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VIBROACOUSTIC MODELLING OF ENCLOSURE COUPLED TO A FLEXIBLE WALL WITH ATTACHED SPRING-MASS-DAMPER SYSTEM

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ABSTRACT

Generally, solutions to improve the vibration and noise problems are to redesign or modify the system such as increasing the thickness of the wall panels, enhancing the elasticity of the structure, and increase the damping mechanism of the structure. In this study, the application of vibroacoustic modelling of enclosure coupled to a flexible wall was investigated and its effectiveness was further researched by attaching single and multiple spring-mass-damper (SMD) system for the structural vibration control and sound pressure attenuation. The SMD system is designed to minimize the sound pressure amplitude of a flexible wall of a rigid walled enclosure. The sound pressure characteristics of rigid walled enclosure, such as natural frequency and mode shape were determined using two approaches which are simulation (ANSYS®) and theoretical (MATLAB®). In the preliminary study, the theoretical equation derived in MATLAB® such as rigid walled enclosure coupled to flexible wall and rigid walled enclosure with attached SMD were used to validate finite element analysis (FEA) conducted using ANSYS®. The result indicates that the theory and FEA showed in a good agreement. Thus, proved that the FE model was accurate and can be applied in the subsequent analysis. As the mass and damping of the SMD were changed, the sound pressure of enclosure was also affected. The study showed that with attached SMD to a flexible wall, the sound pressure of enclosure was dropped significantly. From these result, single SMD with mass of 20 kg and damping coefficient of 10Ns/m provides the best option. As the study extended to multiple SMDs attachment at different location and configuration, the outcomes showed that the center point attachment on flexible wall at coordinate (1,0.25,0.15) and the combination between location one, two, three and four produce the highest reduction. Finally, it can be concluded that multiple SMDs were able to reduce the sound pressure compared to single SMD. However, for the structures that take weight into consideration, such as in aerospace, automotive and machine system, adding numbers of SMDs will result excess weight to the structure, thus reduce fuel efficiency.

ABSTRAK

Umumnya, penyelesaian untuk memperbaiki masalah getaran dan kebisingan adalah dengan merekabentuk semula atau mengubah suai sistem seperti menambah ketebalan pada dinding panel, mempertingkatkan keanjalan struktur, dan menambah mekanisma redaman pada struktur. Dalam kajian ini, penggunaan pemodelan vibroacoustic ruang terbuka yang ditutup dengan dinding fleksibel adalah untuk menyiasat keberkesanannya dan mengkaji dengan meletakkan pegas-jisim-peredam (SMD) sistem tunggal dan berganda untuk mengawal getaran struktur dan tekanan bunyi didalam ruangan tertutup. Sistem SMD direka untuk mengurangkan amplitud tekanan bunyi dinding fleksibel rongga dinding tegar. Ciri-ciri tekanan bunyi di dalam ruang berdinding, seperti kekerapan semula jadi dan bentuk mod ditentukan dengan dua pendekatan iaitu simulasi (ANSYS®) dan teori (MATLAB®). Dalam kajian awal, persamaan teori yang diperolehi dalam MATLAB® seperti ruang berdinding yang ditutup dengan dinding fleksibel dengan dilampirkan SMD telah digunakan untuk mengesahkan analisis unsur terhingga FEA dijalankan menggunakan ANSYS®. Hasilnya menunjukkan bahawa teori dan FEA menunjukkan dalam perjanjian yang baik. Oleh itu, terbukti bahawa FE model adalah tepat dan boleh digunakan dalam analisis seterusnya. Jisim dan redaman daripada SMD telah diubah telah mengurangan tekanan bunyi di dalam ruang tertutup. Kajian ini menunjukkan bahawa dengan dengan meletakkan SMD pada dinding, tekanan bunyi ruang tertutup telah menurun dengan ketara. Dari hasil ini, SMD tunggal dengan 20 kg jisim dan pekali redaman 10 Ns/m adalah menghasilkan pilihan yang terbaik. Kajian diperluaskan kepada gandaan SMD pada lokasi dan konfigurasi yang berbeza, hasil menunjukkan bahawa SMD yang diletak di titik pusat di dinding fleksibel pada koordinat (1,0.25,0.15) dan gabungan antara lokasi satu, dua, tiga dan empat menghasilkan penurunan tertinggi. Akhirnya, dapat disimpulkan bahawa penyerap getaran berganda dapat mengurangkan tekanan bunyi berbanding dengan penyerap tunggal. Walaubagaimanapun, bagi struktur yang mengambil kira berat sebagai pertimbangan, seperti dalam aeroangkasa, automotif dan sistem mesin, penambahan SMD akan menyebabkan berat badan yang berlebihan kepada struktur, dengan itu mengurangkan kecekapan bahan api.

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LIST OF SYMBOLS

ν Poisson Ratio

ω_{n_e} Natural Frequency

$\Psi_m(y)$ uncoupled vibration mode shape function

$\psi_n(x)$ uncoupled acoustic mode shape function

ρ Density

ρ_s Poisson Ratio

ε_{n_1} normalization factors

ζ_e Damping Ratio

ζ_m Viscous Damping

$a_n(\omega)$ complex amplitude of acoustic pressure

$b_m(\omega)$ complex amplitude of the vibration velocity

c Damping Coefficient

c Speed of sound

c_e Damping Coefficient

DOF Degree of Freedom

$DOFs$ Displacement

E Young's Modulus

F Force

F_m external force applied on the flexible wall

$FLUID30$ Acoustic fluid with structure present

h Thickness

k Spring Stiffness

K_a dynamic stiffness

k_e Spring stiffness

Lx Length

Ly Width

Lz Height

m Mass

m Mode Number

m_e Mass

n Mode Number

P Sound Pressure

$SHELL63$ Flexible wall

APDL Ansys Parametric Design Language

BEM boundary element method

Cmn Modal coupling

DOE design of experiments

fn Natural Frequency

m structural modal integers

n enclosure modal integers

p pressure

u velocity

CHAPTER 1

INTRODUCTION

1.1 Research Background

Nowadays, vibroacoustic or noise and vibration have become important issues to modern society who are looking for a better quality of life. These indirectly make vibration and acoustic characteristics into important criterions to be considered in many engineering design problems. For example, in the automotive and aerospace industries, the level of vibration and noise emissions has become an important asset in manufacturing because its affect on passenger's comfort.

In fact, noise in real applications mostly involves structural vibrations which generate sound wave and these combinations are also known as vibroacoustic. The vibroacoustic study shows that when an elastic structure is in contact with a fluid, the structural vibrations and acoustic pressure in the fluid are influenced by the mutual vibroacoustic coupling interaction. Although these issues have been known over the time, there are still lacks of study in the vibroacoustic fields.

The vibroacoustic are also a major contributor heath disorders concerns. These disorders are caused by vibration and noise. The effect of vibration can cause fatigue, insomnia, stomach problems, headache and "shakiness" shortly after or during exposure. The symptoms are similar to those that many people experience after a long car or boat trip. After daily exposure over a number of years, whole-body vibration can affect the entire body and result in a number of health disorders. Sea, air or land vehicles cause motion sickness when the vibration exposure occurs in the 0.1 to 0.6 Hz frequency range. Studies of bus and truck drivers found that occupational exposure to whole-body vibration could have contributed to a number of circulatory, bowels, respiratory, muscular and back disorders. The combined effects of body posture, postural fatigue, dietary habits and whole-body vibration are the possible causes for these disorders. Studies show that whole-body vibration can increase heart rate, oxygen uptake and respiratory rate, and can produce changes in blood and urine. East European researchers have noted that exposure to whole-body vibration can produce an overall ill feeling which they call "vibration sickness." [1] For some other cases, the effect from noise affects the hearing organs (cochlea) in the inner ear. That is why noise-induced hearing loss is sensory neural type of hearing loss. Certain medications and diseases may also cause damage to the inner ear resulting in hearing loss as well. Generally, it is not possible to distinguish sensory-neural hearing loss caused by exposure to noise from sensory-neural hearing loss due to other causes. Medical judgement, in such cases, is based on the noise exposure history. Workers in noisy environments who are also exposed to vibration (e.g., from a jack hammer) may experience greater hearing loss than those exposed to the same level of noise but not to vibration [1].

Therefore, this research project aims to introduce the problem solving of vibroacoustic system of rigid walled enclosure coupled to a flexible wall with attached spring-mass damper system by using analytical approach and finite element method (FEM). In this research, the properties of the spring-mass-damper system are adapted in order to minimize the sound pressure level in the rigid walled enclosure.

1.2 Problem Statement

Enclosure is the physical separator between the interior and the exterior environments of a building. It serves as the outer shell to help maintain the indoor environment (together with the mechanical conditioning systems). However, the sound reflection and transmitted into rigid walled enclosure cannot be controlled easily when there is a noise source either from inside or outside of rigid walled enclosure.

The traditional treatment for this problem is by using flexible additional linings attached to the wall, in order to have better sound insulation. Nevertheless, this approach is not really effective as it involves highly cost, complicated and not operative in reducing noise at wide frequency spectrum. In the past, many studies have been devoted to develop a feasible method to reduce vibroacoustic (a combination of vibration and acoustic that produces noise). These include: (i) modifying the system, so that the natural frequency does not coincide with the operating speed, (ii) apply damping to prevent large response, (iii) installing isolating devices between adjacent sub-systems, and (iv) adding discrete masses into equipment to reduce the response and absorb vibration.

However, there is still lack of research in the vibroacoustic analysis. One such example is rigid walled enclosure. This research concerns on the above mentioned problems and need of a fundamental study with regard to vibroacoustic modelling of rigid walled enclosure, in particular deriving mathematical formulas to represent this model. It is to discover a better solution on reducing and minimize the sound pressure level of enclosure.

1.3 Objectives of Study

Based on research, there are four objectives need to achieve:

i. To develop a vibroacoustic modelling of rigid walled enclosure coupled to a flexible wall.

ii. To determine the sound pressure reduction of rigid walled enclosure with attached a single SMD systems to a flexible wall.

iii. To investigate the effect of location, mass and damping properties of a single SMD system attached to a flexible wall on sound pressure reduction of rigid walled enclosure.

iv. To investigate the effect of multiple SMD system attached to a flexible wall on sound pressure reduction of rigid walled enclosure.

1.4 Scopes of Study

The research is limited according to the scopes below:

i. Derive a mathematical model for rigid walled enclosure and rigid walled enclosure with attached absorber on a flexible wall

ii. The theoretical model for each case of study will be formulated using Matlab

iii. The finite element analysis model for each case of study will be simulated using Ansys.

iv. The size of rigid walled enclosure that is considered is 1m x 0.5m x 0.3m of width, length and height.

v. The dimension of flexible wall that is considered is 0.5m x 0.3m x 0.01m of width, height and thickness.

vi. Only setting spring stiffness, k at the first mode of natural frequency.

vii. Only maximum five SMDs system will be considered in the study of multiple SMDs.

viii. Literature search on vibration, acoustic, vibroacoustic, spring-mass damper system, sound radiation, rigid walled enclosure and a flexible wall coupled to rigid wall enclosure.

1.5 Expected Outcomes

Several contributions to the body of knowledge presented in this research are; 1) the developments of multiple spring-mass damper system which are used to target wide frequency range are able the reduce sound pressure level of enclosure, 2) provide guidelines for optimum numbers of spring-mass damper systems used and its placement in order to have substantial sound attenuation.

1.6 Significant of Study

In engineering history, excessive vibration and noise emissions has been a common problem in causing the fatigue life of structures shorter. The intensity of vibration sources around us in increasing and tolerances on allowable vibration levels are becoming more and more stringent. From this phenomenon, we know that vibration affects the machines and structure life span. Due to this, it is necessary to come out for a solution by solving from its root.

Vibration and noise also can be harmful and therefore should be avoided. The most effective way to reduce unwanted vibration is to suppress the source of vibration. Above this condition, this research was carried out to understand the vibration characteristic in order to design a dynamic vibration SMD due to the needs of vibration protection itself. As a result, it gave an idea on how to produce an effective SMD. The knowledge gained from this research can be used to minimize the the sound pressure level in the enclosure.

A complete understanding of vibration and noise are needed involves in the analysis and design of a vibration SMD devices so this are the importance why this study should be conducted. This research also has its own novelty in theories and knowledge whereas the finding of this research is instrumental in terms of identifying key of theoretical and mathematical model in development of Dynamic Vibration SMD for multi degree of freedom systems. The other benefit comes from this research in specific or potential application aspect is it could control sound pressure in building and airplane wing flutter control. Therefore, it is judge to be important for doing this research.

CHAPTER 2

LITERATURE REVIEW

In this chapter of literature will explain about vibration, acoustic, vibroacoustic, spring-mass damper system, sound radiation, flexible wall, rigid walled enclosure, Matlab and FEA.

2.1 Vibration

Vibration is a periodic motion of the particles of an elastic body or medium in alternately opposite directions from the position of equilibrium where that equilibrium has been disturbed. The physical phenomena of vibration that take place more or less regularly and repeated themselves in respect to time are described as oscillations. In other words, any motion that repeats itself after an interval of time is called vibration or oscillation. The theory of vibration deals with the study of oscillatory motion of bodies and the associated forces [2].

2.1.1 Classification of Vibration

Vibration can be classified into four categories. First, free and forced where if a system after initial disturbance is left to vibrate on its own, the ensuing vibration is called free vibration, when the system is subjected to an external force (often a repeating type of force) the resulting vibration is known as forced vibration. For second are damped and undamped. Damping is present, then the resulting vibration is damped vibration and when damping is absent it is undamped vibration. The damped vibration can again be classified as under-damped, critically-damped and over-damped system depending on the damping ratio of the system.

Third are linear and nonlinear vibration where if all the basic components of a vibratory system – the spring the mass and the damper behave linearly, the resulting vibration is known as linear vibration. While, if one or more basic components of a vibratory system are not linear then the system is nonlinear.

For deterministic in Figure 2.1(a) , if the value or magnitude of the excitation (force or motion) acting on a vibratory system is known at any given time, the excitation is called deterministic. The resulting vibration is known as deterministic vibration. Random vibration at figure 2.1(b) is the value of the excitation at any given time cannot be predicted. Ex. Wind velocity, road roughness and ground motion during earth quake.

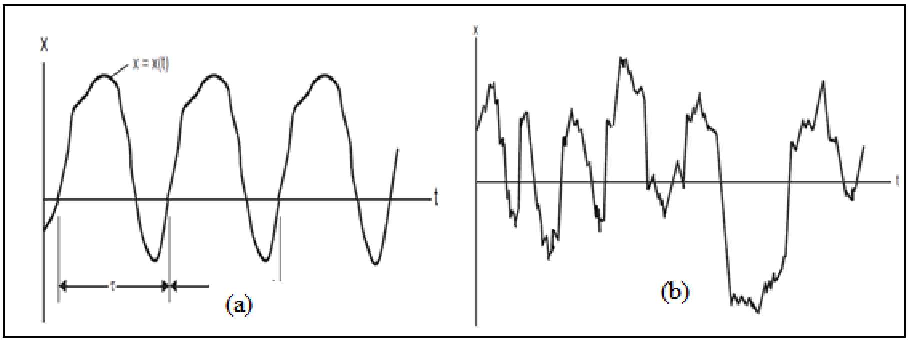

Figure 2.1: (a) deterministic (b) Random.

2.1.2 Vibration Characteristic

This subchapter describes two of vibration characteristics such as natural frequency and mode shape.

2.1.2.1 Natural Frequency

If a system, after and initial disturbance, is left to vibrate on its own, the frequency which it oscillates without external forces is known as its natural frequency as figure 2.2 showed [3]. Each degree of freedom of an object has its own natural frequency, expressed as ω_n. Frequency is equal to the speed of vibration divided by the wavelength, $\omega = \nu \cdot \lambda$. Other equations to calculate the natural frequency depend upon the vibration system. Natural frequency can be either undamped or damped, depending on whether the system has significant damping. The damped natural frequency is equal to the square root of the collective of one minus the damping ratio squared multiplied by the natural frequency, $\omega_d = \sqrt{1-\zeta^2 \cdot \omega_n}$.

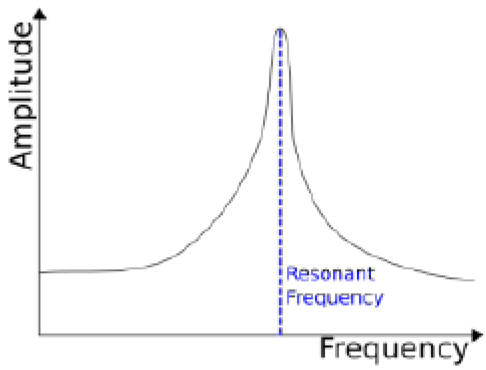

Figure 2.2: Natural Frequency.

Whenever the natural frequency of vibration of a machine or structure coincides with the frequency of the external excitation, there occurs a phenomenon known as resonance, which leads to excessive deflections and failure. The literature is full of accounts of system failures brought about by resonance and excessive vibration of components and systems in 2.3 [3].

Figure 2.3: Tacoma Narrows bridge during wind-induced vibration. The bridge opened on July 1, 1940, and collapsed on November 7, 1940. (Farquharson photo, Historical Photography Collection, University of Washington Libraries.)

Frequency is the number of wave cycles or revolutions per second. The Formula for period (T) in terms of frequency is given by:

$$f = \frac{1}{T} \tag{2.1}$$

From the radian frequency, the natural frequency, f_n , can be found by simply dividing ω_n by 2π. Without first finding the radian frequency, the natural frequency can be found directly using:

$$f_n = \frac{1}{2\pi}\sqrt{\frac{k}{m}} \tag{2.2}$$

2.1.2.2 Mode Shape

Any complex body (e.g., more complicated than a single mass on a simple spring) can vibrate in many different ways. There is no one "simple harmonic oscillator". These different ways of vibrating will each have their own frequency, that frequency determined by moving mass in that mode, and the restoring force which tries to return that specific distortion of the body back to its equilibrium position [4].

It can be somewhat difficult to determine the shape of these modes. For example one cannot simply strike the object or displace it from equilibrium, since not only the one mode liable to be excited in this way. Many modes will tend to excited, and all to vibrate together. The shape of the vibration will thus be very complicated and will change from one instant to the next.

However, one can use resonance to discover both the frequency and shape of the mode. If the mode has a relatively high Q and if the frequencies of the modes are different from each other, then we know that if we jiggle the body very near the resonant frequency of one of the modes, that mode will respond a lot. The other modes, with different resonant frequencies will not respond very much. Thus the resonant motion of the body at the resonant frequency of one of the modes will be dominated by that single mode.

Doing this with strings under tension, we find that the string has a variety of modes of vibration with different frequencies. The lowest frequency is a mode where the whole string just oscillates back and forth as one with the greatest motion in the centres of the string as illustrated in Figure 2.4.

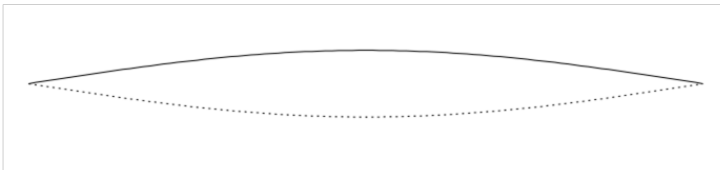

Figure 2.4: The motion in the centre of the string.

The diagram gives the shape of the mode at its point of maximum vibration in one direction and the dotted line is its maximum vibration in the other direction. If we increase the frequency of the jiggling to twice that first modes frequency we get the string again vibration back and forth, but with a very different shape. This time, the two halves of the string vibrate in opposition to each other as shown in Figure 2.5. As on half vibrates up, the other moves down, and are vice versa.

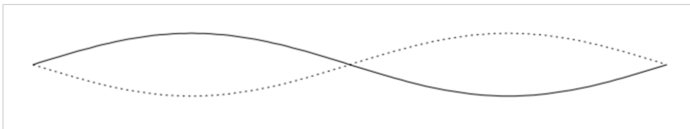

Figure 2.5: The two halves of the string vibrate in opposition each other.

Again the diagram gives the shape of this mode, with the solid line being the maximum displacement of the string at one instant of time, and the dotted being the displacement at a later instant (180 degrees phase shifted in the motion from the first instant). If we go up to triple the frequency of the first mode, we again see the string vibrating a large amount, example at the resonant frequency of the so called third mode. Figure 2.6 shows the string is divided into three equal length sections, each vibrating in opposition to the adjacent piece.

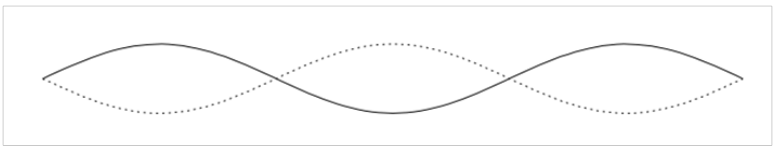

Figure 2.6: The string is divided into three equal length section.

As we keep increasing the jiggling frequency we find at each whole number multiple of the first modes frequency another mode. At each step up, the mode gets an extra “hump” and also an extra place where the string does not move at all. Those places where the string does not move are called the nodes of the mode. Nodes are where the quality (in this case the displacement) of a specific mode does not change as the mode vibrates.

The modes of the string have the special feature that the frequencies of all of modes are simply integer multiples of each other. The n^{-th} mode has a frequency of n times the frequency of the first mode. This is not a general feature of modes. In general the frequencies of the modes have no simple relation to each other. As an example let us look at the modes of a vibrating bar free bar. In Figure 2.7, we plot the shape of the first five modes of a vibrating bar, together with the frequencies of the five modes. Again the solid lines are the shape of the mode on maximum displacement in one direction and the dotted the shape on maximum displacement in the other direction. Note that these are modes where the bar is simply vibrating, and not twisting. If one thinks about the bar being able to twist as well, there are extra modes. For a thin bar, the frequencies of these modes tend to be much higher than these lowest modes discussed here. However the wider the bar, the lower the frequencies of these modes with respect to the vibrational modes.

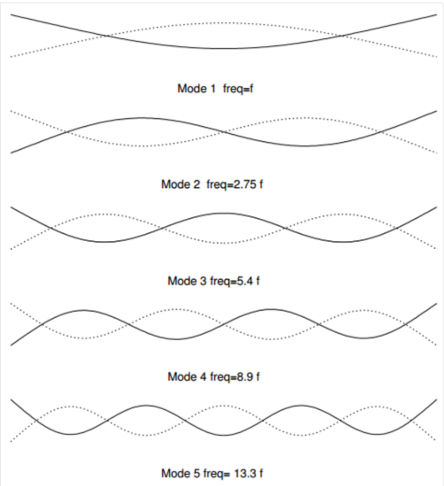

Figure 2.7: The string mode with difference frequencies.

We note that if we lightly hold a finger or other soft item against the vibrating object, it will vibrate against the finger unless the finger happens to be placed at a node where the bar does not vibrate at that node. We can see that the lowest mode and the fifth mode both have nodes at a point approximately 1/4 of the way along the bar. Thus if one holds the bar at that point and strikes the bar, then all of the modes will be rapidly damped except the first and fifth modes, which have a node there. Similarly, if one holds the bar in it's centred, the second, fourth modes both have nodes there while the others do not. Thus only those two will not be damped out.

We note that these modes do not have any nice relation between the frequencies of their modes. We note also that if we strike the bar, we can hear a number of different pitches given off by the bar. For example if we hold it at the 1/4 point, we hear two frequencies, one a very low one and another very high (13.3 times the lowest).

On the other hand if we strike or pluck a string, we hear only one pitch, even if we do not damp out any of the modes. Is there something strange about how the string vibrates? The answer is no. The string vibrates with all of its modes, just as the bar does. It is our mind that is combining all of the frequencies of the various modes into one pitch experience.

2.2 Spring-Mass-Damper Systems

The vibration analysis can be understood by studying the simple spring–mass–damper model. This subchapter describes simple and dynamic of SMD systems.

2.2.1 Simple Spring-Mass-Damper

The simple spring–mass–damper model important in the vibration analysis. Indeed, even a complex structure such as an automobile body can be modeled as a "summation" of simple spring–mass–damper models. The spring–mass–damper model is an example of a simple harmonic oscillator in Figure 2.8. The mathematics used to describe its behavior is identical to other simple harmonic oscillators.

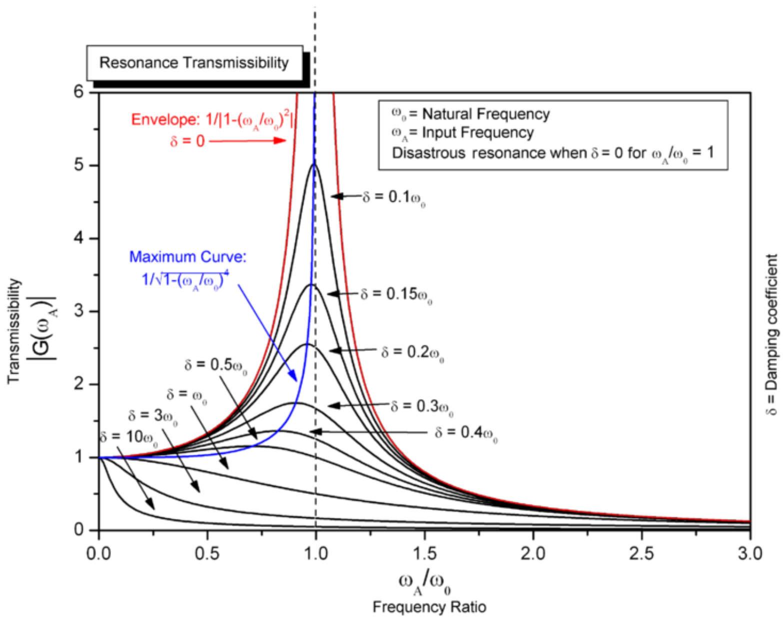

Figure 2.8: A simple harmonic oscillator.

In engineering, an understanding of the vibratory behaviour of mechanical and structural systems is important for the safe design, construction and operation of avarietyy of machines and structures. the failure of most mechanical and structural elements and systems can be associated with vibration.

The spring–mass-damper system in Figure 2.9 is a common control experimental device frequently seen in teaching lab. The design of three different controllers for this system and present robust stability and robust performance analysis of the corresponding closed-loop systems, respectively.

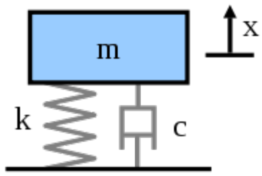

Figure 2.9: Elements of vibratory system where: m-mass (stores kinetic energy); k-spring (stores potential energy, support load) and c-damper (dissipates energy, cannot support load).

The equation of motion applied for this system is given by [5–7] :

$$m\ddot{x} + c\ddot{x} + kx = F \tag{2.3}$$

Taking the Laplace transform of a general second order differential equation with initial conditions, the transfer fuction of the system is:

$$\frac{x(s)}{F(s)} = \frac{1}{ms^2 + cs + k} \tag{2.4}$$

by applying the following definitions $\zeta = \frac{1/m}{2\sqrt{km}}$ and $\omega_n^2 = \frac{k}{m}$, Equation can be simplified and rewriting as [5–7]:

$$\frac{x(s)}{F(s)} = \frac{1/m}{s^2 + 2\zeta\omega_n s + \omega_n^2} \tag{2.5}$$

where ω_n is the undamped natural frequency and ζ is teh damping ratio. The damped natural frequency also known as resonance frequency is given in Equation :

$$\omega_d = \omega_n\sqrt{1-\zeta^2}\,(rad/s) \tag{2.6}$$

When considering the mechanical vibrations of machine elements and structures one generally utilises either the lumped or the distributed parameter approach to study the normal modes of vibration of the system. Engineers are often only concern with the estimation of the first few natural frequencies of a large variety of structures. When modelling the vibrational characteristics of a structure via ANSYS® and MATLAB® approach, the elements that constitute the model include a mass, a spring, a damper and an excitation[8].

The excitation force provides the system with energy which is subsequently stored by the mass and the spring, and dissipated in the damper. The mass, m, is modelled as a rigid body and it gains or loses kinetic energy. The spring (with stiffness Ks) is assumed to have a negligible mass, and it possesses elasticity. A spring force exists when there is a relative displacement between its ends, and the work done in compressing or extending the spring is converted into potential energy – i.e. the strain energy is stored in the spring [8].

2.2.2 Dynamic Spring-Mass-Damper

A dynamic vibration SMD is a device consisting of an auxiliary mass-spring system which tends to neutralize the vibration of a structure to which it is attached. The dynamic vibration SMD has certain advantages over other methods of vibration suppression. It is external to the structure, so no re-installation of equipment necessary. A dynamic vibration absober can be designed and tested before installation. In many scenarios, this offer an economical vibration reduction solution.

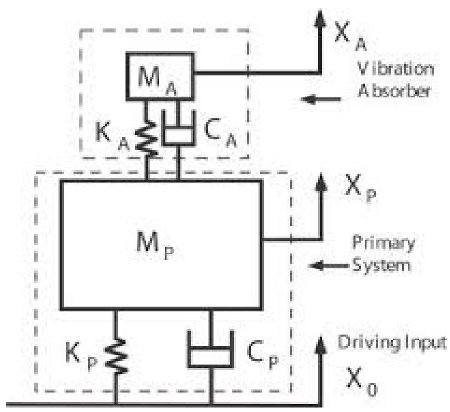

Figure 2.10: Example of Dynamic Vibration SMD.

Figure 2.10 depicts a dual dynamic vibration SMD mounted on identical primary systems. M_A and M_P are the corresponding mass, K_A and K_P are the coresspond-ing stiffness, C_A and C_P are the corresponding damping. It is assumed that identical forces with two harmonic components are applied to the primary mass of the system[4].

A vibration SMD is useful for situations in which the disturbance has a contant frequency. As opposed to a vibration isolar, which contains stiffness and damping elements, a vibration SMD is a device consisting of another mass and a stiffness element that are attached to the main mass to be protected from vibration. The new system consisting of the main mass and the SMD mass has two degrees of freedom, and thus the new system has two natural frequencies.

If the SMD is tuned so that its natural frequency coincides with the frequency of the external forcing, the steady state vibration amplitude of the main device becomes zero. From a control perspective, the SMD acts like a controller that has an internal model of the disturbance, which therefore cancels the effect of the disturbance. If the frequency of the disturbing input and the natural frequency of the original system, that can select the values for the SMD's mass and stiffness so that the motion of the original mass is very small, which means that its kinetic and potential energies will be small. In order to achieve this small motion, the energy delivered to the system by the disturbing input must be "absorbed" by the SMD's mass and stiffness. Thus the resulting motion SMD will be large.

2.3 Acoustic

Acoustic is the science of sound, including its production, transmission, and effects. In present usage, the term sound implies not only the phenomena in air responsible for the sensation of hearing but also whatever else governed by analogous physical principles. Thus, disturbances with frequencies too low (infrasound) or too high (ultrasound) to be heard by a normal person are also regarded as sound. Acoustic is distinguished from optics in that sound is a mechanical, rather than electromagnetic, wave motion [9].

A variety of applications, in basic research and in technology, exploit the fact that the transmission of sound is affected by, and consequently gives information concerning, the medium through which it passes and intervening bodies and inhomogeneities. The physical effects of sound on substances and bodies with which it interacts present other areas of concern and of technical application [10].

The term 'frequency' in acoustics is bound to pure tones, meaning a sinusoidal wave form in the time-domain. Such a mathematically well-dened incident can only rarely be observed in natural sound incidents.

2.3.1 Fundamental Acoustic Concepts

One of the characteristics of fluids, that is, gases and liquids, is the lack of constraints to deformation. Fluids are unable to transmit shearing forces, and therefore they react against a change of shape only because of inertia. On the other hand a fluid reacts against a change in its volume with a change of the pressure. Sound waves are compressional oscillatory disturbances that propagate in a fluid. The waves involve molecules of the fluid moving back and forth in the direction of propagation (with no net flow), accompanied by changes in the pressure, density and temperature; see figure 2.11.

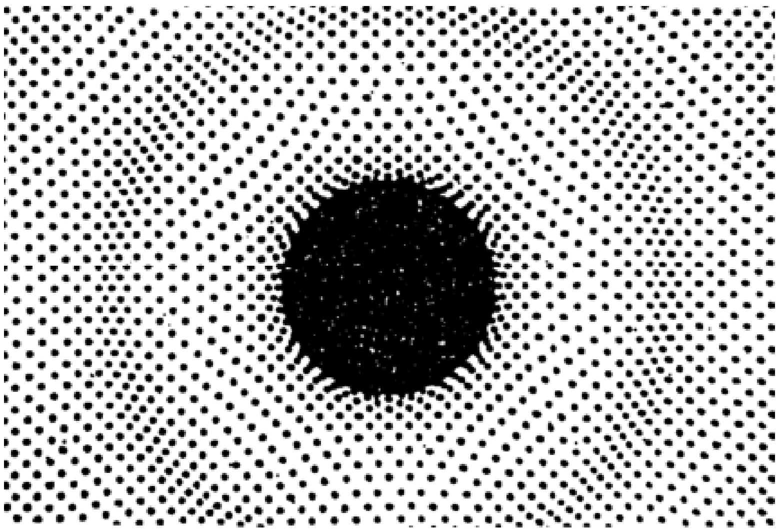

Figure 2.11: Fluid particles and compression and rarefaction in the propagating spherical sound field generated by a pulsating sphere.[11].

The sound pressure, that is, the difference between the instantaneous value of the total pressure and the static pressure, is the quantity we hear. It is also much easier to measure the sound pressure than, say, the density or temperature fluctuations. Note that sound waves are longitudinal waves, unlike bending waves on a beam or waves on a stretched string, which are transversal waves in which the particles move back and forth in a direction perpendicular to the direction of propagation.

2.3.2 Acoustic measurement

The most important measure of sound is the rms sound pressure,defined as:

$$p_{rms} = \sqrt{p^2(t)} = \left(\lim \frac{1}{T} \int p^2(t)dt\right)^{1/2} \tag{2.7}$$

However, as we shall see, a frequency weighting filter is usually applied to the signal before the rms value is determined. Quite often such a single value does not give sufficient informa- tion about the nature of the sound, and therefore the rms sound pressure is determined in frequency bands. The resulting sound pressures are practically always compressed logarithmically and presented in decibels [12].

2.4 Vibroacoustic

Vibroacoustic is a coupling between sound waves and structural vibrations which are of practical importance in many fields concerning acoustic and structural engineering. The problem is to the conservative case, which means in particular that both the solid and the fluid are assumed to be finite extent, and limited by perfectly reflecting boundaries. Presentation is first focused on one-dimensional systems, which are amenable to analytical and semi-analytical solution based on the modal synthesis method [13] and [10].

Several examples are worked out to highlight interesting physical features of vibroacoustic coupling and illustrate a few numerical aspects of practical relevance concerning the computed vibroacoustic modes. In this respect, the coupled problems will be formulated in terms of non-symmetrical as well as symmetrical equations, depending on field variables used to describe the fluid [13] .

2.5 Flexible Wall

A flexible wall in Figure (2.12) a flat structural element for which the thickness is small compared with the surface dimensions. The thickness is usually constant but may be variable and is measured normal to the middle surface of the plate. Wall structures often contain or are in contact with liquids or gases. Vibrating shell and wall structures are not only encountered by the civil, aeronautical, and astronautical engineer, but also by the mechanical, nuclear, chemical, and industrial engineer.

Parts or devices such as engine liners, compressor shells, tanks, heat exchangers, life support ducts, boilers, automotive tires, vehicle bodies, valve read plates, and saw disks, are all composed of structural elements that cannot be approximated as vibrating beams. Shells especially exhibit certain effects that are not present in beams or even plates and cannot be interpreted by engineers who are only familiar with beam-type vibration theory.

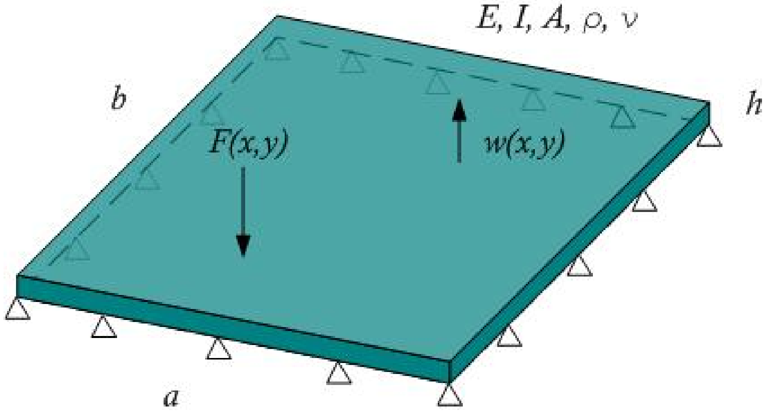

Figure 2.12: Schematic of flexible wall coupled to rigid walled enclosure [14]

The equation of motion of a simply-supported plate can be written as [15, 16]:

$$EI\left(\frac{\partial^4\omega}{\partial x^4}+2\frac{\partial^2\omega}{\partial x^2\partial y^2}+\frac{\partial^4\omega}{\partial y^4}\right)+\rho h\frac{\partial^2\omega}{\partial t^2}=-F\left(x,y,t\right) \tag{2.8}$$

where E is the Young's modulus, I is the area moment of inertia, ρ is the density of plate and h is thickness of plate. The area moment of inertia for plate is defined, where ν is the Poisson's ratio.

$$I=\frac{h^3}{12\left(1-\nu^2\right)} \tag{2.9}$$

The solution of transverse modal displacement for a plate is given by the summation of all of the individual modal amplitude responses multiplied by their mode shapes at that point [15, 16].

$$w(x,y,t) = \sum_{m=1}^{\infty}\sum_{n=1}^{\infty} W_{mn} \cdot \Psi_{mn}(x,y)\, e^{j\omega_n t} \tag{2.10}$$

where W_{mn} is the modal amplitude, $\Psi_{mn}(x,y)$ is the mode shape of plate, and m and n are modal integers.
The general mode shape of a simply-supported plate can be calculated with [15, 16]:

$$\Psi_{mn}(x,y) = 2\sin\left(m\pi x/a\right)\sin\left(n\pi y/b\right) \tag{2.11}$$

where a and b are the length and width of a plate, respectively.
The natural frequencies of a simply-supported plate can be calculated from [15, 16]:

$$\omega_n = \sqrt{\frac{EI}{\rho h}}\left[\left(\frac{m\pi}{a}\right)^2 + \left(\frac{n\pi}{b}\right)^2\right] \qquad (rad/s) \tag{2.12}$$

By neglecting the exponential time varying term, an expression of the total response of simply-supported plate incorporating the viscous damping ζ and structural damping η is given [15, 16].

$$w(x,y,t) = \frac{F}{\rho hab}\sum_{m=1}^{\infty}\sum_{n=1}^{\infty}\frac{\Psi_{mn}(x,y)\,\Psi_{mn}(x_i,y_i)}{\omega_{mn}^2 - \omega^2 + j2\zeta\omega\omega_{mn}} \tag{2.13}$$

$$w(x,y,t) = \frac{F}{\rho hab}\sum_{m=1}^{\infty}\sum_{n=1}^{\infty}\frac{\Psi_{mn}(x,y)\,\Psi_{mn}(x_i,y_i)}{\omega_{mn}^2(1+j\eta) - \omega^2} \tag{2.14}$$

The solution method for the harmonic analysis that is applied in ANSYS® is the same as in MATLAB®, which is mode superposition method.

2.5.1 Flexible wall analysis attached with Spring-Mass-Damper System

Model of a simply-flexible wall attached with a single DOF spring-mass-damper (SMD) system subjected by a harmonic load, $F(x_i,y_i)$ is shown in Figure 2.13. The flexible wall has a uniform thickness h, lenght a, width b, the Young's modulus E, the area moment of inertia I, the density $\rho$and Poisson's ratio ν, while the SMD system has a mass m_e, spring stiffness $k_e$and damper c_e. The spring-mass-damper system is applied at the center of the wall.

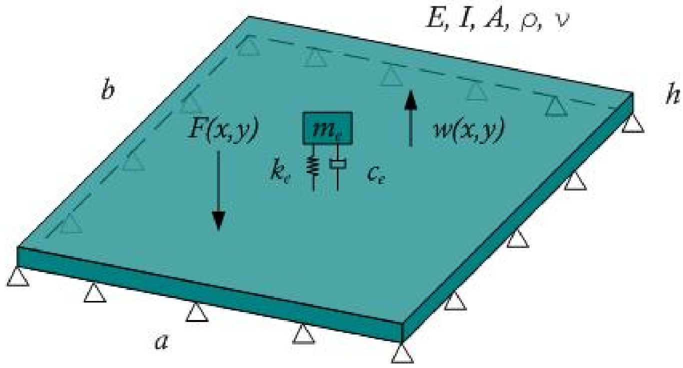

Figure 2.13: A model of simply supported plate attached with SMD .

The theory used to solve this analysis is quite similar. By using the previous equation, the response of a plate at point $(x_1, y_1)$with a spring-mass-damper system attached at point $(x_2, y_2)$is given by [6, 17, 18] :

$$\omega_1 (x,y) = \left[\alpha_{11} - \frac{\alpha_{21}^2}{\alpha_{22} + \beta_{22}} \right] F(x,y) \tag{2.15}$$

where

$$\alpha_{11} = \frac{4}{\rho abh} \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} \frac{\sin^2 (k_m x_1) \sin^2 (k_n y_1)}{\omega_n^2 - \omega^2 + j2\zeta \omega \omega_n} \tag{2.16}$$

$$\alpha_{21} = \frac{4}{\rho abh} \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} \frac{\sin (k_m x_1) \sin (k_n y_1) \sin (k_m x_2) \sin (k_n y_2)}{\omega_n^2 - \omega^2 + j2\zeta \omega \omega_n} \tag{2.17}$$

$$\alpha_{22} = \frac{4}{\rho abh} \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} \frac{\sin^2 (k_m x_2) \sin^2 (k_n y_1)}{\omega_n^2 - \omega^2 + j2\zeta \omega \omega_n} \tag{2.18}$$

where ω_n is the n-th natural frequency of plate, ζ is the viscous damping of plate, m and n are the mode number, $k_m$and k_n is given by Equation :

$$k_m = \frac{m\pi}{a} \tag{2.19}$$

and

$$k_n = \frac{n\pi}{b} \tag{2.20}$$

2.5.2 Sound Radiation from Flexible Wall

Among the most important sources of noise pollution are transport means, that is, cars, trucks, trains, planes, boats, etc. All these vehicles are essentially composed of thin vibrating structures. The simplest thin structure is the thin plate, then comes the circular cylindrical thin shell and the spherical thin shell. These basic structures provide a set of examples which make it possible to understand the basis of the physical phenomena of vibrations and sound radiation.

To estimate the radiated sound power by vibrating structure computer software can be used. In order to be sure that a correct model is used, the results have to be compared with theory. Theories exist about simple structures like a rectangular wall. Therefore in this report a baffled rectangular wall (without damping) is investigated, see Figure 6. The wall is simply supported at the edges; there are no translational degrees of freedom at the edges, see Figure 2.14.

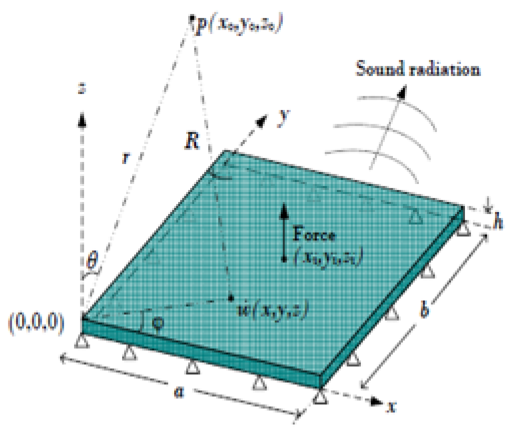

Figure 2.14: Sound radiation from a flexible

The simplest approach to calculate the sound field radiated by a vibrating surface that is surrounded by a rigid infinite panel is the evaluation of the Rayleigh Integral, which given as follows [16, 19].

$$p\left(x_\circ, y_\circ, z_\circ\right) = \int_s \frac{j\omega\rho_a \dot{w}\left(x,y,z\right)\exp\left(j\omega t\right)\exp\left(-jkR\right)}{2\pi r}\,\mathrm{d}S \tag{2.21}$$

where $\dot{w}\left(x,y,z\right)$ is component of the complex velocity normal to the surface, ρ_a is the density of the acoustic medium, ω is frequency in rad/s, r is the distance from the observation point $\left(x_\circ, y_\circ, z_\circ\right)$ to the coordinate origin and $R^2 = \left(x - x_\circ\right)^2 + \left(y - y_\circ\right)^2 + \left(z - z_\circ\right)^2$

The classical assumption made in order to evaluate the far-field pressure is that the value of R is approximated by [16, 19, 20]:

$$R \backsimeq r - x\sin\theta\cos\phi - y\sin\theta\sin\phi \tag{2.22}$$

where x and y define the coordinate position on the plate, and $\left(r, \theta, \phi\right)$ are the coordinate of the field point. This assumption is valid for provided $R \geqslant a, b$.

A particular form of out-of-plane vibration for a simply-supported rectangular plate with the above assumptions, leads to an analytically tractable form of equation which is given by [16, 19, 20]:

$$\dot{w}\left(x,y,z\right) = \sum_{m=1}^{\infty}\sum_{n=1}^{\infty} \dot{W}_{mn}\sin\left(\frac{m\pi x}{a}\right)\sin\left(\frac{n\pi y}{b}\right) \tag{2.23}$$

By substituting Equation given, the sound pressure radiated by a simply-supported plate in an infinite baffle then can be written as:

$$\begin{aligned} p\left(r,\theta,\phi\right) = \sum_{m=1}^{\infty}\sum_{n=1}^{\infty} & \frac{j\omega\rho_a \dot{W}_{mn}\exp\left(j\omega t\right)\exp\left(-jkr\right)}{2\pi r} \\ & \times \int_0^b\int_0^a \sin\left(\frac{m\pi x}{a}\right)\sin\left(\frac{n\pi x}{b}\right)\exp\left[j\left(\frac{\alpha x}{a} + \frac{\beta y}{b}\right)\right]\mathrm{d}x\,\mathrm{d}y \end{aligned} \tag{2.24}$$

where $\alpha = ka\sin\theta\cos\phi$ and $\beta = kb\sin\theta\sin\phi$

This integral equation has been evaluated by [21] who gives the solution:

$$p(r,\theta,\phi) = \frac{j\omega\rho_a \dot{W}_{mn} \exp(-jkr)}{2\pi r} \cdot \frac{ab}{mn\pi^2} \times \left[\frac{(-1)^m \exp(-j\alpha) - 1}{(\alpha/m\pi)^2 - 1}\right] \times \left[\frac{(-1)^n \exp(-j\beta) - 1}{(\beta/m\pi)^2 - 1}\right] \quad (2.25)$$

2.6 Rigid Wall Enclosure

Enclosure or cavity at Figure 2.15 is the physical separator between the interior and the exterior environments of a building. It serves as the outer shell to help maintain the indoor environment (together with the mechanical conditioning systems). Enclosures can be improved for the purposes of better sound insulation using exible additional linings, each elevated on separate bearings. The improvement becomes noticeable above the spring-mass resonance frequency of the resonator given by the enclosure between the walls and the lining.

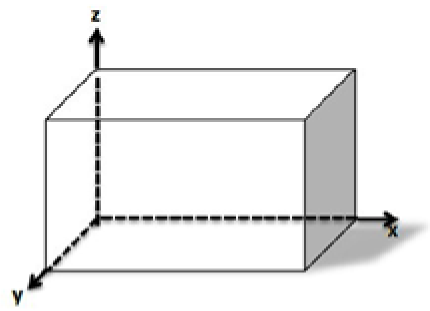

Figure 2.15: The definition of rigid-walled enclosure dimensions

Figure (2.16) illustrates the rectangular enclosure and the coordinate system adopted. The dimensions of the enclosure are $Lx1$(length) $Lx2$(width) and $Lx3$(height) which were chosen such that $Lx1$: $Lx2$:$Lx3 = 1$:$e/\pi : 1/\pi$so as to avoid the degenerate acoustic modes. The enclosure consists of five acoustically rigid walls and a simple supported flexible panel at$x2 = 0$.The primary enclosed sound fields are due to the interaction between the interior acoustic space and the structural vibration on the flexible panel excited by an external plane wave Sp of frequency, ω.

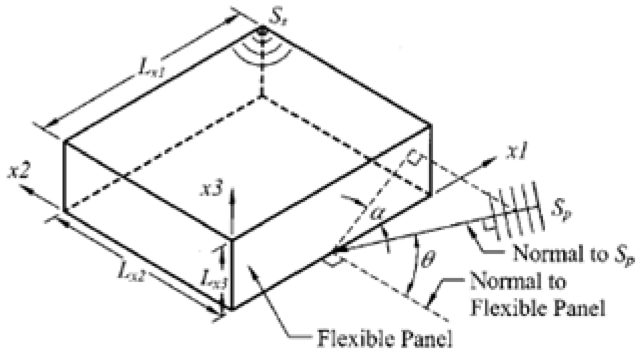

Figure 2.16: Rectangular enclosed space and coordinate system [22]

The propagation direction of Sp is defined by the incidence angle $\theta$and azimuth α as shown in Fig. 10. The incidence angle $\theta$is defined as the angle between the lines normal to the external plane wave and the flexible panel, while the azimuth$\alpha$is the angle between the projected plane of the line normal to the external plane wave on the panel and the $x1$ axis. In the present investigation, the secondary acoustic control source, $Ss(Lx1, Lx2, Lx3)$) in order to avoid the nodal lines of any acoustic mode in the rectangular enclosure. Two new dimensionless parameters $n_c$and φ, are introduced. Lx(max) is the maximum perpendicular separation between two parallel walls inside the rectangular enclosure, and Ly3Ly2 are the dimensions of the flexible panel. In the present numerical model, Lx (max) = Ly1 = Lx1 and Ly2 = Lx3. Table 1 shows some possible values of $n_c$and φ[22].

TABLE I. Possible values of η_c and φ.[a]

Flexible structure	L_{x1} (m)	η_c	φ
6 mm glass (Ref. 21)	1	0.01	1.04
6 mm glass (Ref. 21)	5	0.26	5.21
12 mm glass (Ref. 21)	5	0.06	2.60
6 mm alumina (Al_2O_3) (Ref. 21)	5	0.10	3.05
Kim and Brennan (Refs. 10 and 22)	1.5	0.04	0.81

Figure 2.17: Possible Value of Dimension

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