Notes on Honeycomb Sandwich Structures
Revision B

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Introduction

Honeycomb sandwich structures are designed to have a high stiffness-to-mass ratio.

The transmission loss depends on many factors including:

1. Whether the honeycomb sandwich is in the form of a flat panel or a cylindrical shell
2. Geometrical dimensions
3. Materials
4. The critical and coincidence frequencies
5. The ring frequency in the case of a cylindrical shell and whether it is acoustically thick or thin
6. Shear-to-bending transition
7. Boundary conditions
8. Damping
9. Modal density, natural frequencies, wavelengths, wavenumbers, phase speeds
10. Classification of modes in terms of subsonic (acoustically slow) or supersonic (acoustically fast)
11. Waveform dispersion
12. Potential dilatational mode
13. Acoustic field type and angle of incidence
14. External flow type if applicable
15. Spectral content and corresponding wavelengths of the acoustic field
16. Pressure differential between the external and internal air volumes

Note that the critical frequency is the frequency at which the speed of the free bending wave in a structure becomes equal to the speed of the airborne acoustic wave. The corresponding wavelengths are likewise equal.

The coincidence frequency depends on the critical frequency and on the angle of incidence.

Response by Frequency Range

Furthermore, honeycomb sandwich structures have three ranges characterizing their response behavior.

Low Frequencies	Bending of the entire structure as if were a thick, homogenous plate or shell
Mid Frequencies	Transverse shear strain in the honeycomb core governs the behavior
High Frequencies	The structural skins act in bending as if disconnected

Mass or Stiffness Controlled

Authors differ as to whether the transmission loss tends to be mass or stiffness controlled. They likewise differ as to whether sandwich structures offer "good" or "poor" attenuation. The correct assessment depends of the interplay of the factors given above.

The high stiffness and low weight of certain sandwich structures can result in supersonic wave propagation at relatively low frequencies, which adversely affects the acoustical performance at these frequencies. The result of these factors tends to be poor acoustic attenuation.

Other sandwich structures, however, have shear phase speeds in the shear-controlled region which are lower than the speed of sound. The combination of low mass and high transverse stiffness provides good attenuation at low frequencies.

Improving Transmission Loss by Filling the Core

The transmission loss can be improved by filling the core with granular material according to Reference 4. This lowers the fundamental frequency and increases the mass density. These changes force the response into the “mass controlled region” of the transfer function magnitude.

Improving Transmission Loss by Adding Acoustic Blankets

Another approach is to use acoustic blankets to supplement the honeycomb-sandwich structure,

Consider a launch vehicle payload fairing. Acoustic blankets may be mounted on the inside walls of the fairing to reduce the fairing's interior acoustics, as well as the resulting vibration response of the payload. The blankets convert the acoustic energy to heat.

These blankets are usually made of fiberglass batting or a combination of fiberglass sheets and batting which are of different thicknesses and are layered together. Melamine foam is another material used in blankets.

The blankets provide little attenuation in the lower frequency range, say, below 100 Hz. The wavelengths are relatively long at low frequencies, and there is not a sufficient depth of material in blankets to absorb a significant portion of the energy. As an example, a 100 Hz acoustic wave has a length of 3.43 meters assuming the ground, ambient speed of sound at 343 m/sec.

Also note that the blanket thickness is limited by mass and volume constraints.

Critical Frequency for a Flat Plate

The shear and bending wave speeds should be lower than the speed of sound to avoid coincidence in the frequency band of interest. The critical frequency must thus be as high as possible.

The critical frequency of a flat panel can be increased by:

1. Decreasing the thickness
2. Decreasing the elastic modulus
3. Increasing the mass density

Potential design changes must be weighed against the other factors in the list in the Introduction section.

Doubling the thickness has the disadvantage of reducing the critical frequency by one-half. But this change would also increase the transmission loss by 6 dB below the critical frequency.

Critical and Ring Frequencies for a Cylindrical Shell

Note that a flat panel has a well-defined critical frequency. A cylindrical shell does not have a unique critical frequency, however. Assume that the critical frequency of a cylindrical shell is the same as that of a flat plate with the same shell thickness.

The transmission loss of a cylinder tends to be minimal at its ring frequency. The radiation efficiency is the highest at this frequency.

The transmission loss may also be minimal at the critical frequency if the facesheets are aluminum per Reference 6. This characteristic does not appear to be true for composite facesheets, however.

The cylinder tends to vibrate as a flat panel above its ring frequency because the curvature effects are less important.

A cylindrical shell is acoustically thin if its ring frequency is less than the critical frequency.

A cylindrical shell is acoustically thick if its ring frequency is above its critical frequency.

Further information is given in Reference 2.

Cylindrical Shell Response

The response of a cylindrical shell to an external acoustic field is discussed further in Appendix A, as taken from Reference 5.

Acoustic Field Types

The common acoustic field types are:

1. Normal incidence
2. Oblique incidence
3. Field incidence
4. Random incidence
5. External airflow

Note that field incidence approximates a diffuse incidence sound field with a limiting angle of about 78 degrees.

The transmission loss tends to be the greatest for a given frequency when the incidence is normal. The transmission loss tends to be the least for random incidence.

References

1. T. Irvine, Natural Frequencies of a Honeycomb Plate, Rev F, Vibrationdata, 2008.
2. T. Irvine, Vibroacoustic Critical and Coincidence Frequencies of Structures, Rev G, Vibrationdata, 2008.
3. T. Irvine, Transmission Loss of a Plane Wave through a Multi-Layer Partition, Vibrationdata, 2009.
4. Vibration, Shock, Acoustics; McDonnell Douglas Astronautics Company, Western Division, 1971. (See section 3-1.8)
5. T. Irvine, Vibroacoustic Response of a Cylinder, Rev A, Vibrationdata, 2009.
6. Tang, Robinson, & Silcox; Sound Transmission through a Cylindrical Sandwich Shell with Honeycomb Core, NASA Document: 20040110815.

APPENDIX A

Introduction

A homogeneous, single-layer cylindrical shell was modeled in Reference 5. It was modeled as a single-degree-of-freedom system with its natural frequency equal to the ring frequency.

Excerpts of the analysis from Reference 5 are given in this appendix. The conclusions can be applied to a honeycomb cylindrical shell with proper caution.

The acoustic transmission loss is assumed to be inversely proportional to the displacement.

Example 1

Consider an aluminum cylinder idealized as a single-degree-of-freedom system.

The cylinder has the following properties for each of two cases:

Diameter	36 inch
Damping	5%
Wall Thickness	0.125 inch for case 1 0.250 inch for case 2
Surface Mass Density	0.0125 lbm/in^2 for case 1 0.025 lbm/in^2 for case 2

The ring frequency is 1792 Hz for each case.

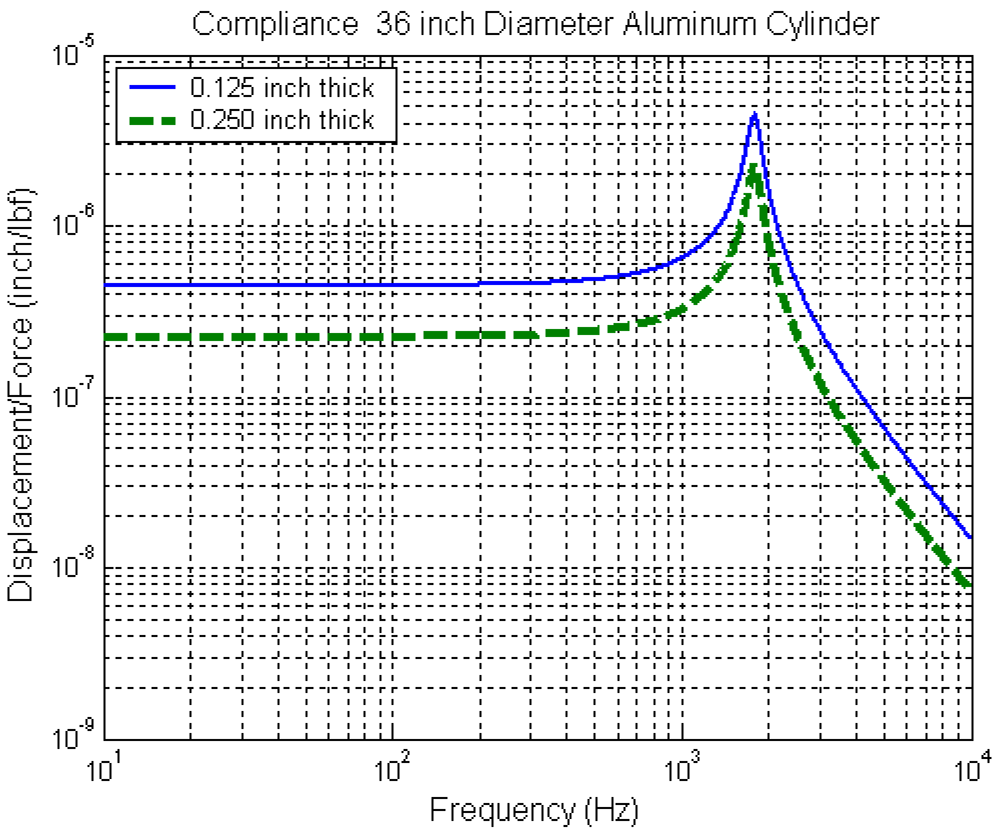

Figure A-1.

Doubling the wall thickness, and hence the surface mass density, decreases the response by 6 dB for a fixed natural frequency.

Example 2

Consider an aluminum cylinder idealized as a single-degree-of-freedom system.

The cylinder has the following properties for each of two cases:

Diameter	36 inch for case 1 72 inch for case 2
Damping	5%
Wall Thickness	0.250 inch
Surface Mass Density	0.025 lbm/in^2
Ring Frequency	1792 Hz for case 1 896 Hz for case 2

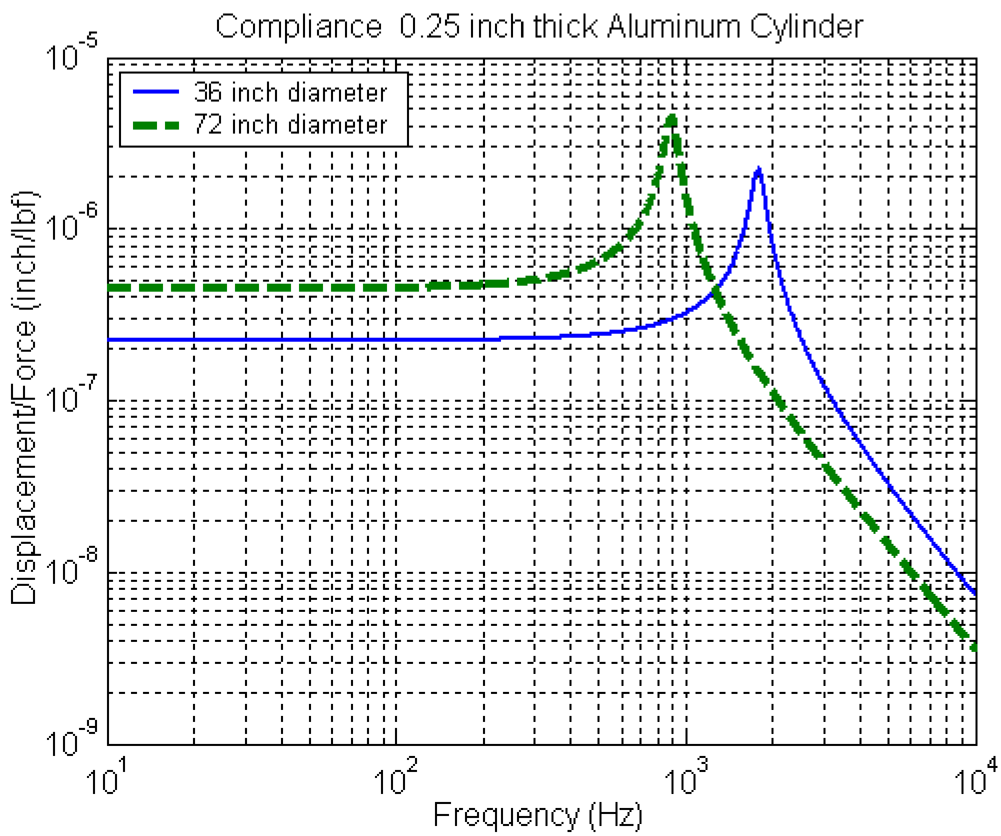

Figure A-2.

The comparison shows the trade-offs involved by changing the diameter and hence the ring frequency.

The following statements apply to a cylinder with a constant wall thickness and constant surface mass density:

a. A stiffer cylinder offers better attenuation at frequencies well below the ring frequency.

b. A more compliant cylinder provides better attenuation at frequencies well above the ring frequency.

Furthermore, the above statements assume normal incidence.