TRANSMISSION LOSS THROUGH A HONEYCOMB-SANDWICH CYLINDRICAL SHELL

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This is an unfinished tutorial….

Variables

c	= speed of sound in the surrounding air medium
c_L	= longitudinal wave velocity
d	= diameter
D	= bending stiffness
E	= elastic modulus
E_f	= face sheet elastic modulus
f	= forcing frequency
f_c	= critical frequency
f_{mn}	= bending frequency for mode (m, n)
f_r	= ring frequency
h	= thickness
TL	= random-incidence transmission loss
TL_0	= normal incidence transmission loss
TL_{cyl}	= transmission loss for the honeycomb-sandwich cylindrical shell
ρ	= mass per volume of the shell
$\bar{\rho}$	= surface mass density of the cylindrical shell
ρc	= characteristic impedance of air
η	= loss factor
ν	= Poisson ratio

Note that the characteristic impedance of dry air at 20 °C is ρc = 415 (Pa • sec/m).

Introduction

Certain payload fairings on rocket vehicles are constructed from honeycomb-sandwich material. The overall geometry is typically that of a cylindrical shell with a nosecone.[1]

The fairing protects the payload from a variety of environments. The liftoff acoustics and the aerodynamic-induced sound pressure are particular concerns in this tutorial. The acoustic energy from these sources can damage the payload and its electronic components. Solar panels, antennas, and other appendages are also at risk.

The fairing's transmission loss must be determined in order to calculate the resulting sound pressure level within the fairing. The purpose of this tutorial is to develop a calculation method. The method is an extension of the Franken method, where the Franken method is calibrated using Reference 1.

Mass Law

Reference 1 states:

> The potential of the basic sandwich panel as an airborne sound insulator is limited by the mass law. Departure from mass law occurs because of resonance conditions in some frequencies, such as the normal resonance frequencies, the coincidence frequencies, the dilational resonance frequencies, and the double wall resonance frequency. The effect of resonances can be avoided by adequate design procedures.

Honeycomb-sandwich shells have the potential of yielding greater attenuation than that predicted by the mass law at frequencies well below the cylinder's ring frequency, particularly due to the stiffness and damping of the shell.

Further information regarding the mass law is given in Reference 5.

Shear and Bending Waves

The shear and bending wave speeds should be lower than the speed of sound to avoid coincidence in the frequency band of interest. The bending wave coincidence frequency must thus be as high as possible. See Appendix A for further information.

Franken Method Assumptions

The Franken method is an empirical method for calculating the vibration response of a cylindrical skin to an external acoustic pressure field.

The Franken method assumes

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[1] There are many possible geometrical variations. As further examples, the nosecone might be biconic, or it might be an ogive.

1. All flight vehicles to which the procedure is applied have similar dynamic characteristics to the Jupiter and Titan 1 vehicles.
2. Vibration is primarily due to the acoustic noise during liftoff or other pressure fields during flight which can be estimated.
3. The vibration magnitude is directly proportional to the pressure level of the excitation and inversely proportional to the surface weight density of the structure.
4. Predominant vibration frequencies are inversely proportional of the diameter of the vehicle.
5. Spatial variations in the vibration can be considered as a random variable.

<u>Steps</u>

Each of the following steps represents a simplifying assumption.

Step 1. Calculate the Transmission Loss at the Ring Frequency

Assume a loss factor. Note that the loss factor is twice the fraction of critical damping.

Calculate the normal-incidence transmission loss TL_0 at the ring frequency, per Reference 1, equation (26).

$$TL_0(f) = 10 \log \left\{ \left[1 + \frac{\eta \pi \bar{\rho} f}{\rho c} \left(\frac{f_{mn}}{f} \right)^2 \right]^2 + \left(\frac{\pi \bar{\rho} f}{\rho c} \right)^2 \left[1 - \left(\frac{f_{mn}}{f} \right)^2 \right]^2 \right\} \tag{1}$$

Equation (1) could be used to calculate a transmission loss for each modal frequency and for each forcing frequency. Nevertheless, a single transmission loss value is needed. Thus, set both f and f_{mn} equal the ring frequency f_r for this calculation.

$$f = f_{mn} = f_r \tag{2}$$

The normal-incidence transmission loss at the ring frequency is thus

$$TL_{0,ring} = 10 \log \left\{ \left[1 + \frac{\eta \pi \bar{\rho} f}{\rho c} \right]^2 \right\} \tag{3}$$

Assume normal incidence for liftoff acoustics. Otherwise, the random-incidence TL can be evaluated from the following approximate expression:

$$\mathrm{TL} \approx \mathrm{TL}_0 - 5\,\mathrm{dB} \quad (4)$$

Step 2. Calculate the Uncalibrated Franken Function

Assume that the internal noise results directly from vibration of the shell.

Calculate the PSD (G^2/Hz) function for the given cylinder geometry assuming that the SPL has unity dB amplitude in each of the one-third octave bands.

Convert the PSD function to dB format, where the zero dB reference is arbitrary.

Step 3. Normalize the Franken Function

Add a dB margin to the uncalibrated Franken function so that the level at the ring frequency is zero dB. Apply this margin uniformly in each one-third octave band.

Step 4. Calibrate the Normalized Franken Function

Use the $\mathrm{TL}_{0,\mathrm{ring}}$ value[2] to convert the normalized Franken vibroacoustic curve into a transmission loss function for the cylindrical shell $\mathrm{TL}_{\mathrm{cyl}}$. The value is added uniformly to each one-third octave band.

Step 5. Conservative Limits

Apply limits to the maximum $\mathrm{TL}_{\mathrm{cyl}}$ magnitude for conservatism using engineering judgment. For example the transmission loss below and above the ring frequency could be limited to a certain dB level.

<u>Example</u>

An example of this method is given in Appendix C.

<u>Issues</u>

Modal density and modal damping….unknown…

Field type modeling ….

[2] Or the equivalent random-incident value as appropriate.

Interior field may be reverberant, but complicated by the geometry of the payload. See Fill factor in Reference 8.

Franken…

Bending and shear wave coincidence with acoustic wavelengths at low frequencies.

Non-linearity due to large deflections. Membrane and bending stiffness….

Blankets – Reference 6.

Flanking noise

Vent noise

References

1. H. Wen-chao & Ng Chung-fai, Sound Insulation Improvement using Honeycomb Sandwich Panels, Applied Acoustics, Vol. 53, No. 1-3, 1998.

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3. L. Beranek & I. Ver, Noise and Vibration Control Engineering, Principles and Applications, Wiley, New York, 1992. (See page 297).

4. T. Irvine, Vibration Response of a Cylindrical Skin to Acoustic Pressure via the Franken Method, Revision E, Vibrationdata, 2004.

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APPENDIX A

Bending Wave, Critical Frequency

The critical (or coincident) frequency is the frequency at which the airborne acoustic wavelength matches the panel bending wavelength. The critical frequency f_c for a homogeneous panel is

$$f_c = \frac{c^2}{2\pi h}\sqrt{\frac{12(1-\nu^2)\rho}{E}} \tag{A-1}$$

$$C_L = \sqrt{\frac{E}{\rho}} \tag{A-2}$$

$$f_c \approx \frac{c^2}{1.8\, c_L\, h} \tag{A-3}$$

Equation (A-3) is taken from Reference 2, chapter 6, equation (56a).

The corresponding equation for a honeycomb-sandwich shell is not immediately available.

A cylinder shell is considered as “acoustically thin” if $f_r < f_c$.

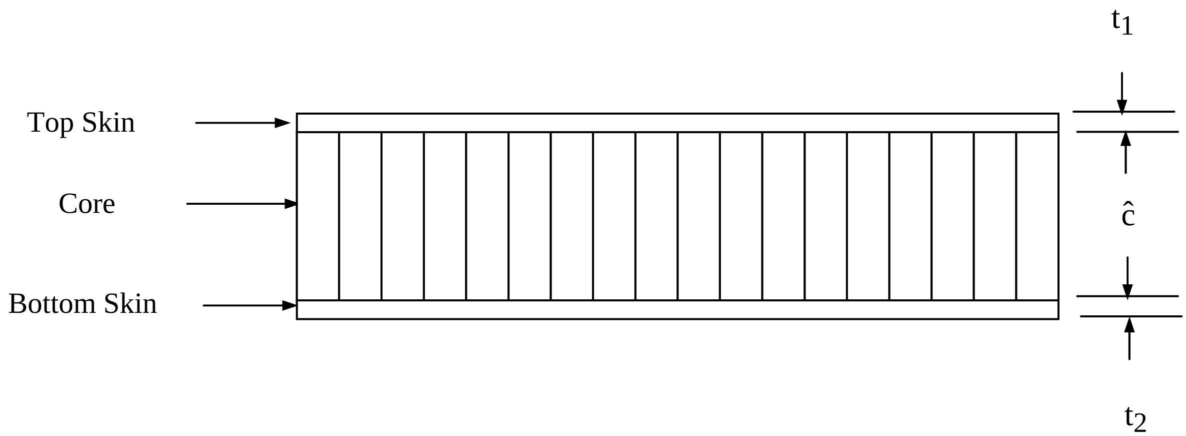
The ring frequency f_r is given in Appendix B.

Furthermore, the cylindrical shell tends to vibrate as a flat plate above the ring frequency because the shell’s curvature is less important.

$$f_c = \frac{{c_{air}}^2}{2\pi\, h}\sqrt{\frac{12\,(1-\mu^2)\rho}{E}} \qquad \text{(A-4)}$$

$$f_c = \frac{{c_{air}}^2}{2\pi\, h}\sqrt{\frac{\rho\, h^3}{D}} \qquad \text{(A-5)}$$

$$f_c = \frac{{c_{air}}^2}{2\pi}\sqrt{\frac{\rho\, h}{D}} \qquad \text{(A-6)}$$



The bending stiffness from Reference 7 is

$$D = \left[\frac{E_f}{1-v^2}\right]\left[\frac{t_1\, t_2\, \hat{h}^2}{t_1 + t_2}\right] \qquad \text{(A-7)}$$

where

$$\hat{h} = \hat{c} + \frac{1}{2}\left[t_1 + t_2\right] \qquad \text{(A-8)}$$

$$f_c = \frac{c^2}{2\pi}\sqrt{\frac{\rho\, h\left[1-v^2\right]\left[t_1 + t_2\right]}{E_f\left[t_1\, t_2\;\, h^2\right]}} \qquad \text{(A-9)}$$

$$f_C = \frac{c^2}{2\pi}\sqrt{\frac{\rho\left[1-v^2\right]\left[t_1+t_2\right]}{E_f\, h\left[t_1\ t_2\right]}} \qquad \text{(A-10)}$$

APPENDIX B

The ring frequency f_r is the frequency at which the longitudinal wavelength in the skin material is equal to the vehicle circumference.

$$f_r = \frac{C_L}{\pi d} \qquad \text{(B-1)}$$

where

C_L is the longitudinal wave speed in the skin material

d is the diameter

APPENDIX C