### INVESTIGATION OF INVERSE ACOUSTICAL CHARACTERIZATION OF POROUS MATERIALS USED IN AIRCRAFT NOISE CONTROL APPLICATION

A Thesis by

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The following faculty members have examined the final copy of this thesis for form and content, and recommend that it be accepted in partial fulfillment of the requirement for the degree of Master of Science with a major in Mechanical Engineering.

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### DEDICATION

To my beloved parents and sister

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V

#### ABSTRACT

Sound propagation through porous media such as foams and fibers is governed by five parameters that describe the geometry of the porous frame: porosity, tortuosity, flow resistivity, viscous characteristic length, and thermal characteristic length. The conventional laboratory methods for measuring these geometric properties are prone to errors and can be highly cumbersome.

In this work, an alternative method of determining the geometric properties of porous materials, based on an inverse acoustical technique, was investigated for materials used in aircraft noise-control applications. This technique is incorporated in commercial software codes, such as FOAM-X (ESI Group) and Comet Trim<sup>TM</sup> (Comet Acoustics), which require the absorption coefficient and/or transmission loss (TL) to be measured in Brüel and Kjær (or equivalent) standing wave tubes as inputs. The estimated geometric properties are required to define the porous material for complex vibroacoustic analysis in commercial code such as AutoSEA2 (ESI Group).

One of the goals of this work was to evaluate the accuracy of the estimated geometric properties. A closed-loop validation technique was previously developed where the absorption coefficient and transmission loss were predicted using AutoSEA2 and compared with the standing wave tube measurements. Good agreement between the measured and predicted absorption coefficient was observed for both foams and fibers. However, in the case of transmission loss, good agreement was observed for fibers but not for foams.

In order to eliminate inconsistencies, the existing validation loop was modified by incorporating Comet Trim<sup>TM</sup> inverse characterization software that took both the normal

incidence absorption coefficient and transmission loss as sequential inputs to estimate the geometric properties.

To complete the modified loop, sound absorption and transmission loss of porous materials was predicted using the performance analysis module in Comet  $Trim^{TM}$  and compared with the test results. In general, the absorption coefficient of most of the foams and fibers, prediction using both validation loops was in good correlation with the measured data. On the other hand, the correlation in normal incidence transmission loss was better using the modified loop.

In the process of investigating the repeatability of estimating the physical properties, previously measured porous material samples were re-measured for their absorption coefficient and transmission loss. A possible effect of sample aging was discovered and reported. As an alternate method to the forward TL calculation, a finite element model of the standing wave tube was also developed. This could be used to study the effect of boundary conditions on acoustic properties. Finally, individually validated samples were combined to develop optimized multilayer aircraft noise-control treatments and were experimentally demonstrated to produce excellent acoustical performance.

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#### **CHAPTER 1**

#### **INTRODUCTION**

#### **1.1 Background**

Aircraft interior noise, its generation, and propagation have been an increasingly important issue in the development of modern aircraft [1]. Much progress has been made in recent years in the eternal quest for quieter aircraft interiors. To reduce the noise level within an aircraft, a systems-level technique must be devised in the initial design phase that handles both the airborne acoustic energy and structure-borne vibration energy.

#### **1.2 Aircraft Cabin Noise**

Noise can be defined as unwanted sound and is among the most pervasive pollutants today. Noise in an aircraft cabin is caused by the acoustic and aero-acoustic excitation of the fuselage structure and covers a large surface area on the fuselage over a wide range of frequencies. Understanding the noise-source spectrum and its propagation modes, illustrated in Figure 1.1, are two important aspects of cabin noise-control and depend upon the type of aircraft.



Figure 1.1 Airborne and structure-borne sound

In a propeller aircraft, the exterior noise source arises from the propeller, exhaust, and engine vibration. Propeller noise consists of both broadband and low-frequency discrete components, but the pressure spectrum is dominated by discrete frequency components at the blade passage frequency  $f_b$  of the propeller and its subsequent harmonics [2]. Sound pressure levels on the fuselage of multiengine general aviation aircraft are typically on the order of 130 dB at the blade-passage frequency. High-speed propellers with supersonic tip speeds have sound pressure levels of 150 dB.

The advent of the jet engine changed the form of acoustic excitation of an aircraft's interior. Jet noise is generated by interaction between the turbulent exhaust of the jet or engine and the surrounding air. The noise field is non-stationary (varies with time) but can be considered stationary over short time periods. Jet noise spectra are broadband and peak at different frequencies for different locations in the near field. The airborne transmission of jet noise is mainly associated with aircraft that have wing-mounted engines, and it affects cabin regions aft of the engine exhaust, especially when the engines are mounted close to the fuselage [2].

The structure-borne noise caused by unbalance forces within the engine is another source of cabin noise. These forces induce vibrations into the aircraft structure that subsequently radiate acoustic energy into the interior cabin. The structurally excited noise components of the aircraft occur at the rotating frequencies of the fan and compressor and typically lie in the range of 75 Hz to 200 Hz [2].

Another source of cabin noise is from aerodynamic noise or, more specifically, turbulent-boundary layer noise. The dominant fluctuating pressures acting on an aircraft in high-speed flight are associated with the development of a turbulent boundary layer on the external surfaces of the vehicle. Similar fluctuating pressure fields are also encountered on other moving vehicles including automobiles, particularly around the windshield, and high-speed elevators. The pressure spectrum for a turbulent boundary layer is broadband and is a function of flow Mach number, flow velocity, and the boundary layer displacement thickness [2]. The graph in Figure 1.2 is representative of the noise distribution components for typical aircraft.



Figure 1.2 Noise distribution components for aircraft (NASA FS–1997–07–003–LeRC)

Aircraft utilize engine exhaust to augment lift generated by the wing and increase the effectiveness of the control surfaces. By doing so, the surfaces of the aircraft are exposed to high sound pressure levels that are a combination of acoustic and aero-acoustic pressures. Sound pressure levels increase to 165 dB on an airplane with upper surface blowing, and the structure heats up to a temperature of 500 °F to 700 °F [2].

Impulsive sounds, such as sonic booms generated by airplanes in supersonic flight and blast waves from explosions, can cause transient vibration of a structure. Supersonic jet exhausts result in the generation of additional broadband noise and discrete frequency screech because of shock wave interaction.

#### **1.3 Noise-Control Techniques**

Noise-control is the process of obtaining an acceptable noise environment by controlling the source, the path, the receiver, or all three. As illustrated in Figure 1.1, noise into the aircraft cabin propagates as airborne waves through the atmosphere and as structure-borne vibration waves through the fuselage structure. The sources of sound and vibration are many and coupled to one another in a complex manner. Their propagation paths are also of great interest and can be either direct or indirect. In order to reduce cabin noise, a thorough understanding and quantification of the sources and transmission paths are required.

Transfer path analysis (TPA) [www.lmsintl.com] is a powerful tool used to trace the flow of vibroacoustic energy from a source to the receiver through a set of known structure-borne and airborne pathways and quantify the dominant paths. A simplified TPA model representing the source, path, and receiver are illustrated Figure 1.3. A suitable control strategy must be applied to the dominant paths to reduce the overall sound pressure level inside the aircraft cabin. Noise-control can be categorized as active, passive, or active/passive.



Figure 1.3 Simplified source-path-receiver model

#### 1.3.1 Active Noise-control

Technological development of precision electronics over the past decades has enabled active control methods to emerge with prominence. Active noise-control is sound field modification and cancellation by electro-acoustical means. The two basic approaches to active control are active noise cancellation (ANC) and **a**ctive structural acoustic control (ASAC).

In ANC, the actuators are acoustic sources that produce an out-of-phase signal to "cancel" the disturbance. If the noise is caused by the vibration of a flexible structure, the coupling between the structural vibration and the radiated sound field can be used to advantage. This methodology is referred to as active structural acoustic control and was first introduced by C.R. Fuller. In ASAC, the actuators are vibration sources, which can modify the manner in which a structure vibrates, thereby altering the way it radiates noise. An example of active noise-control at the source is the use of the piezo-ceramic actuator to cancel discrete tonal noise from the aircraft engine.

#### **1.3.2** Passive Noise-control

Passive noise-control results from modification of the environment in which the sound sources operate and requires no input power to reduce noise and vibration. Passive control is inexpensive and easy to implement. However, performance is limited to the mid and high frequencies (>2000 Hz). Fortunately, active noise-control works best for low-frequency sound fields that are spatially simple. A combination of active and passive control is a common noise-control practice to combat noise over broad frequency range.

Passive noise-control systems use one or more of the following control types: absorbers and barriers for the control of airborne noise, and vibration isolators and vibration dampers for the control of structure-borne noise. The working and application of each type of noise-control material are illustrated in Figure 1.4 and will be described in Section 1.4.



Figure 1.4 Passive noise-controls [3]

#### 1.4 Materials used for Noise-control

Noise sources into the aircraft cabin can be distinguished as airborne and structureborne. Some airborne noise sources are caused by the turbulent boundary layer flow over the fuselage structure and the exhaust flow. Vibration from an unbalanced engine or wing flutter excitation of the fuselage structure causes structure-borne noise. A popular way to reduce the undesirable or harmful effects of sound is to block the sound transmission paths with passive acoustic material treatments.

Sound absorbers and sound barrier materials control airborne noise transmission paths. On the other hand, the structure-borne noise path is controlled by vibration isolators and structural dampers. As shown in Figure 1.4, sound absorbers reduce noise by the dissipation of sound energy to heat, whereas sound barriers control noise by a nondissipative mechanism (reflection). This figure also shows that the vibration isolator isolates the source by a non-dissipative mechanism and vibration dampers reduce the vibratory motion by dissipative mechanism. A detailed description of the mechanism behind each kind of passive control technique is discussed in this section.

#### **1.4.1 Sound Absorbers**

Freely propagating sound energy is dissipated by a combination of viscous and thermal mechanisms, also referred to as sound absorption. Sound absorbers are installed to dissipate sound energy and to minimize the reflection of sound. This is illustrated in Figure 1.4. The absorption coefficient  $\alpha$  is the index for measuring the sound absorption of a material and is a function of the frequency of the incident sound. It is defined as the ratio of energy absorbed by a sample to the energy incident upon the sample. Mathematically,

$$\alpha = 1 - \frac{I_{\text{Re flected}}}{I_{\text{Incident}}} \tag{1.1}$$

where

I<sub>Incident</sub> = intensity of sound incident on the surface

 $I_{\text{Reflected}}$  = intensity of sound reflected from the surface

It can be inferred from equation (1.1) that the absorption coefficient of materials ranges from 0 to 1. The absorption coefficient of commercial materials is specified in terms of a noise reduction coefficient (NRC) which is the average of absorption coefficients at 250 Hz, 500 Hz, 1,000 Hz, and 2,000 Hz. Based on the construction, sound absorbers are categorized as non-porous and porous absorbers. The characteristic frequency dependent behavior of the absorption coefficient is illustrated in Figure 1.5.

#### • Non Porous absorbers

Two types of non-porous absorbers are membrane absorber and resonators. Membrane absorbers are light, non-porous sheets or panels that are tuned to absorb sound waves over a specific frequency range. The resistance of the membrane to rapid flexing helps in sound absorption. Resonators or cavity absorbers are perforated material containing very tiny holes; a Helmholtz resonator is a classic example. The size of the opening, the length of the neck, and the volume of air trapped in the cavity govern the resonant frequency and hence the absorption.



Figure 1.5 Typical behavior of absorption versus frequency [4]

#### • Porous Absorbers

Porous absorbers are any material where sound propagation occurs in a network of interconnected pores in such a way that viscous and thermal effects cause acoustic energy to be dissipated as heat [5]. The propagation of sound is governed by physical properties of a porous medium, namely porosity, tortuosity, flow resistivity, viscous characteristic length, and thermal characteristic length. Absorptive treatment like glass fiber, mineral wool, or open-cell foam reduces reverberant sound. Porous absorbers are excellent thermal materials and usually not good sound barriers. The need for significant thickness compared to wavelength makes porous absorbers inefficient and not particularly useful at low frequency.

#### 1.4.2 Sound Barriers

As shown in Figure 1.4, a sound barrier blocks the transmission of airborne sound waves entering the aircraft cabin by interrupting the path of the sound wave and by acting as an acoustical reflector. An effective barrier material should be impervious, limp, and have high surface density. The effectiveness of the barrier material is specified with sound transmission loss (STL) measured in decibels (dB). STL is calculated in terms of sound transmission coefficient t<sub>c</sub>, which is the ratio of sound energy of a given frequency transmitted through a surface to that incident on it [8]. Mathematically,

$$STL = 10\log\frac{1}{t_c} \mathbf{d}B \tag{1.2}$$

Transmission loss is a function of the surface density of the panel, the frequency of the incident sound wave, and the angle of incidence. As illustrated in Figure 1.6, an infinite homogenous panel exhibits three control regions accounting for transmission loss phenomena across the entire frequency spectrum:



Figure 1.6 Transmission loss for an infinite homogeneous panel [7, 8]

In Region I the sound transmission loss is governed by the stiffness until the resonant frequency  $f_r$  (Hz) of the panel. In this region, the transmission loss decreases at the rate of 6 dB per octave increase in frequency. Beyond this point, at slightly higher frequencies, the resonance of the panel begins to control its transmission behavior and STL is determined by the panel's damping ratio.

In Region II, the STL is controlled primarily by the mass and not by damping and stiffness of the panel. Doubling the mass or the frequency results in a 6 dB/octave increase in transmission loss. This is known as the mass law of sound transmission. In this region, the normal incidence transmission loss can be approximated by

$$TL_0 = 10\log\left[1 + \left(\frac{\omega\rho_s}{2\rho c}\right)^2\right] dB$$
(1.3)

where

 $\omega$  = sound frequency (rad/sec)

 $\rho c$  = characteristic impedance of medium

 $\rho_s$  = mass of panel per unit surface area

In Region III, at a particular frequency of incident sound wave, the wavelength of sound in air coincides with the structural bending wavelength, and the transmission loss theoretically goes to zero. This is called the coincidence effect, and represents a coupling of structural vibration to acoustic radiation of the panel into the air. According to Fahy [6], for a given angle of incidence ( $\varphi$ ), there is a unique coincidence frequency ( $\omega_{co}$ ), which is a function of mass of the panel (m), bending stiffness (D), and the speed of sound in the panel (c). The coincidence frequency is given by

$$\omega_{co} = \left(\frac{m}{D}\right)^{1/2} \times \left(\frac{c}{\sin\phi}\right)^2 \tag{1.4}$$

The lowest frequency of the coincidence phenomenon occurs for a normally incident sound wave ( $\varphi$ =90°). This frequency is called critical frequency f<sub>c</sub> (Hz). Beyond this frequency, efficient radiation of sound occurs. Hence, to obtain maximum STL, an attempt should be made to obtain resonant frequencies preferably well below the audible range and the critical frequency preferably well above the audible range.

#### **1.4.3** Vibration Isolators

Vibration isolators are flexible components used to reduce transmitted vibratory motion or forces from one structure to another. Transmissibility, defined as the ratio of the transmitted force to the disturbing force quantifies performance of the isolator. Transmissibility is a function of the damping factor of the system ( $\xi$ ), natural frequency of the vibrating system ( $\omega_n$ ), and the frequency of the forcing function ( $\omega$ ).

Vibration isolators such as elastomeric mounts, bushings, and pads are used in aircraft engine mounts and also between structural joints. The frequency range over which the isolator must be effective is an important design criterion. Isolation of noise in the frequency range above 250 Hz requires a relatively stiff, low-deflection mount. Isolation of very low-vibration frequencies, such as the fundamental rotation speed of a jet engine at 125 Hz, requires much greater deflection capability from the mount [9].

#### **1.4.4 Structural Dampers**

Structural damping provides a means for eliminating vibration energy by converting it to heat. As illustrated in Figure 1.4, damping reduces the amplitude of oscillation. Damping materials are typically applied directly to the aircraft skin to reduce the effects of structural vibration. They are also applied to the surfaces of the interior trim structure to reduce the effects of induced airborne vibration and engine vibration. Another place of application is the windows of an aircraft. Aircraft windows are a significant noise path that contributes to interior noise. Damping treatments applied on windshield and forward cabin windows offers good potential for cabin noise reduction [10].

#### **1.5 Acoustic Material Testing**

Generally, the terms acoustic materials or noise-control materials refer to sound absorbers and sound barriers. The acoustic performance of these materials is better at some frequencies than others. For this reason, a frequency performance evaluation of the noise-control material is almost always required. Sound absorbers and barriers are evaluated in terms of the frequency-dependent acoustical indicators, namely, absorption coefficient and transmission loss, respectively. Further, acoustic material testing assists in selecting the most adequate treatment and provides input information for vibroacoustic tools used for the prediction of sound fields in acoustic cavities. The standard test methods to evaluate the frequency-dependent absorption coefficients and transmission loss of noise-control materials are as follows:

- ASTM C384, Standard Test Method for Impedance and Absorption of Acoustical Materials by the Impedance Tube Method.
- ASTM E1050, Standard Test Method for Impedance and Absorption of Acoustical Materials Using a tube, two microphones and a digital frequency analysis system in an impedance tube.
- ISO 10534, Acoustics—Determination of sound absorption coefficient and impedance in impedance tubes—Part 1: Method using standing wave ratio, Part 2: Transfer-function method.

- ASTM C423, Standard Test Method for Sound Absorption and Sound Absorption Coefficients by the Reverberation Room Method.
- ASTM E90, Standard Test Method for Laboratory Measurement of Airborne Sound Transmission Loss of Building Partitions and Elements.
- ISO 140-3, Acoustics—Measurement of sound insulation in buildings and of building elements—Part 3: Laboratory measurements of airborne sound insulation of building elements.
- ASTM E336-05, Standard Test Method for Measurement of Airborne Sound Insulation in Buildings.

#### **1.6 Problem Statement**

Porous materials such as foams and fibers are widely used as passive noise-control materials. The study of acoustical performance of these materials is important to efficiently combat aircraft cabin noise. To address this issue, the Department of Mechanical Engineering at Wichita State University and the National Institute for Aviation Research (NIAR), in collaboration with the aerospace industries in Kansas, developed a broad acoustic database of commercially available noise-control materials that are used in aircraft interior applications [37].

The objective of their study was to measure the normal incidence acoustical performance, namely, absorption coefficient and transmission loss of various porous noise-control materials using a Brüel and Kjær impedance tube kit Type 4206 and transmission loss tube kit Type 4206T, respectively. The measurements were recorded in the frequency range of 50 Hz to 6,400 Hz, which is the focus of interest in the aerospace industry.

The acoustical performance of these materials is dependent on geometric physical and mechanical properties. It is common practice in the aerospace industry to use a statistical energy analysis-based code called AutoSEA2 [32, see Appendix] to predict noise levels in an aircraft cabin. The geometric physical properties along with the elastic properties of the porous noise-control materials are required to predict the noise levels.

A software code called FOAM-X [32, see Appendix], is based on an inverse acoustical characterization algorithm takes the impedance tube test data as input to estimate the physical properties of porous materials. A closed-loop validation technique was developed in order to validate the physical properties estimated in FOAM-X. In that process, the estimated physical properties in conjunction with the measured elastic properties of the porous materials were used as inputs in AutoSEA2 to predict the random incidence absorption coefficient and transmission loss of each material. The predicted data were compared to the measured test results to complete the validation loop.

It was observed that measured absorption coefficients of foams and fibers were in close correlation with the predicted data. In the case of transmission loss, a good correlation between the measured and predicted data was observed for fibers (see section 4.2.1). However, for most of the foam samples, a poor correlation was observed in transmission loss over a wide range of frequencies (see section 4.2.2).

#### 1.7 Study Objective

One possible method to address this issue was to modify the existing validation loop to include the effects of elastic frame porous material. To accomplish this, an inverse acoustical tool called Comet Trim<sup>TM</sup> [41, see Appendix] was implemented to estimate the physical properties of porous materials. Comet Trim<sup>TM</sup> used both absorption coefficient

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and normal incidence transmission loss test data for inverse characterization. Further, the performance analysis module in Comet Trim<sup>TM</sup> was utilized to predict the normal incidence acoustical performance of porous materials, which were compared with the test results to complete the modified validation loop.

The key motivation for this study was to examine the precision of inverse acoustical techniques and develop confidence in the estimated physical properties that are often used in complex vibroacoustic analysis such as those used in an aircraft cabin system. Hence, the objectives of this work were as follows:

- 1. To investigate the closed-loop validation procedure that was previously developed and explore reasons for poor correlation between the measured and predicted acoustical properties of materials based on porous material models.
- 2. To evaluate the performance of the modified validation loop as applied to characterization of porous materials having rigid, limp and elastic frames.
- 3. To study the effect of potential sample aging on acoustical measurements and the impact on estimated physical properties of the porous material.
- 4. To investigate the acoustical performance of an optimized multilayer noisecontrol treatment that can be used to design quieter aircraft interiors.

#### **CHAPTER 2**

#### LITERATURE REVIEW

#### 2.1 Overview

Porous materials such as plastic foams, or fibrous, and granular materials are widely used in several noise-control applications. These materials are frequently used in the automotive and aeronautics industries. A good understanding of the vibroacoustic behavior of porous materials is an integral part of interior noise-control.

Air-saturated porous materials are inherently two-phased, consisting of an elastic solid frame surrounded by air that can flow through the pore channels. Sound wave propagation in a porous material can be very complicated as acoustic energy is transmitted through elastic frame and air pores. These materials are known for their ability to dissipate the energy of sound wave propagating though them by altering the speed of propagation of plane waves from its free wave value and also by attenuating the wave as it propagates.

#### 2.1.1 Porous Material Properties

In order to understand the research involved in vibroacoustics of porous material, it is important to review the properties governing the propagation of sound in porous media. The sound propagation in porous media is governed by properties that depend on the geometry of the porous material, the property of air in the pores, and the mechanical properties of the solid skeleton. The detailed description of the meaning and the physical origin of these parameters are outline here [5].

#### • Macroscopic Physical Properties

Five parameters describe the geometry of the porous frame and define the complexity in the propagation of sound in porous media. They are as follows:

- Porosity  $\varphi$  is a physical property defined as the ratio between the air volume in open pores V<sub>a</sub> and the total volume V<sub>t</sub>, i.e.,  $\varphi = V_a/V_t$ . Porosities are typically grater than 0.95. Good sound absorbers have porosity nearly equal to one.
- The tortuosity  $\alpha_{\infty}$  is a physical property that describes how well the porous material prevents direct flow through the medium. It relates to the average fluid path length through the material normalized by thickness of the material. Sound energy dissipation is high in a material with high tortuosity. Tortuosity results in inertial coupling between solid and fluid phases.
- The flow resistivity  $\sigma$  is the pressure drop required to force a unit flow through the material and expresses the viscous losses to the propagating sound waves inside the porous material. For a wide range of porous materials, the flow resistivity is the most important parameter for sound absorption. The unit of flow resistance is Ns/m<sup>4</sup> or rayls/m.
- The viscous characteristic length  $\Lambda$  (µm) is the average macroscopic dimension of the cells related to viscous losses and may be seen as an average radius of the smaller pores of a porous aggregate.
- The thermal characteristic length  $\Lambda$ ' ( $\mu$ m) is the average macroscopic dimension of the cells related to thermal losses and may be seen as an average radius of the larger pores of a porous aggregate.

#### • Air Properties

The air in the pores can be considered an ideal gas. The density of air  $\rho_0$ , speed of sound c in the air pore, viscosity of air  $\eta$  (pa.s), Prandtl number Pr and specific heat ratio  $\gamma$  define the air properties.

#### • Mechanical Properties of Frame

Following the Biot theory [12], the frame of an isotropic open-cell elastic porous material is defined by three macroscopic elastic properties: the loss factor and two of the following properties: shear modulus G, Young's modulus E, or Poisson's ratio v, together with the density  $\rho$  of the solid frame. These properties are usually complex and frequency-dependent due to the viscosity of the frame.

#### 2.1.2 Plane Waves in Porous Media

According to Fahy [6], the modified equation for plane wave sound propagation in air contained within rigid porous materials is given by

$$\frac{\partial^2 p}{\partial x^2} - \left(\frac{\alpha_{\infty} \rho_0}{k_{eff}}\right) * \frac{\partial^2 p}{\partial t^2} - \left(\frac{\sigma \phi}{k_{eff}}\right) * \frac{\partial p}{\partial t} = 0$$
(2.1)

where

p = sound pressure within the cylindrical pores of material

 $\rho_0$  = density of air at standard temperature and pressure

 $k_{eff}$  = effective bulk modulus of air

$$\alpha_{\infty}$$
 = tortuosity

$$\varphi = \text{porosity}$$

 $\sigma$  = flow resistivity

The acoustical behavior of homogenous porous material can be also be determined from its fundamental quantities: the complex wave number and characteristic impedance [6]. These quantities are the harmonic solution of the modified plane wave equation and can be used to determine the acoustical performance of porous materials, namely absorption coefficient, surface impedance, and transmission loss.

The wave number k is seen as the spatial analog of frequency. As illustrated in Figure 2.1, it is a measure of how often the wave repeats per unit distance. The system of units (SI) unit of wave number is reciprocal meters (m<sup>-1</sup>). The wave number k' in the porous material is a complex term and takes the form

$$k' = \gamma - j\beta \tag{2.2}$$

The real part of the complex wave number is called the propagation constant and represented as  $\gamma$ , and the imaginary part of the complex wave number is referred to as the attenuation constant  $\beta$ , since it is responsible for attenuation of sound waves. It is noted that the amplitude of the sound waves decreases with length inside a porous material; also, the wave speed is less than the adiabatic value. The complex wave number is a property of the material itself.



Figure 2.1 Pictorial representation of wave number [6]

The characteristic impedance  $Z_c$  is defined as the ratio of amplitude of sound pressure to the associated particle velocity at the surface of a material of infinite depth on which a plane sound wave is falling perpendicular to the surface. The characteristic impedance of the porous material is a complex term indicating that the pressure and velocity are not in phase. Song and Bolton [11] developed an experimental method for determining the complex wave number and characteristic impedance of isotropic porous materials in a modified standing wave tube using a transfer matrix formulation.

#### 2.2 Modeling Porous Media

Extensive work has been done in the past to model the vibroacoustic behavior of porous materials. The most significant contribution to the investigation of wave propagation in air saturated porous materials was proposed by Biot [12, 13, 14]. Later researchers tailored Biot's theory to a range of applications and completed the description of porous materials. The mechanism of sound propagation in porous materials depends on the type of frame. Based on the motion of the frame, porous materials were classified as rigid foam, limp foam, fibrous foam, and elastic foam.

#### 2.2.1 Delany and Bazley Model

Delany and Bazley [15] developed a semi-empirical model to describe sound propagation in fibrous materials with porosity close to 1. They came up with a single parameter empirical expression that permits the estimation of characteristic impedance  $Z_c$ and propagation constant  $\gamma$  of porous materials, if a subsidiary measure of the flow resistivity can be made. Subsequently, this model was improved by Morse and Ingard [16] and Attenborough [17]. By using regression analysis, the empirical laws to estimate the propagation constant  $\gamma$  and characteristic impedance  $Z_c$  were derived from measurement data as

$$\gamma = j \frac{\omega}{c} \left[ 1 + 0.0978 \left( \frac{\rho_0 f}{\sigma} \right)^{-0.700} - j0.189 \left( \frac{\rho_0 f}{\sigma} \right)^{-0.595} \right]$$
(2.3)

$$Z_{c} = j\frac{\omega}{c} \left[ 1 + 0.0571 \left(\frac{\rho_{0}f}{\sigma}\right)^{-0.754} - j0.087 \left(\frac{\rho_{0}f}{\sigma}\right)^{-0.732} \right]$$
(2.4)

where

- $\rho_o$  = density of air at standard temperature and pressure
- $\sigma$  = flow resistivity
- f = frequency

 $\omega = 2\pi f$ 

#### 2.2.2 Rigid Porous Model

In a rigid-frame porous material, the solid frame is motionless and the frame bulk modulus is significantly greater than that of air. This type of material can be modeled as an effective fluid using the wave equation for a fluid with complex effective fluid density and complex effective bulk modulus. While propagating through the motionless porous network, sound waves are attenuated due to viscous and thermal dissipation mechanisms. The dynamic density accounts for the viscous losses and the dynamic bulk compression modulus for the thermal losses. The effective density  $\rho_{eff}$  in terms of the dynamic tortuosity  $\alpha$  ( $\omega$ ) is given by

$$\rho_{\text{eff}} = \alpha_{\infty} \left( 1 + \frac{\sigma \phi}{j \omega \rho_0 \alpha_{\infty}} G(\omega) \right) \rho_0$$
(2.5)

where

 $G(\omega)$  = viscous correction factor

Johnson et al. [18] worked out an analytical model G ( $\omega$ ) that links the high- and lowfrequency behavior of the effective density and acts as a correction factor for viscosity in the pores for which the flow differs from Poiseuille flow with increasing frequency. It is expressed as

$$G(\omega) = \sqrt{1 + j \frac{4\alpha_{\infty}^2 \eta \rho_0 \omega}{\sigma^2 \Lambda^2 \phi^2}}$$
(2.6)

where

 $\eta$  = viscosity of air

Champoux and Allard [19] introduced a similar function G' ( $\omega$ ) in the dynamic bulk compression modulus to account for the thermal interactions in the porous material due to the thermal exchanges between the acoustic wave front, which propagates in the fluid phase and the solid frame. The expression for the bulk modulus is

$$k_{eff} = \frac{\gamma p_0}{\gamma - (\gamma - 1) \left[ 1 - j \frac{8\eta}{\Lambda^2 \operatorname{Pr}^2 \omega \rho_0} G'(\omega) \right]}$$
(2.7)

where

$$G'(\omega) = \sqrt{1 + j \frac{\Lambda^2 \operatorname{Pr}^2 \omega \rho_0}{16\eta}}$$

Pr = prandtl number of air

 $\gamma$  = specific heat ratio of air

#### 2.2.3 Limp Porous Model

A material is considered limp if the frame of a porous material has no stiffness but its inertia effects have a significant effect on acoustic energy propagation. In a limp porous material the solid phase moves and the frame bulk modulus is significantly less than that of air. Sound propagation through a limp material is described by Goransson [20]. The acoustic wave propagates at constant temperature conditions inside the porous material, and the speed of sound is given the isothermal value. The bulk mass density of the material is given by

$$M = \phi \rho_0 + (1 - \phi) \rho_s \tag{2.8}$$

where

 $\rho_s$  = density of the frame material,

The limp-frame model is better suited for materials having a low Young's modulus like glass wool and can be described by an effective density and effective bulk modulus derived using fluid volume displacements.

#### 2.2.4 Elastic Porous Model

The fourth type of porous-material model is the elastic frame model. In this model, the sound wave propagation occurs through air in the pores and through the solid frame. Hence, both phases are in dynamic motion. The description of sound propagation is more complicated than the rigid-frame or limp-frame porous material. Biot developed the stress and strain relationship equations for the structural and fluid partition to completely describe the phenomenon of sound propagation [12,13,14].

Following Biot's theory, three waves may propagate simultaneously in an airsaturated, open-cell poroelastic material—two compression waves and one shear wave. It has been noted that for many materials having an elastic frame and set on a rigid floor, the frame can be almost motionless for large ranges of acoustical frequencies, thus allowing the use of the models worked out for rigid-framed materials. An elaborate study on sound propagation in elastic framed porous materials is presented by Allard [5].
### **2.3 Laboratory Characterization of Porous Materials**

Several research studies have been conducted to relate the macroscopic physical properties of porous materials to their effective properties [18, 19]. With knowledge of the five geometric parameters and the material models, the calculation of desired acoustical indicators as well as optimization of several noise-control treatments can be done rapidly. Several different laboratory techniques are involved to measure these physical properties.

Porosity is one of several important parameters required by acoustical theory to characterize a porous material. Champoux et al. [21] devised a technique to measure the porosity  $\varphi$  of a material by measuring the isothermal pressure change in a closed volume containing the sample for a known change in the volume using a porosimeter. The apparatus used for measuring the porosity is illustrated in Figure 2.2.



Figure 2.2 Apparatus used for measuring porosity [21]

Panneton and Gros [22] proposed another technique to measure the open porosity of materials when the mass density of the solid phase constituent is unknown. This method is based on the measurement of the apparent and true masses of a porous solid, where a

missing mass was found and related to the volume of the solid phase through Archimedes' principle.

Flow resistivity  $\sigma$  is another important parameter required by acoustical theory to characterize porous materials like plastic foams and fibrous or granular materials. Stinson and Diagle [23] developed an electronic system to measure the flow resistance of porous materials using a variable-capacitance pressure transducer, as illustrated Figure 2.3.



Figure 2.3 System for the measurement of flow resistance [23]

Fellah et al. [24] proposed a method for measuring flow resistivity of porous materials having a rigid frame using acoustic reflectivity method. It was found that flow resistivity has significant sensitivity on reflected waves at low frequencies. Hence, an inverse scattering problem for waves reflected by a slab of air-saturated porous material was solved to estimate the flow resistivity. As an extension to this work, Fellah et al. [25] proposed an acoustic transmissivity method to measure flow resistivity of porous materials having a rigid frame. Henry et al. [26] used the standard Brunauer, Emmett, and Teller (BET) method that is used for measuring the bulk modulus of porous materials to evaluate the characteristic dimensions  $\Lambda$  and  $\Lambda$ ' of porous materials.

Alternative methods, based on acoustical ultrasound measurements, have been developed to circumvent the difficulties inherent in the direct characterization of porous materials. Moussatov et al. [27] developed an ultrasonic method to estimate tortuosity, the viscous and thermal characteristic lengths of porous materials saturated by air. This method is based on the evaluation of speed of sound and the attenuation inside the material when the static pressure of the gas saturating the material is changed. Fellah et al. [28] proposed an ultrasonic method to characterize air-saturated porous materials by solving the inverse problem using experimental data of both reflected and transmitted waves to determine simultaneously all the physical parameters intervening in the propagation.

The elastic properties of a porous material can have an important effect on the acoustical performance of noise-control materials. Langlois et al. [29] proposed a quasistatic compression test method to determine the elastic properties of isotropic poroelastic materials. Polynomial equations that related the compression stiffness, Young's modulus, Poisson's ratio, and shape factor were developed using axisymmetric finite element simulations under static compression. The compression stiffness was measured using a compression test setup and consisted of a disk-shaped poroelastic sample sandwiched between two rigid plates, as illustrated in Figure 2.4. An accelerometer was fixed on the bottom plate, and a force transducer was mounted on the top plate. A shaker was used to excite the bottom plate. Using an FFT analyzer, the transfer function mechanical impedance was determined. The measured compression stiffness is the real part of the transfer function. The Young's modulus and Poisson's ratio of the poroelastic material were obtained from the stiffness and the polynomial relations.



Figure 2.4 Measurement setup for compression test [29]

# 2.4 Inverse Acoustical Characterization

The conventional laboratory method for measuring tortuosity and characteristic lengths of the porous materials was found to be cumbersome and costly, and may yield large errors. Atalla and Panneton [30] developed an alternative acoustical method based on an inverse characterization procedure to estimate the geometric physical properties of open-cell porous media using the impedance tube measurements described in ASTM E1050 (see Introduction). In this method, the frame of the porous sample was assumed to be rigid under acoustical excitation. Hence, the porous material was described as an equivalent fluid with effective density and bulk modulus [18, 19].

The solution involved fitting data to a porous material model that depended on independent adjustable parameters of porosity, tortuosity, flow resistivity, and viscous and thermal characteristic lengths. A multidimensional optimization merit function was developed to match the observed measurements and predict acoustical quantities obtained from the iteratively estimated physical properties. The merit function was subjected to the following bounds on adjustable parameters during the optimization process:

- tortuosity  $\alpha_{\infty}$  between 1 and 4
- $\Lambda \leq \Lambda'$
- $1 \mu m \leq (\Lambda \text{ and } \Lambda') \leq 2000 \mu m$
- $1000 \text{ Ns/m}^{-4} \le \sigma \le 5*10^6 \text{ Ns/m}^{-4}$
- $0.7 \le \phi \le 1$

It was found that the minimization of the merit function leads to an accurate determination of the intrinsic physical properties when the three zones (I, II, and III) of the absorption coefficient curve, as shown in Figure 2.5, are covered. A powerful proprietary minimization algorithm was designed based on the physics of the studied problem to obtain a unique solution in terms of the geometric parameters.



Figure 2.5 Frequency zones of a typical sound absorption curve [32]

# 2.5 Validation Loop

The identification algorithm developed by Atalla and Panneton was incorporated in the software code FOAM-X [32]. Sharma et al. [31] developed a closed-loop process, as illustrated in Figure 2.6, to validate the physical properties of porous materials estimated using FOAM-X and impedance tube test data.



Figure 2.6 Closed-loop validation process [31]

The impedance tube test data was input into FOAM-X to estimate the macroscopic physical properties of the porous materials, which include porosity, flow resistivity, tortuosity, viscous characteristic length, and thermal characteristic length. The computed physical properties in conjunction with the elastic properties were used as input for the AutoSEA2 model to predict the absorption coefficient and transmission loss of the porous material. The AutoSEA2 [32] predicted results were compared with the actual test data to complete the validation loop. A good correlation was observed between the predicted and measured absorption coefficients of foams and fibers. However, when comparing the predicted and measured transmission loss of each material, a close match was observed for fibers, but poor correlation was observed for most foam samples across a wide frequency range (see section 4.2).

#### 2.6 Effect of Boundary Condition on Acoustical Measurements

One of the main uncertainties regarding the accuracy of acoustical measurements in a tube may arise from the potentially poor fit between the sample and tube diameters: the sample's diameter can either be smaller or larger than the tube inner diameter.

Song and Bolton [33] examined the effects of boundary condition on the measured normal incident acoustical properties of absorption coefficient and transmission loss in a standing wave tube. The measurements revealed that the constraint of the samples around their edges significantly altered their transmission loss and other acoustical properties at low frequencies. The major effect of the degree of edge constraint was the shift in first resonant frequency of the sample to higher frequency, consequently enhancing the lowfrequency transmission loss and reducing the magnitude of the absorption coefficient.

Dependence of the absorption coefficient and transmission loss on the material parameters of the porous sample was also studied. It was found that shear modulus controls the location of resonance features in the spectrums, and this phenomenon can be attributed to the shearing resonance of the solid phase of the porous material. The loss factor controls the width and height or depth of the features in the spectrum. Finally, it was noted that the flow resistivity has a significant impact on transmission loss, both below and above the frequency of the first-shearing mode of the sample. Similarly, it was found that flow resistivity has a significant impact on the absorption coefficient over the complete frequency range considered.

Cummings [34] performed a study in which he examined theoretically and experimentally the effect of air gaps between a sound-absorbing material and the impedance tube wall, and found that air gaps had a less noticeable effect on the measured absorption coefficient at low frequencies than at high frequencies It was also noted that air gaps tended to have a greater significance in the case of media having relatively high flow resistivities.

#### **CHAPTER 3**

# **METHODOLOGY**

#### 3.1 Background

Measurements of the acoustical performance of porous noise-control materials can be accomplished with normal incidence or random incidence sound waves. Although random incidence more nearly approximates conditions of actual use, measurements at normal incidence are much easier and valuable for ranking acoustical materials. The normal incidence measurement technique using impedance and transmission loss tube was adopted in this thesis work and discussed in sections 3.2 and 3.3.

Commercial software such as AutoSEA2 requires the input of macroscopic physical properties for modeling the vibroacoustic behavior of porous materials. Hence, an inverse characterization method using Foam-X was utilized to estimate the physical properties of porous materials. A validation loop was developed in a previous study [37] in order to build confidence on the inverse characterization method and discussed in section 3.4.

The limitations in the original validation loop created inconsistencies in the correlation of the acoustical performance of porous materials. In order to overcome these limitations, a modified validation loop using Comet Trim<sup>TM</sup> was proposed as a part of the current study and discussed in section 3.5.

# **3.2 Acoustical Measurements: Theoretical**

# 3.2.1 Absorption Coefficient—Transfer Function Method

There are two well-established methods of measuring the normal-incidence absorption coefficient and other properties in an impedance tube terminated by a sample of the material under test. The first method is based on the measurement of the standing wave ratio frequency by frequency standardized to ASTM C384 and ISO 10534-1 and described in section 1.5. The second method is based on pressure measurements at two microphone locations and excitation with a broadband signal, and involves the measurement of the transfer function between two microphone signals. This method is standardized to ASTM E1050 and ISO 10534-2.

The transfer function technique of measuring normal-incidence absorption coefficients was developed by Chung and Blaser [35, 36]. Figure 3.1 illustrates the theory underlying this technique, which involves the decomposition of a broadband stationary random signal (generated by an acoustic driver) into its incident and reflected components using a simple transfer-function relation between the acoustic pressures at two locations on the tube wall.



Figure 3.1 Two microphone transfer function method [37]

This technique required a two-channel FFT analyzer and two closely spaced microphones placed along the wall of the tube. The microphones were previously calibrated for phase and gain matching with a proper microphone switching technique for the validity of the transfer function method. In this method, the measurement is possible only under normal incidence conditions ( $\theta = 0$ ), on non-diffusing surfaces, and inside a

tube of proper diameter and length, as per ASTM E1050, to ensure that both the incident and reflected waves are plane and in the same direction. The usable frequency range depends on the diameter of the tube and the spacing between the microphone positions.

The schematic of this two-microphone method, as illustrated in Figure 3.1, shows that the sound source (loudspeaker) is mounted at one end of the impedance tube, and a sample of the material is placed at the other end on an adjustable rigid plunger. The loudspeaker generates broadband, stationary, random sound waves that propagate as plane waves in the tube. As the incident plane wave hits the test sample, some of the incident sound energy is absorbed by the sample and the rest is reflected off the surface. Because of the loss in sound energy due to absorption, the incident and reflected plane waves are out of phase. Consequently, a standing-wave interference pattern is produced due to the superposition of forward- and backward-traveling waves inside the tube.

A pair of microphones is mounted flush with the inner wall of the tube near the sample end of the tube to measure sound pressure at two fixed locations. A two-channel digital frequency spectrum analyzer is used to obtain the transfer function (frequency response function) between the microphones. From the transfer function  $H_{12}$ , the pressure reflection coefficient R, absorption coefficient  $\alpha$ , and the surface impedance z of the material can be determined as

$$R = \frac{H_{12}e^{jkX_1} - e^{jkX_2}}{e^{-jkX_2} - H_{12}e^{-jkX_1}}$$
(3.1)

$$\alpha = 1 - \left| R \right|^2 \tag{3.2}$$

$$z = \rho_0 c \frac{1+R}{1-R}$$
(3.3)

where

X<sub>1</sub>= distance from the sample surface to microphone 1 X<sub>2</sub>= distance from the sample surface to microphone 2  $H_{12}=\frac{P_2}{P_1}$ , transfer function between the microphones P<sub>1</sub>= pressure measured by microphone 1 (reference channel) P<sub>2</sub>= pressure measured by microphone 2 k= wave number

 $\rho_o c$ = characteristic impedance of air

# 3.2.2 Transmission Loss--Transfer Matrix Method

The four microphone transfer function method employs a modified standing wave tube having a downstream tube section (receiving tube) mounted with a pair of microphones and a changeable anechoic termination. A loudspeaker is mounted at one end of the source tube, and a test sample is installed in a sample holder connecting the source and receiving tubes, as shown in Figure 3.2.



Figure 3.2 Schematic diagram of the standing wave tube [38]

The measurement of normal incident sound transmission loss of a plug of noisecontrol material is based on either wave field decomposition methods [35, 36] or the transfer matrix method [11]. Both techniques use a four-microphone standing wave tube and an FFT analyzer. The two-load [39] and two-source [40] methods are the most commonly used transfer matrix approaches. The transfer matrix defines a relationship between the complex sound pressures and the complex normal acoustic particle velocities on the two faces of a porous layer extending from x=0 to x=d as

$$\begin{bmatrix} p \\ v \end{bmatrix}_{x=0} = \begin{bmatrix} T_{11} & T_{12} \\ T_{21} & T_{22} \end{bmatrix} \begin{bmatrix} p \\ v \end{bmatrix}_{x=d}$$
(3.4)

When the plane waves generated by the loudspeaker hit the sample in the holder, part of the wave is reflected back into the source tube, part of it gets absorbed by the material, and some passes through the material to the receiving tube. The complex pressure measured at each microphone location is the sum of the positive X-going and negative Xgoing going plane wave. If the pressure amplitude of the positive X-going plane wave upstream is A and the negative X-going plane wave is B then the complex pressures at the two microphone locations upstream will be

$$p_{i} = Ae^{-jkx_{i}} + Be^{jkx_{i}}\Big|_{i=1,2}$$
(3.5a)

Similarly, if the pressure amplitude of the positive-going plane wave downstream is C and the amplitude of the negative-going plane wave downstream is D, then the complex pressures at the downstream microphone locations will be

$$p_{i} = Ce^{-jkx_{i}} + De^{jkx_{i}}\Big|_{i=3,4}$$
(3.5b)

The upstream and downstream pressures and particle velocities on the two surfaces  $(P_{x=0,d} \text{ and } V_{x=0,d})$  of the porous layer are determined from the measured pressure amplitude of the plane wave along with the characteristic impedance of air  $\rho_0 c$ . It is of interest to solve for the elements of the transfer matrix as it can be related to acoustic

properties such as characteristic impedance, wave number, absorption coefficient, and transmission loss of the material layer under test. Equation (3.4) represents two equations and four unknowns:  $T_{11}$ ,  $T_{12}$ ,  $T_{21}$ , and  $T_{22}$ . Two additional equations are required in order to be able to solve for the elements of the transfer matrix. These equations can be obtained through two methods: the two-load method and one-load method. They are described as follows:

#### Two-Load Method

In this method [38], two additional independent equations can be generated by making another measurement using a different impedance condition at the downstream termination of the standing wave tube, as shown in Figure 3.2. If the superscripts a and b represent the two termination conditions, then the elements of the transfer matrix can be determined by solving

$$\begin{bmatrix} p^{a} & p^{b} \\ v^{a} & v^{b} \end{bmatrix}_{x=0} = \begin{bmatrix} T_{11} & T_{12} \\ T_{21} & T_{22} \end{bmatrix} \begin{bmatrix} p^{a} & p^{b} \\ v^{a} & v^{b} \end{bmatrix}_{x=d}$$
(3.6)

# • One-Load Method

Song and Bolton [11] reported that, for symmetric homogeneous and isotropic porous material, the plane wave reflection and transmission coefficients from the two surfaces of the sample are the same, i.e.,  $T_{11}=T_{22}$ . Further, by applying the principle of reciprocity, the determinant of the transfer matrix in equation (3.4) turns out to be unity. The two additional equations produced are

$$T_{11}T_{22} - T_{12}T_{21} = 1 \tag{3.7}$$

$$T_{11} = T_{22} \tag{3.8}$$

The elements of the transfer matrix in equation (3.6) can be solved to obtain the transmission loss according to the two-load method. The one-load method can be solved using the matrix equation (3.4) along with the reciprocity and symmetric conditions in equations (3.7) and (3.8), respectively. In both methods, a perfect anechoic termination is used for the transmission loss calculation. In that case, the amplitude of the plane wave reflection at the downstream is zero, i.e., D=0, then the normal incident sound transmission loss for both the methods is calculated as

$$TL_{n}(\omega) = 10\log\left(\frac{1}{4}\left|T_{11} + \frac{T_{12}}{\rho_{0}c} + \rho_{0}cT_{21} + T_{22}\right|^{2}\right)$$
(3.9)

Equation (3.9) is incorporated in PULSE acoustic material testing module to compute the normal incidence sound transmission loss.

#### **3.3 Acoustical Measurements: Experimental**

#### 3.3.1 Hardware Setup

The Brüel and Kjær Impedance Tube kit Type 4206 and Transmission Loss Tube kit Type 4206T were used to measure the normal incidence sound absorption coefficient and transmission loss of noise-control materials, respectively. The complete acoustic material test system consisted of a large tube 100 mm in diameter, a small tube 29 mm in diameter, two sample holders 29 mm and 100 mm in diameter, two extension tubes 29 mm and 100 mm in diameter, two extension tubes 29 mm and 100 mm in diameter, four <sup>1</sup>/<sub>4</sub>-inch microphones Type 4187, and four <sup>1</sup>/<sub>4</sub>-inch preamplifiers Type 2633.

The usable frequency range of measurement was dependent on the diameter of the tube and the spacing between microphone positions. The components were assembled in two different setups to measure all acoustical parameters over the entire frequency range of 50 Hz to 6.4 kHz. A large-tube setup was used to measure the acoustical parameters in

the frequency range of 50 Hz to 1.6 kHz, and a small-tube setup was used to measure the acoustical parameters in the frequency range of 500 Hz to 6.4 kHz. The microphones were closely spaced in the small-tube setup because the wavelength of the propagating plane waves was very small. On the other hand, microphones were comparatively widely spaced in the large-tube setup.

The hardware flow of the PULSE system, shown in Figure 3.3, consisted of a PULSE multianalyzer, front-end, 4/2 input/output module data acquisition system Type 3109 built on frame Type 3560C, a power amplifier Type 2706, a personal computer (PC) loaded with the PULSE Labshop software version 11.1, and the impedance tube/transmission loss tube system. PULSE Material Testing software module Type 7758 was the interface to measure the acoustical performance of noise-control materials. It worked in conjunction with the impedance tube and transmission loss tube system.



Figure 3.3 Schematic of hardware setup [37]

The front-end data acquisition system was interfaced to the PC using a local area network (LAN) connection. The front-end generated a random plane wave sound signal which was amplified in the power amplifier and drove the loudspeaker in the source tube. The filter knob in the loudspeaker controlled the frequency range of the propagating plane waves and was set to high-pass for the small-tube setup and linear-pass for the large-tube setup.

### 3.3.2 Data Acquisition and Measurement—Absorption Coefficient

The impedance tube experimental setup consisted of two specially designed quarterinch phase-matched microphones Type 4187, along with two quarter-inch preamplifiers Type 2633 and three dummy microphones. The small- and large-tube setups for absorption measurements are shown in Figures 3.4 and 3.5, respectively. The microphone mounting positions and front-end connections are described in Table 3.1. Dummy microphones were inserted on other locations of the impedance tube to avoid leaks. The frequency selection knob was set at the appropriate position depending on the size of tube.



Figure 3.4 Absorption measurement: small-tube setup



Figure 3.5 Absorption measurement: large-tube setup

Т	1 1	1	2	1
	ah	0	-	
T	au	lU	э.	т

#### MICROPHONE POSITION: IMPEDANCE TUBE

Microphone ID	Small-tube	Large-tube	Front-End Position
	<b>Microphone Position</b>	<b>Microphone Position</b>	
1886539	4	2	Channel 3
1886538	5	3	Channel 4

A PULSE Labshop software version 11.1 absorption coefficient template was used to measure the absorption coefficient of test materials. The microphone positions and frequency filter were set according to the tube type. The front-end data acquisition system and the amplifier were switched on, with the microphones and loudspeaker input signal connected to the front end. The microphones were then calibrated using a Brüel and Kjær sound level calibrator Type 4231 at sound pressure level (SPL) of 94 dB at 1,000 Hz to adjust the gain and consequently the actual sensitivity of the microphone.

Before the absorption measurement of the actual sample, three measurements were taken using a standard reference sample: background measurement with no signal, original microphone measurement, and switched microphone measrement. The reference sample was flush mounted in the rigid piston end of the tube. Subsequently, the signal-tonoise ratio (SNR) was calculated from the background and signal measurements. SNR is the measure of signal strength relative to background noise. Next, a transfer function calibration was obtained from two measurements: one with interchanged microphone positions and one with normal microphone positions. Transfer function calibration was recommended in ASTM E1050 to eliminate the effects of phase and amplitude mismatches between the two measurement channels. Finally, the frequency-dependent absorption coefficient of the reference sample was measured and compared to the Brüel and Kjær standard absorption coefficient to ensure the accuracy of measurement. After measurement of the reference sample, the actual test sample was measured for the absorption coefficient.

This procedure was adopted for both small and large tubes. The data were combined over the overlapping frequency range from 500 Hz to 1,600 Hz in order to obtain the frequency-dependent absorption coefficient over the entire frequency range of 50 Hz to 6,400 Hz. The results could be post processed for octave extraction, complex reflection coefficient, complex impedance ratio and complex admittance ratio.

#### 3.3.3 Data Acquisition and Measurement—Transmission Loss

The transmission loss tube kit included four quarter-inches microphones Type 4187, along with four quarter-inch preamplifiers Type 2633 and four dummy microphones. The small- and large-tube setups for transmission loss are shown in Figures 3.6 and 3.7, respectively. The microphone mounting positions and front-end connections are described in Table 3.2. Dummy microphones were inserted on other locations of the impedance tube to avoid leaks. The frequency selection knob was set at the appropriate position depending on the type of tube.

A PULSE Labshop software transmission loss template was used to measure the sound transmission loss of the test materials. The microphone positions and frequency

filter were set according to the tube type. The front-end data acquisition system was turned on along with the amplifier, and, the microphones and loudspeaker input signal were connected to the front end. The microphones were calibrated using a Brüel and Kjær sound level calibrator Type 4231 at sound pressure level of 94 dB at 1,000 Hz to adjust the gain and consequently the actual sensitivity of the microphone.



Figure 3.6 Transmission loss measurement: small-tube setup [adapted from test setup]



Figure 3.7 Transmission loss measurement: large-tube setup [adapted from test setup]

Table 3.2

# MICROPHONE POSITION: TRANSMISSION LOSS TUBE

Microphone ID	Small-tube	Large-tube	Front-End Position
	<b>Microphone Position</b>	Microphone Position	
1886539	4	1	Channel 1
1886538	5	3	Channel 2
1882050	12	9	Channel 3
1882051	13	11	Channel 4

The loudspeaker in the source tube generated random signal plane waves. When the plane waves hit the sample in the holder, part of the sound energy was reflected, part was absorbed, and some passed through the material to the receiving tube. The receiving tube was changeable with an open termination or closed anechoically terminated. Four measurements with different impedance termination conditions at the receiving tube were required to solve the elements of the transfer matrix in equation (3.6) and to obtain the transmission loss of a material.

The first measurement was made without the sample in the sample holder, and the sound pressures at each microphone location were measured in an open-tube termination condition at the receiving end of the tube. This measurement was called background calibration open (BCO). The second measurement was also made without the sample. However, the sound pressures at each microphone location were measured in an anechoically terminated closed tube. This measurement was called background calibration closed (BCC).

The third measurement was taken to determine the transmission loss of the actual test sample. The Brüel and Kjær standard reference sample was placed inside the sample holder located in between the source tube and the receiving tube. Sound pressures were measured at each microphone location with an open-tube termination condition at the receiving end of the tube. This measurement was called total loss open (TLO). Finally, with the sample still held inside the sample holder, sound pressures were measured at each microphone location with a closed tube anechoic termination condition at the receiving end of the tube. This measurement was called total loss closed (TLC). With the help of these four measurement conditions, the transmission loss of the material was calculated using equations (3.9) after the solution of the transfer matrix in equation (3.6). Following this first transmission loss measurement, it was sufficient to perform just the last two measurements of TLO and TLC for determining the transmission loss of other materials.

This procedure as adopted for both small and large tubes. Small and large tube data were combined over the overlapping frequency range from 500 Hz to 1,600 Hz using fourth-order curve fitting in order to obtain the frequency dependent transmission loss over the entire frequency range of 50 Hz to 6,400 Hz.

# 3.4 Validation Loop

The macroscopic physical properties, namely, porosity, tortuosity, flow resistivity, viscous characteristic length, and thermal characteristic length are required to describe porous noise-control materials. The inverse characterization tool, FOAM-X, was utilized to estimate the physical properties of porous materials using the measured impedance tube test data.

A closed-loop methodology, shown in Figure 2.6, was developed to validate the physical properties estimated in FOAM-X. The estimated physical properties along with the mechanical properties of the materials were used in AutoSEA2 (see Appendix) to predict the absorption coefficient and transmission loss of all the foams and fibers. The predicted results were compared to the measured data to complete the validation loop described in Section 2.5.

### 3.4.1 Inverse Characterization Using FOAM-X

FOAM-X [32] is an inverse characterization tool to estimate the macroscopic physical properties of open-cell porous materials using impedance tube test data, measured according to ASTM E1050 and ISO 10534-2. It has the capability to characterize all five physical properties or just a few of them, based on the number of known measured physical properties.

In FOAM-X, limp porous material was characterized using the General Model and rigid porous materials using the Fiber Model. Based on the thickness and the number of available measurements, either the one-thickness or two-thickness method was used. Generally, the two-thickness method was considered more accurate. Room pressure, temperature, and humidity conditions recorded during the measurement were taken as input, since these parameters highly affect the air properties used in the identification algorithm.

For each material the frame type, bulk density, frequency range of analysis, ASCII format impedance tube data file, and sample thickness were taken as inputs in FOAM-X for inverse characterization. The impedance tube input data file was distributed into six columns as follows: frequency in Hz, sound absorption coefficient  $\alpha$ , real part of reflection coefficient Re(R), imaginary part of reflection coefficient Im(R), real part of impedance ratio Re(Z), and the imaginary part of impedance ratio Im(Z).

#### 3.4.2 Two-Room AutoSEA2 Model

A two-room model of the ASTM E90 two-room random incidence transmission loss measurement was developed in the software code AutoSEA2 [32] to predict the random incidence sound absorption coefficient and transmission loss of the porous noise-control material layer. AutoSEA2 uses statistical energy analysis (SEA) to analyze structural acoustic systems. The foam module in AutoSEA2 [32], which is based on Biot's theory of porous media [12], was used to predict the structural acoustic effects of porous noise-control treatments. Four different porous material models of foam and fibrous materials were represented by elastic porous (foam) model, limp porous (fiber) model, rigid (fiber) porous model, and Delany and Bazley (fiber) model. The treatment media was composed of thin elastic plates, thin mass layers (or septa), fluid layers and viscoelastic materials, which could be modeled as elastic plates with frequency dependent material properties. The properties relevant to model the vibroacoustic response of either foam or fibrous materials are given in Figure 3.8.



Figure 3.8 Pertinent material properties [32]

The schematic of an acoustic model in AutoSEA2 to predict the random incidence sound absorption coefficient and transmission loss is illustrated in Figure 3.9. This model

consisted of two acoustic cavities of dimension 1 m by 1 m by 1 m and a 1-mm-thick aluminum panel. Area junctions were created between the aluminum panel and the adjacent acoustic cavities to ensure that the three subsystems sharing the common boundaries were connected and that all of the appropriate wave fields occurring between the panel and the acoustic cavities would be accounted for in the transmission loss calculation.



Figure 3.9 AutoSEA2 acoustic model [31]

The source room was excited by applying an acoustic constraint, which fixed the response of the system to a known level. The physical properties estimated in FOAM-X (see chapter 4) along with the measured mechanical properties (see chapter 4) were used to define porous noise-control materials in AutoSEA2.

The frequency range of analysis was from 50 Hz to 6,300 Hz in a one-third octave domain. Then the model was solved for absorption coefficient and transmission loss without applying the noise-control treatment on the panel. Subsequently, a layer of noise-

control treatment was applied on the receiving room surface of the panel, and the model was solved again.

The absorption coefficient of the noise-control treatment was directly obtained from the second run. Transmission loss was obtained by subtracting the transmission loss of the aluminum panel from that of the noise-control treatment backed by the panel. This procedure was repeated for all materials. The random incidence absorption coefficient and transmission loss predicted in AutoSEA2 were compared with the normal incidence test data, to complete the validation loop presented in Figure 2.6.

# **3.5 Modified Validation Loop**

In the original validation loop, the estimation of physical properties in FOAM-X was limited to rigid- and limp-framed materials. Further, the measured normal incidence test data was compared to the random incidence prediction in AutoSEA2. These limitations created inconsistencies in the correlation.

In order to overcome these limitations, a modified closed-loop validation, illustrated in Figure 3.10, was proposed. The existing validation loop was modified by incorporating Comet Trim<sup>TM</sup> inverse characterization software that took both absorption coefficient and transmission loss as inputs to estimate the physical properties. Further, Comet Trim<sup>TM</sup> was capable of characterizing a range of material types, such as elastic foam, rigid foam, limp foam, and fibrous foam.

The estimated physical properties along with the mechanical properties of the materials were used in the performance analysis module of Comet  $Trim^{TM}$  to predict the normal incidence absorption coefficient and transmission loss of all foams and fibers and compared to the normal incidence measured data to complete the validation loop.

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Figure 3.10 Modified closed-loop validation process

# 3.5.1 Inverse Characterization Using Comet Trim<sup>TM</sup>

The inverse characterization module in Comet Trim<sup>TM</sup> requires the measured absorption coefficient and/or transmission loss spectrum along with the sample thickness to estimate physical properties of porous materials. Better property estimates could be achieved with the inputs of multiple measurements of the same sample. The bandwidth of the original test data was reduced from 2 Hz to 10 Hz to comply with the maximum number of input data points allowed in this software. The reduced test data was keyed into the database by appropriately selecting the type of input spectrum, and a unique name was specified to identify this.

The first step in the inverse characterization process was to select the material type based on the frame of porous material. Upon selecting the material type, Comet Trim<sup>TM</sup> displayed a material property table with relevant properties, as represented in Table 3.3. The values of all known parameters were entered by check-marking the appropriate material property. An initial approximate guess value of the unknown properties enhanced the estimation accuracy.

### Table 3.3

	Elastic Foam	Fibrous Foam	Rigid Foam	Limp Foam
Porosity	х		х	х
Flow Resistivity	х	Х	Х	Х
Tortuosity	Х		Х	Х
Viscous Length	Х		Х	Х
Thermal Length	Х		Х	Х
Density	Х			Х
Young's Modulus	Х			
Poisson's Ratio	Х			
Loss Factor	Х			
Anisotropy		Х	Х	Х

# INPUT PHYSICAL PROPERTY FOR EACH MATERIAL TYPE IN Comet $\mathrm{Trim}^{\mathrm{TM}}$

The physical properties were estimated by two methods. Initially, both absorption and transmission loss test spectrums were simultaneously used as inputs to estimate the physical properties of the porous materials, as shown in Figure 3.10. But the property estimates were not good as they did not validate the original test measurements.

In the later method, shown in Figure 3.11, two sets of physical properties were estimated. The first set of physical properties was estimated using the absorption test data spectrum. Subsequently, with the above estimated physical properties as an initial estimate, a second set of physical properties was estimated with the transmission loss test data. This method increased the accuracy of inverse characterization. The inverse characterization algorithm in Comet Trim<sup>TM</sup> is controlled by the number of iterations and a tolerance factor. Hence, a converged solution can be achieved by increasing the number of iterations, tolerance value, or both.



Figure 3.11 Sequential estimation of physical properties in Comet Trim<sup>TM</sup>

# 3.5.2 Performance Analysis Using Comet Trim<sup>TM</sup>

The Comet Trim<sup>TM</sup> performance analysis tool is capable of predicting the random incidence and normal incidence acoustical characteristics of single and multilayer porous materials. This tool is based on the transfer matrix solution method for layered media, which relates the pressures and velocities on either side of the porous material surfaces [11]. In the original validation loop, the random incidence acoustical characteristics originally predicted using AutoSEA2 was again predicted at normal incidence using Comet Trim<sup>TM</sup>.

First, the estimated physical properties and the measured mechanical properties were keyed into the database of the Comet Trim<sup>TM</sup> performance module, and a unique name was assigned for each material. Then, the material was added to the layer configuration table, and the thickness of the sample was specified.

Analysis options were set up to define the performance analysis. A frequency range of analysis from 25 Hz to 6,300 Hz was set at a one-third octave band interval. The

incidence angle was set to zero degrees normal incidence and the hard-wall absorption termination condition was selected. The test environment conditions, namely, speed of sound, density, specific heat ratio, Prandtl number, and fluid viscosity were set to their default values.

The normal incidence absorption coefficient and transmission loss were predicted in Comet Trim<sup>TM</sup> using the physical properties estimated in FOAM-X. Then, the first set of physical properties estimated in Comet Trim<sup>TM</sup>, using the absorption test data shown in Figure 3.11, was used to predict the absorption coefficient. The second set of physical properties data, estimated using the normal incidence transmission loss measurements, was used to predict normal incidence transmission loss. The predicted absorption coefficient and transmission loss were compared with the tube-measured data to complete the modified validation loop.

# 3.6 Finite Element Modeling of Sound Transmission

The effects of boundary conditions such as circumferential air gap and edge constraint on transmission loss measured in a standing wave tube can be studied using a finite element model (FEM) of sound transmission loss through a porous material. This model was developed in HyperMesh and analyzed in Comet Safe®.

#### 3.6.1 HyperMesh Model

A simple FEM mesh of the circular-tube and elastic-porous samples was created using HyperMesh in Nastran bulk data format. The model was divided into three components: source tube, receiving tube, and porous sample. Hexahedral brick (CHEXA) elements with a material property of air was applied to all three components in HyperMesh. Large-tube and small-tube models were developed to represent exactly the

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measurement procedure. Tubes were modeled with a resolution of five elements per wavelength at the highest frequency of analysis.

HyperMesh was used to model a large-tube with an inner diameter of 100 mm, as shown in Figure 3.12. The length of the source and the receiving tube was 0.45 m. A circumferential mesh density of 16 was used, with eleven three-dimensional hexahedral brick elements along the length for an upper frequency of 1,600 Hz. Similarly, HyperMesh was used to model a small-tube with an inner diameter of 29 mm, as shown in Figure 3.13. The length of the source and the receiving tube was 0.30 m. A circumferential mesh density of 16 was used, with twenty-eight three-dimensional hexahedral brick elements along the length for an upper frequency of 6,400 Hz.



Figure 3.12 HyperMesh model of large-tube transmission loss setup



Figure 3.13 HyperMesh model of small-tube transmission loss setup

# 3.6.2 Comet SAFE® Analysis

The sound transmission through porous cylindrical samples was analyzed using the code Comet SAFE® [41, see Appendix]. This software is based on a finite element implementation of the Biot theory for wave propagation in elastic porous materials. The Nastran format bulk data file, modeled in HyperMesh, was imported into Comet SAFE®.

Three zones were created: the source, receiver and material. Zonal properties consisting of the material type and spatial dimension of the element were assigned, corresponding to each zone. The source and receiver zones were associated with three-dimensional acoustic brick elements and air properties. The material zone was also associated with three-dimensional brick elements but with elastic porous properties based on the frame of the porous material and chosen from Table 3.3. After assigning the material type, relevant material properties corresponding to each zone were assigned. The air density and speed of sound were used to define the source and receiving tube. The

physical properties estimated using Comet Trim<sup>TM</sup> along with the mechanical properties (see chapter 4) was used to define the properties of the porous material.

The frequency range of analysis and the boundary conditions were specified in the model. A velocity boundary condition of 1m/s was applied at the input end of the tube and a plane-wave impedance of 415.03 Rayls was applied to both ends to ensure plane wave condition in the tube, as shown in Figure 3.14. The radial displacements of both the solid and fluid phases were set to zero as shown in Figure 3.15, at the duct circumference to model the hard-wall boundary condition. The tube air-domain was coupled to the elastic-porous material to describe the physical connection between the two domains. The model was solved for the frequency response function. Subsequently, the frequency response of the baseline model consisting of the tube without the sample was determined. The normal incidence transmission loss of the material was computed using these results.



Figure 3.14 Boundary condition at tube terminations



Figure 3.15 Boundary condition at tube material interface

#### **CHAPTER 4**

# **RESULTS AND DISCUSSION**

A database of acoustic materials, such as foams and fibers, were evaluated for absorption coefficient and transmission loss using the Brüel and Kjær impedance and transmission loss tube systems, respectively. In order to obtain reliable data, multiple measurements of the same samples were taken and averaged.

The physical properties were estimated using an inverse acoustical technique, which took impedance tube test data. Estimated physical properties and the measured mechanical properties were used to predict the absorption and transmission loss in Comet Trim<sup>TM</sup> and compared with the measured test data, correlation validated the estimated properties. Finally, best-candidate single layer-materials were organized into multiple-layered treatment configurations and tested for absorption for transmission loss.

During the course of investigating multiple-layered measurements and a repeatability study of physical properties estimation, a significant change was observed in the estimated physical properties of individual samples. Table 4.1 presents the old and new physical properties estimated using FOAM-X and two impedance tube measurements, measured in a span of 30 months. This change in physical properties as well as measured absorption coefficient and transmission loss may be due to aging of the sample. This will be discussed further in section 4.3. The recently estimated physical properties in conjunction with elastic properties in Table 4.2 were used in AutoSEA2 model to predict the acoustical properties of the porous materials.

#### Table 4.1

#### **Viscous** length Thermal length Flow Resistivity Ns/m<sup>4</sup> **Tortuosity Porosity** (Micrometer) (Micrometer) Sample OLD **NEW** OLD **NEW NEW** OLD OLD NEW OLD NEW AC 550 0.8060 0.8656 78335.130 18899.000 1.6084 1.8974 44.259 71.502 109.980 195.620 MicAA\_0.6 pcf\_1" 0.9157 0.9999 50869.420 20596.000 1.0000 1.0000 92.583 84.005 185.170 168.010 MicAA\_1.5 159157.740 0.8287 0.9999 55799.000 1.0000 1.0000 44.149 37.634 88.300 75.270 pcf\_0.375" 8898.210 1.2694 1.0284 115.838 Sonex Mini\_0.5" 0.9633 0.8772 11994.000 119.825 149.110 119.830 Sonex One\_0.9" 0.9999 0.9999 13200.170 17054.000 1.0000 1.0000 105.472 121.850 273.190 91.846 Sonex Classic\_1.7" 0.9999 0.9999 12065.220 11181.000 1.0000 1.0000 89.407 98.591 364.140 378.300 UAI 1.5 pcf 0.5" 0.8400 0.7685 115130.230 101689.000 1.0000 1.0000 26.170 41.864 52.340 83.730 SOUNDFOAM 259.260 114.170 0.9999 0.9999 14304.610 11271.000 1.0000 101.410 114.169 1.0000 ML 1"

# COMPARISON OF ESTIMATED PHYSICAL PROPERTIES IN FOAM-X [37]

#### Table 4.2

Material	Young's Modulus (N/m²)		Loss Factor		Poisson's Ratio	
Trade Name	Mean Value	Standard Deviation	Mean Value	Standard Deviation	Mean Value	Standard Deviation
Ensolite ALC	535850	7148	0.517	0.004	0.425	0.029
AC-550	36378	790	0.096	0.034	0.458	0.003
Sonex Classic	91068	1668	0.078	0.024	0.430	0.005
Soundfoam ML	87079	876	0.031	0.002	0.441	0.006

# MECHANICAL PROPERTIES OF FOAM SAMPLES [37]

However, inconsistencies in correlation of the predicted transmission loss prompted modification to the existing validation loop, recognizing a difference in normal incidence acoustical performances measured in the tube and that of random incidence acoustical performances predicted in a two-room AutoSEA2 model. A modified loop was developed which utilized the Comet Trim<sup>TM</sup> inverse characterization tool to estimate the physical properties of individual samples. As seen previously in Figure 3.11, two sets of physical properties were estimated. The first set was estimated using absorption test data spectrum. Subsequently, with the above estimated physical properties as an initial estimate, the second set of physical properties was estimated with the transmission loss test data. This method increased the accuracy of inverse characterization and is tabulated in Table 4.3. These properties were used in the performance module in Comet Trim<sup>TM</sup> to predict the acoustical performance of the porous layer.

# Table 4.3

Material Name	Input Data	Porosity	Flow Resistivity (Ns/m <sup>4</sup> )	Tortuosity	Viscous Char. Len. (m)	Thermal Char. Len. (m)
AC 550_0.5"	Absorption	0.9900	45517.0000	1.0010	0.00007771	0.00007771
	TL	0.9900	30761.0000	1.0010	0.00007484	0.00007771
AC 550_1"	Absorption	0.9900	46681.0000	1.0010	0.00001845	0.00010978
	TL	0.9900	28716.0000	1.0010	0.00004082	0.00010978
MicAA_0.6 pcf_1"	Absorption	0.9900	40957.0000	1.0010	0.00058365	0.00058365
	TL	0.7228	49704.0000	1.0010	0.00738000	0.00058365
MicAA_1.5 pcf_0.375"	Absorption	0.9470	233330.0000	2.3628	0.00045853	0.00045853
	TL	0.9900	95292.0000	1.0010	0.00005198	0.00045853
UAI_1.5 pcf_0.5"	Absorption	0.9900	99407.0000	1.0010	0.00002747	0.00040384
	TL	0.9900	101240.0000	1.0010	0.00004502	0.00004502
Sonex One_0.9"	Absorption	0.9900	19161.0000	1.0010	0.00009039	0.00009515
	TL	0.9900	15197.0000	1.0010	0.00008943	0.00009515
Sonex Mini_0.5"	Absorption	0.9900	16706.0000	1.0010	0.00013329	0.00013329
	TL	0.9900	14043.0000	1.0010	0.00010262	0.00010262
Sonex Classic_1.7"	Absorption	0.9900	11405.0000	1.0010	0.00013336	0.00013866
	TL	0.9900	9281.8000	1.0010	0.00010914	0.00010914

# PHYSICAL PROPERTIES ESTIMATED USING Comet $\mathrm{Trim}^{\mathrm{TM}}$

The normal incidence absorption coefficient and transmission loss were predicted in the performance analysis module of Comet Trim<sup>TM</sup>. The estimated physical properties along with thickness and mechanical properties were taken as inputs to predict the normal incidence acoustical performance.
The graphs in sections 4.1 and 4.2 compares the measured and predicted normal incident acoustic properties of individual samples estimated in the validation loops. The curve New.Foam\_X, shown in the graphs, was predicted in Comet Trim<sup>TM</sup> performance analysis using the physical properties estimated in FOAM-X and recently measured test data. The curve Trim.Inverse, shown in the graphs, was predicted in Comet Trim<sup>TM</sup> performance analyswas using the physical properties estimated in Comet Trim<sup>TM</sup> inverse characterization and recently measured test data. The predicted data are compared against the recently measured test data curve, TEST DATA.

## 4.1 Absorption Coefficient of Individual Samples

The graphs presented in sections 4.1.1 and 4.1.2 show the absorption coefficient versus frequency plots comparing the measured and predicted data for various foams and fibers.

#### 4.1.1 Absorption Coefficient of Fibers

It is observed from Figure 4.1 that, the absorption coefficient of Microlite AA\_1.5 pcf predicted in the curve New.Foam\_X does not correlate well with the measured data over the full range of frequencies. This could be because the thickness of the sample is less than the minimum thickness required by the FOAM-X inverse characterization algorithm.

The predicted and measured absorption coefficient of Microlite AA\_0.6 pcf and UAI 1050\_1.5 pcf in Figures 4.2 and 4.3, respectively, shows a good correlation over the entire frequency range because fibers are considered to be rigid- or limp- frame material and both FOAM-X and Comet Trim<sup>TM</sup> use a limp-frame model to characterize materials.



Figure 4.1 Absorption coefficients of Microlite AA, 1.5 pcf, 0.375"



Figure 4.2 Absorption coefficients of Microlite AA, 0.6 pcf, 1"



Figure 4.3 Absorption coefficients of UAI 1050, 1.5 pcf, 0.5"

## 4.1.2 Absorption Coefficient of Foams

Although foams are elastic-frame porous material, when set on a rigid backing, they behave like a rigid-frame porous material having a motionless solid phase. Hence, the rigid-frame model in FOAM-X is a good approximation for foam samples on a rigid backing. Consequently, the predicted absorption coefficient of foam samples exhibited a close correlation with test results, as shown in Figures 4.4 through 4.8.

It is observed that the measured absorption coefficient of Sonex Classic\_1.7" in Figure 4.4, correlates well with Comet  $Trim^{TM}$  over the entire frequency range. The FOAM-X prediction correlated well, except in the frequency range of 2,500 Hz to 5,000 Hz.



Figure 4.4 Absorption coefficients of Sonex Classic, 1.7"

From Figures 4.5 and 4.6, it can be observed that the absorption coefficient predicted by FOAM-X shows a closer correlation for Sonex One 0.9" and Sonex Mini 0.5", respectively, over the entire frequency range.



Figure 4.5 Absorption coefficients of Sonex One, 0.9"



Figure 4.6 Absorption coefficients of Sonex Mini, 0.5"

It can be observed in Figures 4.7 and 4.8 that characterization of AC 550 samples using FOAM-X is more accurate when compared to Comet Trim<sup>TM</sup>.



Figure 4.7 Absorption coefficients of AC 550, 0.5"



Figure 4.8 Absorption coefficients of AC 550, 1"

The overall conclusion from these analyses was that foams and fibers mounted on a rigid backing essentially behaved as rigid-frame material, independent of whether the sample was foam or fiber.

## 4.2 Transmission Loss of Individual Samples

The graphs presented in sections 4.2.1 and 4.2.2 show the transmission loss versus frequency plots comparing the measured and predicted data for various foams and fibers.

In general for all samples, a close correlation was observed for transmission loss below 1,000 Hz, predicted in Comet Trim<sup>TM</sup> using FOAM-X- and Comet Trim<sup>TM</sup> - estimated physical properties. It is worthwhile to understand that the Comet Trim<sup>TM</sup> - predicted transmission loss is based on sound transmission through a laterally infinite sample, and the laterally infinite condition was a close approximation for the low-frequency measurements recorded in the large 100 mm diameter samples.

For all samples, the transmission loss predicted in Comet Trim<sup>TM</sup> using FOAM-Xestimated physical properties was lower than the test data for frequencies greater than approximately 1,000 Hz. This was taken care of in the modified validation loop that uses Comet Trim<sup>TM</sup> for inverse characterization and performance analysis. The inverse characterization module in Comet Trim<sup>TM</sup> took the transmission loss test data as one of the inputs along with the absorption test data. In addition, elastic porous materials, such as foam, were able to be more accurately represented in Comet Trim<sup>TM</sup> using an elastic frame model incorporating the elastic properties of the material.

#### 4.2.1 Transmission Loss of Fibers

From the graph in Figure 4.9, it was observed that the Trim.Inverse-predicted transmission loss of Microlite AA\_1.5 pcf\_0.375" in the frequency range from 2,000 Hz to 6,400 Hz did not correlate with the measured data. This could be due to distortion in thickness of the small-tube sample over time which implies that, the transmission loss of a thinner sample is lower compared to a thicker sample. Analytical results using finite element simulation was reported by Ohadi and Moghaddami [42]. The graphs in Figures 4.10 and 4.11 for Microlite AA\_0.6 pcf\_1" and UAI\_1.5 pcf\_0.5", respectively, show a closer correlation of transmission loss using Comet Trim<sup>TM</sup>.



Figure 4.9 Transmission loss of Microlite AA, 1.5 pcf, 0.375"



Figure 4.10 Transmission loss of Microlite AA, 0.6 pcf, 1"



Figure 4.11 Transmission loss of UAI 1050, 1.5 pcf, 1"

## 4.2.2 Transmission Loss of Foams

It was observed in past research [11] that the effect of frame motion highly affects the sample transmission loss measurement, especially for foams, since the sample is simply supported in the impedance tube without a rigid support. The effects of sample-mounting boundary conditions were apparent in small-tube transmission loss measurement.

Good correlation was observed between the measured and predicted transmission loss for Sonex and AC 550 using the new validation loop in Comet Trim<sup>TM</sup>, as shown in Figures 4.12 through 4.16. It was also observed that, the transmission loss measurement of Sonex One 0.9" in Figure 4.13 showed minor oscillations in the mid- to high frequency range. This may be due to the circumferential air gap around the sample.



Figure 4.12 Transmission loss of Sonex Classic, 1.7"



Figure 4.13 Transmission loss of Sonex One, 0.9"



Figure 4.14 Transmission loss of Sonex Mini, 0.5"



Figure 4.15 Transmission loss of AC 550, 0.5"



Figure 4.16 Transmission loss of AC 550, 1"

#### 4.3 Investigation of Sample Aging

In the process of investigating the repeatability of estimating physical properties, previously measured porous material samples were re-measured for their absorption coefficient and transmission loss. Differences in measurements were noticed between the present and past test data of the same sample measured after a span of 30 months. This may be due to sample aging and/or variation in boundary condition. Aging can include effects such as sample compression, fiber distortion, which may affect the physical properties as well as boundary condition during tube measurement. Further investigation is recommended to explore these effects. The measurements of absorption coefficient and transmission loss were repeated two to three times and generally did not show much variation. Hence, this variation may be due to the sample aging effects.

#### **4.3.1** Effects on Absorption Coefficient

From the measurement of the absorption coefficients of foams and fibers originally measured 30 months ago, it was observed that potential sample aging resulted in a change in the absorption coefficient of several fiber samples in the low- to mid-frequency range. Aging of Microlite AA\_0.6 pcf\_1" showed a decrease in the absorption coefficient in the frequency range of 500 Hz to 2,500 Hz, as shown in Figure 4.17. Figure 4.18 illustrates a comparison of absorption coefficients of UAI\_1.5 pcf\_0.5". It was observed that the absorption coefficient decreases in the frequency range of 1,000 Hz to 3,500 Hz. The absorption measurements were repeated two to three times and generally did not show much variation. This variation may be due to the sample aging effects.



Figure 4.17 Aging effect on absorption in Microlite AA, 0.6 pcf, 1"

This sample aging did not indicate any change in absorption coefficient for Sonex foams. This is possibly because the samples did not significantly degrade owing to their high stiffness and modulus properties. The graph in Figure 4.19 shows the comparison of

old and new test data of Sonex Classic 1.7". AC 550 0.5", indicating a decrease in absorption coefficient in the frequency range of 500 Hz to 4,000 Hz. An increase in absorption coefficient from frequency 4,200 Hz to 6,400 Hz is shown in Figure 4.20



Figure 4.18 Aging effect on absorption in UAI, 1.5 pcf, 0.5"



Figure 4.19 Aging effect on absorption in Sonex Classic, 1.7"



Figure 4.20 Aging effect on absorption in AC 550, 0.5"

# 4.3.2 Effects on Transmission Loss

A decrease in transmission loss due to potential aging was observed for Microlite AA fiber samples, as shown in Figure 4.21. The transmission loss of UAI\_1.5 pcf\_0.5", shown in Figure 4.22, was however unchanged. Surprisingly, the Sonex foam showed an increase in transmission loss, both in predicted and measured results, as seen in Figure 4.23.



Figure 4.21 Aging effect on transmission loss in Microlite AA, 1.5 pcf, 0.375"



Figure 4.22 Aging effect on transmission loss in UAI, 1.5 pcf, 0.5"



Figure 4.23 Aging effect on transmission loss in Sonex Mini, 0.5"

## 4.4 Acoustical Performance of Multilayer Treatment

Noise-control materials, such as fibers and foams, are often combined to form a multilayer treatment. Such materials can offer remarkable performance advantages over a wide range of operating frequencies. An appropriate selection of a multilayer treatment

often can help to control both noise and vibration by blocking, redirecting, and absorbing the airborne sound energy and reducing the structural vibration.

The ultimate objective of the NIAR/Industry/State (NIS) research project [37] was to identify optimized multilayer treatments for reducing noise in aircraft interiors. A multilayer test matrix was developed by identifying candidate single-layer materials based on the acoustical database compiled for all individual materials. Table 4.4 presents the multilayer test matrix comprised of 16 different layup configurations. Each layup configuration was tested for both absorption coefficient and transmission loss.

#### Table 4.4

Id	Material 1	Material 2	Material 3	Total thickness
1.	2.0" Microlite AA 0.6 pcf	-	-	2.000''
2.	2.0'' Sound Foam ML	-	=	2.000''
3.	0.9" Sonex One	1.125" Microlite AA 1.5 pcf	-	2.025"
4.	0.9'' Sonex One	0.5 psf Loaded Vinyl 1.125" Microlite AA 1.5 pcf		2.065"
5.	1.125" Microlite AA 1.5 pcf	0.9'' Sonex One		2.025''
6.	1.125" Microlite AA 1.5 pcf	0.5 psf Loaded Vinyl	0.9" Sonex One	2.065"
7.	1.0'' AC 550 Foam	1.0" Sound foam ML	-	2.000''
8.	1.0" Sound foam ML	1.0" AC 550 foam	-	2.000''
9.	0.5" Sonex Mini	1.125" Microlite AA 1.5 pcf	0.5" AC 550 foam	2.125''
10.	0.5" Sonex Mini	0.5 psf Loaded Vinyl	1.125'' Microlite AA 1.5 pcf / 0.5'' AC 550	2.165"
11.	1.7" Sonex Classic	0.375" Microlite AA 1.5 pcf	-	2.075''
12.	0.375" Microlite AA 1.5 pcf	1.7" Sonex Classic	-	2.075''
13.	0.5" Sonex Mini	1.125" Microlite AA 1.5 pcf	0.5psf Loaded vinyl / 0.5"AC 550 foam	2.165"
14.	0.5" Sonex Mini	1.125" Microlite AA 1.5 pcf	0.25" Ensolite(ALC) / 0.5" AC 550	2.375"
15.	1.0" UAI 1050 1.5 pcf	0.5" Sonex Mini	0.5" UAI 1050,1.5 pcf	2.000''
16.	0.5" UAI 1050 1.5 pcf	0.5" Sonex Mini	1.0" UAI 1050 1.5 pcf	2.000''

#### MULTILAYER MATERIAL TEST MATRIX [37]

For absorption measurement, the treatment layup faced the acoustic source, with the fuselage at the other end, as shown in Figure 4.24. For the transmission loss measurement, the fuselage sample faced the acoustic source. The rationale behind this positioning scheme was to simulate the actual aircraft noise-control condition for the test layup as an aircraft interior sound absorber and a sound barrier.

For the purpose of illustration multilayer configurations having almost the same total thicknesses (configurations Id's 4, 6, 11 and 12) were chosen from the test matrix in Table 4.4. Configuration Id-5 was also considered to illustrate the effect of a limp, impervious septum. The measurement result of the entire matrix is published in the NIS report [37].



Figure 4.24 Test layup for combination measurement

Multilayer Id-4 was comprised of 0.9" Sonex One, 0.04" limp vinyl having a surface density 0.5 psf, and 1.125" Microlite AA 1.5 pcf fiberglass giving a total thickness of 2.065". Multilayer Id-6 was essentially comprised of same group of materials as Id-4, but was tested in the reverse direction. Both multilayer Id-4 and Id-6 were estimated to be 2.26 grams. Multilayer Id-5 consisted of 1.125" Microlite AA 1.5 pcf fiberglass and 0.9" Sonex One giving a total thickness of 2.025" and mass estimated as 0.64 grams. This multilayer can be viewed as Id-6 without the 0.04" limp vinyl sheet. Multilayer Id-11 was

comprised of 1.7" Sonex Classic and 0.375" Microlite AA 1.5 pcf fiberglass giving a total thickness of 2.075". Multilayer Id-12 was essentially the reverse of multilayer Id-11. Masses of multilayer Id-11 and Id-12 were estimated to be 0.71 grams.

## 4.4.1 Absorption Coefficient of Multilayer Configuration

Figure 4.25 shows results comparing the frequency-dependent absorption coefficients of the various multilayer configurations tested. From this graph, it can be observed that the absorption coefficients of the multilayer configurations Id-4 and Id-6, having reversed layup configurations almost follow a similar trend up until 1,800 Hz, beyond which the performance of multilayer Id-4 was better. The absorption coefficient of multilayer Id-5 was pretty much same as multilayer Id-6, especially in the high-frequency zone after 2,500 Hz. As a result of reversing the treatment layer in multilayer Id-11 and multilayer Id-12, a better performance was observed for the former, especially beyond 1,000 Hz.



Figure 4.25 Absorption coefficient of multilayer configuration

#### 4.4.2 Transmission Loss of Multilayer Configuration

Figure 4.26 shows test results comparing the frequency-dependent transmission loss of multilayer configurations. Comparing the reverse pair multilayer configurations Id-4 and Id-6 along with the reverse pair multilayer configurations Id-11 and Id-12, it was observed that the effect of reversing the layup had no significant change in transmission loss.



Figure 4.26 Transmission loss of multilayer configuration

It was observed that the transmission loss of multilayer Id-5 was 5 to 7 dB higher than of multilayer Id-11 and Id-12, over the entire frequency spectrum. This enhanced transmission loss in Id-5 was achieved because of the limp Microlite AA fiberglass which has low stiffness, but its inertia effects have a significant effect on the propagation of airborne acoustic energy. Based on the mass calculation, it can be asserted that multilayer Id-5 produced a better noise reduction, weighed lower and occupied lesser space compared to multilayer Id-11 and Id-12.

Now comparing multilayer Id-4 and Id-6 with multilayer Id-5, it was observed that the addition of 0.04" vinyl layer significantly enhanced the transmission loss by of 5 to 15 dB over the bandwidth of 500 Hz to 6,400 Hz. However this increase was achieved with a two and one-half time in crease in mass of the multilayer treatment because of the insertion of the 0.04" thick layer of vinyl. This increase in transmission loss can be attributed to the mass law of sound transmission loss provided by the flexible vinyl sheet, which acts as an excellent barrier that decouples the airborne acoustic energy from the panel.

#### 4.5 Finite Element Simulation of Sound Transmission Loss

The model development and analysis technique for sound transmission through a porous material in a standing wave tube was discussed in section 3.6. The intent of the model was to correlate normal incidence transmission loss measured in a standing wave tube to a model of a standing wave tube using normal incidence, rather than correlating the data to a two-room AutoSEA2 random incidence model. Representative samples consisting of Microlite AA 0.6 pcf 1" fiber, Sonex One 0.9" foam and AC 550 1" foam were chosen for the analysis.

The physical properties estimated using Comet Trim<sup>TM</sup> was used to define the porous material sample in the finite element analysis. A limp-frame material model was used to represent fibers and an elastic-frame material model was used to represent foams. The finite element model was developed to predict the sound transmission loss of the porous material.

Figures 4.27, 4.28, and 4.29 shows the comparison of measured and the predicted transmission loss data obtained using Comet Trim<sup>TM</sup> performance analysis and Comet SAFE® FEM. In case the of Microlite AA 0.6 pcf 1" fiber, Figure 4.27 shows that the transmission loss predicted using Comet Trim<sup>TM</sup> was in closer correlation to test data than Comet SAFE®, especially beyond 3,000 Hz. This could be because of the use of imprecise (assumed viscous characteristic length = thermal characteristic length) porous material input properties used in the finite element simulation. However it was observed that the transmission loss of foams, shown in Figures 4.28 and 4.29, predicted with Comet SAFE®, gave similar correlation over the entire frequency range than Comet Trim<sup>TM</sup> predictions.



Figure 4.27 Transmission loss of Microlite AA, 0.6 pcf, 1" in Comet SAFE®



Figure 4.28 Transmission loss of Sonex One, 0.9" in Comet SAFE®



Figure 4.29 Transmission loss of AC 550, 1.0" in Comet SAFE®

#### **CHAPTER 5**

#### **CONCLUSIONS AND FUTURE WORK**

## 5.1 Conclusions

- A new validation-loop design tool was created which includes the effects of elastic-framed porous material and also uses the input of transmission loss measurement data for inverse characterization. This design tool predicted normal incidence absorption coefficient and transmission loss and provided a closer and consistent correlation with the standing wave tube measurement data for most of the foams and fibers.
- Possible effects of sample aging or variation in tube installation boundary conditions were discovered during the process of investigating the inverse characterization techniques.
- It was found that the measured absorption coefficient for most of the sample foams and fibers examined were unchanged. However, the transmission loss of most samples decreased. This was due to aging or boundary condition variation as reflected through a change in physical properties of the porous materials. One exception was the melamine foam Sonex, which showed an increase in transmission loss.
- Sensible selection of multilayer treatment, when optimized to its weight and thickness, can yield very good acoustic performance and reduce aircraft cabin noise over a wide span of frequencies. The performance trade-off for choosing the right noise control treatment based on the overall noise reduction, weight and the space occupied was demonstrated. Five multilayer noise control configurations

having Id's 4, 5, 6, 11 and 12 were studied. It was concluded that Id-5 had the lowest weighed and occupied least space, while providing good transmission loss over the entire frequency spectrum.

• A finite element model of the standing wave tube was successfully developed and validated with the transmission loss tube measurements. This model can be used for investigating the effects of sample aging and variation of boundary condition.

#### 5.2 Future Work

- Although the estimated physical properties of porous materials with the closedloop validation design tool gives a close prediction of absorption coefficient and transmission loss, further validation studies will have to be conducted to ensure the accuracy of estimation. Actual laboratory measurement of the physical properties for the samples tested would provide additional insights into correlating the design tool with measured absorption and transmission loss.
- One possible alternative direction to porous materials characterization is to estimate the characteristic impedance and wave number through the material as studied by Song and Bolton [11], which can be empirically related to the physical properties of porous materials.
- The changes in measured absorption and transmission of identical samples measured between a span of 30 months may be due to aging effects. The changes could also be due to variations in boundary conditions when mounting the sample, but the variations would be expected to be constant with time. With more test data, the nuances between possible effect of aging and boundary conditions can be identified by using statistical tools.

• The finite element model of the standing wave tube and the porous material can be customized to study the effects of boundary condition and also study aging effects on acoustic materials. REFERENCES

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APPENDIX

#### **APPENDIX**

# 1. AutoSEA2 [32]

AutoSEA2 is a vibroacoustic design evaluation tool based on the method of statistical energy analysis (SEA) and was developed by ESI Group. In SEA, mathematical energy flow models of dynamic systems made of coupled acoustic cavities and structures can predict the vibration and acoustic responses of the system. The whole system is partitioned into many subsystems, and power-balanced equations that couple the power injected by the external loads and the energy of the various subsystems are solved for modal energy as

$$\omega \begin{bmatrix} \eta_1 + \sum_{i \neq 1} \eta_{1i} \end{bmatrix} n_1 - \eta_{12} n_1 & -\eta_{1k} n_1 \\ \dots & \dots & \dots \\ -\eta_{k1} n_k & -\eta_{k2} n_k \begin{bmatrix} \eta_k + \sum_{i \neq k} \eta_{ki} \end{bmatrix} n_k \begin{bmatrix} \frac{E_1}{n_1} \\ \dots \\ \frac{E_k}{n_k} \end{bmatrix} = \begin{bmatrix} P_1 \\ \dots \\ P_k \end{bmatrix}$$

where

 $\eta_i$ =subsystem damping loss factor

 $\eta_{ii}$ =junction coupling loss factors

n<sub>i</sub>=modal density

E<sub>i</sub>=energy of each subsystem

# 2. FOAM-X [32]

FOAM-X is advanced software used for defining acoustical properties of open-cell porous materials, based on impedance tube measurements following ASTM E1050 or ISO 10534-2, and was developed by the ESI Group. The parameters found with FOAM-X are compatible with poro-elastic material modules of commercially available programs such as VA One, AutoSEA2, Rayon, Comet, and NOVA. The characterization module of FOAM-X uses ASCII impedance files. The data are distributed into six columns in the format illustrated in the figure below:

🗾 test_moyen_sample1.tbx - Notepad									
Elle Edit Format Help									
%Frequency 1.0066562E+002 1.0596381E+002 1.1126200E+002	Alpha Re_R -1.2951000E+000 -6.4015554E-001 -9.4374954E+000	Im_R Re_Z 1.3534061E+000 1.2688193E+000 2.6434999E+000	IM_Z 6.8072889E-001 -1.7393430E-001 -1.8572571E+000	-2.2014738E+000 -6.2443871E+000 -1.5344284E+000	2.3142719E+000 -3.3932788E+000 -6.0393736E-001				
4									

- 1. Column #1 gives the frequency in Hertz.
- 2. Column #2 gives the sound absorption coefficient  $\alpha$ .
- 3. Column #3 gives the real part of the complex reflection coefficient Real(R).
- Column #4 gives the imaginary part of the complex reflection coefficient Imag(R).
- Column #5 gives the real part of the normalized complex surface impedance Real(Z).
- Column #6 gives the imaginary part of the normalized complex surface impedance in Imag(Z).

# 3. Comet $Trim^{TM}$ [41]

Trim, an acronym for transfer impedance method, is the software developed by Comet Technology Corporation to perform four types of analyses. Two of them are as follows:

• **Performance Analysis:** Used to calculate acoustical performance indicators such as transmission loss, absorption coefficient, acoustic impedance, and equivalent acoustic medium properties of multilayered materials including structures, elastic porous materials, and fluids. The multiple layers may consist of any or a combination of the following: elastic solid, plate panel, elastic foam, fibrous foam, rigid foam, limp foam, resistive screen, impervious screen, perforated panel, fluid, and air gap.



Inverse Characterization of Macroscopic Properties: Used for the inverse characterization of macroscopic properties using measured absorption and/or transmission loss values for materials such as elastic foam, rigid foam, limp foam, etc. In order to get the best estimation, multiple measured inputs are required. This can be a combination of transmission loss and an absorption coefficients spectrum corresponding to various thicknesses of the same material.

TL (dB)	Performance Analysis 0	atimization Sensiti	vity Inverse Char	acterizatio	
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	Test Data	Test Data			
DE03	Data Name Type	Start Freg En	d Freg Thickness	Select	
				-	
. DE02 -					
: 0E02					
0E02					
2 DE02 -	setup				
	Material Properties				
	Property Name	Value	Units	Knowr	
0.0 200.0 400.0 600.0 600.0 1000.0	Porosity	9.4700E-01			
	Flow Resistivity	1.4278E+05	(N.s/m4)		
Frequency (Hz)	Tortuosity	1.60D0E+00	1 (100 (100 (100 (100 (100 (100 (100 (1		
Comet Tri	m Viscous Char. Len.	9.5238E-05	(m)		
11 0 12	Thermal Char. Len.	2.89B1E-04	(m)		
Abs. Coen.	Density	3.00D0E+01	(Kg/m3)		
	Young's Modulus	4.4000E+05	(Nm2)		
0E82	Poisson's ratio	3.0000E-01		-	
	Loss Factor	2.0000E-01		- Intel	
0502			10 10		
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0E02	-Control Parameters-				
			Run Jol	2	
2 0E02	Max. Relations 75				
	Tolerance 1 00	0E-03	Save Rest	<i>i</i> its	
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0.0 200.0 400.0 600.0 800.0 1000.0					
Francisco de Albert					
Frequency (Hz)					
Comet Tri	m				

## 4. Comet SAFE® [41]

Structural Acoustic Foam Engineering (SAFE) or COMET SAFE® is a general purpose analysis software developed by Comet Technology Corporation for solving coupled acoustic problems in elastic-solid (structural), elastic-porous (foam), and fluid media. Problems are formulated and discretized in terms of the finite element method (FEM or FEA) and hence must be of a finite domain size. It has an accompanying graphical user interface-based pre- and postprocessor called COMET/Vision.

- 1. Analysis types in Comet SAFE®
  - Frequency response analysis
  - Acoustic Eigen frequency analysis
  - Structural modal analysis
  - Uncoupled acoustic analysis
  - Coupled structural elastic porous acoustic analysis
    - Pure displacement-displacement formulation
    - o Mixed pressure displacement formulation
  - Static analysis for structures and elastic solids
- 2. Boundary Condition Types
  - Acoustic boundary conditions
    - o Pressure, Particle velocity, and Impedance
  - Structural boundary conditions
    - o Displacement, Acceleration, Rotation, Force, and Moment
  - Randomly distributed load
  - Frequency dependent boundary conditions