# Vibro-Acoustics Analysis for Prediction of Disturbance/noise in Machine Workshop

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*Abstract*—**A noise source can be very complex in nature. In noise control engineering an essential first step is to identify the strongest contributing noise sources. There are many tools available for predicting noise levels in industrial workrooms. These comprise simple theoretical formulae or empirical models as well as more complex methods of image or ray-tracing approaches. These more complex methods, such as the ray-tracing approaches, are proven prediction methods, but they involve considerable calculation times. Simple formulae and empirical models involve reduced computation times, at the cost of reduced performance.**

***Keywords: Noise Sources, Octave bands, Sound propagation curve***

## I. INTRODUCTION

Most machinery and manufacturing processes generate noise as an unwanted by-product of their output. Basic sources of industrial noise can generally be classified into one of the four groups. They are, continuous machinery noise, high-speed repetitive actions that create intense tonal sounds, flow induced noise, and the impact of a working tool on a work piece. Some typical examples of noise sources in the industrial environs include the combustion processes associated with furnaces, impact noise associated with punch presses, motors, generators and other electro-mechanical devices, unbalanced rotating shafts, gear meshing, gas flows in piping systems, pumps, fans, compressors, etc. There are only few basic noise producing mechanisms, and recognizing this allows for a systematic approach to be adopted. As an example, for a very noisy punch press, noise originates from several basic sources such as metal to metal contact, gear meshing, and high velocity air.

## II. LITERATURE REVIEW

Noise can have a wide range of effects on people, depending upon the level and on the situation in which people are exposed. The effects of noise are seldom catastrophic and are often only transitory, but the adverse effects can be cumulative with prolonged or repeated exposure. High-level sounds occurring in heavy industries and similar situations can damage hearing; lower level sounds occurring in community context cause significant annoyance and can potentially damage non-auditory health. Noise sometimes causes constriction of blood vessels and lead to hypertension. It affects physiological and psychological health by generating stress through annoyance and by disturbing rest and sleep. In addition, noise can interfere with the teaching and learning process, disrupt performance of certain tasks and increase the incidence of antisocial behavior. Short term exposure to continuous sounds at lower sound levels (than 140 dB) can lead to temporary loss of hearing sensitivity, known as temporary threshold shift (TTS). It occurs because of fatigue of the sensitive hair cells in the inner ear. Prolonged exposure at similar sound levels can cause permanent threshold shift and can lead to a permanent loss of hearing sensitivity associated with progressive atrophy of the sensitive hair cells in the inner ear. The three most important factors are the noise level, the total time of exposure to the noise and individual susceptibility [4].The results of air-conditioned noise in classrooms have been discussed by Lilly [5].He showed that the noise reduces speech intelligibility. A reduction in the overall learning capacity of the students was observed. However, people more or less are relatively unconcerned about their effects. In another study [6] it is found that in industries, very little attention is given to protect the hearing of the worker population, as only 25% of the workers who need hearing protection (28.5% of the surveyed population) habitually wear hearing protection leading to hearing loss in 7% of the working population. High frequency threshold shifts were found to be common among young workers, under 35 years, with over 1/4 of these workers experiencing a noise-induced permanent threshold shift (NIPTS) greater than 30 dB. Zhang Bangjun, Shi Lili and Di Guoqing [7] found in his study that under the same acoustic environment, when the source of noise can be seen the corresponding noise annoyance is higher. Noise can also constitute a serious threat to mechanical and electrical systems by producing excessive vibration, stress, fatigue damage or malfunction. It adversely affects industrial product quality, marketability and commercial competitiveness.

## III. REGULATIONS

Most countries have enacted some form of regulations or statutes regarding noise. The regulations often include performance standards for motor vehicles and the acoustic interaction among residences, commercial businesses, and heavy industries. A number of organizations have developed methods and policies for community noise measurement, recommendations for land use planning, general guidelines for noise mitigation, and various interpretations of noise effects on communities. In order to limit high level occupational noise, maximum permissible occupational noise exposure limit in the range of 90–85 dB (A) $L_{eq}$ for 8 h/day (40 h/week) have been allowed by the International Standards Organization (ISO) [8]. Also these environmental noise regulations are getting more and more demanding. Hence factories are facing the problem of reducing the noise to diminish their acoustic pollution. Hence there is a need to control the noise. A systematic approach to a noise control problem should always involve three stages.

**A:** analysis of the problem and identification of the noise sources.

**B:** an investigation as to whether source modification is possible.

**C:** recommendations for appropriate modification.

The first stage involves defining the problem, identification of noise sources and establishing acceptable limits and restraints.

## Modeling of Sound Fields in Industrial Workrooms

### Relationship between sound pressure levels at two points in free-field sound propagation for point sources

A sound source can generally be modeled as a point spherical sound source if its diameter is small compared with the wavelength that it generated, or if the measurement position is at a large distance away from the source. For a sound source of power *W,* sound intensity at distances $R_1$ and $R_2$ is given by

$I_{R1} = \frac{W}{4\pi R_1^2}$ and $I_{R2} = \frac{W}{4\pi R_2^2}$ respectively. Hence,

$$L_{I_{R1}} = 10\log_{10}\left(\frac{W}{4\pi R_1^2} \frac{1}{I_{ref}}\right), \text{ and} \qquad (1)$$

$$L_{I_{R2}} = 10\log_{10}\left(\frac{W}{4\pi R_2^2} \frac{1}{I_{ref}}\right), \qquad (2)$$

Rearranging $L_{I_{R2}}$,

$$L_{I_{R2}} = L_{I_{R1}} - 20\log_{10}\left(\frac{R_2}{R_1}\right) \qquad (3)$$

Hence for far-field conditions, assuming $L_I \approx L_P$,

$$L_{P2} = L_{P1} - 20\log_{10}\left(\frac{R_2}{R_1}\right) \qquad (4)$$

### Combination of several sources

Total intensity produced by several sources as illustrated in Figure 1 is given by

$$I_T = I_1 + I_2 + I_3 + \ldots\ldots\ldots \qquad (5)$$

where intensity levels are known as ( $L_1, L_2, \ldots\ldots$)

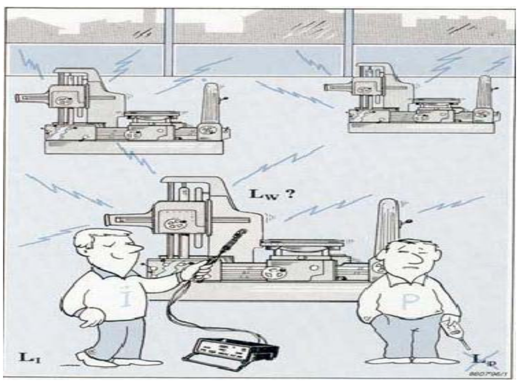


Figure 1

If intensity levels of each of the N sources is same,

$$L_T = 10Log\left(N 10^{\frac{L_1}{10}}\right) \qquad (6)$$

Which is equal to $L_T = 10LogN + L_1$ (7)

### Addition of coherent sound pressures

Often, combinations of sounds from many sources contribute to the observed total sound. In general, the phases between sources of sound will be random and such sources are said to be incoherent. However, when sounds of the same frequency are to be combined, the phase between the sounds must be included in the calculation.

For two sounds of the same frequency, characterized by mean square sound pressures $p_1{}^2{}_{rms}$ and $p_2{}^2{}_{rms}$ and phase difference $\beta_1 - \beta_2$, the total mean square sound pressure is given by the following expression:

$$p_t{}^2{}_{rms} = p_1{}^2{}_{rms} + p_2{}^2{}_{rms} + 2[p_1 p_2]_{rms}\cos(\beta_1 - \beta_2) \qquad (8)$$

### Addition of incoherent sound pressures (logarithmic addition)

When bands of noise are added and the phases are random, the limiting form of the previous equation reduces to the case of addition of incoherent sounds; that is

$$p_t{}^2{}_{rms} = p_1{}^2{}_{rms} + p_2{}^2{}_{rms} \qquad (9)$$

Incoherent sounds add together on a linear energy (pressure squared) basis. The procedure accounts for the addition of sounds on a linear energy basis and their representation on a logarithmic basis.

### Sound-propagation Curve

Sound – propagation curve SP (r) is the variation with distance r from a point source of sound, the pressure level $L_p$ (r) minus the source sound level $L_w$ [11].

Hence, SP (r) = Lp (r) - $L_w$ in dB (10)

Since the sound power level of a machine is a constant, the variation of sound propagation curve with distance r , in actual, is the variation of sound pressure level.

### Normalized Sound-propagation Curve

Normalized sound-propagation curve NSP (r) can be defined as

NSP(r) = SP (r) – SP (1), in dB (11)

Normalized sound-propagation curve NSP (r) gives the variation of sound pressure level as a function of distance r from the source, with 0 dB at 1 meter distance from the source. So with a common value at 1 meter of distance, it is easy to compare the variation of sound pressure level with r for different set of values.

### Sound Absorption Coefficient

Sound absorption coefficient, $\alpha$, of a surface is the ratio of absorbed to incident sound intensities on the surface. Values of $\alpha$, for different materials, are shown in Table 1. Values of sound absorption coefficient for concrete floor, brick wall, and window glass and plywood panel are taken from different acoustics textbooks and that for e-board is taken from manufacturer's site.

**Table 1:** Sound absorption coefficient of different materials.

| | Octave band central frequency (Hz) | | | | | | |
|---|---|---|---|---|---|---|---|
| | 125 | 250 | 500 | 1000 | 2000 | 4000 | 8000 |
| Concrete floor | 0.01 | 0.01 | 0.02 | 0.02 | 0.02 | 0.02 | 0.02 |
| Brick wall | 0.01 | 0.01 | 0.01 | 0.02 | 0.02 | 0.02 | 0.02 |
| Window glass | 0.25 | 0.25 | 0.18 | 0.12 | 0.07 | 0.05 | 0.04 |
| Eboard ceiling | 0.18 | 0.20 | 0.20 | 0.20 | 0.22 | 0.22 | --- |
| Plywood panel | 0.28 | 0.22 | 0.17 | 0.09 | 0.10 | 0.11 | 0.08 |

**Sound-propagation Curve by Sabine Theory**

In a completely diffused sound field, following Sabine theory [12], the sound pressure level in a room, produced by a sound source at a given point at distance $r$ from the source is given by,

$$L_P(r) = L_W + 10\log\left(\frac{1}{4\pi r^2} + \frac{4}{R}\right), \quad \text{in dB} \qquad (12)$$

), in dB (13)

Here S is the total surface area of the room and $\alpha_{avg}$ is the average sound absorption coefficient of the room surface. Assuming each surface , $S_n$, of the room has a different sound absorption coefficient, $\alpha_n$ , the average absorption coefficient, $\alpha_{avg}$, in the room is given by,

$$\alpha_{avg} = \frac{S_1\alpha_1 + S_2\alpha_2 + \ldots + S_n\alpha_n}{S_1 + S_2 + \ldots + S_n}, \qquad (14)$$

Sound propagation curve, $SP(r)$, for the diffused field, following the Sabine theory is $SP(r) = 10\log\left(\frac{1}{4\pi} + \frac{4}{R}\right)$, in dB

(15)

**Sound-propagation Curve by Thompson Model**

Thompson has proposed a modification to the expression describing steady-state levels according to diffuse field theory to allow its application to irregularly proportioned workrooms [13]. Sound-propagation curve by Thompson model is given by

$$SP(r) = 10\log\left[\frac{\exp(-mr)}{4\pi r^2} + \left(\frac{4V}{rS}\right)\frac{1}{((\alpha_{avg}S_w + K) + 4mV)}\right] + 10\log\left[\frac{(T+460)}{527}\,\frac{30}{BP}\right] \qquad (16)$$

In which $r$ is the between source and receiver; $\alpha_{avg}$ is the average room absorption coefficient; $S_w$ is the wall surface area; $m$ is the air absorption coefficient; $V$ is the room volume; S is the total surface area of the room, $T$ is the temperature, and $BP$ is the barometric pressure. $T$ and BP may be assumed to be constants for measurements under same conditions. Hence the variation of sound propagation curve and the variation of sound pressure level will be same.

**Sound-propagation Curve by Sergeyev Model**

Osipove, Sergeyev and Shubin developed a model, which predicts octave band sound levels [14]. Sound-propagation curve is determined by the formula,

$$SP(r) = 10\log\left[\frac{1}{2} + \frac{(1-\alpha_{eff})(r+W)J(\alpha_{eff},\rho)}{HW(r+H)}\right], \text{ in dB} \qquad (17)$$

With

$$J(\alpha_{eff},\rho) = 0.1/[\alpha_{eff} + \rho^2\exp(0.065\rho)] \qquad (18)$$

In which $\rho = -\left[\frac{r \times S \times \ln(1-\alpha_{eff})}{4V}\right]$; $r$ is the distance between source and the receiver. And $\alpha_{eff}$ is effective absorption coefficient. Effective absorption coefficient was determined for typical empty workrooms and for fitted workrooms. For metalworking industry, effective absorption coefficient, averaged over frequency range 0 Hz to 4000 Hz, is determined to be 0.34. V is the room volume, S is the total surface area of the room, W is the room width and H is the room height. Hodgson [15] found in his study that both the Thomson model and the Sergeyev model work well in empty workrooms. The Thompson model overestimates the levels if workroom is acoustically treated and the Sergeyev model underestimates the levels if workroom is fitted.

## IV. CASE STUDY

**Different Parameters for Calculating Normalized Sound-propagation Curves**

**The dimensions of the ME workshop, where sound pressure level readings were taken are as follows:**

1. Length of the room = 29.4 meters
2. Width of the room = 10.5 meters
3. Height of the room = 3.7 meters
4. Total surface area the room = 912.66 square meters
5. Volume of the room = 1142.19 cubic meters
6. There are a total of 14 glass windows in the workshop, out of which dimensions of 8 windows are (1.82 × 1.65) square meters each and dimensions of remaining 6 windows are (1.21 × 1.65) square meters each. Eighteen glass surfaces are at upper portion of workshop of dimensions (0.6 × 0.6) square meters each. Glass surface attached with plywood is of dimension of (10.5 × 0.34) square meters. Total area of glass windows, which remains open, is 6.127 square meters. Total area of glass surface, including window glass, which remains closed, is 40.03 square meters.
7. Total surface area of plywood on the two sides of the workshop is 56.40
8. Total open area, including open part of window, two gates, one shutter and upper part of plywood is 34.06 square meters.
9. Surface area of ceiling is 308.70 square meters.

10. Surface area of floor is 308.70 square meters.

11. Surface area of plastered wall is 152.47 square meters.

12. Total surface area of wall, including plywood panel is 208.87 square meters.

**Average Sound Absorption Coefficient of the ME Workshop and Air Absorption Coefficient**

Average sound absorption coefficient $\alpha_{avg}$ for the room is given by,

$$\alpha_{avg} = \frac{S_1\alpha_1 + S_2\alpha_2 + S_3\alpha_3 + S_4\alpha_4 + S_5\alpha_5 + S_6\alpha_6}{S_1 + S_2 + S_3 + S_4 + S_5 + S_6}; \quad (19)$$

Where $S_1$, $S_2$, $S_3$, $S_4$, $S_5$, $S_6$ are the total surface area of the floor, ceiling, wall, windows glass , plywood panel and open space respectively. $\alpha_1$, $\alpha_2$, $\alpha_3$, $\alpha_4$, $\alpha_5$, $\alpha_6$ are the sound absorption coefficient of the floor, ceiling, wall, window glass, plywood panel and open space respectively. Floor with tiles is approximated to be concrete floor. and absorption coefficient of all the open area is assumed to be one. Average sound absorption coefficient is calculated in octave bands by putting all the surface areas and their sound absorption coefficients (in that octave band) in the equation ( 19 ). For calculating the average sound absorption coefficient at 8000 Hz, the value of sound absorption coefficient for e-board ceiling is taken to be that at 4000 Hz. Values of the calculated sound absorption coefficients are shown in Table 2.

**Table 2:** Average sound absorption coefficient in different octave bands.

| Octave band centre frequencies (Hz) | 125 | 250 | 500 | 1000 | 2000 | 4000 | 8000 |
|---|---|---|---|---|---|---|---|
| $A_{avg}$ | 0.1320 | 0.1350 | 0.1340 | 01264 | 0.1317 | 0.1314 | 0.1310 |

Values of air absorption coefficient, *m,* at 303 ° K of temperature and 70% humidity for different frequencies are given in Table 3.

**Table 3:** Air absorption coefficients at different frequencies.

| Frequency(Hz) | 2000 | 4000 | 6000 | 8000 | 10000 | 12500 |
|---|---|---|---|---|---|---|
| Air absorption coefficient, *m* | 0.00205 | 0.00517 | 0.00957 | 0.0150 | 0.0210 | 0.030 |

Sound absorption coefficient $\alpha_{avg}$ used in Thompson model and Sergeyev model is the octave band sound absorption coefficient. Value of the sound absorption coefficient which is used to calculate the total (broadband) *NSP(r)* is the average value of the octave band sound absorption coefficients calculated in 125 – 4000-Hz octave bands. In the present case it is 0.1318. Surface materials in the workshop are concrete, glass brick wall, e-board and plywood. For these materials variation of the sound absorption coefficient with frequency is less and except for the tool grinders and wood sawing machine, sound due to most of the machines is concentrated in the 0 - 3000 Hz frequency range. Hence average sound absorption coefficient averaged over 125 – 4000-Hz octave bands can be used for the calculation of *NSP(r)* with a very marginal error, which is acceptable. At low frequencies (<2000 Hz) variation of air absorption coefficient with frequency is less, compared to that at higher frequencies. Magnitude of air absorption coefficient at these low frequencies is also very less. Sound due to most of the machines is concentrated in the low frequency region. Hence to calculate the total (broadband) *NSP(r)* , value of the sound absorption coefficient is taken to be that at 2000 Hz. Sound due to tool grinders and wood sawing machines is distributed all over the frequency region (0 – 20 KHz) and the air absorption coefficient is more at higher frequencies. Hence the value of NSP*(r)* will be overestimated for the tool grinders and the wood sawing machines.

## V. Methodology of Measurement

Modeling of the mechanical workshop has been done to find out the total (broadband) and octave-band normalized sound propagation curve for the machines. Different models have been investigated to choose the best model that can predict the normalized sound propagation curve accurately. For modeling of sound fields in the workshop, sound pressure levels are measured by Bruel & Kjaer's sound level meter - mediator 2238. Broadband sound pressure level and octave band sound pressure levels are measured. For calculating the normalized sound propagation curve for a machine, measurements are taken when only the machine under consideration is running. Firstly, measurement is taken at 1 meter distance from the machine. Secondly, in order to find out its variation with source distance *r,* consequent measurements are taken at increasing distance of 0.6 meters from the machines. Same procedure is repeated for some of the machines in the workshop.

## VI. Results and Analysis

At low frequencies (<2000 Hz) variation of air absorption coefficient with frequency is less, compared to that at higher frequencies. Magnitude of air absorption coefficient at these low frequencies is also very less. Sound due to most of the machines is concentrated in low frequency region. Hence to calculate the total (broadband) NSP ( r ) value of air absorption coefficient is taken to be that at 2000 Hz. Sound due to tool grinders and wood sawing machines is distributed all over the frequency region (0 – 20 KHz) and the air absorption coefficient is more at higher frequencies. Hence the value of NSP (r) will be overestimated for the tool grinders and the wood sawing machines.

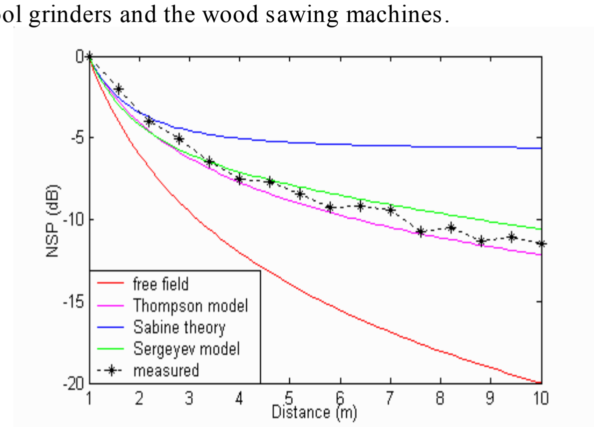


**Figure 2:** Measured values and predicted values of normalized

sound propagation curve by different models for milling machine (M11).

Figure 2 shows the measured values and the predicted values of the normalized sound propagation curve. Values predicted by assuming free field condition are underestimates of the measured values, while the values predicted by Sabine theory are overestimates of the measured values. Thompson model and Sergeyev models are very close to the actual measured values, and the actual values lie in between the values predicted by these two models. At lower frequencies, attenuation of sound due to air is less, while it is more at higher frequencies. Hence it is important to consider the attenuation of sound due to air, because sound due to some of the machines is distributed over the high frequencies also. The Sergeyev model does not account for the attenuation of sound due to air. Hence if this model is giving good results for machines having sound concentrated in low frequency region, it does not give good results for machines which have high sound levels at higher frequencies also. Results predicted by Thompson model are underestimates of the actual measured values. With increase in the absorption of sound due to surrounding surface area, there should be a corresponding decrease in the actual value of the normalized sound propagation curve. This indicates that there is overestimation of the term $\alpha_{avg} S_w$

in the expression for the sound propagation curve. Areas of shutter gate and that at upper portion of plywood are assumed to be open spaces with their absorption coefficient values to be 1. Another source of over estimation of the value of $\alpha_{avg} S_w$ may be due to the assumption of treating these spaces to be open spaces. This may be attributed to the fact that the entire energy incident upon it may not be absorbed or pass through the open space and a part of the energy in fact might be reflected back. There is another possibility of the Thompson model being inaccurate.

In order to predict the sound propagation curve accurately, it is proposed to introduce a correction factor K *to* account for this error in estimating the value of $\alpha_{avg} S_w$. Thus by introducing a correction factor $K$, the Tho mpson model becomes,

$$SP(r) = 10\log\left[\frac{\exp(-mr)}{4\pi r^2} + \left(\frac{4V}{rS}\right)\frac{1}{\left((\alpha_{avg}S_w + K) + 4mV\right)}\right] + 10\log\left[\frac{(T+460)}{527} + \frac{30}{}\right] \quad (20)$$

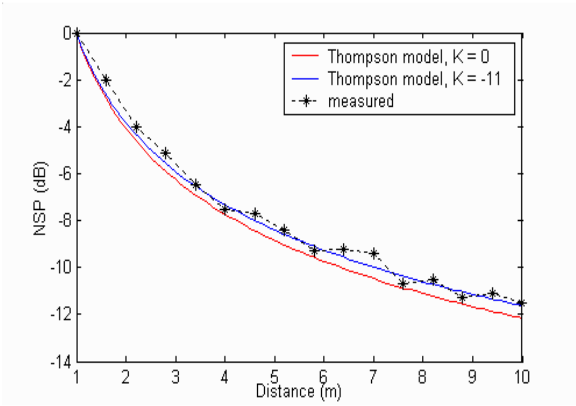


**Figure 3:** Measured values and predicted values of normalized sound propagation curve for milling machine (M11) using $K$ = -11.

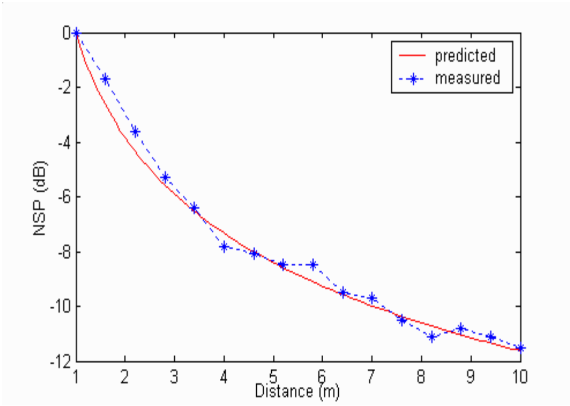


**Figure 4:** Measured values and predicted values by Thompson model normalized sound propagation curve for lathe (M5).

With a value of K = -11, the prediction from the Thompson model and the measuredvalues match. Thus one can conclude that the overestimation of the value of $\alpha_{avg} S_w$ in the Thompson model can be corrected using a constant $K$ = -11.

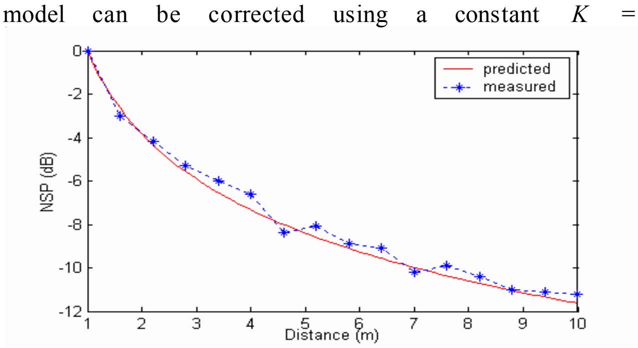


**Figure 5:** Measured values and predicted values by Thompson model of normalized sound propagation curve for shaper (M8).

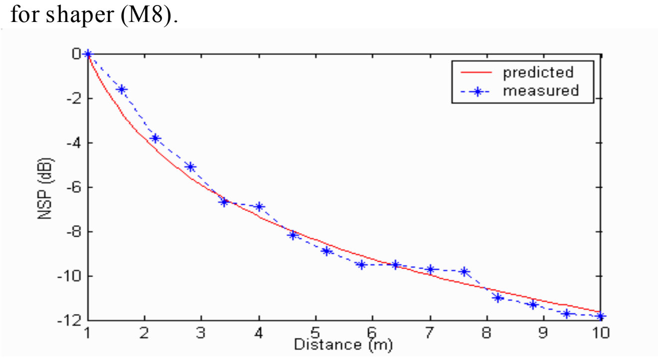


**Figure 6:** Measured and predicted values by Thompson model of normalized **sound** propagation curve for milling machine (M10).

Figures 4, .5 and 6 show the measured values and the predicted values of normalized sound propagation curves for the lathe, shaper and the milling machines. The Thompson model is showing good agreement with the measured values. Reason of deviation of the measured values from the predicted values may be because of the following reasons 1. Sound fluctuates with time and the measurements are not taken simultaneously at all the points, 2. Measurements which are taken near some fitting surfaces, give more values than the expected one. It may be due to reflections of the sound from the surface. Hence measurements are taken along a line which has minimum number of nearby reflecting surfaces.

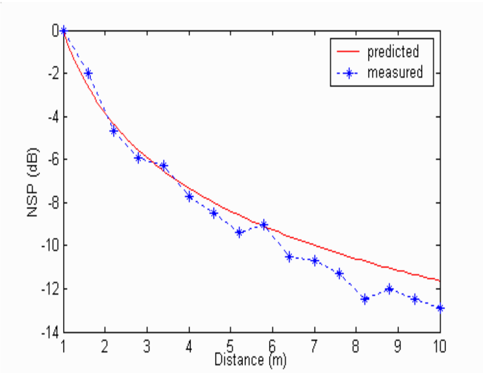


**Figure 7:** Measured values and predicted values by Thompson model of normalized sound propagation curve for tool grinder (M12).

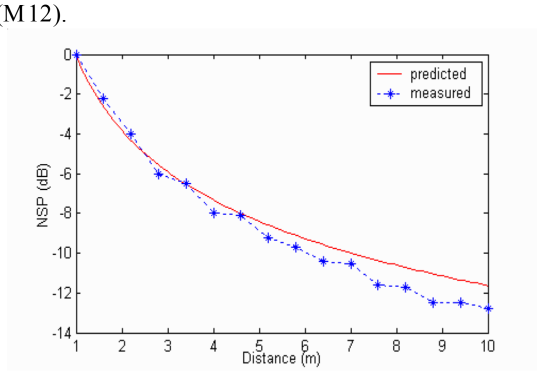


**Figure 8:** Measured values and predicted values by Thompson model of normalized sound propagation curve for wood sawing machine (M13).

Figures 7 and 8 shows the normalized sound propagation curve for tool grinder and the wood sawing machines respectively. The predicted values of the normalized sound propagation curves by the Thompson model are overestimates of the measured values in both the cases. For both tool grinder and wood saw machine, sound is distributed over higher frequencies also, which have significant contribution in total sound pressure level. Hence more attenuation of sound at higher frequencies might be the cause of overestimating the measured values.

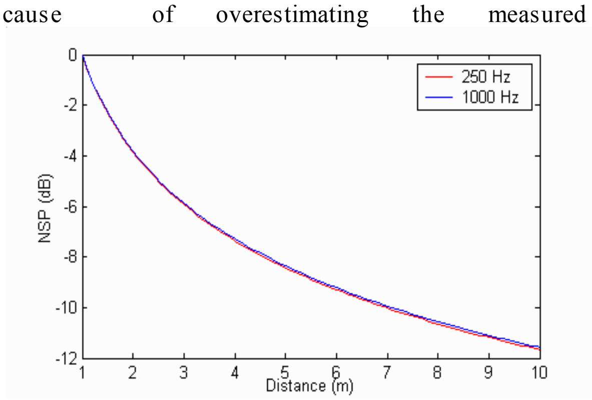


**Figure 9:** Normalized sound propagation curve by Thompson model at 250 Hz and 1000 Hz octave bands.

Figure 9 shows the normalized sound propagation curve, predicted by Thompson model for 250 Hz and 1000 Hz octave bands. From Table 2, for the 125 – 8000 – Hz octave bands, the value of the average sound propagation $\alpha_{avg}$ is minimum at 250 Hz and maximum at 1000 Hz. Hence the variation in $\alpha_{avg}$ over frequency range of 125- 8000 Hz is very less and its variation can be neglected for the calculation of the normalized sound propagation curve. Hence for calculating normalized sound propagation curve for a given workroom, according to Thompson model, the only parameter which is changing with frequency is the sound absorption coefficient *m* . The variation of sound absorption coefficient *m* with frequency *f* is shown in Table 3.

For $f > 2000$, these values can be approximated by empirical expression

$$m(f) = 1.1667 \times 10^{-4} + (6.6875 \times 10^{-7})\, f + (1.4896 \times 10^{-10})\, f^{2} \qquad (21)$$

The values of air absorption coefficient, with that predicted by equation 21 are in table 4.

**Table 4:** Air absorption coefficients and the predicted values at different frequencies

| *Frequency (Hz)* | 2000 | 4000 | 6000 | 8000 | 10000 | 12500 |
|---|---|---|---|---|---|---|
| *Air absorption coefficient, m* | *0.00205* | *0.00517* | *0.00957* | *0.0150* | *0.021* | *0.030* |
| *Predicted by expression 21* | *0.00205* | *0.00517* | *0.00949* | *0.0150* | *0.0217* | *0.0318* |

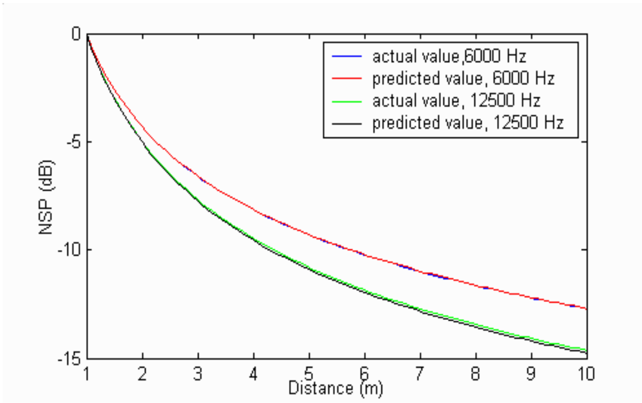


**Figure 10:** Normalized sound propagation curve by Thompson model by using actual and predicted values of air absorption coefficient at 6000 Hz and at 12500 Hz.

Figure 10 shows the normalized sound propagation curve with actual value of air absorption coefficient and that predicted by equation 21. The difference between the two is negligibly small. Hence it can be concluded that equation 21 can be used to estimate the value of air absorption coefficient at any frequency between 2000 and 12500 Hz accurately.

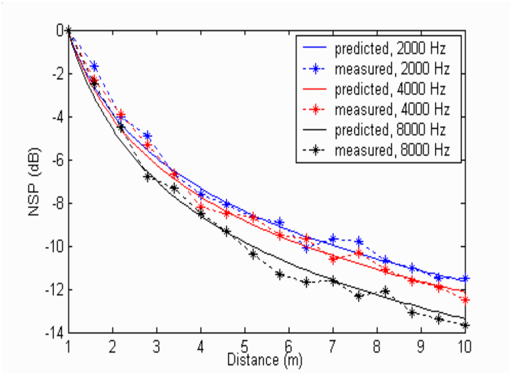


**Figure 11:** Measured values and predicted normalized sound propagation curve for tool normalized sound propagation curve grinder (M12).

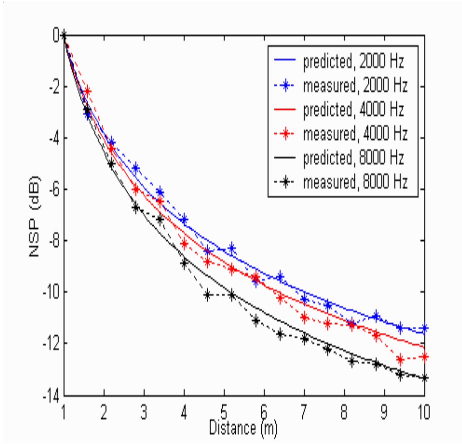


**Figure 12:** Measured values and predicted values by Thompson model of octave band values by Thompson model of octave band for wood saw machine (M13).

Figure 11 and 12 shows normalized sound propagation curve for 2000 – 8000-Hz octave bands. Due to higher attenuation of sound due to air at higher frequency bands, the predicted normalized sound propagation curve has progressively lower values at higher frequency bands. The corresponding measured values are also lower at higher frequency octave bands. Hence, for frequencies f > *2000* Hz, the Thompson model given in the equation (20) for the sound propagation curve can be used for finding the normalized sound propagation curve with the value of air absorption coefficient given by the equation (21). This will eliminate the procedure of calculating the average sound absorption coefficient in each octave band and finding the value of air absorption coefficient in that octave band. For a given workroom, the only variable in this expression is the air absorption coefficient, which depends upon only the frequency, thus the expression can be used for calculating any narrow band sound propagation curve by substituting the central frequency of that narrow band in the expression for air absorption coefficient $m$. The workroom, in which the study has been done, all the surface areas are acoustically hard and there was no acoustic treatment to reduce the noise level. Sound absorption coefficient of acoustically hard surfaces like concrete and plastered walls do not change much with change in frequency, while this change in sound absorption coefficient for materials which are used for acoustic treatment, like foam, carpet and fiber glass, is more. Hence the above expression for calculating the normalized sound propagation curve may not be suitable for industrial workrooms which are acoustically treated.

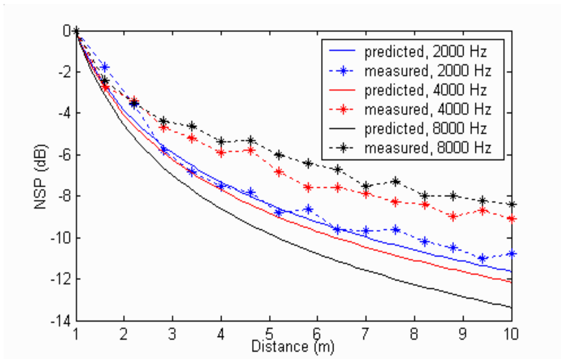


**Figure 13:** Measured values and predicted values by Thompson model of octave band normalized sound propagation curve for shaper (M8).

**Figure 14:** Measured values and predicted values by Thompson model of normalized sound propagation curve for octave band lathe (M5)

Figures 13 and 14 shows the measured and predicted normalized sound propagation curve for lathe and shaper respectively in 2000 – 8000-Hz octave bands. There is an underestimation of the value of NSP by the Thompson model. This is due to the presence of background noise. The difference between machine noise and background noise is decreasing with increasing frequencies. At distances close to machines, the noise is sufficiently higher than the background noise but as the distance between source and receiver increases, the effect of background noise increases. Hence the Thompson model should not be used for the situations when the difference between source noise and the background noise is less.

Sound pressure levels measured at one meter distance from different machines are shown in Table 5.

**Table 5:** Sound pressure levels measured at one meter distance from the machines

| Machine | SPL (dB) |
|---|---|
| Precision lathe (M1) | 82.5 |

| | |
|---|---|
| Centre lathe (M3) | 84.8 |
| Radial drilling machine (M7) | 78.0 |
| Shaper (M8) | 75.3 |
| Horizontal milling machine (M10) | 81.7 |
| Universal milling machine (M11) | 83.1 |
| Tool grinder (M12) | 92.4 |
| Wood sawing machine (M13) | 104.8 |
| Wood turning lathe (M14) | 74.2 |
| Pillar drilling machine (M15) | 76.9 |
| Column drilling machine (M16) | 75.2 |
| Surface grinder (M17) | 71.0 |
| Power hack saw (M18) | 72.3 |
| Wood planer (M19) | 89.4 |

## VII. CONCLUSIONS

Different models for normalized sound propagation curves are investigated . An improvement in the Thompson model is proposed by introducing a correction factor K. The results from the improved Thompson model are compared with the measured values and found to be more accurate. Prediction of sound propagation curve by the Thompson model is found to be better than the other models as the Thompson model incorporates the attenuation of sound due to air, which is important parameter for sound propagation at high frequencies. Improvements in the Thompson model are proposed, and the performance of the proposed model is studied by comparing the measured values and the predicted values. It allows one to predict the normalized sound propagation curve without calculating the average sound absorption coefficient and the air absorption coefficient in each frequency band. This is a major contribution in the present work.

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