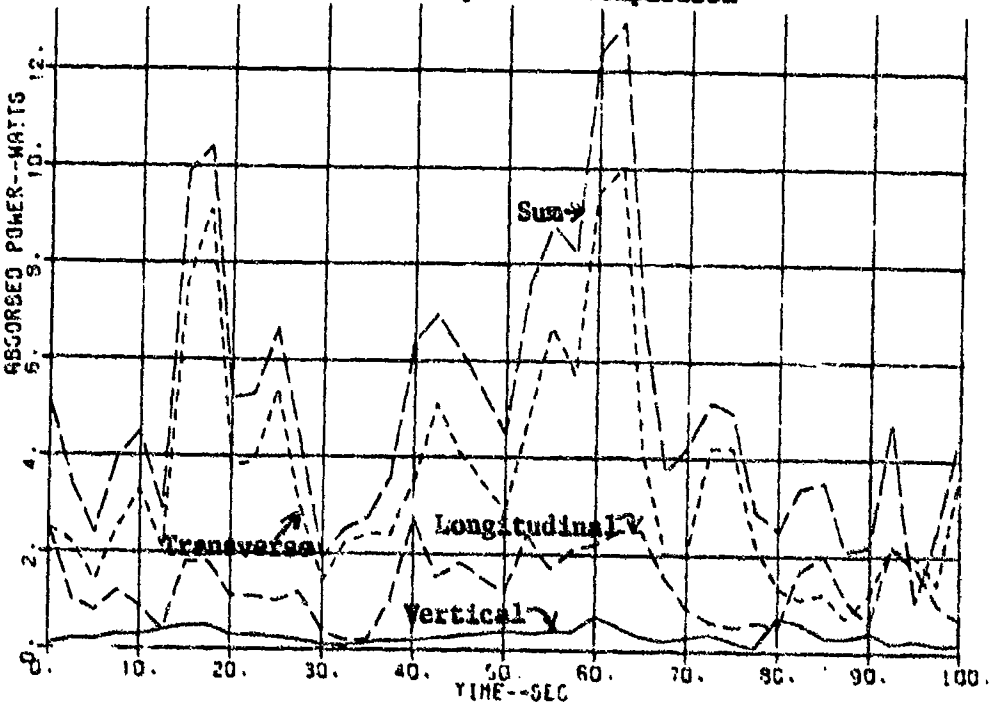


Fig. 7 M35 Three Axis Power Comparison