Designing for Quieter Living

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The author's journey from a very humble beginning in a refugee camp during the time of the Partition (1947) to joining the Indian Institute of Science, Bengaluru, is recollected on page 368 under Personal Reflections.

Keywords
Noise, noise control, quieter living, noise measurement, acoustic enclosures, mufflers, silencers, speech interference, noise limits.

Quieter living requires knowledge of the nature of noise, its measurement, frequency analysis, implications of logarithmic addition (and subtraction) of decibel levels, noise limits prescribed by the Government, speech interference, etc. This tutorial article touches briefly on all these aspects, and goes on to the sound field in enclosed spaces, acoustics of partitions and barriers, design of acoustic enclosures and mufflers/silencers. Finally, general noise control strategies are listed with particular mention of the control of noise of the ubiquitous fans and blowers.

Noise is unwanted sound either because of its loudness or frequency characteristics. Excessive or prolonged exposure to noise may lead to several physiological effects like annoyance, headache, increase in blood pressure, loss of concentration, speech interference, loss of working efficiency, or even accidents in the workplace. Persistent exposure of a worker to loud noise in the workplace may raise his/her threshold of hearing.

1. Decibel Level

The human ear is a fantastic transducer. It can pick up pressure fluctuations of the order of 10^{-5}Pa to 10^{3}Pa; that is, it has a dynamic range of 10^{8}! Therefore, a linear unit of measurement is ruled out. Instead, a logarithmic unit of decibels has been universally adopted for measurements of Sound Pressure Level (SPL), Sound Intensity Level (SIL) and Sound Power Level (SWL). These are defined as follows [1–3]:

$$\mathrm{SPL} \equiv L_p = 10\ \log\ \frac{p_{\mathrm{rms}}^2}{p_{\mathrm{th}}^2} = 20\ \log\left(\frac{p_{\mathrm{rms}}}{2\times 10^{-5}}\right),\ \mathrm{dB} \tag{1}$$

$$\mathrm{SIL} \equiv L_I = 10\ \log\ \frac{I}{I_{\mathrm{ref}}} = 10\ \log\ \left(\frac{I}{10^{-12}}\right)\ ,\ dB \tag{2}$$

$$\mathrm{SWL} \equiv L_w = 10\ \log\ \frac{W}{W_{\mathrm{ref}}} = 10\ \log\ \left(\frac{W}{10^{-12}}\right),\ \mathrm{dB} \tag{3}$$

where 'log' denotes log to the base 10, and $p_{\mathrm{th}} = 2\times 10^{-5}$ Pa represents the threshold of hearing. This standard quantity represents the root-mean-square pressure of the faintest sound of 1000 Hz frequency that a normal human ear can just pick up.

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2. Frequency Analysis: Octave Band Filters

The human ear responds to sounds in the frequency range of 20 Hz to 20,000 Hz (20 kHz), although the human speech range is 125 Hz to 8000 Hz. Precisely, male speech lies between 125 Hz to 4000 Hz, and female speech is one octave higher, that is, 250 Hz to 8000 Hz.

The audible frequency range is divided into octave bands and 1/3rd octave bands. For an octave band, the lower frequency f_{l}, mean frequency f_{m} and the upper frequency f_{u} are related as

$$f_{\mathrm{u}}/f_{\mathrm{l}} = 2 \text{ and } f_{\mathrm{m}} = (f_{\mathrm{u}}\cdot f_{\mathrm{l}})^{1/2}\,. \tag{4}$$

Thus,

$$f_{\mathrm{l}} = \frac{f_{\mathrm{m}}}{2^{1/2}} = 0.707 f_{\mathrm{m}} \quad \text{and} \quad f_{\mathrm{u}} = f_{\mathrm{m}}\cdot 2^{1/2} = 1.414 f_{\mathrm{m}} \tag{5}$$

and the bandwidth of the filter $= (1.414 - 0.707)f_{\mathrm{m}} =$ 70.7 % of the mean frequency.

Persistent exposure of a worker to loud noise in the workplace may raise his/her threshold of hearing.

A frequency of 1000 Hz has been recognized internationally as the standard reference frequency, and the mid frequencies of all octave bands and 1/3rd octave bands have been fixed around this frequency.

Incidentally, the standard frequency of 1000 Hz happens to be the geometric mean of the human speech frequency range; that is, $1000 = (125 \times 8000)^{1/2}$. Standard octave filters are centred around the mean frequencies of 31.5, 63, 125, 250, 1000, 2000, 4000, 8000 and 16000 Hz.

3. Weighted Sound Pressure Level

The human ear responds differently to sounds of different frequencies. Extensive audiological surveys have resulted in weighting factors[1] for different purposes. These are shown in *Figure* 1.

The A-weighted network is now used exclusively in most measurement standards and the mandatory noise limits.

4. Logarithmic Addition, Subtraction and Averaging

The total sound power level (SWL) of two or more incoherent sources of noise may be calculated as follows:

[1] Human ear is not equally sensitive to all frequencies. This non-uniform frequency response is accounted through appropriate weighting factors as shown in *Figure* 1. A weighted sound pressure level L_{pA} is used for mandatory limits for environmental as well as industrial and automotive noise. C-weighting is preferred for measurement of high level noise.

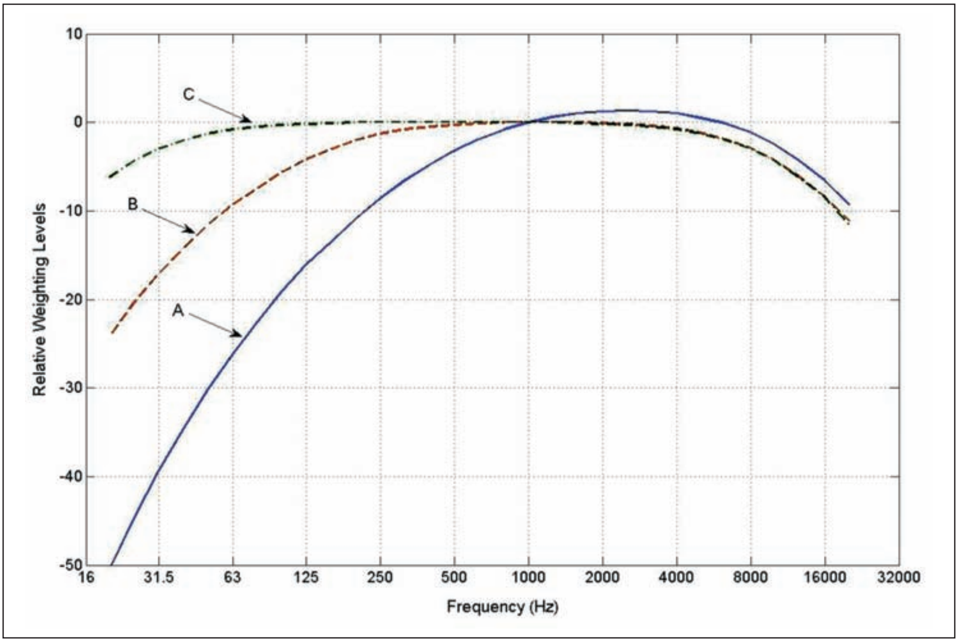

Figure 1. Approximate electrical frequency response of the A-, B-, and C-weighted networks of sound level meters [2].

$$W_t = \sum_{i=1}^{n} W_i \tag{6}$$

or

$$L_{w,t} = 10 \log \left[\sum_{i=1}^{n} 10^{0.1 L_{w,i}} \right] . \tag{7}$$

Here, n denotes the total number of incoherent sources, such as machines in a workshop or different sources of noise in an engine, etc. Similarly, the corresponding total SPL at a point is given by

$$L_{p,t} = 10 \log \left[\sum_{i=1}^{n} 10^{0.1 L_{P,i}} \right] . \tag{8}$$

Incidentally, equations (6–8) would also apply to the logarithmic addition of SPL or SWL of different frequency bands in order to calculate the total level. The logarithmic addition of sound power levels or pressure levels has some interesting implications for noise control. It may easily be verified from (7) and (8) that $100 \oplus 100 = 103\,\text{dB}$, $100 \oplus 90 = 100.4\,\text{dB}$ and $x \oplus x = x + 3\,\text{dB}$.

Similarly, 10 identical sources of x dB would add up to $x + 10\,\text{db}$. Perception wise, a 3 dB increase in SPL is hardly noticeable; a 5 dB increase in SPL is clearly noticeable; and a 10 dB increase in SPL appears to be twice as loud. Similarly, a 10 dB decrease in SPL would appear to be half as loud, indicating 50 % reduction in SPL.

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The concept of addition can also be extended to the averaging of the sound pressure level in a community location. Thus, the equivalent sound pressure level during a time period of 8 hours may be calculated as an average of the hourly readings; that is,

$$L_{p,8\text{h}} = 10 \log \left[\frac{1}{8} \sum_{i=l}^{8} 10^{0.1 L_{p,i}} \right], \ \text{dB}. \tag{9}$$

This averaging is done automatically in an integrating sound level meter or dosimeter used in factories in order to ensure that a worker is not subjected to more than 90 dBA of equivalent sound pressure level during an 8-hour shift. Here, dBA denotes A-weighted decibels (see *Figure* 1). Similarly, one can measure L_{d}, the day time average (6 AM to 9 PM) and L_{n}, the night time average (9 PM to 6 AM). Making use of the fact that one needs a quieter environment at night, the day-night average (24-hour average) is calculated as follows:

$$L_{\mathrm{dn}} = 10 \log \left[\frac{1}{24} \left\{ 15 \times 10^{0.1 L_{\mathrm{d}}} + 9 \times 10^{0.1(L_{\mathrm{n}}+10)} \right\} \right], \mathrm{dB}. \tag{10}$$

It may be noted that L_{n} has been increased by 10 dBA in order to account for our increased sensitivity to noise at night.

5. Loudness

The loudness index L is measured in terms of sones and the loudness level P in phons. They are related to each other as follows:

$$L = 2^{(P-40)/10}, \; P = 40 + 33.2 \log L \; . \tag{11}$$

Roughly, loudness level in phons corresponds to the A-weighted sound pressure level. Then, equations (11) lead to *Table* 1.

Thus, loudness starts from 40 dBA, which is indeed the lowest ambient SPL in the urban environment.

6. Noise Limits in India

The Ministry of Environment and Forests (MOEF) of the Government of India, on the advice of the National

Table 1.

A-weighted SPL (dBA)	40	50	60	70	80	90	100
Loudness Index (sones)	1	2	4	8	16	32	64

Committee for Noise Pollution Control (NCNPC), has been issuing Gazette Notifications prescribing noise limits as well as rules for regulation and control of noise pollution in the urban environment. These are summarized in *Box* 1.

Box 1. The Noise Pollution (Regulation and Control) Rules, 2000 [3]

These rules make use of *Table* A for the ambient air quality standards. They are more or less the same as in Europe and USA.

1. A loud speaker or a public address system shall not be used except after obtaining written permission from the authority.
2. A loud speaker or a public address system or any sound producing instrument or a musical instrument or a sound amplifier shall not be used at night time except in closed premises for communication within, like auditoria, conference rooms, community halls, banquet halls or during a public emergency.
3. The noise level at the boundary of the public place, where a loudspeaker or a public address system or any other noise source is being used, shall not exceed 10 dB(A) above the ambient noise standards for the area (see *Table* A) or 75 dB(A), whichever is lower.
4. The peripheral noise level of a privately owned sound system or a sound producing instrument shall not, at the boundary of the private place, exceed by more than 5 dB(A) the ambient noise standards specified for the area in which it is used.
5. No horn shall be used in silence zones or during night time in residential areas except during a public emergency.
6. Sound emitting fire crackers shall not be burst in a silence zone or during night time.
7. Sound emitting construction equipment shall not be used or operated during night time in residential areas and silence zones.

Table A. Ambient air quality standards in respect of noise.

Category of Area/Zone	Limits in Leq (dBA)	
	Day Time	Night Time
Industrial area	75	70
Commercial area	65	55
Residential area	55	45
Silence zone	50	40

Note:

1. Day time shall mean from 6:00 a.m. to 10:00 p.m.
2. Night time shall mean from 10:00 p.m. to 6:00 a.m.

6.1 *Permissible Noise Exposure for Industrial Workers*

In keeping with the practice in most countries, India has adopted the international limit of 90 dBA during an 8-hour shift for industrial workers.

If we want to evaluate the maximum time that a technician may be asked to work in a noisy environment without risking overexposure, we may make use of an integrating sound level meter to evaluate the 8-hour average of the A-weighted SPL $L_{\mathrm{Aeq,8h}}$ as follows:

$$L_{\mathrm{Aeq,8h}} = 10 \log \left[\frac{1}{8} \int_0^8 10^{L_{pA(t)/10}} \;\mathrm{d}t \right] , \qquad (12)$$

where t is in hours. Then, the maximum allowed exposure time to an equivalent SPL, $L_{\mathrm{Aeq,8h}}$, would be given by

$$T_{\mathrm{allowed}} = 8/D \, , \qquad (13)$$

where D, the daily noise dosage with reference to the base level criterion of 90 dBA, is given by

$$D = 2^{(L_{\mathrm{Aeq,8h}} - 90)/3} . \qquad (14)$$

Here, the constant 3 represents the decibel trading level which corresponds to a change in the exposure by a factor of two for a constant exposure time (10 log 2 = 3). For use in USA, this constant would be replaced with 5.

7. Masking

Masking is the phenomenon of one sound interfering with the perception of another sound. Often environmental noise masks a warning signal. This is why honking has to be considerably louder than the general traffic noise around.

Masking can be put to effective use in giving acoustic privacy to employees located in open cubicles in large call centres or similar large offices under the same roof.

Playing soft instrumental music in the background is enough to mask the conversation in the neighbouring cubicles.

Maintenance of adequate speech communication is often an important aspect of the problem of dealing with occupational noise. The degree of intelligibility of speech depends on the level of background noise in relation to the level of the spoken words. Additionally, the speech level of a talker will depend upon the talker's subjective response to the level of the background noise. Both effects can be quantified [4], as illustrated in *Figure* 2.

The degree of intelligibility of speech depends on the level of background noise in relation to the level of the spoken words. Additionally, the speech level of a talker will depend upon the talker's subjective response to the level of the background noise.

The figure also shows the voice level that the talker would automatically use (expected voice level) as a result of the background noise level. The range of the expected voice level represents the expected range in a talker's subjective response to the background noise. If the talker is wearing an ear protection device such as earplugs or earmuffs, the expected voice level will decrease by 4 dB.

For face-to-face communication with average male voices, the background noise levels shown by the curves in *Figure* 2 represent upper limits for just acceptable speech

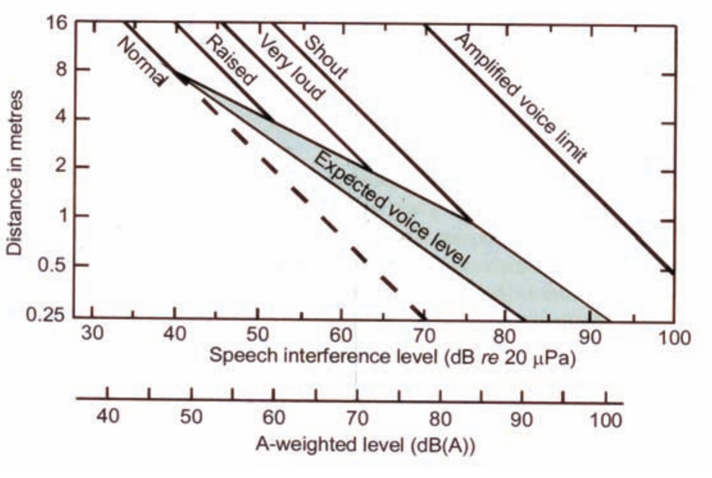

Figure 2. Rating noise with respect to speech intererence [4].

communication, i.e., 95% sentence intelligibility, or alternatively 60% word-out-of-context recognition. For female voices, the speech interference level, or alternatively the A-weighted level shown on the abscissa, should be decreased by 5 dB, i.e., the scales should be shifted to the right by 5 dB [4].

For industrial situations where speech and telephone communication are important, such as in foremen's offices, control rooms, etc., an accepted criterion for background noise level is 70 dBA.

8. Sound Field in a Room

For a machine with a sound power level of $L_w(f)$ at a frequency f, the sound pressure level $L_p(r, f)$ at a distance r from the radiating surface or the acoustical centre of the machine (see *Figure* 3) is in general given by the following formula [1]

$$L_p(r, f) = L_w(f) + 10 \log \left[\frac{Q(f)}{4\pi r^2} + \frac{4}{R(f)} \right] . \qquad (15)$$

Here, $Q(f)/4\pi r^2$ is the contribution from the direct field and $4/R(f)$ is the contribution from the diffuse field; $Q(f)$ is the directivity factor:

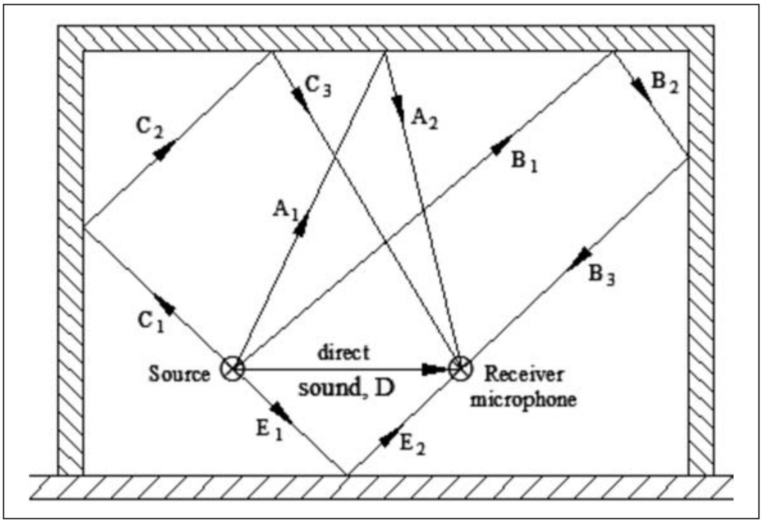

Figure 3. Schematic of the direct sound field D superimposed on the reverberant sound field $A_1 - A_2$, $B_1 - B_2 - B_3$, $C_1 - C_2 - C_3$ and $E_1 - E_2$. The shortest distance between the source and the receiver microphone is denoted by *r*.

$$Q(f) = Q_l \cdot Q_i(f) \ , \tag{16}$$

where $Q_i(f)$ is the inherent directivity of the source as a function of frequency f, and Q_l is the locational directivity factor:

$$Q_l = 2^{n_s} \ , \tag{17}$$

n_{s} is the number of surfaces touching or enclosing the machine location or source. $R(f)$ is the room constant [1,4]:

$$R(f) = \frac{S\bar{\alpha}(f)}{1 - \bar{\alpha}(f)} \, , \tag{18}$$

where S is the total surface area of the room including floor, ceiling, walls, furniture and human occupancy, and $\bar{\alpha}(f)$ is the overall (surface-averaged) acoustic power absorption coefficient:

$$\bar{\alpha}(f) = \frac{\sum_i S_i \alpha_i(f)}{S} \, , \quad S = \sum_i S_i \, . \tag{19}$$

Here, S_i and $\alpha_i(f)$ are the area of the i^{th} surface and its absorption coefficient, respectively.

Incidentally, it may be noted from (15) that in free field (where there is no reflection), $\bar{\alpha} = 1$ and $R(f)$ tends to infinity. Then, (15) yields the following direct field equation

$$L_p(r, f)_{\mathrm{direct\,field}} = L_w(f) + 10 \log \left[\frac{Q(f)}{4\pi r^2} \right] \, , \tag{20}$$

where $Q(f)$ is given by (16) above.

Equation (20) defines the inverse-square law for the direct acoustic pressure field p_{rms} and intensity I:

$$p_{\mathrm{rms}}^2(r), \ I(r) \, \alpha \, \frac{1}{r^2} \tag{21}$$

provided r is large enough to satisfy the far-field criteria. In terms of levels, the inverse square law becomes

$$L_p(r_1) - L_p(r_2) = 20 \log(r_2/r_1), \mathrm{dB} \, . \tag{22}$$

Equation (22) implies that the SPL in the far-field decreases by 6 dB per doubling of the distance, since 20 log 2 = 6. Thus, the decrease is sharper nearer the source but is comparatively milder as one moves farther away from the source [1–3].

In a room or in an industrial shed or a workshop, as we move away from the source, the direct field term of (15) decreases progressively as per the inverse square law, so that close to the walls, it becomes negligible with respect to the reverberant or diffuse field term 4/R. Then, (15) reduces to the diffuse field equation

$$L_p(f)_{\text{diffuse field}} = L_w(f) + 10\log\left[\frac{4}{R(f)}\right]. \qquad (23)$$

Note that the diffuse field equation (23) is independent of the distance parameter r, while the direct field equation (20) is independent of the room surfaces.

It may be noted from (15) that if r tends to zero, the direct sound field would predominate over the diffuse field. This is why a whisper in the ear can be heard even in the noisiest of environments.

The absorption coefficients of different types of walls, ceilings, furnishings, linings and panels at the centre frequencies of different octave bands, measured in specially designed reverberation rooms, are listed in handbooks; see, for example, [1, 4, 5].

9. Acoustics of a Partition Wall

Often a noisy source in a room is separated from the receiver by means of a partition wall. When this wall does not extend up to the ceiling, it is called a barrier.

A partition wall reflects a part of the incident acoustic power or energy back to the source side of the room, absorbs a little of it, and lets the rest pass through to the other side, as shown in *Figure* 4. Then,

$$W_i = W_r + W_\alpha + W_t \qquad (24)$$

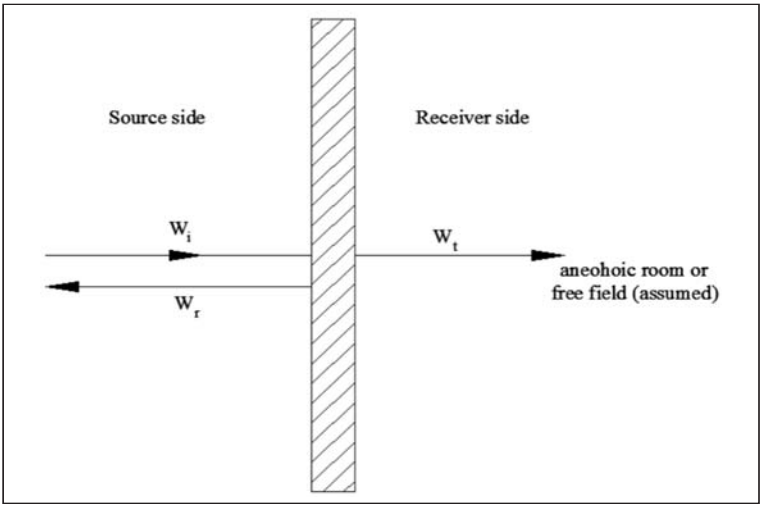

Figure 4. Schematic of a partition wall for normal incidence transmission loss $TL = 10 \log (W_i/W_t)$.

$$\text{Transmission coefficient}, \tau = \frac{W_t}{W_i}, \tag{25}$$

$$\text{Reflection coefficient}, R = \frac{W_r}{W_i}, \tag{26}$$

$$\text{Absorption coefficient}, \alpha = \frac{W_a}{W_i}, \tag{27}$$

$$\text{Transmission loss}, \mathrm{TL} = 10 \log \frac{W_i}{W_t} = -10 \log(\tau) . \tag{28}$$

A double-glaze window with a substantial airgap ($\geq$ 50 mm) between the two layers yields a substantially higher TL than a single-glaze window of double the thickness, because the impedance mismatch happens twice in the case of a double-glaze window, or for that matter, in double-skin doors, double masonry walls, stud partitions, etc. This is a desirable design feature. A well-known example is the construction of aircraft windows.

10. Design of Acoustic Enclosures

Performance of an acoustic enclosure is measured in terms of its insertion loss. It is defined as reduction of SPL at the receiver due to location of the source (machine) in an acoustic enclosure.

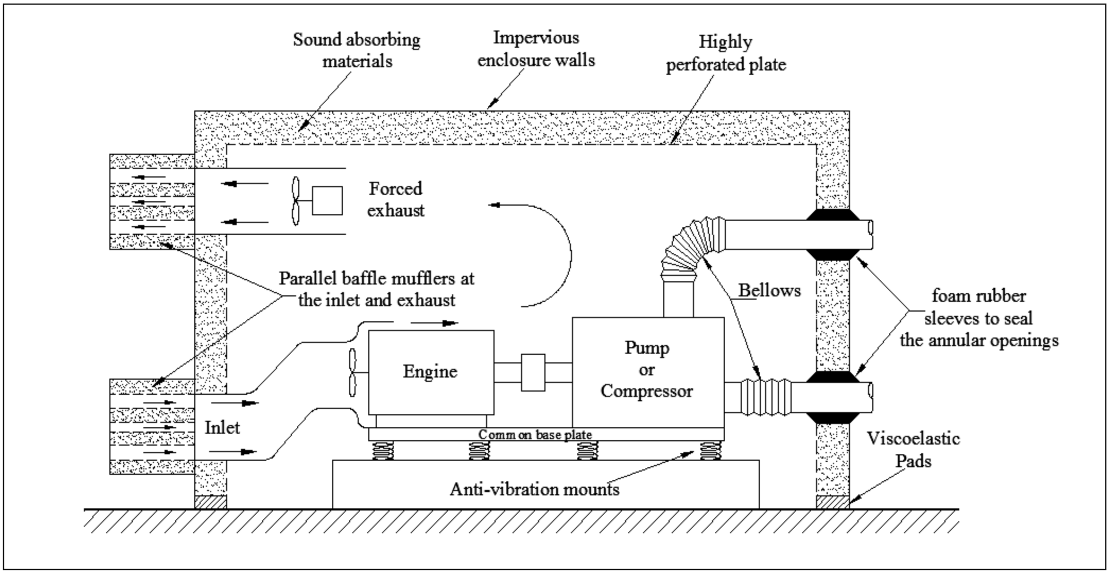

Figure 5. Schematic of an accoustic enclosure used for noise control of an engine-driven pump or a compressor.

The insertion loss (IL) is given by the following, rather approximate, relationship:

$$\mathrm{IL} = 10\log(\bar{\alpha}/\bar{\tau}) = \mathrm{TL} + 10\log\bar{\alpha}\ . \qquad (29)$$

Here $\bar{\tau}$ and TL are the transmission coefficient and transmission loss of the impervious (often metallic) layer of the enclosure walls, and $\bar{\alpha}$ is the random-incidence absorption coefficient of the absorptive lining on the inner or source side of the enclosure walls.

It may be noted from (29) that IL of the enclosure is always less than TL of the walls because $\bar{\alpha}$ is always less than unity. It also indicates the importance of the absorptive lining of all the inner surfaces of the acoustic enclosure. The design and construction of an acoustic enclosure is illustrated in *Figure* 5.

11. Acoustics of Barriers

Acoustic barriers are often used for audio privacy between adjacent cabins in an office layout, or for the partial protection of road-side housing colonies from traffic noise, etc. Trees and bushes are also planted sometimes for landscaping as well as environmental noise control.

Sound diffracts around a finite barrier from all the three sides (top and two side edges). Low-frequency (or large-wavelength) waves bend around more efficiently than high-frequency (or small-wavelength) waves. The effectiveness of barriers increases with Fresnel number, N_i, defined by [1, 4]

$$N_i \equiv 2\delta_i/\lambda\,, \tag{30}$$

where δ_i is the difference between the diffracted path around the ith side or edge and the direct path between the source and the receiver, and λ is wavelength, C_0/f.

For a long acoustic barrier (extending several wavelengths on either side of the source) like a highway barrier, (29) reduces to

$$\mathrm{IL(longbarrier)} = -10\log\left[\frac{\lambda}{3\lambda + 20\delta_1}\right]\,. \tag{31}$$

Referring to *Figure* 6, where subscript 1 refers to the vertical plane,

$$\delta_1 = (c+d) - (a+b)\ . \tag{32}$$

For such a barrier, height is the only design parameter.

The trigonometric implications of equation (32) and *Figure* 6 are as follows:

1. The greater the barrier height h,

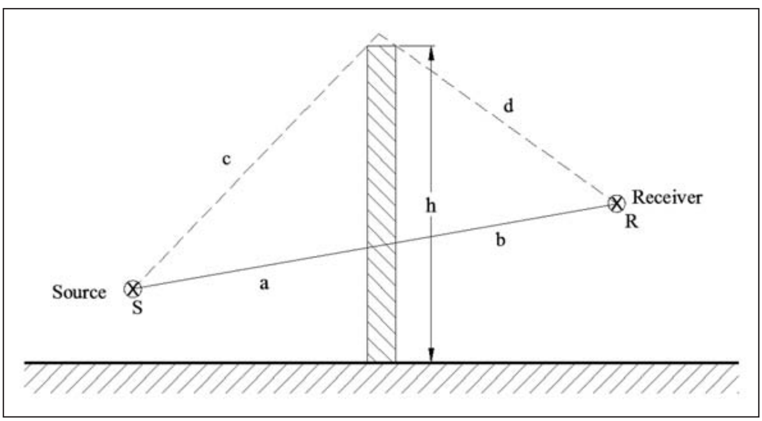

Figure 6. Schematic of a long (semi-infinite) barrier of height *h*.

the greater would be its insertion loss IL. As an important corollary, a highway barrier blocks tyre noise more effectively than the engine noise, since the effective tyre-noise source is situated lower than the effective engine-noise source.

2. Increasing the height beyond (say) 5 m is generally not a very cost-effective measure. One could then explore the feasibility of building the acoustic barrier on the top of an intervening hill or earthen mound.

3. The barrier should be located so that the source or the receiver falls in its shadow, as it were. For example, a railway barrier should be located as close to the railway line or the railway colony as logistically feasible.

4. When a highway or a railway line has acoustic barriers on both sides, the barriers should be lined with an acoustically absorptive layer which can withstand the elements (sun, rain, snow, etc.)

12. Mufflers and Silencers

The terms 'muffler' and 'silencer' are often used interchangeably. These devices are used extensively on the intake as well as exhaust systems of reciprocating internal combustion engines used in automobile or captive generator. compressors, fans, blowers, gas turbines, heating ventilation and air-conditioning (HVAC) systems, high-pressure vents and safety valves. Practically, the intake and/or exhaust (or discharge) systems of all flow machinery are fitted with mufflers. Automotive engines are invariably provided with exhaust mufflers. Therefore, the theory and practice of exhaust mufflers has a history of over a hundred years.

Passive mufflers, that do not make use of a secondary source of noise, are of two types: the reactive (or reflective) type and the absorptive (or dissipative) type. Reactive mufflers work on the principle of impedance mismatch. Making use of sudden changes in the area of cross-section, perforated elements, resonators, etc., the incoming or incident energy is reflected back to the source. In fact, a combination of such elements helps to reduce the acoustic load resistance[2] faced by the source to much less than the characteristic impedance[3] of the exhaust (or intake) pipe so that the source produces much less noise than it would produce into an anechoic load (or termination). An absorptive muffler, on the other hand, does not alter the sound produced by the source, but converts it into heat as sound propagates through its absorptive passages – acoustically absorptive linings [7].

[2] Standing wave accoustic wave pressure divided by the mass velocity.

[3] Progressive wave accoustic pressure divided by the corresponding mass velocity.

Active noise control in a duct consists of making use of a secondary source of noise and an adaptive digital control system in order to produce nearly zero acoustic load (impedance at the junction). Thus, an active noise control system produces an acoustical short-circuit, effectively muffling the primary as well as the secondary sources of sound. Active noise control system is most effective at low frequencies (50 to 500 Hz) whereas passive mufflers work best in the middle and high frequencies. Reactive mufflers, however, can be specially configured for low frequency attenuation.

Design or selection of a muffler is based on the following considerations [7]:

- Adequate insertion loss so that the exhaust (or intake) noise is reduced to the level of the noise from other components of the engine (or compressor or fan, as the case may be), or as required by the environmental noise pollution limits;
- Minimal (or optimal) mean pressure drop so that the source machine does not have to work against undue or excessive back pressure. (This is particularly applicable to fans or blowers which would stall under excessive back pressure);
- Size restrictions, particularly under the vehicle;
- Weight restrictions;

• Durability, particularly in view of sharp thermal gradients in rain or on wet roads;
• Cost effectiveness is often the most important design criterion.

12.1 *Simple Expansion Chamber*

The simplest muffler configuration is a Simple Expansion Chamber (SEC) shown in *Figure* 7. It was first conceptualized in the lumped-element model as a compliance (capacitance) sandwiched between two inertances (inductances), analogous to a low-pass filter in the electrical network theory. For a stationary medium in a chamber of length l, it can be shown that [7]

$$\mathrm{TL} = 10 \log \left[1 + \left\{ \frac{m-(1/m)}{2} \sin(k_0 l) \right\}^2 \right] \quad , \qquad (33)$$

where m is the area expansion ratio given by $m = (R/r)^2$, and the product $k_0 l$ is called the Helmholtz number or the non-dimensional frequency.

Equation (33) is plotted in *Figure* 8 for the area ratio $m = 25$. The following features are noteworthy:

• The TL curve of a simple expansion chamber consists of periodic domes, with sharp troughs occurring at integral multiples of π, and peaks occurring at odd multiples of $\pi/2$.
• Peak value of TL is approximately $20 \log(m/2)$ for $m >> 1$. Thus, larger the expansion ratio, higher the peaks (or domes).

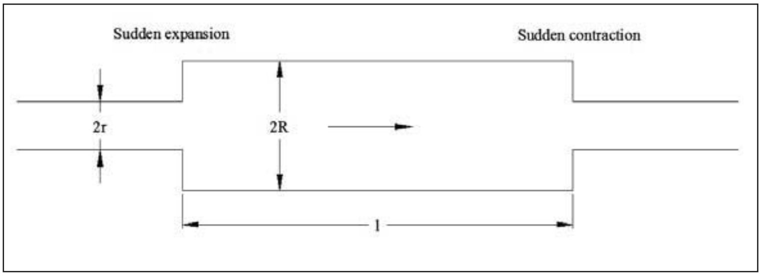

Figure 7. A simple expansion chamber.

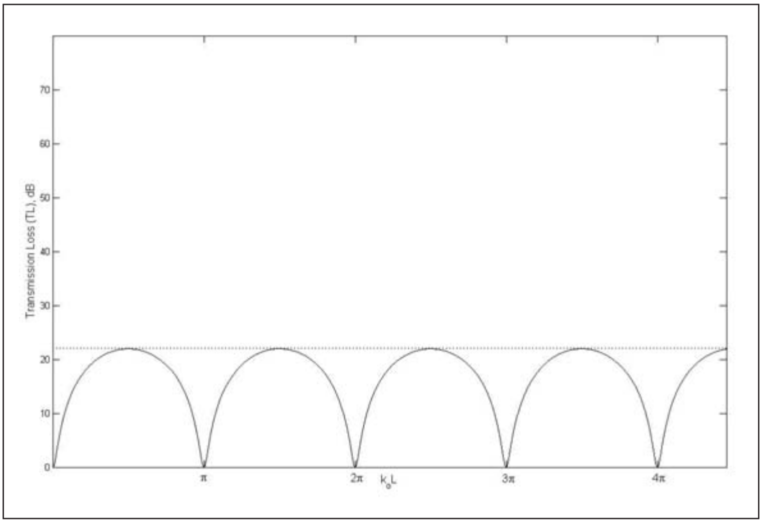

Figure 8. TL of a simple expansion chamber.

• As $m \to 1$, TL $\to 0\,$dB. This shows that TL of a uniform pipe is zero.
• Observations (ii) and (iii) above, when read together, indicate that in reactive mufflers, a sudden change in the area of cross-section S results in a sudden change or jump in the characteristic impedance $Y_0(= c_0/S)$. In fact, it can be shown [7] that in a stationary medium for a sudden expansion and a sudden contraction, (both characterized by equality of pressure and mass velocity across the junction), TL is given by the common expression [7]:

$$\mathrm{TL} = 10 \log \left[\frac{(S_\mathrm{u} + S_\mathrm{d})^2}{4 S_\mathrm{u} S_\mathrm{d}} \right] . \tag{34}$$

where subscripts ‘u’ and ‘d’ denote upstream and downstream, respectively. This is the basis of the concept of ‘impedance mismatch’ which is a fundamental principle ofreactive or reflective mufflers.

Commercial automotive mufflers, however, are much more complex than the simple expansion chamber shown in *Figure* 7 and need special analytical and numerical techniques [7].

12.2 *Absorptive Ducts and Mufflers*

An acoustically lined duct converts or dissipates acoustic energy into heat as a progressive wave moves along the duct, forward or backward. Common absorptive (or dissipative) materials are glass wool, mineral wool, ceramic wool, polyurethane (PU) foam, polyamide foam, etc. All these materials are highly porous with volume porosity of more than 95%. An acoustical material must have open (interconnected) pores unlike thermal foam which has closed pores.

The most important design parameter is $\hat{h}$, defined as

$$\hat{h} \equiv \frac{\text{cross-section of the flow passage}}{\text{wetted (or lined) perimeter}} \quad . \tag{35}$$

Referring to *Figure* 9, it may be noted that for a rectangular duct lined on two sides with an absorptive layer of thickness d, $\hat{h} = b.2h/2b$. Therefore, the flow passage height is denoted by '$2\hat{h}$'. In other words, $\hat{h}$ is half the flow passage height.

Applying (35) to a circular duct lined all around with flow passage diameter D, yields

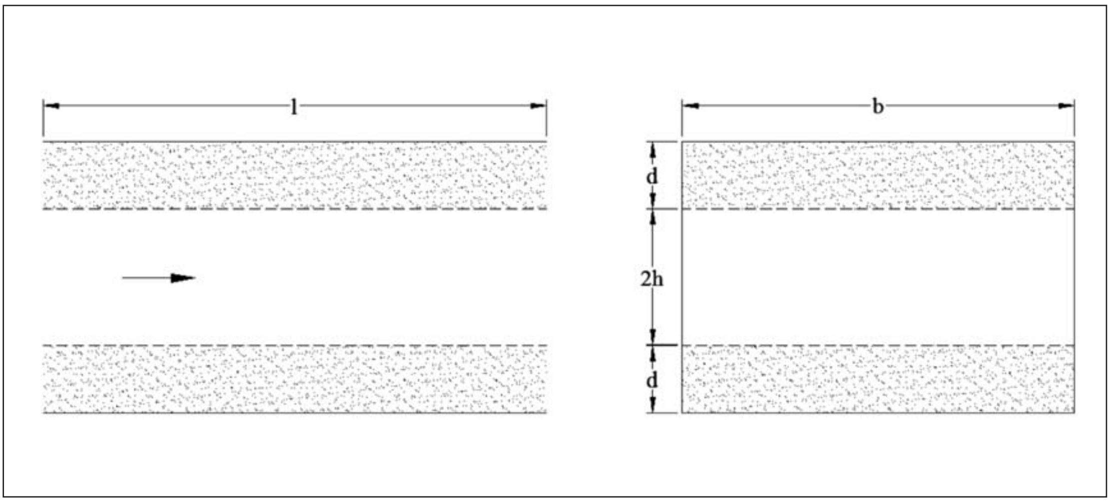

Figure 9. Schematic of a rectangular duct lined on two sides.

$$\hat{h} = \frac{\pi/4D^2}{\pi D} = \frac{D}{4} \, . \tag{36}$$

A similar relationship would hold for a square duct lined on all four sides; $\hat{h}$ would be one-fourth of the clear passage height. The most common practice in the heating, ventilation and air-conditioning (HVAC) systems is to make use of a rectangular duct lined on two sides, as shown in *Figure* 9.

Transmission loss of a lined duct of length l is given by

$$\mathrm{TL} = \mathrm{TL}_h \cdot l/\hat{h} \, , \tag{37}$$

where $\hat{h}$ is defined by (35) and TL_h is the specific TL of a lined duct of length equal to $\hat{h}$. This relationship is very helpful inasmuch as it indicates that TL of a duct is directly proportional to its length l and inversely proportional to $\hat{h}$, defined by (35). For the configuration of *Figure* 9, $\hat{h} = h$, half the passage width.

Equations (35) and (37) collectively indicate that the TL of a square duct lined on all four sides would be nearly double that of the duct lined only on two opposite sides.

Sometimes a designer or consultant needs to do some quick hand calculations of the effectiveness (in terms of transmission loss TL) of a lined duct. The datum available is $\bar{\alpha}$, the absorption coefficient of the material of the lining, defined as the fraction of the normally incident plane-wave energy absorbed by the given thickness of the lining, backed by a rigid wall. The value of the absorption coefficient $\bar{\alpha}$ supplied by the manufacturer is an average value over a certain frequency range. There are a number of empirical formulae for quick hand calculations. One popular example is Piening's empirical formula [7], according to which

$$\mathrm{TL} \approx 1.5 \frac{P}{S} \bar{\alpha} l \text{ (dB)} \, , \tag{38}$$

Manufacturing to closer tolerances is a good engineering practice and makes economic sense in the long run. In other words, the lifetime cost of a quieter, though costlier, machine is relatively lower.

where $\bar{\alpha}$ is the absorption coefficient of the material, P is the lined perimeter, and S is the free-flow area of the cross section.

Thus, for a circular duct of radius r_0 or a square duct with each side $2r_0$ long, lined all over the periphery,

$$\mathrm{TL} \approx 3\bar{\alpha}(l/r_0) \approx 3\frac{l}{d_0} \ . \tag{39}$$

Formula (39), which implies $\bar{\alpha} = 0.5$, is indeed very useful for a quick estimate of the effectiveness of an acoustically lined duct. For example, it indicates that if a material with $\bar{\alpha} = 0.5$ were used to line a circular or a square duct, it would yield a 3-dB attenuation across a length equal to one diameter or side length. Therefore, it is a common design practice to acoustically line a duct length of about 10 diameters upstream and/or downstream of a fan (or air-handling unit, AHU) in an air-conditioning system.

13. General Noise Control Strategies

Noise may be controlled at the source, along the path or at the receiver end. The cost of noise control increases as one moves away from the source. In fact, designing for quietness is the most cost effective way; prevention is better than cure!

Select a quieter machine from the market even if it is relatively costlier. Often, this additional cost is less than the cost and hassle of retrofit noise control measures.

A machine would generally be quieter if its moving parts were fabricated to closer tolerances. Manufacturing mating parts to closer tolerances reduces micro-impacting between the parts. This in turn

- reduces the mechanical (impact) noise;
- reduces vibrations;
- reduces wear and tear;
- increases fatigue life;

- increases the interval between maintenance outings; and
- increases the accuracy of a machine tool.

Overall, manufacturing to closer tolerances is a good engineering practice and makes economic sense in the long run. In other words, the lifetime cost of a quieter, though costlier, machine is relatively lower.

Human desire to obtain large power from small power packs, particularly for aeroengines, motorbikes and automobiles has resulted in a race for high speed engines and turbines. However, noise control at the source can be achieved by selecting large, slow machines rather than small, faster ones, particularly for stationary installations of captive diesel generators, compressors, etc.

13.1 *Maintain for Quietness*

Often, it is observed that a machine that has been in use for a while is no longer as quiet as it was when it was first installed. In order to avoid this additional noise at the source, it is necessary to

- have the rotating parts of the machine balanced on site, not at the supplier's premises;
- monitor the condition of the bearings continuously and have them lubricated regularly;
- replace or adjust the worn or loose parts as soon as detected; and
- follow the periodic maintenance schedules specified by the supplier.

If a machine is thus maintained, its noise level would not unduly increase with use or age.

13.2 *Control of Noise Along the Path*

For any existing noisy machine, the option of noise control at the source as well as designing for quietness is not available. The next option is to control the noise along the path, making use of (a) acoustic barriers,

	Reflective measures	Absorptive measures
Airborne sound	Sound barrier	Sound absorber
Structure-borne sound	Vibration isolator	Vibration damper

Table 2.

hoods, enclosures, etc., for the casing radiated noise, (b) mufflers or silencers for the duct borne intake/exhaust noise, (c) vibration isolators to reduce propagation of unbalanced forces to the foundation or the support structure, and (d) structural discontinuities (impedance mismatches) to block the propagation of structure borne sound.

A damper plays the same role in the absorption of structure-borne sound as the acoustically absorptive material does in the dissipation of airborne sound. This is illustrated in *Table* 2.

Finally, it may be noted that in practice, despite the cost disadvantage, noise is controlled more often along the path than at the source, because of logistical convenience and the ready availability of acoustic enclosures, mufflers or silencers, vibration isolators and dampers.

13.3 *Noise Control at the Receiver End*

This option is the last resort, as it were. The receiver may be protected from excessive noise exposure by means of:

ear plugs or muffs; a cabin for the operator, driver or foreman, and; a control room for the supervisory personnel in noisy test cells.

Ear plugs and muffs are readily available in the market. They provide an insertion loss of 5 to 15 dB for the user, but then the user's functionality may be compromised. He/she may not be able to detect a malfunction in the machine, or may not be forewarned of the damages arising from a malfunction in a machine in the vicinity.

It may be noted that noise control at the receiver end may protect a particular receiver from the noise, but would not help others in the vicinity. This comment would also apply to some of the path control measures. Therefore, the noise control at the source or designing for quietness is the most cost-effective way that would help everybody in the vicinity, not just the operator or driver of the machine.

It is surmised that nearly 50% of all noise in the world is due to air moving devices like fans and blowers.

14. Control of Noise of Fans and Blowers

It is surmised that nearly 50 % of all noise in the world is due to air-moving devices like fans and blowers.

Design of quieter fans (for automotive engines, for example) would normally involve

- Reducing the fan impeller tip speed by reducing the rotational speed (RPM) and/or impeller diameter.
- Increasing the number of blades so as to reduce the required RPM for a given flow rate and static pressure rise.
- Using a thermal (temperature controlled) drive for automobile fans so that the fan would switch off automatically at higher automobile speeds.
- Using airfoil blades in order to increase the aerodynamic efficiency.
- Using non-metallic blades (for increased loss factor).
- Increasing the cooling efficiency so that a fan with a lower tip speed would suffice.
- Minimizing the restriction to the airflow so that the fan could work under lower back pressure.
- Increasing the radiator frontal area and improving the radiator design in order to reduce the cooling load on the fan.
- Using a shroud with minimal radial blade tip clearance in order to reduce the recirculation of flow around the tip.

For a given fan installation, noise can be reduced by

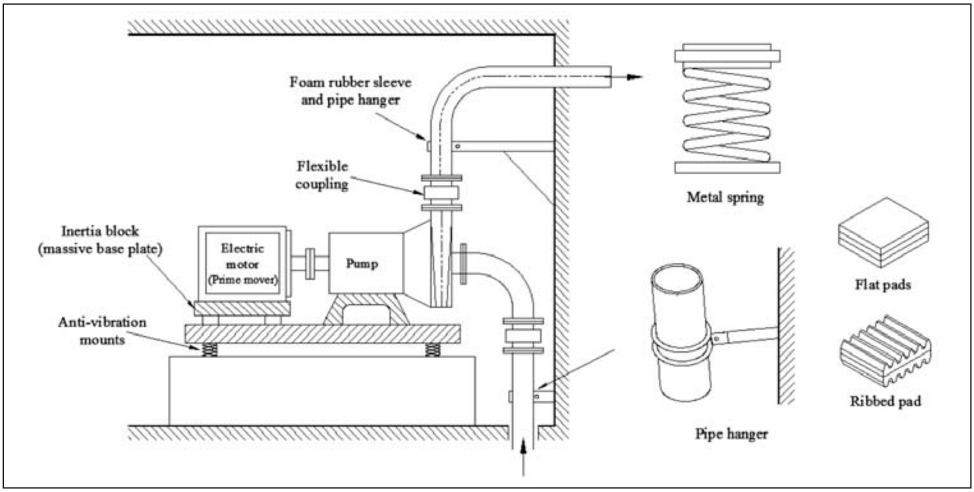

Figure 10. Schematic of different measures for reduction of structure-borne sound [8].

means of

- intake silencers,
- discharge or exhaust silencers,
- acoustic wrapping by means of a hood or an enclosure around the fan in order to contain and absorb the casing noise.

A stand-alone acoustic enclosure would incorporate all three of these measures. The intake and exhaust silencers would then take the shape of parallel baffle mufflers or acoustic louvres.

Noise from a cooling tower can be reduced by providing it with an acoustic enclosure with the two longer opposite sides made entirely of acoustic louvers with sufficient air passages to provide adequate air for the cooling operation.

Quieter technologies are only half the solution. We must develop quieter living habits. This involves discrete use of honking on the roads, loud speakers for celebrations, and television for home entertainment.

Operation of a pump creates pressure pulsations which can be spread by pipes as structure-borne noise and by the liquid medium as fluid-borne noise around the whole pumping system. *Figure* 10 illustrates some measures for the reduction of structural vibration [8].

15. Concluding Remarks

In this tutorial paper, the reader has been informed about environmental noise, noise measurement, speech interference, direct sound, diffuse sound, design of partition walls, acoustic enclosures, barriers, reflective mufflers, absorptive silencers, and general strategies for noise control with particular emphasis on designing for quieter living. The noise of heating, ventilation and air-conditioning systems affects our living quarters. Therefore, the control of noise from fans and blowers is specially touched upon.We must however realize that quieter technologies are only half the solution.We must develop quieter living habits. This involves exercising discretion when honking on roads, using loud speakers for celebrations, and using the television for home entertainment.

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