A Comparative Study of Performances of Different Types of Barrier Constructions for Attenuating Low Frequency Noise

Abid Hossain Khan[1*], Md. Mostofa Hossain[2]
Department of Mechanical Engineering, Sonargaon University (SU)
*Corresponding author: khanabidhossain@gmail.com, khanabidhossain@su.edu.bd

Abstract

Low frequency sounds and vibrations are associated mostly with rotary machines, which are the most common ones, seen from workplaces like factory lathe machines, milling machines, etc., to machineries like pumps, fans, generators etc., to vehicles like cars, trains, airplanes etc., in one words, almost everywhere. When the sound pressure level of these low frequency sounds is not within a tolerable range, it is considered as noise and it must be controlled in order to protect human ear from its hazardous effects. Low frequency noise can be controlled in many ways; one of the ways is restricting the noise from reaching human ear i.e. using barriers. The problem faced by barriers in restricting of low frequency noises is that they have a high wavelength, making them likely to pass through barriers easily and less chance of reflection. Also, the absorption coefficient of a material is inversely proportional to the wavelength, therefore low absorption possibility for low frequency noises. To solve these problems, the mass of the barrier is increased following mass low, which increases cost. In the present study, it has compared the transmission losses due to two types of commonly used barrier constructions, Double layer type barriers and Sandwich type barriers and determined which one has better performance in attenuating low frequency noise. Also, for construction, two different types of absorbing materials, glass wool and PE foam is used to account for material variations.

Keywords: Noise Barrier; Transmission Loss; Low Frequency Noise; Noise Attenuation; Sound Pressure Level, Mass Law, Critical Frequency; Coincidence region

Nomenclature

Notation	Details	Notation	Details
τ	Transmittance	ω_c	Critical frequency of the medium
γ	Adiabatic coefficient	$\Delta\omega$	Frequency bandwidth
u	Velocity of medium particles	B	Bending stiffness per unit area
ρ	Density of the medium	E	Modulus of elasticity
ρ_s	Mass per unit area	t	Thickness of the panel
c	Speed of sound	θ	Angle of incident sound
η	Structural loss factor	θ_{CO}	Coincidence angle

υ	Viscous coefficient of the material	f	Frequency
σ_{rad}	Radiation efficiency	f_c	Coincidence frequency
k	Stiffness per unit area	L_x, L_y	Panel dimensions
ω	Frequency of incident sound		

1.0. Introduction

A single homogeneous barrier needs a thick layer to restrict greater noise. But in many cases, the size of the barrier is a limiting issue. For that reason, compact double layer barriers have been studied for better performance in lesser space. Many have thought of backing the front rigid layer with flexible natural fibrous materials, others have used the fibrous material in between two rigid layers [1-2]. There have been many studies regarding performance of each of the two kinds, but the number of comparative study between them is very few, and the number is even lower if the studies conducted on low frequency noise are only considered. Khedari et. Al. [3] has developed new insulating barriers with Durian (*Durio zibethinus*) peels and coconut (*Cocos nucifera*) coir fibers are used as the raw material to manufacture particleboards. The cost of the barriers is later reduced [4]. Earlier than that, Larbig et. Al. [5] has worked on natural fiber reinforced foam based on renewable resources for automotive interior applications. Multi-layer absorbers [6] have been studied by Lee et al. Nor et al. later did a similar study to find the sound absorption using coconut coir fiber layers [7]. Many other researches on natural fibers for double layer walls have been conducted to identify their effectiveness [8-9].

Over the years, a great deal of research has been carried out in identifying the TL characteristics of different sandwich panel constructions. Bending deformation in a sandwich panel construction [10] has been characterized by Ross et. al. The description of bending rigidity is utilized by Holmer [11] in the development of the coincidence wall design. Manning [12] has studied procedures for optimizing the performance of the coincidence wall and developed expression for the effective damping in panels with 3, 4, and 5 layers. The first effort at describing the effects of the thickness deformation on panel TL is attributable to Lord [13]. Panel dynamics is characterized using a variation approach with assumed displacement fields for the face sheet sand core of a three-layer panel configuration. The face sheet sand core consisted of homogeneous isotropic materials.

Subsequent investigators, Smolenski et. al. [14] has applied essentially the same approach in studying the effects of core compliance on panel TL. The approach of Lord et al. is extended by Moore [15]. This study specifically focuses on low frequency noises as they are the hardest ones to control. In this study, both types have been experimentally studied to identify which one is superior for reducing low frequency noise since both have very good performance for high frequency noise as seen from the previous studies.

2.0. Mathematical modeling

2.1. Transmission loss in single panels

When sound is incident on a surface that is an interface between two mediums having different densities and speed of sound, a portion of the sound is absorbed in the second medium, a portion is reflected back to the previous medium and the rest of the sound is transmitted through the medium. This is shown in the Fig. 1.

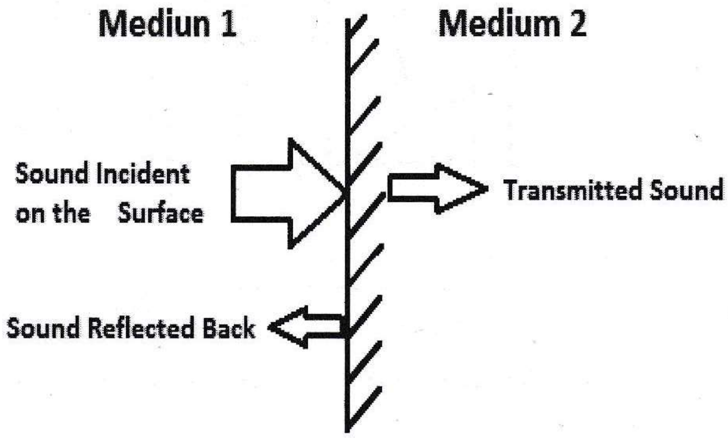

Fig. 1. Transmission of incident sound.

Now, let us consider I_i, intensity of the incident sound, I_a, intensity of the absorbed sound, I_r, intensity of the reflected sound and I_t, intensity of the transmitted sound, then

$$I = I_a + I_r + I_t \tag{1}$$

The transmittance of a medium is given by the ratio of the intensity of sound transmitted to the intensity of the sound incident on a surface, This equation is valid for normal-incidence transmission losses greater than 15dB and it represents n incident diffusion field with a limiting angle of 78 degrees. A random incidence mass law can also be obtained by averaging equation (6) overall all angles from 0 to 90 degrees. If the normal-incidence transmission loss is defined as TL_o then the random-incidence transmission loss is

$$TL_R = TL_0 - 10log_{10}(0.23TL_0) \tag{8}$$

Likewise, the field-incidence transmission loss may be re-expressed as,

$$TL_R = TL_0 - 5\ dB \tag{9}$$

Equation (9) may also be used to obtain a qualitative understanding of the behavior of panels or barriers at frequencies above the critical frequency. It cannot, however, be used in practice because incident sound waves generally involve a broad range of frequencies and angles of incidence; the latter are generally indeterminate. A close examination of the equation indicates that transmission loss is the minimum when

$$\frac{B\omega^2}{2\rho_0 c^4} sin^4\theta = 1 \tag{10}$$

This condition is referred to as the coincidence condition and it corresponds to a situation where the trace wavelength, ($\lambda/\sin\theta$) of the incident sound wave equals a free bending wavelength, λ_B, at the same frequency. For finite panels, free bending waves only occur at natural frequencies; for infinite panels, they can occur at any frequency. Thus, for finite panels there will be certain coincidence angles, θ_C's, and corresponding coincidence frequencies, ω_C, at frequencies above the critical frequency for which there is very efficient transmission of sound. For finite flat panels, the coincidence frequencies are in fact natural frequencies. From equation (5) and (10),

$$sin\theta_{CO} = \left(\frac{\omega_C}{\omega}\right)^{1/2} \tag{11}$$

At these coincidence angles, the panel transmission loss is obtained by substituting equation (19) into (10). It is,

$$\tau = \frac{I_t}{I_i} \qquad (2)$$

This transmittance is a function of the densities and speed of sound in two mediums and is given by equation,

$$\tau = \frac{4(\rho c)_1(\rho c)_2}{\{(\rho c)_1 + (\rho c)_2\}^2} \qquad (3)$$

Now, transmittance can be used to defined another term called Transmission Loss (TL) which is the measure of how much sound is passing through a medium and it is given by,

$$TL = 10log_{10}\left(\frac{1}{\tau}\right) \qquad (4)$$

The characteristics of a bounded homogeneous barrier are shown schematically in Fig. 2.

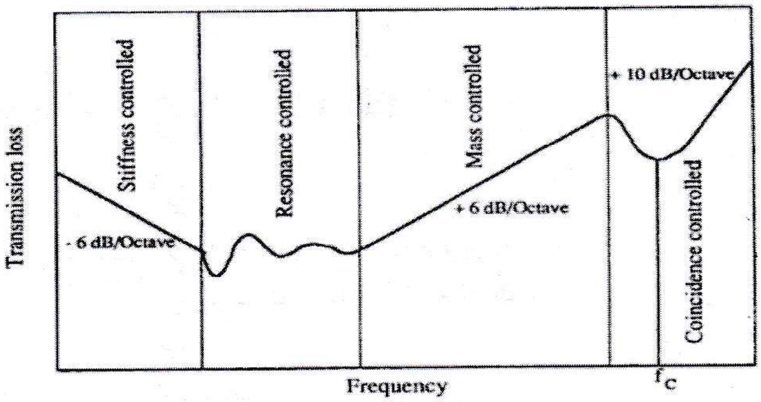

Fig. 2. Transmission loss in a homogenous barrier

There are four regions of interest: stiffness controlled, resonance controlled, mass controlled and coincidence controlled. Because of the fact that the barrier is finite and bounded, it has a series of natural frequencies. It is important to note that these frequencies are not always relevant to sound transmission. If the panel is mechanically excited, or if the incident sound field is not diffuse, then the resonant structural modes control the sound transmission through the barrier. Under these conditions, the addition of suitable damping material would increase the TL. If the barrier is acoustically excited below the critical frequency and the incident sound field is diffuse, then the forced bending waves at the excitation frequencies dominate the sound transmission through the panel and the resonant structural modes are relatively unimportant.

Again, at frequencies well below the first fundamental natural frequency, it is the stiffness of the material that dominates its sound transmission characteristics. At this region, the addition of mass or damping will not affect the transmission loss characteristics. The transmission loss at this region is given by the equation,

$$TL = 10log_{10}\{1 + (\frac{k/\omega}{2\rho_0 c}cos\theta)\} \quad (5)$$

Where k is the stiffness per unit area and ω is the frequency of incident sound. Doubling the frequency i.e. an octave increase produces a decrease in transmission loss by 6dB. Doubling the stiffness will increase the transmission loss by 6dB.

At frequencies above the first few natural frequencies but below the critical frequency, the response is mass controlled. In that region, the equation of transmission loss is,

$$TL = 10log_{10}\{1 + (\frac{\rho_s\omega}{2\rho_0 c}cos\theta)^2\} \quad (6)$$

Here ρ_s is the mass per unit area. It may be shown from the equation that there is a 6dB increase in transmission loss per octave increase in frequency. There is also a 6dB increase in transmission loss if the mass is doubled. Damping and stiffness do not control the sound transmission loss in this region. This is called the mass law equation.

Equation (6) is valid only for a specific angle of incidence ranging from 0 to 90 degrees. When the incident sound field is diffuse, as is generally the case in practice with the exception of certain confined spaces, an empirical field-incidence mass law is commonly used in place of the opaque-incidence mass law. It is,

$$TL = 10log_{10}\{1 + \eta(\frac{\rho_s\omega}{2\rho_0 c}cos\theta)^2\} - 5dB \quad (7)$$

$$TL = 10log_{10}\{1 + \eta\,(\frac{\rho_s \omega}{2\rho_0 c} cos\theta_{CO})\}^2 \qquad (12)$$

At the critical frequency, θ = 90, and the barrier offers no resistance to incident sound waves. At other coincidence angle, the transmission loss is limited by the amount of damping that is present. At angles of incidence that do not correspond to a coincidence angle, the transmission loss is obtained from equation (5). Here, both stiffness and damping limit the transmission of sound through the panel.

Because of random nature of the frequency composition of the incident sound waves and the associated angles of the incidence, equation (5) must be solved by numerical integration procedures to obtain a field-incidence transmission loss for frequencies above the critical frequency. Alternatively, an empirical relationship can be used. It is,

$$TL_R = TL_O + 10log_{10}(\frac{f}{f_c} - 1) + 10log_{10}\eta - 2dB \qquad (13)$$

The equation indicates a 10dB increase per octave increase in frequency. It also suggests that structural damping plays an important part in maximizing the transmission loss in this frequency range.

2.2. Transmission loss through a multi-layer barrier

In case of barriers having more than one layer, it is difficult to obtain a universal characteristic equation. There are basically three reasons behind it,

I. Firstly, with addition of each layer, an additional surface interface is introduced, which causes reflection of a portion of transmitted sound which overlaps with the incoming sound wave and creates standing wave. Therefore, coincidence regions far before the end side of the barrier are observed, which cause change in transmission loss characteristics.

II. Secondly, in case of addition of absorbing material with rigid material to form a composite barrier, the equivalent absorption and reflection coefficient is greatly changed for different frequencies.

III. Finally, if the barrier is backed by reflective material, the reflection of sound is increased, decreasing transmission loss.

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σ_{rad}	Radiation efficiency	f_c	Coincidence frequency
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Due to these three cases, the transmission losses are different in multi-layer barriers from single layer barriers. The non-homogeneity of the barrier material makes it difficult to derive any generalized equation since each barrier has different composition. The transmission loss of these barriers can be approximated using equation (6) and (7).

Also, the multi-layer barriers follow the mass law differently from single layer barriers. In case of multi-layer barriers, with addition of each layer, the slope of the mass dominant region is found by adding the slopes of each layer, thus increasing transmission loss at higher frequencies. Again, addition of damping materials will increase the transmission loss if the barrier is acoustically excited and decrease if the barrier is mechanically excited.

2.3. Equation of low frequency mechanical noise:

For a solid vibrating surface, driven or in contact with a prime mover or linkage, radiated sound power (W in Watts) is proportional to the vibrating area S and the mean square vibrating velocity $< v^2 >$, which is given by,

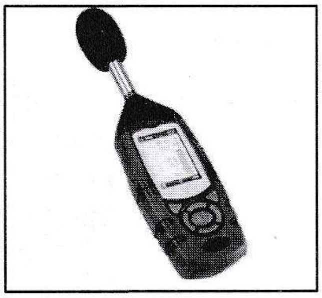
$$W = \rho c S < v^2 > \sigma_{rad} \tag{14}$$

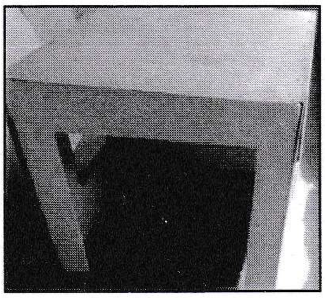
Where, ρ is the air density (kg/m3), cis the speed of sound (m/s) and σ_{rad} is the radiation efficiency.

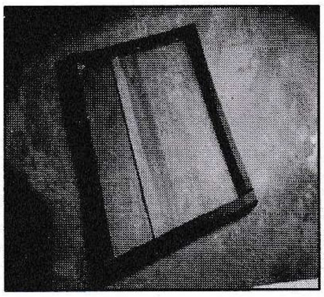
Usually mechanical noise is a low frequency noise having frequency ranging from 100 to 500Hz. In our present study, the focus is to control this low frequency noise. Therefore, the work mainly focuses on mechanical noise.

3.0. Materials and methods

To conduct the experiment of the effect of barriers to transmission loss a noise enclosure, a noise source and a sound level meter have been used. The noise source is placed inside the enclosure and the noise level is measured from outside the barrier. Measurements are done in no barrier and with barrier condition. The differences between these two values are the desired transmission loss. The noise enclosure that was built for this experiment has 3 fixed walls and 1 flexible wall. The flexible wall is the front wall where different noise attenuating barrier panel can be attached with







(a) (b) (c)

Fig. 3. (a) Sound Level Meter; (b) The noise enclosure; (c) the removable panel for attaching the barriers CASELLA CEL-62X sound level meter has been used to measure the sound pressure level at different frequency.

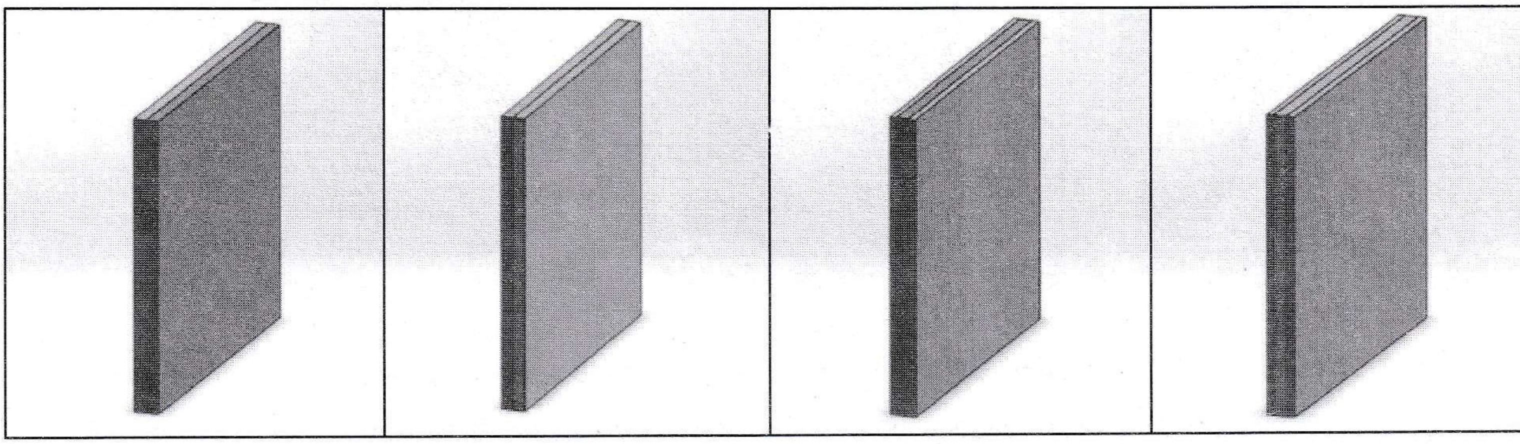
screws and nuts. The fixed walls along with the roof are made of steel sheets. The inner surfaces of them are covered with foam as a noise absorbing material. The dimension of the enclosure is 2ft x 2ft x 2ft. Fig. 3. shows different components of the experimental setup.

4.0. Objective

The objective of this experiment is to observe the transmission loss through barriers of different absorbing materials of different setup of layers. 4 panels are made with 2 different types of construction. Materials used were plywood, glass wool and foam with Aluminum cover. The thickness of every panel has been kept constant at 1 inch for better comparison of transmission loss. The four constructions are,

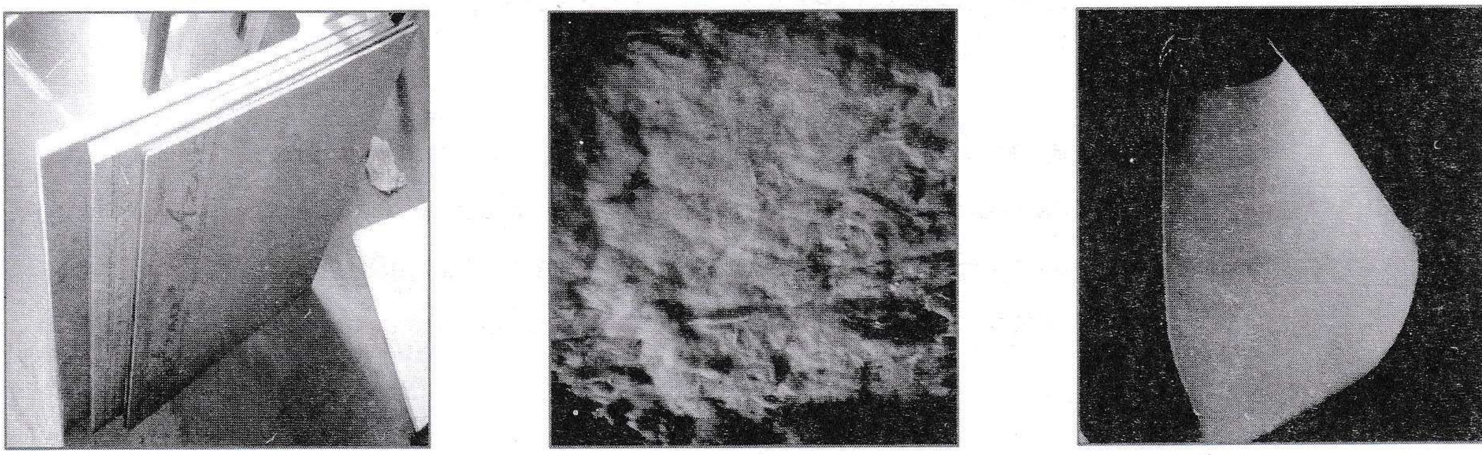
- Glass wool-backed plywood barrier: ½ inch plywood board was backed with ½ inch layer of glass wool.
- Foam-backed plywood barrier: ½ inch plywood board was backed with ½ inch PE foam layer.
- Plywood-glass wool sandwich barrier: This panel was constructed by keeping a ½ inch glass wool layer between to ¼ inch plywood board.
- Plywood-PE foam sandwich barrier: This panel was constructed by keeping a ½ inch PE foam layer between to ¼ inch plywood board.

The four constructions are shown in Fig. 4 and the materials used in making the barriers are shown in Fig. 5.



(a) (b) (c) (d)

Fig. 4. (a) Glass wool-backed plywood barrier; (b) Foam-backed plywood barrier; (c) Plywood-glass wool sandwich barrier; (d) Plywood-PE foam sandwich barrier



(a) (b) (c)

Fig. 5. (a) Plywood of different thickness; (b) Glass wool; (c) PE foam

During the experiment, the noise source (a drilling machine) is placed in the enclosure. The front wall is kept open i.e. first measurement was done without placing a barrier so that the next data collected with barriers can be compared to it and calculate the transmission loss. The sound level meter is placed 1 meter away from the front wall. The sound pressure level (SPL) meter is set to 1/3 octave bands to capture A-weighted frequency. The SPL meter is switched on and it measures the noise level in decibel and shows in the display. The machine is kept on for about 20 seconds. Later, the data has been collected from the same distance for the same settings of the SPL meter but after placing different barriers in the front wall. The collected data for different conditions have been taken directly to the PC using USB cable and used for further analysis.

5.0. Results and discussions

Table A-1 in the Appendix shows the Sound pressure level data collected by using different types of barriers in the noise enclosure. The SPL data of in the table is an average of minimum of 7 sets of data measured for each barrier so that the random error possibility is minimized. An interesting thing to notice from the table is that the reduction in SPL is quite high at both lower and higher frequencies and minimum for a frequency near 80Hz to 100Hz. This indicates that the frequency is the first natural frequency of the corresponding barrier. Also, it can be observed that at low frequencies from 50Hz to 630Hz the SPL reduction, which is actually the transmission loss, is very low compared to the higher frequency region ranging from 1000Hz to 20000Hz. This proves that for any type of barrier, the low frequency noise attenuation will always be lower than that of high frequency noise attenuation.

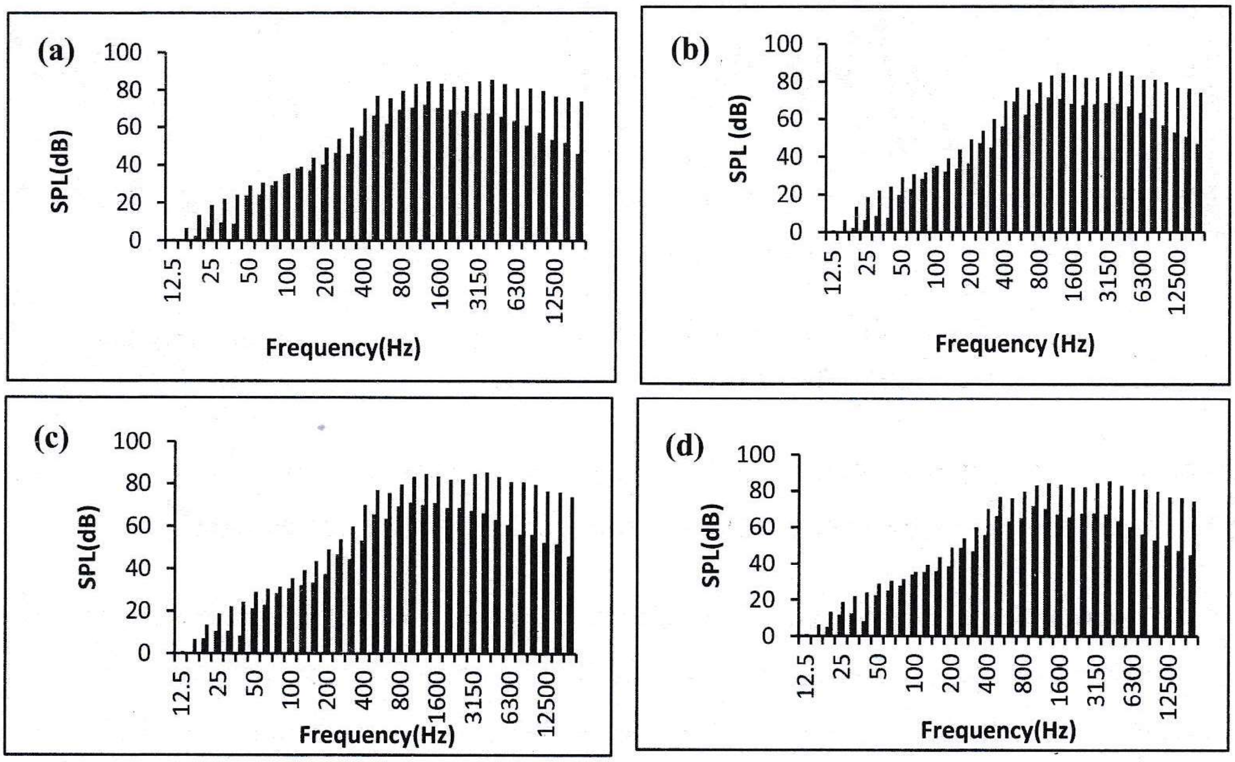


Fig. 6. SPL data for (a) Glass wool-backed plywood barrier; (b) Foam-backed plywood barrier; (c) Plywood-glass wool sandwich barrier; (d) Plywood-PE foam sandwich barrier.

Fig. 6. shows the sound pressure level data collected for different types of barriers and compared with that of no barrier condition. Here, it has focused on the frequency from 100Hz to 500Hz since this is the machine induced low frequency noise and hard to attenuate. Also, we are not ignoring the other high frequency noises since it requires a barrier which is suitable for both cases.

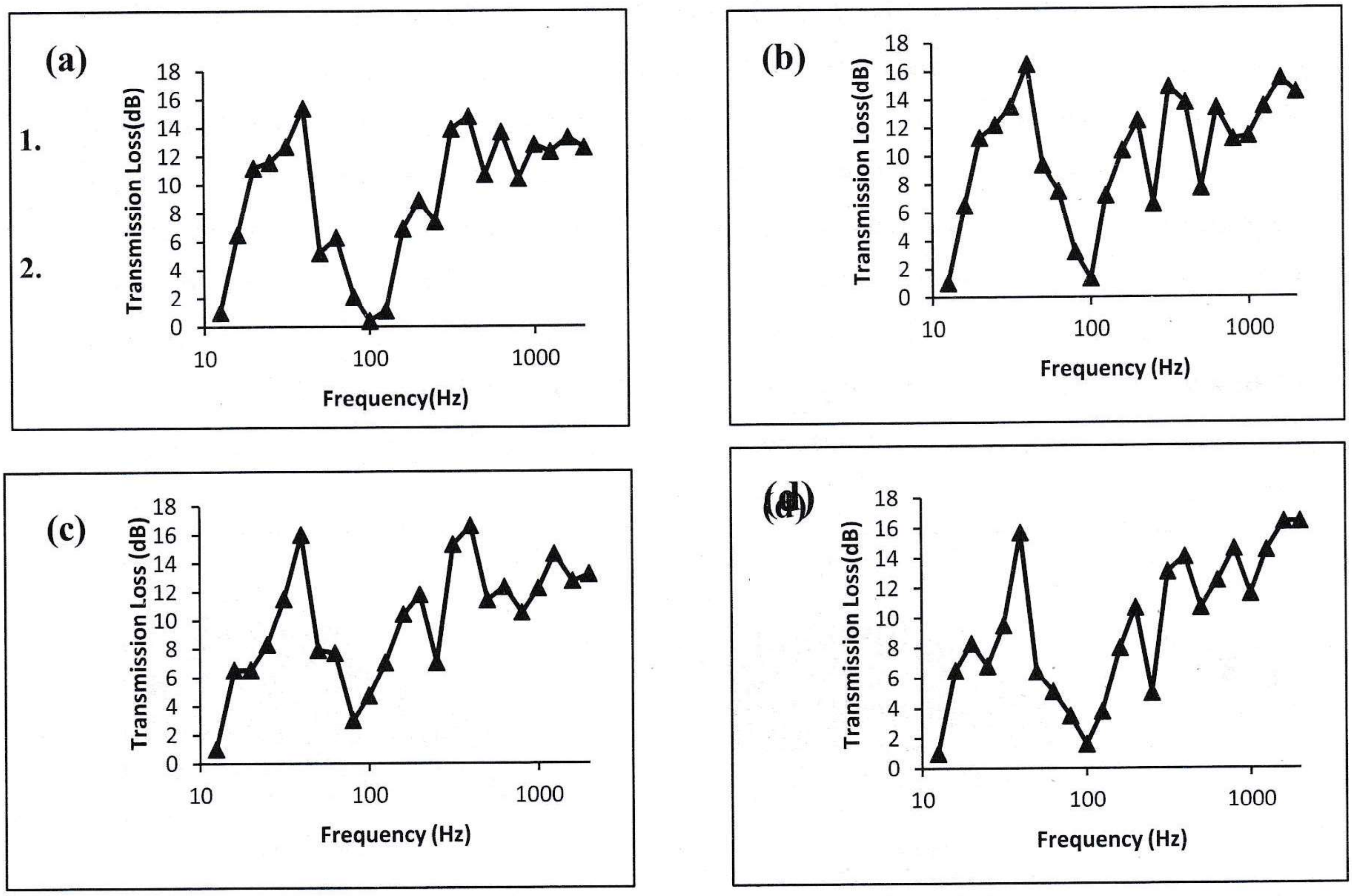


Fig. 7. Transmission Loss characteristics for (a) Glass wool-backed plywood barrier; (b) Foam-backed plywood barrier; (c) Plywood-glass wool sandwich barrier; (d) Plywood-PE foam sandwich barrier

Fig. 7. shows the transmission loss characteristics for different barriers. From this figure, it is evident that for each case, there is a frequency where the transmission loss is minimum and usually it has been seen to be near 100Hz to 125Hz, which is most probably the natural frequency of the barrier where attenuation is usually the lowest.

Fig. 8. is presented to compare results for different barriers within the range of mechanical noise. Machine induced noise is usually within the range 100Hz to 500Hz. It is obvious that at the lowest frequency, 100Hz, transmission loss is the lowest. But that is not true in case for the highest frequency, 500Hz where the transmission loss again starts decreasing gradually.

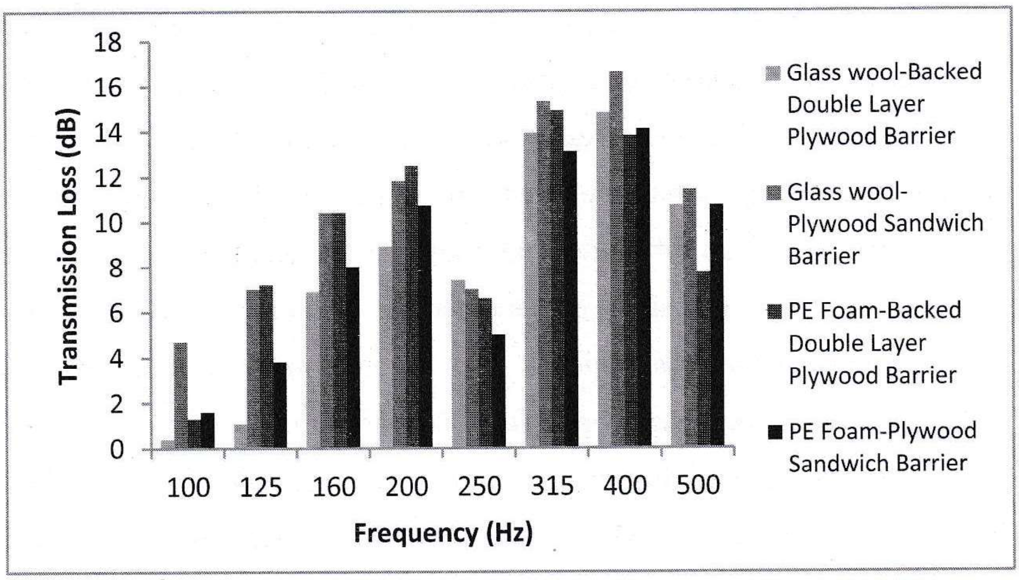


Fig. 8. Comparison of Transmission Loss for all barriers

From this figure, it is found that the transmission loss is the highest for Glass wool-Plywood sandwich barrier except for at frequency 200Hz, where PE Foam backed double layer plywood barrier has better performance. Both Glass wool- Plywood sandwich barrier and PE Foam backed double layer plywood barrier have good attenuation properties throughout the frequency range. On the other hand, Glass wool backed double layer plywood barrier has the minimum transmission loss amongst the four types.

It is also interesting to see that in case of glass wool, the double layer barrier is less efficient compared to sandwich barrier, whereas for PE foam, from 125Hz to 400Hz, double layer is better performing compared to sandwich barriers. However, for higher frequencies, the sandwich barriers are mostly more efficient for both glass wool and PE foam.

6.0. Conclusions

From this experimental study, the following conclusions may be drawn:

i. Transmission loss first rises and then falls almost linearly for 40-63Hz indicating that the later region is stiffness controlled. From 63Hz-1000Hz a fluctuation of transmission loss occurs which resembles the region is resonance controlled. According to transmission loss characteristic curve for a homogenous barrier the next region is mass controlled where transmission loss increases linearly with increase of frequency. In this experiment, higher frequency regions have not been considered. Therefore, the data between 10Hz to 2000Hz is only plotted here, which does not show the characteristics of the mass controlled region. So, it can be assumed that the low frequency region may not follow the mass law characteristics of transmission loss, which it does at much higher frequencies. Also, the possibility of observing coincidence region is not possible from the data representation of our study since it occurs at a much higher frequency range, which is out of scope of this study.

ii. At lower frequencies, Glass wool- plywood sandwich panel works the best, PE foam backed double layer plywood barrier also works considerably well. The other ones have not been found consistent in case of low frequency noise attenuation. This finding is very crucial as the objective of this experiment is to find which barrier has better performance in low frequency machine noise (100-500Hz).

iii. At higher frequencies, however, the performance of Glass wool-plywood sandwich is the best. PE-foam sandwich barrier also works satisfactorily. Glass wool-backed plywood panel and PE-foam backed plywood panel also gives good performance but single and double layer plywood panel has not provided much of attenuation. From this, it may be concluded that if attenuation of both low frequency and high frequency noise is required, cotton-plywood sandwich barrier is recommended since it does well in a wider range compared to double layer plywood barrier.

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Appendix

Table A-1: Transmission Losses for different barriers.

Frequency	No Barrier Condition	Glass wool-Backed Double Layer Plywood Barrier	Glass wool-Plywood Sandwich Barrier	PE Foam-Backed Double Layer Plywood Barrier	PE Foam-Plywood Sandwich Barrier
12.5	1	0	0	0	0
16	6.5	0	0	0	0
20	13.5	2.3	7	2.2	5.2
25	18.7	7.1	10.4	6.5	11.9
31.5	22.2	9.5	10.7	8.7	12.7
40	24.2	8.8	8.2	7.7	8.5
50	29.1	23.9	21.2	19.7	22.7
63	30.5	24.2	22.8	23	25.4
80	31.5	29.4	28.5	28.3	28
100	35.4	35	30.7	34.1	33.8
125	39.3	38.2	32.3	32.1	35.5
160	43.8	36.9	33.4	33.4	35.8
200	49.1	40.2	37.3	36.6	38.4
250	53.8	46.4	46.8	47.2	48.8
315	59.9	46	44.6	45	46.8
400	69.9	55.1	53.3	56.1	55.8
500	76.9	66.2	65.5	69.2	66.2
630	75.7	62	63.4	62.3	63.2
800	79.6	69.2	69.1	68.4	65
1000	83.1	70.3	70.9	71.7	71.5
1250	84.4	72.1	69.8	70.9	69.9
1600	83.4	70.1	70.7	67.9	67
2000	81.8	69.2	68.6	67.3	65.4
2500	82.1	68.7	68.5	67.7	67.4
3150	84.4	67.4	67.3	68.4	67.7
4000	85.3	67.3	66.1	68	67
5000	83.1	65.6	63	66.4	63.4
6300	80.9	63.5	60.6	63	60.2
8000	80.8	60.9	56.1	60.4	56
10000	79.6	57.2	56.1	56.5	53
12500	76.6	53.4	52.6	52.7	50
16000	76.1	51.8	51.9	50.4	47.3
20000	74	46	46.2	46.9	44.8