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PREDICTION OF LIGHT AIRCRAFT INTERIOR NOISE

By

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16. Abstract

At the present time, predictions of aircraft interior noise depend heavily on empirical correction factors derived from previous flight measurements. However, to design for acceptable interior noise levels and to optimize acoustic treatments, analytical techniques which do not depend on empirical data are needed. This paper describes a computerized interior noise prediction method for light aircraft. An existing analytical program (developed for commercial jets by Cockburn and Jolly in 1968) forms the basis of some modal analysis work which is described. The accuracy of this modal analysis technique for predicting low-frequency coupled acoustic-structural natural frequencies is discussed along with trends indicating the effects of varying parameters such as fuselage length and diameter, structural stiffness, and interior acoustic absorption.

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By James T. Howlett and David A. Morales

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INTRODUCTION

Single engine propeller-driven aircraft have high levels of interior noise, particularly in the low frequency region, when compared with other forms of transportation (ref. 1). Reduction of these levels is desirable for comfort of the crew and passengers. In the development of methods for reducing the noise, for example by optimizing noise reduction treatments, accurate analytical methods for predicting interior noise can be of great assistance. Such prediction methods can be used to identify the most important sources and transmission paths of the noise and to determine the best combination of structural parameters, such as skin thickness and stringer spacing, for reducing the noise. At present, suitable prediction methods for light aircraft are not available. The work described in this paper was undertaken, therefore, as an initial step in development of prediction methods for light aircraft. This work concentrates on the low frequency noise that has been shown (ref. 1) to be important for small, single-engine, propeller-driven light aircraft and emphasizes analytical methods that account for the details of the structure such as ring and stringer stiffness, and for the details of the modal behavior that is expected to be important in the low frequency range. The analysis described in this paper uses the modal approach to interior noise prediction and has sufficient detail to indicate the effects on interior noise level of variations of the structural mass and stiffness, structural damping, and acoustic absorption.

CONSIDERATIONS FOR PREDICTION OF LIGHT AIRCRAFT INTERIOR NOISE
(Figure 1)

In attempting to predict aircraft interior noise, an adequate description of the noise sources, the noise transmission paths, and the interior treatment is fundamental(figure 1). In this paper, information on the noise sources is assumed to be available and the emphasis is placed on noise paths and interior treatment. This initial prediction effort considers those features of the paths and interior treatment that are felt to be most essential for low-frequency noise. Noise which is transmitted into the interior by windows, structural vibration, acoustic leaks, and flanking is not included. Interior treatments, like trim panels and fiberglass, are not treated in detail but their overall effects are included by incorporating acoustic damping in the analysis. On the other hand, stiffness of the fuselage sidewall, including effects due to curvature, circumferential frames, and longitudinal stringers, is considered very important and included in the analysis. Likewise, structural damping and interior absorption are included since they are important to the resonant response in the low-frequency range.

CONSIDERATIONS FOR PREDICTION OF LIGHT AIRCRAFT INTERIOR NOISE

SOURCES	PATHS	INTERIOR
PROPELLERS	FUSELAGE SIDEWALL	ABSORPTION
ENGINE EXHAUST	WINDOWS	STRUCTURAL DAMPING
ENGINE VIBRATION	STRUCTURE	TRIM PANELS
BOUNDARY LAYER	ACOUSTIC LEAKS	
	FLANKING	

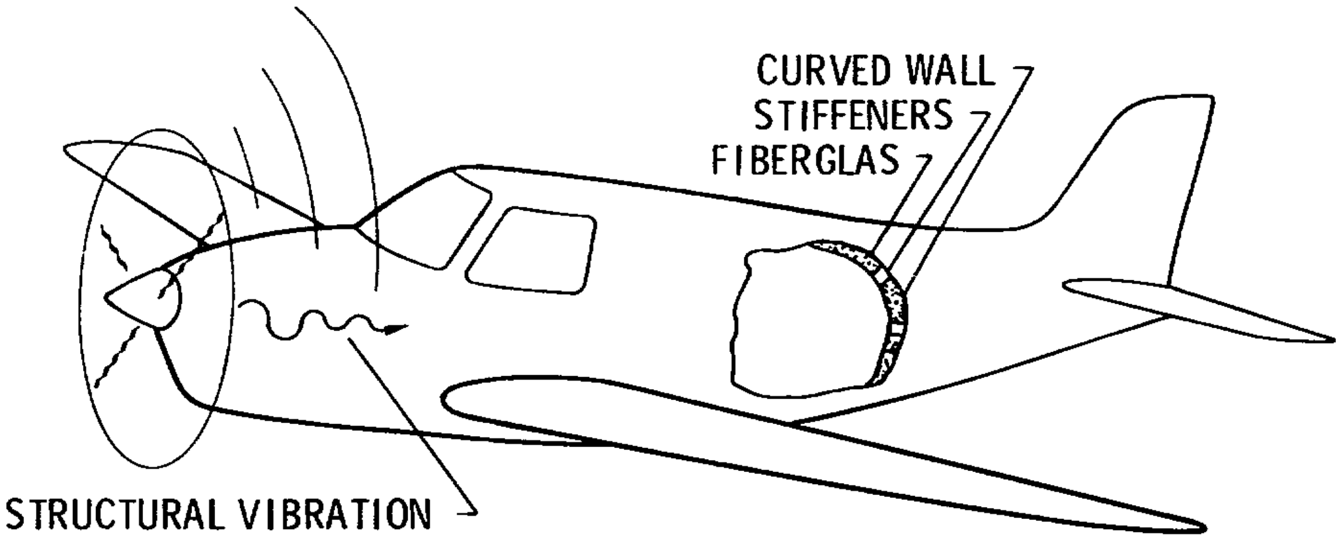
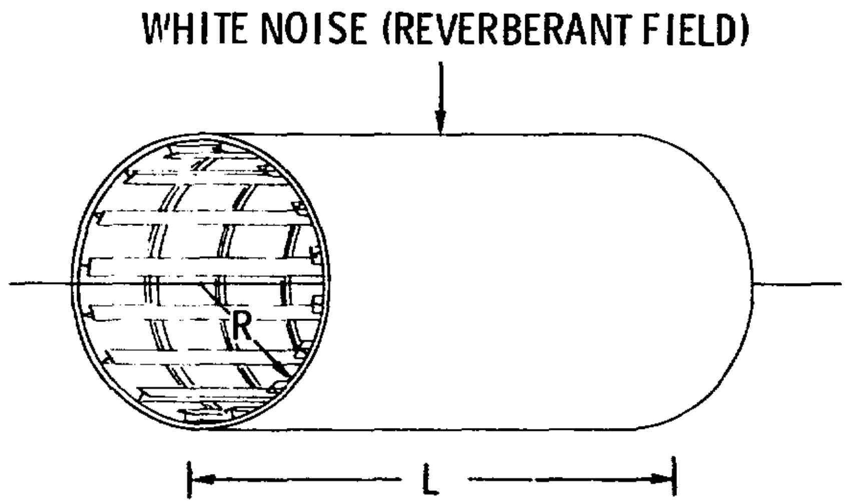


Figure 1

DESCRIPTION OF THE ANALYSIS
(Figure 2)

A sketch of the model analyzed in this study is shown in figure 2(a). The vehicle structure is modeled as a cylindrical shell with stiffness in both the circumferential and longitudinal directions. The analysis couples the interior volume of air to the structure by constraining the air particle displacement at the shell surface to be equal to the radial displacement of the shell. In the present study, the exterior pressure field is assumed to be reverberant white noise. A modal anslysis technique is used to obtain the space averaged interior pressure response for low frequency noise. This approach was used by Cockburn and Jolly (ref. 2) to analyze jet transport interior noise. The same techniques are used in the present study but with emphasis on values of the parameters that represent interior noise in light aircraft. The results in this paper discuss: (1) the interior noise in terms of the noise reduction (defined in fig. 2(b)), an (2) the effect of shell length and radius variations on the coupled frequencies. The parameters which are varied in the noise reduction studies are listed in figure 2(c). The ranges over which these parameters are varied include values appropriate for the aircraft considered.

DESCRIPTION OF ANALYSIS



a) SKETCH OF SHELL-ACOUSTIC SYSTEM

COUPLED MODES OF SHELL AND INTERIOR ACOUSTIC SPACE

$$\text{NOISE REDUCTION, NR, dB} = 10 \text{ LOG } \frac{PSD_{EXTERIOR}}{PSD_{INTERIOR}}$$

b) APPROACH

SHELL STIFFNESS
SHELL MASS
STRUCTURAL DAMPING
INTERIOR ABSORPTION
SYSTEM LENGTH AND RADIUS

c) PARAMETERS VARIED

Figure 2

FREQUENCY EQUATION FOR COUPLED ACOUSTIC STRUCTURAL SYSTEM
(Figure 3)

The frequency equation for the coupled acoustic-structural system which is derived in reference 2 is shown in figure 3. The unknown frequency is denoted by Ω. The frequencies of the stiffened shell are ω_{mn} and the organ pipe frequencies (i.e., axial acoustic resonances with constant radial pressure) of the interior air are ω_m. The speed of sound and density of air inside the fuselage are denoted by c and ρ, respectively. The radius and density of the shell are denoted by R and M. The nth order Bessel function is J_n and the prime denotes differentiation. Note that setting ρ=0 eliminates the last term of this equation. If this is done, a solution of the frequency equation yields the uncoupled shell frequencies (ω_{mn}), the organ pipe frequencies (ω_m), and the frequencies obtained by solving $J_n'=0$. The latter frequencies correspond to the acoustic frequencies for a hard wall boundary condition and, thus, the solution of the frequency equation for ρ=0 will be called the hard wall case. The solution for the hard wall case allows a comparis between exact frequencies, numerical results for the hard wall case, and coupled frequencies.

FREQUENCY EQUATION FOR COUPLED ACOUSTIC–STRUCTURAL MODES

$$\left(1-\frac{\Omega^2}{\omega_{mn}^2}\right)\sqrt{1-\frac{\omega_m^2}{\Omega^2}}\; J_n'\left(\frac{\Omega R}{c}\sqrt{1-\frac{\omega_m^2}{\Omega^2}}\right)+\frac{\rho c \Omega}{M\omega_{mn}^2}\, J_n\left(\frac{\Omega R}{c}\sqrt{1-\frac{\omega_m^2}{\Omega^2}}\right)=0$$

Ω	-	FREQUENCY OF COUPLED MODES
ω_m	-	FREQUENCY OF ACOUSTIC ORGAN PIPE MODES
ω_{mn}	-	FREQUENCY OF SHELL STRUCTURE
ρ, c	-	DENSITY AND SPEED OF SOUND OF INTERIOR AIR SPACE
R	-	RADIUS OF SHELL
M	-	DENSITY OF SHELL
J_n	-	BESSEL FUNCTION

Figure 3

EFFECT OF COUPLING ON MODAL FREQUENCIES
(Figure 4)

Figure 4 shows the effect of acoustic-structural coupling on natural frequencies. The higher frequencies are included in the figure to demonstrate the relationship between the three cases shown in the figures but are not important for the forced response calculations to be discussed later. As the figure indicates, the hard wall results are identical to the frequencies obtained from the exact solution (ref. 2). This indicates that the approximate numerical technique used to solve the equation in figure 3 is producing results of acceptable accuracy. For practical purposes, the coupled natural frequencies are the same as those obtained for the hard wall condition. In all the cases for which this comparison has been made, the coupled frequencies are very close to the hard wall values. Thus, for the ranges of parameters considered in this paper, coupling the shell structure to the interior air does not appear to affect the natural frequencies of the model analyzed. This suggests that uncoupled modes might be used in an analysis of this problem. Coupled modes were used in the present analysis to provide the mechanism for inducing motion in the enclosed air space in response to the motion of the shell. This mechanism is required because the exterior noise field acts only on the shell structure and not on the enclosed air.

EFFECT OF COUPLING ON MODAL FREQUENCIES

MODE	FREQUENCY, Hz		
	EXACT	HARDWALL	COUPLED
1st ORGAN PIPE	76	76	76
2nd ORGAN PIPE	152	152	152
STRUCTURAL	479	479	475
ACOUSTIC RADIAL	797	797	796
ACOUSTIC RADIAL	1153	1153	1151

Figure 4

VARIATION OF MODAL FREQUENCY WITH LENGTH AND RADIUS
(Figure 5)

Figure 5 indicates the effects of varying the length and radius of the shell on natural frequencies. The ranges of length and radius chosen correspond to values which are considered reasonable to model a light aircraft. Figure 5(a) indicates the meaning of the mode numbers m, n, and s. In figure 5(b), note that the frequencies of the (1, 0, 0) and (2, 0, 0) modes are not changed by increasing the radius. This result is expected because these are the first and second organ pipe modes involving primarily longitudinal motion of the air inside the shell and, hence, the frequencies should not be affected by changing the radius. The frequencies of these modes do, of course, decrease with increasing length as shown in figure 5(c). The results shown in the rest of the paper are for a shell which is 225 cm long and has a radius of 48 cm. The frequency range chosen for the noise reduction studies was 50 to 200 Hz. Figure 5 shows the only two acoustic modes, the (1, 0, 0) and (2, 0, 0) organ pipe modes, which lie in this frequency range. There are no structural modes in this frequency range. Thus, for this analysis the shell is acting in its "stiffness-controlled" region while the interior air space is acting in its resonant-response region.

VARIATION OF MODAL FREQUENCY WITH LENGTH AND RADIUS

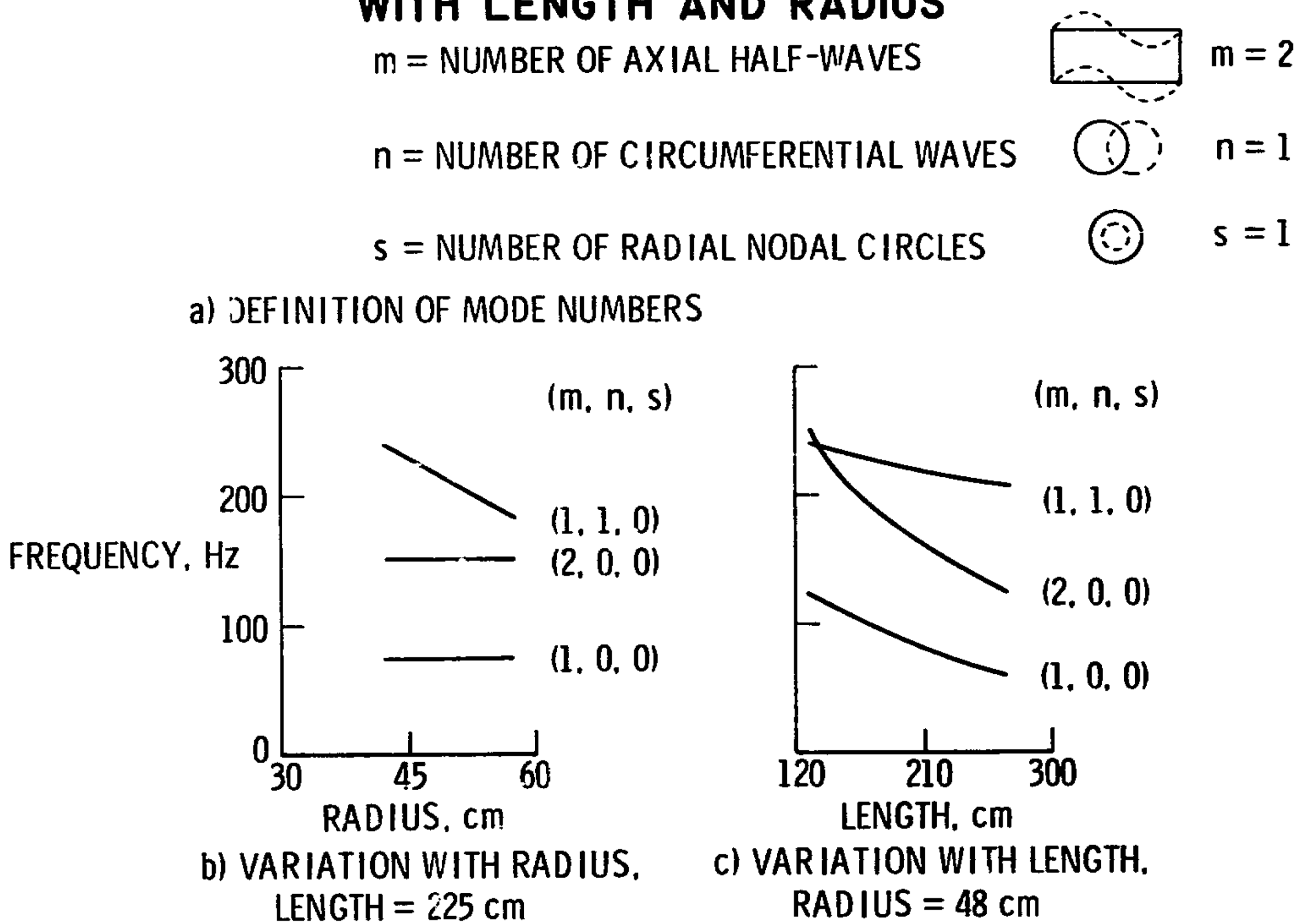


Figure 5

EFFECT OF DAMPING ON NOISE REDUCTION
(Figure 6)

Figure 6 shows the calculated effects of increasing structural damping and acoustic absorption on noise reduction (NR) for the frequency range 0 to 200 Hz. The results shown are for an aluminum shell 225 cm long and 48 cm in radius. The shell has ring stiffeners approximately 81 cm apart and longitudinal stiffeners 36 cm apart. Figure 6 shows that for all three cases, there are two locally minimum values of noise reduction at approximately 75 Hz and 150 Hz. These frequencies correspond to the first two organ pipe modes for this system. Comparing figure 6(a), 6(b), and 6(c) shows that the principal effect is due to the increased acoustic absorption, which increases the noise reduction in the region of these organ pipe frequencies. For example, comparing figure 6(a) with figure 6(c) shows that increasing the absorption increased the noise reduction in the 150 Hz region from about 5dB to 15dB. The noise reduction at nonresonant frequencies, however, was not greatly affected by changes of damping (e.g., noise reduction at about 100 Hz is about 32dB for all three values of damping). Thus, the interior noise reduction from increased damping can be expected to be limited because of the low number of acoustic modes in the frequency range studied.

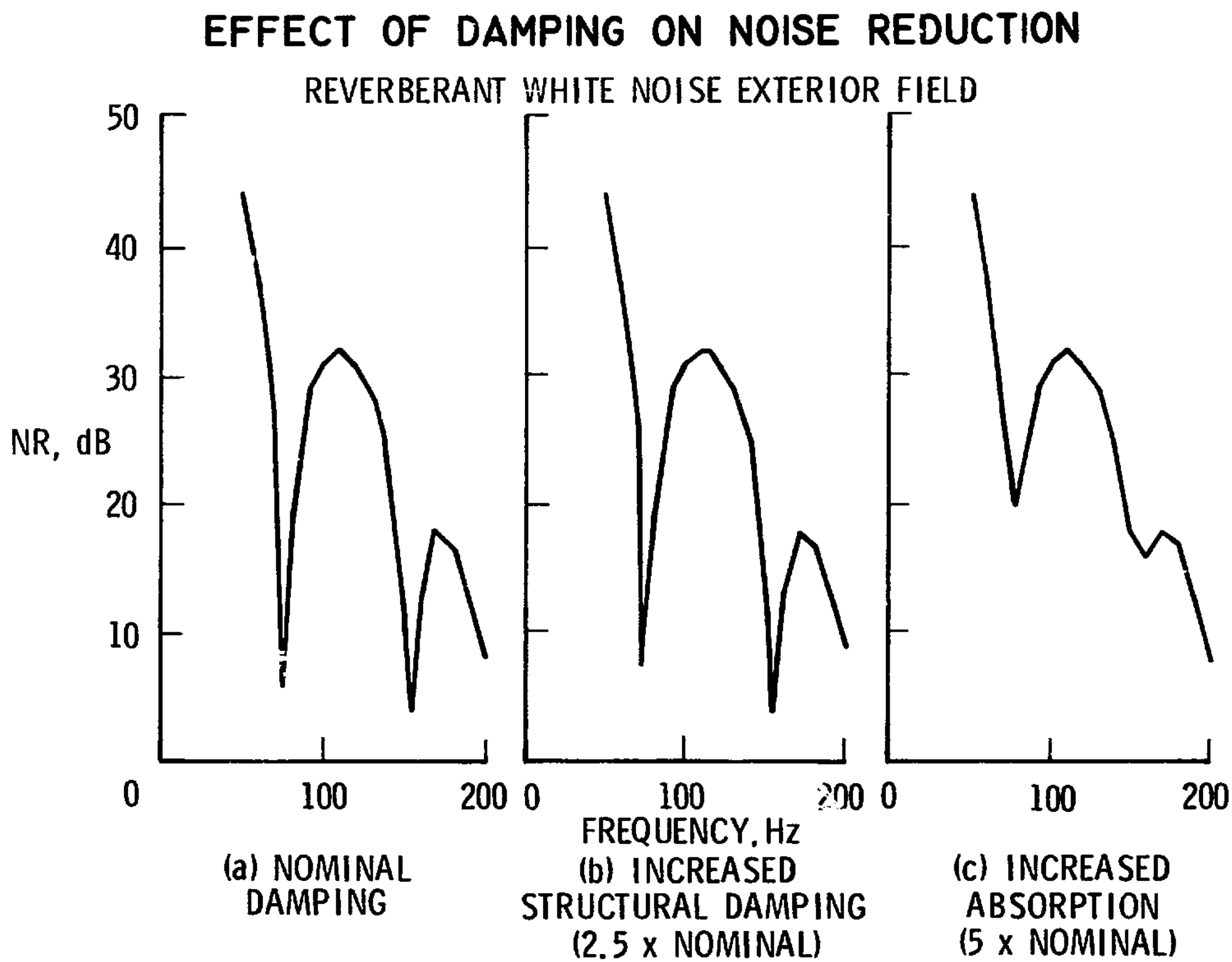

EFFECT OF DAMPING ON NOISE REDUCTION
REVERBERANT WHITE NOISE EXTERIOR FIELD
NR, dB
50
40
30
20
10
0
100
200
0
100
200
0
100
200
FREQUENCY, Hz
(a) NOMINAL DAMPING
(b) INCREASED STRUCTURAL DAMPING (2.5 x NOMINAL)
(c) INCREASED ABSORPTION (5 x NOMINAL)

Figure 6

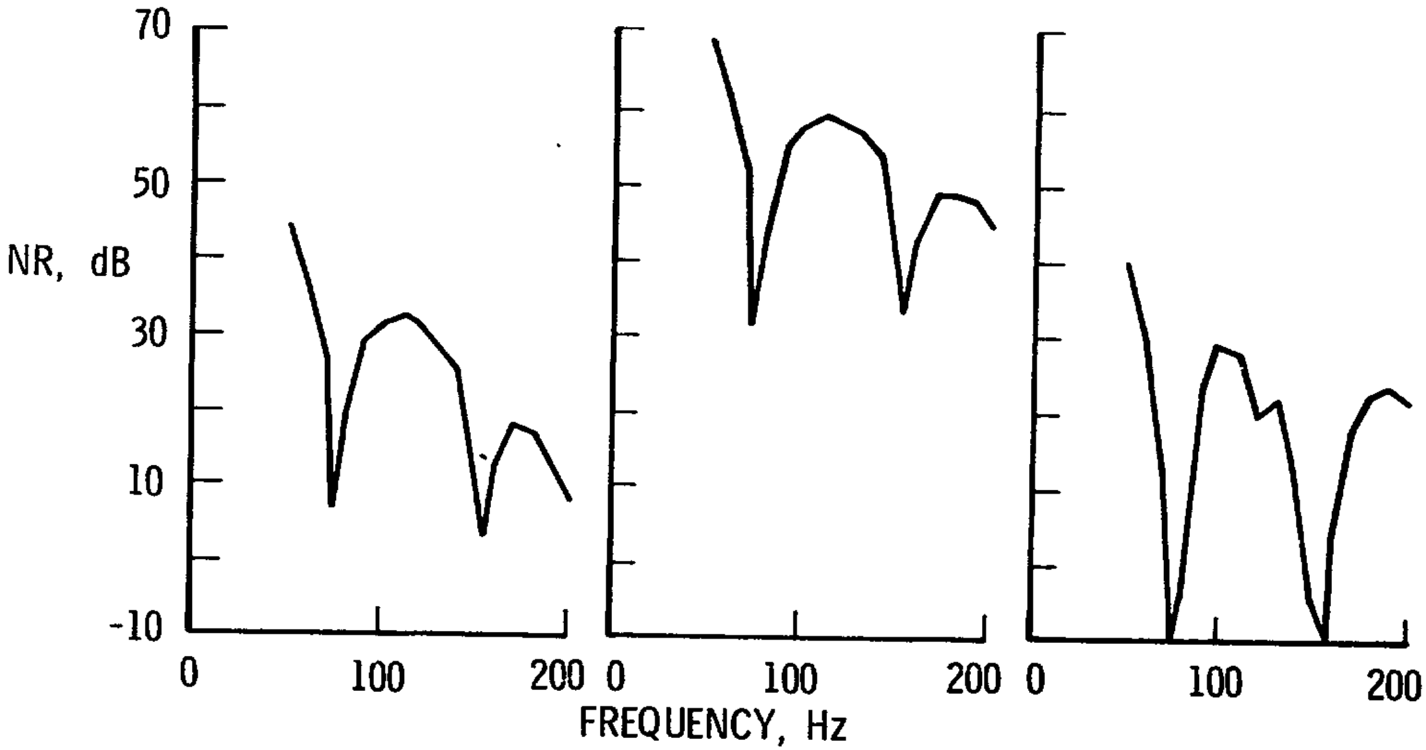
EFFECT OF SHELL STIFFNESS AND MASS ON NOISE REDUCTION (Figure 7)

The noise reduction curve shown in figure 7(a) has been reproduced from figure 6(a) for reference. Figure 7(b) shows the noise reduction calculated with 10 times the shell stiffness as used for figure 7(a), but with all other parameters, including mass and damping, having the same values as for figure 7(a). Figure 7(b) shows that the noise reduction has been increased by about 30dB by the increased stiffness, and further that the NR has been increased by about the same amount at all frequencies. Figure 7(c) shows that increasing the mass decreased NR to negative values at frequencies near the modal resonances. The amplifications shown in figure 7(c) by the negative values of NR at 75 and 150 Hz are thought to result from structural modes in this frequency range that are present with the increased mass, figure 7(c), but not with the nominal values, figure 7(a).

Figure 7(a) shows that calculated values of NR in this frequency range are from about 5 to 30dB. The values of the parameters used to calculate this NR curve were chosen to be representative of a small single engine light aircraft. Measured values of NR in a reverberation room (unpublished data) for a particular light aircraft were in the range of 10-20dB in this frequency range.

EFFECT OF SHELL STIFFNESS AND MASS

REVERBERANT WHITE NOISE EXTERIOR FIELD



(a) NOMINAL (ALUMINUM)

(b) INCREASED STIFFNESS (10 x NOMINAL)

(c) INCREASED MASS (10 x NOMINAL)

Figure 7

CONCLUSIONS AND RECOMMENDATIONS
(Figure 8)

A summary of the conclusions resulting from this initial study is shown in figure 8. Because of the low modal density, the modal analysis technique is promising for calculating low frequency interior noise in light aircraft. The calculated results indicate that increased stiffness is the best way to obtain increased NR over this entire 50-200 Hz frequency range considered. Although preliminary experimental results indicate that the predicted values of noise reduction are reasonable, a controlled experiment on an actual vehicle in a reverberant field is a desirable next step in order to determine whether the analysis correctly predicts the actual values of noise reduction.

CONCLUSIONS

MODAL DENSITY IS LOW IN LOW FREQUENCY RANGE FOR LIGHT AIRCRAFT SIZE STRUCTURE

INCREASED STIFFNESS INCREASED NR OVER WHOLE 50 - 200 Hz RANGE

INCREASED MASS DECREASED NR AT MODAL FREQUENCIES

Figure 8

REFERENCES

1. Catherines, John J. and Mayes, William H.: Interior Noise Levels of Two Propeller-Driven Light Aircraft. NASA TM X- 72716, July 1975.

2. Cockburn, J. A. and Jolly, A. C.: Structural-Acoustic Response, Noise Transmission Losses and Interior Noise Levels of an Aircraft Fuselage Excited by Random Pressure Fields. Air Force Flight Dynamics Laboratory Technical Report. AFFDL-TR-68-2, August 1968.