VIBRATION TESTING

USE OF FORCE AND ACCELERATION MEASUREMENTS IN SPECIFYING AND MONITORING LABORATORY VIBRATION TESTS

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The usual practice of basing vibration test specifications on an envelope of the equipment base acceleration levels experienced in the service environment can often result in excessive levels of overtesting. This results from the large difference between the mechanical impedance of the vehicle structure and that of a fully equalized shaker. A possible solution to the problem is the generation of test specifications that are based on the knowledge of both the accelerations and the forces transmitted to the equipment in the service environment.

Results are presented of a program devoted to the development of improved procedures for defining the vibration levels experienced by equipment attached to flight vehicles and providing realistic laboratory test levels. The program involved the measurement of the accelerations and forces experienced by simulated equipment attached to a vibrating aircraft fuselage. The equipment was later removed from the vehicle, placed on a shaker, and vibrated to a test specification based on an envelope of the acceleration peaks measured in the fuselage. Comparisons were then made of the acceleration levels that a resonant element in the equipment received in the fuselage and on the shaker. The tests on the shaker were repeated with the exception that, at those frequencies corresponding to an equipment resonance, the force transmitted to the equipment base was not allowed to exceed the maximum force developed (at any frequency) when the equipment was attached to the vibrating fuselage. It was found that this procedure largely eliminated the high levels of overtesting introduced by the conventional approach.

INTRODUCTION

Standard methods for specifying laboratory vibration tests often result in subjecting the equipment being tested to a damage potential that is much greater than it will experience in service. The overtesting involved arises from the practice of specifying laboratory test levels on the basis of an envelope of the acceleration peaks measured in the service environment and employing narrow-band equalization to eliminate any spectral notches that may result from equipment antiresonance effects during laboratory testing. This procedure, in effect, causes the shaker to present an infinite mechanical impedance to the attached equipment, while in the

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service environment the impedance of the supporting structure may in some cases be less than that of the attached equipment.

In a previous paper [1], the author discussed the possibility that this problem could be resolved if the force transmitted to the equipment by the vehicle supporting structure could be measured both in the service environment and during laboratory testing. The suggested force measurements were to be performed in addition to the acceleration measurements normally taken. The referenced paper pointed out that it would be necessary for the force transducers to have very low mass and compliance to preclude the possibility of modifying the dynamic response of the system under study. The design and performance characteristics of an experimental washer type force transducer were described and it was reported that although experimental transducers fulfilled most of the requirements, additional development was required to reduce sensitivity to moment loading.

Efforts to measure both forces and accelerations experienced by equipment attached to a vehicle have continued since the presentation of the previous paper, but further developmental efforts to develop a satisfactory miniature transducer were discontinued when miniature quartz transducers became commercially available.

The present paper is derived from a Lockheed-sponsored program that had the following objectives:

1. To evaluate the accuracy of miniature quartz force transducers when they are used for the measurement of dynamic forces transmitted between equipment and supporting structure in an airplane;

2. To measure the forces and accelerations experienced by equipment attached to a typical vibrating aircraft structure and to determine the degree of overtesting that would be involved if laboratory vibration tests were based on an envelope of the measured acceleration peaks; and

3. To determine the degree of improvement in service environment simulation that can be realized if the laboratory test is correctly based on a knowledge of both the forces and accelerations that exist in the service environment.

FORCE TRANSDUCER EVALUATION

The initial phase of the program was devoted to an examination of the validity of the force measurements that could be obtained in a typical equipment-supporting structure combination. The force transducers used were obtained from the Kistler Instrument Corp. and are shown in Fig. 1. Although the manufacturer of the transducer supplied a detailed performance specification, this information was considered to be not necessarily applicable to the intended application. One would expect, for instance, to be able to reproduce the output versus applied axial force relation specified by the manufacturer if corroborative testing were carried out under idealized conditions which assured that a uniform axial load was applied. The loading conditions that would be imposed in a typical vehicle are complex combinations of moment, shear and axial loading. Other conditions that might affect transducer performance are supporting structure flexibility and attachment point misalignment.

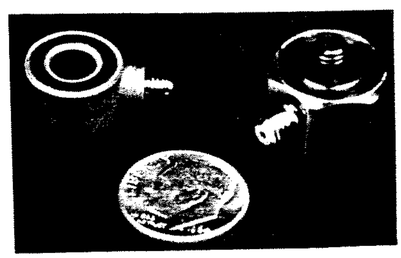


Fig. 1 - Force transducers

To examine the accuracy of the force transducers under loading conditions similar to those just described, the following experimental program was conducted.

A special structure was designed and built to simulate an equipment item. This structure, which will hereinafter be called the "equipment," is shown in Fig. 2. Its basic parts consisted of a base plate and a simply supported beam with a lumped mass in the center. The beam was attached to the base plate at each end through

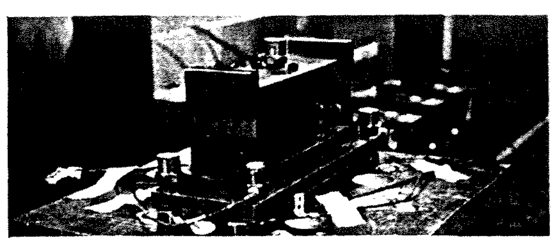


Fig. 2 - Simulated equipment

flexures. A 15-ft long section of an F-104 fuselage was obtained to simulate a typical aircraft structure. An electrodynamic shaker was attached to the fuselage to provide vibratory excitation (Fig. 3). The interior of the fuselage incorporated a number of brackets for the attachment of various pieces of electronic equipment. One of these was chosen as the supporting structure for the simulated equipment mentioned previously. This bracket is shown in Fig. 4. Figure 5 shows the simulated equipment attached to it.

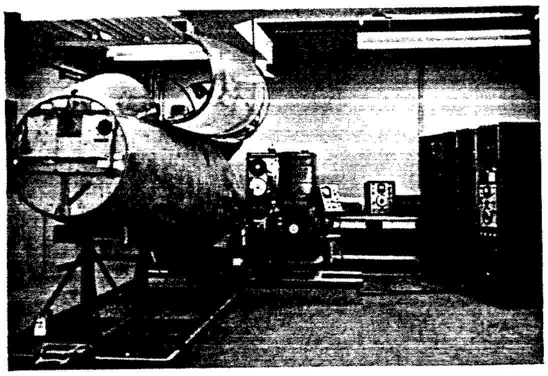


Fig. 3 - F-104 fuselage section with shaker attached

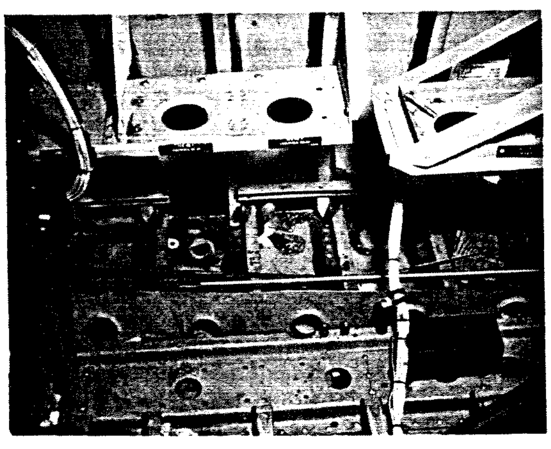


Fig. 4 - Equipment attachment bracket, position 1

Force transducers were placed between the base plate of the equipment and the supporting bracket at each of the four corners. Accelerometers were located at each of the four corners of the base plate, and an additional accelerometer

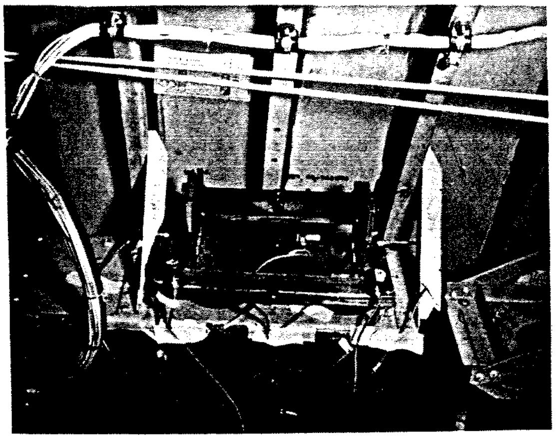


Fig. 5 - Simulated equipment attached to bracket, position 1

was attached on the lumped mass located at the center of the equipment beam.

This arrangement allowed sufficient information to be obtained to allow two independent determinations of the total normal force acting on the base of the equipment. One of these determinations could be made on the basis of the total signal produced by the force transducers. The second could be obtained from the acceleration signals and a knowledge of the equipment masses. Force measurements based on acceleration readings were readily obtained by an analog computer programmed to multiply the incoming acceleration signals by the appropriate constants (representing the base plate and beam masses) and to perform the necessary summing operations.

The force transducer signals were also summed by an operational amplifier. No special care was exercised as to attachment surface condition or bracket flexibility. The conditions were similar to those that would be present in a typical practical application. When the equipment was attached to the supporting bracket, one side of the force transducers pressed against an angle bracket which was only 0.04-in. thick. The surface of the base plate, which was adjacent to the opposite side of the transducers, had a milled finish.

Following the attachment of the instrumented simulated equipment to the support in the vehicle, the fuselage was subjected to a slow sinusoidal sweep vibration excitation and the force and acceleration measurements were plotted versus frequency with an x-y recorder. Typical results obtained are given in Fig. 6. Although both curves in Fig. 6 give the variation

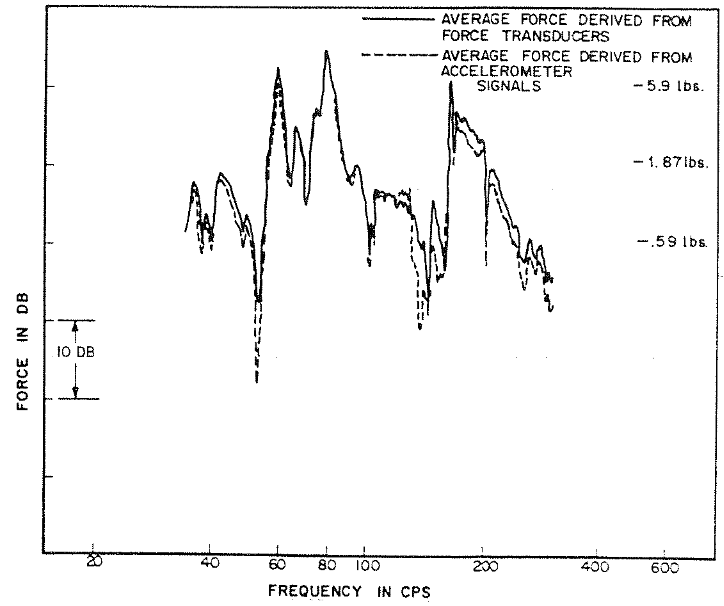


Fig. 6 - Comparison of measured normal force with force computed from acceleration measurements, position 1

of the total normal force (acting on the base plate) versus frequency, one of them was derived from force transducer measurements and the other from acceleration measurements. The agreement between the two curves is considered to be remarkably good and is typical of the correlation between force transducer derived and accelerometer derived data obtained with the simulated equipment attached to the vehicle at various other locations.

PROGRAM PLAN

Once the accuracy of the force transducers had been evaluated, attention was directed to an examination of the benefits force measurement could provide in the formulation of improved laboratory environmental test procedures.

The investigation involved the following steps:

1. The fuselage was subjected to a sinusoidal sweep excitation with the equipment located at various positions. (Details of the attachment locations will be described later.)

2. At each position, the force and acceleration signals were recorded as described above. Of particular interest were (a) the acceleration of equipment base plate based on an average of the four accelerometer readings, (b) the normal force acting on the base plate derived from the sum of the four force transducer signals, and (c) the acceleration of the mass attached to the center of the equipment beam.

3. Hypothetical laboratory vibration test specifications were established for each of the equipment locations. In each case, the test selected was a sinusoidal sweep having a constant acceleration amplitude. The acceleration level chosen was based on the equipment support response in the vehicle and was equal to the value of the major peak base plate response that followed or preceded the antiresonance introduced by the "tuned damper" action of the beam. This procedure is considered to be representative of the practice often followed in generating test specifications.

4. The equipment was then removed from the vehicle structure, placed on a shake table and subjected to the excitation levels discussed in step 3. The equipment was instrumented, as before, with four force transducers located between the equipment base plate and the table and with five accelerometers placed in the positions already described.

5. The acceleration of the mass on the equipment beam was chosen as a measure of the damage potential to which the equipment was subjected. Since the accelerations experienced by the same mass when the equipment

was attached to the vehicle structure were known, it was possible to establish a ratio between the relative damage potential experienced by the equipment on the shaker to that encountered in the vehicle. The ratio of damage potential was named the "Overtest Index." Two different overtest indexes, designated as Type 1 and Type 2, were derived and defined as follows: (a) Type 1 overtest index, $(O.I.)_1$, is the ratio of the maximum acceleration that the resonant element (beam) experienced during the shake table test to the maximum acceleration that it received in the airplane irrespective of frequency; and (b) Type 2 overtest index, $(O.I.)_2$, is the ratio of the maximum acceleration that the resonant element received during the shake table test to that which it received in the vehicle when the exciting frequency was equal to the natural frequency of the resonant element.

6. After completion of the tests during which the shake table was maintained at a constant acceleration amplitude over the frequency range of interest, a second set of tests was conducted. The table acceleration was again maintained at the same amplitude as before except for a narrow frequency band centered at the natural frequency of the equipment beam. In this frequency region, the beam introduced a notch in the table acceleration spectrum. Instead of increasing the command signal to maintain a constant table acceleration amplitude (as was done in the previous tests), a change was made from acceleration control to force control. The input signal to the shaker was adjusted to produce a transmitted force to the base plate which was equal to the maximum force measured (irrespective of frequency) when the equipment was attached to the vehicle structure.

7. Comparisons were made of the damage potentials associated with the two test procedures described in steps 4 and 6.

EQUIPMENT AND VEHICLE CHARACTERISTICS

The simulated equipment used in this investigation has been described in the first section and can be idealized as the vibration system shown in Fig. 7. The beam attached to the base plate behaved as a tuned damper when the exciting frequency was equal to the natural frequency of the beam. Although the base plate itself could be considered as a lumped mass over the frequency range of interest, the mechanical impedance of the composite base plate-supporting structure system was springlike at low frequencies and masslike at sufficiently higher frequencies.

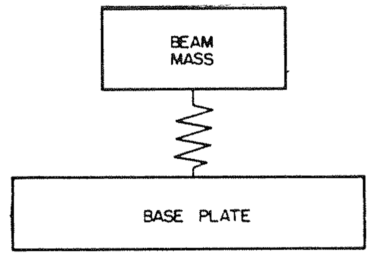


Fig. 7 - Vibration system represented by simulated equipment and supporting structure

The data reported herein were obtained with the equipment located at four different positions on the fuselage. Position 1 was a lightweight bracket on the fuselage interior normally used to support a standard piece of flight hardware. This position was the same as used in the force transducer evaluation program described in the first section. Positions 2 and 3 were on the ring structure of the fuselage. These locations, shown in Fig. 8, were chosen on the fuselage exterior for convenience. Position 4, also on the fuselage exterior, was immediately over the intersection of a highly rigid bulkhead and the fuselage skin. The support at position 1 was by far the most flexible of the four selected. Position 4 was the most rigid and position 2, which was located adjacent to position 4, presented the second most rigid support.

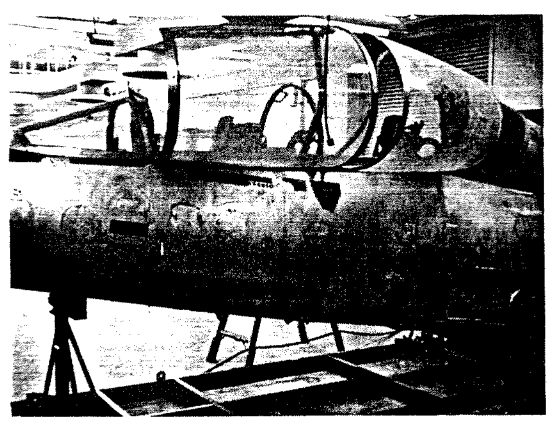


Fig. 8 - Equipment attachment location on fuselage exterior

The results reported here were all obtained with a beam mass of 1.7 lb and a beam natural frequency of 137 cps. The weight of the base plate was approximately 5 lb. Tests were also conducted using other values of mass and natural frequency. The results from these additional tests do not lead to any modifications of the conclusions reached and, for the purpose of brevity, will not be reported in this paper.

Irrespective of the supporting structure impedance, the beam could be expected to introduce a notch in the base plate acceleration spectrum as the exciting frequency was swept through 137 cps (the natural frequency of the beam). Furthermore, the base plate could be expected to experience peak responses on each side of the notch frequency. Before presenting test results, it will be useful to make certain observations concerning the behavior of an idealized system consisting of a primary simple resonator to which is attached a mass-spring system that can behave as a dynamic damper. Although the support locations cannot be considered to behave as simple resonators over a broad frequency range, the idealization selected is probably valid within the frequency region of interest.

The support can be expected to experience response peaks at frequencies preceding and following the antiresonance (notch) response. The proximity of these peaks to the notch frequency can provide considerable insight into the impedance of the support. If the preceding peak is much closer to the notch than the following peak, it can be concluded that the support is springlike within the frequency range of interest. A masslike support impedance is indicated when the peak support response that follows the notch is closer to it than the preceding peak. If the preceding and following support peak responses are symmetrically located on either side of the notch, the support impedance is damperlike. All other factors remaining constant, the frequency range between the support response peaks becomes greater as the ratio of the tuned damper mass to the effective supporting structure mass is increased.

The observations just cited can be demonstrated analytically, and the reader is referred to the literature for further details [2]. In the present instance these considerations are useful in providing a qualitative evaluation of the beam response at the various attachment locations.

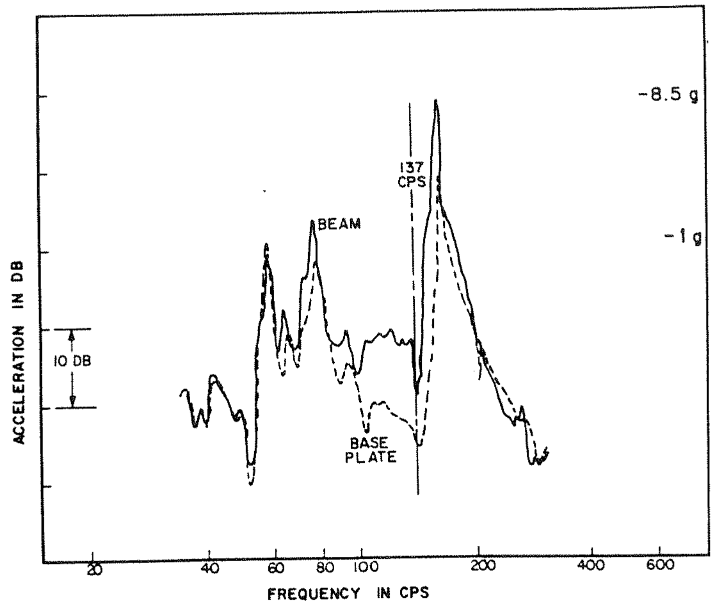
In this regard, attention is called to Figs. 9 through 12 which show the variation of beam and base plate accelerations with frequency at attachment positions 1, 2, 3 and 4, respectively. Position 1, which was considered to be the most typical equipment support, provided a masslike mechanical impedance to the beam system in the vicinity of 137 cps. Here the mass was essentially the 5-lb base plate since the bracket was of very light weight. It can be seen (Fig. 9) that both the beam and base plate peak responses occur approximately 45 cps above the notch frequency. The structure at positions 2 and 3 (Figs. 10 and 11) was springlike in the vicinity of 137 cps as is indicated by the sharp decline in the base plate response as the exciting frequency rose to 137 cps. Position No. 4 (Fig. 12) is seen to have also provided a springlike support impedance.

OVERTEST LEVEL CRITERIA

The problem of defining the relative severity of a laboratory vibration test and of the service environment is difficult. For instance, one does not, in general, know if the damage that the equipment may experience is due to excessive stress, displacement, velocity or acceleration. In the present instance we have rather arbitrarily assumed that the acceleration of an equipment resonant structure provides a measure of the damage potential. The simulated equipment used in the experimental program incorporated such a resonant structure in the beam and the acceleration of the beam was assumed to provide the required measurement of test severity.

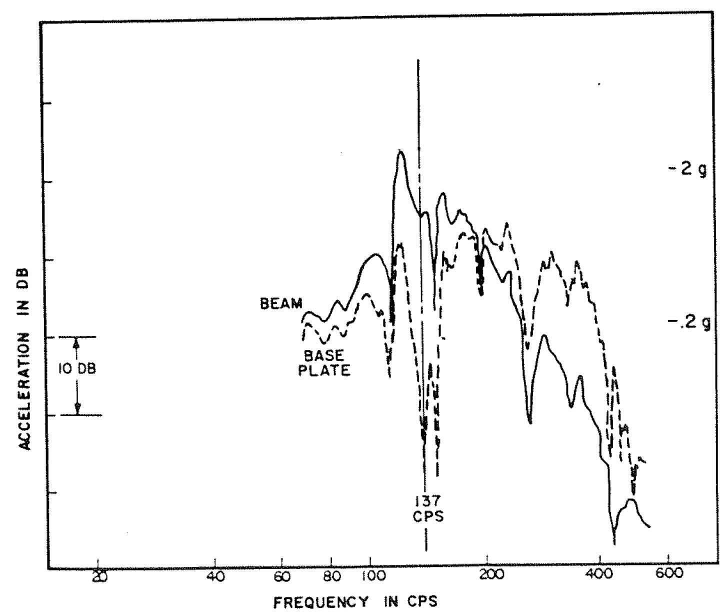
It was also necessary to establish a method for choosing the acceleration input level that would be called out in a hypothetical test specification requiring a constant amplitude of acceleration sinusoidal sweep. Since attention was confined to the frequency region that involved the resonant frequency of the equipment and the adjacent peak responses of the supporting structure, it was decided to use the higher of the two peak base plate (support) responses as the input level to be imposed when the equipment was placed on the shake table.

Finally, it was also necessary to provide a basis for quantifying the relative severity of the acceleration experienced by the equipment beam when the equipment was attached directly to a shake table with that imposed by the vehicle. There were obviously an infinite number of frequencies at which the acceleration levels differed but there were two frequencies that were considered to be of particular importance. One of these was the natural frequency of the equipment beam. At this frequency, the beam experienced the peak acceleration when the equipment was vibrated on the shake table. With the equipment attached to the vehicle, the base plate



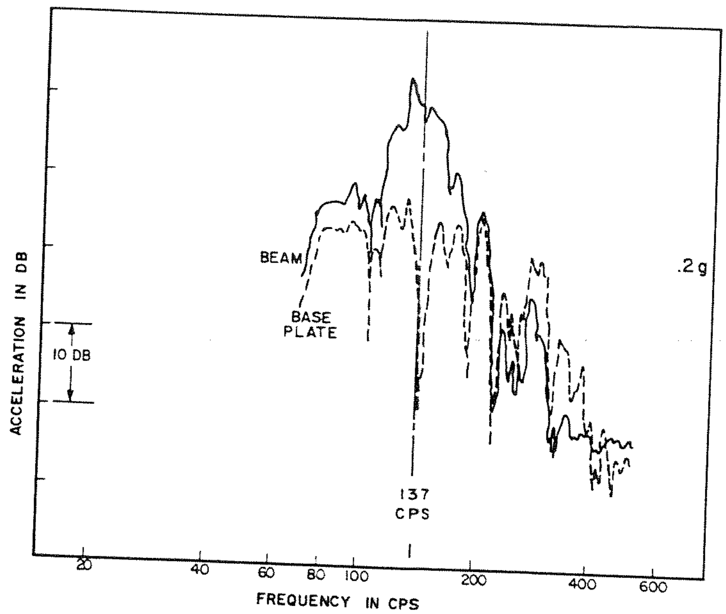
masslike mechanical impedance

Fig. 9 - Comparison of base plate and beam accelerations, position 1



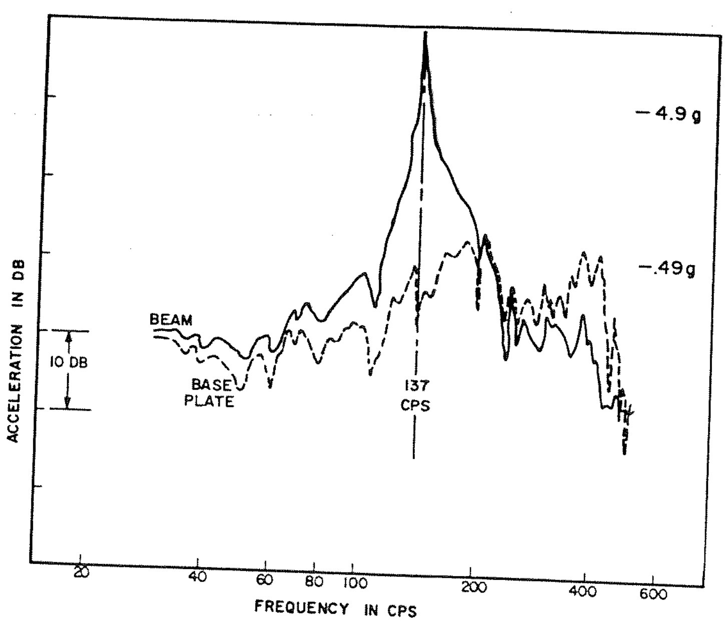
Springlike mechanical impedance

Fig. 10 - Comparison of base plate and beam accelerations, position 2



spring like mechanical impedance

Fig. 11 - Comparison of base plate and beam accelerations, position 3



spring like mechanical impedance

Fig. 12 - Comparison of base plate and beam accelerations, position 4

support assembly experienced an antiresonance notch at the same frequency. The second frequency of interest was associated with the peak response of the base plate-support assembly when the equipment was attached to the vehicle. This was also the frequency at which the beam experienced its maximum acceleration (in the vehicle).

For the shake table tests it was logical to choose the resonant acceleration of the beam as the measurement of test severity. For the simulated service environment tests (equipment attached to fuselage), two values of severity were selected. One of these was the maximum beam acceleration measured, irrespective of frequency. The second was the beam acceleration occurring at the resonant frequency of the beam.

The above measurements allowed two ratios, called "overtest indexes," to be established. A precise definition of these quantities has already been given in step 5 of the section on the Program Plan.

OVERTEST LEVELS

Comparisons of beam accelerations measured with the equipment attached to the vehicle and those produced on the shake table at constant input acceleration amplitude are provided in Figs. 13 through 16. Overtest indexes for the various attachment locations are indicated on the figures.

The greatest overtest levels were associated with position 1 where the Type 1 and 2 indexes were found to be 21 and 1400, respectively. At position 3 (this support was next to position 1 in rigidity) index levels of 10 and 20 were obtained. At the most rigid locations (positions 3 and 4), overtest index pairs of 3, 6 and 2, 3 were measured.

In assessing the significance of these results, it is of interest to note that the highest levels of overtesting were associated with the most typical equipment support structures used. Positions 2 and 4 were highly rigid locations that would rarely be approximated in practice. Furthermore, it is important to remember that the effective mass of the beam was only 1.7 lb. Larger values of mass in the resonating structure could be expected to produce higher overtest levels.

A method for obtaining an approximation of the overtest indexes based on the peak-notch characteristics of the supporting structure and the Q of the resonant element is given in the Appendix.

VIBRATION TESTING WITH FORCE CONTROL

Following completion of the shake table tests at constant amplitude, the merits of a force-controlled test were examined.

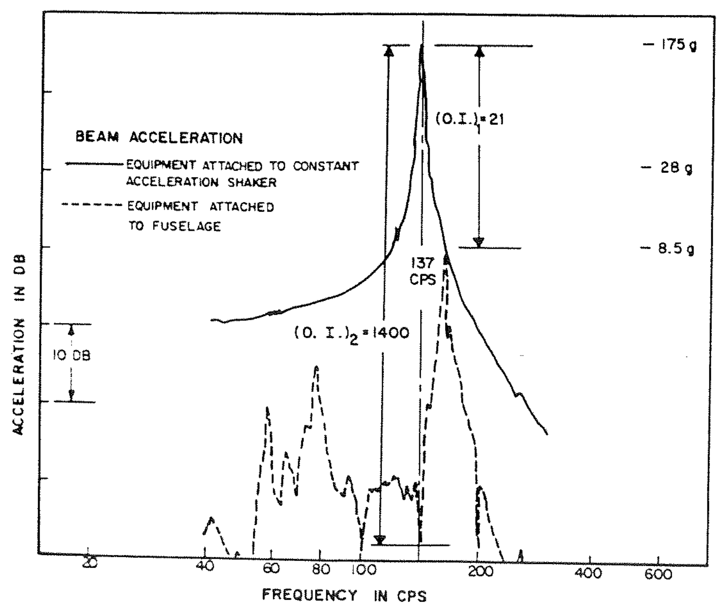


Fig. 13 - Overtest indexes associated with position 1

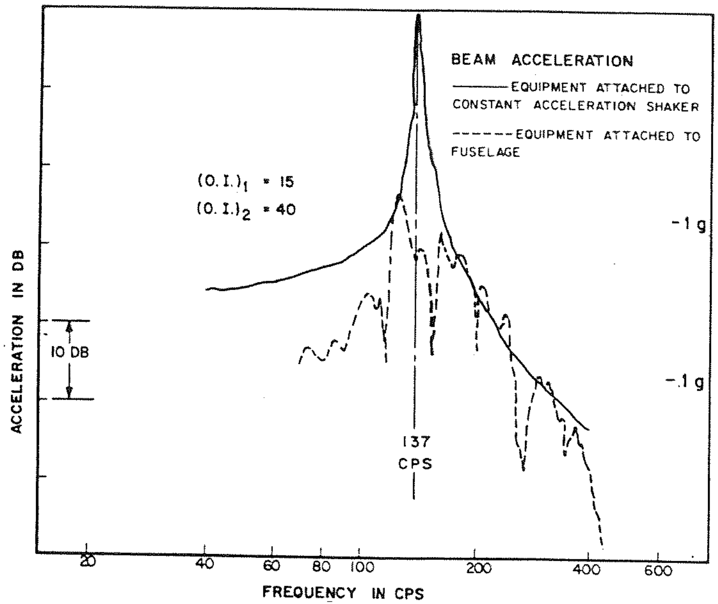


Fig. 14 - Overtest indexes associated with position 2

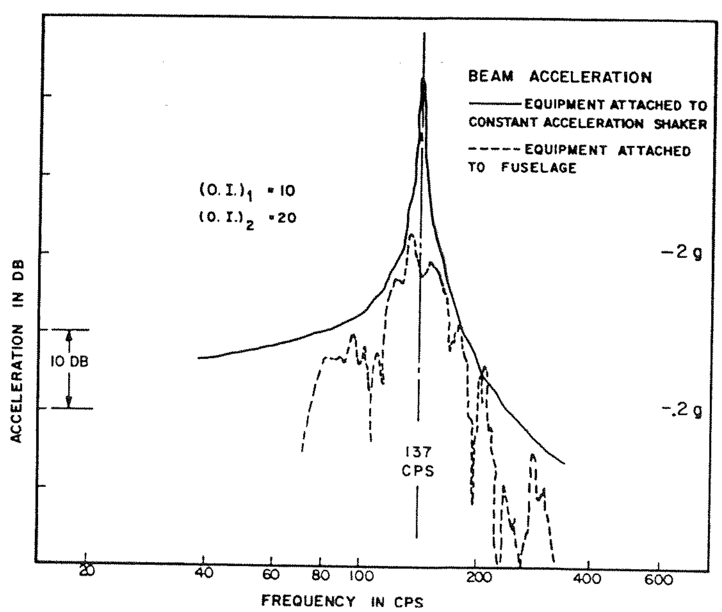


Fig. 15 - Overtest indexes associated with position 3

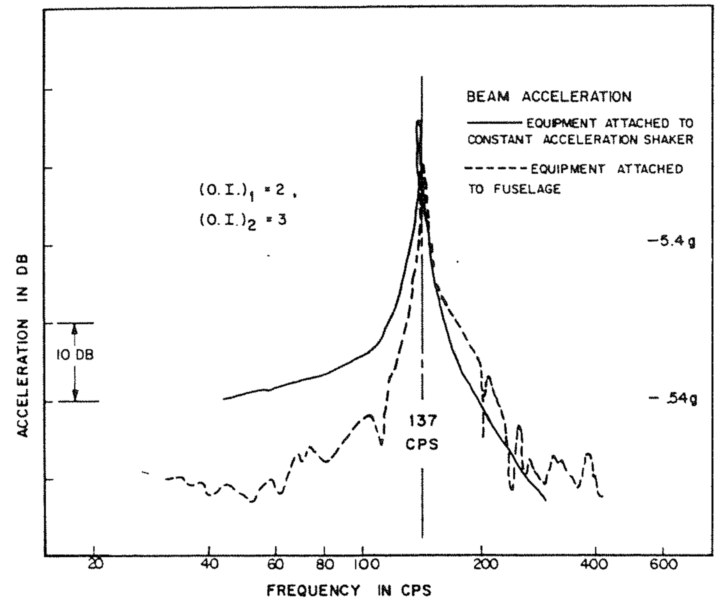


Fig. 16 - Overtest indexes associated with position 4

From the preceding discussion, it is evident that the principal difficulty encountered in constant acceleration testing arises at major equipment resonances. At an equipment resonance, an antiresonance occurs in the table spectrum unless the input signal amplitude is increased to compensate for this effect. In the vehicle, the antiresonance effect always produces a notch in the support acceleration spectrum. It follows that the force transmitted by the support in the vehicle to the equipment will be less than would be transmitted by the shake table.

If the force transmitted by the support in the vehicle is known and if the force transmitted by the shake table to the equipment base can be measured, it is a simple matter to adjust the shaker acceleration in the vicinity of an equipment resonance so as to provide an approximate simulation of the dynamic loads applied in the service environment. In general, maximum dynamic loads that the equipment will experience in the vehicle will not occur at an antiresonance. (If they did occur at an antiresonance, the two overtest indexes would be equal.) To assure that the laboratory shake table test is conservative, the force transmitted by the table at an equipment resonance should not fall below the maximum force measured in the vehicle, irrespective of frequency.

The force-controlled tests, which followed completion of the constant acceleration amplitude tests, were based on the considerations cited above. With one important exception, these tests were similar to the constant acceleration tests. Except at a frequency band in the vicinity of the beam resonance, the table acceleration was maintained at the same constant value as before. As the beam resonance began to occur, a change was made from acceleration control to force control and the table acceleration was reduced to provide a transmitted force that approximated the maximum value measured in the vehicle. As the exciting frequency was further increased and the resonance subsided, a return was made to acceleration control.

The procedure just described could be expected to reduce the Type 1 overtest index to unity. The Type 2 overtest index would be reduced by an amount equal to the change in the Type 1 index. A typical result obtained during a force-controlled test is shown in Fig. 17. Shown here are beam accelerations as measured: (a) in the vehicle with the equipment attached at position 1, (b) during a constant acceleration amplitude shake table test, and (c) during a force-controlled test. The force-controlled test is seen to be conservative but the overtest level at the beam resonant frequency was reduced by a factor of 20.

Figure 18 shows the variation of the transmitted force as measured: (a) with the equipment attached to the vehicle at position 1, and (b) during the force-controlled test. These results provide additional evidence that the force-controlled test was conservative.

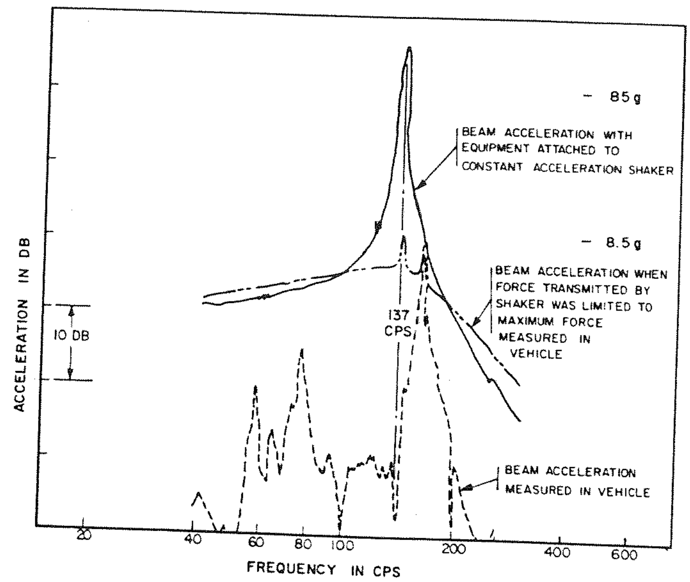


Fig. 17 - Comparison of overtest levels introduced with: (a) shake table acceleration held constant, and (b) force limiting employed at equipment resonant frequency

position 1

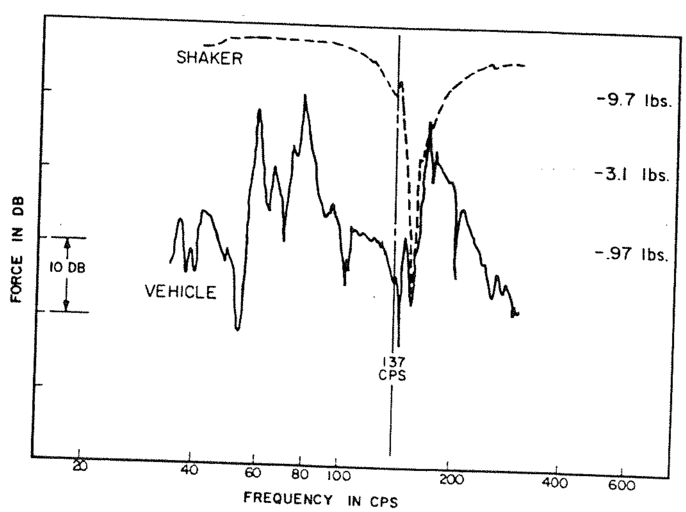


Fig. 18 - Comparison of transmitted normal force measured in vehicle (position 1) with force applied by shake table when table spectrum was notched to limit force at resonant frequency of equipment beam

CONCLUDING REMARKS

This paper has presented a portion of the data obtained during a program designed to examine the feasibility of obtaining and using transmitted force information to provide laboratory vibration tests that are not overly conservative. Due to time and budget limitations, the investigation was confined to sinusoidal rather than random excitation. The reader should be cautious in drawing any conclusions as to what the overtest levels would have been had the excitation been random. It is the author's opinion that the results obtained with random excitation would not have been greatly different if all measurements of power spectral density were obtained with a very narrow-band filter and if narrow-band shake table equalization were employed.

REFERENCES

1. G. W. Painter, "Use of Miniature Force Transducers in the Measurement of Shock and Vibration Environments," Shock and Vibration Bull. No. 34, Part 4, pp. 45-53, Feb. 1965

2. C. H. Powell, "Graphical Treatment of Vibration and Aircraft Engine Dampers," Bookcraft, N.Y.

Appendix

For the idealized system shown in Fig. A-1, it is possible to approximate the Type 1 and Type 2 overtest indexes on the basis of the peak-notch behavior of the support and the Q of the resonant element.

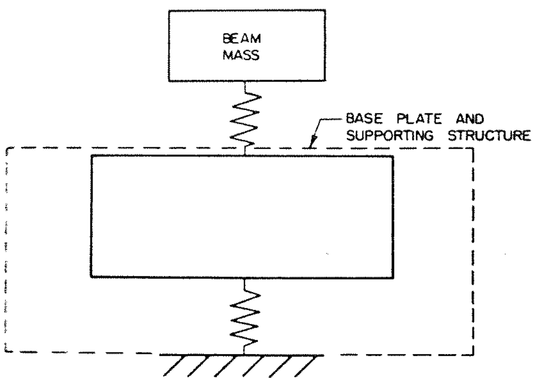


Fig. A-1 - Idealization of equipment vehicle vibration system

The Type 1 overtest index, $(O.I.)_1$, can be determined from

$$(O.I.)_1 = \frac{Q_2}{\left[\frac{\ddot{X}_2}{\ddot{X}_1}\right]_{\omega_p}}, \qquad \text{(A-1)}$$

where

Q_2 = quality factor of the resonant element, and

$[\ddot{X}_2/\ddot{X}_1]_{\omega_p}$ = ratio, $\ddot{X}_2/\ddot{X}_1$, at the frequency (ω_p), where the peak support response occurs.

$[\ddot{X}_2/\ddot{X}_1]_{\omega_p}$ can be determined from the well-known single-degree-of-freedom transmissibility equation, or

$$\left[\frac{\ddot{X}_2}{\ddot{X}_1}\right]_{\omega_p} = \sqrt{\frac{1 + \left(\frac{\omega_p}{\omega_n Q_2}\right)^2}{\left[1 - \left(\frac{\omega_p}{\omega_n}\right)^2\right]^2 + \left[\frac{\omega_p}{\omega_n Q_2}\right]^2}}, \qquad \text{(A-2)}$$

where ω_n is the natural frequency of the resonant element.

When

$$\frac{\omega_p}{\omega_n Q} << \left|1 - \left(\frac{\omega_p}{\omega_n}\right)^2\right| \quad \text{and} \quad \frac{\omega_p}{\omega_n Q_2} << 1,$$

$$\left[\frac{\ddot{X}_2}{\ddot{X}_1}\right]_{\omega_p} \approx \frac{1}{1 - \left(\frac{\omega_p}{\omega_n}\right)^2} \qquad \text{(A-3)}$$

and, therefore,

$$[O.I.]_1 \approx Q_2 \left[1 - \left(\frac{\omega_p}{\omega_n}\right)^2\right]. \qquad \text{(A-4)}$$

The Type 2 overtest is simply equal to the depth of the support (X_1) notch as measured from the maximum adjacent support response peak.

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