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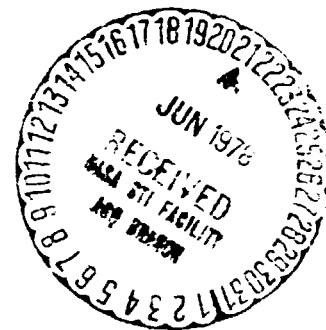
Low Frequency Cabin Noise Reduction Based on the Intrinsic Structural Tuning Concept

The Theory and the Experimental Results

G. SenGupta

March 1978

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16 Abstract A fundamental concept called Intrinsic Structural Tuning for reducing low frequency cabin noise and sonically induced stresses in an aircraft fuselage is presented. According to this concept, the low frequency response of the fuselage structure may be reduced by intrinsic tuning of the various structural members such as the skin, stringers, and frames and then applying damping treatments on these members. The concept has also been useful in identifying the key structural resonance mechanisms controlling the fuselage response to broadband random excitation and in developing suitable damping treatments for reducing the structural response in various frequency ranges. In this report, the mathematical proof of the concept and the results of some laboratory and field tests on a group of skin-stringer panels are described. The results of this study show that in the so-called <i>stiffness-controlled</i> region, the noise transmission may actually be controlled by stiffener resonances, depending upon the relationship between the natural frequencies of the skin bay and the stiffeners. Therefore, cabin noise in the <i>stiffness-controlled</i> region may be effectively reduced by applying damping treatments on the stiffeners.					
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LOW FREQUENCY CABIN NOISE REDUCTION BASED ON THE INTRINSIC STRUCTURAL TUNING CONCEPT

THE THEORY AND THE EXPERIMENTAL RESULTS

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1.0 SUMMARY

The fuselage of an aircraft is exposed to convected random pressure fields caused by noise from the jet engines and the turbulent boundary layer. In addition to satisfying strength, durability, and fail safe requirements, the fuselage structure should be designed to have a high transmission loss at low and mid frequencies and a long sonic fatigue life, all at low weight and cost. For this purpose, a concept called Intrinsic Structural Tuning has been developed. According to this concept, low frequency cabin noise and sonically induced stresses can be reduced by matching the first natural frequency of the skin bay, bounded by frames and stringers, with the corresponding natural frequency of the stringer segment supported between two frames, and then applying damping treatment on the stringer flanges.

In this report, the mathematical proof of the concept and the results of some laboratory test and field test on a group of skin-stringer panels are described. The initial analytical work and the laboratory tests were conducted as a part of the Independent Research and Development (IR&D) program of The Boeing Company. The field test was conducted in conjunction with the full-scale test of the YC-14 upper surface blown (USB) propulsion system, under the joint NASA/USAF contract.

A good correspondence between the analytical predictions and the results of the laboratory and field tests has been observed. The following points have emerged from the study:

1. Analytical studies indicate that when the structural elements are intrinsically tuned, the response of a skin-stringer panel does not pass through a peak at a frequency close to the fundamental frequency (f_p) of the individual skin bay clamped along the stringers and simply supported along the frames. On the contrary, the response of the panel is reduced considerably around this frequency. This has been verified by laboratory and field tests.

Analytical studies also indicate that two other modes appear at frequencies above and below the frequency. The responses of both these modes can be reduced by applying damping treatment on the stringers. This has been verified by the laboratory and the field tests.

2. Laboratory test data indicate that with no external damping treatment, the panel damping loss factors of the low frequency modes pass through a maximum when the skin and stringers are intrinsically tuned.

Laboratory test data also indicate that in the absence of any external damping treatment, the skin stringer panel response of the most dominant low frequency mode passes through a point of diminishing return when the tuning condition is satisfied.

3. A significant reduction of low frequency noise radiation and bending stress response can be achieved by designing the structure so that the skin panel frequency is higher than the stringer frequency and then applying damping treatment on the stringer flanges. This was demonstrated by both the laboratory and field test results which were in good agreement with the theoretical analysis.
4. Good agreement was obtained between the predicted and measured natural frequencies and mode shapes. The predicted and measured root mean square (rms) velocity responses at the skin panel center were compared for the most dominant low frequency modes and an agreement was obtained within a factor of two. The predictions were based on the assumption that the test panels were infinitely long. Considering this and other factors that were not taken into account in the simplified theory, the agreement is reasonable. A better agreement could be obtained by including the above factors in a more detailed analysis.
5. The past attempts on reducing low frequency cabin noise were based on increasing the structural stiffness since it was believed that low frequency cabin noise was controlled by the fuselage structural stiffness. The present study has pointed out that reduction of low frequency cabin noise by using this approach follows a law of diminishing return. Secondly, in what is usually regarded as a *stiffness controlled* region, the noise transmission may actually be controlled by stiffener resonances, depending upon the relationship between the natural frequencies of the skin bay and the stiffeners. Therefore, cabin noise in the *stiffness controlled* region may be effectively reduced by applying damping treatment on the stiffeners.

Low frequency cabin noise and sonically induced stresses can, therefore, be effectively reduced by using the following criteria based on the Intrinsic Structural Tuning concept. Damping should be applied on the skin if the skin-bay frequency is lower than the stringer frequency. Damping can be applied on the skin and on the stringers, if the structure is intrinsically tuned. For maximum low frequency noise reduction, the structure should be designed so that the skin frequency is greater than the stringer frequency, and then damping should be applied on the stringers. Additional skin damping tapes can also be applied to the skin for additional benefit in the mid frequency range.

2.0 INTRODUCTION

The fuselage of an aircraft is exposed to convected random pressure fields caused by noise from the jet engines and the turbulent boundary layer. Because of high power and light weight requirements, the aircraft designer is severely challenged to achieve acceptable interior noise levels and sonic fatigue life. In the past, cabin design for minimum noise levels normally was based on experience from past airplane design, inflight measurement, and subsequent application of acoustic treatment to meet some pre-determined noise level. Application of acoustic treatments always follow a law of diminishing return. Therefore, the need for designing the fuselage structure to act as an efficient noise barrier with a satisfactory sonic fatigue life is becoming more and more apparent with current trends in aircraft design toward maximum structural efficiency and fuel economy. Although commercial transports do not experience sonic fatigue on fuselage structure, this can be a problem in STOL and military aircraft.

Control of low frequency interior noise has been difficult in all commercial and general aviation aircraft. The YC-14 upper surface blown (USB) and the YC-15 externally blown flap (EBF) STOL aircraft are expected to have higher levels of low frequency interior noise and sonically induced loads, because of the proximity of the engines to the fuselage, dominance of low frequency components in the USB or EBF environment, and the high degree of convection and correlation of the fluctuating pressure field exciting the fuselage structure, possibly under coincidence conditions (refs. 1 and 2). Control of low frequency interior noise in the fuel efficient, propfan aircraft is also likely to be a difficult task. The existing sound attenuation techniques (i.e., application of lead vinyl and fiberglass insulation) are less effective at low frequencies. Therefore, low frequency interior noise and sonically induced stresses can mainly be reduced by a proper design of the fuselage structure and by making the fuselage structure an integral part of the sound proofing system.

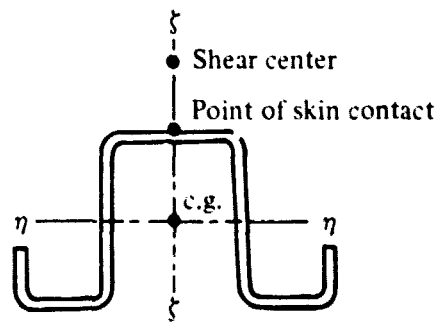
For this purpose, a concept called Intrinsic Structural Tuning, for designing a fuselage structure with reduced cabin noise and sonically induced stresses was developed (ref. 3). According to this concept, the low frequency response and sound radiation from the skin panel can be reduced by designing the structure so that the skin panel and stringers are intrinsically tuned to each other, and then applying damping treatment on the stringer flanges. The theory is discussed in detail in section 4.0. Based on this theory, four skin-stringer panels were built and tested in the laboratory. In that test, the panels were excited by a set of electromagnetic shakers, connected in series to simulate highly coherent near field jet noise and fed by current from a white noise generator. The results of the laboratory test confirmed the analytical predictions. These results are summarized in section 5.0.

Three of these panels were then exposed to the near field noise environment generated by the full-scale, YC-14 engine-wing-flap test rig, under a joint NASA/USAF contract. The details of this field test and the primary results are presented in sections 6.0 and 7.0. The final conclusions are presented in section 8.0.

3.0 SYMBOLS AND ABBREVIATIONS

A, B, C, D	Coefficients in equation (12)
$\{A\}$	A column matrix consisting of A, B, C, D , equation (15)
A_{st}	Cross sectional area of the stringer
a	Stringer spacing
b	Frame spacing
C_{ws}	Warping constant of the stringer cross section, about the point of skin contact
C_t	Trace velocity of the excitation field
c, c_c	Damping constant, critical damping
D_{sk}^*	$= D_{sk} (1 + i\eta_{sk})$
D_{sk}	Skin flexural rigidity
E_{sk}, E_{st}	Young's moduli for skin and stringer materials, respectively
$[E(x)]$	A matrix as defined in equation (15)
$[E(a)]$	$= [E(x)]$ at $x = a$
f_c	Frequency of acoustic coincidence excitation
f_p	Fundamental frequency of the basic panel of dimensions $a \times b$, with clamped edges at the stringers and simply - supported edges at the frames
f_s	Fundamental bending frequency of the stringer, simply - supported at the frames
f_{ST}	Fundamental torsional frequency of the stringer simply - supported at the frames
G_{st}	Shear modulus of the stringer material
$[I]$	Identity matrix
I_s	Polar moment of inertia of the stringer cross section about the point of skin contact
I_η	Moment of inertia of the stringer cross section in bending
i	$= \sqrt{-1}$
J_{st}	St. Venant constant of uniform torsion for the stringer cross section
k	Wave number $= \frac{\omega}{C_t}$
K_R	Dynamic rotational stiffness of the stringer
K_T	Dynamic bending stiffness of the stringer
M_1, M_2	Bending moments at stations 1 and 2, respectively (fig. 1b)
n	Number of flexural half waves between two frames
P	$= P_O / \{ D_{sk}^* (k^2 + \lambda_1^2) (k^2 - \lambda_2^2) \}$

p_0	Amplitude of the pressure wave, assumed to be unity in the computation
$[S]$	A matrix, dependent on stringer bending and rotational stiffnesses
S_1, S_2	Shear forces at stations 1 and 2, respectively (fig. 1b)
$S_v(\omega), S_M(\omega), S_p(\omega)$	Power spectral densities of velocity response, bending moment response, and pressure field, respectively
$S_p(\xi, \omega)$	Cross spectral density of the excitation at two points separated by ξ
t	Skin thickness
w	Deflection
x	Coordinate perpendicular to the stringer axis
y	Coordinate parallel to the stringer axis
$\{Z\}, \{Z_0\}, \{Z_1\}, \{Z_2\}$	State vectors at x , and at stations 0, 1, 2 respectively (fig. 1b)
$\xi - \xi$	Axis of the stringer passing through its c.g., and perpendicular to the skin
$\eta - \eta$	Axis of the stringer passing through its c.g., and parallel to the skin
η_{sk}	Skin loss factor
θ_1, θ_2	Slopes at stations 1 and 2, respectively, (fig. 1b)
λ	$= \left(\frac{\rho_{sk} t}{D_{sk*}} \right)^{1/4} \omega^{1/2}$
λ_1^2	$= \lambda^2 + \left(\frac{\pi}{b} \right)^2$
λ_2^2	$= \lambda^2 - \left(\frac{\pi}{b} \right)^2$
ν	Poisson's ratio of the skin material
ξ	Distance of separation
π	$= 3.14159265$
ρ_{sk}, ρ_{st}	Densities of skin and stringer materials, respectively



4.0 THEORY

During takeoff, the aft section of an aircraft with wing-mounted engines is subjected to near field noise from the jet engine operating at maximum power. The incident jet noise generates structural wave motion in the skin. These structural waves in turn excite acoustic waves in the cabin interior. When the convection or the trace velocity of the pressure field coincides with the natural flexural wave speed in the stiffened fuselage skin at a particular frequency, large *coincidence* peaks appear at that frequency in the structural response and radiated noise spectra. A simplified skin stringer model (fig. 1a) consisting of an infinitely long, flat and unpressurized skin, simply supported along the frames and stiffened by stringers at regular intervals is often used to predict the natural frequency distribution and the response spectra. In this model, the stringers are also assumed to be simply supported at the frames. For the sake of simplicity, the same model will be used in the present analysis. However these simplifying assumptions are not in any way essential for the concept to be applicable in a more general situation.

When the flexural waves travel along the length of the structure and meet a discontinuity such as provided by a stringer, a part of the energy associated with the waves is reflected back, a part is absorbed by the stringers executing torsional and bending vibration, and the remaining part is transmitted to the next panel (fig. 1b). This wave propagation mechanism in a periodic structure has been studied extensively (refs. 4 through 25). To summarize, a periodic skin-stringer structure acts like a band pass filter in certain frequency zones, bounded by *out-of-phase* and *in-phase* modes of the stiffened structure. Within these frequency bands, known in the literature as *free propagation zones* (FPZ), a special class of free flexural waves can be excited under coincidence conditions (fig. 2). The coincidence frequencies fall within the free propagation zones. Usually, the peak at the lowest coincidence frequency is the largest and is of major concern. Outside the free propagation zones, no coincidence excitation of flexural waves is possible.

It should be noted that these free propagation zones are the characteristics of periodically stiffened structures only. In an infinite beam without any stiffener, coincidence excitation is theoretically possible at a single frequency for a given trace velocity (fig. 3). In an infinitely long unstiffened strip of a finite width supported at the frames, coincidence excitation is possible at two frequencies, due to a given convection velocity, for a given number of half-waves between the frames (fig. 4). The lower frequency, in most cases, is close to, or of the order of the natural frequency of a finite beam of the same thickness, supported at the frames. The higher frequency is close to the coincidence frequency that would be calculated by treating the strip as an infinitely long beam.

The presence of the stringers, therefore, drastically alters the range of frequencies within which structural waves may be excited under coincidence condition. At the coincidence frequency (f_c), the waves reflected by any two successive stringers reinforce each other and the amplitude of panel vibration builds up to a large value. For an infinite trace velocity, the coincidence (f_c) is close to the natural frequency (f_p) of the panel with clamped edges at the stringers and simply supported edges at the frames (fig. 2). The process of reflection of energy by the stringers depends on the stringer stiffness in bending and torsion. In order to understand this process, let us take a look at a flexural wave as it

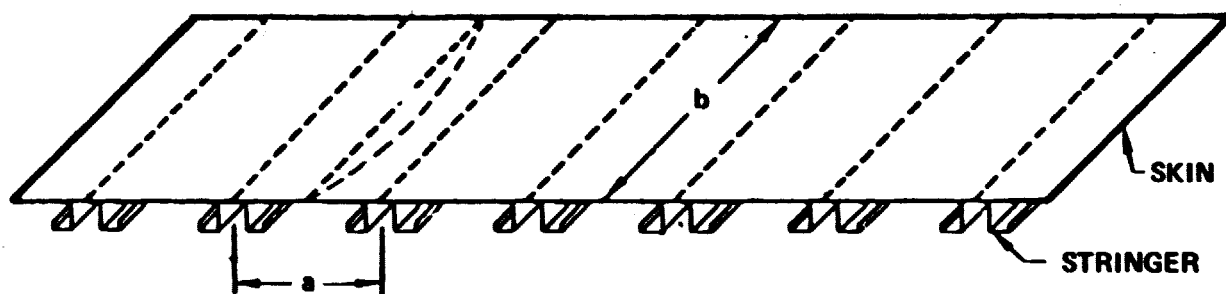


Figure 1a.—Periodic Skin - Stringer Model

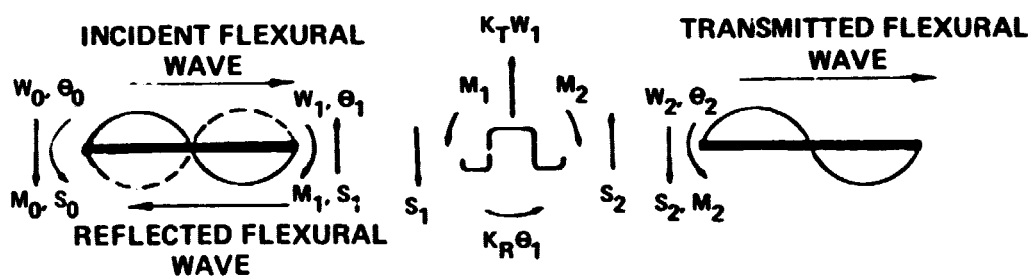


Figure 1b.—Reflection and Transmission of Flexural Waves by a Stringer

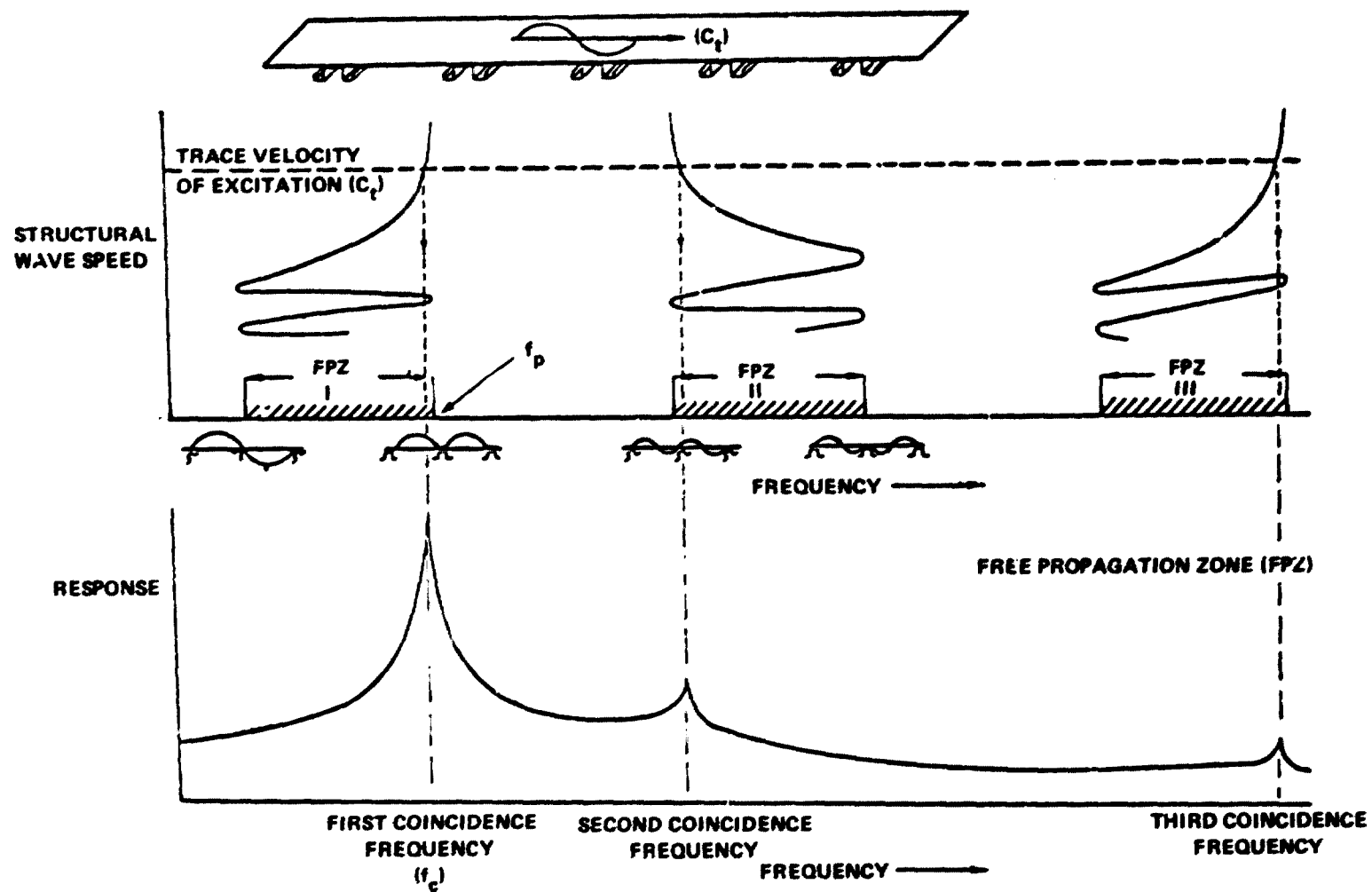


Figure 2.—Mechanism of Coincidence Excitation of a Periodically Stiffened Pane.

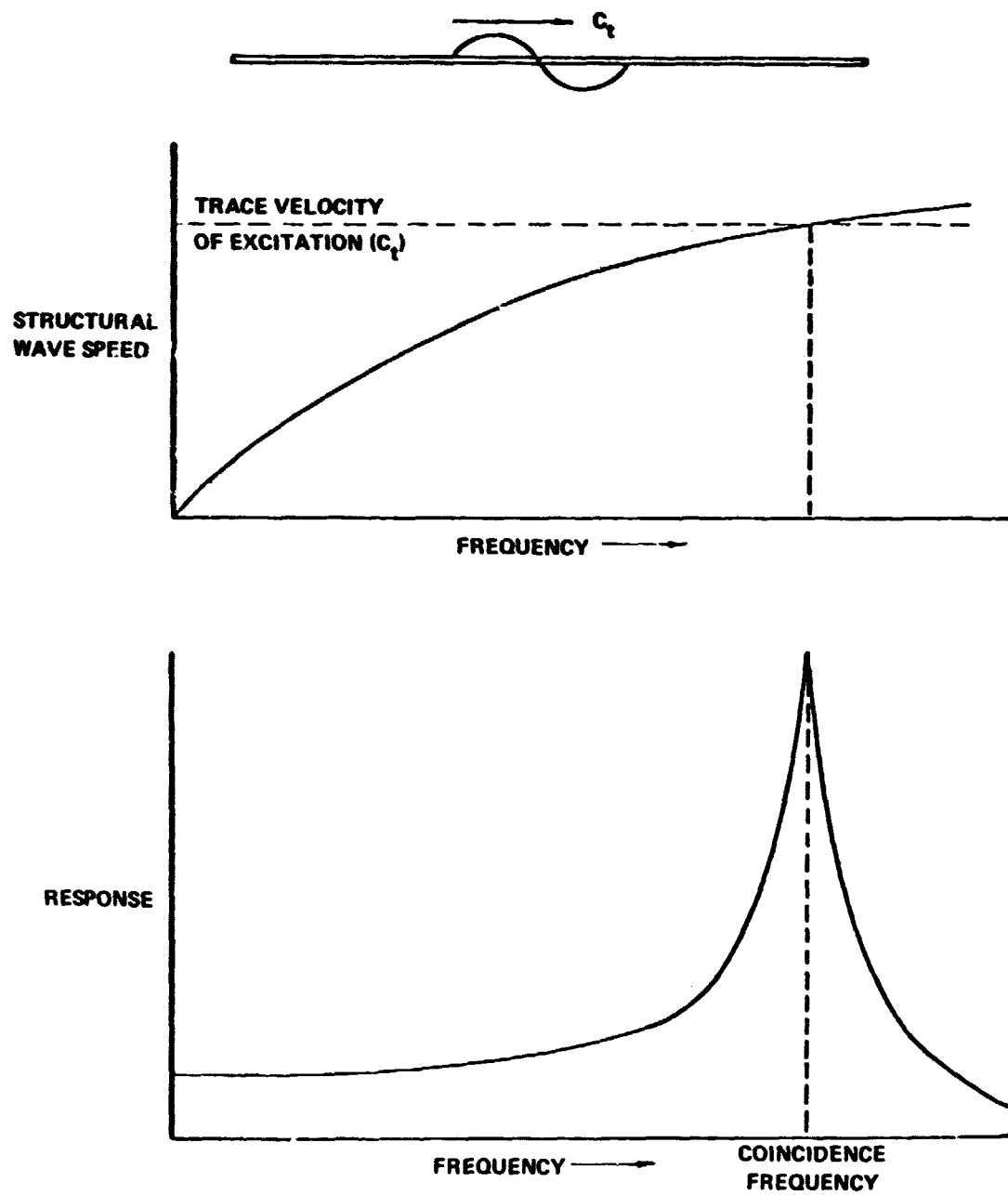


Figure 3.—Coincidence Excitation of an Infinitely Long Beam

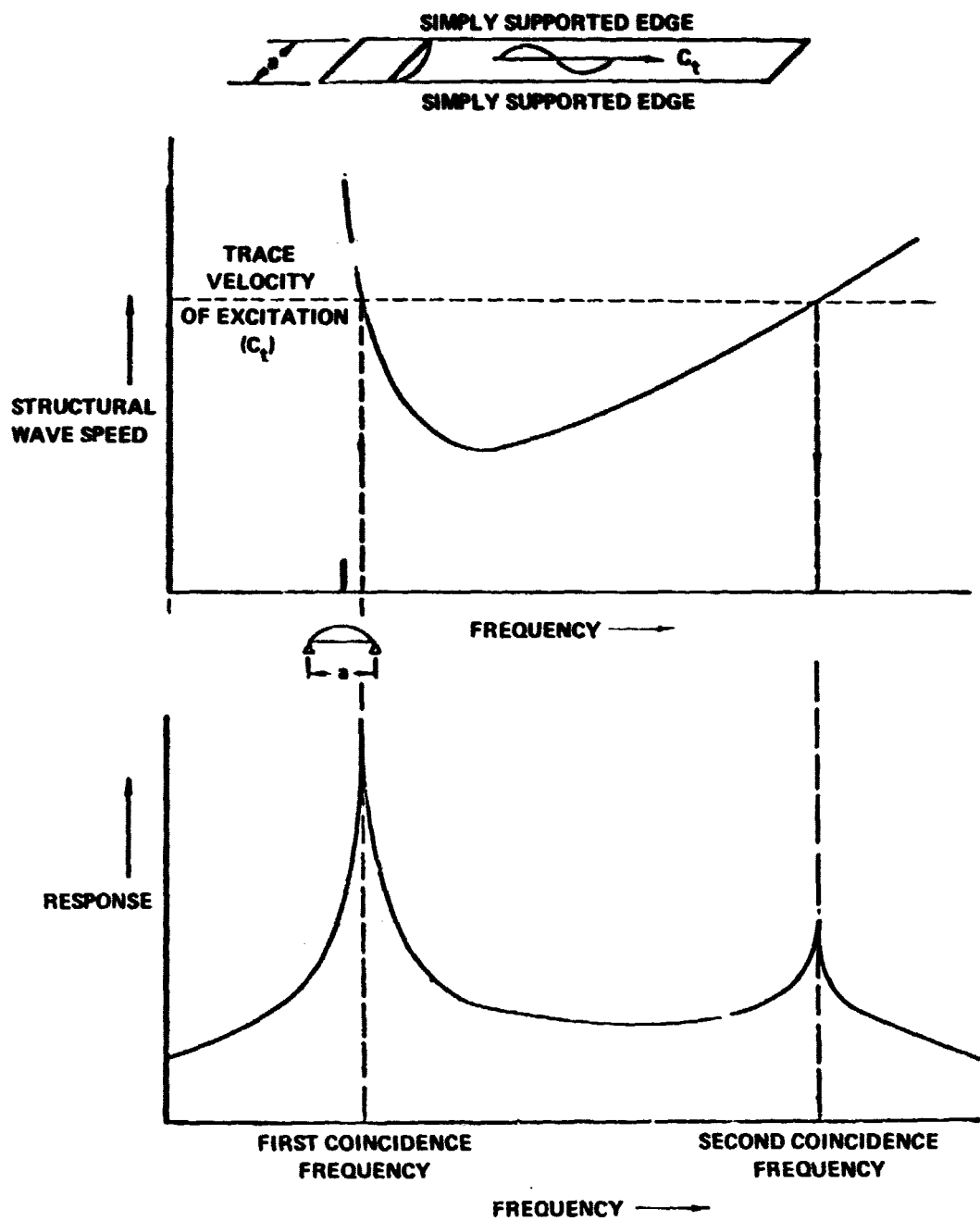


Figure 4.—Coincidence Excitation of an Infinite Strip of Finite Width

passes from one side of the stringer to the other. The displacement and the slope associated with this bending wave are the same on the two sides of a stringer because of continuity. Therefore, referring to fig. 1b,

$$\begin{aligned} w_2 &= w_1 \\ \theta_2 &= \theta_1 \end{aligned}$$

In addition, the moments and the shear forces on the two sides of a stringer will have to satisfy the equations of equilibrium.

Therefore,

$$\begin{aligned} M_2 - M_1 &= K_R \theta_1 \\ S_2 - S_1 &= -K_T w_1 \end{aligned}$$

where K_R and K_T are the rotational and translational stiffnesses, respectively, of unit length of the stringer. These equations apply strictly to stringers of symmetric cross section, such as a top-hat section. Expressed in a matrix form, these equations reduce to

$$\begin{Bmatrix} w_2 \\ \theta_2 \\ M_2 \\ S_2 \end{Bmatrix} = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \\ 0 & K_R & 1 & 0 \\ -K_T & 0 & 0 & 1 \end{bmatrix} \begin{Bmatrix} w_1 \\ \theta_1 \\ M_1 \\ S_1 \end{Bmatrix} \quad (1)$$

In the present case, an excitation field with an infinite trace velocity has been assumed. Further simplification is possible under such conditions. For white noise at normal incidence, the trace velocity C_t is infinity, and the wave number $k (= \frac{\omega}{C_t})$ is zero at any frequency ω . This means that at any frequency, the adjacent panels vibrate in phase and the stringers cannot rotate. Then the slope at each stringer location becomes zero, and the following equation applies:

$$\begin{aligned} w_2 &= w_1 \\ M_2 &= M_1 \\ S_2 - S_1 &= -K_T w_1 \end{aligned} \quad (2)$$

The corresponding matrix equation is then obtained as:

$$\begin{Bmatrix} w_2 \\ M_2 \\ S_2 \end{Bmatrix} = \begin{bmatrix} 1 & 0 & 0 \\ 0 & 1 & 0 \\ -K_T & 0 & 1 \end{bmatrix} \begin{Bmatrix} w_1 \\ M_1 \\ S_1 \end{Bmatrix} \quad (3)$$

If the stringers were now removed, equation (3) would reduce to

$$\begin{Bmatrix} W_2 \\ M_2 \\ S_2 \end{Bmatrix} = \begin{bmatrix} 1 & 0 & 0 \\ 0 & 1 & 0 \\ 0 & 0 & 1 \end{bmatrix} \begin{Bmatrix} W_1 \\ M_1 \\ S_1 \end{Bmatrix} \quad (4)$$

The presence of the stringers then contributes to one element of the transfer matrix when the trace velocity is infinity. This term arises because of the stringer bending stiffness K_T .

It is now apparent that if, somehow, we could get rid of this particular element in the matrix, there would be no strong reflection (and hence no reinforcement) in the panel response. One way to achieve this would be to physically remove the stringers and build a stringerless structure. However, the stringers are generally required because of static strength considerations. The purpose of the present study is to present an alternative way, so that the effect of reflections caused by the stringers can be minimized.

In order to introduce this alternative way, let us look at the problems a little more closely. According to Lin (ref. 26), the expression for K_T for a stringer considered to be simply supported at the two successive frame locations is given by

$$K_T = E_{st} I_{\eta} \left(\frac{\pi}{b} \right)^4 - \rho_{st} A_{st} \omega^2 \quad (5)$$

where the various parameters are as defined in the List of Symbols. (In equation (5), K_T is identical to the parameter "e" of equation (14) of ref. 26. The notation K_T is used here for the sake of clarity, since it denotes the translational stiffness of the stringers.) K_T therefore is a function of frequency. As frequency increases, K_T decreases. At the fundamental natural frequency of the stringer in bending, K_T becomes zero. This frequency is obtained by setting $K_T = 0$ and is given by

$$f_s = \frac{1}{2\pi} \left(\frac{E_{st}}{\rho_{st}} \right)^{1/2} \left(\frac{I_{\eta}}{A_{st}} \right)^{1/2} \left(\frac{\pi}{b} \right)^2 \quad (6)$$

Therefore at this frequency (typically 300 to 400 Hz), the presence of the stringers is not felt by the skin, and the skin appears stringerless. At frequencies below f_s , K_T is positive. Above f_s , K_T is negative; i.e., stringer impedance is then primarily caused by its mass. The coincidence frequency f_c caused by an infinite trace velocity is generally close to the panel natural frequency f_p . For takeoff situations with no pressurization, f_p is typically 100 to 200 Hz. When the panel frequency is much smaller than the stringer frequency, the stringers appear too stiff, and there is a strong panel response at a frequency close to the panel natural frequency (f_p). On the other hand, if the stringer frequency is much smaller than the panel frequency the stringers appear too massive for the panel vibrating at f_p . Therefore, it should now be obvious that if the stringer and panel dimensions were chosen in such a way that the condition

$$f_p = f_s \quad (7)$$

was satisfied, the stringers would not offer any impedance to the bending waves propagating at f_p . The mechanism of build-up of flexural energy at the panel natural frequency would then be destroyed, and the large response at f_p would disappear. Therefore, under such conditions, the panel would act like a stringerless structure at the frequency f_p . If the stringers and the panels are designed so that they satisfy the usual static strength requirements and also are tuned to each other according to equation (7), we will then have a structure that is stiffened for static purposes and stringerless from the dynamic point of view; i.e., the structure will act efficiently under both static and dynamic loading conditions.

It should be pointed out that equation (7) gives the condition to be satisfied when the trace velocity of excitation is very high. This would usually be the situation in most cases. However, if the excitation trace velocity is such that there is coincidence excitation of the *stringer torsion mode* of the stiffened panel (fig. 2), the situation becomes entirely different. (In this mode, the skin bays on the two sides of a stringer vibrate out of phase, with the stringers undergoing torsional oscillation.) The stringer torsional stiffness K_R then plays the most important part in impeding the flexural waves. The panel deflection at the stringers being negligible, the effect of the stringer bending stiffness K_T is no longer significant at this frequency. The expression for K_R from reference 26, is given by

$$K_R = E_{st}C_{ws} \left(\frac{\pi}{b}\right)^4 + G_{st}J_{st}\left(\frac{\pi}{b}\right)^2 - \rho_{st}I_s \omega^2 \quad (8)$$

where the parameters are as defined in the List of Symbols. (In this equation, K_R is related to the parameter "C" of equation (14) of ref. 26. The notation K_R is used here for the sake of clarity, since it denotes the rotational stiffness of the stringers.) It is seen that K_R is frequency-dependent and it goes to zero at

$$f_{ST} = \frac{1}{2\pi} \left[\left(\frac{E_{st}}{\rho_{st}} \right) \left(\frac{C_{ws}}{I_s} \right) \left(\frac{\pi}{b} \right)^4 + \left(\frac{G_{st}}{\rho_{st}} \right) \left(\frac{J_{st}}{I_s} \right) \left(\frac{\pi}{b} \right)^2 \right]^{1/2} \quad (9)$$

Proceeding in the same manner, it can be shown that in this case the stringer torsion frequency of the stiffened panel. In general, if the coincidence frequency is somewhere in between the *out-of-phase* mode (stringer torsion) and *in-phase* mode (stringer bending) frequencies of the stiffened panel, the stringers and the panel should be designed so that K_R and K_T go to zero around that frequency.

It should be noted that in all these cases, the behavior of the stringers is analogous so that of the tuned dampers. However, in the present case the stringers suppress the coincidence excitation of flexural waves and, therefore, they may be viewed as acting like tuned wave dampers. A stiffened structure in which the components are tuned to each other may be called an *Intrinsically Tuned Structure*.

Based on these ideas, a theory for verifying the concept was developed. This is reproduced in section 4.1.

4.1 DERIVATION

The theory uses the simplified skin-stringer model shown in figure 1. Let us consider the response of this structure to random acoustic plane waves simulating jet noise, travelling

with a trace velocity C_t along the length of the panel. The spectral density of the acoustic pressure at any point on the structure is denoted by $S_p(\omega)$. The cross-spectral density of the pressures at two points on the structure, separated in the streamwise direction by ξ , is

$$S_p(\xi, \omega) = S_p(\omega) \cdot e^{-i\frac{\omega}{C_t}\xi}$$

where ω is the circular frequency.

The power spectral density (PSD) of skin velocity response $S_v(\omega)$ or skin bending moment response $S_M(\omega)$ is related to that of the excitation $S_p(\omega)$, in the following manner:

$$\begin{aligned} S_v(\omega) &= |Y_v(\omega)|^2 S_p(\omega) \\ S_M(\omega) &= |Y_M(\omega)|^2 S_p(\omega) \end{aligned} \quad (10)$$

where $Y_v(\omega)$ and $Y_M(\omega)$ are the response admittance functions; i.e., velocity and bending moment response of the skin caused by a travelling sinusoidal pressure field

$$e^{i(\omega t - \frac{\omega}{C_t} x)}$$

of unit amplitude.

Section 4.2 describes how $Y_v(\omega)$ or $Y_M(\omega)$ is determined. It is assumed (for simplicity in this presentation) that the pressure and associated responses vary across the skin in proportion to $\sin \frac{\pi y}{b}$. The analysis is then carried out in terms of the amplitudes of the fundamental components of the pressure and responses. The total response over a wide frequency range is, of course, given by the summation of all the components associated with all the half waves between the frames.

4.2 RESPONSE TO AN ACOUSTIC PLANE WAVE

A harmonic pressure field of frequency ω and wave number $k = \frac{\omega}{C_t}$ convected along the structure exerts pressures of equal amplitude to all points. But at points separated by "a" (equal to the stringer spacing which determines the periodic length) in the direction of propagation, the pressures have the phase difference "ka". The responses of the infinitely long structure, at these two points are also equal in magnitude and also have the same phase difference. Therefore, it follows that the deflection (W), slopes (θ), bending moments (M), and shear forces (S) and the points 0 and 2 (fig. 1) must be related by

$$\begin{aligned} W_2 &= W_0 e^{-ika} \\ \theta_2 &= \theta_0 e^{-ika} \\ M_2 &= M_0 e^{-ika} \\ S_2 &= S_0 e^{-ika} \end{aligned} \quad (11)$$

Let us now consider the equation of motion of the skin panel

$$\gamma^4 W + \frac{\rho_{sk} t}{D_{sk}^*} \ddot{W} = \frac{p(x,y,t)}{D_{sk}^*}$$

Since the skin edges at the frames are simply supported, the solution of this equation is given by

$$W = Ae^{\lambda_1 x} + Be^{-\lambda_1 x} + Ce^{i\lambda_2 x} + De^{-i\lambda_2 x} + \frac{p_0 e^{-ikx}}{D_{sk}^* (k^2 + \lambda_1^2) (k^2 - \lambda_2^2)} \quad (12)$$

where:

$$\lambda_1^2 = \lambda^2 + \left(\frac{\pi}{b}\right)^2$$

$$\lambda_2^2 = \lambda^2 - \left(\frac{\pi}{b}\right)^2$$

$$\lambda^4 = \left(\frac{\rho_{sk} t}{D_{sk}^*}\right) \omega^2$$

It should be noted that the coefficients A, B, C, and D are associated with the waves reflected by the stringers.

From equation (12) for the deflection, the equations for the slope, bending moment, and shear force at any point in the panel can be derived. Expressed in a matrix form,

$$\begin{Bmatrix} W \\ \theta \\ M \\ S \end{Bmatrix} = \begin{bmatrix} e^{\lambda_1 x} & e^{-\lambda_1 x} & e^{i\lambda_2 x} & e^{-i\lambda_2 x} \\ \lambda_1 e^{\lambda_1 x} & -\lambda_1 e^{-\lambda_1 x} & i\lambda_2 e^{i\lambda_2 x} & -i\lambda_2 e^{-i\lambda_2 x} \\ \lambda_1^2 e^{\lambda_1 x} D_{sk}^* & \lambda_1^2 e^{-\lambda_1 x} D_{sk}^* & -\lambda_2^2 e^{i\lambda_2 x} D_{sk}^* & -D_{sk}^* \lambda_2^2 e^{-i\lambda_2 x} \\ \lambda_1^3 e^{\lambda_1 x} D_{sk}^* & -\lambda_1^3 e^{-\lambda_1 x} D_{sk}^* & -i\lambda_2^3 e^{i\lambda_2 x} D_{sk}^* & D_{sk}^* i\lambda_2^3 e^{-i\lambda_2 x} \end{bmatrix} \begin{Bmatrix} A \\ B \\ C \\ D \end{Bmatrix} + \frac{p_0 e^{-ikx}}{D_{sk}^* (k^2 + \lambda_1^2) (k^2 - \lambda_2^2)} \begin{Bmatrix} 1 \\ -ik \\ -D_{sk}^* k^2 \\ D_{sk}^* ik^3 \end{Bmatrix} \quad (13)$$

or

$$\begin{Bmatrix} W \\ \theta \\ M \\ S \end{Bmatrix} = \begin{bmatrix} 1 & 1 & 1 & 1 \\ \lambda_1 & -\lambda_1 & i\lambda_2 & -i\lambda_2 \\ \lambda_1^2 D_{sk}^* & \lambda_1^2 D_{sk}^* & -\lambda_2^2 D_{sk}^* & -\lambda_2^2 D_{sk}^* \\ \lambda_1^3 D_{sk}^* & -\lambda_1^3 D_{sk}^* & -i\lambda_2^3 D_{sk}^* & i\lambda_2^3 D_{sk}^* \end{bmatrix} \begin{bmatrix} e^{\lambda_1 x} & 0 & 0 & 0 \\ 0 & e^{-\lambda_1 x} & 0 & 0 \\ 0 & 0 & e^{i\lambda_2 x} & 0 \\ 0 & 0 & 0 & e^{-i\lambda_2 x} \end{bmatrix} \begin{Bmatrix} A \\ B \\ C \\ D \end{Bmatrix} \\ + \frac{p_0 e^{-ikx}}{D_{sk}^* (k^2 + \lambda_1^2) (k^2 - \lambda_2^2)} \begin{Bmatrix} 1 \\ -ik \\ -D_{sk}^* \\ D_{sk}^* ik^3 \end{Bmatrix} \quad (14)$$

or

$$\{Z\} = [\Lambda] [E(x)] \{A\} + p e^{-ikx} \{K\} \quad (15)$$

where:

$$\{Z\} = \begin{Bmatrix} W \\ \theta \\ M \\ S \end{Bmatrix}$$

$$\{A\} = \begin{Bmatrix} A \\ B \\ C \\ D \end{Bmatrix}$$

$$\{K\} = \begin{Bmatrix} 1 \\ -ik \\ -D_{sk}^* k^2 \\ D_{sk}^* ik^3 \end{Bmatrix}$$

$$[\Lambda] = \begin{bmatrix} 1 & 1 & 1 & 1 \\ \lambda_1 & -\lambda_1 & i\lambda_2 & -i\lambda_2 \\ D_{sk}^* \lambda_1^2 & D_{sk}^* \lambda_1^2 & -D_{sk}^* \lambda_2^2 & -D_{sk}^* \lambda_2^2 \\ D_{sk}^* \lambda_1^2 & -D_{sk}^* \lambda_1^3 & -D_{sk}^* i\lambda_2^3 & D_{sk}^* i\lambda_2^3 \end{bmatrix}$$

$$[E(x)] = \begin{bmatrix} e^{\lambda_1 x} & 0 & 0 & 0 \\ 0 & e^{-\lambda_1 x} & 0 & 0 \\ 0 & 0 & e^{i\lambda_2 x} & 0 \\ 0 & 0 & 0 & e^{-i\lambda_2 x} \end{bmatrix}$$

$$P = \frac{P_0}{D_{sk}^* (k^2 + \lambda_1^2) (k^2 - \lambda_2^2)}$$

Equation (11) can now be rewritten as:

$$\{Z_2\} = e^{-ika} \{Z_0\} \quad (16)$$

From equation (15)

$$\{Z_0\} = [\Lambda] [E(0)] \{A\} + P \{K\} = [\Lambda] \{A\} + P \{K\} \quad (17)$$

and

$$\{Z_1\} = [\Lambda] [E(a)] \{A\} + P e^{-ika} \{K\} \quad (18)$$

Consideration of continuity across the stringers and the equilibrium of the stringers yielded equation (1). This equation can be rewritten as

$$\{Z_2\} = [S] \{Z_1\} \quad (19)$$

where:

$$[S] = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \\ 0 & K_R & 1 & 0 \\ -K_T & 0 & 0 & 1 \end{bmatrix}$$

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Substituting for $\{Z_2\}$ from equation (16) in equation (19),

$$e^{-ika} \{Z_0\} = [S] \{Z_1\} \quad (20)$$

From equations (17) and (18), we can substitute for $\{Z_0\}$ and $\{Z_1\}$ in equation (20). We then obtain

$$e^{-ika} [A] \{A\} + Pe^{-ika} \{K\} = [S] [A] [E(a)] \{A\} + Pe^{-ika} [S] \{K\}$$

or

$$\left([S] [A] [E(a)] - e^{-ika} [A] \right) \{A\} = Pe^{-ika} \left([I] - [S] \right) \{K\} \quad (21)$$

where $[I]$ is a unit matrix of order 4×4 .

Equation (21) can now be used to solve for the coefficient matrix $\{A\}$. Substituting for $\{A\}$ in equation (14), any of the response quantities can be calculated. It can be seen the matrix $([I] - [S])$ has only two nonzero elements; i.e., $(-K_R)$ and K_T . Therefore, $\{A\}$ vanishes if at a given frequency both K_R and K_T go to zero. An intrinsically tuned structure should, therefore, be designed so that the frequencies at which K_R and K_T vanish, lie in between the *stringer torsion* and *stringer bending* mode frequencies of the stiffened panel. *Then the matrix A arising because of the reflections from the stringers vanishes, and the response reduces to a small value.*

It should be noted that coincidence excitation takes place, when the pressure field wave number k satisfies the equation

$$\det \left| \left([S] [A] [E(a)] - e^{-ika} [A] \right) \right| = 0 \quad (22)$$

Under such conditions, the matrix A in equation (21) becomes infinitely large for an undamped structure. Equation (22) shows the way in which the periodic structure theory incorporates the mechanism of coincidence excitation of a periodically stiffened structure excited by a travelling pressure field.

4.3 EFFECTS OF FINITE NUMBER OF BAYS

In the above theory, the effect of reflection of flexural wave energy from the ends of a multibay periodic structure is not taken into account and, therefore, it is applicable to infinitely long periodic structures. For finite periodic structures, the reflections from the two ends can be incorporated by considering two groups of flexural waves travelling in opposite directions, as shown in references 5 through 7 and reference 10. The responses of infinite and finite periodic structures were compared in reference 7, and it was shown that the infinite structure theory can reasonably predict the response of structures with five or more number of bays. For this reason, it was decided to build five-bay panels for test purposes, although the analytical predictions were based on equations (21) and (14) which are applicable to infinitely long, periodically stiffened panels.

4.4 THE RESULTS OF INITIAL ANALYTICAL STUDIES

Based on the above theory, a computer program was developed for predicting the response of infinitely long, periodic skin-stringer structures to convected random pressure fields. The program was set up so that skin and stringer damping loss factors measured in the laboratory test could be used as input parameters, in addition to other structural parameters such as skin thickness, stringer spacing, stringer moment of inertia, area of cross section, etc. Damping was introduced through the use of complex modulus: i.e., $E(1+i\eta)$, where η is the loss factor.

In the initial analytical studies, the effect of structural tuning on the dynamic response of a stiffened panel was studied by changing the stringer spacing and also by changing the stringer cross section. It is beyond the scope of this report to go into the details of the analytical results and only the principal findings will be summarized here. As discussed earlier, a typical skin-stringer structure exhibits a mode called *stringer bending mode* which is such that all the adjacent skin bays are in phase (fig. 5). The frequency of this mode is close to the frequency f_p of the individual skin bay, with clamped edges along the stringers and simply-supported edges along the frame. This mode responds strongly, when the structure is excited by a highly correlated and coherent near field engine noise field. It is found that when the structure is tuned, the response near the frequency f_p is reduced but there are two other modes that respond strongly. The lower frequency mode (mode 1) is such that the adjacent skin bays as well as the stringer vibrate in phase, as shown in figure 6. The higher frequency mode (mode 2) is such that although the adjacent skin bays vibrate in phase, the skin and stringers vibrate out of phase, as shown in figure 6. For this reason, a certain amount of stringer damping treatment is necessary. With damped stringers, the responses of these two modes are substantially reduced. It is found that the rms response of the structure can be reduced further, when the structure is fine-tuned; i.e., when the responses of modes 1 and 2 with damping treatment applied on the stringers are equal. An optimum stringer damping is also found to exist. The response increases if the stringer damping is increased beyond this optimum level. For example, if the stringer damping loss factor is infinitely large, the stringers do not respond and there is a strong reflection of skin bending waves caused by the stringers. Modes 1 and 2 then converge, giving rise to a strong response at the fundamental frequency f_p of the individual skin bay.

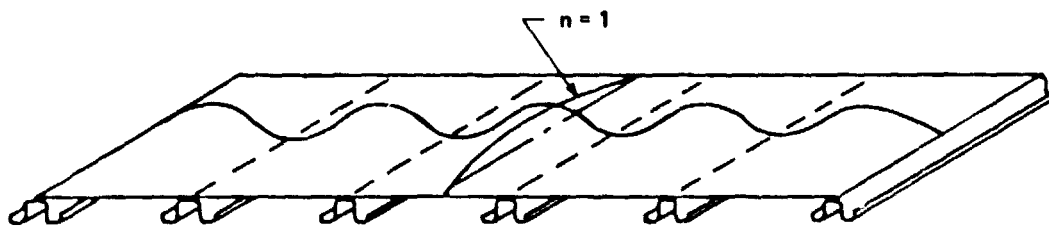


Figure 5.—Classical Stringer Bending Mode of a Periodic Skin - Stringer Panel

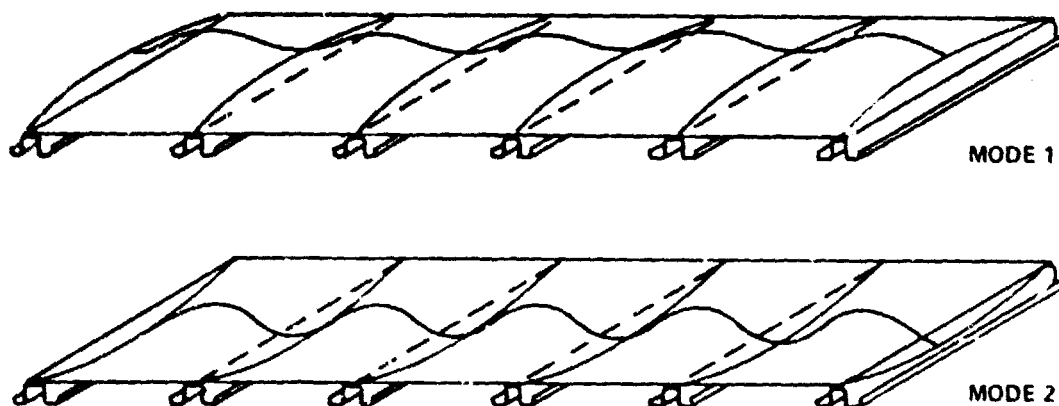


Figure 6.—Two Principal Modes of an Intrinsically Tuned Panel

Analysis also shows that modes 1 and 2 exist in different degrees in panels that are not intrinsically tuned. In panels with a large stringer spacing such that the skin bay frequency is lower than the stringer frequency, the frequency of mode 1 is close to the skin bay frequency f_p , and this mode dominates the response spectrum. The mode 2 frequency is close to the stringer frequency and has a much weaker response. For such panels, application of damping tapes on the skin is very effective in reducing the response of mode 1. On the other hand, application of damping treatment on the stringers is not effective in reducing the response of mode 1, although it is effective in reducing the response of mode 2.

For panels with a narrow stringer spacing, such that the skin bay frequency is higher than the stringer frequency, the frequency of mode 1 is closer to the stringer frequency, and the frequency of mode 2 is closer to the skin bay frequency. In this case, the lower frequency mode is such that the skin acts like a relatively stiff member supported on relatively flexible stringers. With the large stringer deflections and with the entire panel surface vibrating in-phase, this mode is a strong radiator of sound and, therefore, it dominates the radiated noise spectrum. For such panels, application of damping treatment on the skin is effective in reducing the response of mode 2, but it is not so effective in reducing the response of the first mode. On the other hand, application of damping treatment on the stringer is very effective in reducing the response of mode 1, and it is less effective in reducing the response of mode 2.

Furthermore, for panels with very stiff stringers, such that the skin bay frequency is lower than the stringer frequency, the frequency of mode 1 is close to the skin bay frequency f_p . This mode dominates the response spectrum. The mode 2 frequency is close to the stringer frequency and has a much weaker response. For such panels, application of damping tapes on the skin is very effective in reducing the response of mode 1, whereas application of damping treatment on the stringers is not so effective in reducing the response of mode 1.

For panels with very massive stringers, such that the skin bay frequency is higher than the stringer frequency, the frequency of mode 1 is closer to the stringer frequency, and the frequency of mode 2 is closer to the skin bay frequency. However, in this case the stringer deflections are relatively small, and the first mode has a lower response than the second mode. The second mode response dominates the radiated noise spectrum. The response of this mode can be effectively controlled by applying damping treatment on the skin. The first mode response is effectively controlled by stringer damping, but since this is no longer a dominant mode and since the dominant mode around f_p is not significantly affected by stringer damping, the overall level remains relatively unaffected by stringer damping. In this respect, this panel behaves differently from the panel with a narrow stringer spacing in which mode 1 is the dominant mode, although in both panels the skin bay frequency is higher than the stringer frequency. However, in aircraft structures, one is unlikely to encounter very massive stringers, since the goal is to design structures with high stiffness to weight ratios. This case was studied analytically for the sake of completeness, although this was mainly of academic interest. For the same reason, in the experimental study it was decided to look into the effect of varying the stringer spacing instead of varying the stringer cross section.

5.0 LABORATORY TEST RESULTS

After the concept was developed and analytically verified, the next step was to conduct tests on actual hardware representative of a fuselage structural section so that the results of the analysis could be confirmed by test data. For this purpose, four skin-stringer panels were designed and fabricated. Two of them had structural components that were intrinsically tuned. The other two had structural components that were off-tuned. The responses of the panels to an excitation field, simulating the characteristics of near field jet noise, were measured in the laboratory. The purposes of the test were to verify:

1. The disappearance of the strong response around the frequency f_p when the structure is intrinsically tuned
2. The existence of the two principal modes of the tuned panel
3. The effect of applying damping treatment on the stringers on the low frequency structural response
4. The effect of changing the stringer spacing so that the structural components are tuned or off-tuned in different ways

The purpose of this section is to present the method of designing the tuned panel, the principal results, and the conclusions of the laboratory test. This test was conducted as a part of the Boeing Independent Research and Development program on Interior Noise.

5.1 PANEL DESIGN

In order to design the tuned panel, it was necessary to know the exact frequency of the stringer with and without any damping treatment. A series of tests were conducted to develop stringer damping methods (fig. 7). These were based on application of constrained layer viscoelastic damping treatment on the stringer flanges. The results are presented in figure 7. It was found that because of the large bending stiffness of the stringer cross section, the conventional damping tapes applied on the stringer flanges were not very effective, and it was necessary to use a thicker viscoelastic material and a thicker constraining layer of aluminum (fig. 7). Based on these results, a damping treatment consisting of a 0.010 in. (0.00025 m) thick 3M ISD 112 viscoelastic layer and a 3.125 in. x 0.08 in. (0.079 m x 0.002 m) aluminum strip was chosen. The stringer was 24.6 in. (0.625 m) long and was simply supported at the two ends. The natural frequency of the stringer with the damping treatment was found to be 241 Hz, and the change in the stringer frequency between damped and undamped conditions was small. The measured frequency also agreed well with the frequency predicted by equation (6).

The next step was to design the tuned panel for which the natural frequency of the individual skin bay matched with the natural frequency of the stringer as measured from the test. From reference 27 the natural frequency of a rectangular panel with two long sides clamped and two short sides simply supported is given by

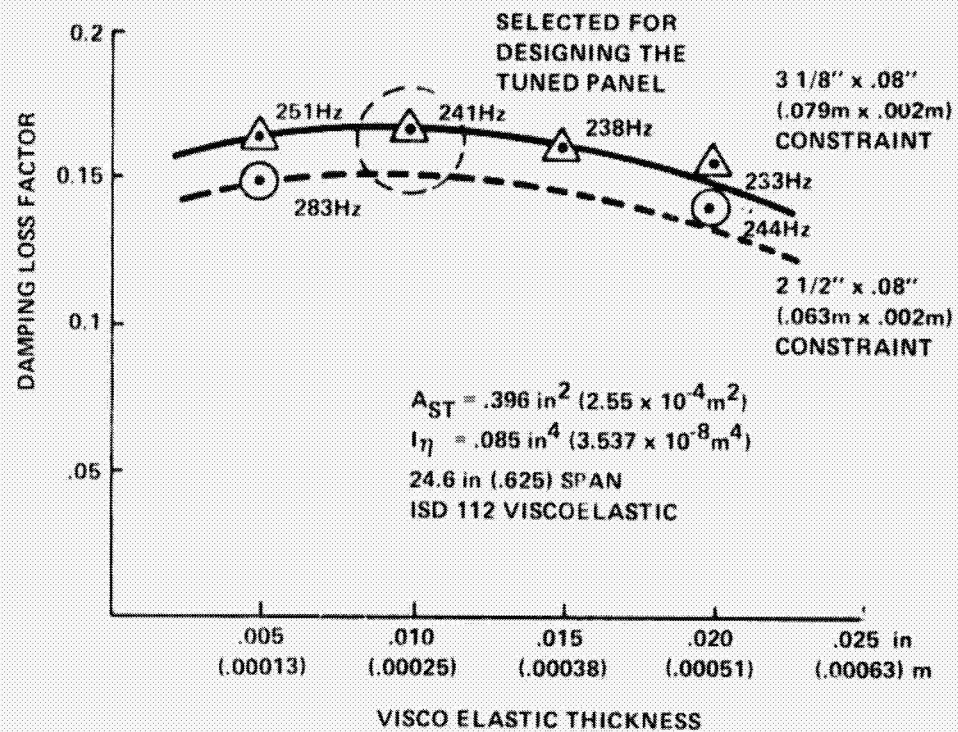
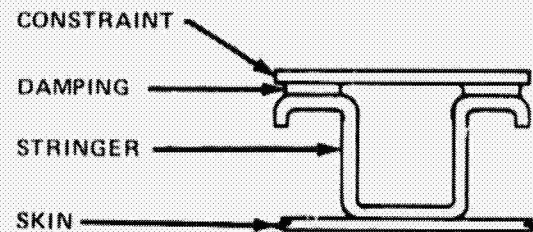
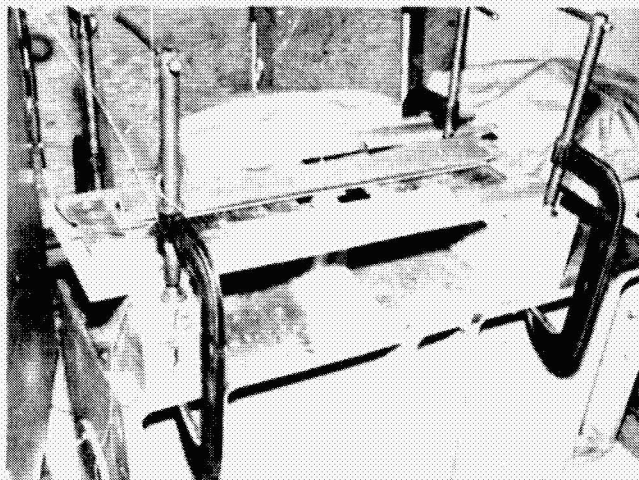


Figure 7.—Stringer Damping Test Results

$$f_p = \frac{\pi}{2 \sqrt{1-\nu^2}} \left(\frac{E_{sk}}{\rho_{sk}} \right)^{1/2} \frac{t}{b^2} \left[\left\{ 1 + (1.506)^2 \left(\frac{b}{a} \right)^2 \right\}^2 - 2.04 \left(\frac{b}{a} \right)^2 \right]^{1/2} \quad (23)$$

where:

a Length of the shorter side

b Length of the longer side

t Panel thickness

E_{sk}, ρ_{sk}, ν = Young's modulus, density, and the Poisson's ratio of the skin material, respectively

In the present case, "a" is the stringer spacing and "b" is the frame spacing (and also the stringer length). The panel thickness was chosen to be 0.063 in. (0.0016 m), and the frame spacing was chosen to be 24.6 in. (0.625 m). The panel material was aluminum. Equation (23) was then solved for "a", so that the panel frequency was equal to the measured stringer frequency; i.e., 241 Hz. In this manner the stringer spacing for the tuned panel was found to be about 7.5 in. (0.19 m).

In order to verify the results of the analytical predictions, four skin-stringer panels were designed and built. They were identical except in stringer spacing. One had a stringer spacing of 7.5 in. (0.19 m), and another had a stringer spacing of 7.25 in. (0.184 m). These two panels were, therefore, intrinsically tuned or very close to being so.

Another panel was built with 9 in. (0.228-m) stringer spacing. Calculations based on equation (23) showed that the skin bay frequency f_p for this panel was about 28% lower than the stringer frequency f_s . Therefore, this panel was not far from being tuned. A panel with a wider stringer spacing could have been chosen to simulate a greater degree of frequency mismatch. However, a stringer spacing of 9 in. (0.228 m) was chosen since that is typical of many commercial airplanes.

The fourth panel had a 5 in. (0.13-m) stringer spacing. Calculations based on equation (23) indicated that the skin bay frequency f_p for this panel was about 2.52 times higher than the stringer frequency f_s . Therefore, this panel was off-tuned to a larger degree.

Each panel was chosen to have five bays. This reduced the effect of reflection of flexural waves from the panel boundaries. In addition, the end bays were extended beyond the terminating stringers by about 3 in. (0.076 m), and standard foil-backed damping tapes (3M No. 428C) were applied on these extensions. A typical skin stringer panel is shown in figure 8. The method of damping the stringers is shown in figure 9.

5.2 RESPONSE TO MULTIPOINT EXCITATION

Each of these panels was then instrumented with 11 accelerometers and 9 strain gages. The panels were mounted between a reverberant room and an anechoic box. A set of lightweight electromagnetic shakers was used to excite the panels (figs. 10 and 11). These shakers were connected in series and fed by current from a white noise generator to

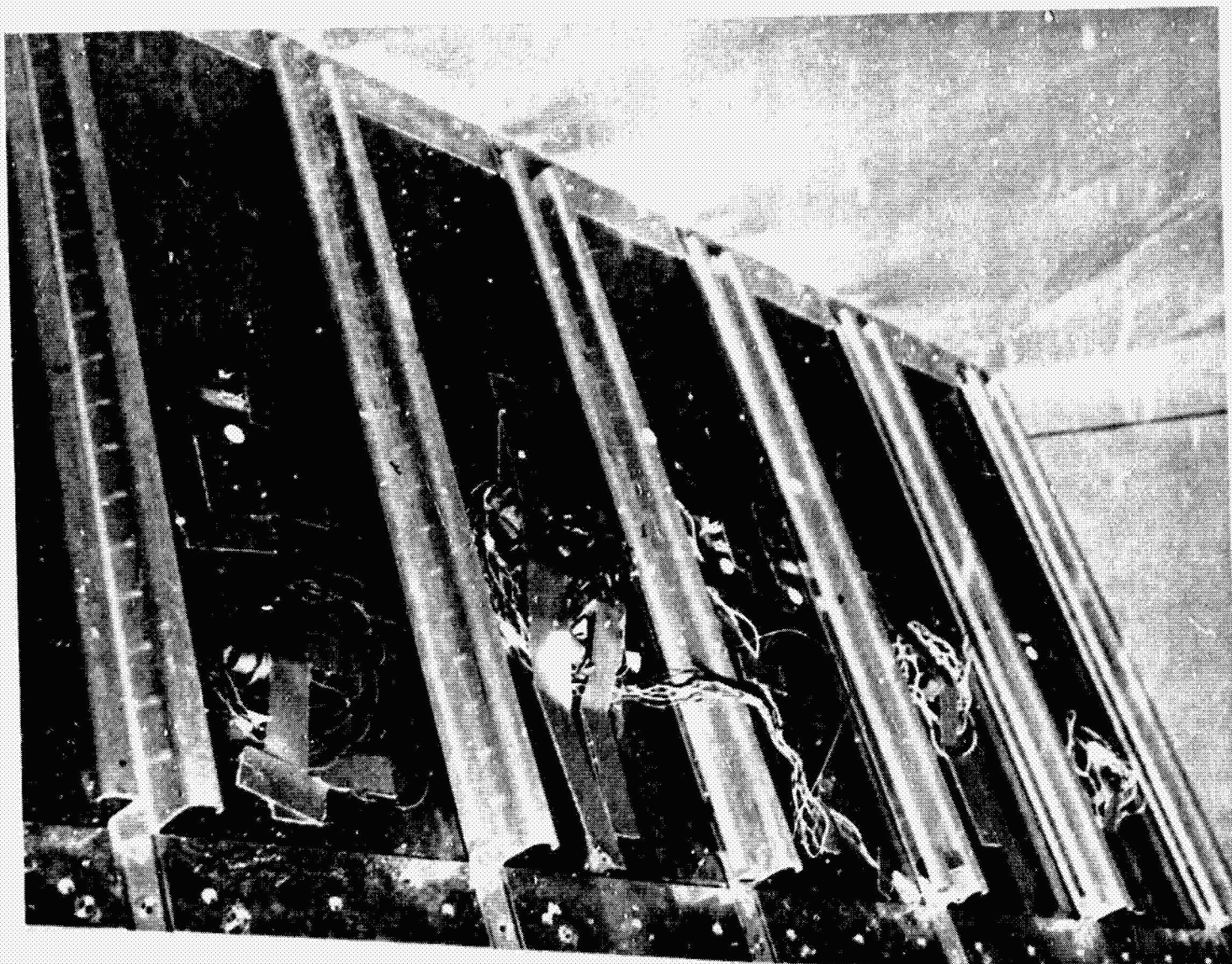


Figure 8.—Typical Skin - Stringer Panel

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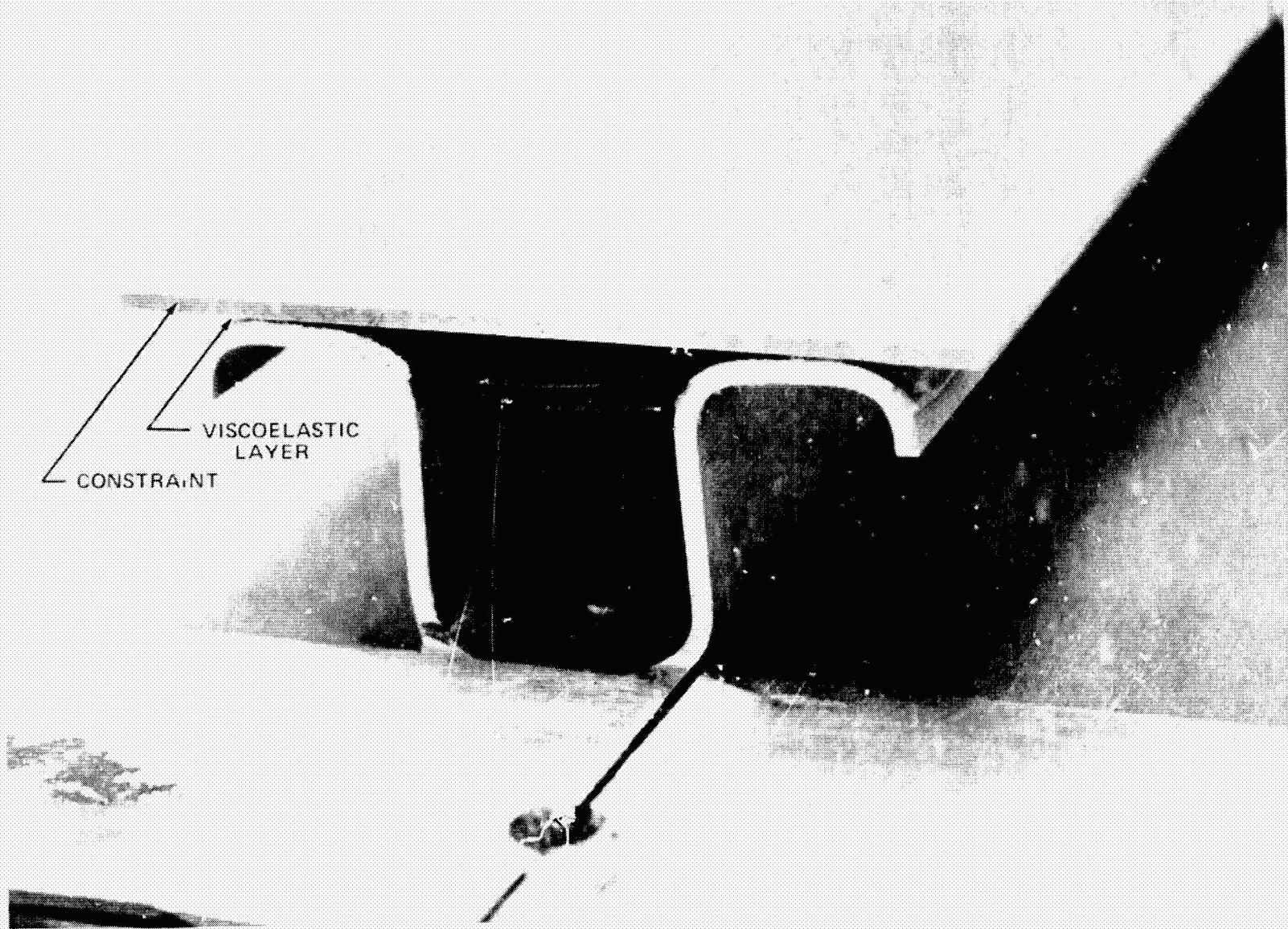


Figure 9.—Use of Constrained Viscoelastic Damping Treatment for Damping the Stringers

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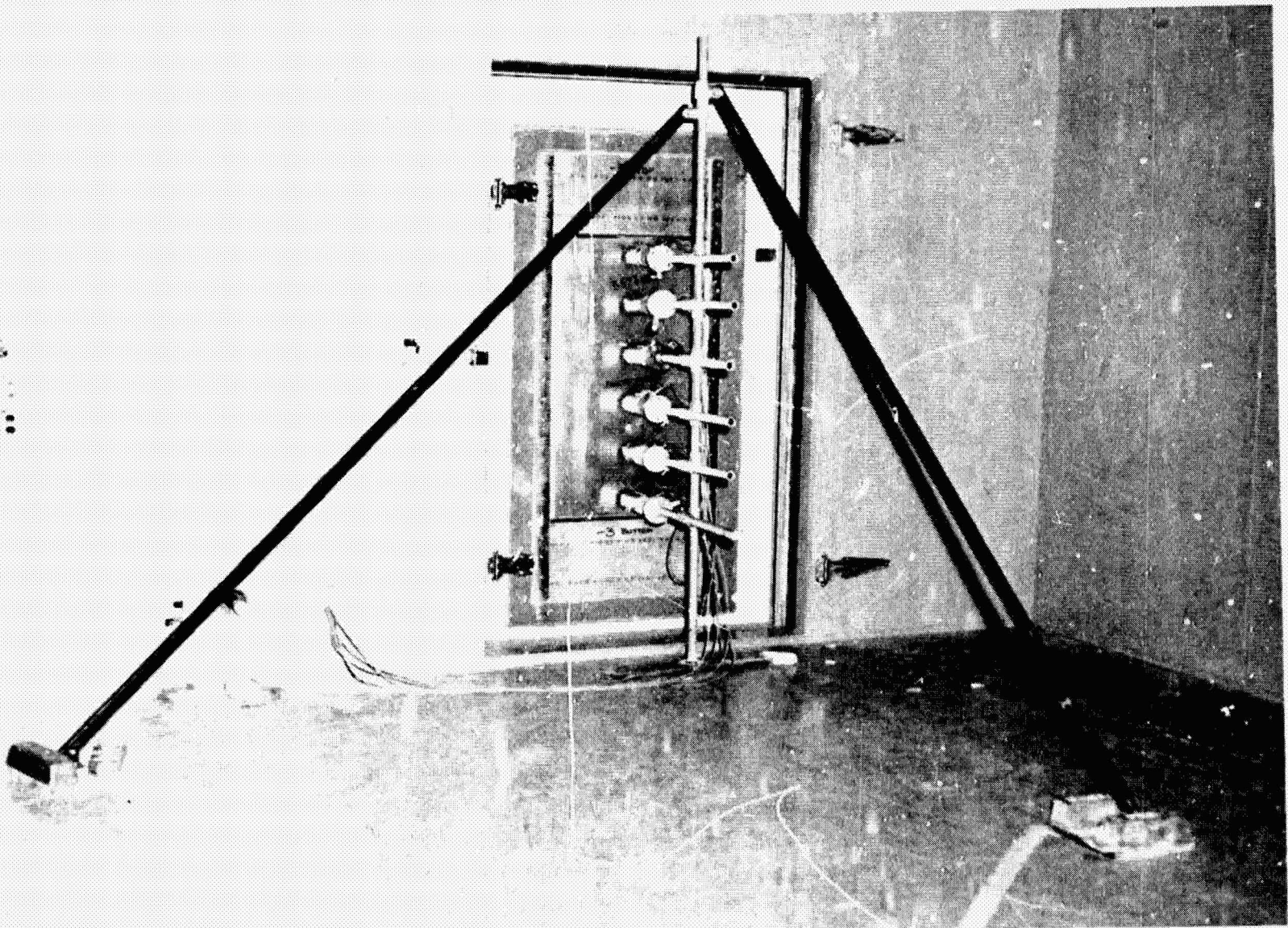


Figure 10.—Laboratory Test Setup

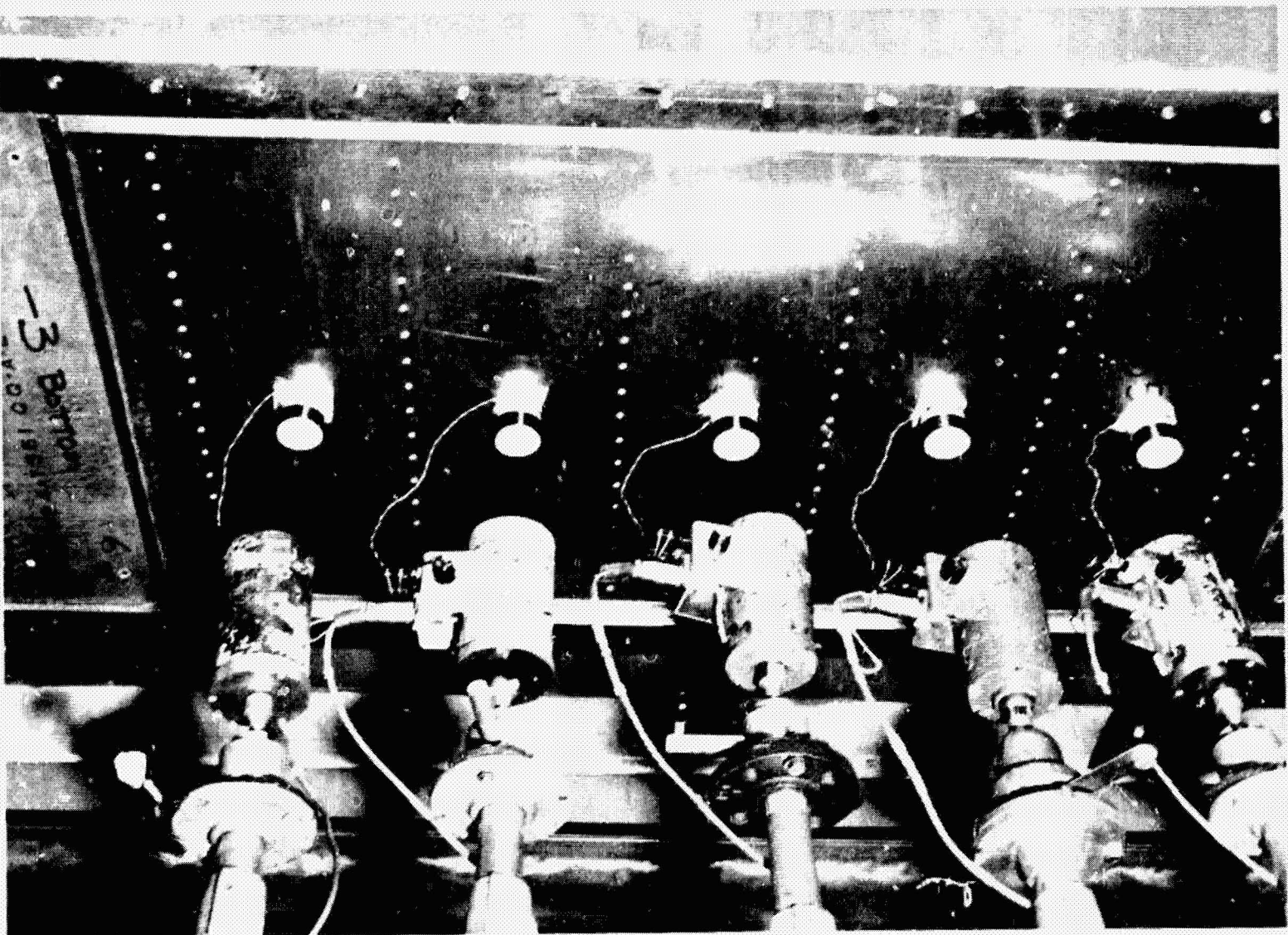


Figure 11.—Use of Inphase, Multipoint Random Excitation to Simulate Near Field Jet Noise

simulate a highly correlated and coherent near field jet noise environment. The frequency range of the white noise was from 0 to 1 kHz. The sound radiation from the panels into the anechoic box was measured by six microphones installed in a vertical plane about 4 in. (0.10 m) away from the panel surface. These microphones were placed along the length of the panel, in two columns, with three microphones in each column.

The panels were excited by using the shakers in two configurations: (1) excitation at the center of each skin bay (five point excitation) and (2) excitation at the center of each stringer (six point excitation). The responses of each panel to both configurations were compared with the predicted response of that panel to white noise excitation at normal incidence, assuming that the panel was infinitely long. The latter was based on the periodic structure theory as discussed in section 4.0. In the computation, p_0 was assumed to be unity in equation (15).

The predicted vibration velocity spectrum of the panel with 9 in. (0.228-m) stringer spacing is compared in figure 12 with the sound radiation spectrum obtained by using five point excitation. The latter was obtained from an FFT analyzer which plotted the transfer function (ratio of the spectrum levels rather than PSDs) between the response and excitation. However, the predicted response is plotted on a PSD basis. For this reason, the numerical values along the vertical axes should not be compared. The main purpose at this point is to compare the frequencies and the shapes of the response spectra, rather than the absolute levels. These comments also apply to figures 13 through 17 (and figs. 42 and 43). The absolute values of the predicted and measured velocity responses are compared in section 7.6.

In figure 12, a good agreement is observed between the predicted and the measured spectra. The peak sound radiation was predicted to occur at 142 Hz; the measured frequency was 135 Hz. The predicted frequencies of the other two modes also agreed with the test data.

However, for the panel with 5 in. (0.13-m) stringer spacing, point force excitation at the bay centers did not excite the modes that were predicted to be excited by the distributed, near field jet noise pressure field. This can be best illustrated by comparing the predicted acceleration spectra with that obtained by using five point excitation (fig. 13). Although the mode at about 200 Hz was excited, the strong peak at about 700 Hz was clearly missing in the measured spectrum. These two modes corresponded to the two principal modes of this panel. On the other hand, there was a strong peak at about 457 Hz in the measured data. The measured mode shape at this frequency showed the existence of three half-waves along the width of the panel.

From the test data, it became clear that for the panel with 5 in. (0.13-m) stringer spacing, point force excitation at each panel center unduly encouraged the mode with more than one half wave along the width of the panel, and the higher frequency mode was not properly excited. To overcome this problem, it was decided to use point force excitations at the center of each stringer.

The measured acceleration transfer function for the panel with 5 in. (0.13-m) stringer spacing is shown in figure 14 where it is compared with the predicted acceleration spectra. It can be seen that with the excitation at the center of each stringer, the 700-Hz mode is strongly

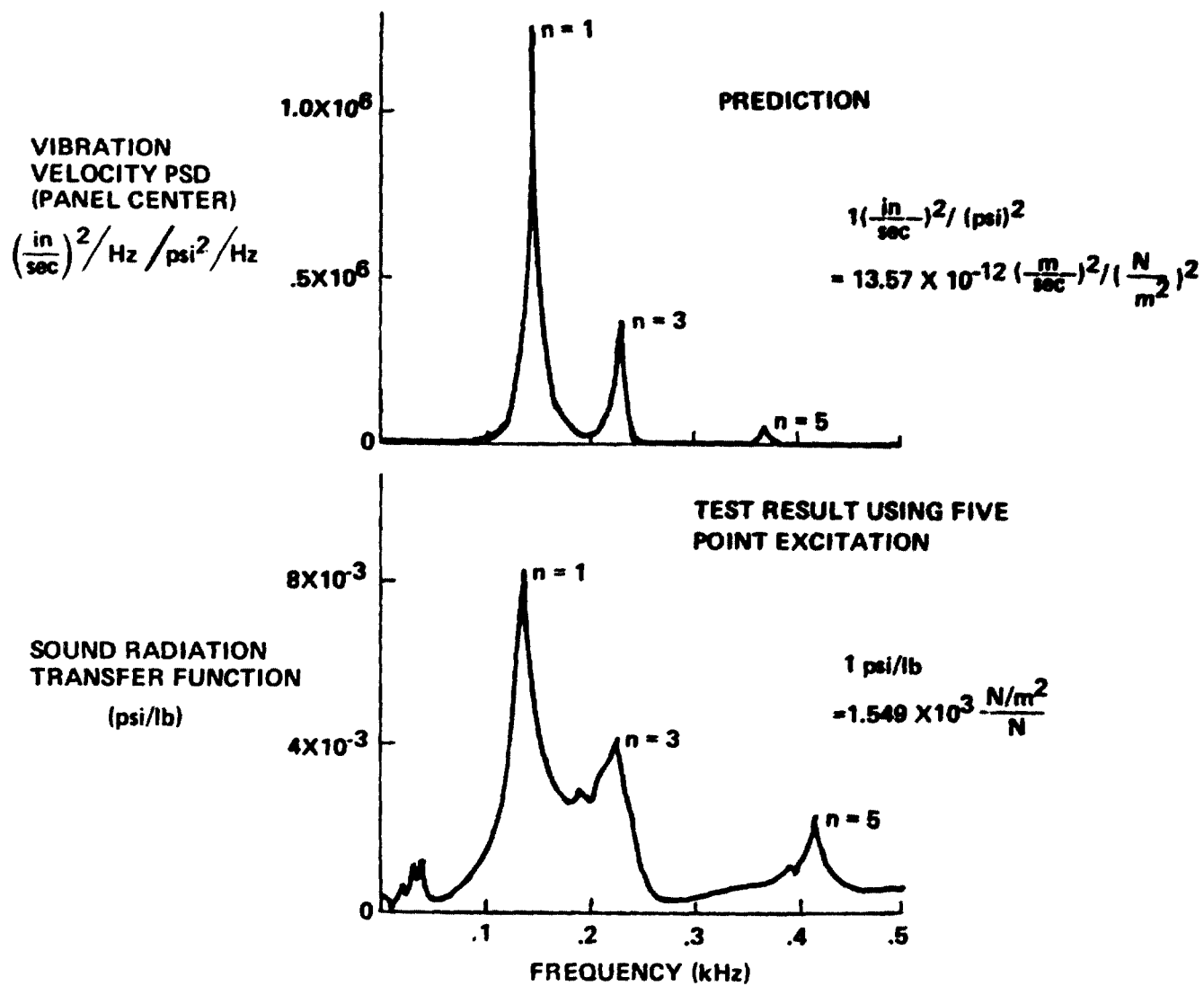


Figure 12.—Response of the Damped Panel with 9-in. (0.228-m) Stringer Spacing

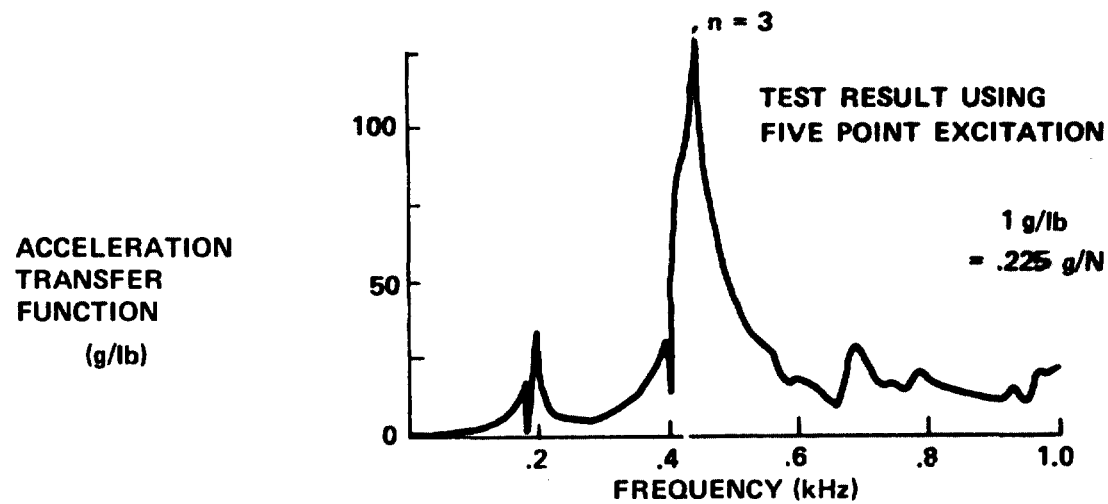
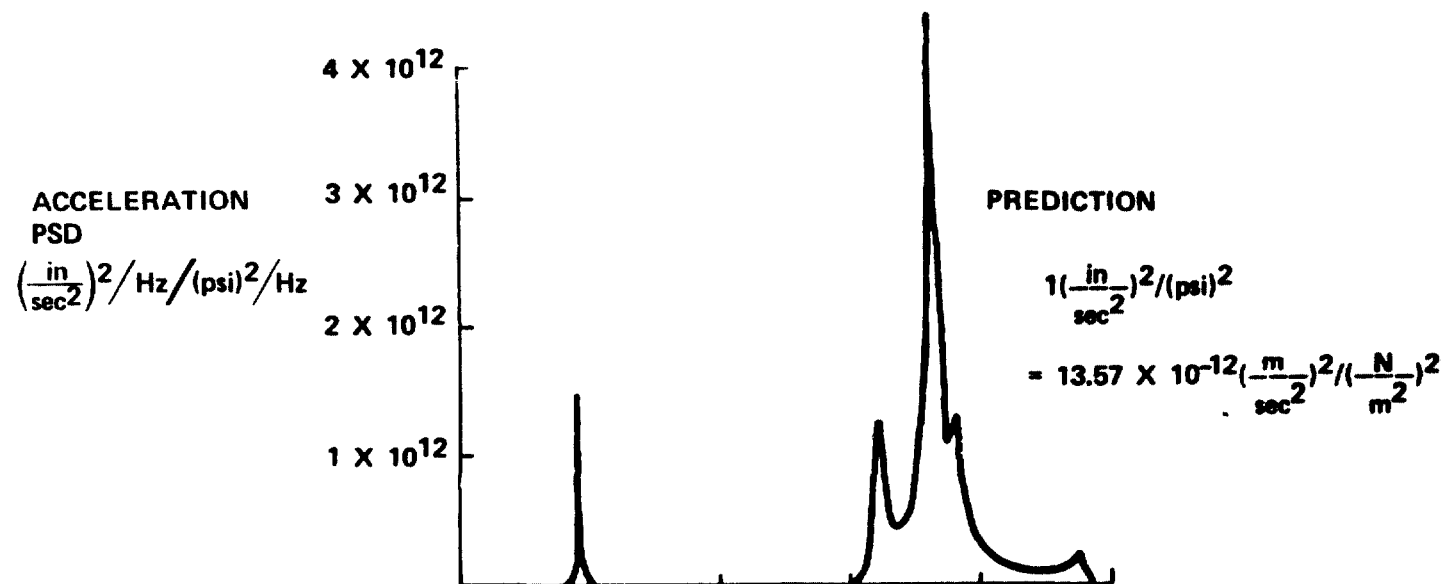


Figure 13.—Response of the Undamped Panel with 5-in. (0.13-m) Stringer Spacing (Panel Center)

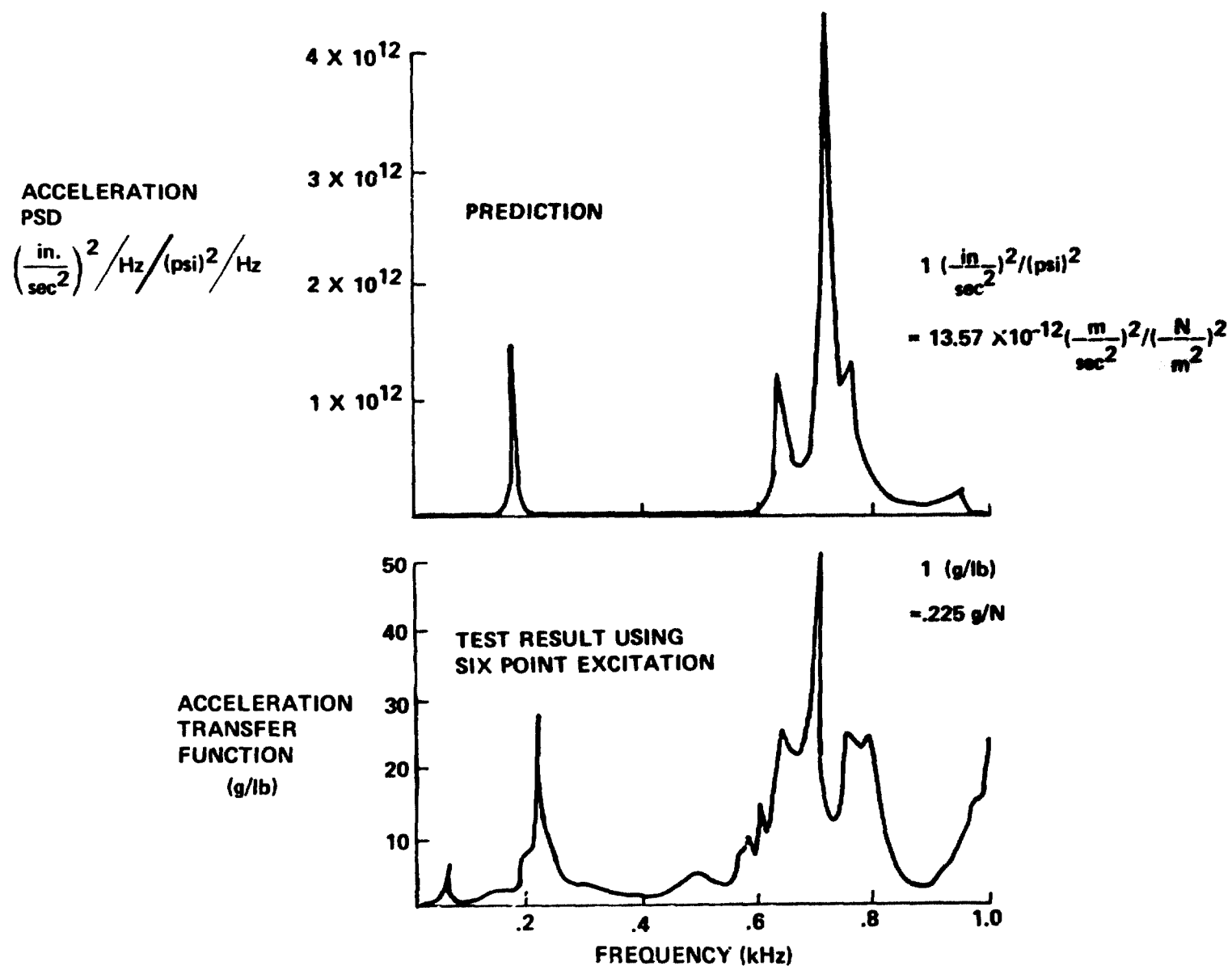


Figure 14.—Response of the Undamped Panel with 5-in. (0.13-m) Stringer Spacing (Panel Center)

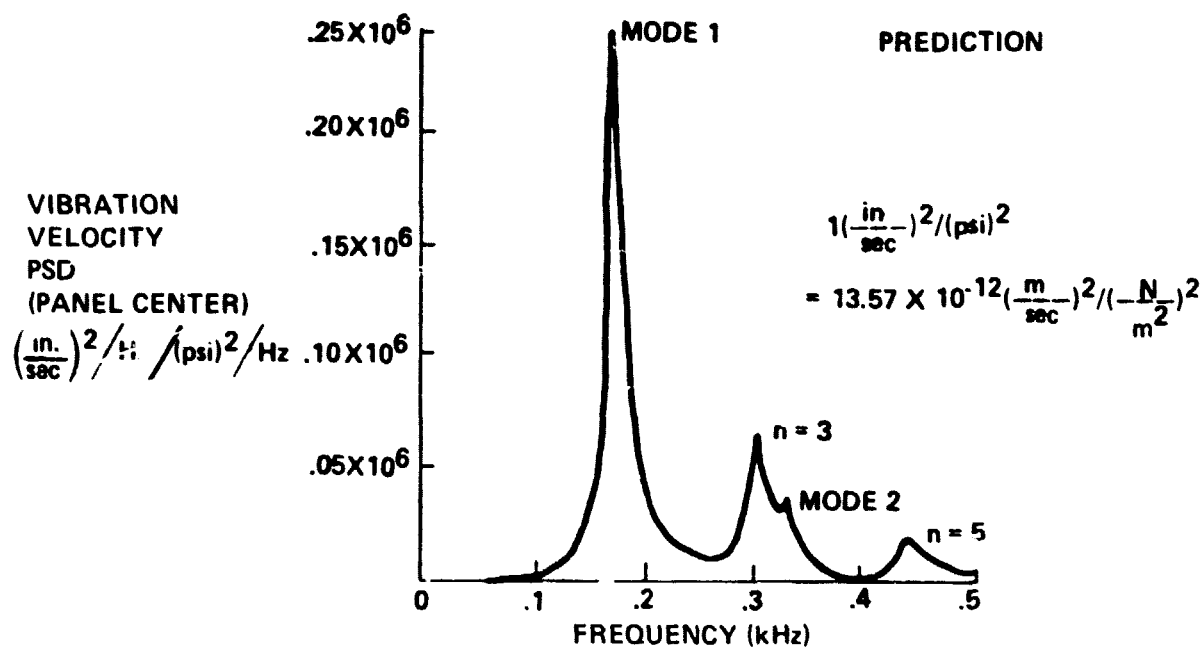


Figure 15.—Predicted Velocity Response of the
Tuned Panel With Damped Stringer

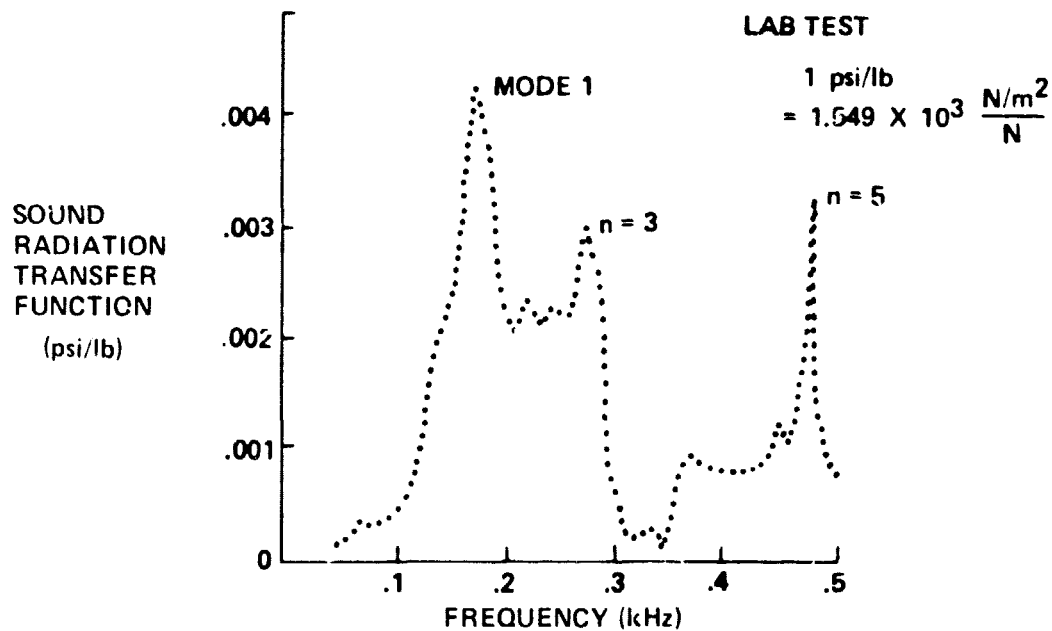


Figure 16.—Measured Response of the Damped
Tuned Panel With Five Point Excitation

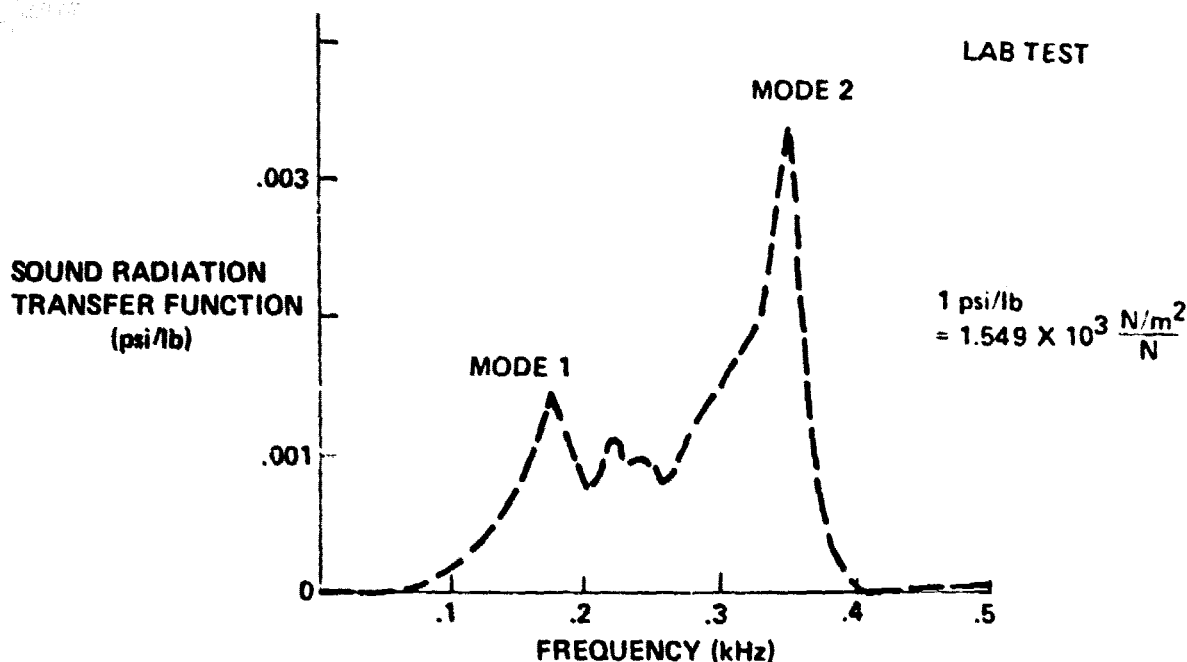


Figure 17.—Measured Response of the Damped Tuned Panel With Six Point Excitation

excited, and there is a very good correspondence between measurement and prediction. Thus, while excitation at panel centers excited the predicted modes of the panel with 9-in. (0.228-m) stringer spacing, excitation at the stringer centers excited the predicted modes of the panel with 5-in. (0.13-m) stringer spacing.

The tuned panel with 7.5-in. (0.19-m) stringer spacing was initially excited by five point excitation, and the results were compared with the predictions. The mode 1 frequency was predicted to be about 165 Hz; the mode 2 frequency was predicted to be 325 Hz (fig. 15). Results with the five point excitation indicated the presence of mode 1 at 175 Hz, but it was found that mode 2 was very difficult to excite with this type of excitation (fig. 16). Another mode with three half-waves along the panel width was predicted to exist around 300 Hz. With five point excitation, this mode was excited in the test panel at 280 Hz (fig. 16). Similar results were observed when the panel with 7.25-in. (0.184-m) stringer spacing was excited by five point excitation.

These panels were then excited by six point excitation. For the panel with 7.5-in. (0.19-m) spacing, the mode 2 was strongly excited at 360 Hz (fig. 17). Although mode 1 was also excited, the response spectrum was now dominated by mode 2. Similar results were observed when the panel with 7.25-in. (0.184-m) stringer spacing was excited by six point excitation.

Notice that no matter how the panel was excited, there was no strong peak around 241 Hz. If the stringers were extremely stiff or extremely massive resulting in a stringer that was much higher or much lower than the panel frequency (i.e., 241 Hz), there would have been a large response around 241 Hz.

Thus, for these panels while the excitation at the bay centers excited some of the predicted modes, some important modes were missing, and these were properly excited by placing the shakers at the center of each stringer. Therefore, all the predicted modes could be excited by a combination of five and six point excitations.

This is also true for all the panels. For the panel with 9-in. (0.228-m) stringer spacing, the frequency of the individual skin bay was lower than the frequency of the stringer, and the response to five point excitation was much stronger than the response to six point excitation. Thus, the combined response was dominated by the response to five point excitation, and the measured response with five point excitation compared well with the predictions. On the other hand, for the panel with 5-in. (0.13-m) stringer spacing, the frequency of individual skin bay was higher than that of the stringer, and the response to six point excitation was much stronger than that due to five point excitation. Thus, the combined response was dominated by the response to six point excitation, and the measured response with six point excitation compared well with the predictions. Therefore, for all the panels the response to near field jet noise excitation could be properly simulated by superposing the responses to five and six point excitations. Therefore, it is more meaningful to compare the response of all the panels to the combined eleven point excitation.

This clearly points out the importance of excitation of appropriate structural modes when discrete forcing functions are used to simulate a distributed pressure field. In the present case, it was necessary to place the discrete excitation near the structural member that had the lower uncoupled natural frequency, since that member acted like a more flexible member and thus determined the overall response. This also shows the necessity of examining and evaluating the test data against predictions based on a mathematical model of the test structure.

5.2.1 PANEL DAMPING LOSS FACTORS

The loss factors of each panel were measured at each accelerometer location, for most of the important modes. All the bolts securing the panels on the door frame of the anechoic chamber were tightened using a measured constant torque, to avoid all variation of damping from panel to panel, caused by frictional energy dissipation in the panel mounting system. The measured values of panel damping showed a considerable variation from one point of a panel to another and from one frequency range to another. Since structural tuning affects the low frequency modes, the damping values of the various panels are compared here only for the low frequency modes. Generally, these modes had somewhat higher damping compared to the modes in 500- to 1000-Hz range.

The panel damping loss factors of the low frequency modes are plotted against stringer spacing in figure 18. The range of variation of the damping loss factor is also shown in this figure. These were obtained by using the method of Kennedy and Pancu (ref. 28). For each stringer spacing, the indicated damping values represent the variations over the first group of low frequency modes, as well as over the entire panel surface. Within each frequency group, the modes in which the adjacent skin bays and stringers vibrated in phase generally had higher damping.

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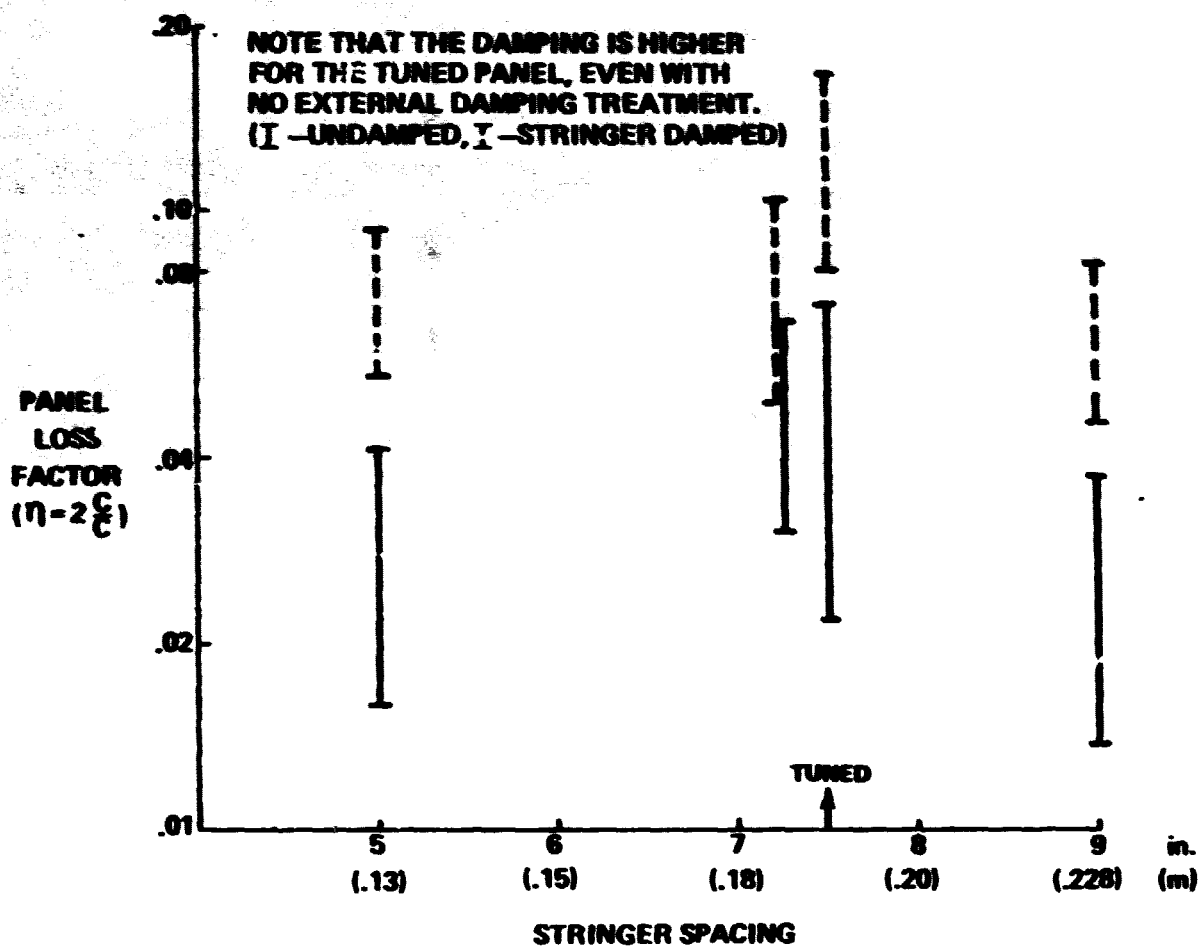


Figure 18.—Variation of Panel Damping Loss Factors for the Low Frequency Modes

From figure 18, it is clear that for the undamped panels, the panel loss factor is higher when the tuning condition is satisfied. This suggests there is increased energy dissipation in the riveted skin-stringer joints when the panel is intrinsically tuned.

The effect of applying damping treatment on the stringers is also shown in figure 18. (Since there was some overlap in the damping values for the panel with 7.25-in. (0.184-m) stringer spacing, the loss factors for the damped condition are plotted with a slight offset, to avoid any confusion.) The overall loss factors of all the panels were increased approximately by a factor of two. The tuned panel still had the maximum overall damping. However, as described in section 5.2.2, stringer damping had the maximum effect on the response of the panel with 5-in. (0.13-m) stringer spacing.

5.2.2 ACCELERATION RESPONSE

The acceleration response was measured by placing accelerometers at the centers of each bay and each stringer. The variation of acceleration response of the most dominant low frequency mode of each panel, as measured in the laboratory test, is plotted as a function of stringer spacing in figure 19. Since the excitation field simulating the near field jet noise excitation was in phase over all the bays, the mode in which all the adjacent bays and the stringers vibrate in phase responded most strongly and, therefore, dominated the response spectrum. This mode is important since it is a good radiator of sound and also has the highest stress response. For these reasons, the acceleration, sound radiation, and the stress responses of all the panels are compared in terms of the response of this mode which has the peak response in the low frequency range.

It is seen that for a given structure, the low frequency acceleration response reaches a minimum level when the skin panel and stringers are intrinsically tuned to each other. For the undamped condition, the response level goes up if the stringer spacing is increased or decreased from the spacing that is required for satisfying the tuning condition.

However, with damped stringers, the panel acceleration response measured in decibels decreased almost linearly as a function of stringer spacing. As the stringer spacing is reduced, a greater part of the vibratory energy was transferred to the stringers. As a result, damping treatment on the stringers became very effective in reducing the response. On the other hand, application of damping treatment on the skin becomes progressively less effective in reducing the response of low frequency modes as the stringer spacing is reduced. This was predicted by the analytical model.

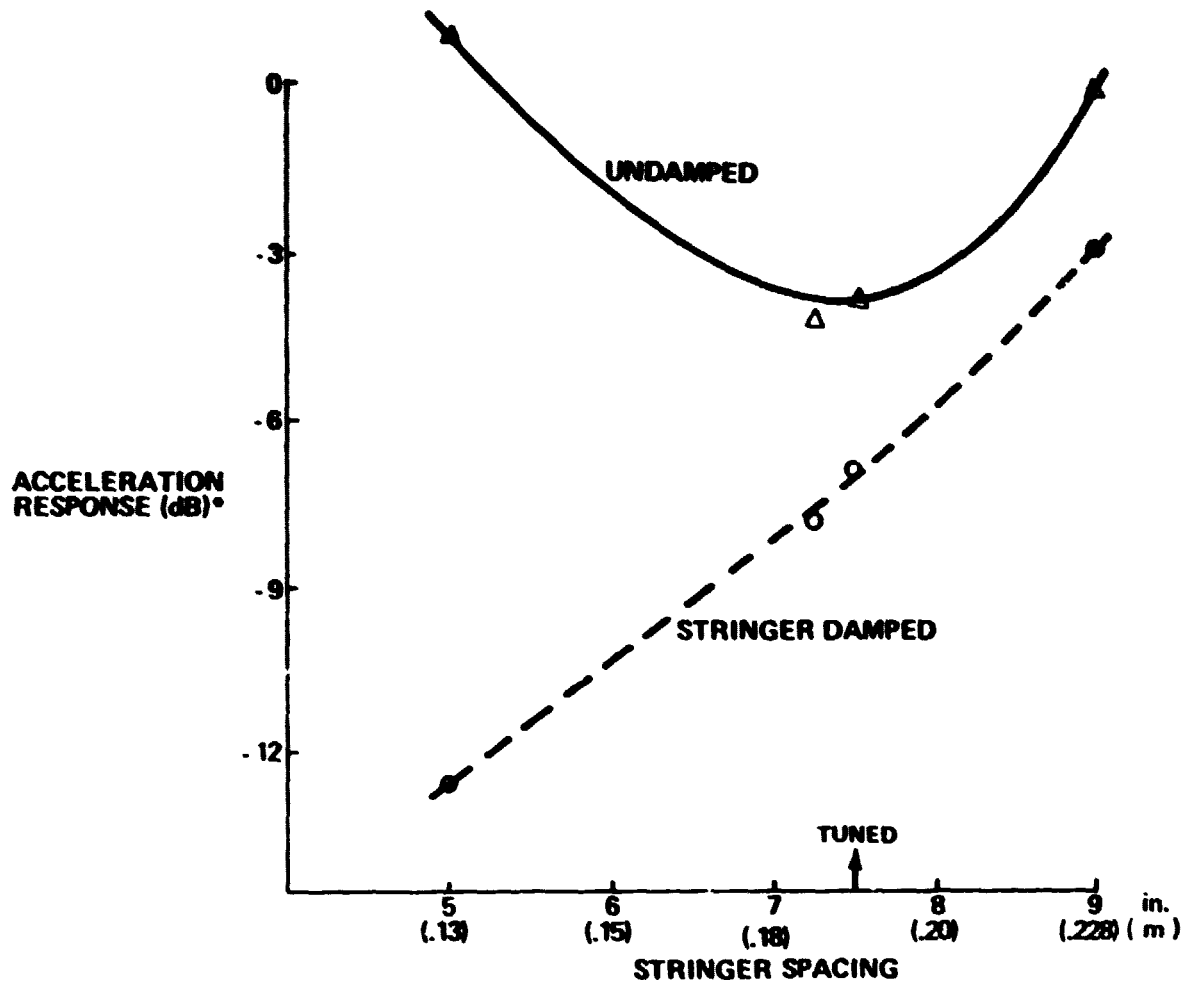
5.2.3 SOUND RADIATION

Figure 20 plots the noise radiated by the most dominant low frequency mode, as a function of stringer spacing. This plot is based on the data obtained by summing up the signals from all the microphones inside the anechoic box. Thus it is a measure of the sound radiated by the entire panel. With no damping treatment, the radiated noise level decreased by more than 3 dB as the stringer spacing was reduced from 9 in. (0.228 m) to 7.5 in. (0.19 m). With a further reduction of stringer spacing, the level of radiated noise remained essentially at the same level. Thus, for an undamped structure, the tuning condition defines the point of diminishing return for reducing cabin noise by fuselage structural modifications aimed at increasing the fuselage structural stiffness.

Further reductions were possible by applying damping treatment on the stringers. Under damped conditions, the radiated noise level measured in decibels decreased almost linearly as a function of stringer spacing. The peak radiated noise levels measured by microphones placed near the panel centers are plotted in figures 21 and 22, again similar trends are observed.

These results show that under undamped conditions, there is a limit to the noise reduction that can be achieved by changing the structural parameters. This limit can be achieved by designing the structure so that the skin panel and stringers are tuned to each other. However, further reductions are possible by choosing structural dimensions so that the

(AVERAGE OF GTIR 624, 626, 628, SEE FIG. 33)



(* RELATIVE TO THE RESPONSE OF THE UNDAMPED PANEL
WITH 9" (.228 m) STRINGER SPACING)

Figure 19.—Reduction of the Peak Low Frequency Acceleration Response of the Skin-Stringer Panels by Tuning and Damping

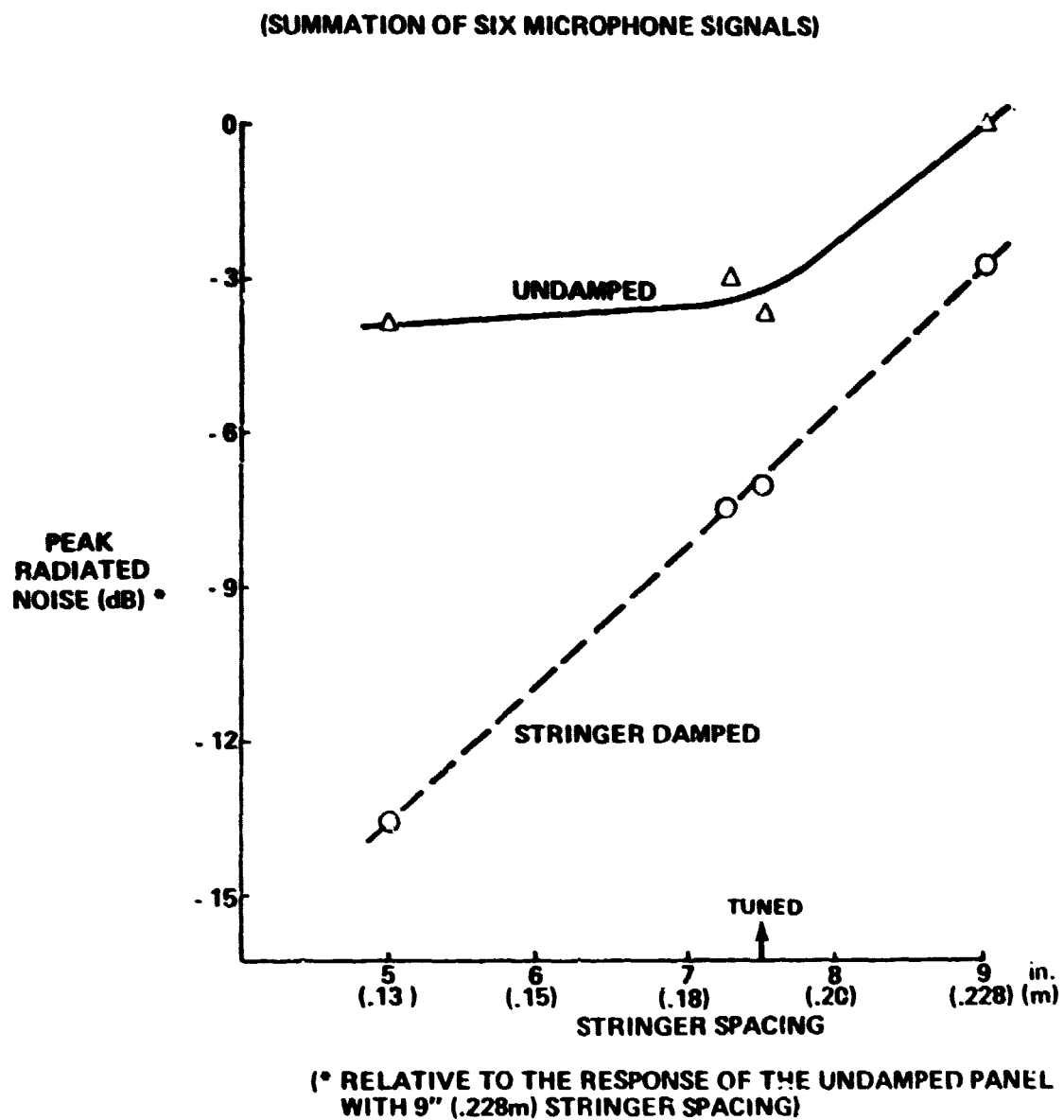


Figure 20.—Reduction of Peak Low Frequency Sound Radiation by Tuning and Damping

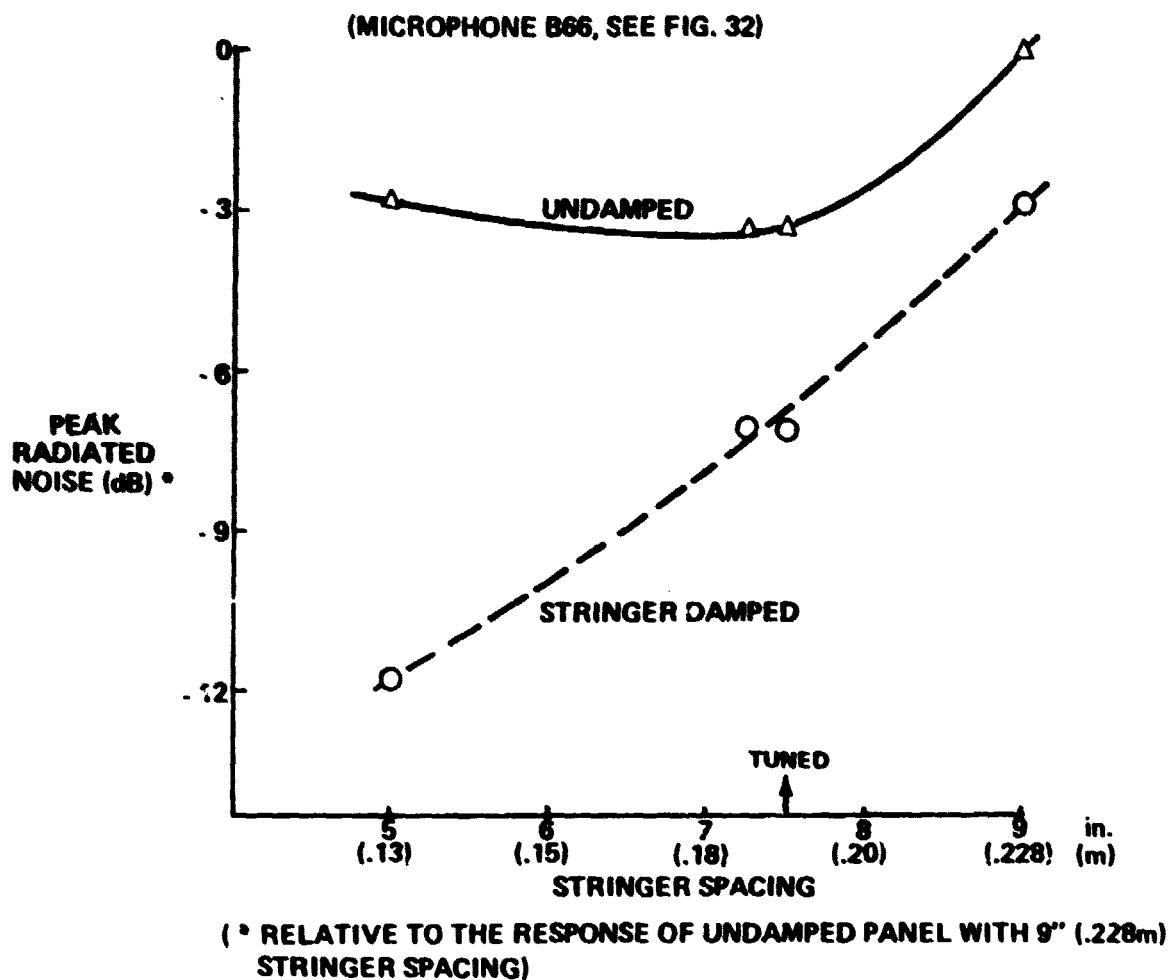


Figure 21.—Reduction of Peak Low Frequency Sound Radiation by Tuning and Damping

uncoupled skin panel frequency is higher than that of the uncoupled stringer frequency and then applying damping treatment on the stringer flanges. Under this condition, the skin acts like a relatively stiff member supported on relatively flexible stringers. As a result, damping treatment on the stringers becomes very effective in reducing the response. In contrast, skin damping would have been less effective in reducing the peak response of the panel with 5-in. (0.13-m) stringer spacings. As discussed in section 4.4, the low frequency response of this panel is primarily controlled by stringer resonance.

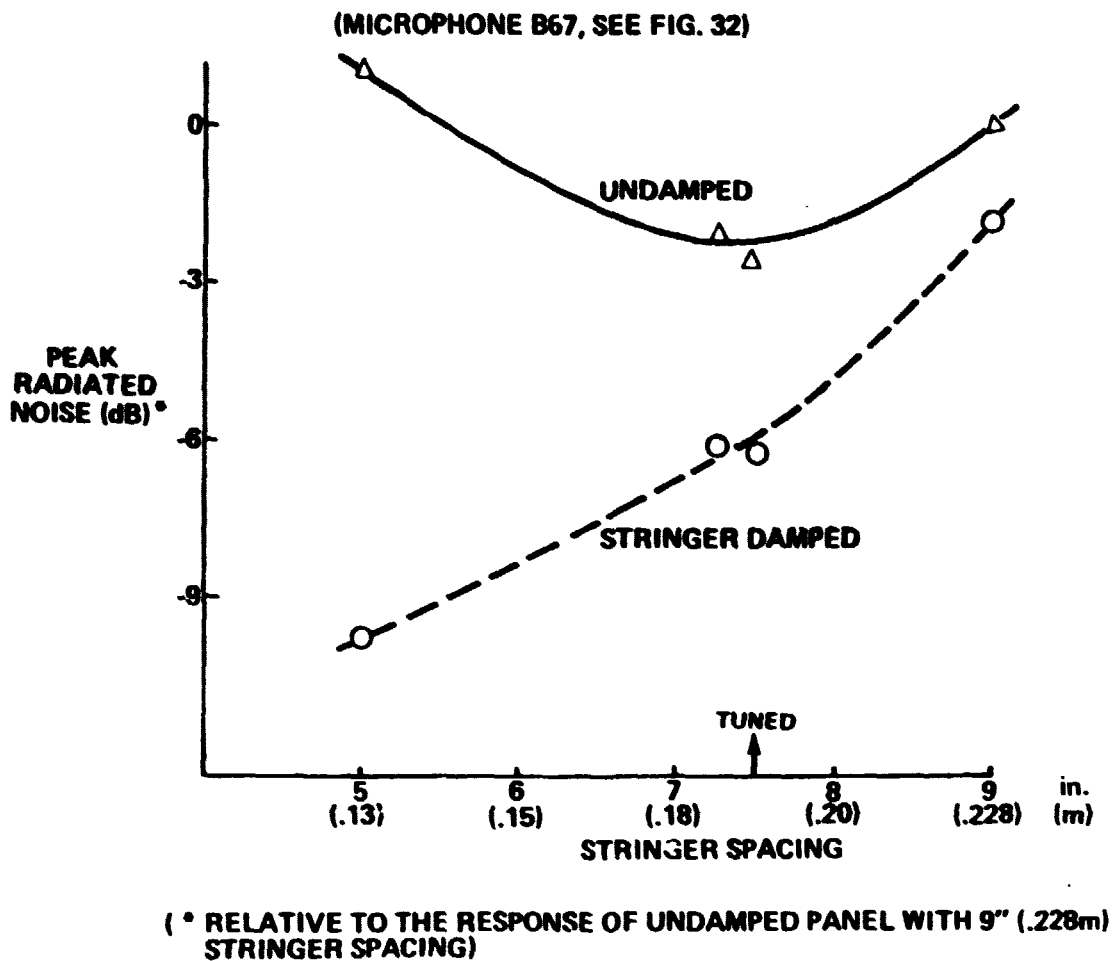


Figure 22.—Reduction of Peak Low Frequency Sound Radiation by Tuning and Damping

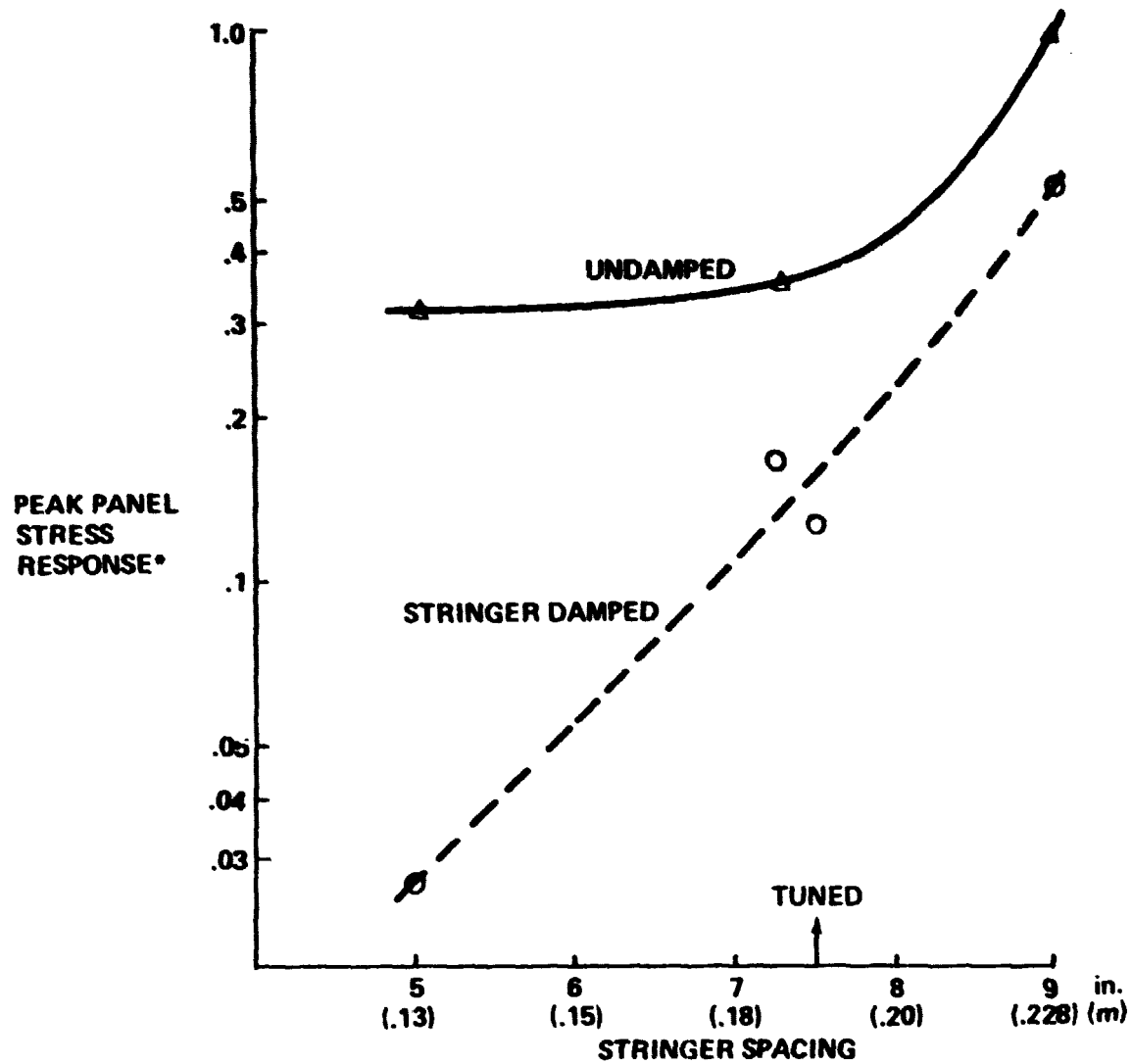
5.2.4 STRESS RESPONSE

In order to get an indication of the effect of intrinsic structural tuning on the sonically induced stresses in the fuselage structure, the bending stress response was measured at various points on each panel. The stress data recorded in this test contained a large amount of noise at 60 Hz and its higher harmonics. For this reason the stress responses are compared on the basis of the PSD responses of the most dominant modes.

5.2.5 SKIN BENDING STRESS RESPONSE

The variation of peak skin bending stress response with the stringer spacing is shown in figure 23. It can be seen that the maximum skin bending stress is reduced by about 60% as the stringer spacing is reduced from 9 in. (0.22 m) to 7.5 in. (0.19 m). The response levels off as the stringer spacing is reduced to 5 in. (0.13 m). Thus, when the tuning condition is satisfied, maximum reduction is achieved with minimum weight penalty.

(AVERAGE OF GAGES 638, 639, & 641, SEE FIG. 34)



(*NORMALIZED WITH WITH RESPECT TO THE UNDAMPED PANEL WITH 9" (.228 m) STRINGER SPACING)

Figure 23.—Reduction of Peak Low Frequency Skin Stress by Tuning and Damping

With damping treatment applied on the stringers, the response is reduced even further. Stringer damping reduced the skin bending stress response of the panel with 9 in. (0.228-m) stringer spacing by about 50%, that of the tuned panel by about 68%, and that of the panel with 5-in. (0.13-m) stringer spacing by about 90%. Thus, maximum skin bending stress reduction was obtained with reduced stringer spacing and increased stringer damping.

5.2.6 STRINGER BENDING STRESS RESPONSE

The variation of peak stringer bending stress response with the stringer spacing is shown in figure 24. Even for the undamped panels, the stringer bending stress response is minimum when the panel is intrinsically tuned. At first glance this is somewhat unexpected, since the stringer participation increases as the tuning condition is approached. However, the tuned panel had the maximum damping in the skin-stringer joints even when no damping treatment was applied on the stringers. This increase in damping reduced the stringer bending stress to the minimum value.

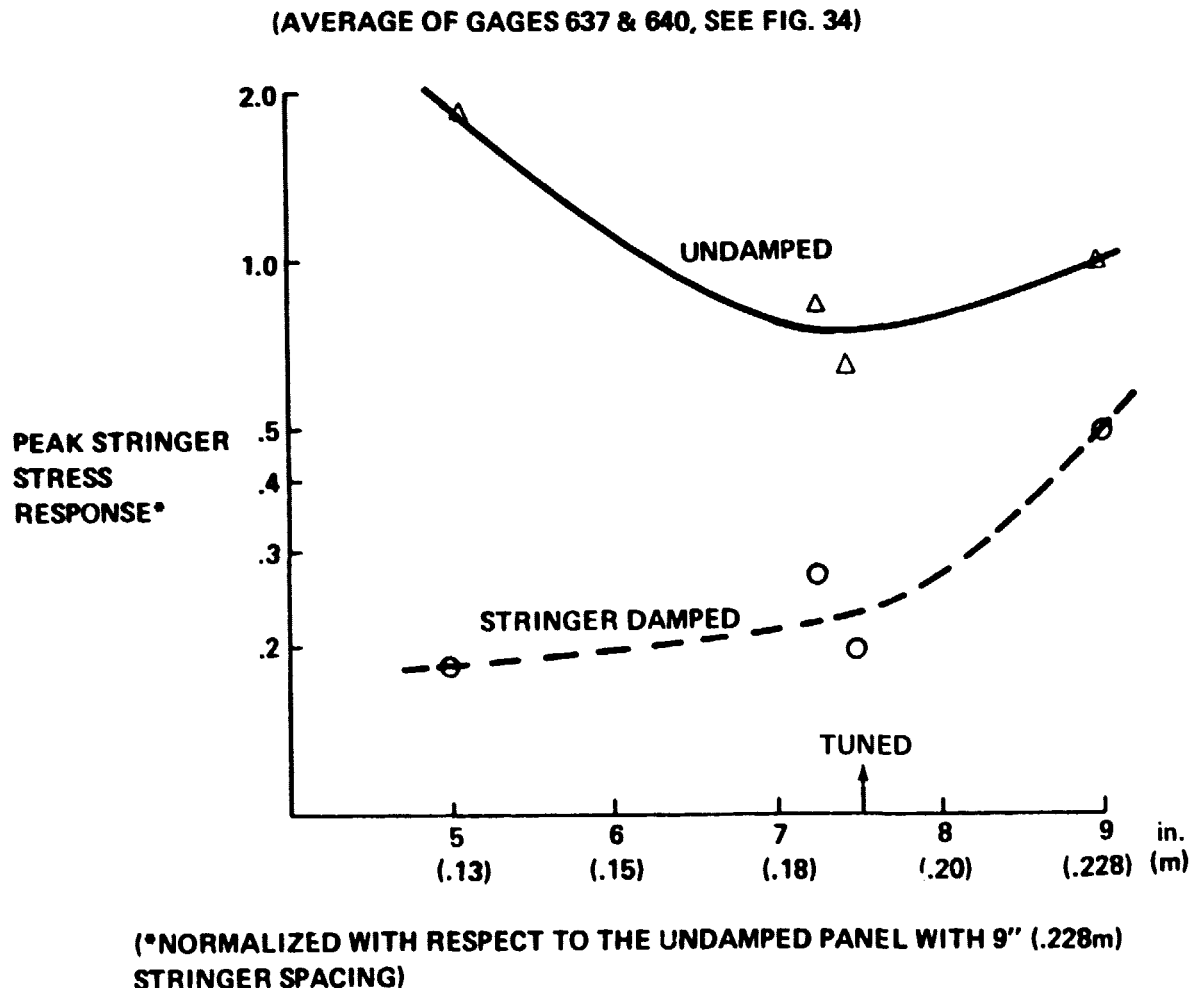


Figure 24.—Reduction of Peak Low Frequency Stringer Stress by Tuning and Damping

Thus, the tuned structure concept holds a significant potential for improving the current sonic fatigue design technology. However, the reduction in the skin and stringer bending stresses caused by tuning is also associated with an increase in structural damping because of frictional energy dissipation in riveted skin-stringer joints which may cause increased fretting in these joints. A certain amount of sonic fatigue testing of tuned and untuned skin-stringer panels will, therefore, be necessary to establish the actual improvements in the sonic fatigue life that can be accomplished by tuning the skin panel and the stringers. In addition, this may suggest application of jointing compounds that allow increased friction with no increase in fretting.

6.0 FIELD TEST DESCRIPTION

6.1 APPROACH

In the laboratory test, discrete point force excitations were used to simulate the distributed near field jet noise field. It was observed that the predicted response spectrum caused by near field jet noise could be simulated by superimposing the response spectra due to excitations at the skin bay centers and at the center of each stringer. This was particularly so in simulating the response of the tuned panel. The field test provided an opportunity to examine the validity of this procedure.

Three of the panels were exposed to the near field noise environment generated by the full scale upper surface blown (USB) YC-14 engine-wing-flap test rig, under the joint NASA-USAF contract. The test panels were mounted in the opening of the portable anechoic box. The overall test setup and the position of the anechoic box relative to the engine are shown in figures 25 and 26. The purpose of the test was to measure the responses of the panels to the near field engine noise environment and to compare the response spectra so obtained with those obtained from the laboratory test. The tuned panel was tested in undamped and damped conditions. The other two panels with 9-in. (0.228-m) and 5-in. (0.13-m) stringer spacing were tested in the damped condition only.

In order to assess the levels, spectrum shapes, and variations of the random pressure field incident on the skin-stringer panel, and also to assess the coherence and phase characteristics of the excitation, a boilerplate panel was mounted in the door of the anechoic box. Thirteen microphones were flush mounted on the boilerplate panel. Four of these were used to monitor the levels and the spectrum shapes at the four corners of the boilerplate. The other nine were used to measure the spectrum shapes, coherence, and phase characteristics of the incident near field noise.

The boilerplate panel layout is described dimensionally in figure 27. Panel thickness was 3/8 in. (0.0095 m). Viscoelastic damping was applied to the inner surface area lying within the mounting frame of the anechoic box doorway. A photograph of the boilerplate panel with the 13 microphones is shown in figure 28.

Each skin-stringer panel required an associated pair of filler panels to fill the opening in the anechoic box. Each filler panel was 3/8 in. (0.0095 m) thick. Each was treated with constrained viscoelastic damping, and each had provisions for flush mounting microphones in the corners adjacent to the test panel. A typical skin-stringer panel with the filler panels is shown in figure 29.

6.2 TEST PANEL INSTALLATION

The same anechoic box used in laboratory tests was used in the field tests. Test panels were installed on a hinged heavy steel door frame designed to fit the anechoic box opening. The hinged frame was closed during test by three screws on the side opposite the hinges. The hinged frame provided convenient access for calibration of microphones inside the box and test panel sensor changes.

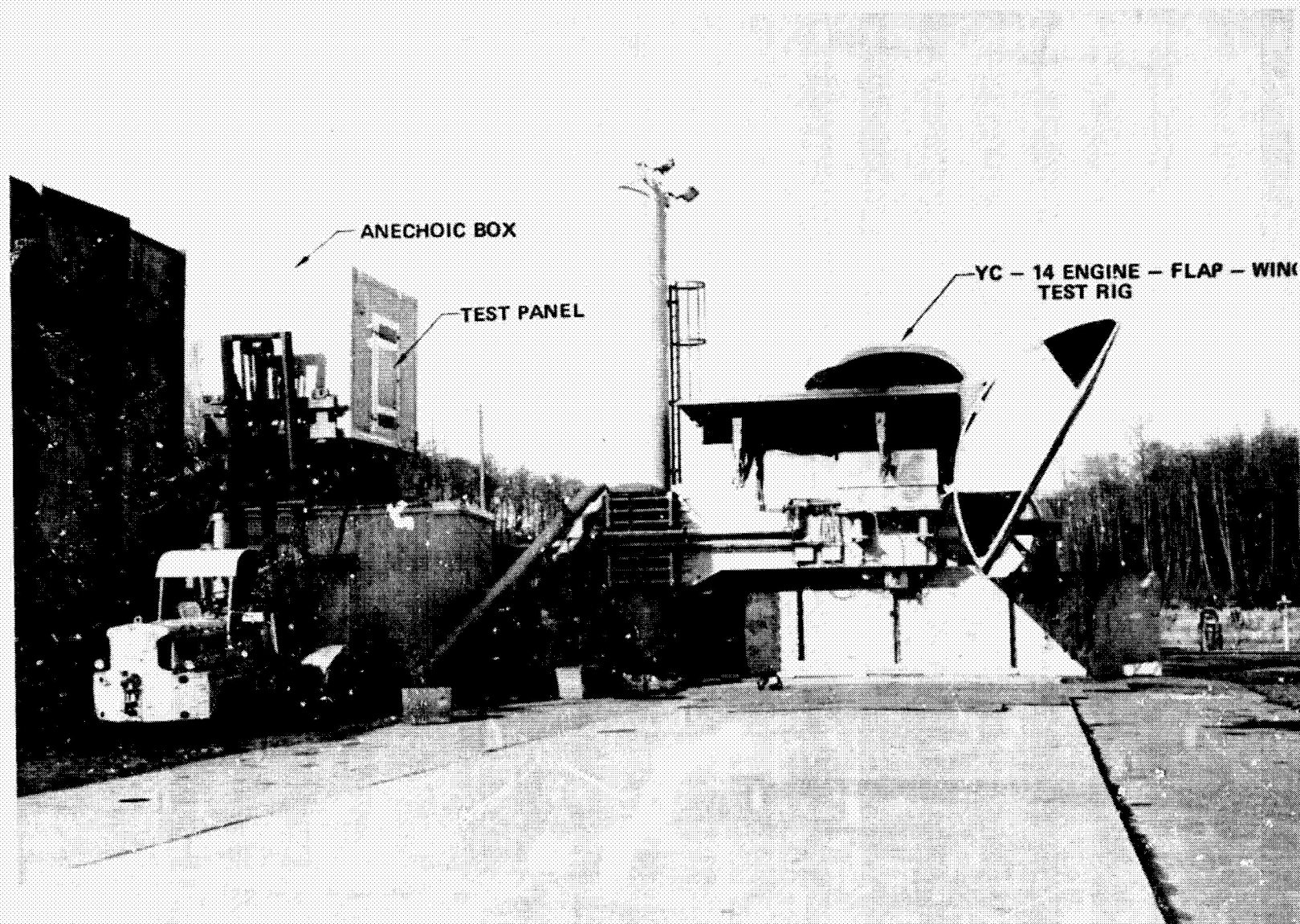


Figure 25.—Field Test Setup

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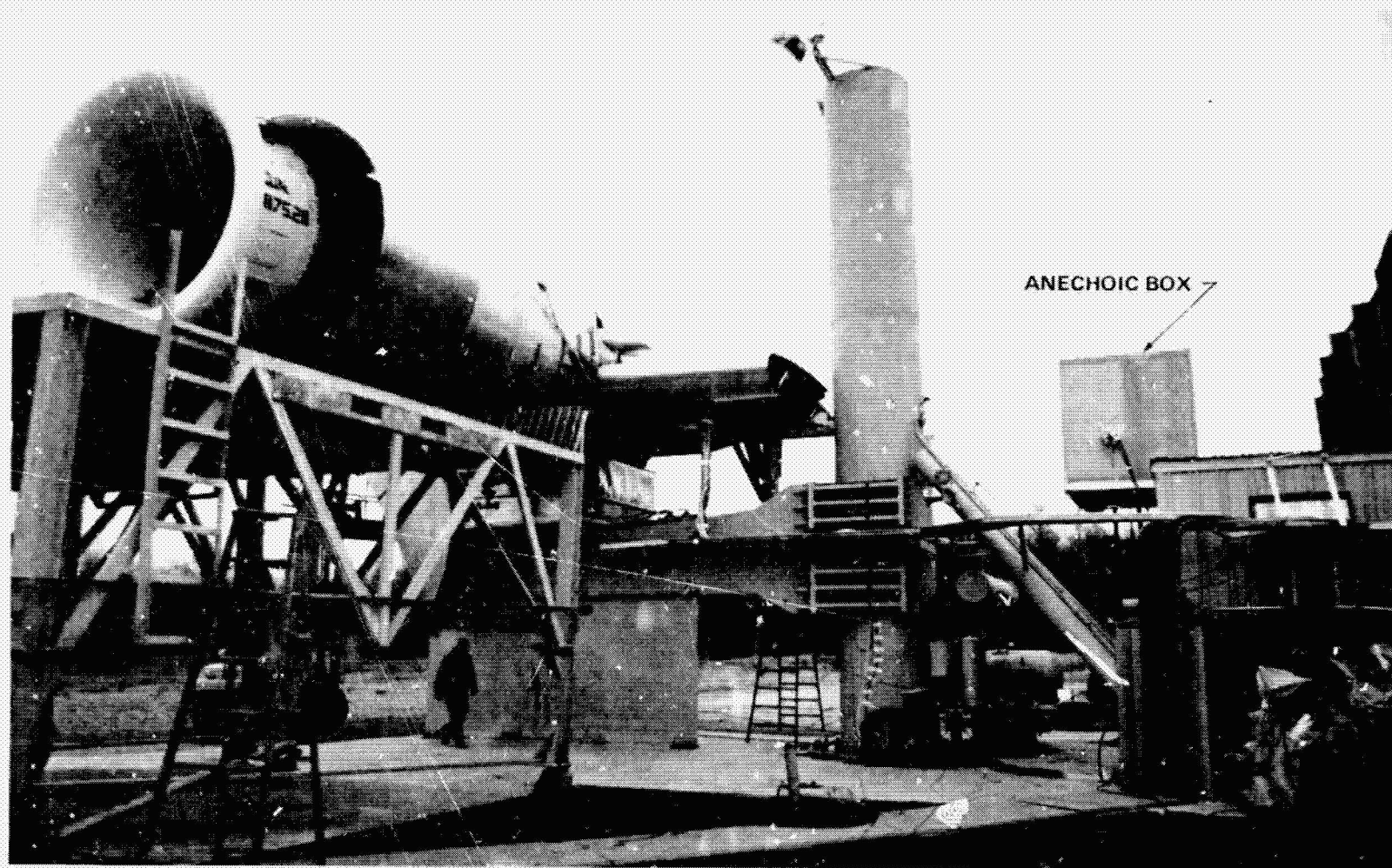


Figure 26.—Test Setup Relative to the Engine Inlet

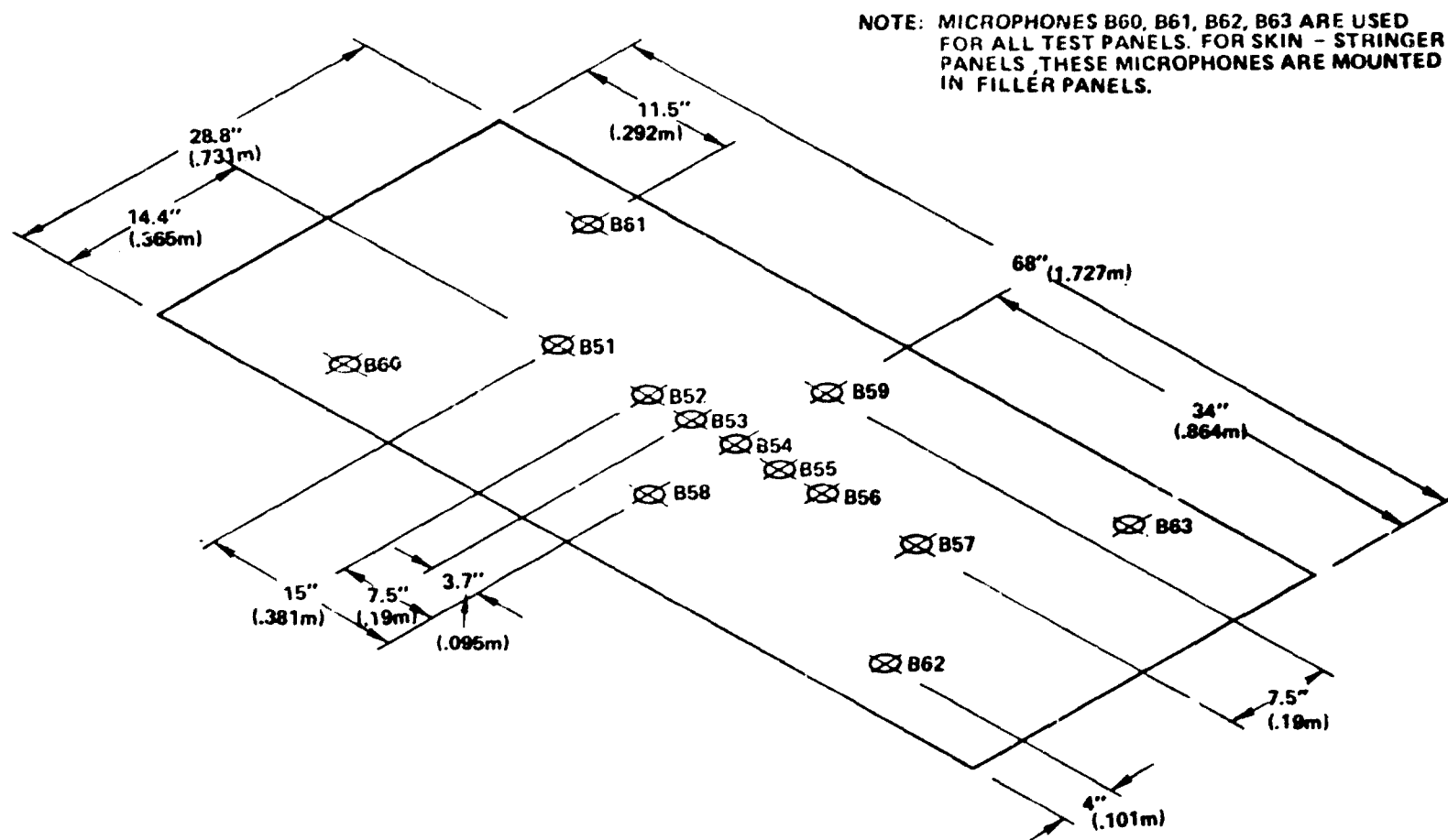


Figure 27.—Boilerplate Panel and Microphone Locations

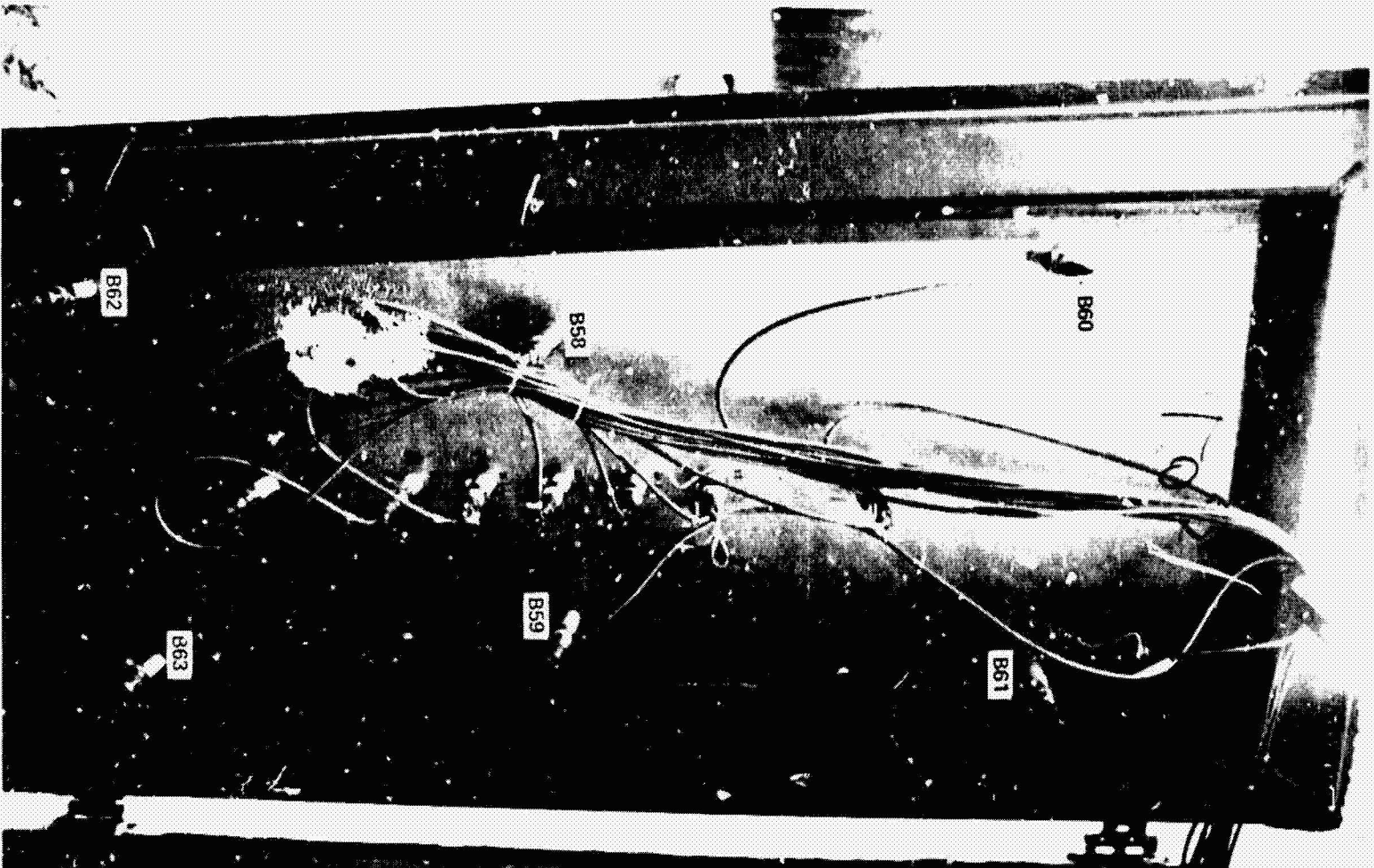


Figure 28.—Boilerplate Panel With Flush-Mounted Microphones

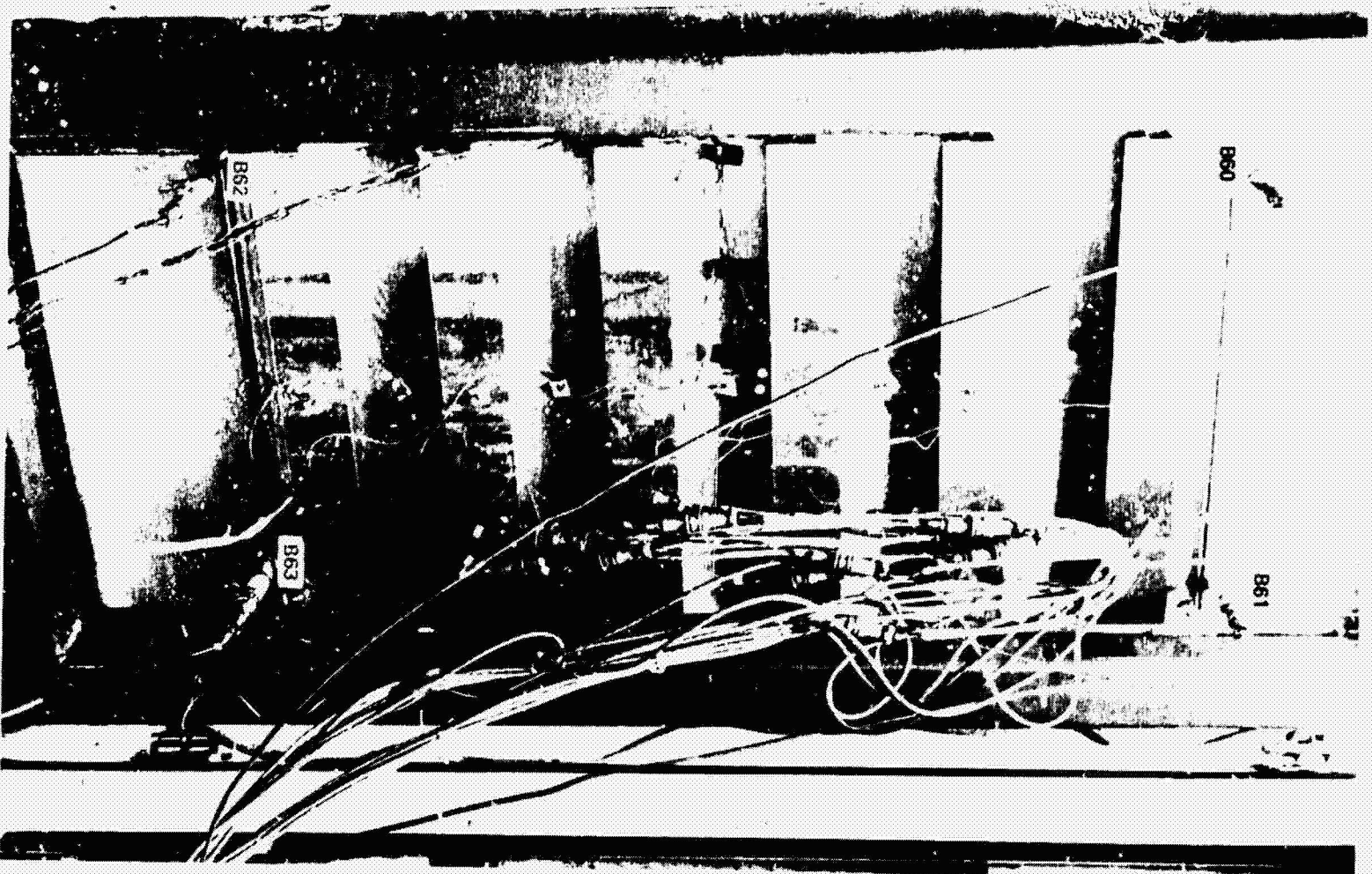


Figure 29—Skin Stringer Panel with Accelerometers, Strain Gages, and Four Corner Microphones. Flush Mounted on the Filler Panels

The same panel installation procedures were used for both laboratory and field tests. Mounting bolts were torqued to 80 lb-in. (9.04 Nm), and filler panels were not in direct contact with the skin-stringer test panels. Standard foil-backed damping tapes were applied over the gaps on the exterior surfaces between test panel end bays, which extended about 3 in. (0.076 m) beyond the terminating stringers and the filler panels. All potential acoustic leaks between panels and mounting frame, and between mounting frame and anechoic box were sealed and covered with foil-backed damping tapes.

The application of damping tape to seal all joints around the panel is illustrated in figure 30 for the boilerplate panel. Prior to entering the anechoic chamber through penetration sleeves, instrumentation cables were routed through sand-filled "U" traps (shown in fig. 30) as a further means for preventing acoustic leaks.

6.3 ANECHOIC BOX LOCATION

The anechoic box used in laboratory and field tests was an Eckel Industries portable AN-ECK-OIC chamber model number 888-150 with a specified cutoff frequency of 150 Hz. The outside dimensions define an 8-ft 7-in. (2.616 m) cube which weighs approximately 7000 lb. (3174.5 kg). Floor design for vibration isolation permitted elevating the box for field testing without impairing transmission loss characteristics. Section joints were waterproofed for the field test environment, and the box interior was heated by a 100-W light bulb to lower the relative humidity.

A fork lift with maximum lifting capacity of approximately 15 000 lb (6803.8 kg) was used as the basic support platform for the anechoic box. This economical system allowed quick, convenient working access to the box for panel changes and instrumentation checkout and calibration. A support cradle with fork pockets and attachment points for guys provided a secure tie between box and lift. The guying installation stabilized the box against motion during test without interfering with the raising/lowering convenience provided by the fork lift.

The position of the anechoic box during panel testing is shown in figure 25. The final box location placed the vertical centerline of a test panel mounted on the box at about 25 ft (7.62 m) downstream from the nozzle exit plane and about 25 ft (7.62 m) to the left (on the pylon stand side) of the nacelle centerline. The horizontal centerline of a test panel mounted on the box was at about the same elevation as the engine centerline.

The location decision, based on model scale data for engine over wing configuration, was influenced by the following factors:

1. No impingement of jet flow on the box
2. Representative sound pressure levels
3. Broad spectrum shape
4. Correlation and convection properties of the excitation field
5. Minimizing effect of ground reflections
6. No significant impact on YC-14 ground test measurements

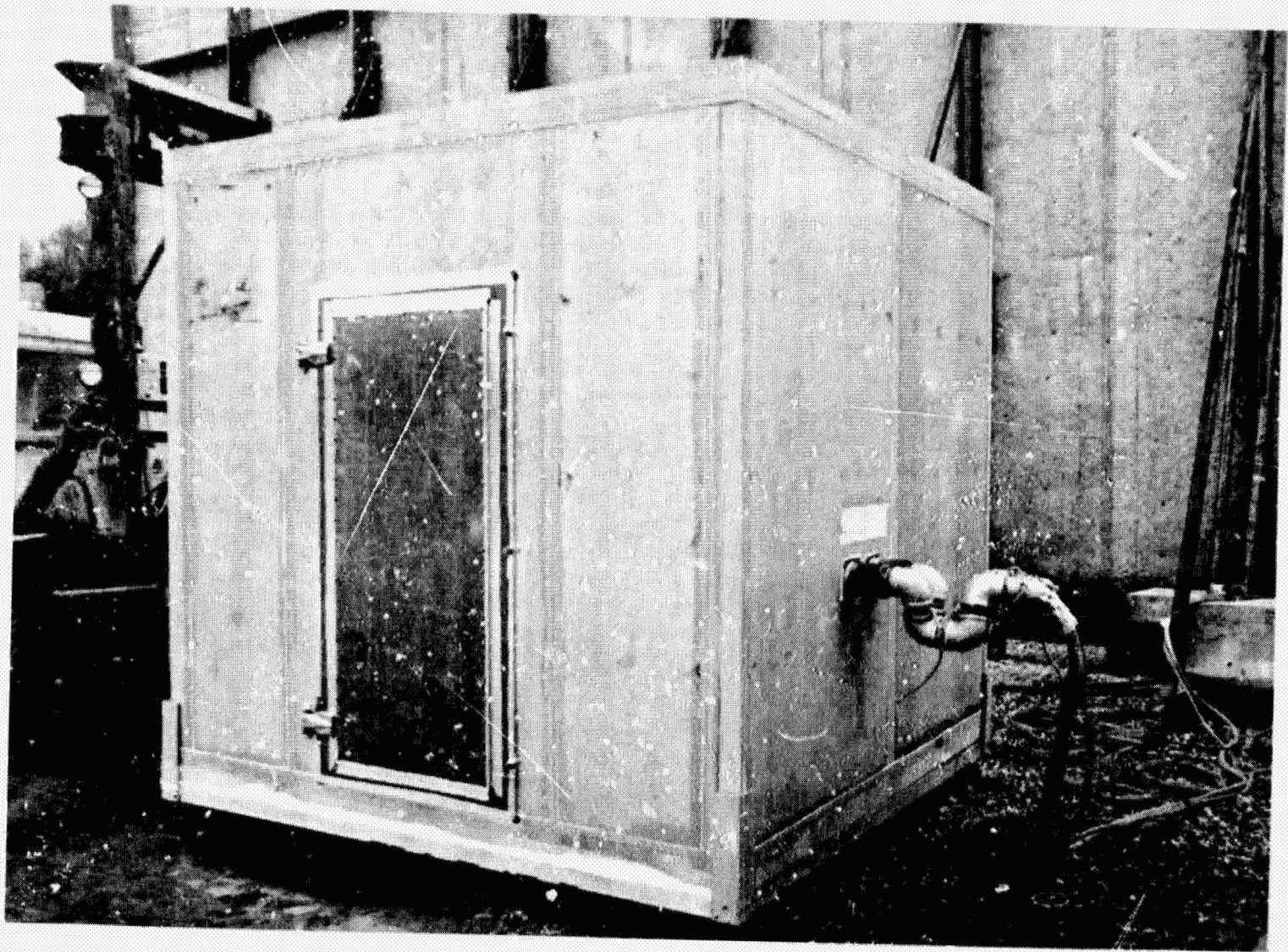


Figure 30.—Anechoic Box With the Boilerplate Panel Installed

Early testing disclosed that with the USB nozzle door open, the jet flow was impinging directly on the test panel even with the anechoic box moved as far from the engine centerline as site configuration would permit. Consequently, all tuned structure data were taken with the USB door closed.

6.4 INSTRUMENTATION

6.4.1 SENSOR DEFINITION AND LOCATION

Microphones

Simultaneous pressure measurements were made at 13 surface points on a rigid panel blank in the area where test panels were mounted. Thirteen 1/4-in. (0.00635-m) diameter B&K microphones were flush-mounted in the boilerplate panel designed to insert in the 24-in. (0.609-m) x 60-in. (1.524-m) opening of the box. The boilerplate microphone locations are shown schematically in figure 27 and with microphones installed in figure 28. Data from the nine centerline microphones were recorded to establish variations in spectrum levels over the panel and also the phase and coherence properties of the pressure field in directions parallel and perpendicular to the stringers.

The four corner microphones were used for monitoring sound pressure levels for all test runs. For skin-stringer panels, the four corner microphones were located on the filler panels associated with each panel.

Simultaneous pressure measurements were made at six points inside the anechoic box to provide data on transmission loss across the test panels. These measurements and the summation of these measurements were recorded for each test condition. Locations of microphones inside the box are shown schematically in figure 31. The microphone installation inside the anechoic box is shown in figure 32.

The data from a microphone located on the upper wing surface was also recorded as a reference signal simultaneously with tuned structure measurements.

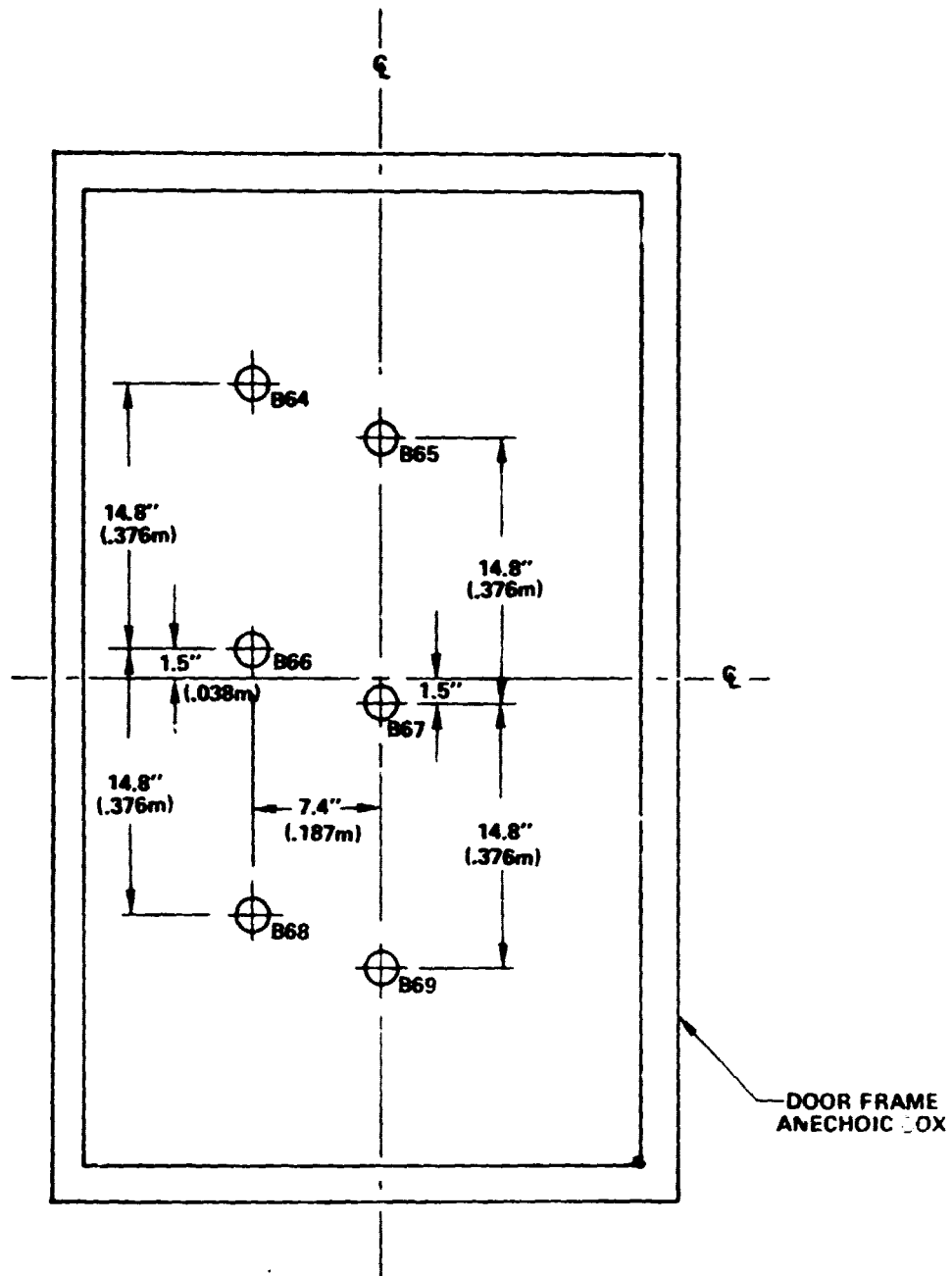
Accelerometers

Simultaneous acceleration measurements were made at 10 points on each skin-stringer panel. Panel accelerometer locations are identified in figure 33.

The 11th accelerometer was mounted on the left hand door frame (viewed from inside) on the test panel horizontal centerline to monitor response of the anechoic box installation to engine fluctuating pressure fields and ground vibration. Figure 29 shows the accelerometers installed on the test panel. Accelerometers were attached with dental cement. Signals from the 11 accelerometers were simultaneously recorded on the same tape.

Strain Gages

Simultaneous strain measurements were made at nine points on the skin-stringer panels. Strain gage locations are defined in figure 34.



FRONT VIEW LOOKING INTO BOX

NOTE: ALL MICROPHONE FACES WERE 4" (.101m) AWAY FROM INSIDE SURFACE OF PANEL 'SKIN'

Figure 31.—Location of Microphones Inside the Anechoic Box

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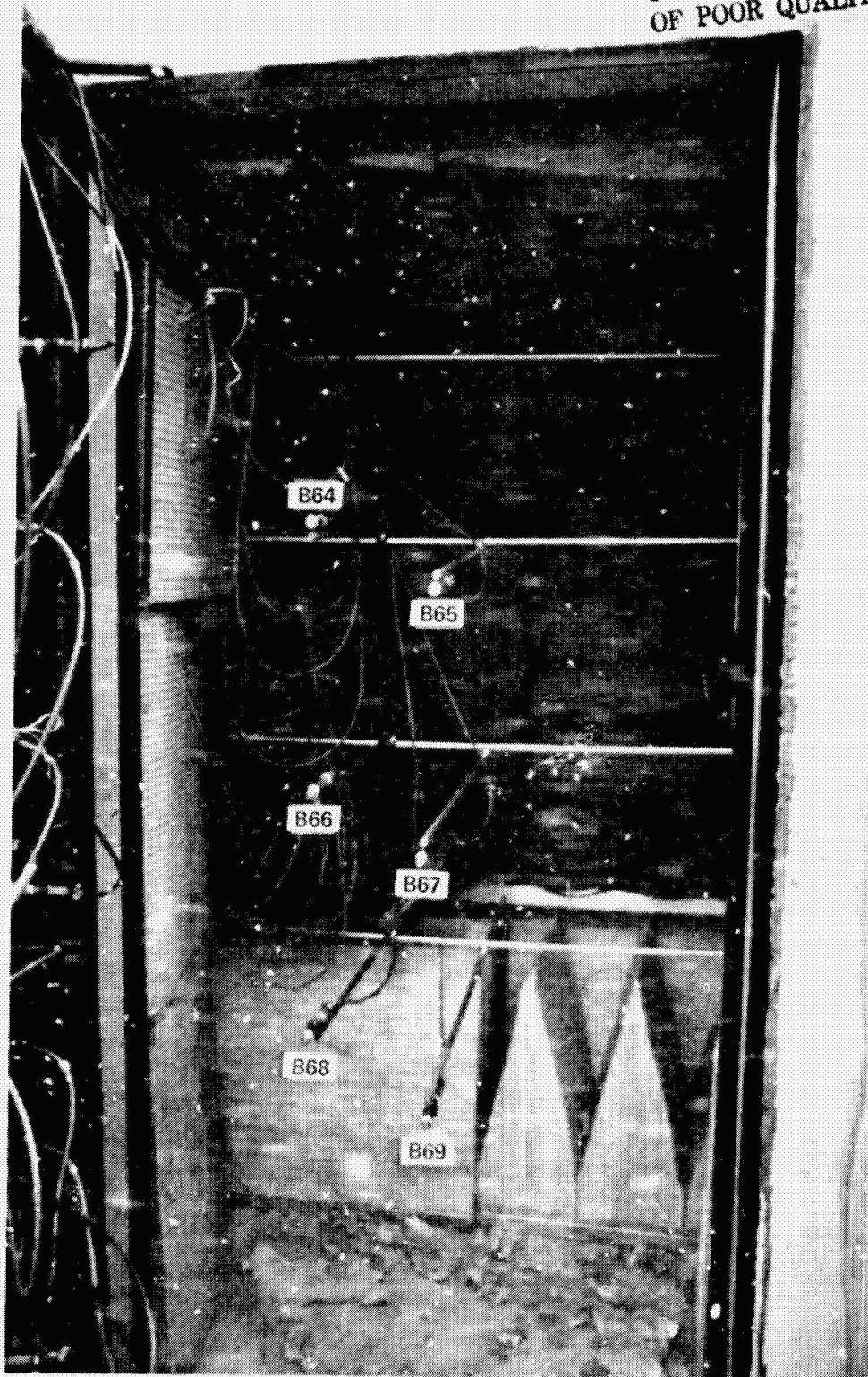


Figure 32.—Microphone Location Inside the Anechoic Box

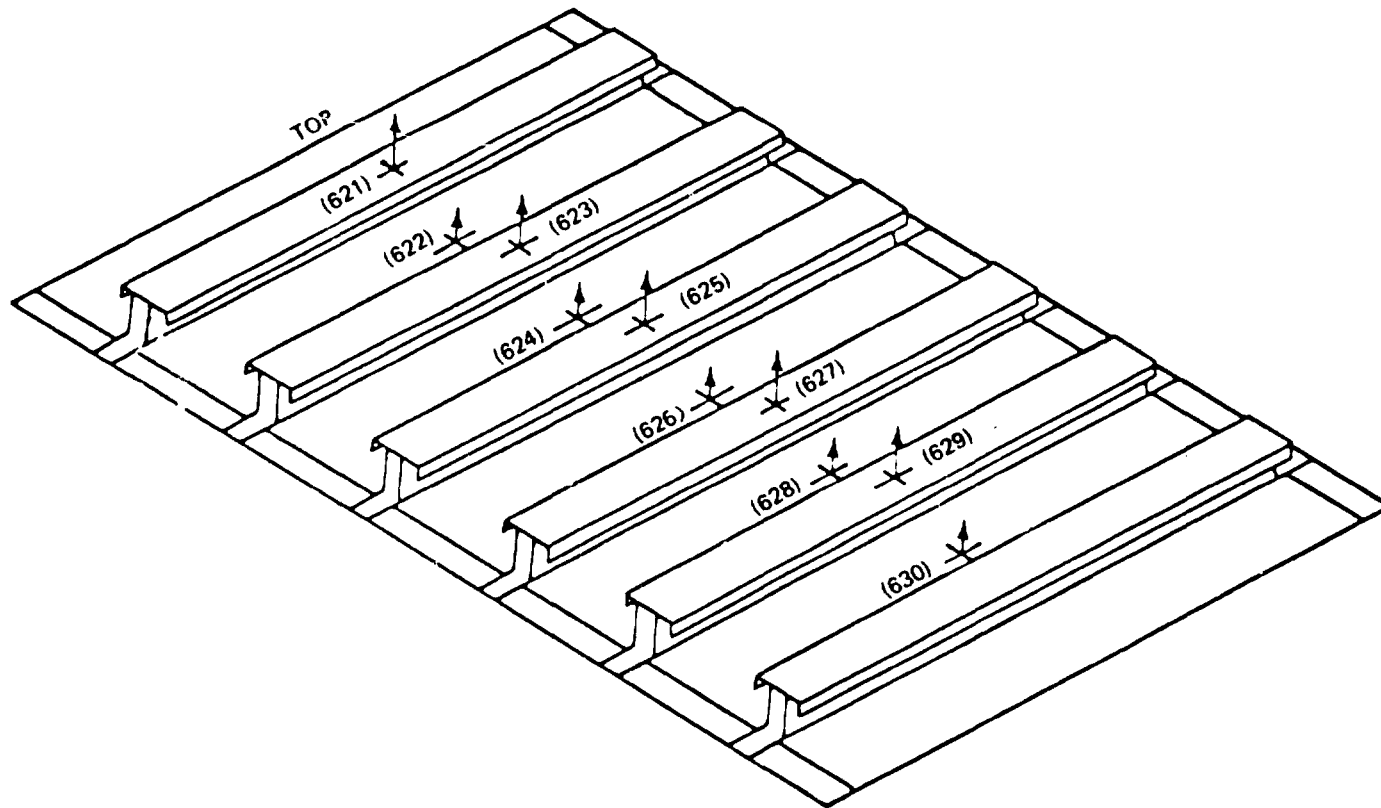


Figure 33.—Location of Accelerometers and Ground Test Instrumentation Requirement (GTIR) Numbers

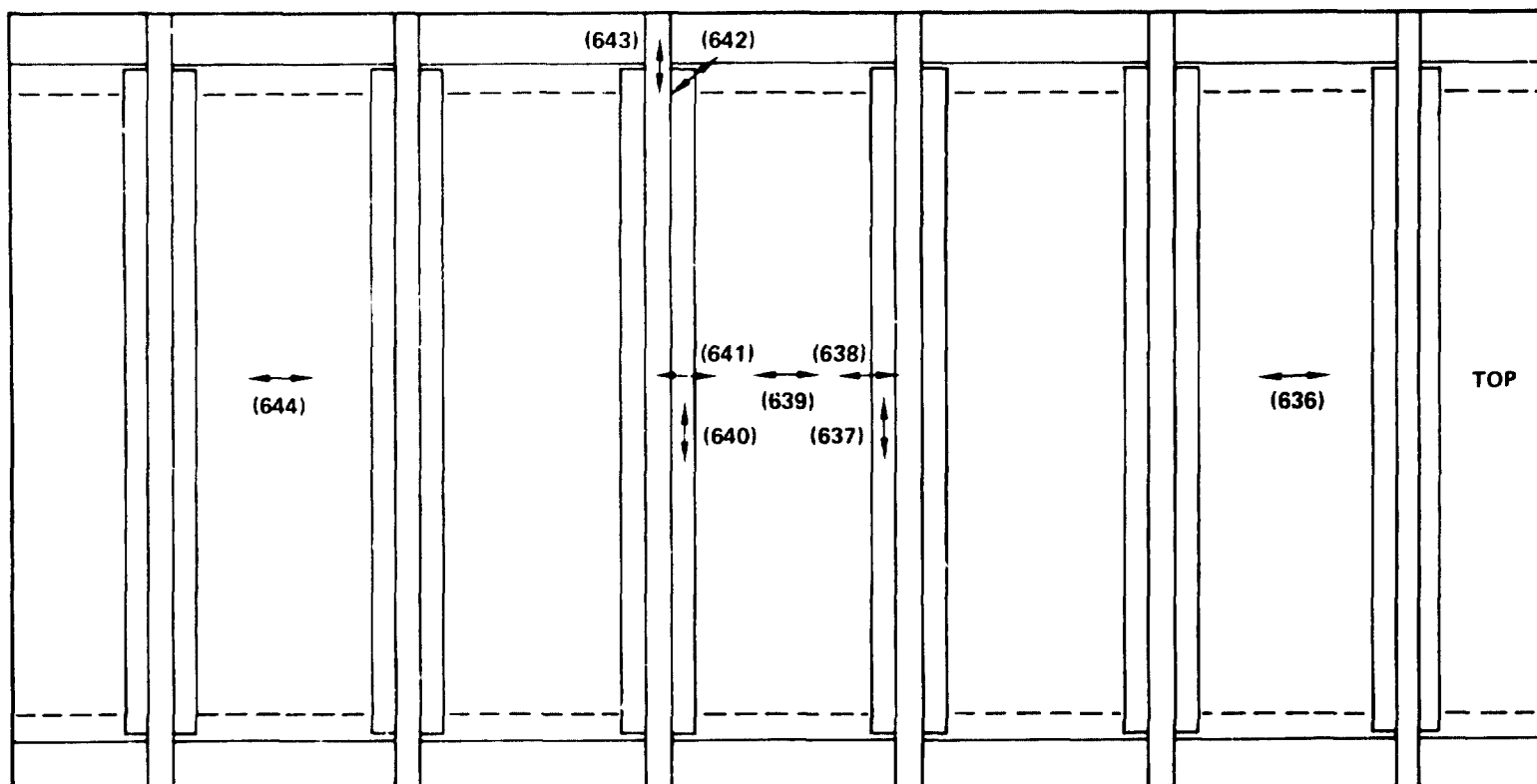


Figure 34.—Location of Strain Gages and Ground Test Instrumentation
Requirement (GTIR) Numbers

6.4.2 MEASUREMENT SYSTEMS

All data including microphone, accelerometer, and strain gage measurements were recorded simultaneously during each test condition. Microphone signal conditioning and recording equipment were located in the acoustics trailer; accelerometer and strain gage signal conditioning and recording equipment were located in the dynamics test trailer. Data recorded with both measurement systems were time-related by means of a common IRIG B time code which was recorded on all data tapes.

The acoustics data acquisition system is illustrated in figure 35. The dynamics data acquisition system is illustrated in figure 36.

6.5 TEST CONDITIONS/CONFIGURATIONS

Field test data for all panels were taken at two consistent engine power settings of 2943 and 3684 rpm (corrected N_1) corresponding approximately to 50% and 90% corrected installed gross thrust.

Test conditions and configuration pertinent to tuned structure data acquisition are summarized in table 1.

Table 1.—Tuned Structure Field Test Conditions

Run	Condition	Corrected N_1 (rpm)	Test panel stringer spacing
10.01	05	2943	9 in. (0.228 m)
	10	3684	(Damped)
11.01	05	2943	7.5 in. (0.19 m)
	12	3684	(Damped)
12.02	10	2943	7.5 in. (0.19 m)
	20	3684	(Undamped)
13.02*	11	2943	5 in. (0.13 m)
	12	3684	(Damped)
15.02*	07	2943	Boilerplate
	08	3684	(Baseline)

Test Configuration

- USB doors closed
- Vortex generators down
- USB flaps up
- Reverser closed
- Bleeds closed
- Bellmouth inlet

* Inlet tube installed

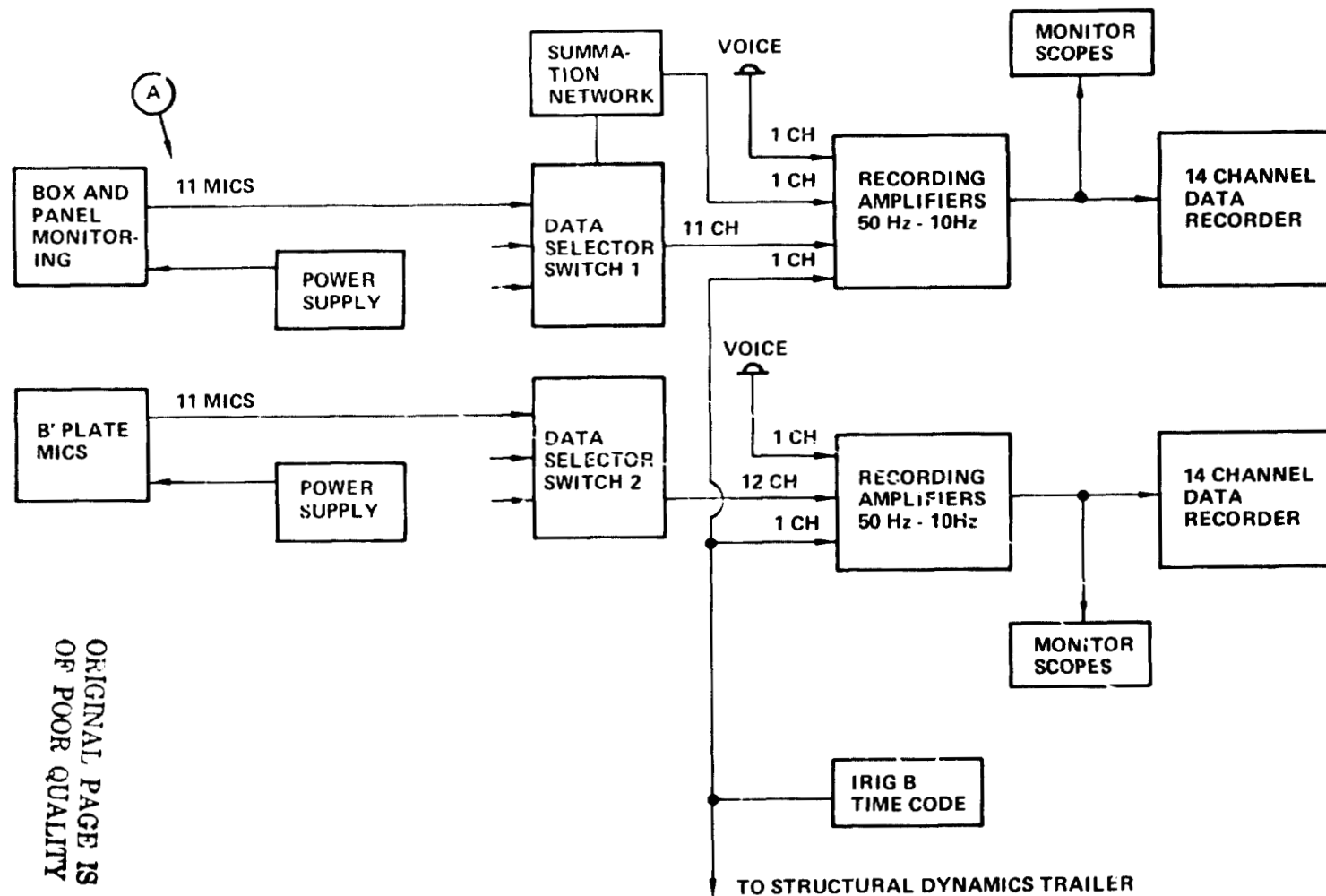


Figure 35.—Block Diagram—Acoustics Data Acquisition System

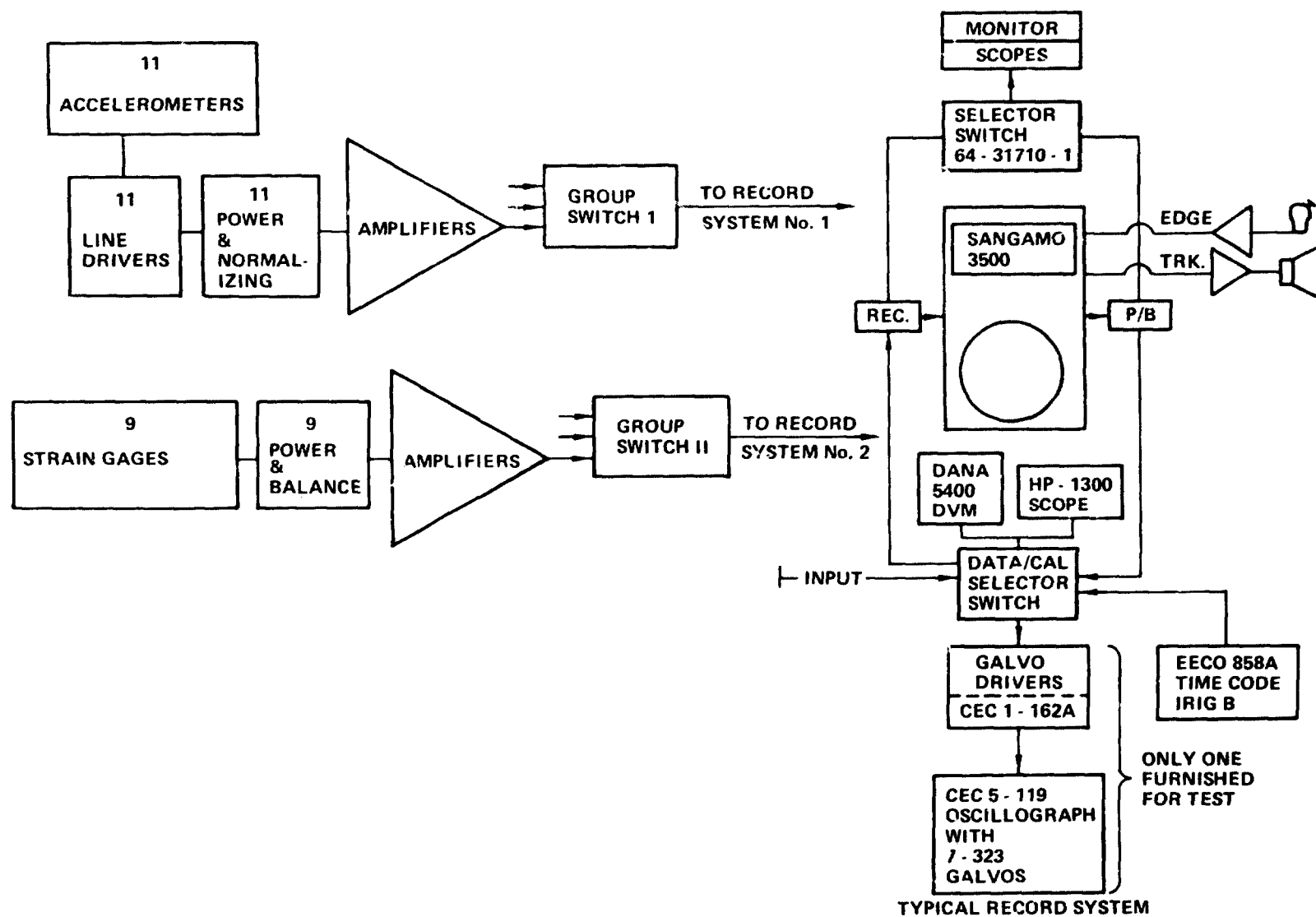


Figure 36.—Block Diagram—Structural Dynamics Data Acquisition System

6.6 ASSESSMENT OF DATA VALIDITY

Early testing with USB door open showed that jet flow was impinging directly on the test panel even with the anechoic box moved as far from engine centerline as site configuration would permit. Therefore, tuned structure data were taken with flaps up and USB door closed. However, the panel microphones were exposed to impinging flow, when YC-14 program conditions with USB door open were run during the same test. As a result, some panel microphone diaphragms were damaged by the impinging flow. The two bottom panel microphone signals were lost for the 5 in. (0.13-m) skin-stringer panel (run 13.02), and the three bottom microphones (corners and center) and left hand microphone on the horizontal centerline were lost on the boilerplate panel (run 15.02). The Kulite reference signal (microphone 43 on the upper wing surface on the engine centerline near the trailing edge) was lost during both conditions of run 10.01 (skin-stringer panel with 9-in. (0.228-m) stringer spacings).

Strain gage data were lost for run 12.02 condition 10 (panel with 7.5-in. (0.19-m) stringer spacing, at lower power setting) when the dynamics recorder mode switch was inadvertently placed in the wrong position.

In the PSD analysis of acceleration and stress data, a frequency bandwidth of 1 Hz was used. The time sample length was chosen so that the data were analyzed with 200 degrees of freedom. With 90% confidence, the true PSD level was between approximately 1.15 times the observed PSD and 0.85 times the observed PSD levels. In the PSD analysis of the acoustic signals, a bandwidth of 1.25 Hz was used. The time sample length was chosen so that the data were analyzed with 96 degrees of freedom. With 90% confidence, the true PSD level was between +1 dB and -1.2 dB of the observed PSD level.

7.0 RESULTS OF THE FIELD TEST

7.1 EXTERNAL PRESSURE SPECTRUM

The external noise level incident on the panels was measured by microphones flush-mounted on the boiler plate panel. Typical PSD plots (fig. 37) show a broadband noise spectrum with a peak level around 100 Hz. For both low and high power settings, the noise PSD levels rolled off approximately at the same rate for frequencies above 100 Hz. In addition to the peak around 100 Hz, there was a peak around 70 Hz associated with USB propulsion system. This peak is more pronounced in the spectrum of the microphone mounted on the wing (fig. 38).

7.2 COHERENCE AND PHASE CHARACTERISTICS OF THE INCIDENT PRESSURE FIELD

The coherence and phase properties of the excitation field were measured by nine microphones flush-mounted on the boilerplate panel (figs. 27 and 28). Three were used to measure the coherence and phase in the direction parallel to the stringers, and seven were used to measure coherence and phase in the direction perpendicular to the stringers. The measured data were analyzed to assess the coherence and phase of each microphone signal with that of the center microphone.

Figure 39 shows a strong coherence of signals up to about 1000 Hz in the direction parallel to the stringers. In the perpendicular direction, the coherence was very high for frequencies up to about 350 Hz. The coherence between microphones B51 and B54 dropped to a minimum value around 550 Hz and reached a fairly high level around 1000 Hz. The reason for this dip in coherence function is not clear. However, since all the panels had their frequencies of peak response below 350 Hz, the panels may be considered to have been excited by a highly coherent noise field.

Figure 40 shows the change of phase of the incident noise field in the directions perpendicular and parallel to the stringers obtained from a cross PSD analysis. In the perpendicular direction, the phased difference between the central microphone and the other microphones is small over the entire frequency range.

Also in the parallel direction, the change of phase is small especially below 350 Hz, although there is evidence of a trace wave travelling at a supersonic trace velocity. Therefore, it may be concluded that during the field test, all the panels were excited by a highly correlated and coherent external noise field, especially at lower frequencies.

7.3 PANEL RESPONSES

All the panels had their most important modes in the frequency range 100 to 350 Hz. This was observed during the laboratory and the field tests. For this reason, the strain response and acceleration response of all the panels had maximum levels above 100 Hz. However, in all cases the spectra of radiated noise inside the anechoic box showed a peak around 50 Hz (fig. 41). This peak was caused by sound transmission through the box walls and was not

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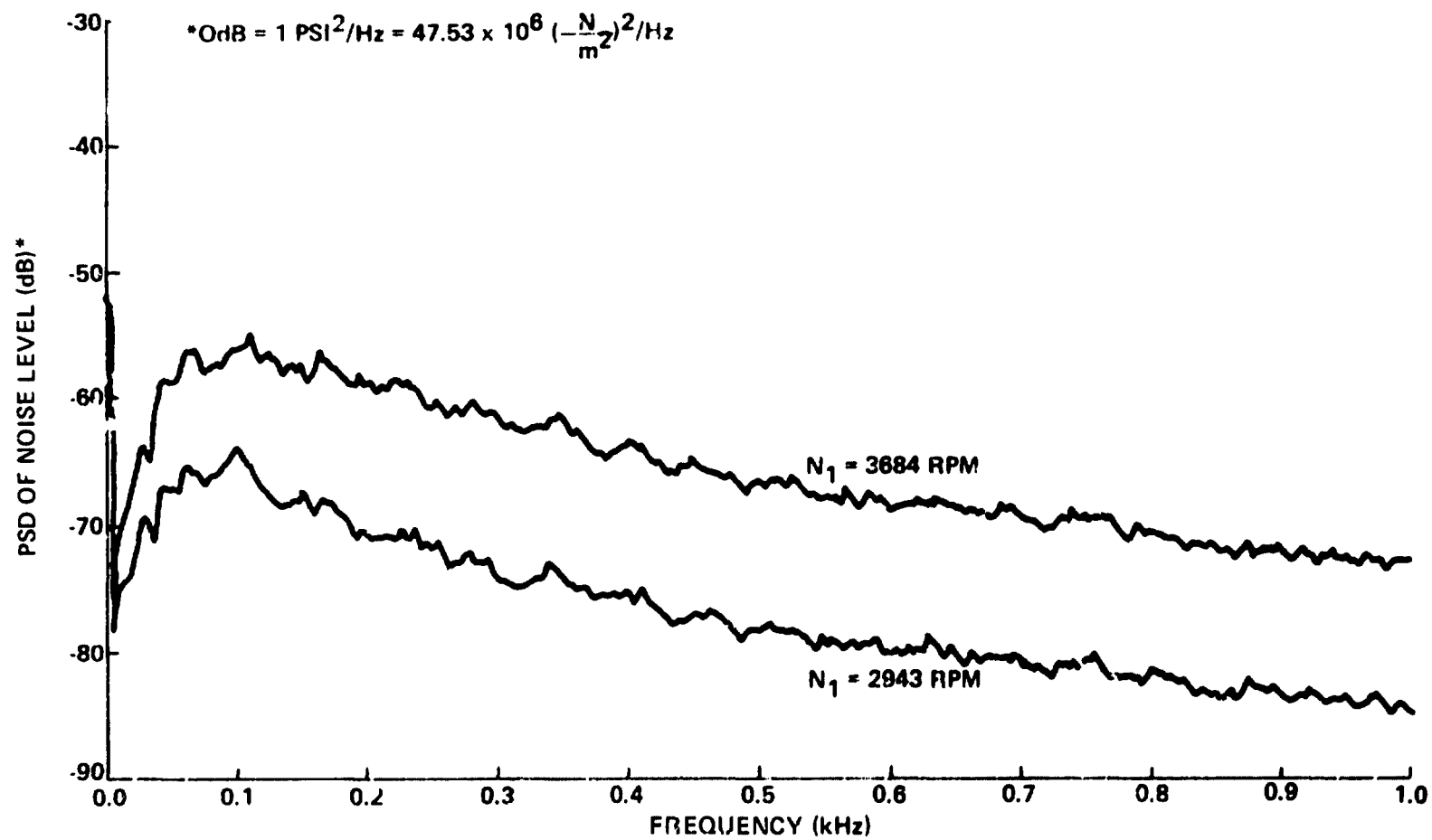


Figure 37. — Power Spectral Density of the Incident Noise

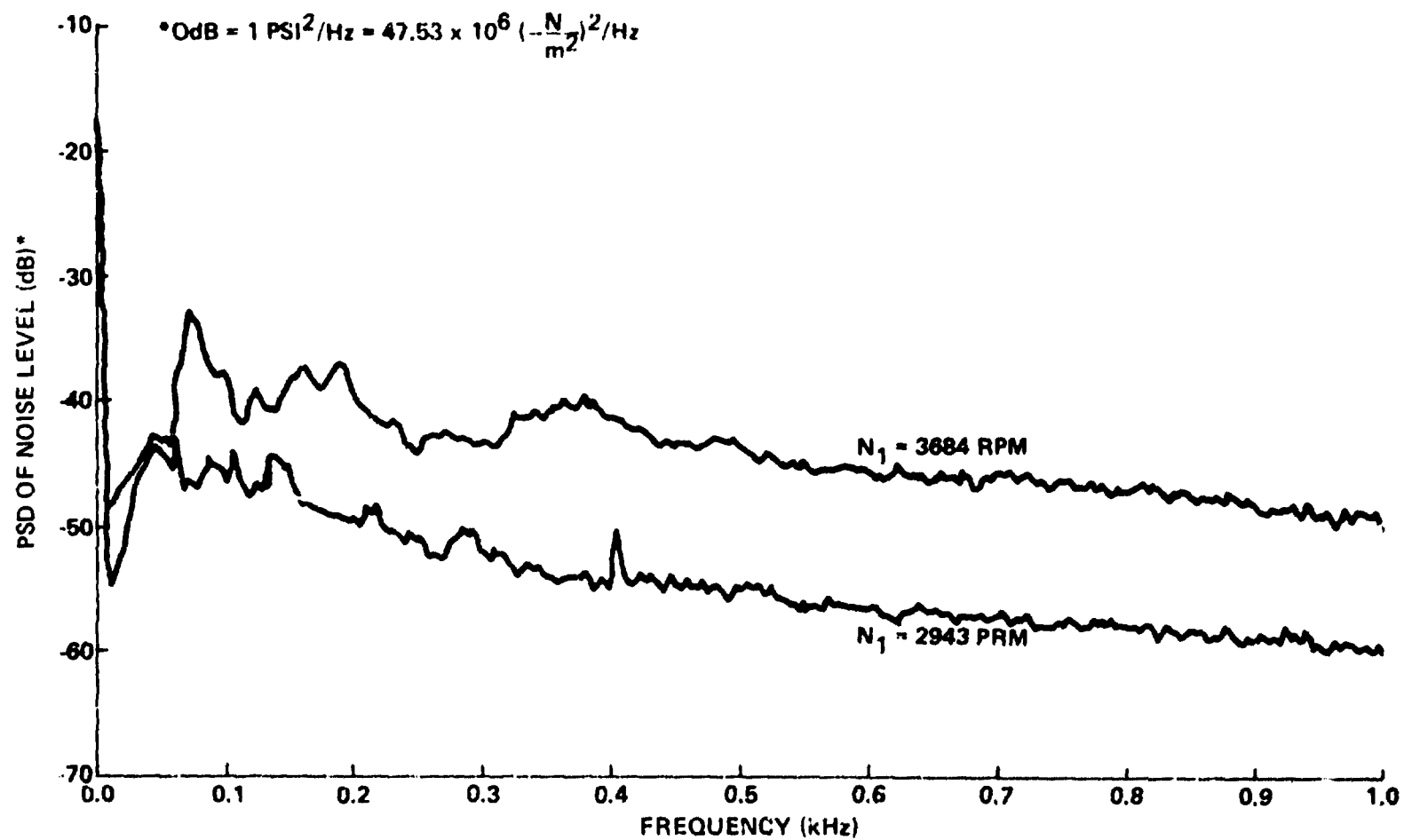


Figure 38.—Power Spectral Density of the Noise Measured by a Microphone Placed on the Wing Next to the Exit Nozzle

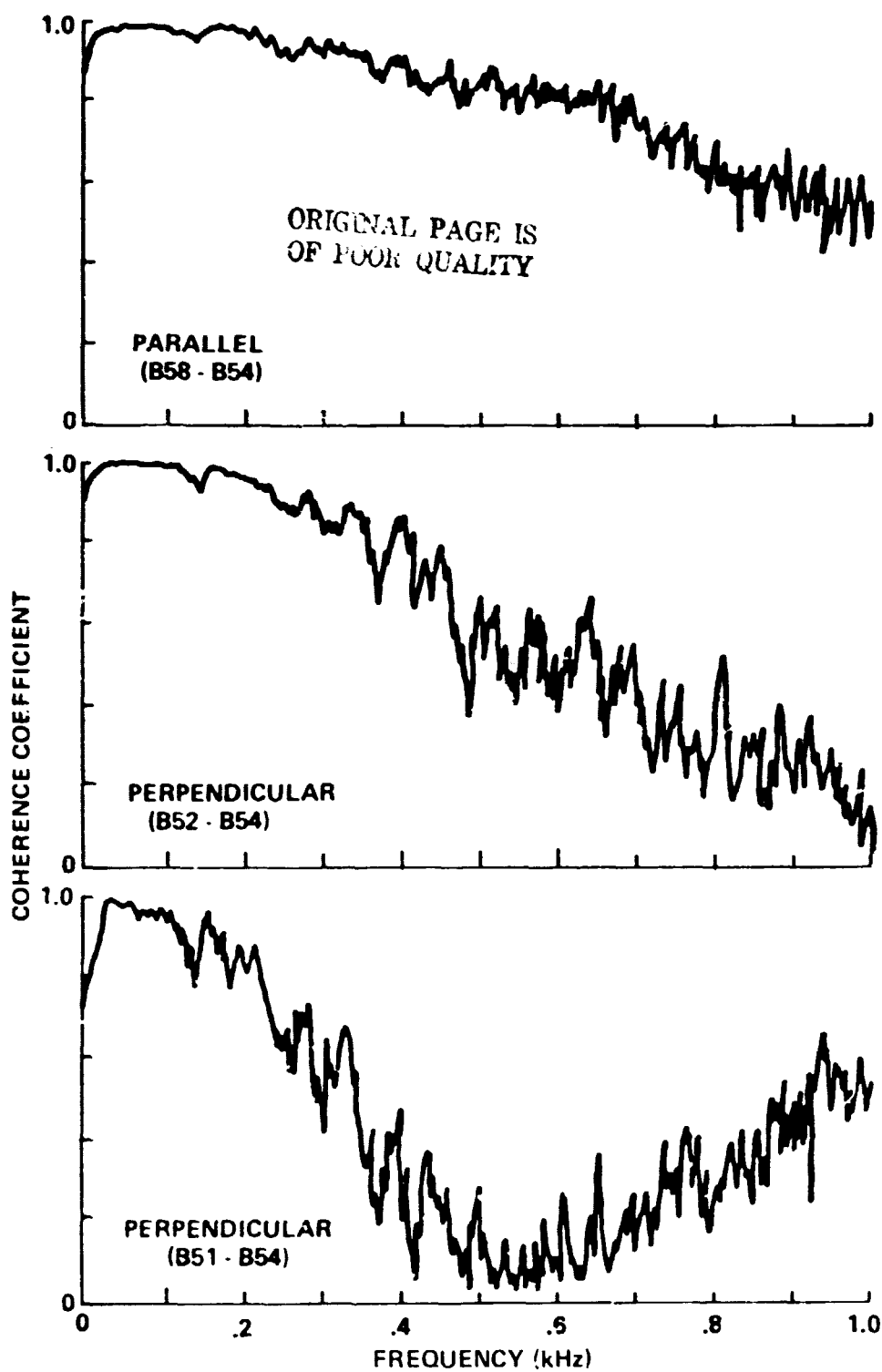


Figure 39.—Coherence of the Incident Noise Field
($N_1 = 3684$ rpm)

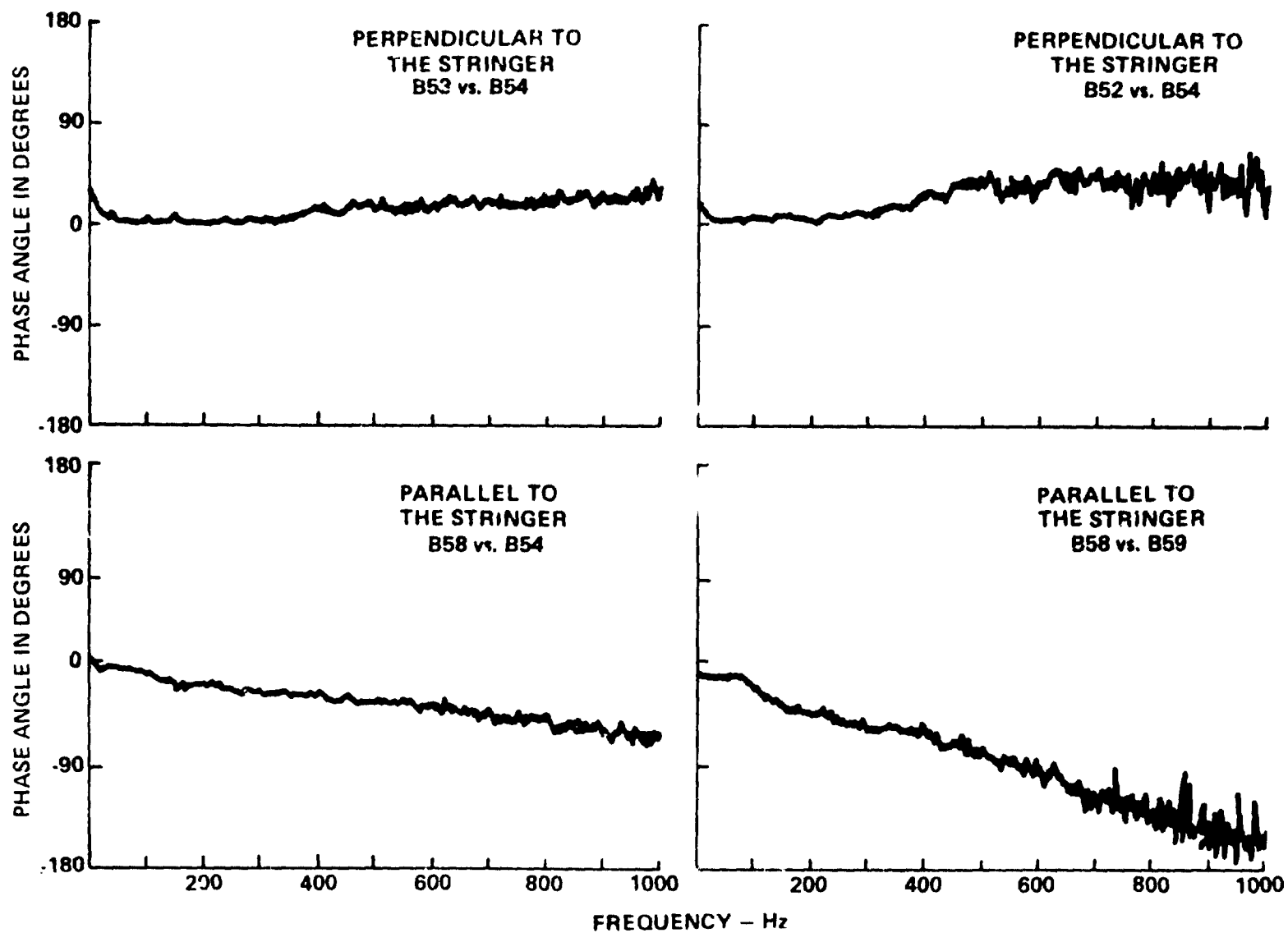
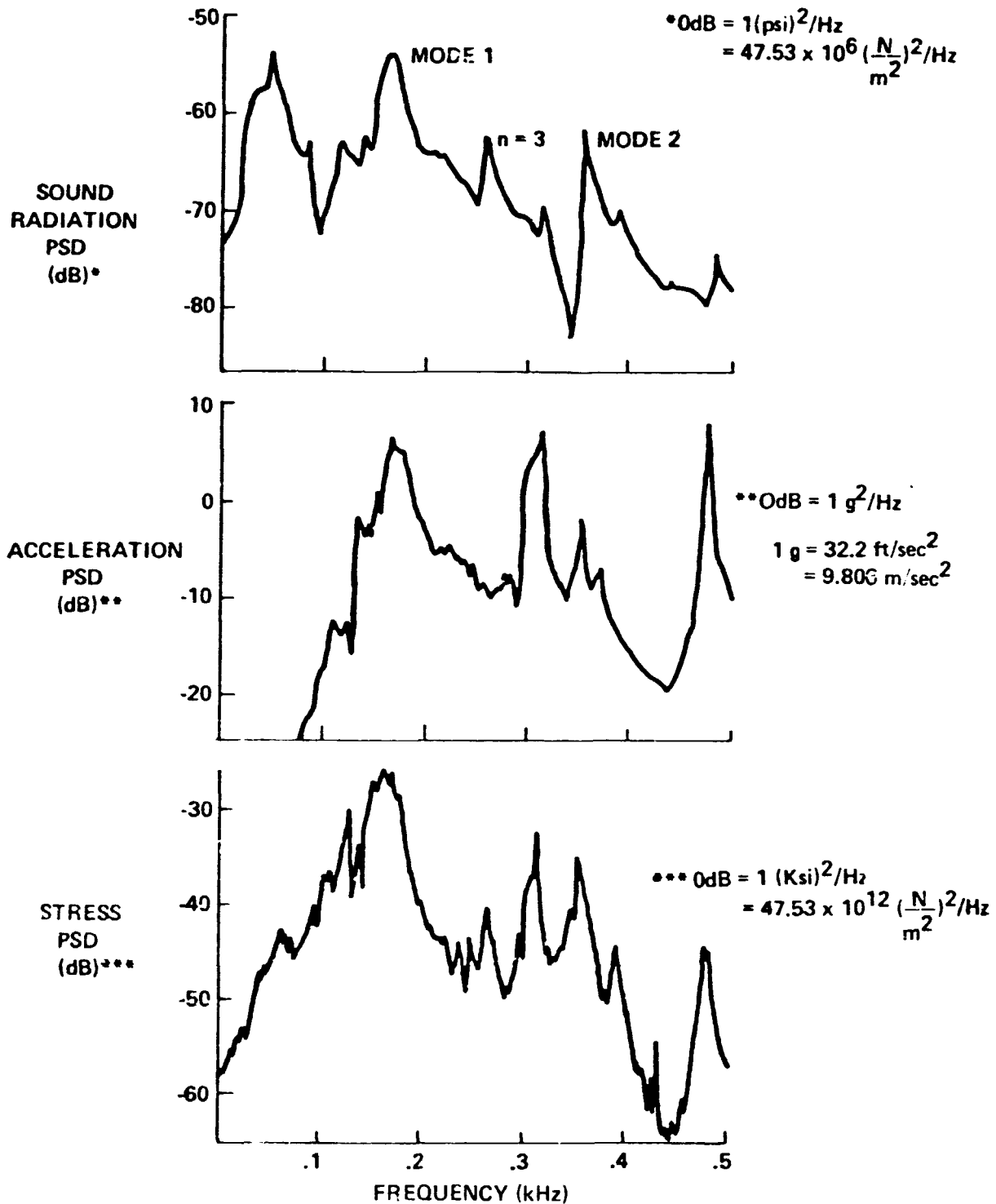


Figure 40.—Variation of Phase Characteristics of the Incident Noise Field
($N_1 = 3864$ rpm)



NOTE: THE 50 Hz PEAK IS DUE TO NOISE TRANSMISSION THROUGH THE WALLS OF THE ANECHOIC BOX AND NOT DUE TO PANEL RESONANCE

Figure 41.—Field Response of the Undamped Tuned Panel

associated with any of the panel response, since the panel stress or acceleration spectra do not show corresponding peaks at 50 Hz. This 50-Hz peak was absent in the laboratory test spectrum, since the excitation was applied on the panels only, with no excitation on the other walls of the anechoic box. Thus in all these cases, the microphone data below 100 Hz were ignored. For this reason the OASPL of sound level inside the anechoic box was also not meaningful, and the panel responses were compared on the basis of the peak responses at frequencies associated with the most important panel modes.

Figure 42 plots the field test spectrum of the noise radiated by the panel with 7.5-in. (0.19-m) stringer spacing, as obtained by the summation of the signals from all the six microphones inside the anechoic box. This is compared with the response of the same panel as measured from the laboratory test (fig. 43). The laboratory test spectrum is based on a combination of the spectra due to five and six point excitation (i.e., from figures 16 and 17). In comparing the two plots, it should be remembered that figure 42 presents the spectrum of the PSD of the field test data plotted on a *logarithmic* scale, and figure 43 presents the spectrum of the transfer function of sound radiation, obtained from the laboratory test, plotted on a *linear* scale. Second, the laboratory test data were based on multipoint excitations, all in phase and fed by current from a white noise generator. In contrast, the excitation spectrum measured during the field test was not flat, and the correlation and coherence characteristics dropped off with increased separation distance and also toward the high frequency part of the spectrum.

In spite of these differences, it can be seen that there is a good correspondence between the two sets of data. Both mode 1 (175 Hz) and mode 2 (360 Hz) were excited during the field test. The mode at 280 Hz observed in the laboratory test can also be identified in the field test data.

Comparison of the response spectra of the other two panels also showed a similar degree of correspondence between the laboratory and the field test data. For the largest panel with 9-in. (0.228-m) stringer spacing, some minor variations were caused by the nonuniformity of the incident noise level over the panel length.

7.4 COMPARISON OF PANEL RESPONSES

7.4.1 ACCELERATION RESPONSE

The PSD of peak low frequency acceleration response of each panel averaged over the three central bays is plotted as a function of stringer spacing in figures 44 and 45. It can be seen that for both power settings, the PSD of acceleration response, plotted on a decibel scale, decreased as the stringer spacing was reduced from 9 in. (0.228 m) to 5 in. (0.13 m). This compares well with the laboratory test data. The dashed line in figures 44 and 45 shows the trend observed from the laboratory test data obtained from figure 19. (This was obtained by superimposing the results of the laboratory and field test results by arbitrarily equating the responses of the damped panel with 9-in. (0.228-m) stringer spacing under laboratory and field conditions. Similar comments also apply to figures 46 through 53.) The response of the panel with 5-in. (0.13-m) stringer spacing was somewhat less than what was expected from the laboratory test data. This was due to two reasons: (1) the field test excitation spectrum rolled off toward higher frequencies, whereas a white noise spectrum was used in the laboratory, and (2) for this panel, the total surface area exposed to the noise field was about 40% less than that of the panel with 9-in. (0.228-m) stringer spacing.

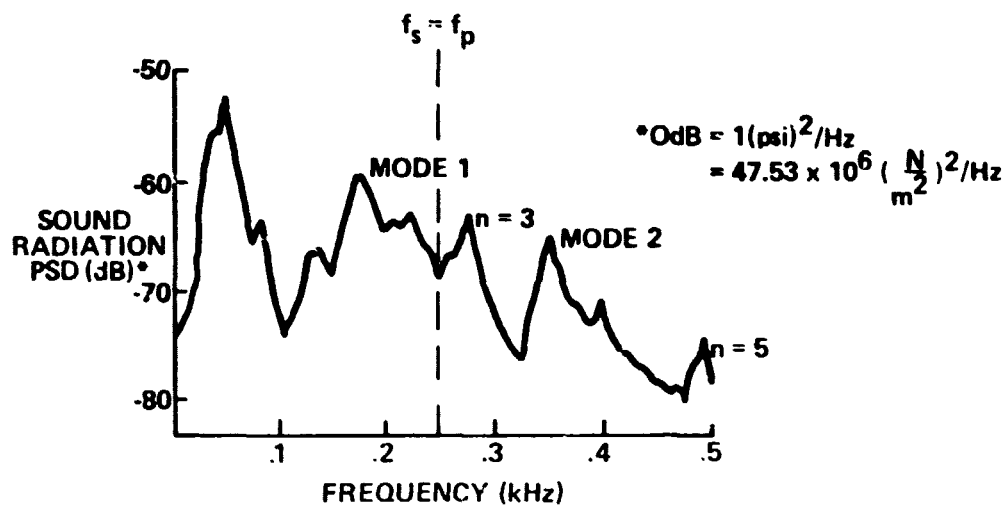
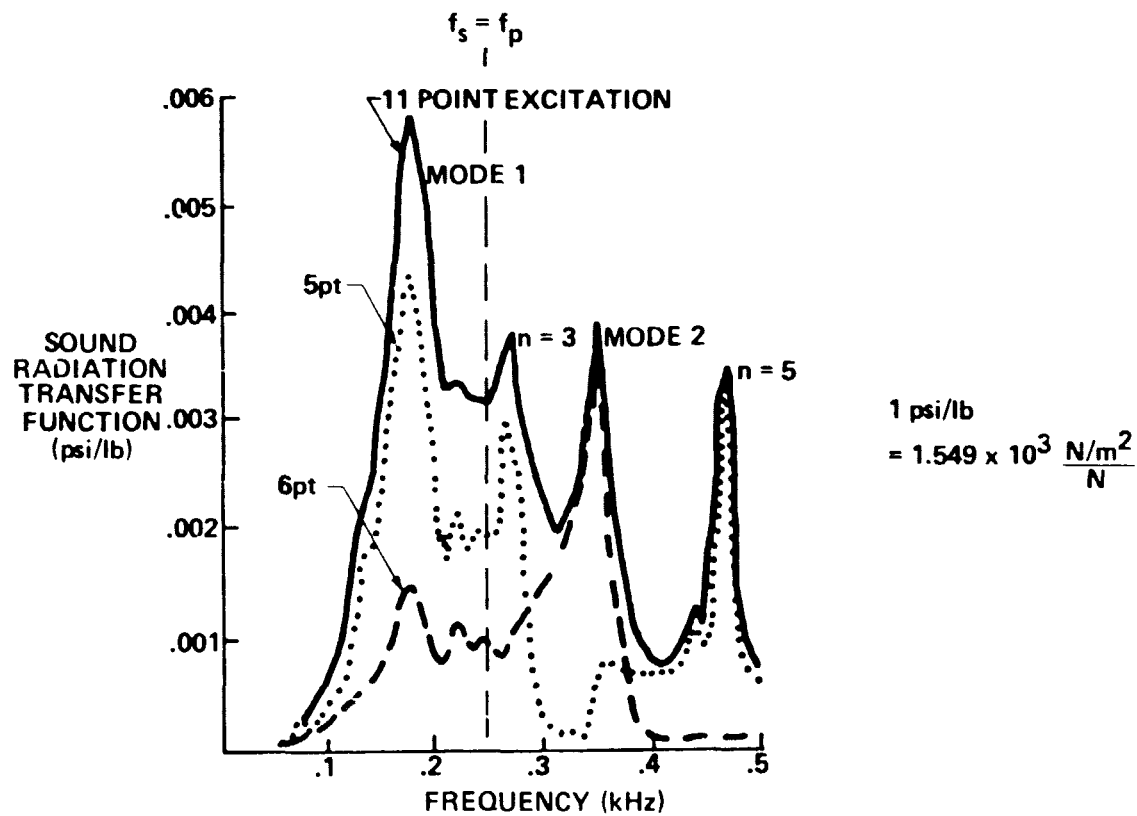


Figure 42. — Sound Radiated by the (Damped) Tuned Panel (Field Test)



(LAB TEST : OBTAINED FROM FIGS. 16 & 17)

Figure 43.—Sound Radiated by the (Damped) Tuned Panel

The change in the response of the tuned panel, between the undamped and damped conditions, was also similar to what was observed from the laboratory test (compare fig. 19 with figs. 44 and 45).

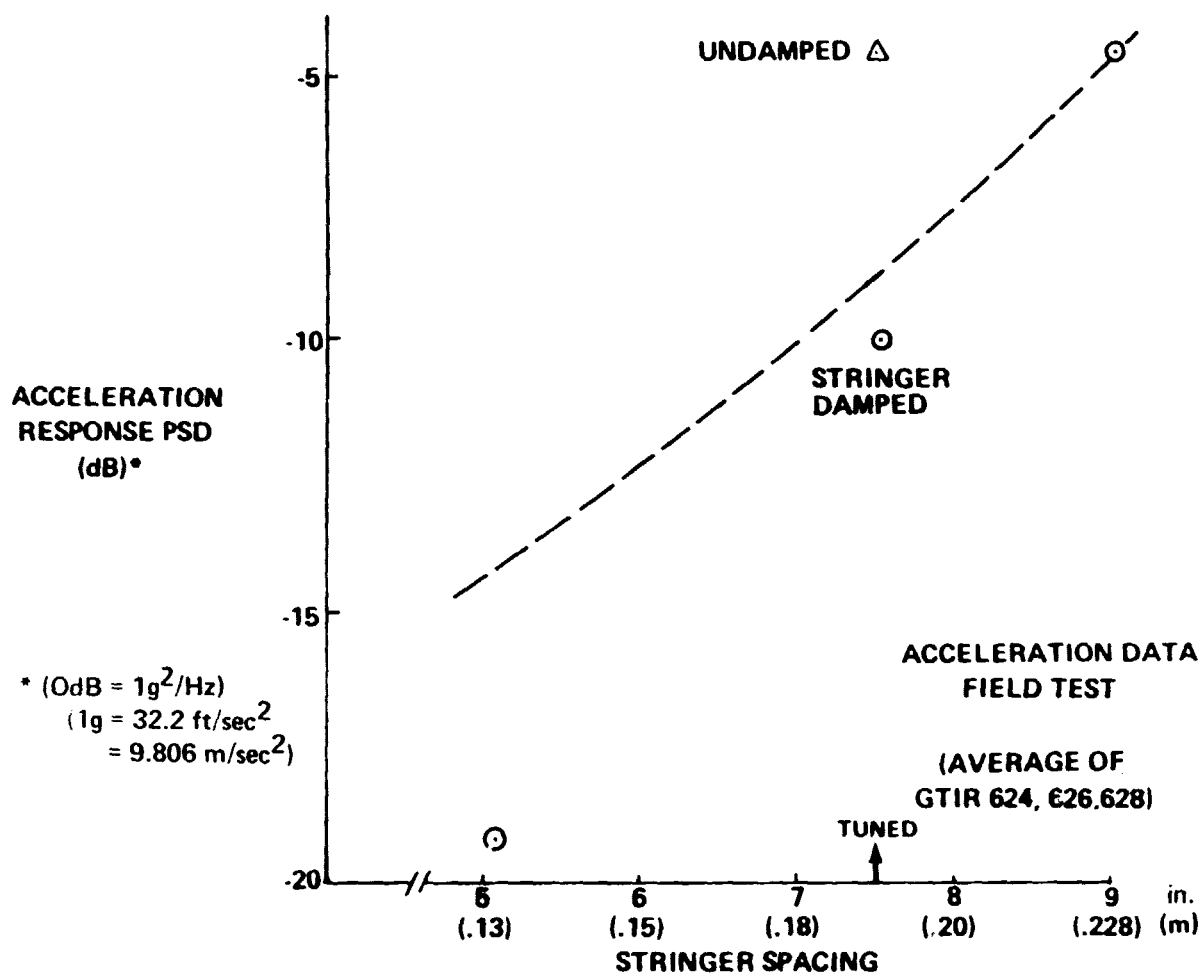
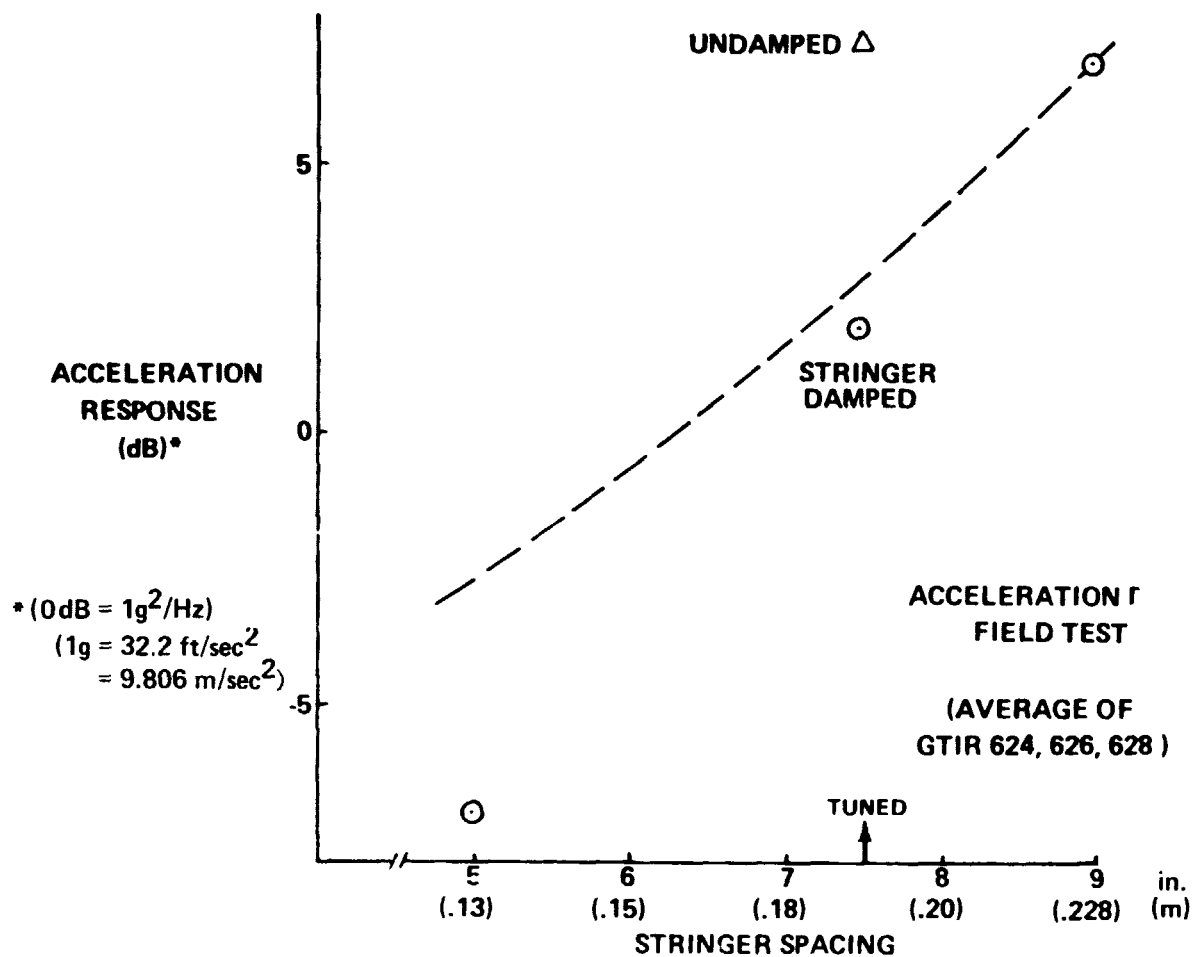


Figure 44.—Variation of Peak Low Frequency Acceleration Response With Stringer Spacing ($N_1 = 2943$ rpm)



NOTE: THE DASHED LINE SHOWS THE TREND OBSERVED FROM THE LABORATORY TEST (FIG. 19)

Figure 45.—Variation of Peak Low Frequency Acceleration Response With Stringer Spacing ($N_1 = 3684 \text{ rpm}$)

7.4.2 SOUND RADIATION

The PSD of peak sound radiation (above 100 Hz) is plotted against stringer spacing in figures 46 and 47. These plots are based on the data obtained by summing up the signals from all the microphones inside the anechoic box. It can be seen that the radiated peak noise level decreased almost linearly with the stringer spacings, and the agreement between the laboratory and the field test is excellent when the spectrum shape of excitation in the field and the difference in panel sizes are taken into account (compare fig. 20 with figs. 46 and 47).

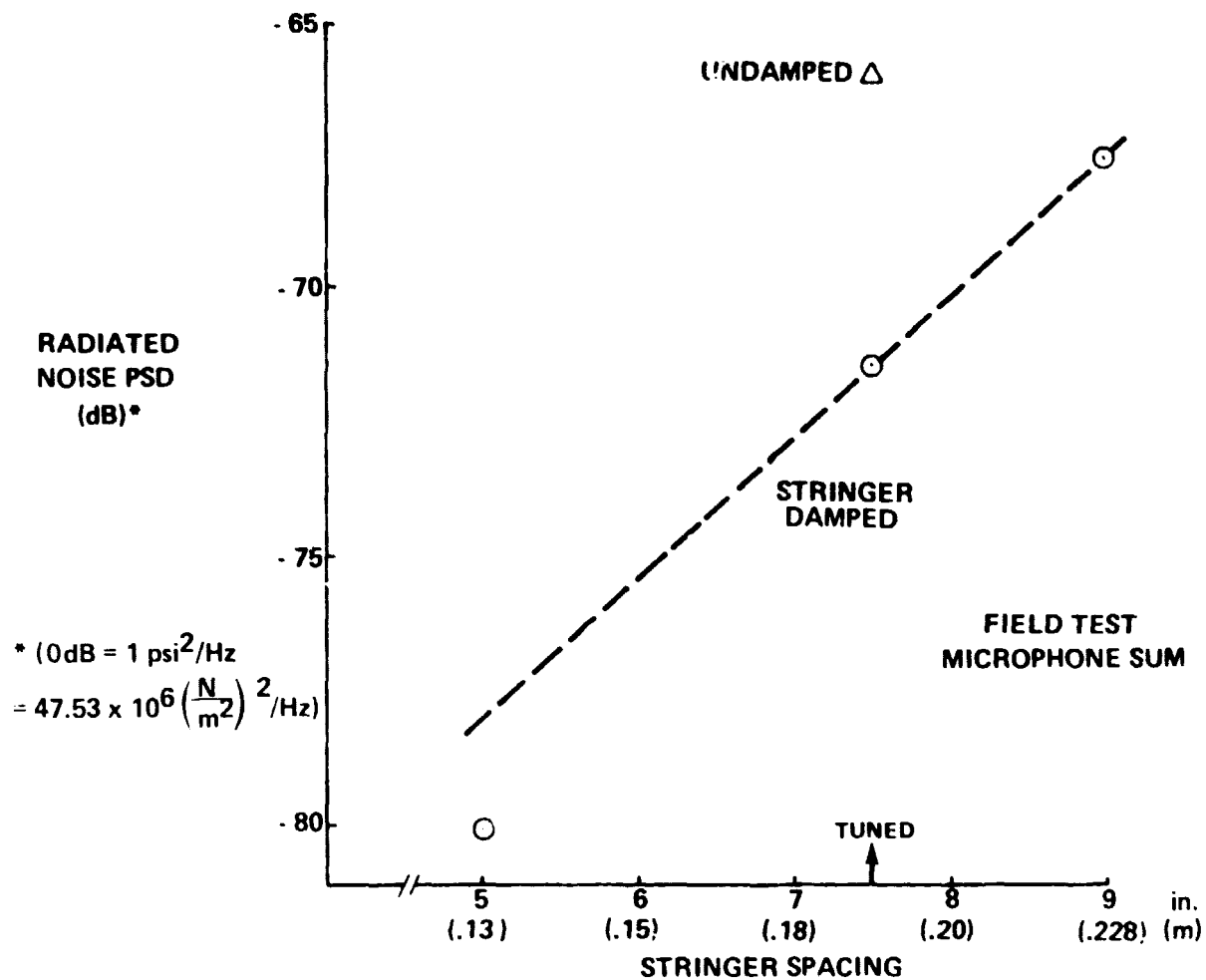
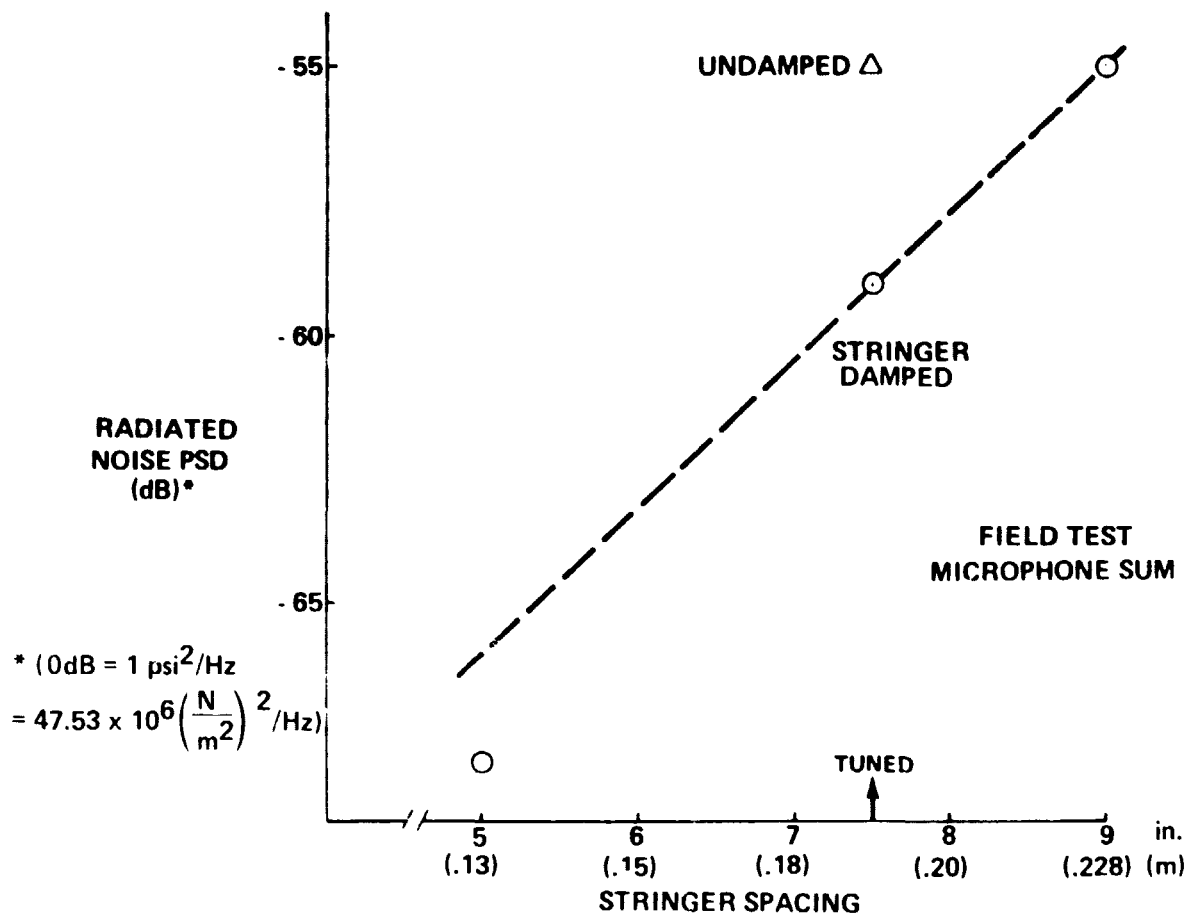


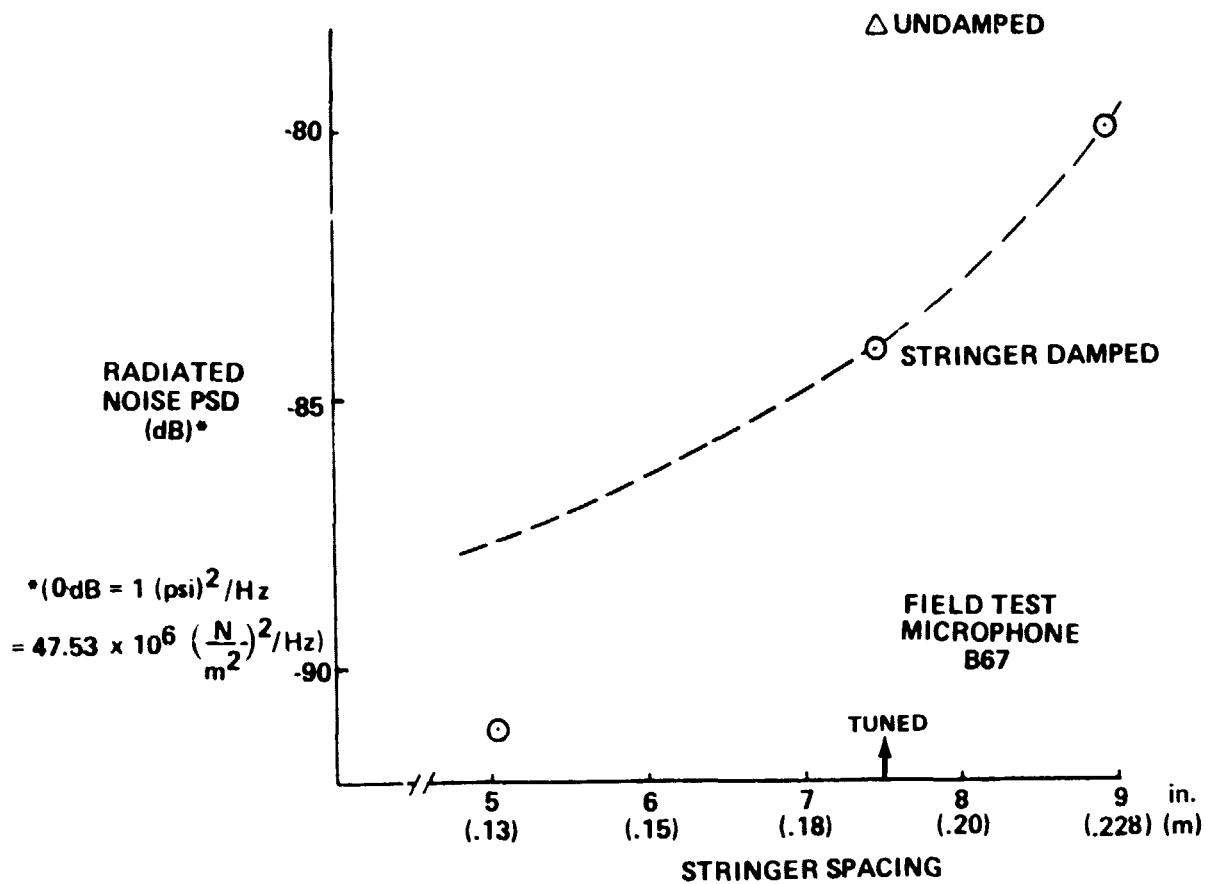
Figure 46.—Variation of Peak Low Frequency Noise Level With Stringer Spacing
($N_1 = 2943$ rpm)



NOTE: THE DASHED LINE SHOWS THE TREND OBSERVED FROM THE LABORATORY TEST (FIG. 20)

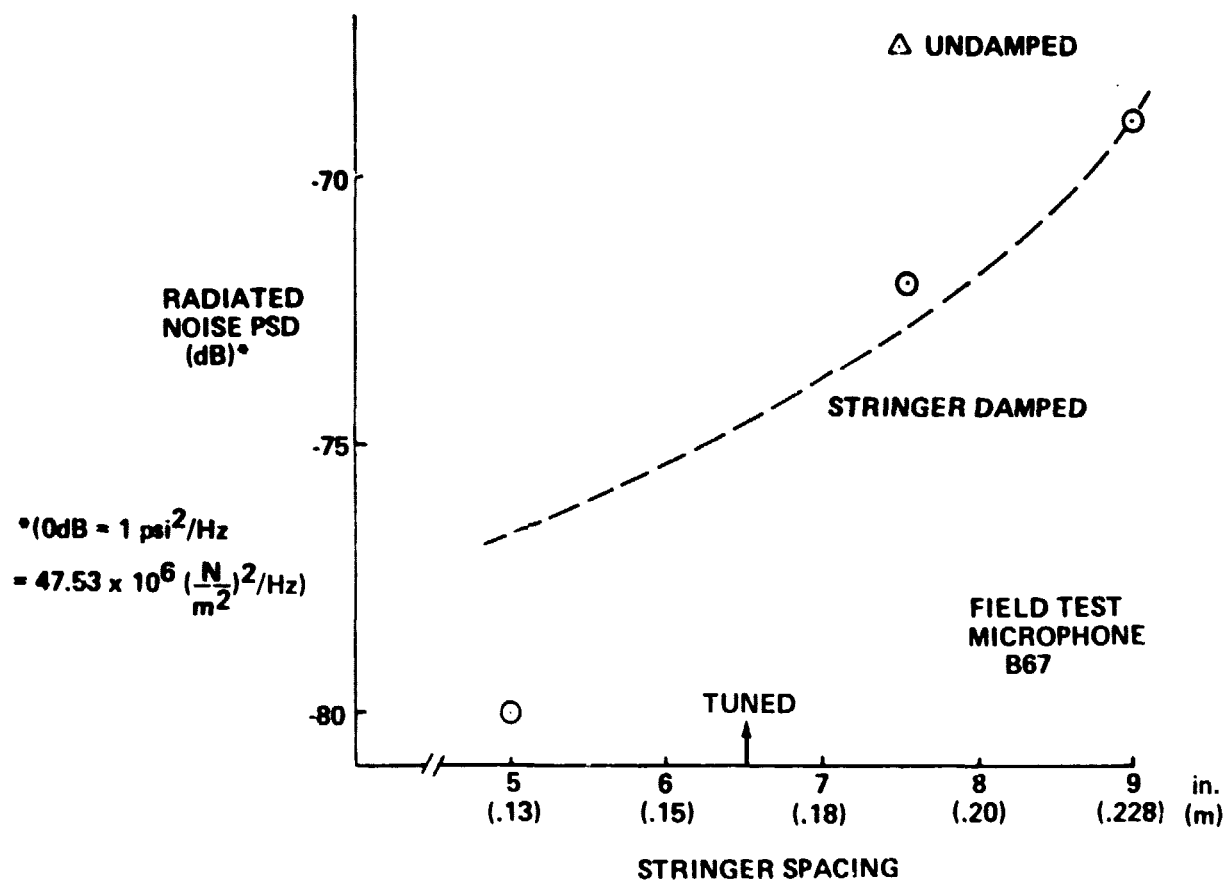
Figure 47.—Variation of Peak Low Frequency Noise Level With Stringer Spacing
($N_1 = 3684$ rpm)

The levels measured by the microphone B67 are also plotted in figures 48 and 49. Again the trends of the laboratory (fig. 22) and field test data are similar. The variation of the sound PSD level measured by E66 with respect to stringer spacing was similar to that shown in figures 48 and 49.



NOTE: THE DASHED LINE SHOWS THE TREND OBSERVED FROM THE LABORATORY TEST (FIG. 22)

Figure 48.—Variation of Peak Low Frequency Noise Level With Stringer Spacing
($N_1 = 2943$ rpm)

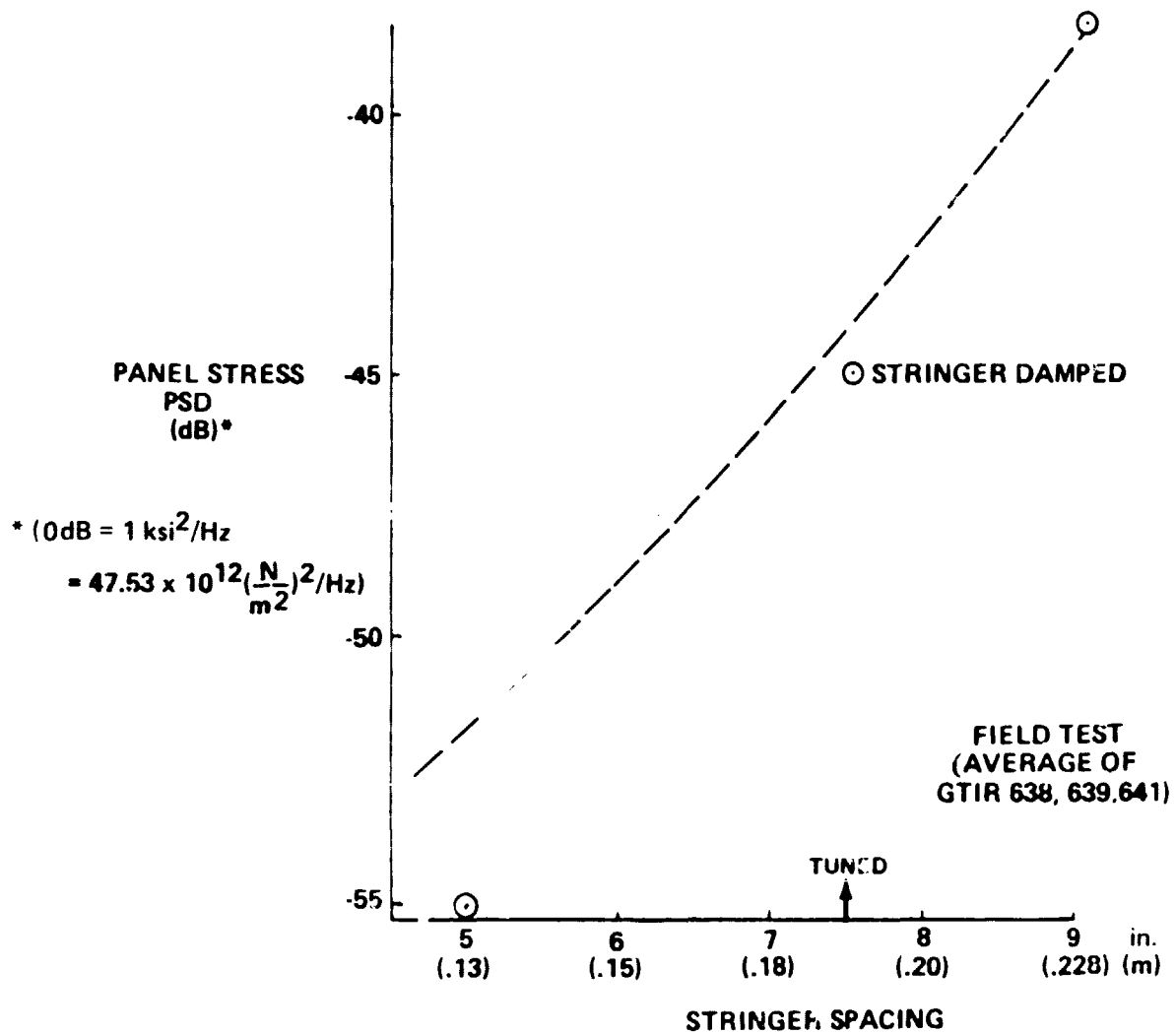


NOTE: THE DASHED LINE SHOWS THE TREND OBSERVED FROM THE LABORATORY TEST (FIG. 22)

Figure 49.—Variation of Peak Low Frequency Noise Level With Stringer Spacing
($N_1 = 3684$ rpm)

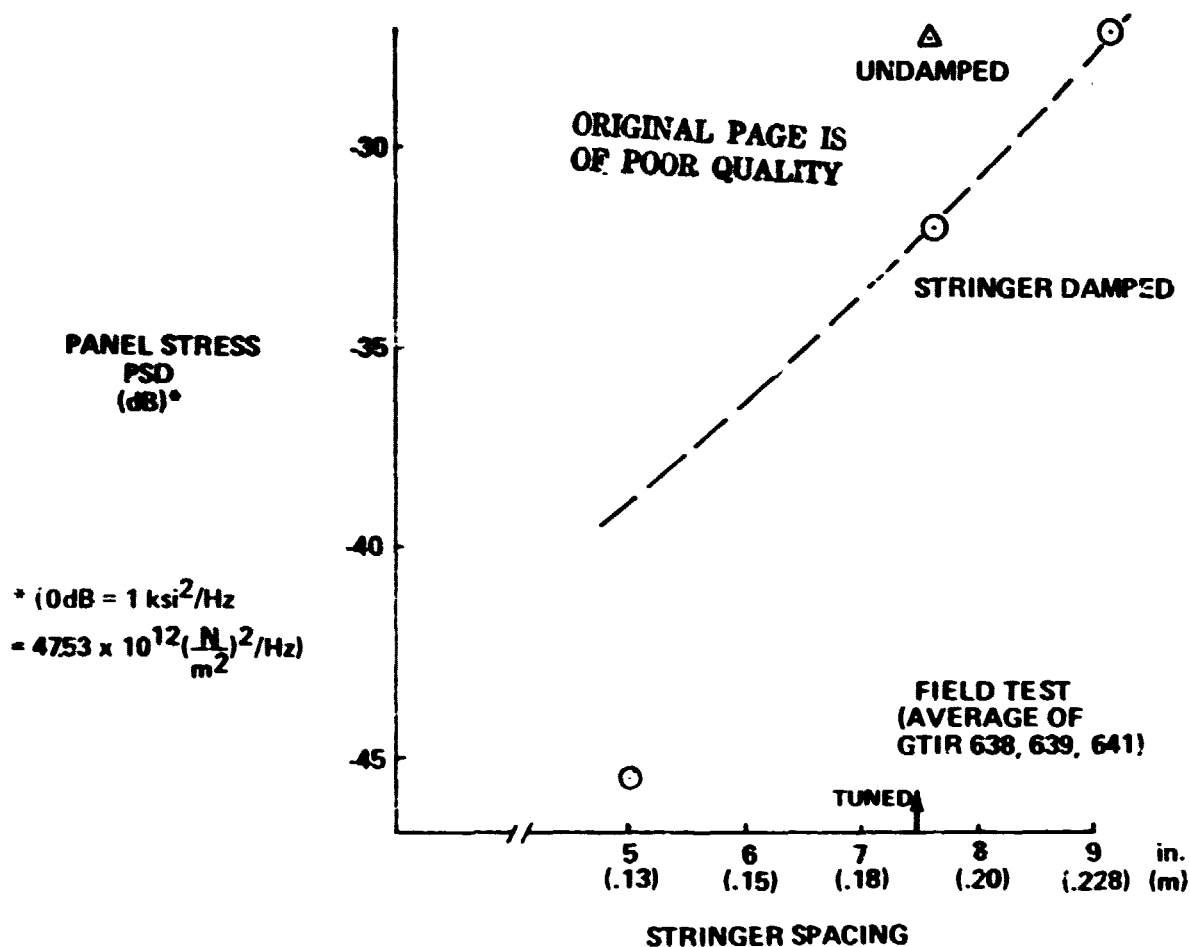
7.4.3 STRESS RESPONSE

The PSD of peak skin panel bending stress response, averaged over the three strain gages in the central bays, is plotted against stringer spacing in figures 50 and 51. Again it can be seen that the peak panel bending stress response can be significantly reduced by reducing the stringer spacing so that the skin frequency is higher than the stringer frequency and then applying damping treatment on the stringers. The trend observed in the field test is similar to that observed in the laboratory, when the field spectrum shape is taken into consideration. The change in the skin bending stress response of the tuned panel, under damped and undamped conditions, is also similar to that observed in the laboratory test (compare with fig. 23).



NOTE: DATA FOR THE UNDAMPED CONDITIONS, INADVERTENTLY NOT RECORDED
THE DASHED LINE SHOWS THE TREND OBSERVED FROM THE LABORATORY TEST (FIG. 23)

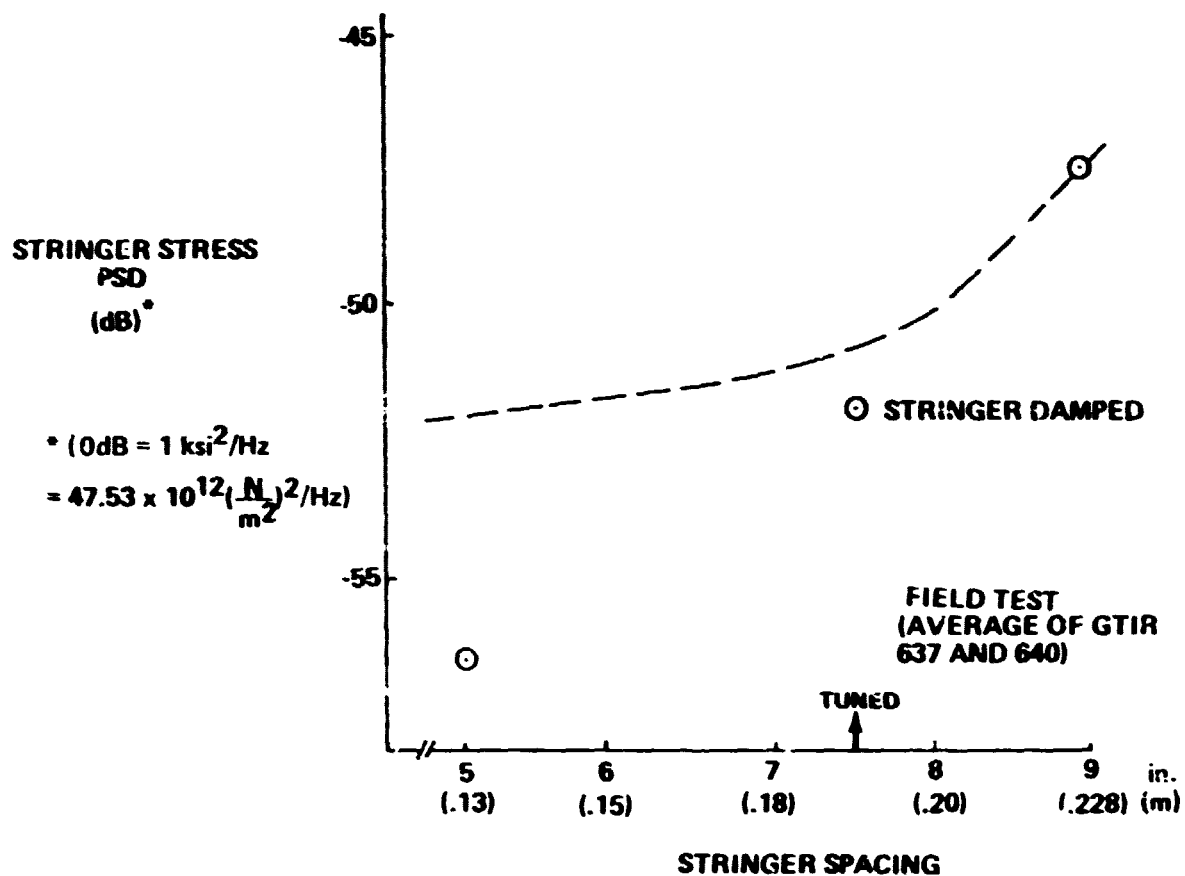
Figure 50.—Variation of Peak Low Frequency Panel Stress Response With Stringer Spacing
($N_1 = 2943$ rpm)



NOTE: THE DASHED LINE SHOWS THE TREND OBSERVED FROM THE LABORATORY TEST (FIG. 23)

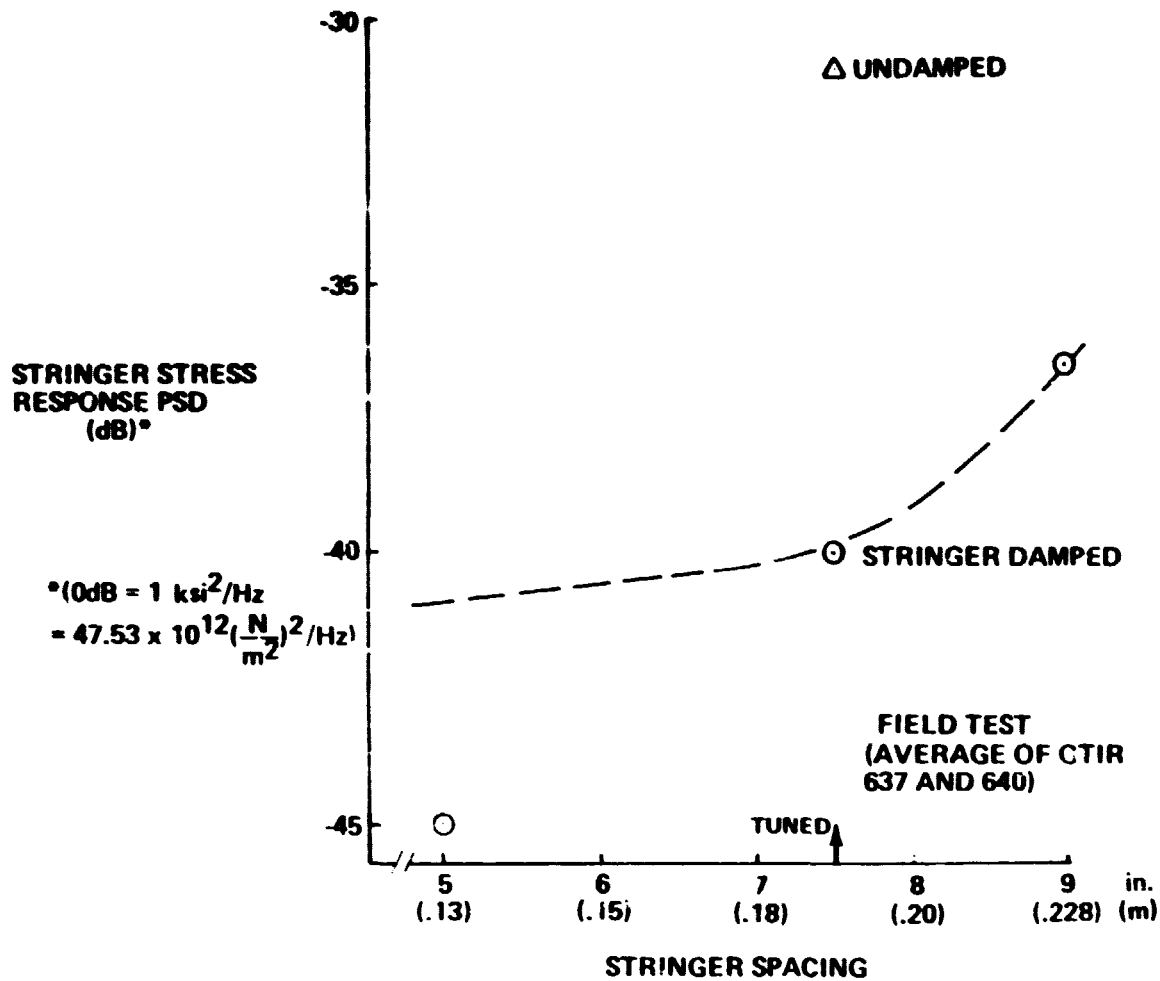
Figure 51.—Variation of Peak Low Frequency Panel Stress Response With Stringer Spacing
($N_1 = 3684$ rpm)

The PSD of peak stringer bending stress response, averaged over the two strain gages on the two central stringers, is plotted against stringer spacing in figures 52 and 53. As the stringer spacing was reduced, the stringer bending stress response was also reduced. However, the rate of change of the stringer stress is much slower than that of the panel stress. For the panel with 5-in. (0.13-m) spacing, the skin and stringer bending stresses are comparable, with damping treatment applied on the stringers. For this panel, any damping treatment applied on the skin would have been much less effective in reducing either the skin or the stringer bending stress response.



NOTE: (1) DATA FOR THE UNDAMPED CONDITIONS, INADVERTENTLY NOT RECORDED
(2) THE DASHED LINE SHOWS THE TREND OBSERVED FROM THE LABORATORY TEST (FIG. 24)

Figure 52.—Variation of Peak Low Frequency Stringer Stress Response With Stringer Spacing
($N_1 = 2943$ rpm)



NOTE: THE DASHED LINE SHOWS THE TRENT OBSERVED FROM THE LABORATORY TEST (FIG. 24)

Figure 53.—Variation of Peak Low Frequency Stringer Stress Response With Stringer Spacing
($N_1 = 3684$ rpm)

The change in the stringer stress response between undamped and damped condition was higher than that observed in the laboratory test (compare fig. 53 with fig. 24). In the field, the panel was initially tested in the damped condition. Then the damping treatment was removed so that the panel could be tested in the undamped condition. It was not easy to remove the damping treatment because of the highly adhesive property of the viscoelastic material, and the 0.08-in. (0.002-m) thick metal constraining strip had to be pried open with considerable difficulty. It is possible that the strain gages on the underside of the stringer flanges were damaged during this process and hence a malfunction is suspected.

7.5 COMPARISON OF THE FIELD TRANSMISSION LOSSES OF THE PANELS

In the field test, the panels were excited by the near field engine noise. This incident noise field is neither reverberant nor a true plane wave. However, since the phase variation over the panel surface was small (fig. 40), the incident noise field may be approximated by a plane progressive wave incident normal to the panel surface.

The noise radiated by the panels was measured inside an anechoic box at points close to the panel surface. Since a plane wave is assumed to be incident on the panel, a plane wave may be assumed to radiate away from the panel and into the highly absorptive receiving space. Although this is questionable at high frequencies, it should be a reasonable assumption at low to mid frequencies.

Under the above condition and within the specified limitations, the ASTM E336 procedure may be used to calculate the field transmission loss (FTL) of the panels. This is defined as

$$FTL = L_{10} - L_2 - 6 \quad (24)$$

where:

L_{10} = Average sound pressure level (dB) at the face of the panel measured by flush-mounted microphones

L_2 = Average sound pressure level (dB) in the receiving side measured at points close to the panel surface

In equation (24), a factor of 6 dB is subtracted to account for the reflection of the incident noise field by the panel surface.

In calculating L_{10} , an average sound spectrum was obtained from the levels measured by the four corner microphones used for each panel. Similarly, L_2 was obtained by averaging the sound levels measured by the microphones in the anechoic box.

The ASTM E336 procedure specifies that although the microphones in the anechoic box should be close to the panel surfaces, they should not be closer than a quarter wavelength from the panel surface. In this test, the microphones were placed about 4 in. (0.101 m) away from the panel surface. It is obvious that the specification was not met, especially at low frequencies. However, a distance of 4 in. (0.101 m) was used in order to readily identify the peaks and the valleys of the radiated noise spectra with those obtained from the acceleration spectra, measured by an accelerometer near the same location. As the microphone distance is increased, noise components coming from different parts of the structure contribute to the noise signal with different phase differences and time delays, which can make such comparison of structural and acoustic data more difficult. The main purpose of the test was to study the changes in the panel transmission loss caused by structural tuning and damping. The procedure described above should be adequate for that purpose.

Figure 54 shows the third octave band field transmission loss of the tuned panel in both undamped and damped conditions. The analysis was carried out in one-third octave bands rather than in full octave bands, in order to identify the valleys associated with the important structural mode groups. For example, the two valleys in the FTL of the undamped tuned panel in 160- and 400-Hz bands are associated with the modes 1 and 2. This detail would be lost in a full octave band analysis. The analysis is limited to frequencies above 100 Hz, since flanking transmission through the anechoic box walls masks the data below this frequency. In addition, a flat panel analysis of unpressurized fuselage structure is not valid below 100 Hz, since the overall modes involving large scale deformations of the fuselage cylinder tend to become important at frequencies below 100 Hz.

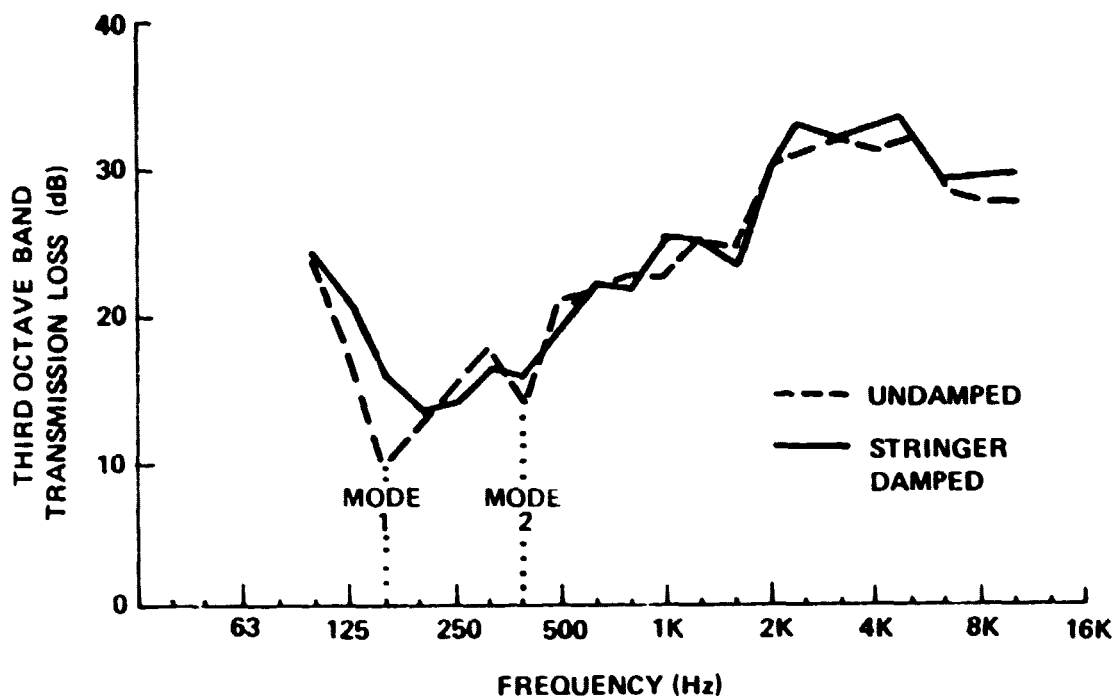


Figure 54.—Field Transmission Loss of the Tuned Panel

It may be noted from figure 54 that the undamped panel has a high transmission loss around 250- and 315-Hz bands. This is to be expected, since vibration energy is transferred from the skin panel to the stringers in these frequency bands because of intrinsic structural tuning. On the other hand, if the stringer spacing was kept constant and the actual stringers were replaced by others that are extremely stiff in bending or extremely massive, a deep valley would have been created in the panel transmission loss around 250-Hz band. Structural tuning has, therefore, effectively increased the panel transmission loss in this frequency band and 315-Hz band. Such a method of increasing the panel transmission loss in certain frequency bands should be particularly useful in situations where the excitation is due to discrete tones; e.g., in turboprop or prop-fan aircraft.

For the undamped tuned panel, the transmission loss passes through a minimum around 160- and 400-Hz. This requires the use of some additional damping treatment. Figure 54 shows a 6-dB improvement of FTL in the 160 Hz band due to stringer damping. The 6-dB improvement around 160 Hz is significant, particularly since the tuned panel had a high loss factor even under undamped conditions. In addition, there was a 4-dB improvement around 125 Hz and a 2.5-dB improvement around 400 Hz. Further reductions should be possible in all these bands by applying damping tapes on the skin.

The damped panel has a slightly lower transmission loss around the 250- and 315-Hz bands. This is to be expected since the damping treatment on the stringers smooths out the peaks and the valleys of the panel response spectrum. Above 800 Hz, the damped panel had about 2-dB higher transmission loss at most frequencies. The only exception is around 1.6 kHz where the FTL of the undamped panel was about 1.5-dB higher. This could be genuine or it could be caused by the scatter of the experimental data. In general the damped panel had about 2-dB higher transmission loss above 800 Hz. This is consistent with the mass law effect, since the stringer damping treatment added about 29% extra weight to the structure. (Lighter stringer damping treatments that are almost equally effective have since been developed.)

The transmission losses of the various panels with damped stringers are compared in figure 55. The data for the panels with 9-in. (0.228-m) and 7.5-in. (0.19-m) stringer spacing are shown for frequencies above 100 Hz. The data for the panel with 5-in. (0.13-m) stringer spacing are shown above 160 Hz. The problem of flanking transmission through the walls of the anechoic box prevented the extrapolation of the test data below these frequencies. In addition, extrapolation of data obtained from testing unpressurized flat panels to frequencies below 100 Hz would have given deceptively encouraging estimates of the fuselage sidewall transmission loss, since overall cylindrical modes are involved at these frequencies.

The transmission losses of the various panels are compared in figure 55. The valleys in the transmission loss curves of each panel can be identified with the resonant modes established earlier from the narrow band PSD data. Several interesting conclusions can be drawn from this plot. With damped stringers, the panel with 5-in. (0.13-m) stringer spacing had the highest transmission loss at low frequencies. This panel had a higher transmission loss even in the 250-Hz band because of the application of damping treatment on the stringers. The results of the computer program based on the theory presented in section 4.0 as well as the modal survey conducted during the laboratory test indicated that, for this panel, the structural modes in the 250-Hz band are such that the skin acts like a relatively stiff member

supported on relatively flexible stringers. For this reason, the stringer damping was highly effective in increasing the transmission loss of this panel in this frequency band. On the other hand, application of any damping tape on the skin would have been much less effective in this frequency band. As predicted by the analysis, the transmission loss of this panel in this frequency band is controlled primarily by stringer resonance.

Therefore, the key to increasing the transmission loss at low frequencies is to design the structure so that the skin bay frequency is much higher than the stringer frequency, and then applying damping treatment on the stringers. This can be achieved by various means besides using a narrower stringer spacing, as was done in the present study. For example, the condition is automatically satisfied in a pressurized fuselage. A typical cabin pressure differential is about 8 or 9 psi (55.16×10^3 or 62.05×10^3 N/m²). This increases the fundamental frequency of the individual skin bay to a value that is much higher than that of the stringer. In the 250- to 450-Hz range, the skin then acts like a relatively stiff structural member supported on relatively flexible stringers. In a pressurized fuselage, application of damping treatment on the stringers should, therefore, be very effective in reducing cabin noise in the above frequency band. Traditionally, noise transmission in this frequency band has been regarded as stiffness-controlled, since this is below the fundamental frequency (typically 500 to 650 Hz) of the individual skin bay carrying inplane loads caused by cabin pressurization, and since application of damping tapes on the skin bays is not very effective in this frequency band.

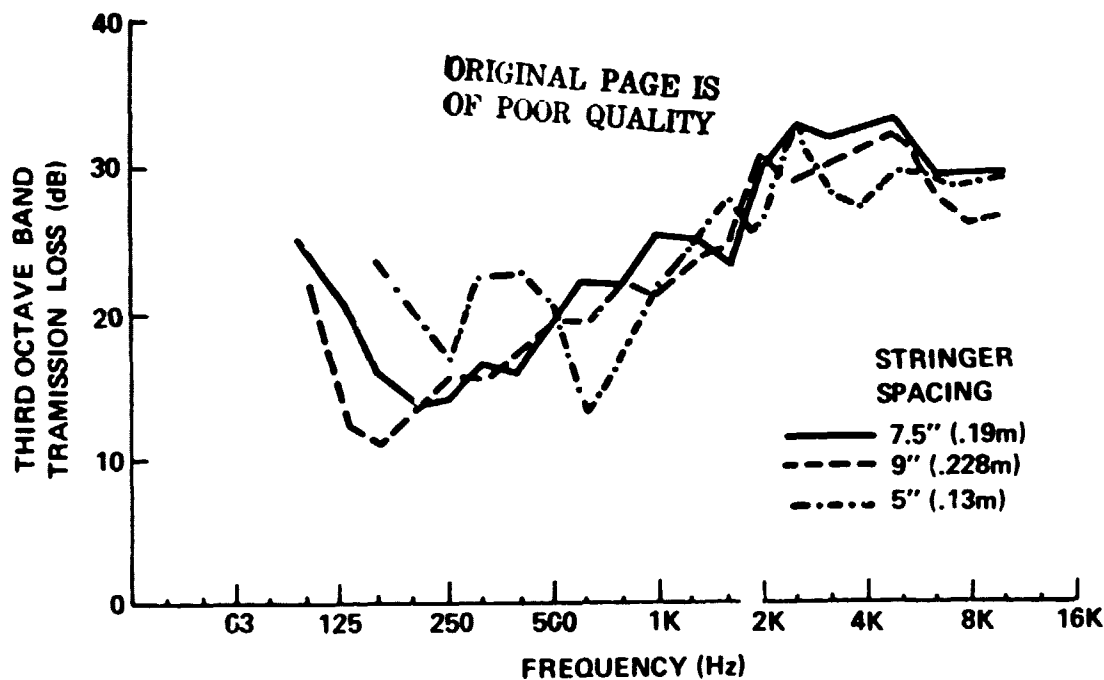


Figure 55.—Comparison of Field Transmission Losses of the Various Panels With Damping Treatment Applied on the Stringers

From figure 55, it may also be observed that the panel with 5-in. (0.13-m) stringer spacing had a high transmission loss in 315- and 400-Hz bands. In these bands, the stringer response is higher than that at the panel center, as it may be seen from figure 56, where the acceleration response of the panel center is compared to that at the center of an adjacent stringer. Notice that the existence of such a band of high transmission loss is most clearly visible for the panel with 5-in. (0.13-m) stringer spacing; i.e., the panel for which the fundamental frequency of the individual skin bay is higher than that of the stringer. Therefore, the transmission loss curve of a pressurized fuselage structure should also exhibit such characteristics. The existence of such frequency bands of high transmission loss may be effectively utilized in situations where the excitation is due to discrete tones; e.g., in prop-fan aircraft. One would then attempt to design the structures so that the excitation tone of greatest concern falls within such a frequency band.

Figure 55 shows that this panel had a poor transmission loss in the 500 Hz – 1 kHz band, even with the damping treatment applied on the stringers. In this frequency band, the structural modes have half wavelengths comparable to or less than the stringer spacing, and the skin bays tend to vibrate as individual rectangular panels. This can be inferred from figure 56. Therefore, in this frequency band, skin damping would be highly effective in increasing the transmission loss. This was verified by the results of the computer program. In addition, the fiberglass insulation becomes effective in this frequency band. The excitation spectrum also tends to roll off at these frequencies.

A similar situation exists in the 500 Hz – 1 kHz band in a pressurized fuselage structure and, for this reason, the application of damping tapes on the skin is quite effective in this frequency band. As a result, in this frequency band, noise transmission in a pressurized fuselage has been correctly regarded as damping controlled.

Another interesting, although somewhat academic point may be observed from figure 55. It may be seen that compared to the panel with 5-in. (0.13-m) stringer spacing, the tuned panel has a higher transmission loss at frequencies above 2 kHz. As discussed earlier, the tuned panel transmission loss at these frequencies is essentially controlled by the mass law, since most of the low order modes occur at much lower frequencies. On the other hand, the panel with 5-in. (0.13-m) stringer spacing shows distinct resonances around 4 kHz which should account for the valley in this frequency band. This is only of academic interest, since application of skin damping tapes would effectively fill up this valley. In addition, the fiberglass insulation would significantly raise the transmission loss at these frequencies.

7.6 COMPARISON WITH PREDICTION

As mentioned in the previous sections, a computer program was developed on the basis of the equations derived in section 4.0, to predict the response of infinitely long, periodic skin-stringer structures excited by a convected random pressure field. The responses of the finite skin-stringer panels used in this study were calculated by using this computer program. A highly correlated white noise excitation was used to simulate the distant near field noise environment generated by the USB YC-14 propulsion system.

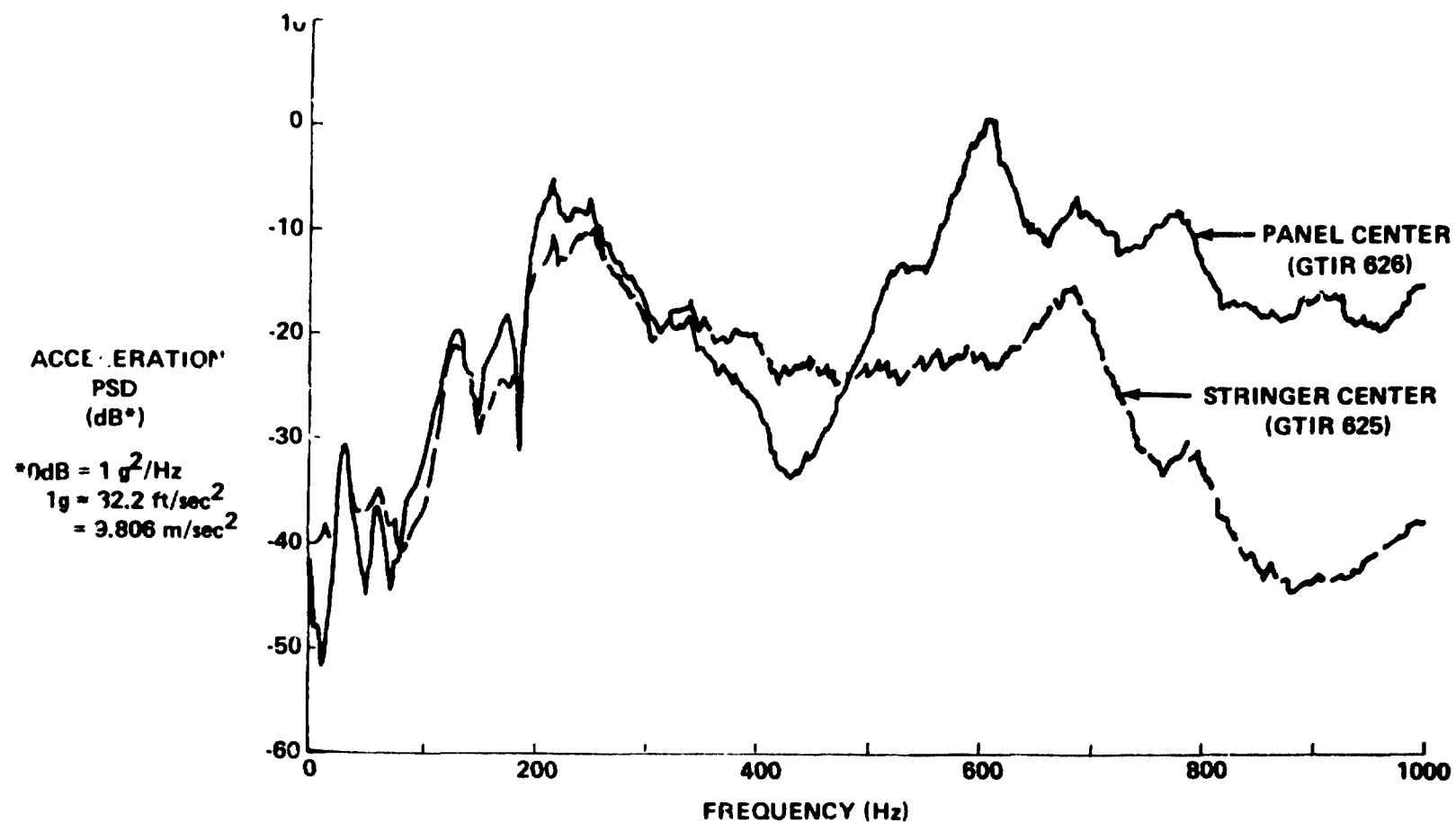


Figure 56.—Comparison of Panel and Stringer Responses for the Stringer Damped Panel with 5-in. (0.13-m) Stringer Spacing

To calculate the response of the undamped panels, the panel loss factors measured in the laboratory test (fig. 18) were used as inputs to the computer program. However, the response of the stringer-damped panels could not be accurately predicted by using the measured loss factors of the stringer-damped panels. The measured data represents an apparent loss factor of each panel, whereas the damping treatment used was of localized nature. Therefore, the responses of the stringer-damped panels were calculated by setting the skin loss factors equal to the measured loss factors of the undamped panel (fig. 18) and setting the stringer loss factor equal to the measured loss factor of the stringer with the constrained layer damping treatment (fig. 7).

In section 5.2, we compared the predicted spectra for the various panels with the corresponding laboratory test data and found some reasonable agreement. In figure 57, the predicted mean square acceleration response over a wide frequency band (50 to 1000 Hz) containing

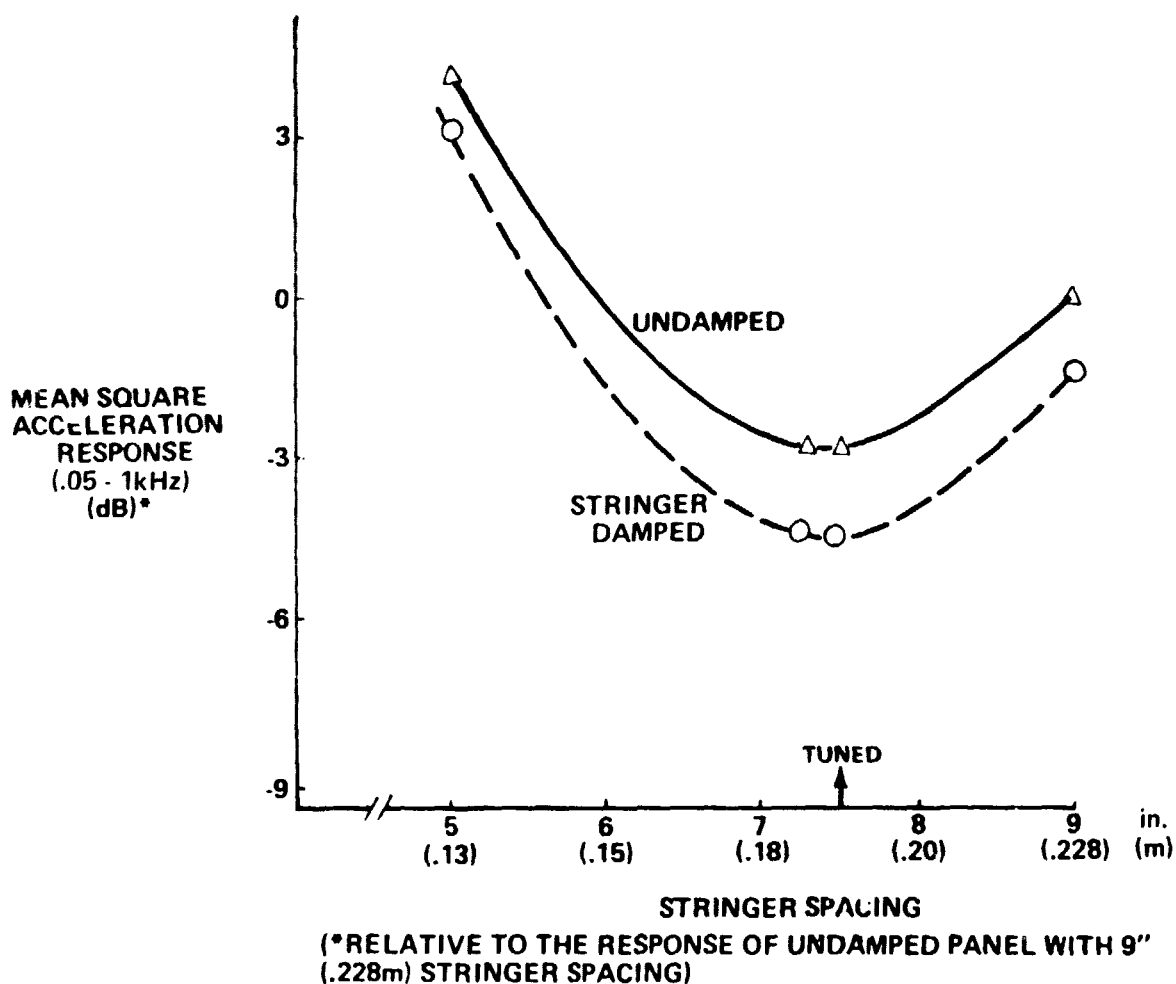


Figure 57.—Predicted Variation of the Mean Square Acceleration Response to Highly Correlated White Noise With Stringer Spacing and Damping

many modes is plotted as a function of stringer spacing. It is seen that the response of the panel goes through a minimum, for undamped as well as damped conditions, when the tuning condition is satisfied. This plot is based on the assumption that the excitation spectrum was flat with a high degree of correlation and coherence over the entire frequency range, and the damping loss factor of each panel was constant over the entire frequency range. However, the field test excitation and the panel damping characteristics were different from above, and for this reason, these results cannot be directly used for comparison.

Therefore, it was decided to compare the measured and predicted results on the basis of the peak low frequency response levels, since this is really more important from the designer's standpoint.

The results are shown in figure 58 in which the variation of the peak low frequency acceleration response is plotted as a function of stringer spacing. With no external damping, the

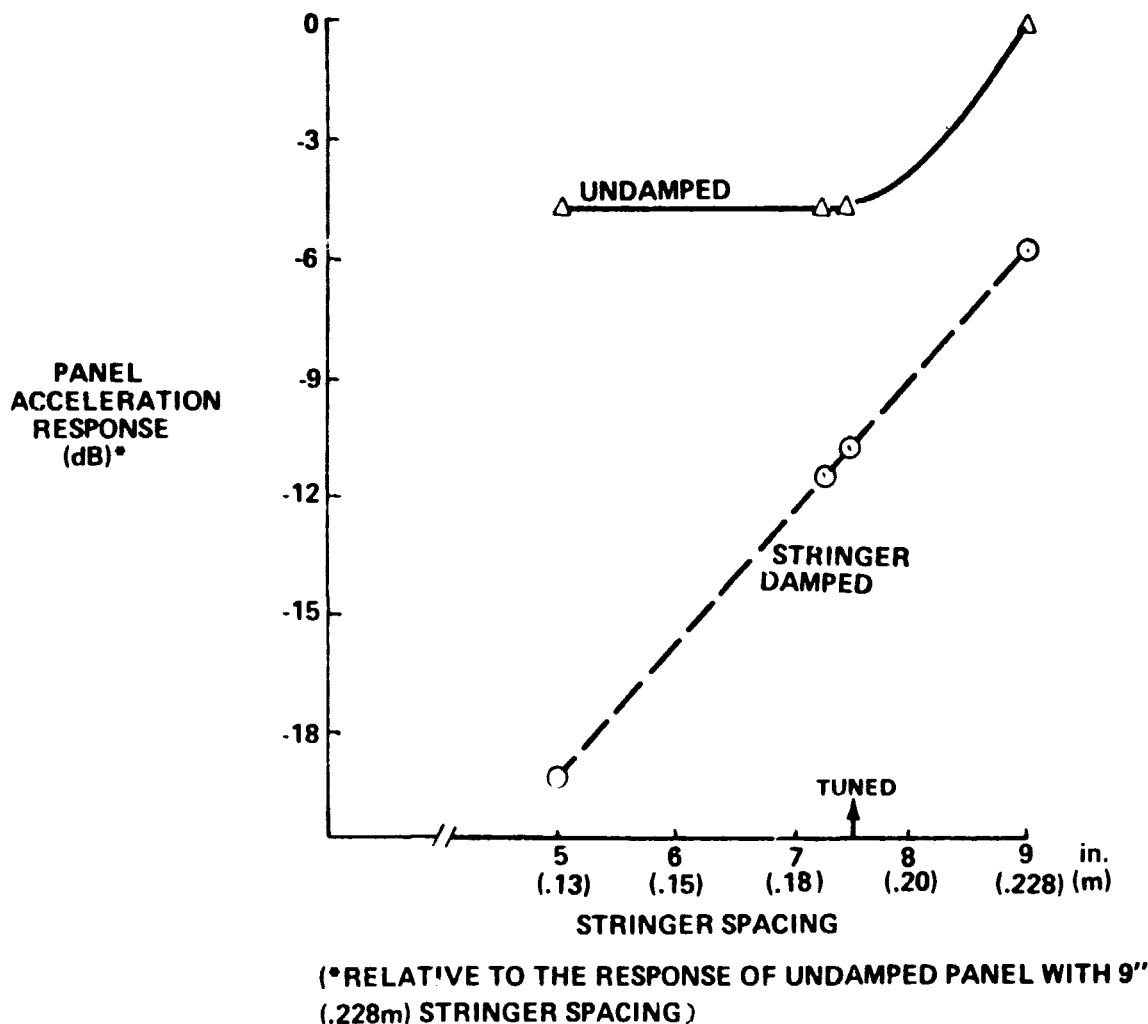


Figure 58.—Predicted Variation of Peak Low Frequency Acceleration Response With Stringer Spacing and Damping

acceleration response passed through a point of diminishing return when the tuning condition was satisfied. With damping treatment applied on the stringers, the panel response, plotted on a decibel scale, decreased linearly with stringer spacing. The maximum reduction was observed for the panel with the narrowest stringer spacings; i.e., the panel for which the uncoupled frequency of the skin bay was greater than that of the stringers.

The variation of the peak low frequency velocity response is plotted as a function of stringer spacing and damping in figure 59. Similar conclusions can also be drawn from this plot.

So far, we have discussed the variation of the predicted and measured response levels of the various panels with respect to those of the undamped panel with 9-in. (0.228-m) stringer spacing. It will now be instructive to compare the absolute levels obtained from the analytical models and the field test.

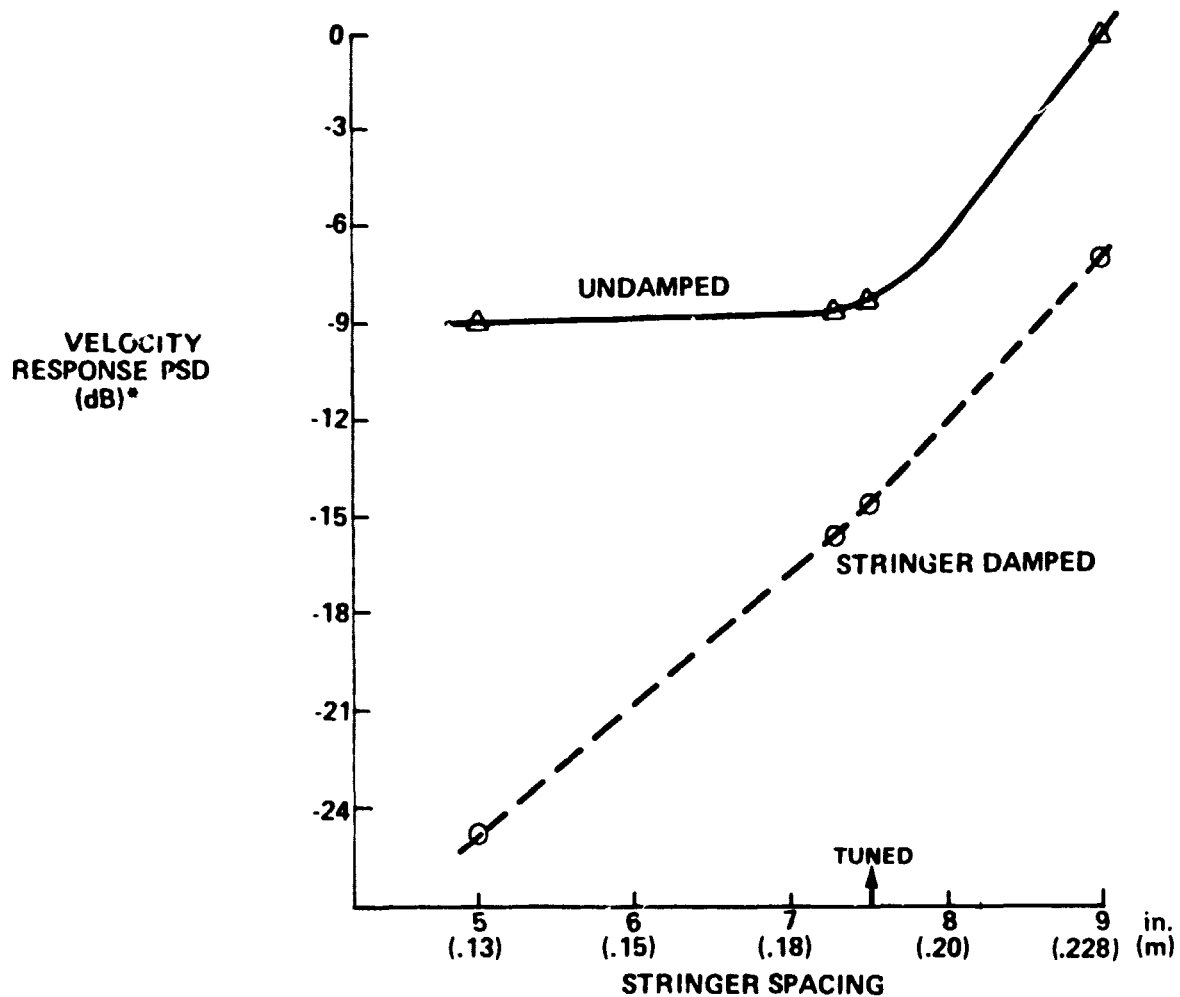


Figure 59.—Predicted Variation of Peak Low Frequency Velocity Response With Stringer Spacing and Damping

The computer program based on the equations presented in section 4.0 can be used to predict the velocity response PSD at each bay center of an infinite panel to unit excitation PSD level. The predicted peak low frequency velocity PSD level was, therefore, obtained by multiplying the above by the average of the excitation PSD levels measured by the four flush-mounted corner microphones. The rms velocity response over a 1-Hz bandwidth centered around the frequency of peak response was then obtained by taking the square root of the above quantity.

The measured rms velocity response was obtained in the following manner. From the field test acceleration data, the peak low frequency acceleration PSD level averaged over panel bay centers (i.e., average of data from the accelerometers GRIR 622, 624, 626, 628, 630 in fig. 33) was calculated. The velocity PSD level was obtained next, by dividing the above by ω^2 where $\omega = 2\pi f$ and f is the frequency of peak low frequency response. The rms velocity response over a 1-Hz bandwidth was then calculated by taking the square root of the velocity PSD level.

The predicted and the measured rms velocity responses are compared in figure 60. If there were a perfect agreement between the predicted and the measured values, the points A, B, C, D in figure 60 would have fallen exactly on the solid line. The dashed line represents a variation by a factor of two. Since the points A, B, C, D lie in between these two bounds, the predicted levels are higher than the measured levels, but the variation is within a factor of two.

A certain amount of discrepancy between the measured and predicted levels is to be expected because of the following factors:

1. The test panels were of finite length, whereas in the analytical model the panels were assumed to be infinitely long.
2. Variation of the panel damping characteristics over the test panel surfaces.
3. Variation of the field test sound pressure levels over the test panel surfaces.
4. Existence of certain amount of warping of the test panels.
5. Nonideal boundary conditions of the test panels, since the simply supported conditions along the frames were only approximately simulated.
6. Existence of tuning fork type modes of the stringer cross section, which was not accounted for in the analysis.

In spite of these differences there was a fair agreement between the prediction and the test data.

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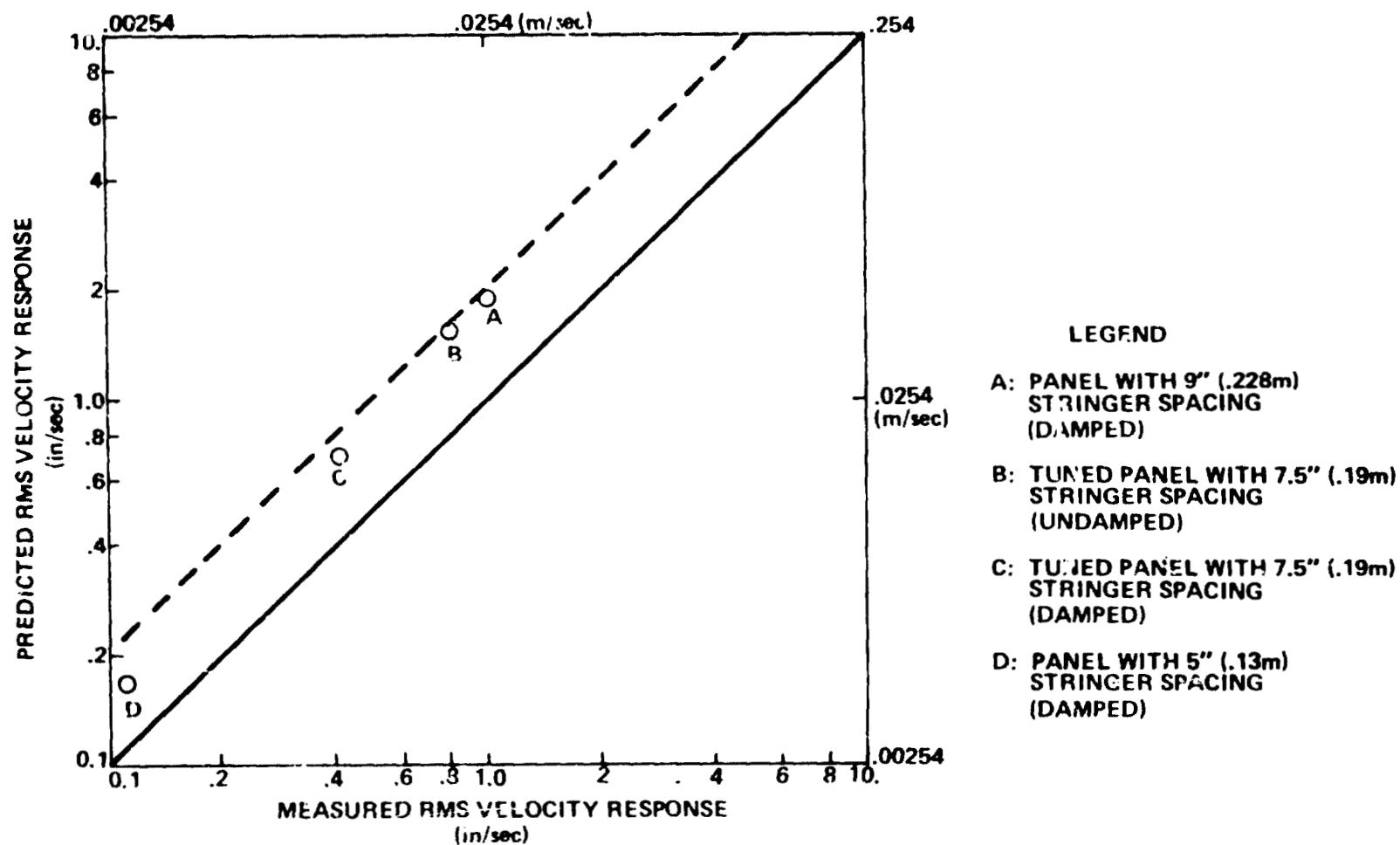


Figure 50.—Comparison of the Predicted and Measured rms Velocity Responses of the Low Frequency Modes

8.0 CONCLUSIONS

A good correspondence between the analytical prediction and the results of the laboratory and field tests has been observed. The following points have emerged from the study.

1. Analytical studies indicate that when the structural elements are intrinsically tuned, the response of a skin stringer panel does not pass through a peak at a frequency close to the fundamental frequency (f_p) of the individual skin bay, clamped along the stringers and simply supported along the frames. On the contrary, the response of the panel is reduced considerably around this frequency. This has been verified by the laboratory and the field tests.

Analytical studies also indicate that two other modes appear at frequencies above and below the frequency f_p , as shown in figure 6. The responses of both these modes can be reduced by applying damping treatment on the stringers, provided the stringer loss factor is below a certain optimum value. The existence of these two modes as well as the effect of stringer damping is verified by tests conducted in the laboratory and in the field.

2. Laboratory test data indicate that with no external damping treatment the damping loss factors of the low frequency modes pass through a maximum when the skin and stringers are intrinsically tuned.
3. Laboratory test data indicate that in the absence of any external damping treatment, the skin-stringer panel response of the most important low frequency mode passes through a point of diminishing return when the tuning condition is satisfied.
4. The predicted modes of the panel with 9-in. (0.228-m) stringer spacing were excited by excitations at the center of each skin bay. On the other hand, the predicted modes of the panel with 5-in. (0.13-m) stringer spacing were excited by excitation at the center of each stringer. The predicted modes of the tuned panel were excited by a combination of both types of excitation. This was further confirmed by the field test data. Care should, therefore, be exercised when discrete forcing functions are used to simulate a distributed pressure field.
5. The field transmission loss of the undamped tuned panel with 7.5-in. (0.19-m) stringer spacing shows two valleys at frequencies corresponding to the two principal modes shown in figure 6. In between these two valleys, the transmission loss passes through a maximum. This occurs in a frequency range in which the vibration energy is transferred from the skin panel to the stringers because of intrinsic structural tuning. Such a method of increasing the panel transmission loss in certain frequency bands should be particularly useful in situations where the excitation is caused by discrete tones; e.g., in turboprop or prop-fan aircraft.

Application of damping treatment on the stringers effectively smoothed out the valleys and the peaks in the transmission loss spectrum. There was a 4-dB improvement around 125 Hz, a 6-dB improvement around 160 Hz, and another 2.5-dB improvement around

400 Hz. Further improvements should be possible in all these bands by applying damping tapes on the skin. Therefore, intrinsic tuning broadens the frequency band over which damping treatment on the stringers and the skin can be effective.

6. The field transmission loss of the stringer damped panel with 5-in. (0.13-m) stringer spacing shows two valleys in two frequency bands containing two of its important modes. Both of these modes were excited in the laboratory test, using excitation at the center of each stringer. The mode shape at the lower frequency is such that the skin and stringers vibrate in phase. At this frequency, the skin acts like a relatively stiff member supported on relatively flexible stringers. For this reason, application of damping treatment on the stringers was highly effective in increasing the transmission loss in the frequency band containing the above mode. On the other hand, application of damping tape on the skin would have been much less effective in this frequency band.

The mode shape at the higher frequency is such that the centers of the skin bay and the stringer vibrate out of phase. The structural wavelength is then comparable to the stringer spacing, and stringer damping is not as effective around this frequency. On the other hand, skin damping should be highly effective in this frequency band.

In between the two valleys at these two frequencies, the transmission loss passes through a maximum. In this frequency band, the velocity response at the stringer center is higher than that at the panel center. The existence of such a frequency band with a high transmission loss may be effectively utilized in situations where the excitation is caused by discrete tones; e.g., in prop-fan aircraft. One would then attempt to design the structure so that the excitation tone of greatest concern falls within such a frequency band.

7. The key to achieving a high transmission loss at low frequencies is to design the structure so that the skin bay frequency is much higher than the stringer frequency, and then applying damping treatment on the stringers. This condition is automatically satisfied in a pressurized fuselage in which the inplane tensile loads induced by cabin pressurization significantly increases the skin bay frequency. In a pressurized fuselage, application of damping treatment on the stringers should, therefore, be very effective in reducing cabin noise in the 250- to 450-Hz band. Traditionally, noise transmission in this frequency band in a pressurized fuselage has been regarded as stiffness-controlled, since this frequency band is below the fundamental frequency (typically 500 to 650 Hz) of the individual skin bay carrying inplane loads caused by pressurization, and since application of damping tapes on the skin bays is not very effective in this band.
8. A good agreement was obtained between the predicted and measured natural frequencies and mode shapes. The predicted and measured rms velocity responses at the skin panel center were compared for the most dominant low frequency modes, and an agreement was obtained within a factor of two. The predictions were based on the assumption that the test panels were infinitely long. Considering this and other factors that were not taken into account in the simplified theory, the agreement is reasonable. A better agreement could be obtained by including the above factors in a more detailed analysis.

9. In an aircraft fuselage, the skin thickness, stringer spacing, and cross section, etc., vary from one part of the fuselage to another. The excitation is also not uniform over the length of the fuselage. Thus, it could be difficult to achieve and maintain a high degree of intrinsic tuning of various structural components over a large section of the fuselage. However, it seems that from the designer's standpoint it is not always essential to have an exact matching of the natural frequencies of the various components. Perhaps the most important finding from the development of the intrinsic structural tuning concept has come from a better understanding of the mechanism of structural response. It is seen that the structural member having a natural frequency lower than the frequencies of the other members becomes the key element controlling the response of the stiffened structure around its fundamental frequency. At higher frequencies, the other members can also be dominant in controlling the response. This is particularly important when the structure responds to broadband excitation sources typical of turbo-jet and turbofan engines and boundary layer turbulence. Therefore, it is important to identify the key structural elements that control the structural response in various frequency ranges so that vibration energy can be transferred from the skin to the stiffeners, if necessary, and damping treatment can be applied to the various elements for effective control of cabin noise and structural vibration in different frequency ranges. In particular, in what is normally regarded as the stiffness-controlled region, the noise transmission may actually be controlled by stiffener resonances, depending upon the relationship between the natural frequencies of this skin bay and the stiffeners. Therefore, cabin noise in the so-called stiffness-controlled region may be effectively reduced by applying damping treatment on the stiffeners.

In contrast, the past attempts in reducing cabin noise at low frequencies were centered around increasing the structural stiffness (see for example, ref. 29). The present study has pointed out that the reduction of low frequency cabin noise by using such an approach follows a law of diminishing return.

Low frequency cabin noise and sonically induced stresses can, therefore, be effectively reduced by using the following criteria based on the intrinsic structural tuning concept. Damping should be applied on the skin if the skin-bay frequency f_p is lower than the stringer frequency f_s . Damping can be applied on the skin and on the stringers, if the structure is intrinsically tuned. For maximum low frequency noise radiation, the structure should be designed so that the skin frequency f_p is greater than the stringer frequency f_s , and then damping should be applied on the stringers. Additional skin damping tapes can also be applied to the skin for additional benefit in the mid frequency (500-1500 Hz) range. It is therefore necessary to know the frequencies of the various structural elements, and these can be easily obtained through analysis and testing.

Thus, the fuselage structure should be designed so that, in addition to satisfying the static strength requirements, it has a high transmission loss at low frequencies and a long sonic fatigue life. The development of the intrinsic structural tuning concept is a step in that direction. The concept is in its initial stages of development, and further analysis and testing of more representative sections of the fuselage structure will be necessary for realizing its full potential. In this study, the effect of tuning and damping the stringers was investigated by analyzing and testing skin-stringer panels. In order to

reduce cabin noise at lower frequencies involving overall cylindrical modes, a similar analysis and testing of periodically stiffened cylinders will be necessary, so that the effect of tuning and damping the frames (ref. 3) may be investigated. The effect of frame resonances on the overall cylindrical modes of the fuselage was discussed in reference 3 and it was observed that for certain combinations of frequencies and circumferential wave numbers, the overall cylinder response is essentially controlled by frame resonances. Therefore, application of damping treatment on the frames should also be effective in reducing low frequency noise transmission into the aircraft cabin.

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