International Journal of Engineering Research and General Science Volume 3, Issue 1, January-February, 2015
ISSN 2091-2730

FROM SOUND TO NOISE INSULATION: A JOURNEY

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INTRODUCTION:

Life is full of sounds and we want to hear the pleasant and vital ones; while shunning the unpleasant and dangerous variety. All told we are becoming steadily more sound conscious, as the relatively enormous growth of the telephone, radio, phonographic recording and talking motion picture industries sufficiently attests. Sounds touch people in different ways. Sound is extremely important to feel the taste of life. It is what we experience through the senses that make life meaningful. Sound is important for communication, signaling system, for finding depth of objects (SONAR) etc.

Sound has been used to study earth's history, to explore for oil and gas, to study undersea earthquakes and volcanic eruptions, to research wind energy, to measure temperature in ocean, to measure global climatic change. Even during the Cold War, the U.S. Navy allowed a small number of oceanographers to make use of the SOSUS (SOund SUrveillance System) for research in tracking soviet submarines at long ranges.

SOUND

The vibrations in machines and structures result in oscillatory motion that propagates in air and/or water and that is known as sound. Sound can also be produced by the oscillatory motion of the fluid itself, such as in the case of the turbulent mixing of a jet with the atmosphere, in which no vibrating structure is involved. The simplest type of oscillation is vibration and a sound phenomenon is known as simple harmonic motion, which is sinusoidal in time.

Sound can propagate through compressible media such as air, water and solids as longitudinal waves and also as a transverse waves in solids. During propagation, waves can be reflected, refracted, or attenuated by the medium. The speed of sound depends on the medium that the waves pass through, and is a fundamental property of the material. According to Newton-Laplace equation the velocity of sound c, is given by

$c=\sqrt{(\frac{K}{\rho})}$; K = elastic bulk modulus of medium, ρ = density of medium.

Those physical properties and the speed of sound change with ambient conditions. Sound waves travel faster and more effectively in liquids than in air and travel even more effectively in solids. This is because the molecules of solids are more tightly packed together than in liquids and those in liquids are more tightly packed than in gases. Vibrating effects are more easily passed on from one molecule to the next when they are in close proximity. It has been observed that velocity of sound is about 4 times faster in water than in air. Velocity of sound in a steel bar is about 16 times greater than velocity of sound in air as calculated by Rayleigh[1]. Velocity of sound is affected by many parameters like temperature, density, wind, frequency of sound waves, amplitude of sound waves. Rayleigh showed that in case of hydrogen the velocity of sound is greater than for air in the ratio 3.792:1

The vibrations of a bar are of three kinds longitudinal, torsional, and lateral. Of these the last are the most important, but at the same time the most difficult in theory. The unit of sound is decibel (dB) and is measured in room as Sound pressure level (L) by Sound Level Meter (SLM) and calculated manually

$L = 10 \log \frac{p1^2+p2^2+ \ldots +pn^2}{n\, p0^2}$ ------------------------------- EN ISO140-6:1998

where p_1, p_2, …… p_n are the rms sound pressures at n different positions in the room and p_0=20μPa is the reference sound pressure. The difference (D) in space and time average sound pressure levels produced in two rooms by one or more sound sources in one of them is calculated by

$D = L_1 - L_2$ ----------------------------- IS : 9901(Part IV) - 1981

Where L_1 is the average sound pressure level in the source room, L_2 is the average sound pressure level in receiving room.
If sound power W is measured in watts then sound pressure level in dB is

$L = 10 \log \frac{W}{10^{-12}}$

The total sound produced by a number of machines of same type can be determined by adding 10 log n to the sound produced by one machine alone. That is $L_p(n) = L_p + 10 \log n$
Thus sound produced by two individual sound pressure of 80dB will produce a total sound pressure level of (80 + 10 log 2 =) 83dB.

Human ear range of sound intensity: from 10^{-16} W/cm^2 to 10^{-3} W/cm^2
• Zero decibels is the threshold of human hearing and 130dB is the threshold for pain
- 20 – 30dB = whisper
- 40 - 50dB = general office noise
• In open space, when distance is doubled between receiver and sound source intensity level (L) decreases 6dB
- Sound intensity (I) (different from intensity level) will decrease to 1/4
• Doubling the mass = 6dB decrease
• Doubling number of sound sources of equal intensity = increase of 3dB

NOISE

Noise means any unwanted sound. Sounds, particularly loud ones, that disturb people or make it difficult to hear wanted sounds, are noise. Noise exposure can cause two kinds of health effects. These effects are non-auditory effects and auditory effects. Non-auditory effects include stress, related physiological and behavioral effects, and safety concerns. Auditory effects include hearing impairment resulting from excessive noise exposure, Acoustic trauma, Tinnitus, Temporary hearing loss. Noise-induced permanent hearing loss is the main concern related to occupational noise exposure.

SOUND ABSORPTION

The sound-absorbing ability of a material is given in terms of an absorption coefficient, designated by a.. Absorption coefficient is defined as the ratio of the energy absorbed by the surface to the energy incident on the surface. Therefore, a can be anywhere between O and 1. When a = O, all the incident sound energy is reflected; when a = 1, all the energy is absorbed.
The value of the absorption coefficient depends on the frequency. Therefore, when specifying the sound-absorbing qualities of a material, either a table or a curve showing a as a function of frequency is required. Sometimes, for simplicity, the acoustical performance of a material is stated at 500 Hz only, or by a noise reduction coefficient (NRC) that is obtained by averaging, to the nearest multiple of 0.05, the absorption coefficients at 250, 500, 1000, and 2000 Hz.
The absorption coefficient varies somewhat with the angle of incidence of the sound wave. Therefore, for practical use, a statistical average absorption coefficient at each frequency is usually measured and stated by the manufacturer. It is often better to select a sound-absorbing material on the basis of its characteristics for a particular noise rather than by its average sound-absorbing qualities.
Sound absorption is a function of the length of path relative to the wavelength of the sound, and not the absolute length of the path of sound in the material. This means that at low frequencies the thickness of the material becomes important, and absorption increases with thickness. Low-frequency absorption can be improved further by mounting the material at a distance of one-quarter wavelength from a wall, instead of directly on it.

Average absorption coefficient, $\bar{a}$ is calculated as follows:

$$, \bar{a} = \frac{\alpha 1 \square 1 + \alpha 2 S2 + \ldots\ldots\ldots + \alpha n Sn}{S1 + S2 + \ldots\ldots\ldots + Sn}$$

Where $\bar{a}$ = the average absorption coefficient
α_1, α_2, α_n = the absorption coefficient of materials on various surfaces
S_1, S_2, S_n = the areas of various surfaces

SOUND ISOLATION

Noise may be reduced by placing a barrier or wall between a noise source and a listener. The effectiveness of such a barrier is described by its transmission coefficient. Sound transmission coefficient of a partition is defined as the fraction of incident sound transmitted through it.
Sound transmission loss is a measure of sound-isolating ability, and is equal to the number of decibels by which sound energy is reduced in transmission through a partition. By definition, it is 10 times the logarithm to the base 10 of the reciprocal of the sound transmission coefficient.
That is,

$$\text{TL} = 10\ log\frac{1}{\tau}$$

Where TL = transmission loss in Db, τ = Transmission coefficient

$\tau = \frac{W2}{W1}$; W_1 represents the sound power incident on the wall and W_2 the sound power transmitted through the wall.

Transmission loss (TL): It is the difference in decibels between sound power incident on a barrier in a source room and the sound power radiated into a receiving room on the opposite side of the barrier.

Noise reduction (NR): It is the difference in decibels between the intensity levels in two rooms separated by a barrier of a given transmission loss.

- Dependent on the transmission loss of the barrier, area of barrier and absorption of the surfaces in the receiving room

$$\text{NR} = \text{TL}+10\ \log\left(\frac{A}{S}\right)$$

A = total acoustical absorption (sabins), S = area of barrier

Reverberation time (RT): The time which takes the sound level to decrease 60dB after the source has stopped

$$\text{TR} = 0.05\left(\frac{V}{A}\right)$$

TR = reverberation time, V = room volume, A = total acoustical absorption

Recommended reverberation times

Space	**time (sec)**
Auditoriums	1.5 – 1.8
Churches	1.4 – 3.4
Classrooms	.6 - .8
Lecture/conference rooms	.9 – 1.1
Offices	.3 - .6
Opera halls	1.5 – 1.8
Symphony halls	1.6 – 2.1

DOUBLEWALLS

A 4-in. thick brick wall has a transmission loss of about 45 dB. An 8-in. thick brick wall, with twice as much weight, has a transmission loss of about 50 dB. After a certain point has been reached it is found to be impractical to try to obtain higher isolation values simply by doubling the weight, since both the weight and the cost become excessive, and only a 5 dB improvement is gained for each doubling of weight.

An increase can be obtained, however, by using double-wall construction. That is, two 4-in. thick walls separated by an air space are better than one 8-in. wall. However, noise radiated by the first panel can excite vibration of the second one and cause it to radiate noise. If there are any mechanical connections between the two panels, vibration of one directly couples to the other, and much of the benefit of double-wall construction is lost.

There is another factor that can reduce the effectiveness of double-wall construction. Each of the walls represents a mass, and the air space between them acts as a spring. This mass-spring-mass combination has a series of resonances that greatly reduce the transmission loss at the corresponding frequencies. The effect of the resonances can be reduced by adding sound-absorbing material in the space between the panels.

MUFFLERS

Silencers, or mufflers, are usually divided into two categories: absorptive and reactive. The absorptive type, as the name indicates, removes sound energy by the use of sound-absorbing materials.

Fiberglass or mineral wool with density approximately (8-96 kg/m^3)0.5-6.0 lb/ft^3 is frequently used in absorptive silencers. These materials are relatively inexpensive and have good sound-absorbing characteristics. They operate on the principle that sound energy causes the material fibers to move, converting the sound energy into mechanical vibration and heat. The fibers do not become very warm since the sound energy is actually quite low, even at fairly high decibel levels. The simplest kind of absorptive muffler is a lined duct, where the absorbing material is either added to the inside of the duct walls or the duct walls themselves are made of sound-absorbing material. The attenuation depends on the duct length, thickness of the lining, area of the air passage, type of absorbing material, and frequency of the sound passing through. The acoustical performance of absorptive mufflers is improved by adding parallel or annular baffles to increase the amount of absorption. This also increases pressure drop through the muffler, so that spacing and area must be carefully controlled.

Reactive mufflers have a characteristic performance that does not depend to any great extent on
the presence of sound-absorbing material, but utilizes the reflection characteristics and attenuating properties of conical connectors, expansion chambers, side branch resonators, tail pipes, and so on, to accomplish sound reduction. An effective type of reactive muffler, called a Helmholtz resonator, consists of a vessel containing a volume of air that is connected to a noise source, such as a piping system. When a pure-tone sound wave is propagated along the pipe, the air in the vessel expands and contracts. By proper design of the area and length of the neck, and volume of the chamber, sound wave cancellation can be obtained, thereby reducing the tone. This type of resonator produces maximum noise reduction over a very narrow frequency range,

DAMPING

Mass, stiffness and damping are three important parameters that determine the dynamic responses of a structure and its sound transmission characteristics. Mass and stiffness are associated with storage of energy. Damping results in the dissipation of energy by a vibration system. There are many damping mechanisms that convert mechanical energy from a vibratory system into heat and other energy forms. Basically damping mechanisms fall into one of the two categories: external and internal.

External damping mechanisms include acoustic radiation damping, Coulomb friction damping, joint and boundary damping and so on. The dynamic response of a structure couples with the surrounding fluid medium, such as air, water or other liquid, in different ways, for example, by the creation of bending and shear waves.

Internal damping, or material damping, refers to the conversion of vibrational energy into heat within the volume of the material. Any real material subjected to stress/strain cycles dissipates energy. Generally the damping of viscoelastic materials is higher than that of conventional metals. High damping is not the only beneficial property for good noise and vibration control.

DAMPING IN SANDWICH STRUCTURES

A sandwich structure consists of three elements: the face sheets, the core and the adhesive interface layers. The great advantage of sandwich structures is that optimal designs can be obtained for different applications by choosing different materials and geometric configurations of the face sheets and cores. By inserting a lightweight core between the two face sheets, the bending stiffness and strength are substantially increased compared with a single layer homogenous structure, without the addition of much weight. The viscoelastic core has a high inherent damping capacity. When the beam or plate undergoes flexural vibration, the damped core is constrained to shear. This shearing causes the flexural motion to be damped and energy to be dissipated.

When a damping layer is attached to a vibrating structure, it dissipates energy by direct and shear strains. When a solid beam or plate is bending, the direct strain increases linearly with distance from the neutral axis. So unconstrained damping layers which dissipate energy mainly by direct strain are attached to the remote surfaces. On the other hand, the shear stress is the largest at the neutral axis and zero on the free surfaces. Therefore, constrained layers dissipate energy by shear stress. It has been shown that shear damping in viscoelastic materials is higher than in typical structural materials. And the constrained treatment has higher stiffness than unconstrained damping treatment. For these reasons sandwich composite structures are widely used. Besides the three-layer sandwich structures, multi-layer sandwich structures are also used.

SOUND INSULATION SOFTWARES

INSUL is a program for predicting the sound insulation of wall, floors, ceilings, roofs & windows. INSUL uses robust theoretical models that are quick to calculate and only require easily obtainable construction information. The program can make good estimates of the Transmission Loss (TL). Weighted Sound Reduction Index (R_w or STC) and Impact Sound Insulation ($L_{n,w}$ or IIC).

BASTIAN uses a graphical window-user interface and event-controlled programming in its calculation of sound insulation between rooms. It is based primarily on the European Standard series EN 12354 but also utilizes other parameters and definitions from other Standards such as ISO 140 and ISO 717.

REDUCT is another sound insulation program used to calculate the sound reduction index of various building elements. It is primarily based on Kaj Bodlund's[1] report that was done in 1980 and is used within Ingemansson acoustical consultants in Sweden. It is extremely similar to INSUL.

ENC is an acoustical program that is essentially a supplement to the book Engineering Noise Control [2]. The program covers every area outlined within this book ranging from calculations concerning some of the fundamentals of acoustics (such as the addition of decibels) to more complicated calculations involving the power radiated from machines.

WinFlag is a program designed to calculate the sound reduction index, impedance and absorption coefficient for various materials. This program is modeling the acoustic properties of a combination of such layered materials using the transfer matrix method. Basically, each layer in the combination, assumed to be in finite extent, is represented by a matrix giving the relationship between a set of physical variables on the input and the output side of the layer. These matrices may then be combined to give the relationship between the relevant physical variable for the whole combination. Characteristic data as the absorption coefficient, input impedance and the transmission loss (sound reduction index) may then be calculated assuming wave incidence.

ANALYSIS OF VIBRATION

Analysis of the vibratory behavior of a complex structure can be undertaken in two basically different ways: deterministic and statistical analysis.
The two approaches to the problem, deterministic and statistical, are not in competition for the great majority of applications statistical techniques take over from deterministic techniques, both in feasibility and usefulness, as the frequency range of interest rises through the mode series of the structure. With the deterministic approach, computation of individual modal behavior becomes increasingly difficult and unreliable as we go to higher mode numbers (beyond twenty or thirty, perhaps). The statistical approach, on the other hand does not require such detailed calculations and becomes increasingly successful as the resonance frequency spacing gets smaller (compared with the half-power bandwidth of each mode) the more modes one can average over, the more reliable the average becomes as an estimate of what actually happens in the structure There may be an intermediate frequency range where both approaches can be tried, but in such a situation it is possible that neither method will give entirely satisfactory results
There are four reasons for this First, the modes crowd together in frequency so that many more of them need to be considered Secondly, higher frequency (l e shorter length-scale) modes are more sensitive to the inevitable small variations in structural detail even in nominally identical structures, so that they are harder to predict reliably Thirdly, related to the previous point, numerical accuracy decreases as one goes higher up the mode series so that, even if the real structure behaves like the model under study, the numerical predictions from the model may not be reliable

STATISTICAL ENERGY ANALYSIS METHOD

Statistical Energy Analysis was developed in the 1960's, to a great extent to clarify and handle structural acoustic problems in conjunction with space-crafts. The statistical energy analysis (SEA) or power balance method is attractive at high frequencies where a deterministic analysis of all resonant modes of vibration is not practical. In SEA model, a complex structure is virtually divided into coupled subsystems. Energy flows from one subsystem to others. Based on the assumption of power balance of these subsystems, the averaged behavior of the whole structure can be predicted. Because SEA calculates the spatial and frequency averaged response, the SEA model for a complex structure could be quite simple. Modal density, internal loss factor for each subsystem, and coupling loss factors between the subsystems are the basic SEA parameters. The most obvious disadvantage with SEA is that the energy quantities obtained for the different subsystems are statistical estimates of the true energy quantities and accordingly involve some uncertainty.

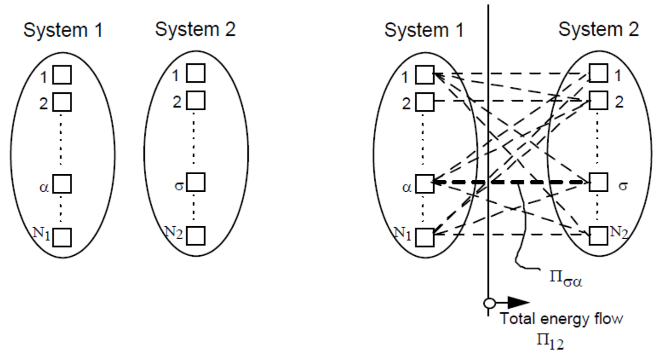

Illustration of interaction and coupling between multi-modal systems

The interaction between two simple oscillators which are assumed to be linear and coupled by linear, non-dissipative elements (no energy lost at the connecting boundaries) constitute the fundamental model from which some important theorems in SEA can be deduced. To summarize it is appropriate to start from the basic equation

$\Pi`2\ 1 = B(E1 - E2)$

Consider the coupled system in Figure above. If the energies in the two subsystems are *E1* and *E2* then the net energy flow between them is found from eqn. above. In this equation *B* is a function of the coupling strength. This coupling is solely governed by the properties of the subsystems and the coupling elements.

From this equation we may state:

• the energy flow is proportional to the actual vibrational energies of the two subsystems

• the coupling function or proportionality is positive, definite and symmetric in the system parameters therefore the system is reciprocal and the energy flows from the subsystem with the higher energy to the one with the lower energy

• if only one subsystem is directly excited the highest possible energy for the indirectly driven subsystem is that of the first.

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