5. SOUND ATTENUATION

5.1 NATURE OF SOUND WAVE

Historically, acoustic is the scientific study of sound. Sound can be considered as a wave phenomenon. A sound wave is a longitudinal wave where particles of the medium are temporarily displaced in a direction parallel to energy transport and then return to their original position [24].

Vibrating objects produces sound. Regardless of what vibrating object is creating the sound wave, the particles of the medium through which the sound moves is vibrating in a back and forth motion at a given frequency. The frequency of a wave is measured as the number of complete back-and-forth vibrations of a particle of the medium per unit of time. A commonly used unit for frequency *(f)* is the Hertz (abbreviated Hz) [24].

The wavelength (λ) of a wave is the distance, which a disturbance travels along the medium in one complete wave cycle. Since a wave repeats its pattern once every wave cycle, the wavelength is sometimes referred to as the length of the repeating patterns [24].

Like any wave, the speed of sound *(v)* refers to how fast the disturbance is passed from particle to particle. While frequency refers to the number of vibrations, which an individual particle makes per unit of time, speed refers to the distance, which the disturbance travels per unit of time. Angular frequency defines the number of radians per second in an oscillating system and is denoted by ω [24].

The speed of a sound wave is mathematically related to the frequency and the wavelength of the wave and is given [24]

$$v = \lambda x\, f \tag{5.1}$$

5.2 MEASURING SOUND WAVES

The amount of energy which is transported past a given area of the medium per unit of time is known as the intensity of the sound wave. The greater the amplitude of vibrations of the particles of the medium, the greater the rate at which energy is transported through it, and the more intense that the sound wave [25].

Intensity is the energy/time/area; and since the energy/time ratio is equivalent to the quantity power it gives

$$Intensity = energy/time \times area \text{ or } intensity = power/area \quad (5.2)$$

Typical units for expressing the intensity of a sound wave are Watts/meter^2.

Since the range of intensities, which the human ear can detect, is so large, the scale, which is frequently used to measure intensity, is a scale based on multiples of 10. This type of scale is sometimes referred to as a logarithmic scale. The scale for measuring sound intensity level *(SIL)* is the decibel scale [25]

$$SIL\ (dB) = 10 \log_{10} (I / I_{ref}) \quad (5.3)$$

The threshold of hearing is assigned a sound level of 0 decibels (abbreviated 0 dB); this sound corresponds to an intensity (I_{ref}) of $1*10^{-12}$ W/m^2. A sound which is 10 times more intense ($1*10^{-11}$ W/m^2) is assigned a sound level of 10 dB and the reference intensity of air ($I_{ref}=10^{-12}$ W/m^2) human ears can detect range of sound levels from a minimum 0 Db to a maximum 130Db.

Another logarithmic scale is the sound pressure level *(SPL)*

$$SPL = 20 \log_{10} (P/P_{ref}) \quad (5.4)$$

where P is the amplitude of the wave and P_{ref} is the standard reference. There is a proportionality between P and I and requires a factor of 20 in order to make the two sound levels similar [25].

5.3 MECHANISM OF SOUND ATTENUATION

The mechanism for reducing sound depends on where the sound comes from. If it is generated within a room then sound is *absorbed*. If it is airborne, it is originating from outside then to keep sound out it is necessary to *insulate* the space. And if it transmitted through the structure then the structure needs to be *isolated* from the source of vibration. Cellular and porous solids can be good absorbing media and they can help isolation. But they are not very good at insulation against sound.

5.3.1 Sound absorption

In this an incident sound wave is neither reflected nor transmitted; its energy is absorbed in the material. There are many ways by which this can happen. 1. By viscous losses as the pressure wave pumps air in and out of the cavities in the absorbers. 2. By thermal elastic damping. 3. By Helmoltz type resonators. 4. Vortex shading from sharp edges. 5. Direct mechanical damping in the material itself.

There are three basic categories of sound absorbers: porous materials commonly formed of matted or spun fibres; panel (membrane) absorbers having an inflexible surface mounted over an airspace; and resonators created by holes or slots connected to an enclosed volume of trapped air. The absorptive property of each type of sound absorber is influenced by the mounting method employed [26].

5.3.1.1 Porous absorbers:

Common porous absorbers include carpets, draperies, spray-applied cellulose, aerated plaster, fibrous minerals wool and glass fibres, open-cell foam, and cast porous ceiling tiles. Generally, all of these materials allow air to flow into a cellular structure where sound energy is converted into heat. Initially, there is the viscous loss as air is pumped into and out of the open porous structures. Sealing the surfaces with paint films greatly reduces the absorption. There is also the damping of the material. Damping refers to the capacity of the material to dissipate energy. It measures the fractional loss of energy of a wave per cycle, as it propagates within the material itself. Damping in metals and ceramics is 10^{-6}-10^{-2}, in polymers it's about 10^{-2}-0.2. Porous absorbers are the most commonly used sound absorbing materials. Fabric applied directly to a hard, massive substrate such as plasterboard does not make an efficient sound absorber due to the very thin layer of fibre. Thicker materials generally show greater damping [26].

5.3.1.2 Panel Absorbers:

Panel absorbers, are typically non-rigid, non-porous materials, which are placed over an airspace that vibrates in a flexural mode in response to sound pressure exerted by adjacent air molecules. Common panel (membrane) absorbers include thin wood panelling over framing, lightweight impervious ceilings and floors, glazing and other large surfaces capable of resonating in response to sound. Panel absorbers are usually most efficient at absorbing low frequencies [26].

5.3.1.3 Resonators

Resonators typically tend to absorb sound in a narrow frequency range. Resonators include some perforated materials and materials that have openings (holes and slots). The classic example of a resonator is the Helmholtz resonator, which has the shape of a bottle. The size of the opening, the length of the neck and the volume of air trapped in the chamber govern the resonant frequency. Typically, perforated materials only absorb the mid-frequency range unless special care is taken in designing the facing to be as acoustically transparent as possible. Slots usually have a similar acoustic response. Long narrow slots can be used to absorb low frequencies [26].

5.3.2 Insulation

Foams are not good at insulation. The degree of insulation depends on the mass law. This law means the heavier the material the better it insulates. Thus the lightweight modern building, which are good for thermal insulation but not good for sound insulation. So it is better to add concrete or brick layers to the wall or floor to improve sound insulation [26].

5.3.3 Isolation

Elastic materials and steel frames can transmit vibrations throughout the building. This type of noise is transmitted by the continuous solid part of the structure, so introducing a float in the floor can reduce it or by putting the building on resilient material and cellular materials could be useful. Putting the whole structure on resilient pads can also isolate buildings [26].

5.4 SOUND ABSORPTION MEASUREMENTS

Experimentally, absorption is measured using a plane-wave impedance tube. When a plane sound wave is put on an acoustic absorber, some energy absorbed and some is reflected. If the pressure p_i is the incident wave is described [27]

$$p_i = A \cos(2\Pi ft) \tag{5.5}$$

The reflected wave p_r by

$$p_r = B \cos(2\Pi ft(t-2x/c)) \tag{5.6}$$

The total sound pressure in the tube can be measured by a microphone and is given by the sum of the two. Here f is the frequency (Hz), t is the time, x is the distance from the sample surface (m), c is the velocity of sound (m/s) and A and B are the amplitudes.

The proportion of sound absorbed by the surface is called the sound absorption co efficient. The absorption coefficient α is

$$\alpha = 1 - (B/A)^2 \tag{5.7}$$

The coefficient can be viewed as a percentage of sound being absorbed, where 1.00 is complete absorption (100%) and 0.01 is minimal (1%).

Sound absorption performance of aluminium depends predominately on the permeability of the structure and is defined by the absorption factor a (the ratio of the unreflected sound intensity at the surface to the incident sound intensity). The factor varies with frequency and angle of incidence. The absorption factor is a function of material thickness, foam density and pore size [27].

5.5 SOUND ABSORPTION IN METAL FOAMS

Metal foams can either contain open or closed cells. Cellular metals with closed cells are very poor sound absorbers, because the fluid within the pores cannot move and dissipate energy. The foam Alporas when studied experimentally in its cast foam, with closed pores, did not absorb sound at all. To enhance absorption it is necessary to allow air motion. This could be done either by rolling or hole drilling. With rolling, some of the faces of the cells break to form small sharp edged cracks. These cracks become passageways for air and thus enhance sound absorption. The other method, hole drilling allows sound to be absorbed at certain frequencies. It also results in small but sharp edged cracks on the cell walls, which became passageways for the movement of air particles. Thus leads to improved sound absorption [28].

Once there is movement in and out of air then mainly viscous loss and thermal damping effects can help to dissipate sound. Analytical models could explain theses mechanisms. Initially Kirchoff solved for sound propagation in a circular tube where the thermal and viscous forces were accounted for [29]. Biot, Zwikker and Kosten model used a model to separate the viscous and thermal effects. They considered a porous metal which had an array of uniform pores a, and porosity (Ω.) The porous medium was saturated with air density ρ_0, thermal conductivity being k_f and occupies a space $0<x>L$ with the assumption that $L>a$ where L is the thickness of the porous material. The aim of the model was to follow the motion of air when the system was subjected to global pressure in the form $\nabla pe^{i\varpi t}$ (where ϖ is the angular frequency and t is the time) where ∇p is the pressure gradient $e^{i\varpi t}$ describes the motion of the wave in model. This can determine the fraction of sound absorbed by the porous medium. [28].

The sound absorption coefficient (α) was plotted as function of frequency f for selected values of cell size a = 0.5, 1, 3 and 5mm for L= 10cm and L= 1cm. The absorption coefficient includes both the viscous and thermal effects. This showed that viscous effect is comparable to the thermal effect when the foam is thin (L =1cm) but it becomes dominant at high frequencies when the foam is thick (L=10cm), Figure 23 [28].

The ratios of the two sound coefficients $\alpha_{\text{viscous}}/\alpha$ was plotted so to normalize the viscous effect and determine the contribution of the viscous effect. Similarly the ratio of the two sound coefficients were plotted as a function of frequency for the same pores sizes a=0.5, 1, 3 and 5mm for L= 10cm and L= 1cm. It showed that when foam was 1cm thin then a thermal effect was dominant, but when the thickness was increased to 10cm thick then viscous effects dominated. Increasing pore size increased heat loss due to air compression, Figure 23 [28].

Figure 23 *Sound absorption coefficient* α *of a model porous material consisting of uniform circular pores plotted as a function of frequency and pore diameter a for foam thickness* ***(a)*** *L = 10cm* ***(b)*** *L = 1cm:normalized sound absorption* $\alpha_{viscous}/\alpha$ *of the same porous model* ***(c)*** *L=10cm* ***(d)*** *L=1cm where material has porosity of 0.9 and is backed by a rigid wall* [28].

T.J. Lu *et al* investigated the use of semi open cells for sound absorption. The foams were processed using negative-pressure infiltration, using perform consisting of soluble spherical particles. The cells of the foam were interconnected by small circular openings with the sizes adjustable due to the variation in the infiltration pressure, particle size, surface tension of molten alloy and wetting angle between molten alloy and particle. The important parameter of the foam was the interconnectivity between the pores, which depended on 2 dimensionless parameters. An analytical model was developed to quantify the dependence of pore connectivity on the processing parameters, including infiltration pressure, particle size, wetting angle and surface tension of molten alloy. It was found that to obtain pore connectivity by adjusting the parameters increasing infiltration pressure is equivalent to decreasing particle size or increase surface tension [30].

Normal sound absorption coefficient was measured experimentally using the impedance tube for 6 samples having different grades of porosity, pore size and pore opening. There was no apparent correlation between sound absorption and porosity or cell size. This may be attributed to the limited parameter range explored. There was an excellent relation between sound absorption and pore size opening with sound absorption increasing as pore opening decreases. Static flow resistance could also measure acoustical performance. The static flow resistance determines that, as the pore size opening is increased then static flow resistance decreases.

In addition to the experimental procedures to calculate sound absorption a theoretical model was also developed. An acoustical model was developed for semi open metallic foams having idealised structures; regular hexagonal prismatic cell was chosen as the unit cell, with one circular aperture placed on eight of its surfaces. The theory depended on the acoustic impedance of the apertures and cavities were developed to establish a structure property relationship having idealised semi open cellular structures. There were great comparisons with the experimental data. Parameter studies showed that there exists an optimal pore size, an optimal pore opening size and optimal pore connectivity for best sound absorption.

An optimal cell size of 1mm tends to maximise frequency in the low frequency range (f < 4000 Hz). The pore connectivity should be 0.3 when the foam is processed. The effect of porosity at 50, 60 and 70 % showed a slight increase in the frequency, Figure 24. Sound absorption was improved by increasing the thickness of the panel but only slightly affected by varying the foam porosity [30].

6. MODEL DEVELOPMENTS FOR SOUND ATTENUATION

The model developed in this project is taken from the classical mass spring harmonic oscillator which when disturbed from its equilibrium or rest position will oscillate with a simple harmonic motion. A simple harmonic motion is when a force tries to restore the object to its rest position and it is proportional to the displacement of the object. This type of motion can be described as a sine wave [31].

The simplest vibratory system can be described by a single mass connected to a spring (and possibly a dashpot). The mass is allowed to travel only along the spring elongation direction. Such systems are called Single Degree-of-Freedom (SDOF) systems [31].

The model used in this project considered a simple single degree of freedom of forced oscillation. When oscillation is forced it shows that the physical behaviour of the oscillator is driven by an alternating force, which in this case is the driving frequency (ω). This model consisted of a heavy base (M) attached by a spring and a viscous damper to a base. Fig 25. Assume the base vibrates at a single frequency (ω) with an amplitude X *so* the displacement is $x=Xe^{i\omega t}$ (where $e^{i\omega t}$ describes the motion of the base). The amplitude of a given oscillator is the maximum displacement between the extreme points and the equilibrium point. Similarly, the deflection of the heavy mass is $y=Ye^{i\omega t}$ where y is the vertical displacement of the base Y is the output amplitude. From the equations of motions it showed that a transfer function could be given as

$$H(\omega) = Y/X = (\omega/\omega_1)^2 / 1-(\omega/\omega_1)^2 + i\eta\,(\omega/\omega_1) \qquad (5.8)$$

where ω_1 is the undamped natural frequency of the oscillator and η is the damping constant. The undamped frequency can be explained, as all objects, when hit or struck or plucked or somehow disturbed, will vibrate. When each of these objects vibrates, they tend to vibrate at a particular frequency or a set of frequencies. The frequency or frequencies, at which an object tends to vibrate with when hit, struck, plucked, or somehow disturbed is known as the natural frequency or the undamped frequency of the object. If the amplitude of the vibrations are large enough and if natural frequency is within the human range frequency then the object will produce sound waves which are audible [32].

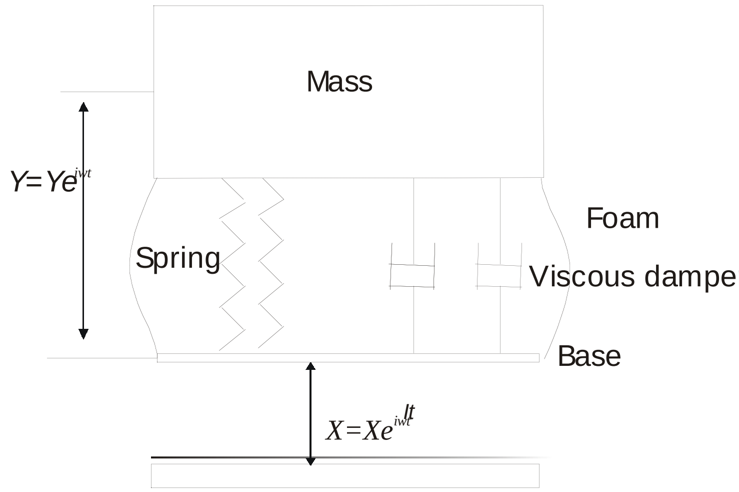

Fig. 24 *Single degree of freedom oscillator subjected to an input frequency ω* [32].

A transfer function can describe as the relative oscillation of the output oscillation to the input of an oscillation. The model (foam) formed was assumed to have both the viscous damper and spring properties.

Initially, the aim of the model was to calculate the undamped frequency for an aluminium circular plate. If the radius of the plate was R and M_1 was mass per unit area were fixed, then

$$\omega_1 = C_2/2\Pi\sqrt{(Et^3)/m_1R^4\ (1-v^2)} \qquad (5.9)$$

where E is Young’s Modulus, v is Poisson’s ratio and C_2 is a constant (for plates 1.44). Poisson’s ratio, describes the relationship between lateral and axial strain. In the linear elastic regime of experiments the lateral strains bear a constant relationship to the longitudinal or axial strains [15].

Poisson's ratio is found by taking the absolute value of the lateral strain divided by the axial strain.

$$v = |Lateral\ strain/axial\ strain| \qquad (5.10)$$

7. RESULTS AND DISCUSSION

For the single degree of freedom oscillator model for aluminium and the porous medium FORTRAN program were written and analysed (see appendix for details of the FORTRAN programs, *soundAl.f* and *soundalpore.f*).

Initially, aluminium metal was plotted as a transfer function for the relative displacement *y* as a function of frequency, taking the Young's Modulus (71GPa), Poisson's ratio (0.34) with radius (0.5m), thickness (0.01m) and M_1 to be 27.10 (as density of Aluminium is 2710kg/m^3). The transfer function gives a measure of the damping capacity of the material. The driving frequency was scaled logarithmically and the graph

was plotted as shown, Figure 25

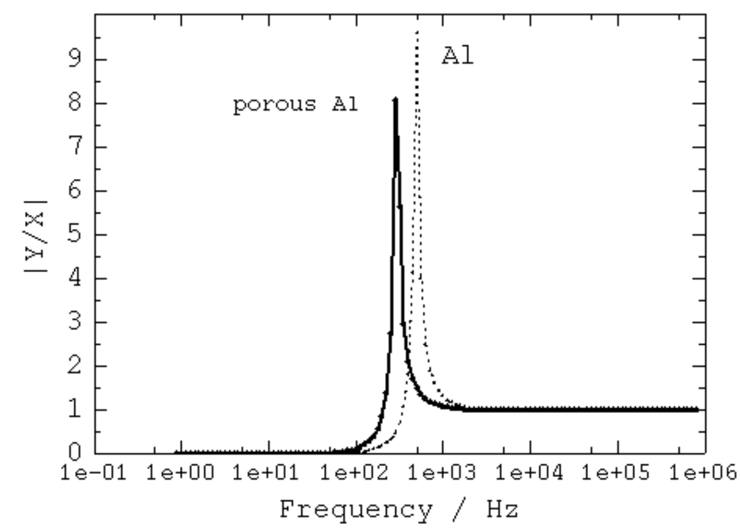

Figure 25 The *transfer function for the relative displacement as a function of natural frequency (Hz)*

Now to convert the aluminium plate to foam need to use the scaling laws specified in [32]. The metals thickness remained constant and the modulus *E* decreased as $(\rho/\rho_s)^2$ and mass varied as (ρ/ρ_s) giving the scaling law

$$\omega_1 / \omega_{1,s} = (\rho/\rho_s)^{-1/2} \qquad (5.11)$$

where ω_1 is the undamped frequency of the foam and $\omega_{1,s}$ was the frequency of the solid (aluminium). The lower the density the higher the natural frequency. (see appendix derivation using scaling laws)

Now to convert the circular plate to the foam then needed to substitute the density scaling laws keeping the thickness constant. The transfer function of a porous metal using 65 % porosity was calculated using similar values for the aluminium above but using density-scaling laws, which then gave the transfer function as a function of frequency. When compared to the transfer function of the circular plate it showed that there was not much difference in the transfer function of the foam and the Al metal over majority of the region of the frequency range. However it showed that the absorption of porous Al is slightly more than the Al metal over a frequency range ~100 Hz – 500Hz, Figure 26.