ACOUSTIC DAMPING MECHANISMS*

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SECTION I

INTRODUCTION

Whenever energy is removed irreversibly from a vibrating system, we say that damping results. When the energy loss occurs through interaction with an adjacent fluid, so-called "acoustic" damping results. The energy need not be converted into heat (immediately) as long as it is made unavailable to the ordered vibratory motion of the panel.

The several types of acoustic damping that we consider in this paper are indicated in the diagram of Fig 1. Perhaps the most familiar type of acoustic damping is the radiation of sound energy to the far field. It is true that the energy in the radiated waves is still in an ordered form. It is, however, unavailable to the panel, and has been lost irreversibly.

When the wavelength of a motion on a large panel is smaller than the wavelength of sound in the adjacent fluid at the same frequency, there is a reactive, oscillatory "near field" adjacent to the panel. If the fluid adjacent to the panel is inviscid, no energy is lost. Instead, the effect of this near field is to present a mass load to the motion of the panel. On the other hand, it is possible to cause energy to be dissipated in the near field if a flow-resistive material is placed in the near-field region. An example of such a material is a glass fiber blanket such as is used for thermal insulations. Energy dissipation takes place as "pseudo radiation" due to the induced oscillatory motions of the fluid in the flow-resistive material.

Other forms of energy removal that may be classed as pseudo-radiation occur when the fluid on one side of the panel is contained by a cavity or compartment. In this case, there are wave reflections from the surfaces of the compartment; and acoustic dissipation means within the compartment can react on the panel.

A final effect that is potentially important in acoustic damping is the effect of motion of the fluid adjacent to the panel. It is known that fluid flow past panels, especially supersonic flow, can give rise to "negative damping", which is evidenced as panel instability and flutter. Thus, in general, we expect that the motion of the adjacent fluid can alter the acoustic panel damping that is observed in the absence of flow.

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SECTION II

RELATIONS BETWEEN ENERGY LOSS AND MEASURES OF DAMPING

Because the term "damping" implies that energy is being extracted from a vibrating system, it follows that a quantitative measure of the amount of damping logically involves the rate of removal of energy. Thus, if we define

D_o = Total energy removed from the system during one cycle,

then,

$P_o = fD_o$ = Average rate of energy removal.

We further define

W_o = Total energy of vibration of the system.

For the types of acoustic damping that we consider in this paper, P_o and W_o are linearly related as follows:

$$P_o = \pi \eta f W_o = \omega \eta W_o \qquad (2\text{-}1)$$

where

η = the damping factor (dimensionless).

For the purposes of this paper, we may regard Equation 2-1 as defining the damping factor. For systems with not-too-high damping, this factor is simply related to the usual measures of damping. We may write the damping ratio (ratio of damping coefficient c to critical damping coefficient c_c) as

$$c/c_c = \eta/2 \qquad (2\text{-}2)$$

The figure of merit Q for resonance sharpness of the system is

$$Q = 1/\eta \qquad (2\text{-}3)$$

The (time-wise) decay rate of a free vibration is given by

$$\Delta_t = 27.3\ \eta\ f\ \text{(db/sec)} \qquad (2\text{-}4)$$

whereas the (space-wise) attenuation of a free-travelling, one dimensional or straight-crested wave is

$$\Delta_\lambda = 13.6\eta\ \text{(db/wave length)} \qquad (2\text{-}5)$$

In this paper we wish to present expressions for the damping factor η for several important types of acoustic damping mechanisms.

SECTION III

RADIATION OF SOUND FROM PANELS

In this section we discuss two types of effects that are important in determining the magnitude of radiation damping of a panel. These effects are principally related to the speed of flexural waves in the panel and to the physical size of the panel.

A. Flexural Waves in Plates

A flexural wave travelling on a plate is characterized by its wavelength λ_b and its speed of propagation c_b. These quantities are related as follows:

$$\lambda_b = c_b/f \text{ (ft)*} \tag{3-1}$$

The speed of propagation of bending waves is given by [1,2]**

$$c_b = (\omega)^{\frac{1}{2}} (B/m)^{\frac{1}{4}} \text{ (ft/sec)} \tag{3-2}$$

where

$\omega = 2\pi f$ = circular frequency (radians/sec)

$B = EH_1{}^3/12(1-\sigma^2)$ = bending stiffness of unit width of plate (lb-ft)

E = Young's modulus of plate material (lb/ft^2)

H_1 = plate thickness

σ = Poisson's ratio for plate material

m = mass of unit area of plate (slug/ft^2)

(3-3)

For steel, aluminum, glass and other materials with approximately the same ratio of Young's modulus to density, we have approximately

$$c_b \doteq 51\sqrt{H_1 f} \text{ (ft/sec)} \tag{3-4}$$

where H_1 is given in inches, and f is given in cycles/sec. This relation is plotted in Fig 2 for a range of plate thicknesses.

* Except where specifically noted, any consistent system of units may be used in the equations in this paper. As examples we use the English units lb (force), slug (mass), ft, sec.

** Numbers refer to references in the bibliography.

The energy of vibration represented by a steady-state flexural wave field on a plate is given approximately by[1,2]

$$W_o = m\,S\,\langle u^2\rangle_{t,S} = \frac{1}{2}\,m\,S\,\langle V^2\rangle_S \qquad (3\text{-}5)$$

where

S = area of the plate;

u = instantaneous velocity of the plate normal to its surface; u is a function of position and time, and

$\langle\ \rangle_{t,S}$ indicates an average over time and over the plate surface.

V = peak amplitude of a normal plate velocity that is sinusoidal in time; V is, in general, a function of position

$\langle\rangle_s$ indicates an average over the surface of the plate.

B. Radiation from Flexural Waves in One Dimension

(1) Plates of Infinite Extent

In order to observe the basic importance of the flexural wave speed in the radiation of sound, let us first review the case for infinite plate. In radiation problems it is useful to define the radiation factor s in accordance with the system sketched in Fig 3[3]. We write

$$s = N/N_o \qquad (3\text{-}6)$$

where

N = Acoustic power radiated from area S of flexural wave having velocity amplitude V

$N_o = \frac{1}{2}\rho c V^2 S$ = Acoustic power radiated from area S of flat plate moving with uniform normal velocity V

Thus, the radiation factor s gives the radiation efficiency of flexural waves relative to the radiation efficiency of a plate with uniform motion over its whole surface.

Gösele[3] has shown that for a free-travelling flexural wave moving on an infinite plate with a bending wave "Mach" number $M_b = c_b/c_o$, where c_o is the speed of sound in the fluid adjacent to the plate, the radiation factor for sound radiation from one side of the plate is written

$$s = M_b \Big/ \sqrt{M_b^2 - 1}\ ,\ M_b > 1$$
$$s = 0\ ,\ M_b < 1 \qquad (3\text{-}7)$$

Since the flexural wave velocity increases with the square root of frequency (see Eq. 3-2), there is a frequency, called the critical frequency f_c, at which the flexural wave velocity equals the speed of sound in the adjacent medium. Therefore we may write $M_b^2 = f/f_c$. The function represented by Eq. 3-7 is plotted in Fig 4 as a function of f/f_c.

We see that there is no radiation of energy when the speed of propagation c_b of the flexural wave is less than c_o, the speed of sound in the adjacent medium. Under these conditions, the medium presents a mass load to the flexural wave motion, but does not withdraw energy. As we shall see later, the simplicity of this result depends upon the fact that the plate is infinite in extent.

Above the critical frequency the radiation factor rapidly approaches a limiting value of 1. That is, for frequencies sufficiently greater than the critical frequency, the flexural wave radiates sound energy as well as would the uniform motion of the plate surface. Incidentally, all of the radiation is in the form of plane waves travelling at an angle $\theta = \sin^{-1} \lambda_o/\lambda_b$ from the normal to the plate. The infinite value of s for $f/f_c = 1$ is a result of calculating for an infinite plate, and of academic interest only.

(2) Plates of Finite Size

Let us consider the radiation from the finite plate shown in Fig 5. For this system Gösele has derived expressions for the radiation factor, both for waves assumed to be free-travelling across the plate, that is, waves appearing at one edge of the plate and disappearing at the other edge, and for standing waves of the plate. In the case of standing waves both free and supported end conditions were considered for the plate. All of these wave systems are made up of waves propagating in the "b" direction on the plate. The finite plate is assumed to be surrounded with an infinite, coplanar rigid surface.

For this system it is useful to introduce a characteristic frequency f_o which is defined as

$$f_o = c_o/b \tag{3-8}$$

At this characteristic frequency the plate length b equals the wave length of sound in the adjacent medium. Gösele's calculations of the radiation factor for travelling waves on plates of several sizes appear in Fig 6. It has been assumed that the plate width, a, is greater than the wavelength of sound in air at the frequencies of interest. The ratio b/λ_c, the number of wavelengths of the flexural waves that fit onto the plate at the critical frequency, is used as an index of the plate size. For the particular case, $b/\lambda_c = 30$, Gösele's calculated results for three different situations are presented in Fig 7.

From the results presented in Fig 7 Gösele concludes that the radiation factor for standing waves on a plate with supported edges does not differ appreciably from that for free-travelling waves, provided that the plate is large enough, i.e., as a large enough value of b/λ_c. On the other hand, at frequencies below the critical frequency, the radiation factor for standing waves on a plate with free edges is less than that for either of the other cases.

For cases in which the characteristic frequency f_o is much smaller than the critical frequency, i.e., $f_o \ll f_c$, we have the following results:

a) $f_o < f < f_c$

$$s \doteq \frac{f_o}{2\pi f_c} (1 - f/f_c)^{-\frac{3}{2}} \quad \text{free travelling wave} \tag{3-9a}$$

$$s \doteq \frac{f_o(2 - f/f_c)}{2\pi f_c (1 - f/f_c)^{\frac{3}{2}}} \quad \text{for standing wave on supported plate} \tag{3-9b}$$

b) $f = f_c$

$$s \doteq 0.94\sqrt{f_c/f_o} \tag{3-10}$$

c) $f > f_c$

$$s \doteq 1/\sqrt{1 - f_c/f} \tag{3-11}$$

C. Damping Factor Due to Radiation of Sound

The radiated acoustic power from a plate with uniform motion (see Fig 3) is given by

$$N_o = \frac{1}{2} \rho c_o S V^2 = \rho c_o S \left\langle u^2 \right\rangle_{t,S} \qquad (3\text{-}12)$$

and from Eq. 3-6 we may write the total acoustic power radiated as

$$N = sN_o = s\rho c_o S \left\langle u^2 \right\rangle_{t,S} \qquad (3\text{-}13)$$

If the radiated power accounts for all of the losses from the plate, we say $P_o = N$. From Eqs 2-1, 3-5, and 3-13 we write for the damping factor

$$\eta = \rho c_o s/\omega m \qquad (3\text{-}14)$$

Thus we see that the radiation factor can be used directly in determining the damping factor of a plate having a surface mass m per unit area, radiating into a medium having characteristic impedance ρc_o. As was mentioned before, the damping due to acoustic radiation has been determined for loading on one side of the plate only. Twice the radiated acoustic power, and therefore twice the damping factor would result from radiation from both sides of the plate. We should remember, however, that in some cases of interest the media on opposite sides of the plate may have different densities and speeds of sound.

SECTION IV

ACOUSTICALLY COUPLED INTERACTIONS WITH SURROUNDINGS

In this section we consider two types of acoustical coupling between a panel and its surroundings. The first of these is the dissipation of energy in the near field near a plate carrying flexural waves below the coincidence frequency. The dissipation results from the introduction of a flow-resistive material in the near field region. The second phenomenon results from the acoustical coupling of a panel to an adjacent cavity or compartment that itself contains means for absorbing acoustic energy.

The dissipation of energy in the near field of a vibrating plate can take place over a wide range of frequencies. However, we expect that the dissipation through coupling to a cavity will be important only near those cavity resonances which present a relatively high acoustic impedance to the surface of the panel.

A. Dissipation in the Near Field of a Vibrating Plate: Pseudo-Radiation

In our consideration of the radiation loading of infinite plates (Section III), we noted that below the coincidence frequency f_c the fluid medium presents only a massive load to the plate. In this case the acoustic field is "reactive" in nature, and no energy is propagated to great distances. Near the plate there is particle motion, both normal to and parallel to the plate. However, this particle velocity u, as well as its associated acoustic pressure p, decay rapidly with increasing distance, from the plate according to the following relation:*

$$p/p_o \text{ or } u/u_o = \exp\left[-2\pi\sqrt{1-M_b^2}\,(z/\lambda_b)\right] \qquad (4\text{-}1)$$

Here p_o and u_o are the values found at the surface of the plate, and the distance from the surface of the plate is expressed as a number of bending wavelengths, i.e., by the ratio z/λ_b. We see, for example, (see Fig 2 and Eq 3-1) for a 1/8 in. thick aluminum or steel plate at 300 cps, c_b = 320 ft/sec, λ_b = 1.07 ft, and in a distance 0.18 $\lambda_b \doteq$ 2.3 in., the near sound field has dropped 10 db from its value at the surface of the plate. For a 1/16 in. thick plate at 1600 cps, the "10-db-down" point occurs at about 0.7 in. from the plate.

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* Equation 4-1 can be derived from information available in Ref 1 or 2. A plot of 20 log p/p_o, but not the expression of Eq. 4-1, is found in Reference 3b.

In Fig 8 we have plotted the quantity 20 log u/u_o or 20 log p/p_o for $f/f_c = 0.5$ and for $f/f_c \ll 1.0$.

Now it appears reasonable to expect if we place a flow-resistive material (e.g., fibrous blanket) in the region of the reactive near field, that energy dissipation will result, due to the induced, oscillatory flow within the blanket. The presence of the blanket will, of course, modify the near field to an extent determined by the blanket properties.

Figure 9 shows experimentally and theoretically determined values of the damping factor η for a 1/16 in. steel plate, to which was cemented a 1.5 in. thick glass fiber blanket (Owens-Corning PF 338). This blanket has nominal density 1.5 lb/ft^3. The plate was rectangular in shape: 3.1 by 4.0 ft. The measured damping of a bare plate of the same size is included in the figure.

The theoretical curve presented in Fig 9 was developed by R. J. McQuillin[4/] for the case of a fibrous blanket of infinite extent, starting from expressions developed by Beranek[5/] for wave motion in porous media. Obviously, at low frequencies we should not expect a blanket only 1.5 in. thick to appear "infinite" in thickness. However, at higher frequencies, say above about 500 cps, this blanket thickness is comparable to the thickness of the region containing most of the reactive energy in the undisturbed near field. Under these circumstances, the theory should give fair agreement with the experiment, as is the case.

The theory used in preparing the calculated curve in Fig 9 neglects the motion of the fibers in the blanket, and considers only the dissipation of energy due to the movement of the air through the flow-resistive blanket. As the blanket is actually cemented to the plate, it is clear that the fibers do move in the immediate vicinity of the plate. A more elaborate theory is required to evaluate the importance of the effect. It is sufficient to say that a relatively simple theory appears to account rather well for the mid- and high-frequency damping due to a fibrous blanket.

Approximate Maximum Damping by a Porous Blanket

It is often valuable to know approximate upper limits for various types of damping phenomena. We speculate here on a probable upper limit of damping due to flow-resistive material adjacent to vibrating panels. We use the very rough physical model of a blanket layer that is forced into motion by the motion of air through the blanket. That is, the blanket does not quite touch the plate. The air motion is induced by the vibrations of a adjacent panel. If the blanket is represented simply by a layer having mass and flow resistance, we can show that the maximum real part of the normal acoustic impedance offered to the plate occurs when this real part of the impedance is equal in magnitude to the reactance of the blanket mass. The maximum real part of the impedance is found to be equal to one half of the blanket mass reactance or

$$R_{max} \doteq \frac{1}{2}\omega m_{Blanket} \qquad (4\text{-}2)$$

Thus the maximum expected damping factor for this mechanism is approximately (see Eq. 3-14, substituting R_{max} for ρcs)

$$\eta_{max} \doteq \frac{1}{2}\frac{\omega m_{Blanket}}{\omega m_{Panel}} = \frac{1}{2}\frac{m_{Blanket}}{m_{Panel}} \qquad (4\text{-}3)$$

The criterion of Eq 4-3 predicts a maximum damping factor of about 0.04 for the experiment reported in Fig 9. This corresponds very nearly to the maximum observed damping factor, which may be entirely fortuitous in this case. We should point out that our physical model is much simpler than the true case. For example, losses in the blanket due to fluid flow parallel to the plate, and frictional losses in the blanket structure have been ignored. Nevertheless, Eq 4-3 may be somewhat useful in giving a rough estimate of the maximum possible damping to be expected from a given blanket.

B. Damping Due to Interaction Between a Panel and an Adjacent Cavity

Whenever a panel forms a part of one wall of a cavity or compartment, it is possible that the panel motion can be influenced by the acoustical character of the cavity. For example, at low frequencies when a cavity may be small with respect to the wavelength of sound, the cavity can exert a stiffness reaction on the panel that, for light panels, can actually shift the resonance frequencies. In the limit of very high frequencies, the influence of a cavity or compartment that is at least somewhat absorptive acoustically tends to disappear, and the panel sees only an appropriate acoustic radiation load determined by the properties of the fluid in the compartment.

In the range of "middle" frequencies, that is, where the cavity dimensions are in the range of about 0.5 to 5 wavelengths of sound, cavity resonances can affect the damping of the panel. A cavity mode that is strongly coupled to a mode of vibration of the panel could present a real component of the acoustic loading impedance on the panel as large as 10 ρc_o, provided that the cavity had only medium-low acoustical absorption (statistical absorption-coefficient of approximately 0.25)[6,1] Such acoustic loading would represent an increase of as much as a factor (10/s) over the radiation damping expected in the absence of the cavity. (s = radiation factor.)

On the other hand, such a resonance would be narrow in frequency, and the frequency of the resonance would vary with cavity contents, temperature, etc. Perhaps more important, it is most unlikely that the resonance frequency of the cavity and panel mode would coincide exactly, giving rise to the strong coupling.

These arguments lead us to conclude that strong panel-cavity interaction is unlikely, and therefore cannot be treated as a controllable factor in panel damping. Such interaction may, however, account for an occasional highly damped panel mode.

We should remember that a pressurized cavity behind a panel could affect the panel motion at altitude by maintaining the value of ρc_o, the characteristic impedance of the air, at a value corresponding to some low altitude. In addition a static pressure differential across the panel may serve to suppress flutter instabilities (see Section V).

SECTION V

EFFECTS OF FLUID MOTION

It is entirely reasonable to ask, "What, if any, effects does flow of fluid past a panel have on the acoustic damping of the panel?" The fact that there can be an interaction between panel motion and the motion of the fluid adjacent to the panel is evident from the existence of cases of panel flutter. In such cases, energy is transferred from the moving fluid to the vibration of the panel, the result being, at least in its incipient stages, equivalent to the presence of negative damping. We should also like to ask whether or not "unflutter" is possible. That is, "Can the motion of the fluid increase the acoustic damping that exists in the absence of flow?"

A. Panel Flutter Studies

The analytical methods appropriate to the analysis of flow-induced damping should be similar to those used in flutter studies. An excellent review of the literature on panel flutter is now available[7]. This review makes clear the complexities and disagreements that characterize the present state of the art. Several researchers [8,9] have chosen to evaluate simplified systems of infinite extent. They have been criticized by others[10] who maintain that a more realistic system must be studied if useful results are to be obtained.

Some of the considerations involved in a complete analysis of the problem of panel flutter are the following:

Exact unsteady aerodynamic forces on the panel

Density, elastic modulus, and physical dimensions of the panel.

Panel edge conditions, e.g., supported, clamped, etc.

Bending and extentional (mid-plane) stresses in the panel

Static pressure differential across the panel

"Acoustic" loading on the side of the panel not exposed to flow

Inherent structural damping of the panel

Buckling and warping of the panel

Naturally enough, different researchers have chosen different physical models for the panels in their flutter studies. The conclusions reached in their studies differ in accordance with the considerations neglected or simplified.

The studies of flutter phenomena that have been reported in the literature appear to have concentrated on the establishment of stability boundaries beyond which the panels become unstable.

The usual approach has been to derive limiting or critical panel thicknesses above which flutter can be avoided.

A more or less uniform conclusion from the various studies reviewed seems to be that if panel flutter occurs, it will occur at supersonic flow speeds. Some of the studies find flutter most likely in the Mach number range $1 < M < \sqrt{2}$. Others find that for high enough dynamic pressure of the flow, flutter can occur at arbitrarily high supersonic Mach numbers.

In general, stabilizing (i.e., anti-flutter) effects are predicted for static pressure differentials across the panel and for positive mid-plane tension in the panel. A tendency to panel flutter is often referred to as negative damping. However, this negative damping is not necessarily completely compensated for by positive structural damping[11]. Although structural damping is expected to inhibit flutter effectively under certain conditions[11], both Miles[8] and Leonard and Hedgepeth[12] state that the stability analysis of unstiffened cylindrical shells must necessarily assume the presence of a small amount of structural damping. It is maintained that this structural damping actually causes a weak instability resulting from phase shifts in the wave on the shell.

B. Effects of Fluid Motion on the Radiation From Standing Waves on Flat Panels

A simplified analysis of the radiation efficiency of a one-dimensional standing wave pattern on an infinite flat plate has been carried out for presentation here. Sinusoidal standing waves have been assumed, with the wavelength determined by the speed of flexural waves in the plate at the frequency of interest.

The relation between acoustic pressure and normal velocity at the surface of the plate was determined for each of the two travelling waves that go to make up the standing wave. The determination was made for each wave in a coordinate system fixed with respect to the fluid that moves past the panel. Appropriate coordinate transformations were then used to transform both sets of results to a set of coordinates fixed with respect to the plate, with the fluid moving past this coordinate system. In a final step, the average radiation efficiency for the standing wave system was determined by integration of an intensity function over a wavelength of the standing wave.

The resulting radiation factor is as follows:

$$s = \frac{1}{2}\left[\frac{M_o - M_b}{\sqrt{(M_o - M_b)^2 - 1}} + \frac{|M_o + M_b|}{\sqrt{(M_o + M_b)^2 - 1}}\right] \qquad (5\text{-}1)$$

where

M_o is the Mach number of the fluid flow past the panel, and

$M_b = c_b/c_o$ is the bending wave "Mach number".

A term in this equation is, by definition, zero if its denominator becomes imaginary.

We note that for zero flow, i.e., $M_o = 0$, Eq. (51) reduces to Eq. (3-7), as it should. On the other hand, as M_o becomes arbitrarily large, the radiation factor given by Eq. (5-1) approaches unity, in agreement with the results of aerodynamic "piston" theory*.

The radiation factor described by Eq. (5-1) is plotted in Fig 10 as a function of M_o for several different values of M_b. We see, in accordance with the results of Section III, that when the relative velocity of a flexural wave and the adjacent medium exceeds the velocity of sound, the radiation efficiency increases. When the Mach number M_o of the flow is high enough so that $M_o - M_b$ is greater than unity, then both the upstream and the downstream travelling flexural waves are moving supersonically relative to the fluid medium. Above this value of M_o the radiation factor approaches its limiting value of 1.0.

We note that the effect of fluid flow on the radiation from a panel for which M_b is greater than 1 is to reduce the radiation factor at first, as the downstream-travelling wave is rendered subsonic. This loss in radiation factor is recovered above $M_o - M_b = 1.0$.

For the case $M_b < 1.0$ the plate is radiating inefficiently for $M_o = 0$. In this case, as M_o is increased, the radiation factor rises as soon as $M_o + M_b = 1.0$, and the upstream-travelling wave becomes supersonic.

In summary, we observe that the effect of flow past the panel can be either to increase or to decrease the radiation efficiency. The increases can be significant, but the decreases are only by about a factor 2. The effect is the same in all cases in the limit of high Mach number. That is, the radiation factor approaches 1.0.

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* A major result of "piston" theory[13/] is that at high Mach numbers the unsteady aerodynamic forces are such that the ratio of fluctuating pressure at the surface of the plate to the normal component of velocity is the same as would be found in front of a piston in an infinitely long straight-walled tube. That is, $p/u = \rho c_o$.

The results presented in Eq (5-1) for an infinitely extended standing wave can be adapted to finite trains of standing waves by fourier synthesis. That is, the finite-standing wave train can be synthesized from a spectrum of infinite wave trains of varying wavelengths (but all having the same frequency). With the spectrum in wavelength as a weighting function, one can integrate the radiation factor over wavelength (or equivalently over M_b), obtaining an effective radiation factor for the wave train. This procedure is quite similar to that used by Gösele in his analysis of the radiation from finite plates without flow, as discussed in Section III.

Equation (5-1) predicts only damping of the panel with adjacent fluid flow, whereas the work of Miles[8/] predicts instability under certain circumstances. Although the two approaches are similar in some respects, the major difference may lie in Miles' inclusion of mid-plane tension forces, which we have neglected here. In any case, our assessment of the importance of any increase in radiation damping due to flow past the panels must be tempered by the considerations given below.

C. Comments on the Significance of Flutter and Radiation Damping

Experiments have shown that in actual cases, flow past panels is capable of introducing negative damping. In some cases, this negative damping can be significant enough to result in absolute instability and consequent panel flutter. In the light of these facts, the possibility of flow-induced instability is important, even in cases where the instability is "sub-threshold", i.e., is not strong enough to overcome the inherent damping in the system. In such cases the tendency toward instability can partially counteract the inherent damping of a panel. Thus, when the motion of the panel is damping limited, such as near resonances, it may be possible to observe an increase in response due to the presence of the flow, all other things being equal.

This possibility of increased response is of importance regarding both acoustic fatigue and noise and vibration control. It is our hope that future work will shed more light on this problem.

The possibility of flow-induced instability, either absolute or sub-threshold, as just discussed, is not necessarily contradictory to the increase in radiation damping predicted by Eq (5-1). The flow-induced instability will be found for certain wavelengths and/or frequencies as determined by the flutter analysis. However, systems having important responses at other wavelengths or frequencies may still experience flow-induced changes in radiation damping as described by Eq (5-1).

SECTION VI

COMPARISON OF ACOUSTIC DAMPING AND DAMPING BY APPLIED TREATMENTS

In assessing the potential importance of acoustic damping, we wish to compare it with the damping obtainable through the use of applied damping treatments. Two important types of applied treatments are the free viscoelastic layer [14] and the constrained viscoelastic layer [15]. Recent work has shown that for relatively low treatment weights the constrained viscoelastic layer is more effective than the free layer.[16] Therefore we shall use the constrained layer in our comparisons here.

In Fig 11 we show the maximum damping factors obtainable from constrained viscoelastic layer treatments applied in various percentages of the weight of a panel. These damping factors apply both for steel and aluminum (or other homogeneous) panels, as long as the constraining layer has the same Young's modulus as the plate material. Because we have chosen to present curves for constant percentage treatment weight, the damping factors expected are independent of the plate thickness.

Also presented in Fig 11 are curves giving the expected damping factors for radiation damping in accordance with Eq. (3-14). The parameter on the curves is the ratio s/f, the ratio of radiation factor to frequency. Since the radiation damping is dependent on plate surface weight, the results in Fig 11 are specific to aluminum plates in air at sea level pressure and a temperature of 59° F. Corrections are given for steel plates, and for altitudes of 25,000 ft and 50,000 ft. The radiation damping factors are for radiation from one side of the plate.

As an example of the application of Fig 11 let us estimate the radiation damping for the steel plate used in the experiments described in Fig 9.

Example: Estimate Radiation Damping of 3.1 x 4 ft x 1/16 in. steel plate at 500 and 2000 cps at sea level, at room temperature (c_o = 1130 ft/sec)

		References
f_c	$= 8000$ cps	Fig 2
λ_c	$= 1130/8000 = 0.14$ ft	Eq (3-1)
b/λ_c	$= 4/0.14 = 28$	
f_o	$\equiv c_o/b = 1130/4 = 280$ cps	Eq (3-2)

	f=500 cps	2000 cps	
f/f_c	0.063	0.25	
10 log s (db)	-22	-21	Fig 7
s	6.3×10^{-3}	8×10^{-3}	
s/f	1.3×10^{-5}	4×10^{-6}	
η(1 side)	0.7×10^{-4}	2.1×10^{-5}	Fig 11 (Extended)
η(Both sides)	1.4×10^{-4}	4.2×10^{-5}	

Referring to Fig 9, we see that the estimated radiation damping lies well below the bare-plate damping actually observed. Actually the estimated damping factors for radiation damping are extremely small, and we would expect considerable difficulty in observing such damping factors. It is more likely that the bare-plate results of Fig 9 indicate damping from other sources, such as plate suspension, accelerometer attachment, etc. There is, of course the possibility that our estimate of radiation factor is low. For example a ten-fold increase in the radiation factor would bring the estimated radiation damping more nearly in line with the measured bare-plate damping, at least at 500 cps. We note that Gösele's analysis leading to the results of Fig 7 assumes that the panel in question is surrounded by a rigid plane surface. In the experiment described in Fig 9 the plate was free hanging.

Reference to Fig 11 shows that a constrained viscoelastic layer treatment weighing only one percent of the plate weight is expected to supply much more damping than was estimated in the example above.

It would be more interesting to compare a higher value of radiation damping with the damping obtainable through the use of applied treatment. Consider, for example, an aluminum panel 0.032 in. thick. If the panel were of an appropriate size to have its fundamental resonance at a frequency of about 100 cps, we might expect a radiation factor of the order of 0.1. (A panel of this size at such a low frequency lies outside the region of validity of Fig 7. However as Skudrzyk 17/ has pointed out, the lowest (odd) modes of a plate can be expected to radiate rather well.)

From Fig 11 we see, for $s/f = 10^{-3}$, that the radiation damping factor at sea level for radiation from both sides of the panel would be about 0.06. This is directly comparable with the maximum damping expected from a constrained-layer treatment having ten percent of the weight of the plate.

The relatively high radiation damping factor estimated for the thin aluminum panel falls off quite rapidly with increasing frequency, both because of increase in ωm, the mass reactance of the panel, and also because of an expected decrease in the radiation factor until the panel reaches the critical frequency (approximately 13kc in this case). Further decreases with increasing altitudes are, of course, expected.

Until such time as more conclusive evidence is available, it appears that radiation damping is more likely to be a relatively uncontrollable side effect rather than a directly useful design parameter. Small amounts of added damping treatment can probably provide higher damping at a stable level in most cases. We must of course realize that quite different conclusions must be drawn in cases where the fluid adjacent to the panel has a characteristic impedance ρc that is appreciably greater than that for air. For example panels in water can experience appreciable radiation damping under certain circumstances.

SECTION VII

CONCLUSIONS AND RECOMMENDATIONS

On the basis of the observations made in this paper, we may draw the following conclusions:

1. Acoustic damping can result from radiation or pseudo-radiation of energy.

2. Flow past the panel can affect the radiation of energy.

3. The maximum radiation damping expected is given approximately by [Eq (3-14) for s = 1]:

$$\eta_{max} = \rho c_o/\omega m$$

We see that this maximum damping factor decreases with increasing frequency.

4. A speculative estimate of the maximum possible damping due to pseudo-radiation into a flow-resistive blanket is the following [Eq. (4-31)]:

$$\eta_{max} \doteq \frac{1}{2}\frac{m_{Blanket}}{m_{Panel}}$$

This must be regarded as tentative and unproven.

5. Properly designed applied damping treatments can give more damping, and, perhaps, more predictable and reliable damping than is provided by acoustic dissipation of energy. However, added treatments increase the weight and probably the cost of the panel.

It seems appropriate to comment on the significance of acoustic damping as related to acoustic fatigue. Presumably, acoustic fatigue refers principally to the effects of panel motion that has been induced through acoustic action, that is, through the influence of a fluid medium in contact with a panel. Qualitatively, we might argue that the manner in which a panel accepts energy from an acoustic field[18] is related to the manner in which the same panel can radiate energy. Thus, it may be appropriate to say that it is probable that when a panel accepts acoustic energy readily, it will also radiate well.

A cautionary remark seems in order. In view of the demonstrated possibility of flutter in a panel exposed to a moving fluid, it seems unwise to count heavily on acoustic damping to be present under all conditions, in spite of the optimistic suggestions presented in Fig 10. On the other hand, it may be possible to show that acoustic damping is effective in frequency or wavelength regions outside the regions of influence of the instability tendencies.

Problems encountered in this attempt to summarize available information on acoustic damping indicate items that need further clarification. We therefore recommend that the following items be considered in future studies by those working in this field:

1. The importance of structural damping in flutter suppression

2. The effect of flow on the radiation of energy from panels

3. The importance of "sub-threshold" instabilities that may serve to increase the response of panels, making the fatigue problem more severe.

4. The relation between acoustic damping and acoustic excitation.

Significant contributions have been made in the fields of the excitation, dynamics and fatigue of structures to date. We hope that the continued efforts of those working in the field will lead to satisfactory answers to the problems that still face us.

LIST OF SYMBOLS

Units are given in terms of Force, Mass, Length, and Time (F, M, L, T) where $F = MLT^{-2}$. Except where specifically stated, any consistent system may be used, e.g., lb, slug, ft, sec. When no dimensions are given, the quantity is dimensionless.

a,b	Plate dimensions, See Fig 5 (L)
c/c_c	Damping ratio: Damping coefficient c to critical damping coefficient c_c
c_b	Speed of propagation of flexural wave (LT^{-1})
c_o	Speed of sound in fluid (LT^{-1})
f	Frequency in cycles per second (T^{-1})
f_o	Characteristic frequency, see Eq 3-8 (T^{-1})
m	Mass per unit area of plate (ML^{-2})
p, p_o	Acoustic pressure (FL^{-2})
s	Radiation factor
u	Acoustic or time-varying velocity (LT^{-1})
z	Distance in directions normal to the plane of the undisturbed panel (L)
B	Bending stiffness of unit width of plate (FL)

D_o	Total energy removed from vibrating system during one cycle (FL)
E	Young's Modulus (FL^{-2})
H_1	Plate thickness (L)
$M_b = c_b/c_o$	Mach number of flexural wave motion
M_o	Mach number of fluid flow
N, N_o	Radiated acoustic power (FLT^{-1}) (See Eq 3-6)
$P_o = fD_o$	Average rate of energy removal (FLT^{-1})
Q	Figure of merit or sharpness of a resonance
R	Real part of acoustic impedance ($FL^{-3}T$)
S	Surface area of the plate (L^2)
V	Velocity amplitude (LT^{-1})
W_o	Total energy of vibration of system (FL)
η	Damping factor (dimensionless)
θ	Angle
λ_b	Wavelength of flexural wave (L)
ρ	Density of fluid adjacent to panel (ML^{-3})
σ	Poisson's Ratio
$\omega = 2\pi f$	Circular frequency in radians/sec (T^{-1})
Δ_t	Time rate of decay of a free vibration in db/sec (T^{-1})
Δ_λ	Space rate of decay of flexural wave in db/wave-length (L^{-1})

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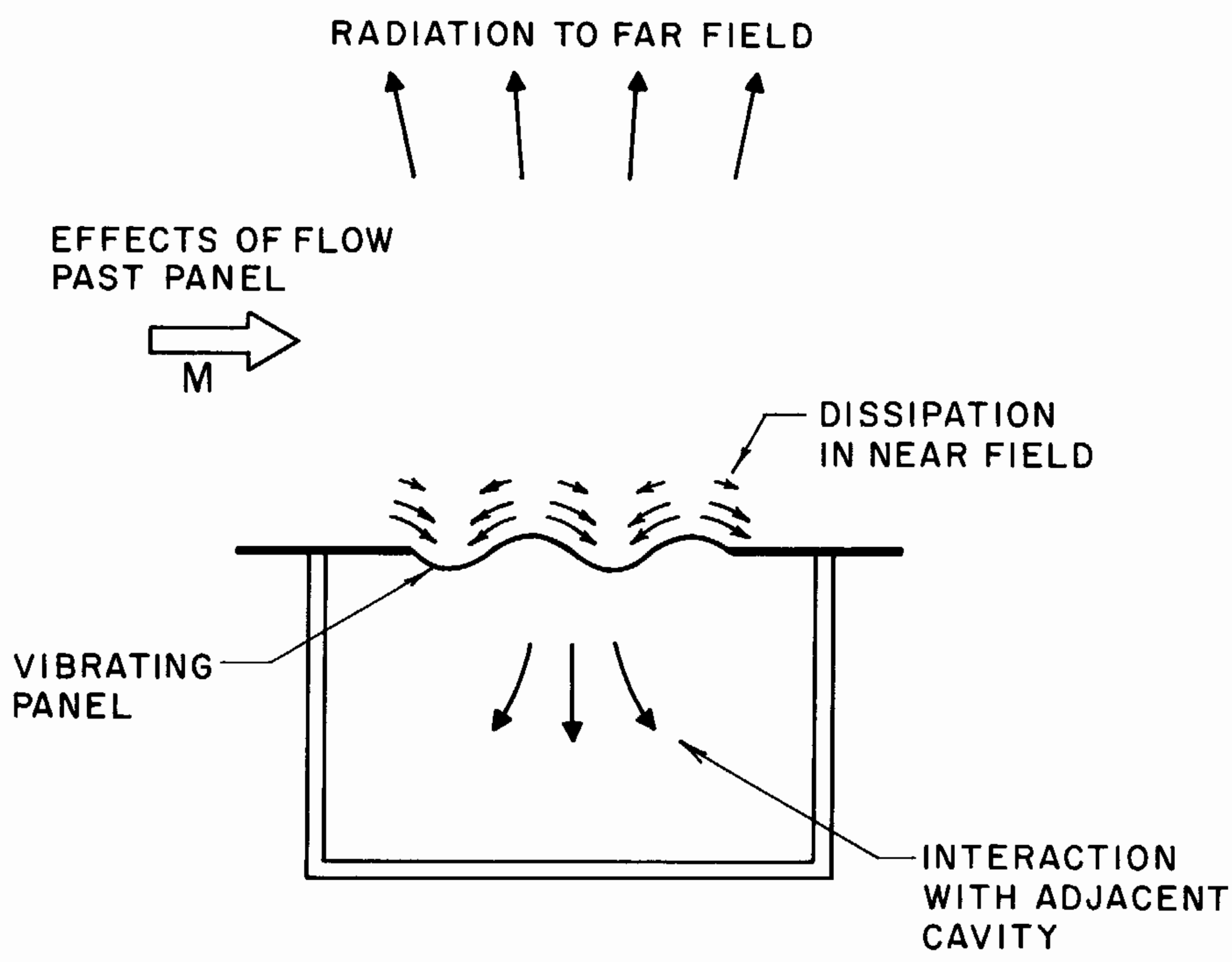

Fig. 1 - Acoustic Damping Mechanisms

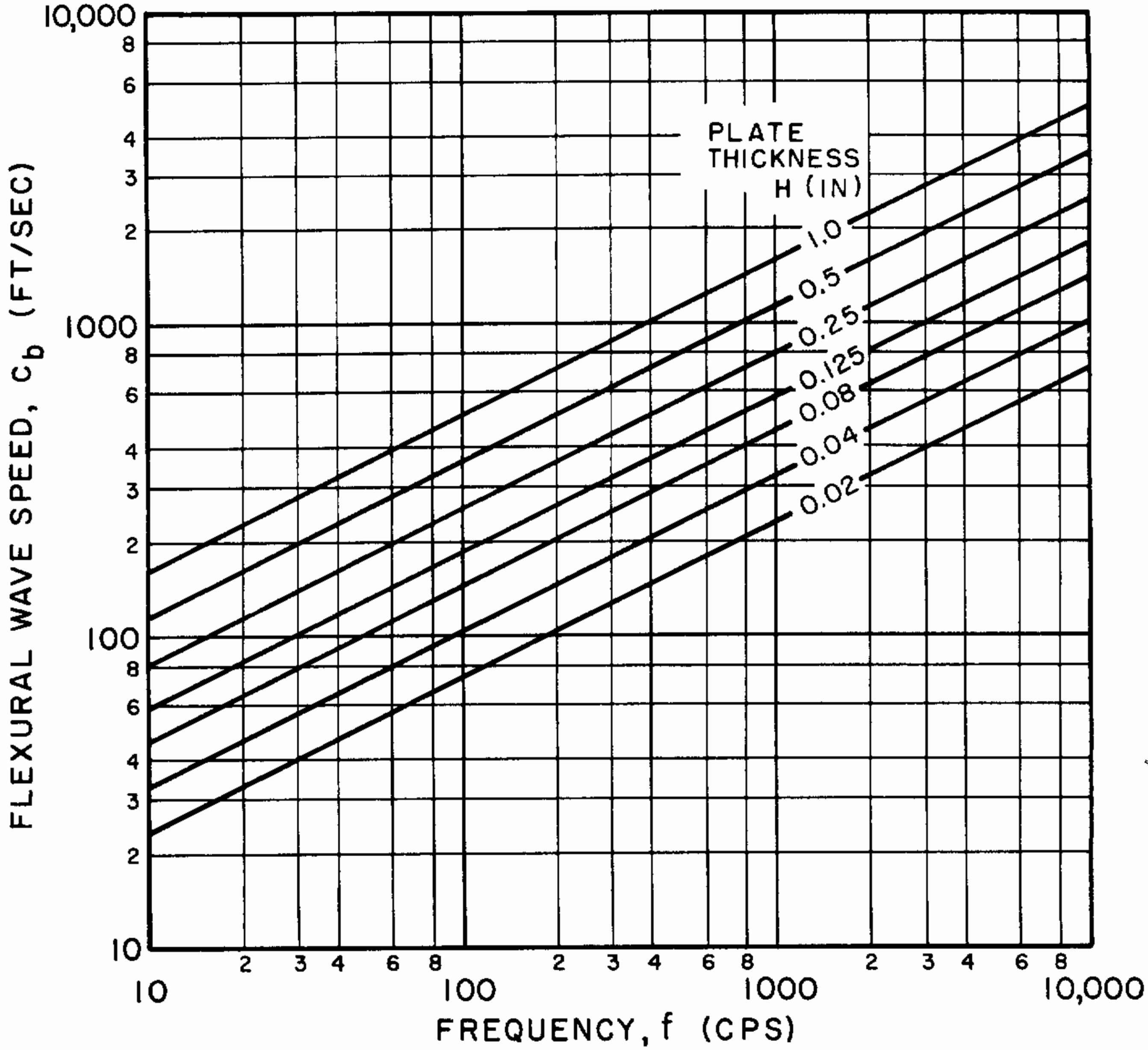

Fig 2 Flexural Wave Speed on Flat Aluminum or Steel Plates

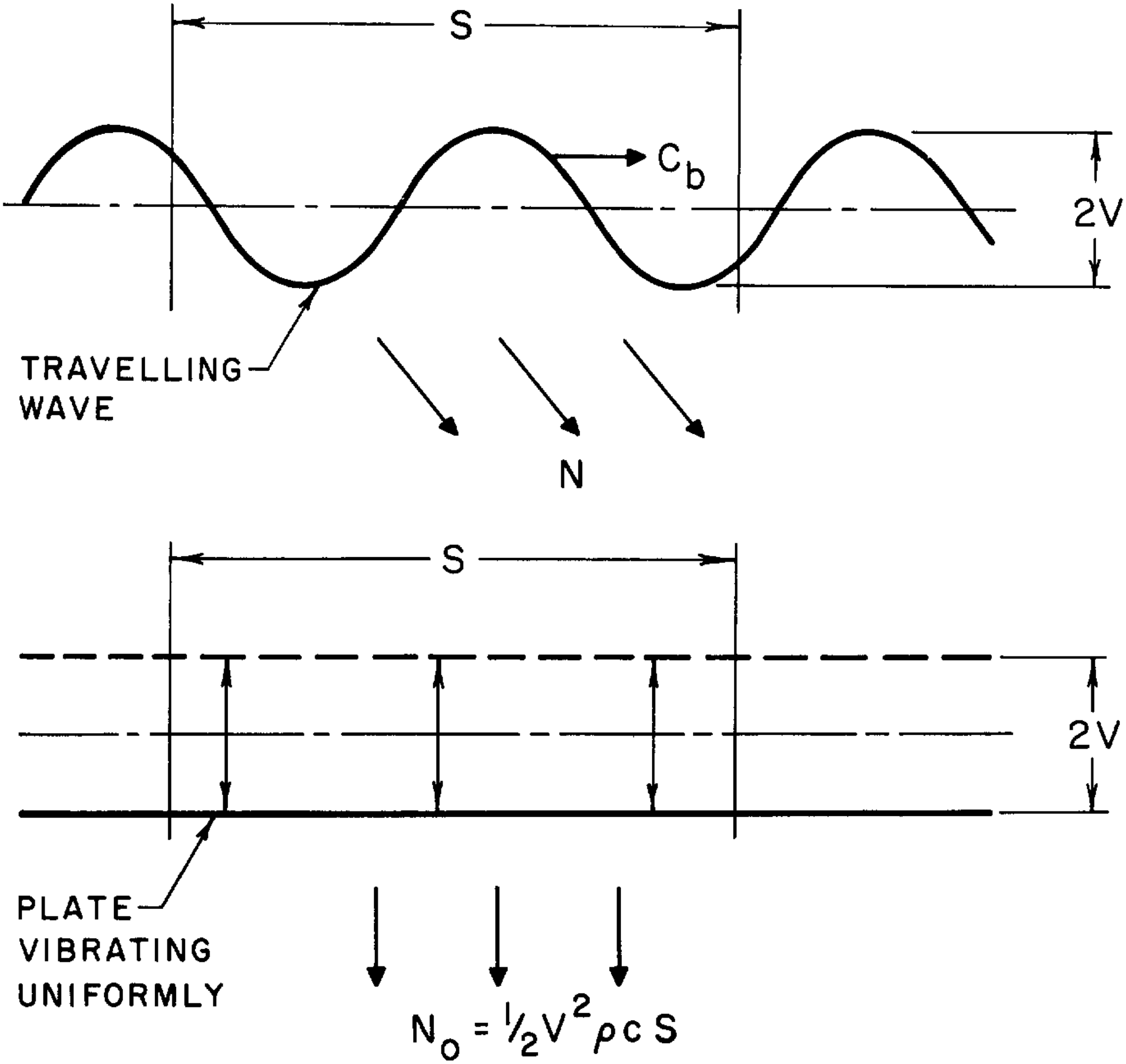

Fig. 3 - Definition of Radiation Factor $s = N/N_o$ for Flexural Waves

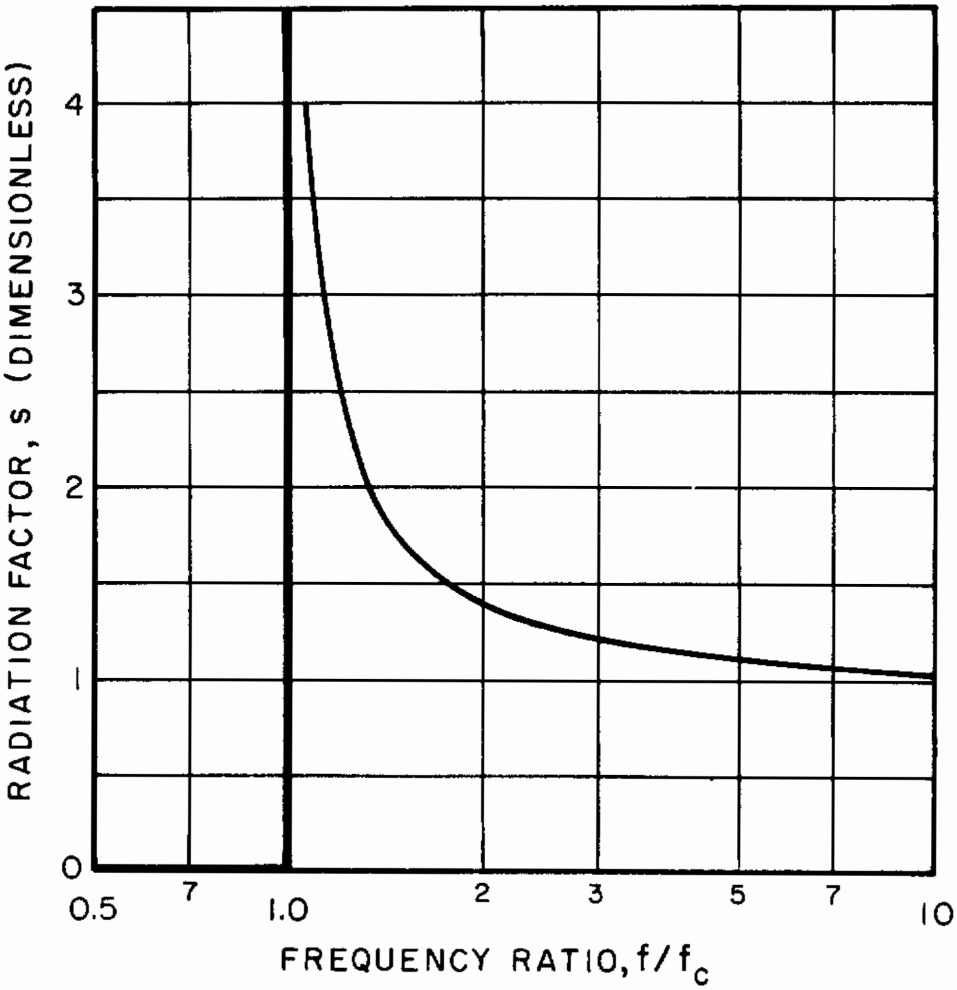

Fig. 4 - Radiation Factor for Infinite Plate

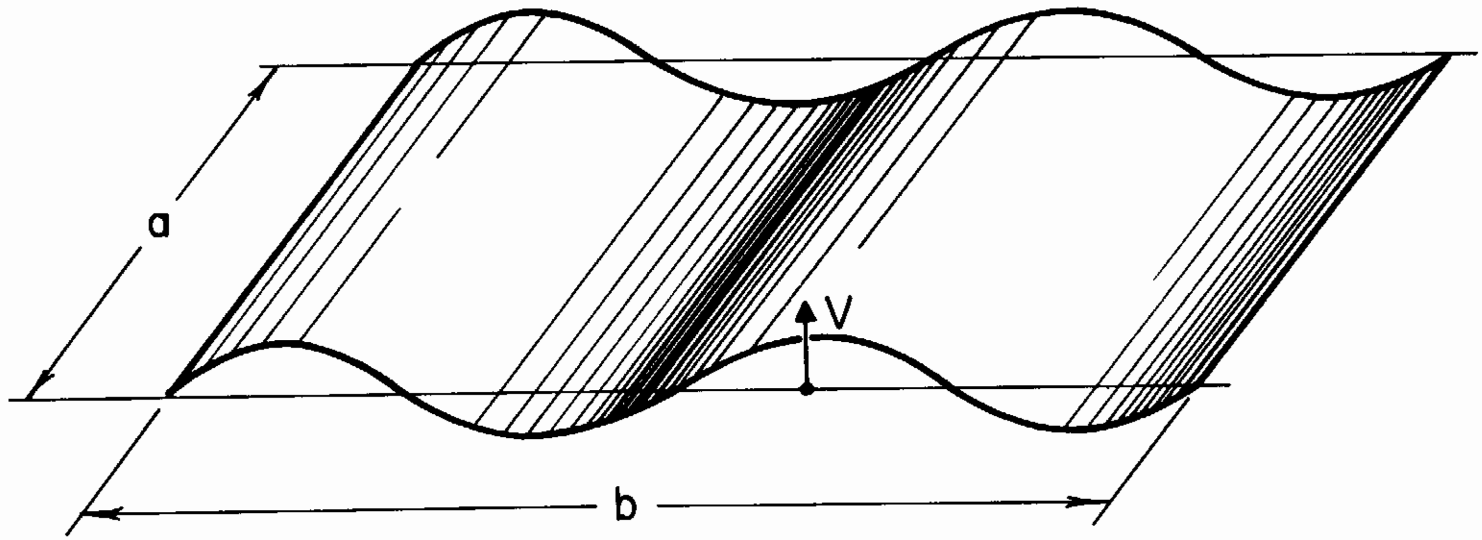

Fig. 5 - One Dimensional Standing Waves on Rectangular Plate

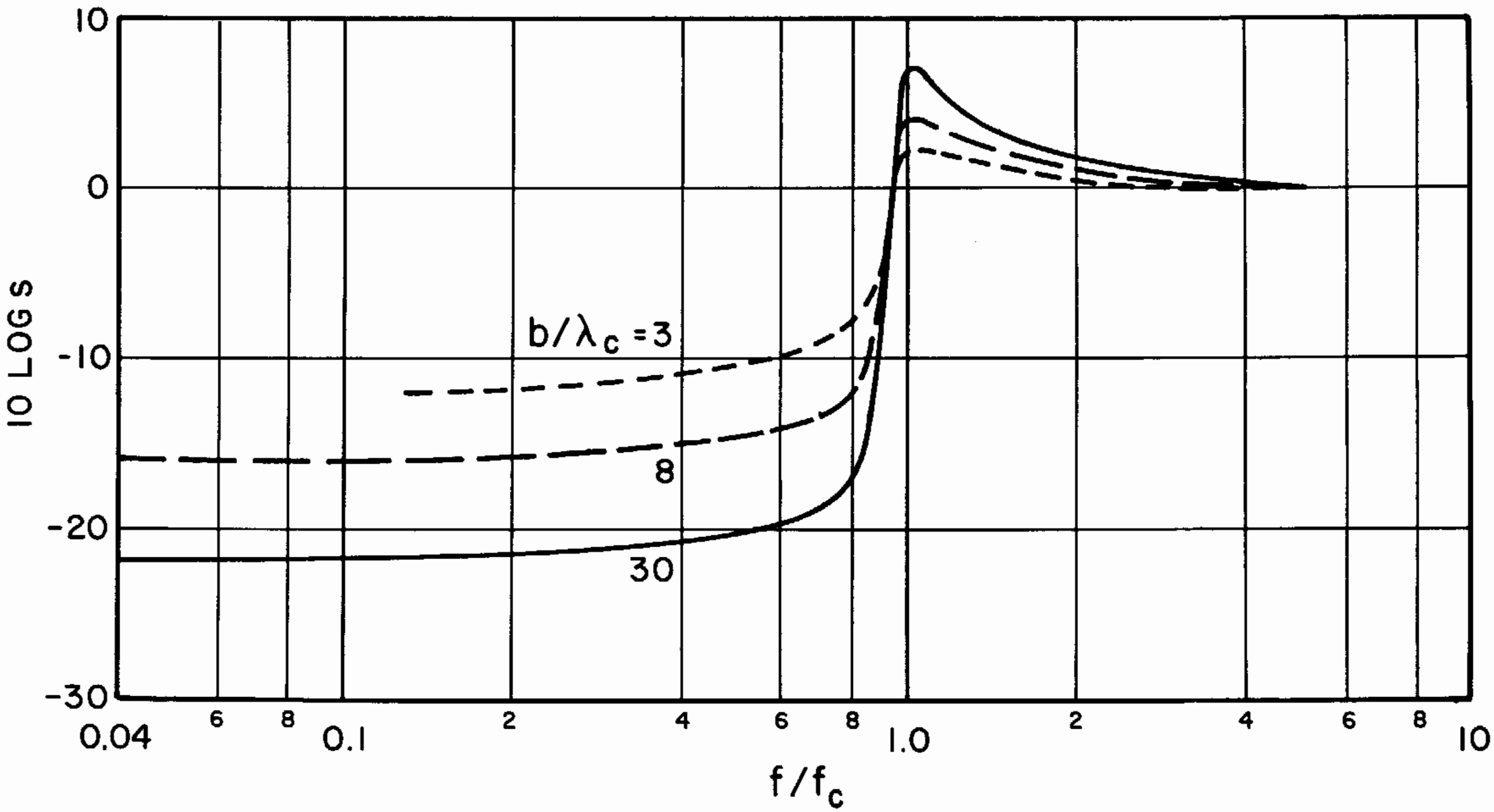

Fig. 6 - Radiation Factor for Travelling Waves on a Finite Plate

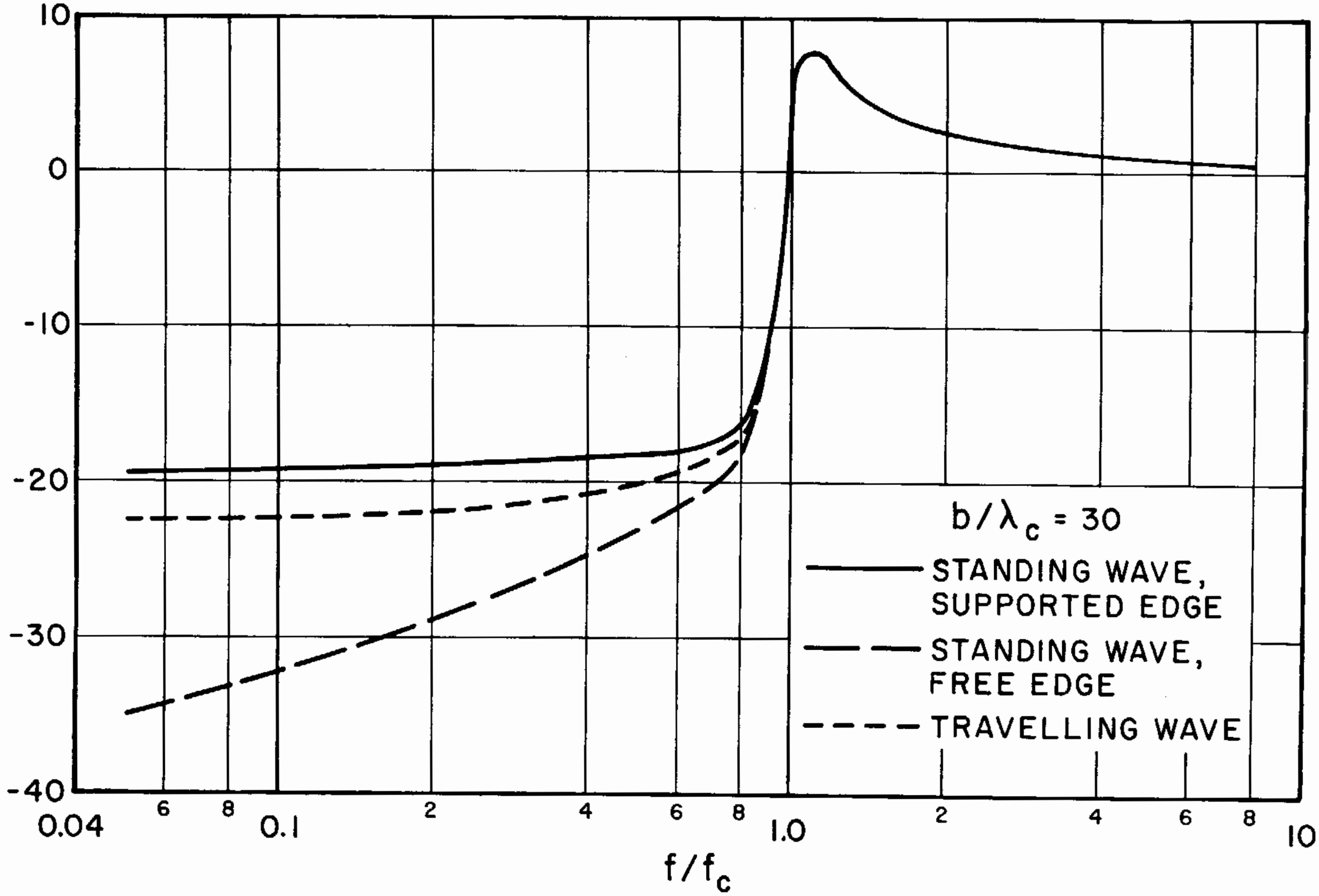

Fig. 7 - Radiation Factor for Standing and Travelling Waves on Finite Plate

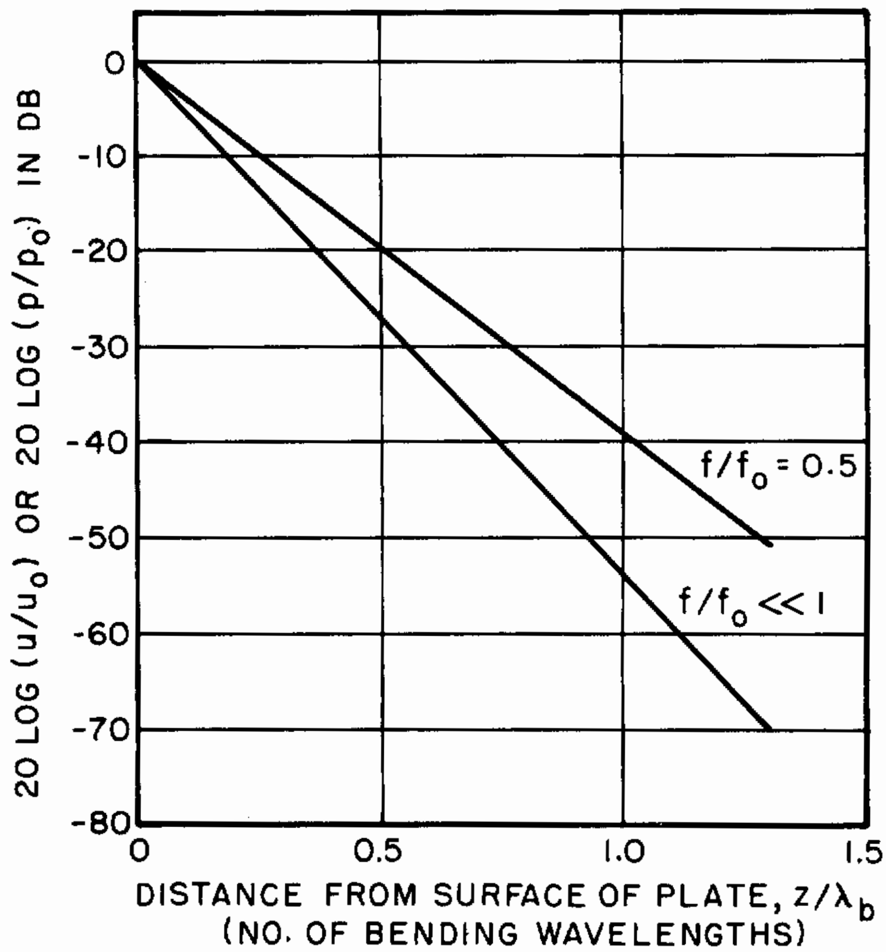

Fig. 8 - Decay of Near-Field Acoustic Pressure p and Velocity u With Distance From Surface of Plate. (p_o and u_o are values at plate surface)

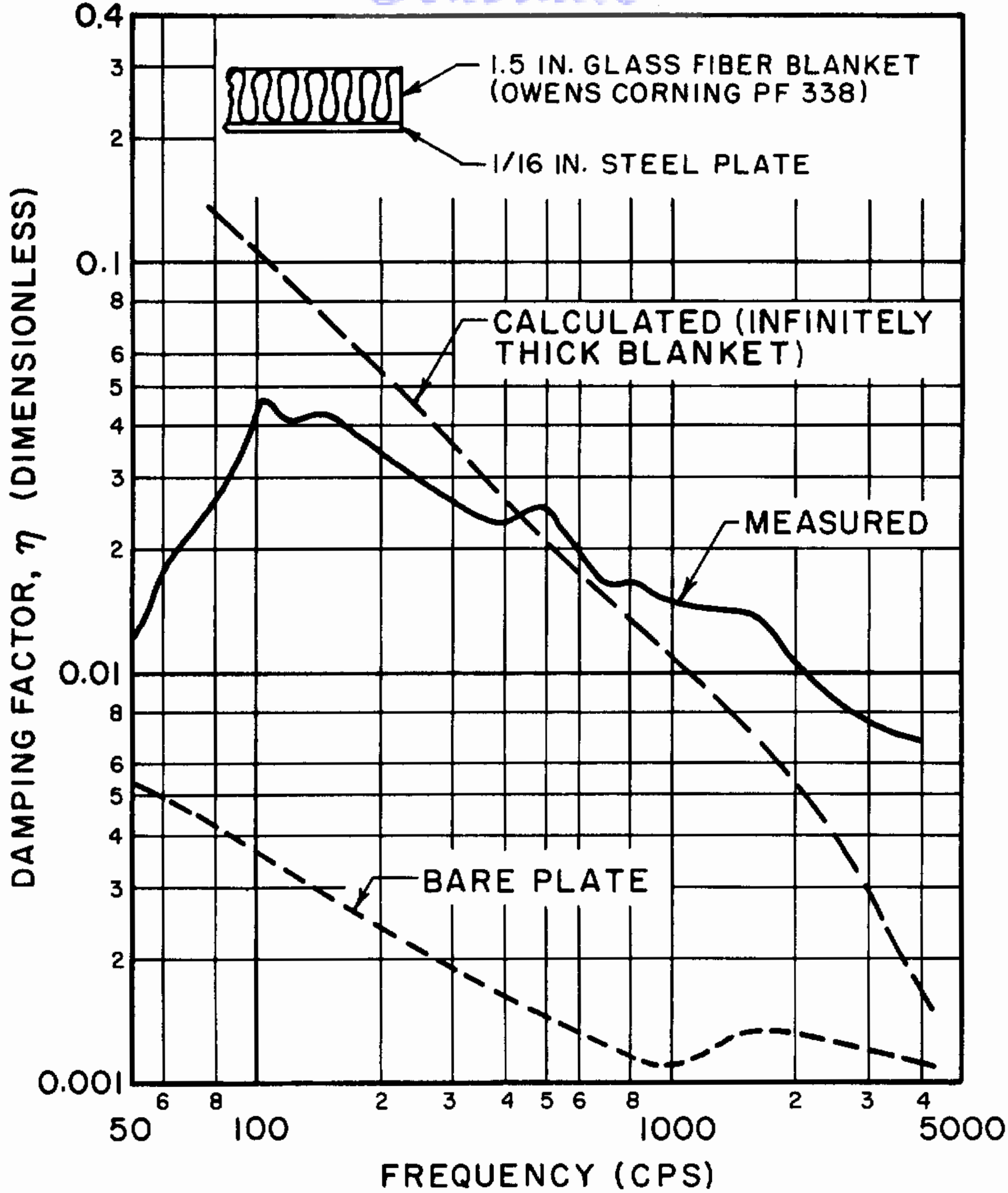

Fig 9 Measured and Calculated Damping Factor for Plate Damped by Fibrous Blanket

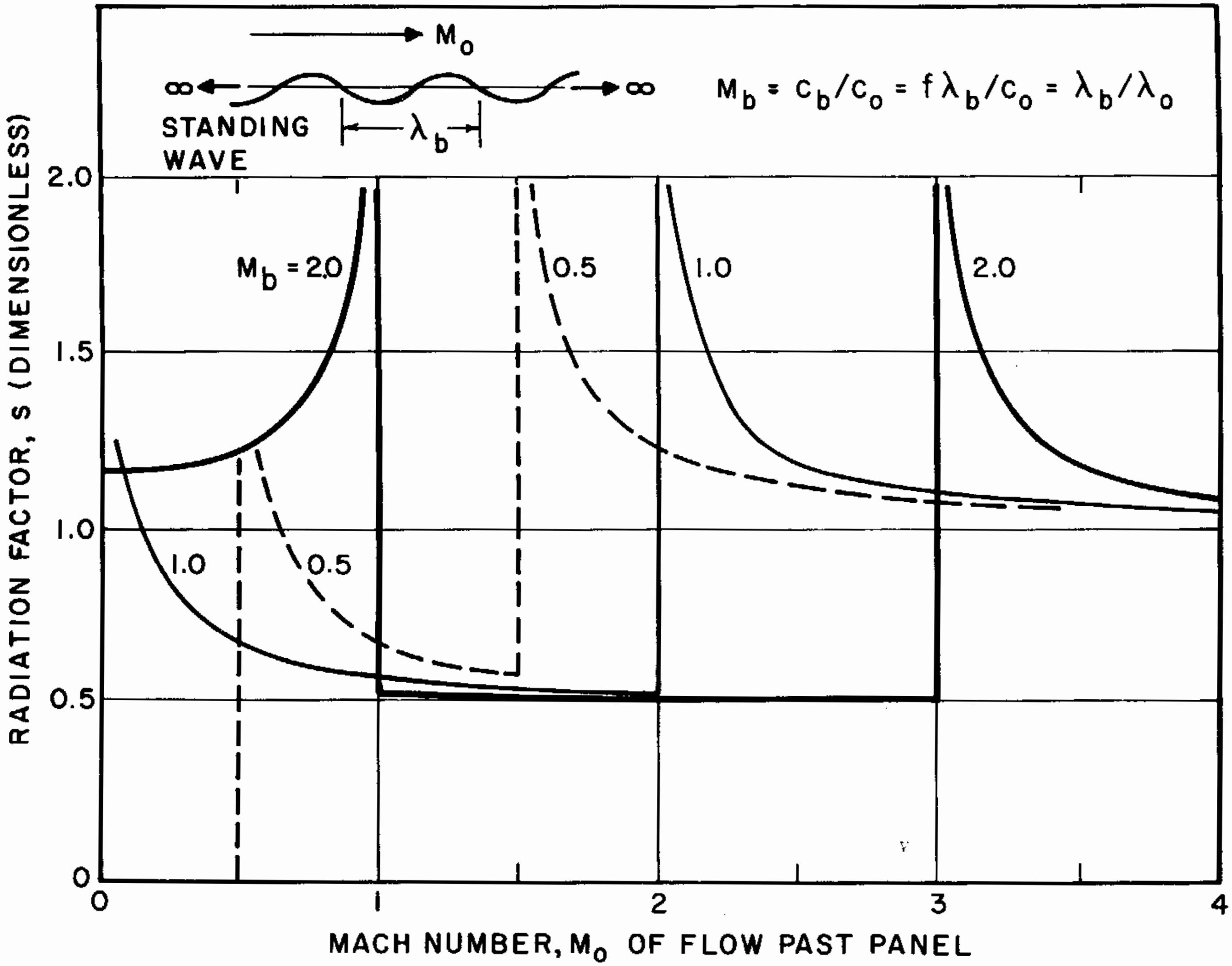

Fig 10 Effect of Fluid Flow on Radiation Factor s for Standing Flexural Waves on Infinite Plate

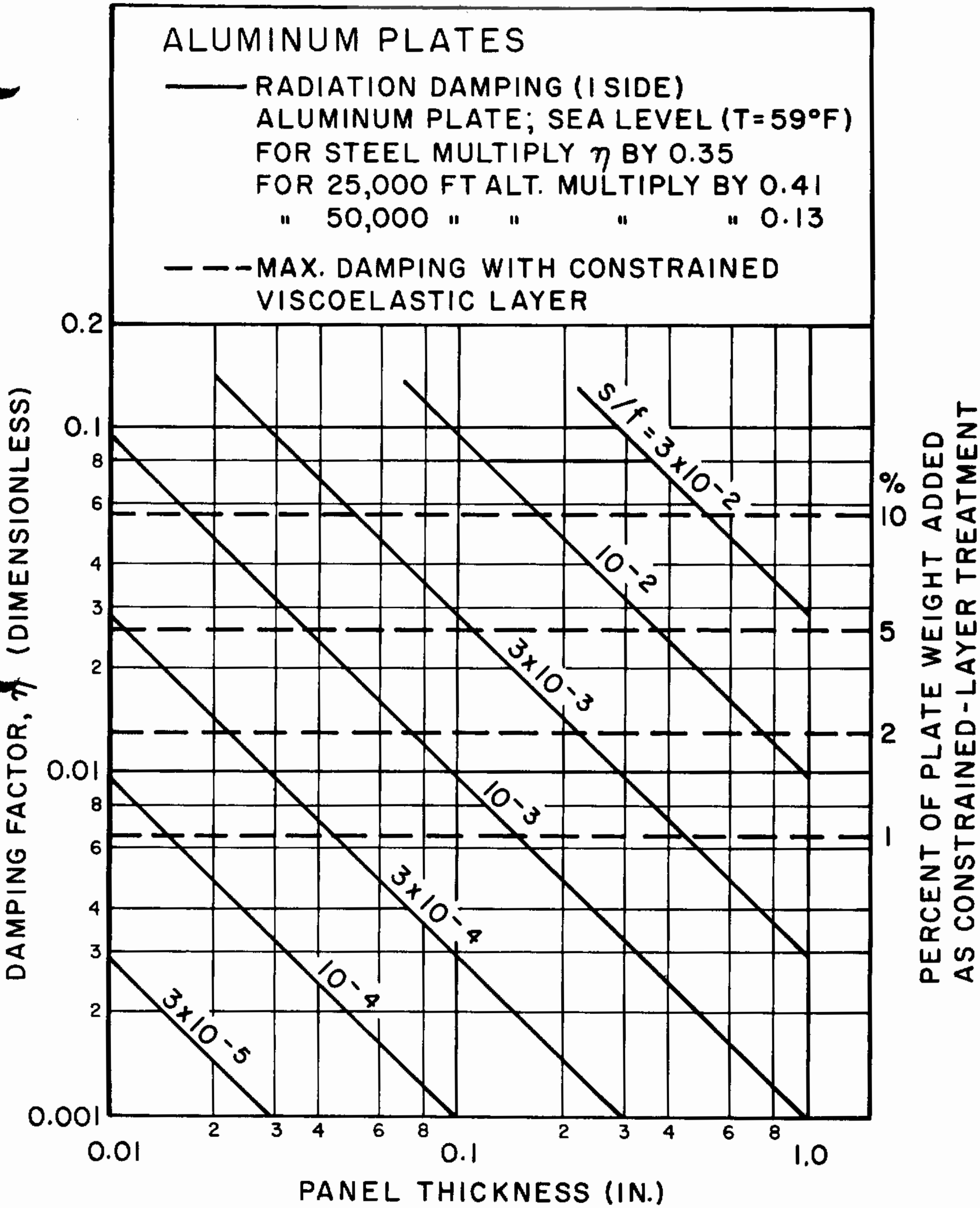

Fig. 11 - Comparison of Estimated Maximum Plate Damping Factors for Radiation Damping and for Applied Constrained Viscoelastic Layers