On Panel Vibration Damping Due to Structural Joints

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Experimental results are summarized which show what effects various panel, joint, and ambient air parameters have on the damping of panel vibrations which is provided by joints connecting panels to stiffeners or other panels. From these results it is concluded that the dissipation of vibratory energy at structurally acceptable multipoint-fastened (riveted, bolted, or spot-welded) joints, at frequencies considerably higher than the panel fundamental, is primarily caused by the "pumping" of air produced as adjacent surfaces between fasteners move away from and toward each other. A method for estimating the damping of built-up panel structures is suggested, based on the concept of the absorption coefficient of a panel discontinuity and on empirical correlations of experimental data.

Introduction

THE capability of a structure to dissipate vibratory energy plays an important role in establishing the levels of the structure's responses to excitations, such as rocket noise, the spectra of which extend over wide frequency bands. The aerospace structural analyst requires means for obtaining realistic values of the magnitude of this energy dissipation capability or "damping" in order to be able to provide meaningful response estimates. In addition, analysts and designers desire to understand the mechanisms responsible for this damping. They need to know which parameters are important, what results may be produced by a given design change, and how to design a structural configuration that combines favorable damping characteristics with low cost and weight.

Much useful information is available concerning the damping properties of homogeneous structures,[1, 2] the design of highly damped structures incorporating viscoelastic materials,[3, 4] and energy dissipation associated with relative motion at some simple structural interfaces.[5, 6] On the other hand, the open technical literature contains little information on the damping of built-up structures (such as aircraft fuselages, which typically consist of a multitude of panels and reinforcing members joined together by various means), particularly for frequencies above the fundamental resonances of the substructural panels. The present paper is intended to provide a step toward filling this void by identifying the dominant damping mechanism and by suggesting practical means for estimating the damping of built-up structures.

The often observed fact that built-up structures tend to be much more highly damped than similar one-piece structures indicates that the damping of built-up structures is primarily caused by the structural joints. Accordingly, this paper deals with the energy dissipation produced at such joints, and with how the action of joints affects the damping of substructural panels.

In order to provide practically useful results, the work presented here focuses on structurally acceptable (tight, nonslipping) joint configurations and on vibrations at amplitudes that are small enough so that the observed damping is independent of amplitude. This restriction on amplitude is not a significant one. For all configurations studied, the damping was found to remain essentially amplitude-independent up to quite large amplitudes, which one would in general want to avoid by proper design. In addition, since damping increases at higher amplitudes, designs based on the lower amplitude values presented here would tend to be conservative.

In order to determine which parameters are important, and thus to identify the dominant energy dissipation mechanism, a series of experiments was undertaken in which some of the panel and joint parameters were varied systematically. This study and its results are presented in the first of the following sections. Thereafter, additional experimental results are summarized and interpreted, and a panel damping estimation technique is suggested on the basis of these findings.

Identification of the Dominant Damping Mechanism

Experimental Arrangement and Technique

All of the measurement results reported here were obtained by use of essentially the same experimental set-up, instrumentation system, and procedures. The test structure in each case consisted of a somewhat irregularly shaped plate, with beams attached at one or more positions, as sketched in Fig. 1. The plate was suspended by means of two long strings attached near two plate corners, as indicated in the figure. The irregular shape was chosen in order to facilitate the establishment of a relatively uniform diffuse wave field.[7] A small helical loudspeaker "voice coil" cemented to the plate and located in the gap of a permanent magnet was used

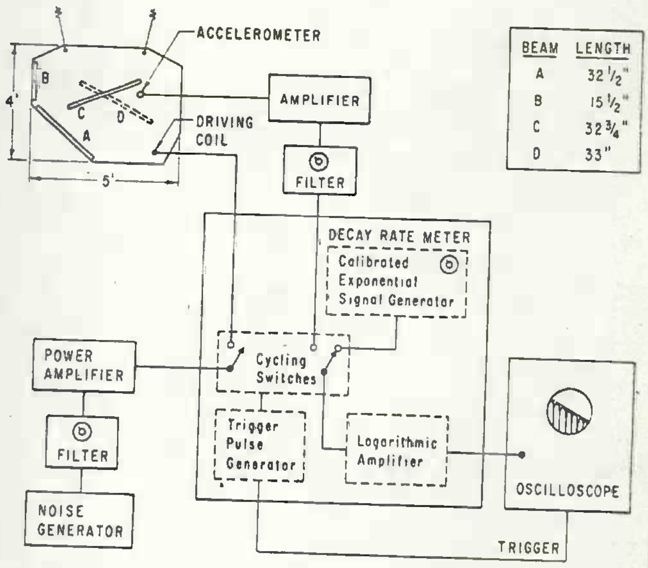

Fig. 1 Test plate arrangement and instrumentation system.

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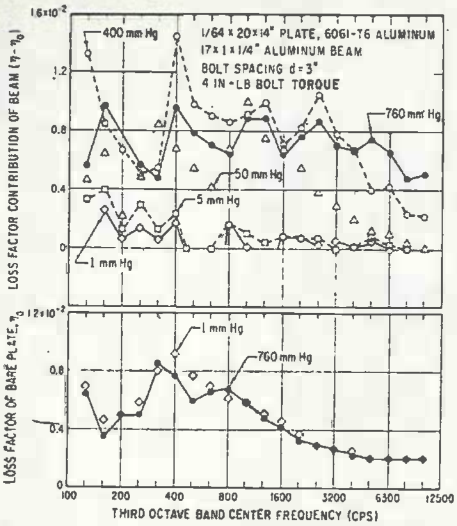

Fig. 2 Effect of reduced atmospheric pressure on contribution to panel damping made by attached beams.

to excite the plate. The plate motions were sensed by a small piezoelectric accelerometer bolted to the plate in one of several locations. Care was exercised so that none of the leads interfered with the plate motions.

Damping was measured by observing the rate of decay of free vibrations.[8] The test plate was excited with a signal obtained by passing white noise through a one-third-octave filter; then the excitation was suddenly turned off, and the rate of decay of the accelerometer signal, filtered in the same band as the excitation, was observed. For each measurement the excitation was applied long enough to permit the plate vibrations practically to attain their steady state, and during each decay observation the voice coil circuit was kept open so that it would contribute no energy dissipation.

The instrumentation system is represented schematically in Fig. 1. Rate-of-decay measurements were made with the aid of a "decay-rate meter" (Spencer-Kennedy Laboratories, Model 507) which repetitively presents on an oscilloscope alternately the logarithm of a decaying accelerometer output and that of an adjustable known signal; by matching the slope of the known signal to that of the envelope of the accelerometer signal, one may read the reverberation time T_{60} directly from the instrument. This time is defined as the interval within which the signal power decays by a factor 10^6 (i.e., by 60 db, corresponding to an amplitude decay by a factor of 10^3), and can readily be shown to be related to the structural loss factor‡ η as

$$\eta \approx 2.2[T_{60}(\text{sec})\,f(\text{cps})]^{-1} \qquad (1)$$

where f denotes the center frequency of the band in which measurements are being taken.

If the envelope of the oscilloscope trace of the logarithm of the decaying acceleration signal (displayed as a function of time) is a straight line, then the corresponding reverberation time and decay rate are amplitude-independent. This straight-line trace is obtained for linear systems; for nonlinear systems the aforementioned trace generally appears curved. Thus, the instrumentation system used here has two advantages, in addition to convenience: it permits one to judge the linearity of the test structure, and the system's repetitive manner of operation permits one to base each measurement on several decay observations.

‡ The loss factor is a commonly employed dimensionless measure of structural damping. It is usually defined for a structure vibrating in steady state, as the ratio between the energy dissipated per cycle and 2π times the (time-wise) maximum total strain energy stored in the structure. For damping that is not too high, say $\eta < 0.2$, (a condition that is met in nearly all practical structures) $\eta \approx 2c/c_c$, where c denotes an equivalent viscous damping coefficient for the structure and frequency of interest, and c_c represents the corresponding critical viscous damping coefficient.[1,8]

Importance of Normal Relative Motion between Beam and Plate Surfaces

A series of loss factor measurements was carried out on $\frac{1}{16}$- and $\frac{1}{32}$-in.-thick aluminum plates, with beams attached by means of bolts. A wide range of bolt-tightening torques was used, ranging from the lowest torque that would result in a tight joint to the greatest torque the bolts could sustain.[9] The measured loss factors were found to be independent of bolt-tightening torque, within the precision of the experiment. This finding agrees with the results of an earlier study,[10] in which damping measurements were performed on samples of aircraft fuselage structures consisting of beams (stringers) with attached sections of skin. In this earlier study it was noted also that similar beam-plus-skin samples provided essentially the same damping, regardless of whether the skin was fastened to the beam by rivets, spot-welds, or well-tightened bolts.

A further series of experiments indicated that the finish of the beam surface in contact with the test plates did not affect the damping significantly. Contrary to what one would expect if interface friction were the dominant mechanism, the damping observed with a very smooth (polished) surface was only very slightly greater than that obtained with a coarsely knurled surface. Also, steel beams were found to produce the same amount of damping as aluminum beams, and changes in beam cross section (or beam flexural rigidity) were found to have no measurable effect, provided that the width of the beam face in contact with the plate remained unchanged and that the beam remained considerably stiffer than a plate strip of the same width. Beams with wider contact faces were found to result in greater damping; it was noted that the loss factor increase obtained by attaching a beam to a test plate was roughly proportional to the aforementioned face width.

As a result of a number of exploratory experiments it was observed that reduced damping resulted from any strategem that restricts (in regions between the bolts or other connectors) the relative motion between the adjacent beam and plate surfaces in the direction normal to these surfaces. For example, reduced damping resulted when washers were clamped between the beam and plate, when drops of a relatively rigid adhesive were placed there (half-way between adjacent connectors), or when additional beams were added so that everywhere two beams were back-to-back, on opposite sides of the plate. Similarly, it was found that continuously welded plate seams and beams welded along their entire lengths contributed no measurable damping.

In experiments in which the beams were mounted so that adjacent beam and plate surfaces were not in direct contact (as, for example, when washers were placed on the bolts, between the beams and plates) the measured damping of the plate with beams attached was almost exactly the same as that of the plate without beams; i.e., the beams again made no detectable damping contribution.

Effect of Ambient Air

An exploration of the effect of interface lubricants[9, 10] revealed that damping decreases as lubricant viscosity increases; dry (i.e., air-lubricated) joints were found to behave at high frequencies as if an oil with a viscosity between 10 and 100 centistokes were present.§

§ As subsequently discussed, significant normal relative motions occur only at frequencies at which the bolt-spacing/plate-wavelength ratio exceeds $\frac{1}{2}$. The kinematic viscosity of air at room temperature and standard atmospheric pressure is about 20 centistokes.

An additional brief series of measurements carried out in a vacuum chamber indicated that the added damping caused by attached beams depends very markedly on the ambient atmospheric pressure. Some of the results of these experiments (performed on a smaller plate than that sketched in Fig. 1, in view of space limitations in the available vacuum chamber) are indicated in Fig. 2. The lower portion of this figure shows that the damping of the plate in absence of attached beams is unaffected by the change in ambient air pressure from one atmosphere (760 mm Hg) to 1 mm Hg. The upper portion of this figure indicates that such a reduction of atmospheric pressure results in reducing the added plate damping provided by an attached beam from a quite significant value to essentially zero.

Although some aspects of the data shown in Fig. 2 (e.g., the generally greater damping contribution at 400 mm Hg than at 760 mm Hg) remain to be explained, the data summarized in this figure do suggest that the ambient air plays an important role in the mechanism that dominates the dissipation of the vibratory energy of the plate because of attached beams.

Dominant Mechanism

The foregoing observations lead one to conclude that the dominant damping mechanism in the plate-plus-beam configurations studied here is not associated with interface slip or friction. Calculations based on expressions derived by Maidanik[11] (and also experimentally verified by him to some extent) indicate that the observed panel damping increase caused by the attached beams cannot be ascribed to the beam's increasing the efficiency with which acoustic energy is radiated from a plate to the ambient air; the observed damping increase considerably exceeds that ascribable to increased radiation efficiency and has a grossly different frequency dependence. Thus, the damping here appears to be primarily caused by a pumping of air, produced by relative motions akin to slapping of the plate surfaces against adjacent beam surfaces.

Estimation of Panel Damping

Absorption Coefficients of Discontinuities

The extremely useful concept of an absorption coefficient of a beam (or of some other linear discontinuity, such as a plate edge or seam) on a plate was introduced by Heckl,[7] in direct analogy to the corresponding quantity used in the study of the acoustics of enclosed spaces. The absorption coefficient of a linear discontinuity is defined as the fraction of the plate bending wave energy impinging on the discontinuity which is dissipated at the discontinuity.¶ Such an absorption coefficient is, of course, intimately related to the damping of the plate because of the discontinuity.

By considering the flexural waves travelling along the plate to be uniformly distributed in direction (i.e., to constitute a "diffuse" field), and by accounting for the mean free path of such waves and for the energy carried by them, Heckl showed that the absorption coefficient γ of a discontinuity on a plate obeys a relation that may be restated as

$$\gamma = \pi^2(\eta - \eta_0)S/L\lambda \qquad (2)$$

Here η_0 denotes the loss factor of the plate being considered, in absence of the discontinuity under study, and η denotes the loss factor of the plate in presence of the discontinuity. The difference $(\eta - \eta_0)$ therefore represents the loss factor contribution made by the discontinuity. The symbol S represents the plate surface area (one side), and λ denotes the average wavelength of the plate flexural waves in the frequency band under consideration. For a uniform plate of thickness h, this wavelength obeys

$$\lambda^2 = \pi[3(1 - \nu^2)]^{1/2}(h\, c_L/f) \qquad (3)$$

where ν and c_L denote, respectively, Poisson's ratio and the longitudinal wave velocity in the plate material.**

The symbol L introduced in Eq. (2) denotes the "effective length" of the discontinuity; i.e., the length on which plate waves can impinge. Since such waves can impinge only on one side of beams or other discontinuities located at a plate edge, the effective length L of discontinuities at plate edges is the same as the actual length of the discontinuities. But, since plate flexural waves can impinge on both sides of discontinuities that are located several wavelengths away from the plate edges (e.g., the beams indicated in the middle of the plate of Fig. 1), L is twice the actual length for such discontinuities.

Clearly, if one knows the absorption coefficient of a given type of discontinuity (at all frequencies of interest), as well as the loss factor of a given panel in absence of dissipative discontinuities, then one can determine the damping of the panel in presence of the discontinuity under consideration. If the absorption coefficient of a discontinuity would depend on the discontinuity length and location, then a considerable amount of data would be required before one could use Eq. (2) to estimate panel damping. But, Heckl's work[7] and the present study[9] have shown that absorption coefficients practically are independent of discontinuity length and position on the plate, provided one restricts oneself to frequencies that are high enough so that the plate flexural wavelengths are considerably shorter than both a characteristic plate surface dimension and the discontinuity length. These short wavelength restrictions are imposed by the requirement of a diffuse wave field on the plate, and in order to avoid the generation of significant wave diffraction effects near the ends of the discontinuities.

Dependence on Bolt-Spacing/Plate-Wavelength Ratio

Heckl[7] observed that the curves representing the variations of his measured absorption coefficients with frequency generally exhibited peaks at frequencies at which the spacing between bolts is an integral multiple of the plate flexural half-wavelength. This dependence of the damping on the bolt-spacing/wavelength ratio was investigated further in the currently reported study, which supplemented Heckl's work by providing experimental data on plate-and-beam configurations in which the bolt-spacing, the plate-thickness, and the plate material were varied systematically.

Figure 3 shows a set of typical experimental results obtained with three different bolt-spacings on a given plate-plus-beam configuration. For the sake of clarity and in order to accent the observed peaks, the data in this figure are presented in terms of the loss factor contribution $(\eta\text{-}\eta_0)$ of the attached beams, rather than in terms of the absorption coefficient. (The Fig. 3 data corresponding to the 6-in. bolt-spacing also are shown in Fig. 4 in terms of absorption coefficients, for illustrative purposes.)

The loss factor contribution curves of Fig. 3 are seen to exhibit marked peaks at bolt-spacing/wavelength ratios $d/\lambda \approx \frac{1}{2}$, 1, and minor peaks at $d/\lambda \approx \frac{3}{2}$, 2. The reasons for the relative magnitudes of the peaks on a given curve are not yet fully known, but there is some indication (Appendix III, Ref. 9) that these magnitudes are correlated with the magnitudes of the displacements (relative to the beam) of the plate portions between the bolts. Figure 3 also shows that some of the damping data for different bolt-spacings fall more

¶ In some applications one finds it useful to define an absorption coefficient that accounts for the energy transmitted past a discontinuity, in addition to the energy dissipated by it. However, in the present paper the definition used is the one involving only the dissipated energy.

** For most structural metals $\nu \approx 0.3$, and the coefficient of (hc_L/f) in Eq. (3) takes on the value 1.9. For steel, aluminum, titanium, and magnesium, $c_L \approx 2 \times 10^5$ in./sec.

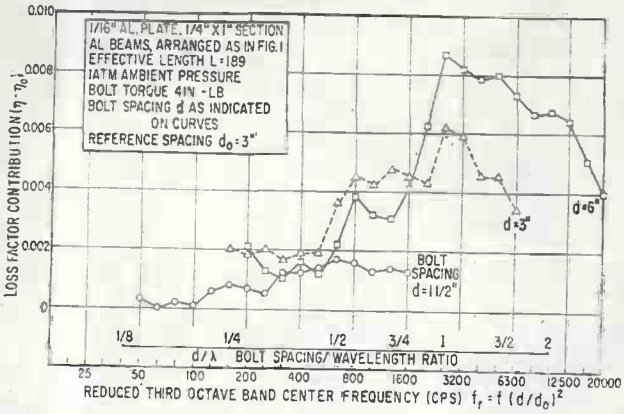

Fig. 3 Effect of bolt-spacing on loss factor contribution made by attached beams.

or less along a common curve when they are plotted against d/λ (or against a function of d/λ), and that the curve for each bolt-spacing deviates from the common one at frequencies that exceed some value that increases with increasing bolt-spacing. The reason for this behavior has not yet been explored, but is thought to be associated with the constraining action the bolts exert on the plate (and which has less effect on longer spans), or possibly with a frequency (or acoustic wavelength) dependence of the air pumping energy dissipation mechanism.

In spite of these deviations from a common curve, the damping data for all of the various configurations studied were found to agree better when plotted against d/λ (or against a function of this ratio) than when plotted against separate functions of d and λ. Thus, it appears reasonable to present a summary of all data in terms of d/λ, and to suggest an estimation technique based on this ratio.

Reduced Frequency and Absorption Coefficient Parameters

Since one usually obtains or requires damping data as a function of frequency, and not as a function of plate flexural wavelength, it is useful to introduce a "reduced frequency" f_r, defined as

$$f_r = f\,(d/d_0)^2\,(h_0/h)\,(c_{L0}/c_L) \qquad (4)$$

where f, d, h, and c_L denote frequency, bolt-spacing, plate thickness, and longitudinal wave velocity, as before, and where the subscript o indicates (constant) reference values of these quantities. This reduced frequency is proportional to the square of the d/λ ratio, as evident from Eq. (3).

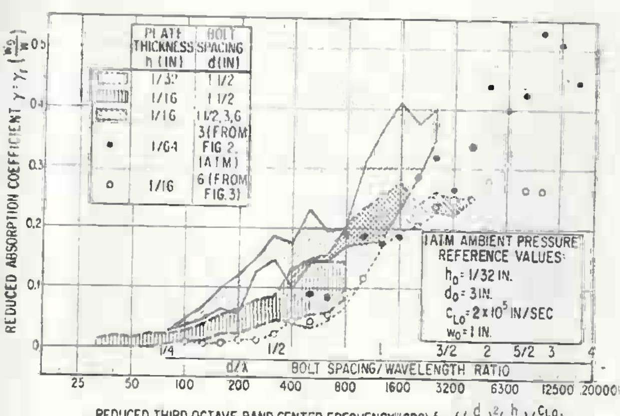

Fig. 4 Summary of absorption coefficient data obtained for 1-in.-wide aluminum beams attached to aluminum plates (for several plate thicknesses and bolt-spacings as indicated, and for bolt-torque between 4 and 25 in.-lb).

The previously defined reduced frequency was used in the construction of Fig. 3, and also appears as the abscissa of Figs. 4 and 5, which summarize the damping data obtained in a large number of experiments. Figure 4 presents absorption coefficients measured with aluminum beams attached to aluminum plates of three different thicknesses, with three different bolt-spacings, and using a wide range of bolt torques. All of the data of Fig. 4 pertain to the same 1-in. width of the attached beams, but several different beam cross sections and effective lengths are represented. Because of the large number of data points involved, these are not shown individually. Instead, regions are indicated into which fall the data for a given set of experimental conditions. Fair agreement between the various experimental results is evident.

As mentioned previously, the damping contribution of a beam was found to be approximately proportional to the width w of the beam face in contact with the plate. It thus is expedient to introduce a reduced absorption coefficient γ_r, which is related to the actual absorption coefficient γ according to

$$\gamma_r = \gamma(w/w_0) \qquad (5)$$

where w_0 denotes an arbitrary reference value of the beam face width.

In Fig. 5 this reduced absorption coefficient parameter is used to summarize experimental data obtained for a considerable range of beam widths. Comparison of these data with that of Fig. 4 (as also indicated in Fig. 5 by the region enclosed by a solid curve) shows that the scatter of the data points here is no greater than that in Fig. 4.

Also indicated in Fig. 5 are data pertaining to steel beams attached to aluminum and to steel plates, as well as data for aluminum beams attached to steel plates. The region occupied by these data points is seen to coincide very nearly with that occupied by the shaded regions of Fig. 4 which pertain to aluminum beams attached to aluminum plates. Absorption coefficient data on plate lap joints with a single row of fasteners[9] are found also to fall largely within the shaded area of Fig. 5, but have not been indicated in this figure.

Estimation of Effect of Reduced Atmospheric Pressure

The data summarized in Figs. 4 and 5 permit one to obtain at least a rough estimate of the absorption coefficient of a multipoint-fastened joint discontinuity for ambient air pressures of one atmosphere. As evident from Fig. 2 and the discussion accompanying it, this estimate must be modified if the ambient air is at lower pressures. Although a detailed understanding of the damping mechanism is still lacking, one may suggest a tentative procedure on the basis of the limited empirical information that has been collected.

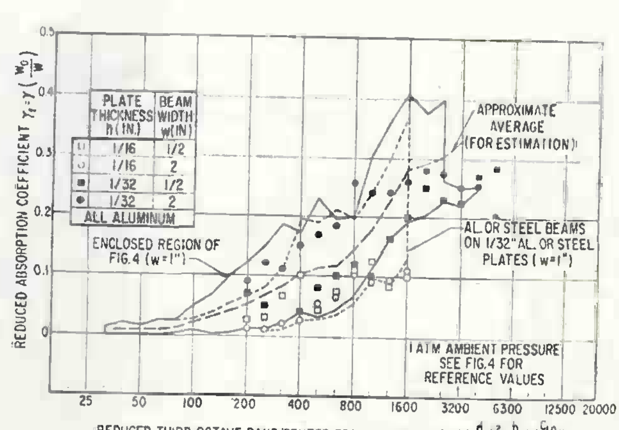

Fig. 5 Summary of reduced absorption coefficient data obtained for beams of various widths and for various beam and plate material combinations.

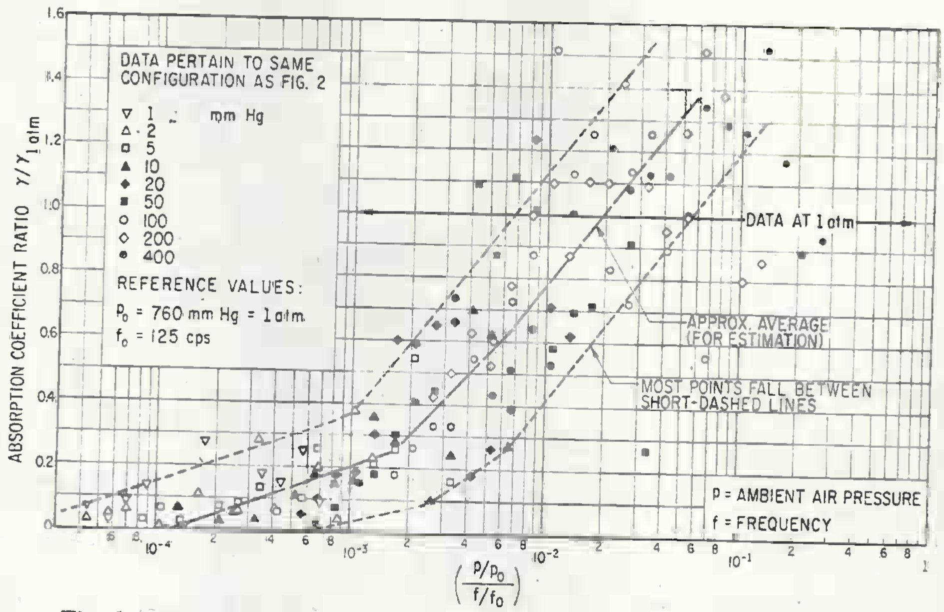

Fig. 6 Dependence of absorption coefficient on pressure/frequency ratio.

The data of Fig. 2 (plus some additional data for the same configuration which have been omitted from Fig. 2 for the sake of clarity) are found to fall within a reasonably well-defined band, if these data are plotted against the ratio of absolute ambient atmospheric pressure p to frequency f, as in Fig. 6. Although presentation of the damping data in terms of other functions of p and f may lead to better clustering of the data, the p/f ratio represents the best physically meaningful parameter with which reasonable correlation of the data has been obtained up to the time at which this paper was submitted. The p/f ratio is, in fact, a quantity that is often used in studies of sound propagation in rarefied gases,[12] and may be shown to be proportional to the ratio of the acoustic wavelength to the molecular mean free path in the gas.

Most of the values of the ratio of the absorption coefficient γ (measured at a given frequency and reduced atmospheric pressure) to the absorption coefficient $\gamma_{1\,atm}$(measured at the same frequency†† and at 1 atmosphere ambient pressure) plotted in Fig. 6 are found to fall within a reasonably well-defined band. This coalescing of the data is most pronounced at low p/f values, whereas the data for high p/f values exhibit considerable scatter. Thus, damping predictions based on Fig. 6 will generally be more reliable for low than for high p/f values.

Summary

The Dominant Damping Mechanism

The experimental results discussed here indicate that at high frequencies (considerably above the fundamental panel resonance) the increase in panel damping resulting from beams or other discontinuities, attached by means of structurally acceptable fastenings, depends very markedly on the ambient atmospheric pressure, on the presence of restraints on the normal relative motions between mating beam and panel surfaces, and on the ratio of connector spacing to panel flexural wavelength. In addition, this damping increase has been found to be very nearly proportional to the total effective length of the discontinuity and to the width of the beam (or discontinuity) face in contact with the panel.

On the other hand, the experimental evidence indicates that the panel damping increase is virtually independent of interface pressure (bolt torque) at the connectors, of connector details, of the beam and plate materials and thicknesses (except insofar as plate thickness and material properties affect the plate flexural wavelengths, and as long as the beam is considerably stiffer than a similar plate strip), and of the smoothness of the mating beam and plate surfaces.

From these observations one may conclude that the dominant mechanism responsible for panel damping caused by bolted, riveted, or spot-welded joints is not associated with interface slip in the vicinity of the connectors. Rather, this damping appears primarily to be caused by a pumping of air, produced as surfaces that are nominally in contact with each other move apart and together (and thus is associated primarily with the regions between connectors).

It is not yet understood precisely how this air pumping removes vibratory energy from a structural configuration; in particular, the observed damping increases produced by pressure decreases appear difficult to explain. However, it is believed that the dominant damping mechanism has been identified in its broad outlines, and that meaningful, although coarse, damping estimates can be made on the basis of available empirical data.

Procedure for Estimation of Damping

Since the energy dissipation contributions of various discontinuities on a plate are additive, and since the absorption coefficient of a discontinuity is a measure of the energy the discontinuity dissipates per unit length, one may solve Eq. (2) for η and generalize the result to obtain

$$\eta = \eta_0 + \left(\frac{\lambda}{\pi^2 S}\right) \sum_i \gamma_i L_i \tag{6}$$

Here γ_i represents the absorption coefficient and L_i the effective length of the ith discontinuity, and the summation is taken over all of the discontinuities present on the plate under consideration.

One may use Eq. (6), in conjunction with the expression (3) for λ, to estimate the loss factor η of a given panel with a given arrangement of discontinuities on it, at a given frequency f (or in a band with center frequency f). The absorption coefficients of bolted, riveted, or spot-welded joints at 1 atm ambient air pressure may be estimated from Figs. 4 and 5. If one is concerned with reduced atmospheric pressures, one may then modify the previously estimated values by use of Fig. 6. In order to obtain the most realistic estimates, one should use the data of Figs. 4 and 5 which pertain

†† Since all of the data of Fig. 6 pertain to the same plate-equal frequencies correspond also to equal plate flexural wavelengths.

to the configuration that most closely resembles the one with which one is concerned. If no directly pertinent data are available in Figs. 4 and 5, or if one only desires a preliminary estimate, then one may use the "approximate average" curves indicated in Figs. 5 and 6.

The loss factor η_0, which characterizes the damping that the panel of interest would exhibit if it had no joints or other discontinuities, accounts for energy dissipation within the panel material and for energy losses from the panel caused by sound radiation from it. (For structures in air, the material damping usually overshadows the acoustic radiation damping.) For most practical configurations, η_0 is negligibly small compared to the total damping produced by the joints. For example, for 2024-T3 aluminum[9] and for magnesium and most steels,[13] η_0 is of the order of 10^{-3}; and even for the relatively highly damped 6061-T6 aluminum alloy, η_0 is only of the order of 5×10^{-3} (see Fig. 2).

Conclusions

The method summarized here provides estimates of the panel damping caused by energy dissipation at structural joints. It is important to note that one requires additional information if one desires to calculate responses in cases where significant energy transmission to other structural components can occur.[14] Thus, the absorption coefficients given in this paper apply, for example, to estimation of the broad-band response of a fuselage section made up of many panels, for the case in which the excitation is approximately uniformly distributed over all panels and in which the bulkheads at the ends of the section are rigid enough so that they permit no significant vibratory energy to flow from the section of interest. On the other hand, if one wishes to calculate the response of a single panel of the aforementioned fuselage section to excitation acting only on the panel, one must account for the energy flow to adjacent panels, as well as for the energy dissipation at the panel edges.

The observed fact that reductions in damping result from reductions in ambient pressure indicates that vibration testing of aerospace structures at ground level atmospheric pressures may be unconservative; greater vibration levels than those observed at zero altitude may occur at the reduced pressures at great altitudes. Further work is in progress, aimed at assessing the practical importance of this damping reduction, at obtaining a better understanding of the air-pumping damping mechanism,[15] and at arriving at configurations with improved damping characteristics based on this mechanism.

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