

VIBRATION CONTROL OF THIN PLATE STRUCTURE WITH
ATTACHED DYNAMIC VIBRATION ABSORBERS

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Dedicated specially to my wife & parents
Anis Nabihah, Mohamed Salleh & Siti Haida & Sufiah
All those who have been a great help in the completion of this thesis

My love to all of you will remain forever...

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ABSTRACT

Vibration of a thin plate structure is one of a major concern in most engineering branches since it is the most widely used structural component in industrial applications, such as mechanical, electronic and civil structures. Thin plate structure exposed to vibration can lead to excessive deflections and eventually failure of the structure. The aim of this research is to investigate the effectiveness of using multiple dynamic vibration absorber for vibration control of a fixed-fixed thin plate. The vibration characteristics of thin plate are determined in the initial stage by using free vibration analysis. Two approaches are employed in the study: finite element analysis and experimental modal analysis. The preliminary results of finite element analysis demonstrate that the first five natural frequency of plate are 43Hz, 162Hz, 281Hz, 387Hz and 519 Hz. These results are found corroborated well with the experiment, although there is a slight discrepancy in the first mode due to noise generated from inconsistent tapping during the impact hammer test. As it is planned to investigate the effectiveness of vibration absorber, a force vibration analysis of plate is carried out by using finite element method running in parallel with the experimental. Again, the simulation results of force vibration analysis are well validated with the experiment. Result shows that the average percentage reduction of plate attached with single vibration absorber is 84%. While when plate attached with dual dynamic vibration absorber, it provide a larger suppression of vibration amplitude which is about 94% at all frequency modes. From these results, it can be concluded that multiple vibration absorber can reduce the overall global structural vibration compared to a single absorber attachment. Nevertheless, the research reveals that the vibration absorber which designed in this study is significant in reducing the vibration. Both result free and force vibration responses of simulation were found corroborated with the experimental.

ABSTRAK

Getaran struktur pada plat nipis adalah salah satu fokus utama di dalam kejuruteraan getaran, kerana ia adalah komponen struktur yang paling banyak digunakan dalam aplikasi industri, seperti struktur mekanikal, elektronik dan sivil. Struktur plat nipis yang selalu terdedah kepada getaran yang boleh menyebabkan kepada pesongan berlebihan dan akhirnya berlaku kegagalan pada struktur. Tujuan kajian ini adalah untuk mengkaji keberkesanan penggunaan satu atau lebih penyerap getaran dinamik untuk mengawal getaran pada plat nipis yang ditetapkan pada setiap bucu. Pada peringkat awal, ciri-ciri getaran plat nipis ditentukan dengan menggunakan analisis getaran bebas. Dua pendekatan yang digunakan dalam kajian ini: analisis unsur terhingga dan analisis menggunakan eksperimen. Keputusan awal analisis unsur terhingga menunjukkan bahawa lima frekuensi semula jadi pertama pada plat adalah 43Hz, 162Hz, 281Hz, 387Hz dan 519 Hz. Analisis dari keputusan ini menunjukkan persamaan yang baik antara kaedah unsur terhingga dan eksperimen analysis mod, bagaimanapun terdapat perbezaan sedikit pada frekuensi mod pertama kerana bunyi yang dihasilkan daripada hentakan tidak konsisten semasa menjalankan ujian tukul impak. Seperti yang dirancang bagi menyiasat keberkesanan penyerap getaran, eksperimen analisis getaran dijalankan dengan menggunakan eksperimen getaran motor. Daripada analisis getaran itu menunjukkan bahawa pengurangan purata apabila plat dipasang dengan penyerap getaran tunggal adalah 84%. Bagaimanapun, apabila plat dipasang dengan dua penyerap getaran dinamik, ia memberi impak yang lebih besar pada pengurangan amplitud getaran iaitu kira-kira 94% pada semua mod frekuensi. Daripada keputusan ini, dapat disimpulkan bahawa penyerap getaran berganda boleh mengurangkan getaran struktur global secara keseluruhan berbanding dengan penyerap tunggal. Selain itu, kajian ini menunjukkan bahawa penyerap getaran yang direka dalam kajian ini adalah penting dalam mengurangkan getaran pada setiap struktur yang bergetar. Daripada kedua-dua hasil kajian dapat disimpulkan bahawa getaran bebas dan getaran daya menggunakan kaedah unsur terhingga didapati disokong dengan eksperimen.

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LIST OF SYMBOLS

a	typical dimension
A	Sectional Area
c_1	Viscous damping of host structure
E	Young's modulus
$f(t)$	harmonic pertubation
h	plate thickness
I	Area moment of inertia
k_1	Stiffness of host structure
L	Length
m_1	Mass of host structure
m_a	Mass of absorber
N	force
ρ	Density
u	force control input
v	Volume
X_1	Displacement of primary structure
X_2	Displacement of absorber

CHAPTER 1

INTRODUCTION

This chapter introduces research background, including problem statement, expected outcomes and the significant of knowledge derived herein. It highlights the aim of this project in context of knowledge gaps identified in this field. The chapter ends with organization of the whole thesis and a brief description for each chapter.

1.1 Research Background

Vibrations are of much importance in a number of different branches of engineering, where in general, they are an undesirable phenomenon. Vibration is caused by environmental factors, rotating components and human activity which has the possibility of causing vibrations in a structure (Breukelman & Eng, 2004). The effects of these vibrations include: (i) an excessive variable stress on machine components, (ii) undesirable noise looseness of parts, (iii) shorten the fatigue life of the structure, and eventually (iv) the failure of the system (Burdisso & Heilmann, 1998). Exposure to excessive vibration can also be harmful to the human body, such as muscular pain and human discomfort (Bosco *et al.*, 2000).

Over the past two decades, there are numerous research works concerning on excessive vibration of light structure and thin panel, such as plate (Carneal *et al.*, 2004; Claeys *et al.*, 2009). The thin plate structures are commonly employed in the engineering applications, for example in automobiles, machinery, building structures and electronic components (Ranjan & Ghosh, 2005). With the trend use of thin plate by the manufacturing industries, environmental vibrations are expected to increase due to its adverse effects on the vibratory behaviour. Therefore, a thin plate stability subjected to dynamic force and controlling its vibration would be a topic of interest to be studied as it could lead to undesirable vibrations (Fuller *et al.*, 1996; Zaman *et al.*, 2013).

Basically, there are many different ways to control vibration. One is to use dynamic vibration absorber (DVA). The idea of DVA emerges due to a common vibration problem existed in engineering structural applications (Den Hartog, 1956; Thomson, 1996). In general, the DVA consists of a mass attached to a structure to be controlled through a spring-damper system. A DVA device system has been widely used to suppress or attenuate vibrations in linear, nonlinear vibrating mechanical systems and many structures (Darus & Tokhi, 2005). The applications of this device can be found particularly in bridges, building structures, aeroplanes, machine tools and others engineering systems (Taniguchi, Der Kiureghian & Melkumyan, 2008; Caetano, Cunha, Moutinho & Magalhães, 2010; Beltran-Carbajal & Silva-Navarro, 2014). In recent years, semi-active and active-passive vibration absorbers have been proposed to suppress harmonic excitations with time-varying frequency. However, a semi-active and active vibration absorbers achieve vibration control with the need of external power supply by changing its dynamic parameters, such as stiffness or damping (Ji & Zhang, 2010). Without an external power source, the vibration absorber become useless, thus lead to uncontrollable structural vibrations.

Here, the research strive to present a passive system of lightweight DVA to reduce the vibration amplitude of a thin plate structure. The first stage of the study analyzes the dynamic characteristics of a plate structure through a finite element method. In this work, a thin plate with all edges clamped is considered. The initial planning (FEM) is to predict the modal characteristics of plate such as natural frequencies, mode shapes and vibration response prior to the attachment of vibration absorber on the plate. Later, the effectiveness of the vibration absorber to reduce the vibration amplitude of plate is examined systematically. Finally, the obtained model is validated with an experimental modal analysis in order to confirm the reliability of the produced results.

1.2 Problem Statement

The rapid advancement of technology has changed the way the manufacturing industry operates, where more lighter structures are being utilized. One of the example of light structure is thin plate, which commonly used in various fields such as aerospace, machinery, civil engineering structure, the boards in electronic equipment, computer peripherals, modern housing and marine industries. However, the trend towards lighter structure has potential to lead structure being more susceptible to vibration. Thus, may lead to problems including fatigue, instability and performance reduction, and eventually may cause damage to the highly stressed structures. These vibration problems may also cause acoustic disturbance due to the coupling of structural vibration and acoustic

fields. As a result, there are needs for vibration control with the adequate approach in order to avert all the aforementioned problems.

There are many studies have been devoted in the past to develop a control technique to reduce vibration that generated by machines. 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 isolation devices between adjacent sub-systems, and (iv) adding discrete masses onto equipment to absorb vibration. In spite of these studies thus far, it has not yet reach a situation where an appropriate control method that can fit to all vibration situations. In fact, the first three aforementioned methods are hard to be implemented and are not really effective because of their design complexity, costly and unfeasible at lower frequency.

The latter approach using discrete masses or known as dynamic vibration absorbers, however, are more sound because they do not contribute additional vibration energy to the structure. On top of that, they are commonly found to be efficient and considered as a low-cost device. Although this method has more practicality, they are concerned of adding extra weight to the host structure. This definitely will not bring any advantage to automotive and aircraft industries since the weight of a host system is vital to overall performance as it affects on the fuel efficiency of vehicle. Apart from that, the dynamic vibration absorber can only tackle or address the vibration at the targeted mode and not in the large frequency range. Therefore, the method still require some room of improvement in order to upgrade the device systems.

1.3 Objectives of Study

In order to address the above aforementioned problems, the objectives of the study are to be designed as follows:

- i. Develop a lightweight dynamic vibration absorber.
- ii. Investigate the effectiveness of lightweight dynamic vibration absorbers to reduce surplus vibration on a thin plate structure by using force vibration.
- iii. Analyze the effect of using dual dynamic vibration absorber to address vibration in large frequency range by using Finite Element Analysis and experimental approach.

1.4 Scopes of Study

The scopes of the study are listed as follows:

- i. Design and build the attachment of the absorber.
- ii. Two absorbers are used for the dynamic vibration absorber study.
- iii. The frequency range of study for a thin plate is 0-600 Hz.
- iv. The plate structure is fixed at all edges.
- v. The dimensions of plate are 450mm x 450mm x 1mm of width, length and thickness, respectively.
- vi. A commercial finite element package Ansys Workbench Software version 14.5 is used to perform the finite element simulation.
- vii. Experimental modal analysis is carried out by using an impact hammer test.
- viii. Force vibration analysis is performed by using a motor shaker which excited at 1200 rpm.

1.5 Significant of Study

The flexible thin rectangular plates structures are the most common used in the industrialized world and in abroad range of engineering applications. However thin structure can be easily influenced by the unwanted structural vibration which may lead to fatigue, instability even failure of the structure. Thus, the discrete mass such as vibration absorber is proposed in the study to suppress the unwanted vibration at all frequency mode. Hence, suppressing and controlling the resonance phenomenon can prevent the structure from the catastrophic damage.

The outcomes of this study are expected to revolutionize the current performance of absorbers over a broad frequency range, thus improve their versatility and effectiveness. The use of dynamic vibration absorber is anticipated to reduce the structural vibration significantly. However, with the use of dynamic vibration absorbers, the results are expectedly enhanced where the vibration amplitude of a host structure will be reduced remarkably over a wide frequency mode. Therefore, a better understanding regarding the role of dynamic vibration absorber in the vibration control can be accomplished, in particular achieving a global structural vibration reduction.

1.6 Thesis Outline

This thesis is organized into five chapters. A brief outline of the thesis contents is as follows:

- Chapter 1 presents an introduction to the research problem. It involves the background, objective and scope of the research as well as the problem statement.
- Chapter 2 focuses on a fundamental theory of the vibration, plate, basic theory about dynamic vibration absorber, theory of finite element method, modal analysis and some of the important experiment tools employed in this study. Apart of that, previous study related to this research about the dynamic vibration absorber attached on structures are also presented in the chapter.
- Chapter 3 describes the specification of plate and dynamic vibration absorber used in the research. This chapter serves the research methodology carried out in this study, including the used instrumentation, step by step procedure of finite element analysis of Ansys Workbench version 14.5 and procedure of experimental modal analysis.

- Chapter 4 begins by laying out the results and an in-depth discussion of finite element analysis and experimental modal analysis of bare plate, plate with attached single vibration absorber and plate with attached dual vibration absorbers, There are variety of methods employed in order to achieve simulation and measure studies, such as modal responses, harmonic responses and transient analysis. These techniques are employed to determine the natural frequency, mode shape, frequency response and time response of the plate.
-
- Chapter 5 is the last chapter that summarises the finding of the research project, as well as outline the directions for future research works.

CHAPTER 2

LITERATURE REVIEW

This chapter provides readers with a general review on the theory of vibration, theory of plate and the principle of dynamic vibration absorber. It also covers description of finite element method, modal analysis and vibration experimental tools used in this study. The last part of this section reviewed previous findings in dynamic vibration absorber, which include comprehensive discussion on design and methodology. This section aims to acquire important informations related to the area of study, in order to give more idea and identified research gaps before the project is implemented. All information was gathered mostly from the text books, journals and internet.

2.1 Theory of Vibration

Vibration is the mechanical oscillation of a particle, member, or a body from its position of equilibrium. It is the study that relates the repetitive motion of physical bodies to the forces acting on them. The rate of the vibration cycles is called 'frequency'. Repetitive motions that occur at relatively low frequencies are commonly called oscillations, while any repetitive motion at high frequencies, even with low amplitudes, and having irregular and random behaviour falls into the general class of vibration (Rao, 2007).

The basic concepts in the mechanics of vibration are space, time and mass. When a body is disturbed from its position, then by the elastic property of the material of the body, it tries to come back to its initial position. In general, it can be seen and felt that nearly everything vibrates in nature. Vibration sometimes can be very weak for identification. On the other hand, there are may be large devastating vibrations that occur because of manmade disasters or natural disasters such as earthquakes, winds, and tsunamis (McDonald, 2003; Peijun, 2005).

There are ‘good vibrations’ which serve a useful purpose. Some example of good vibrations are musical instruments, vehicle suspension system, vibratory testing of materials, and electronic units to filter out unwanted frequencies. Figure 2.1 shows some examples of good vibration that can be found nowadays. Also, there are ‘bad vibrations,’ which can be harmful to structure and human body as shown in Figure 2.2 (Inman & Singh, 2001; Harris *et al.*, 2002; Dilley, 2005). For many engineering systems, operation at resonance would be undesirable and could be harmful to human body. Therefore, suppression or attenuation of unacceptable vibrations through proper and comparative accurate design of machines and structures are appealing.

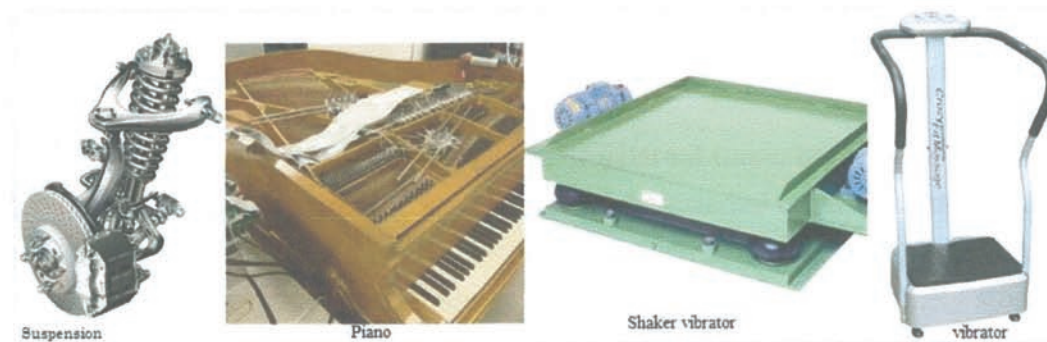


Figure 2.1: Examples of good vibration (Carignano *et al.*, 2014)

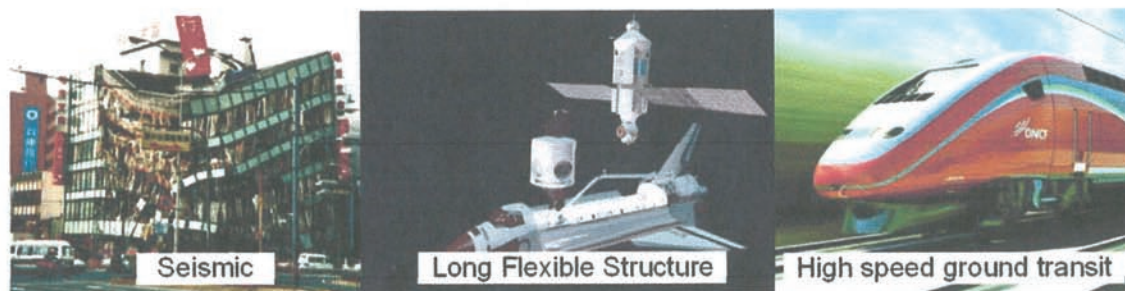


Figure 2.2: Examples of bad vibration (Benaroya & Nagurka, 2009)

2.1.1 Classification of vibration

Vibrations are often unwanted phenomena in structural engineering. Two types of vibrations can be distinguished, being free vibrations and forced vibrations. In free vibrations no energy is exchanged with the environment, while in forced vibrations there is energy exchange.

2.1.1.1 Free vibration

Vibration can be classified by looking at the number of degrees of freedom for their model, the type of force and the assumption by the model. Vibration which caused by initial energy in the system and not from other sources is called free vibration. There are other definition of free vibration which is oscillations where the total energy stays the same over time. This means that the amplitude of the vibration stays the same and no time varying external forces act on the system (De Silva, 2006; Shabana, 2012).

2.1.1.2 Force Vibration

If a system is subjected to a repeating type of external force, the resulting vibration is known as forced vibration. If the frequency of the external force coincides with the natural frequency of the system, resonance condition will take place to make the system vibrates with high amplitude. This must be avoided during the design of any structure and especially rotating machinery. However, if there is a resonance, the damping of the system will limit the amplitude of the resonance condition to some degree as the energy dissipated by friction and fluid structure interaction (Inman & Singh, 2001).

2.1.2 Dynamic Vibration characteristic

2.1.2.1 Natural frequency

If a system, after an initial disturbance, is left to vibrate on its own, the frequency which it oscillates without external forces is known as its natural frequency. 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, $\omega = v\lambda$. 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 undamped natural frequency can be calculated by using Eq. (2.1). While 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 as shown as in Eq. (2.2) (Rao & Yap, 2011; Thomson, 1996):

$$f_n = \frac{1}{2\pi} \sqrt{\frac{k}{m}} \quad (2.1)$$

$$\omega_d = \omega_n \sqrt{1 - \zeta^2} \quad (2.2)$$

When ever 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 as happened at Tacoma bridge shown in Figure 2.3.

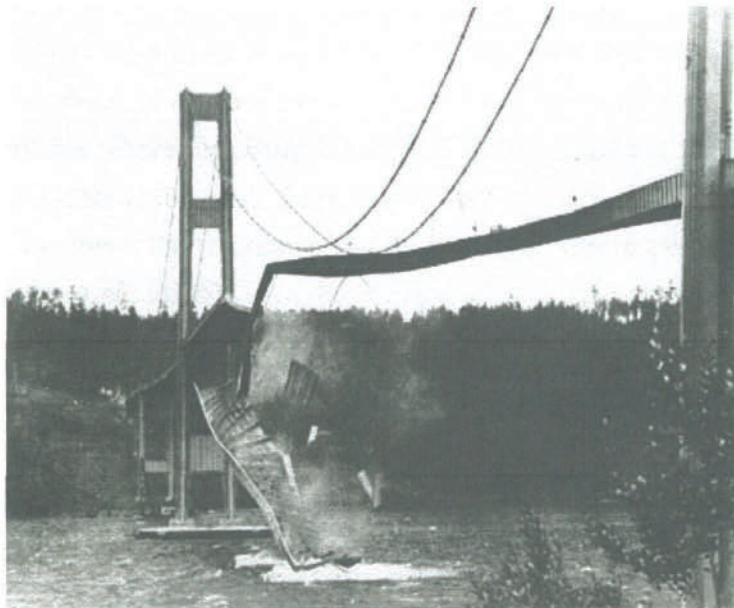


Figure 2.3: Tacoma Narrows bridge during wind-induced vibration (Green & Unruh, 2006)

2.1.2.2 Mode shape

Any complex body can vibrate in many different ways. These different ways of vibrating will have their own frequency and 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. Sometimes it is difficult to determine the shape of these modes. For example one cannot simply strike the object or displaced 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 to the next (Banerjee, 2001; Géradin & Rixen, 2014).

However, to discover both the frequency and the shape of the mode are possible by finding the resonance. If the mode has different frequencies 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 (Gérardin & Rixen, 2014).

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 resonant frequency of the first mode. This is not a general feature of modes. In general, the resonant frequencies of the modes have no simple relation to each other. As an example Figure 2.4 illustrates the modes shape of a free vibrating string. The solid lines are the shape of the mode on maximum displacement in one direction. Note that these are modes where the string is simply vibrating, and not twisting. For a thin string, the resonant frequencies of these modes tend to be much higher. However the thicker the bar, the lower the frequencies of these modes with respect to the vibrational modes (Wirsching *et al.*, 2006).

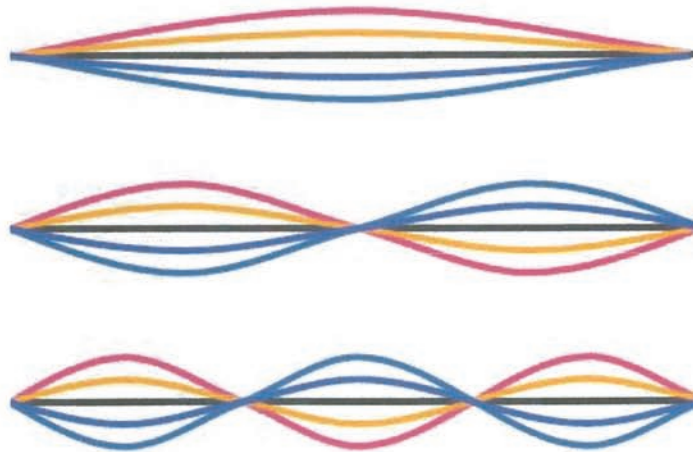


Figure 2.4: Mode shapes of a string (Wirsching *et al.*, 2006)

2.1.2.3 Damping

Damping present in all oscillatory system and the main effect of it is to remove energy from a system. There are two possibility of energy in vibration system which either it dissipated into heat or radiated away. In vibration analysis, it is generally concerned with damping in terms of system response. The energy from oscillation that loss will result in the decaying of amplitude of free vibration as shown in Figure 2.5. Most of the

structures and machinery in the real world nowadays have little or no damping . If it does not possess any damping, no mechanism exist to remove the vibrational energy in it, thus if imply any vibration motion once it set up, it will never stop and continuously forever (Hagood & von Flotow, 1991; Hosaka *et al.*, 1995).

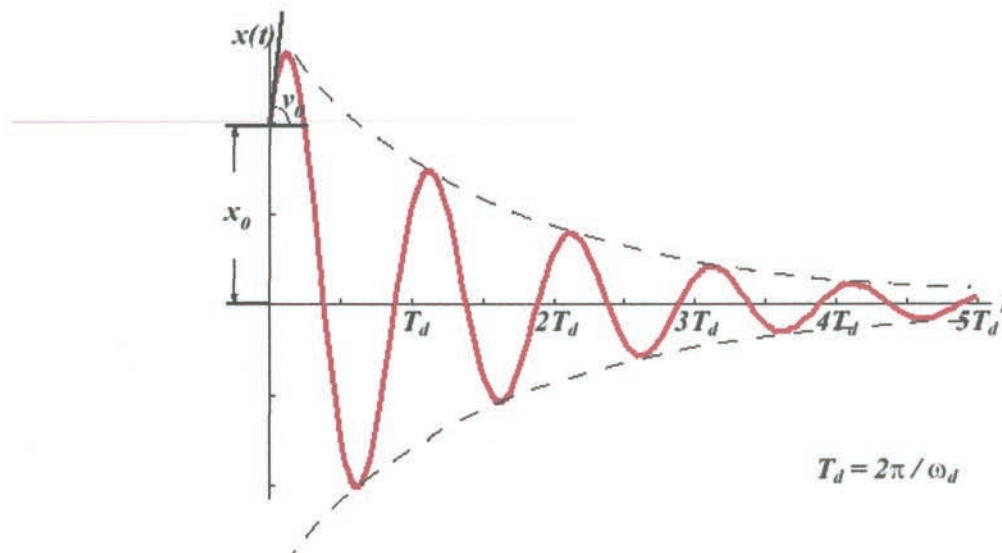


Figure 2.5: Effect of damping in the vibration response (Beranek & Ver, 1992)

2.1.3 Causes of vibrations

There are various source of vibrations in an industrial environment such as listed as follows (Harris *et al.*, 2002):

- Impact process such as pile driving and blasting.
- Unbalance of rotating component in machinery such as engines, compressors and motors.
- Gear wear in transportation vehicles such as trains and aircraft.
- Flow of fluids through pipes and without pipes.
- Natural calamities such as earthquakes and wind.

2.1.4 Analysis of structural vibration

It is necessary to analyze the vibration of structures in order to predict the natural frequencies and the response of the structure. The natural frequencies of the structure must be found because if the structure is excited at one of these frequencies, resonance will occur, thus resulting high vibration amplitudes, dynamic stresses and noise levels. Meaning that resonance should be avoided at the design stage so that it is not encountered during normal conditions. Therefore, it is important for the structure to be analyzed over the expected frequency range of excitation (Fahy & Gardonio, 2007).

It may be possible to analyze the complete structure, thus often leads to a complicated analysis and production of much unwanted information. A simplified mathematical model of the structure is usually sought and hence will produce the desired information as economically as possible and with an acceptable accuracy. The derivation of a simple mathematical model to represent the dynamics of a real structure is not easy if the model is to produce useful and realistic information. Thus it is often desirable for the model to predict the critical location of deform nodes in the structure (Reza *et al.*, 2010).

2.2 Vibration Control

Reducing either the excitation or the response of the structure to that excitation or both can attenuate the level of vibration in a structure. It is sometimes possible at the design stages to reduce the exciting force or motion by changing the equipment responsible, by relocating it within the structure or by isolating it from the structures so that the generated vibration is not transmitted to the supports. The structural response can be altered by changing the mass or stiffness of the structure when moving the source of the excitation to another location or by increasing the damping in the structure. Naturally, careful analysis is necessary to predict all the effects of any changes, whether at the design stage or as modification to an existing structure (Soong & Costantinou, 2014).

A vibration absorber is another common solution to protect the device from steady state harmonic disturbance at a constant frequency. This approach assists the natural frequency of the system by shifting it away from the excitation frequency in order to reduce the resonance and surpluses vibration (Steffen Jr & Rade, 2001; Dayou & Brennan, 2002; Sun *et al.*, 2010; Brennan, 2006; Wright, 2009). There are two types of vibration absorber such as passive and active vibration absorber.

2.2.1 Active vibration absorber

Active vibration control is one of the fundamental control design methodologies other than passive for vibration absorber. Active vibration control achieves dynamic performance by adding degrees of freedom to the system and controlling actuator forces depending on feedback and feedforward real-time information of the system, which obtained from sensor (Williams *et al.*, 2002; Chen *et al.*, 2005; Preumont, 2012). The vibrating mechanical system as shown in Figure 2.6, which consists of an active damped vibration absorber coupled to the primary system. The generalized coordinates are the displacements of both masses X_1 and X_2 , respectively. The u is represents the force control input and $f(t)$ some harmonic perturbation, possibly unknown. Here m_1 , k_1 and c_1 denote mass, linear stiffness and linear viscous damping on the primary system, respectively. Similarly m_a , k_a and c_a denote mass, stiffness and viscous damping of the vibration absorber. When $u = 0$ the active vibration absorber becomes only a passive vibration absorber (Crocker, 2007).

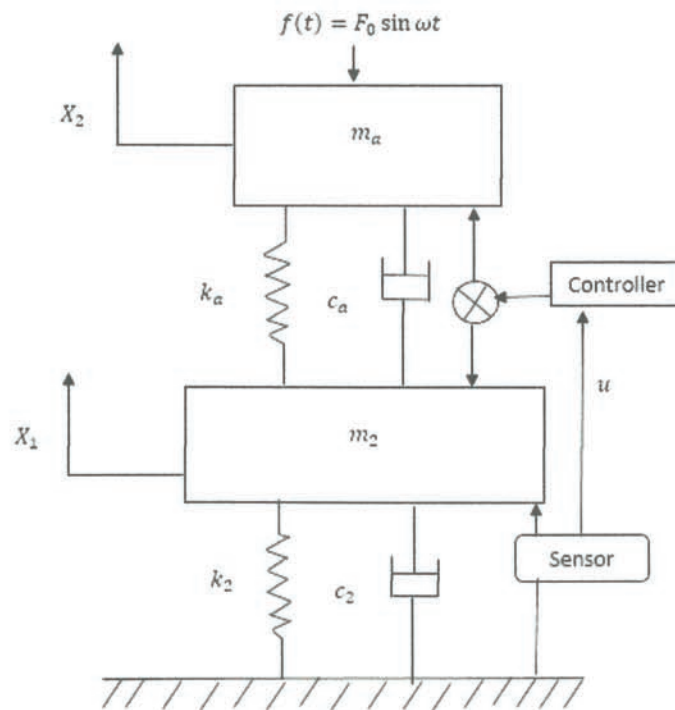


Figure 2.6: Active vibration absorber model

Theoretically, by adding the structural modification, it can produce the passive control that can be thought. Therefore, to improve the vibrational response of the system by choosing α which represents added stiffness, then it can be declared that as a passive control procedure. The use of added power or energy can distinguished between passive control and active control. Generally, the vibration absorber is the

most common passive control device and apart from that, the other methods of passive control are by adding mass and changing stiffness values (Moheimani & Fleming, 2006).

2.2.2 Passive vibration absorber

A mass spring subsystem coupled to a superstructure to control its oscillations under the action of periodic excitation is known as a passive vibration absorber. Figure 2.7 indicates a simple form of passive vibration absorber, where m_1 is a mass emulating the primary structure, k_1 is its mounting spring and c_1 is its viscous damping. The second mass m_a and the coupling spring k_a constitute the absorber system. After that, let X_1 be the displacement of m_1 and X_2 the displacement of m_a and it is assumed that identical forces with two harmonic components are applied to the primary mass of the system (Korenev & Reznikov, 1993; Hartung *et al.*, 2001).

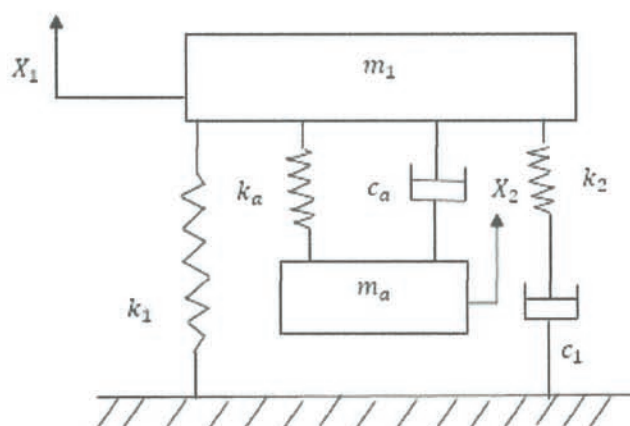


Figure 2.7: Passive vibration absorber model

2.3 Dynamic Vibration Absorber

A dynamic vibration absorber is a passive 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 has certain advantages over other methods of vibration suppression. It is external to the structure, therefore no re-installation of equipment necessary. On top of that, a dynamic vibration absorber can be designed and tested before installation (Wright, 2009; Brennan & Gatti, 2012). In many scenarios, this advantages offer an economical vibration reduction solution to engineering.

A dynamic vibration absorber is useful for situations in which the disturbance has a constant frequency. As opposed to a vibration isolator, which contains stiffness and damping elements, a vibration absorber 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 DVA mass has two degrees of freedom, and thus the new system has two natural frequencies (Lee *et al.*, 2011; Gao *et al.*, 2011).

If the DVA 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 DVA acts like a controller that has an internal model of the disturbance, which therefore cancels the effect of the disturbance. This disturbing input frequency generally match with the natural frequency of the original system, and from there the values for the DVA's mass and stiffness can be selected so that the motion of the original mass is very small. Meaning 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 DVA's mass and stiffness. Thus, the resulting motion of DVA will be large without affecting the motion of primary mass (Carneal *et al.*, 2004).

2.4 Theory of Plate

One of the most commonly used structures in the industrial is plate. Structural elements of a flat plate having a smaller thickness than the dimensions of the sides. Their strength and dynamic response plays an important role in determining their effective behaviour. Knowledge of the relationship between physical structural characteristics and modal parameters can provide considerable great insight into the methods to suppress the effect of undesirable structural responses (Saffry *et al.*, 2013; Carrera *et al.*, 2011).

In general, plates are initially flat structural members bounded by two parallel planes called faces, and a cylindrical surface called an edge or boundary. The generators of the cylindrical surface are perpendicular to plane faces. The distance between the plane faces is called the thickness, h . The flexural properties of a plate depend greatly upon its thickness in comparison with other dimensions. Plates may be classified into three groups according to the ratio a/h , where a is a typical dimension of a plate in plate and h is a plate thickness (Szilard, 2004).

1. The first group is presented by thick plates having ratio $a/h \leq 8 \dots 10$. The analysis of such bodies includes all the components of stresses, strains, and displacements as for solid bodies using the general equations of three-dimensional elasticity.
2. The second group refers to plates with ratio $a/h \geq 80 \dots 100$. These plates are referred to as membranes and they are devoid of flexural rigidity. Membranes carry the lateral loads by axial tensile forces N (and shear forces) acting in the plate middle surface. These forces are called membrane forces and they produce projection on a vertical axis and thus balance a lateral load applied to the plate-membrane.
3. The most extensive group represents an intermediate type of plate, so called thin plate with $8 \dots 10 \leq a/h \leq 80 \dots 100$. Depending on the value of the ratio w/h , the ratio of the maximum deflection of the plate to its thickness, the part of flexural and membrane forces here may be different. In this group, turn into different classes. (a) Stiff plates- a plate can be classified as a stiff plate if $w/h \leq 0.2$. Stiff plates are flexurally rigid thin plates. (b) Flexible plates- if the plate deflections are beyond a certain level, $w/h \geq 0.3$, then the lateral deflections will be accompanied by stretching of the middle surface. Such plates are referred to as flexible plates.

2.4.1 Applications of plate structures

The two-dimensional structural action of plate results in lighter structure, and therefore offers numerous economic advantages. The plate, being originally flat, develops shear forces, bending and twisting moments to resist transverse loads. Because the loads are generally carried in both directions and because the twisting rigidity in isotropic plates is quite significant, a plate is considerably stiffer than a beam of comparable span and thickness (Bercin & Langley, 1996).

So, thin plates combine light weight and form efficiency with high load-carrying capacity and technological effectiveness. Because of the distinct advantages discussed above, thin plates are extensively used in all fields of engineering. Various types of plate structural components like thin plates are commonly found in spacecrafts, missiles, aircrafts, land based vehicles, under-water vessels and structure, chemical processing, instruments, computer peripherals and modern housing. Figure Figure 2.8 shows some of the typical application of plate in industry (Cornwell *et al.*, 1999).



Figure 2.8: Plate application in engineering design (Vavva *et al.*, 2009)

2.4.2 Clamped-edges plate

Clamped-edges plate or known as fixed-fixed end plate is a plate that is clamped at all its boundary edges. In this study, this type of plate is selected as it allows neither vertical movement nor rotation at its boundary supports. This is the basic difference between a fixed-fixed plate and simply supported plate. So in a fixed-fixed plate, the supports will generate reactions at all axes as well as rotational moments. Figure 2.9 illustrates a plate structure with its boundary condition of fixed-fixed end (Bert & Mayberry, 1969).

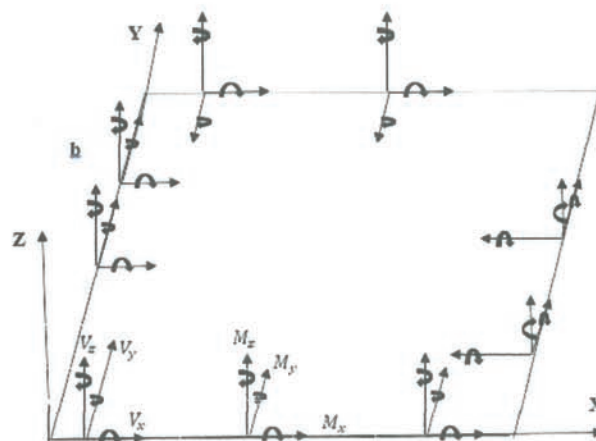


Figure 2.9: Boundary condition of fixed-fixed plate

The equation of motion of a clamped-clamped plate can be written as (Fuller *et al.*, 1996):

$$EI \left(\frac{\partial^4 \omega}{\partial x^4} + 2 \frac{\partial^2 \omega}{\partial x^2 \partial y^2} + \frac{\partial^4 \omega}{\partial y^4} \right) + \rho h \frac{\partial^2 \omega}{\partial t^2} = -F(x, y, t) \quad (2.3)$$

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 as in Eq. (2.4), where ν is the Poisson's ratio.

$$I = \frac{h^3}{12(1 - \nu^2)} \quad (2.4)$$

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 (Fuller *et al.*, 1996).

$$w(x, y, t) = \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} W_{mn} \cdot \Psi_{mn}(x, y) e^{j\omega_n t} \quad (2.5)$$

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 clamped-clamped plate can be calculated with two independent functions (Fuller *et al.*, 1996):

$$\Psi_{mn}(x, y) = X_m(x) \cdot Y_n(y) \quad (2.6)$$

The two independent functions X_m and Y_n can be calculated from Eqs. (2.7) and (2.8):

$$X_m(x) = \cosh(k_{mn}x) - \cos(k_{mn}x) - \beta_{mn} [\sinh(k_{mn}x) - \sin(k_{mn}x)] \quad (2.7)$$

$$Y_n(y) = \cosh(k_{mn}y) - \cos(k_{mn}y) - \beta_{mn} [\sinh(k_{mn}y) - \sin(k_{mn}y)] \quad (2.8)$$

where β_n and k_n are obtained in the respective Eqs. 2.9 and 2.10 (Fuller *et al.*, 1996).

$$\beta_{mn} = \frac{\cosh(k_{mn}L) - \cos(k_{mn}L)}{\sinh(k_{mn}L) - \sin(k_{mn}L)} \quad (2.9)$$

$$\cosh(k_{mn}L) \cdot \cos(k_{mn}L) - 1 = 0 \quad (2.10)$$

The natural frequencies of a clamped-clamped plate can be calculated from:

$$\omega_n = \sqrt{\frac{EI}{\rho h} [(k_m)^2 + (k_n)^2]} \quad (rad/s) \quad (2.11)$$

2.5 Finite Element Method

The finite element method is a numerical method that can be used for the accurate solution of complex mechanical and structural vibration problem. In this method, the actual structure is replaced by several pieces of elements and nodes, each of which is assumed to behave as a continuous structural member called a finite element. The element are assumed to be interconnected at certain points known as nodes (Bathe, 2008).

Finite element method solve engineering problem and it is now widely accepted by the engineering professions as an extremely valuable method of analysis. With the advances in computer technology and Computer Aided Design (CAD) systems, the complex problems can be modelled with relative ease. Several alternative configurations can be tried out on a computer before the first prototype is built. All of this suggests that it is vital to keep pace with these developments by understanding the basic theory, modelling techniques and computational aspects of the finite element method (Hughes *et al.*, 2005).

Nowadays, many engineers use finite element analysis for mechanical engineering design and optimization. It is virtually impossible to spend a day without using product software helped to researchers. Today, there is a lot of finite element software in the market that can be bought. Some of software are easy to use and user friendly. One of the popular finite element software uses is Ansys Workbench. Ansys is a very powerfully built finite element modelling package for numerically solving a large variety of mechanical problems (Lee, 2012).

Figure 2.10 shows some of the problem engineering's example that can be drawn and analyze using this finite element method.

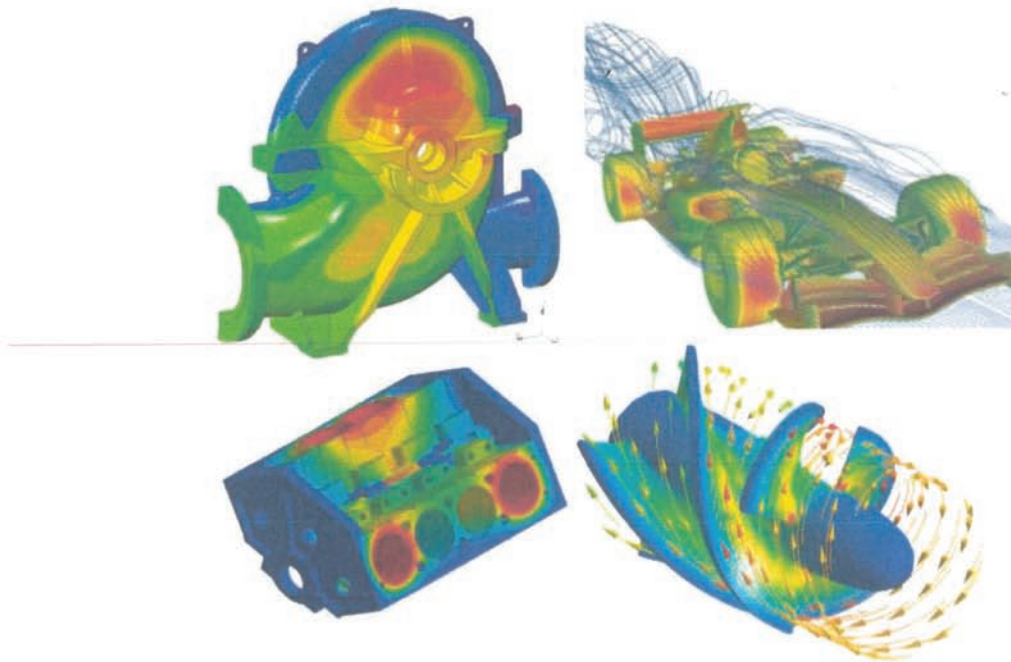


Figure 2.10: Finite element simulation of engineering problems (Hughes *et al.*, 2005)

2.5.1 Process of finite element analysis

Generally, the finite element analysis consists of three major steps. All this step is essential and must be done in order to solve the finite element problem. First step is preprocessing. Preprocessing procedure used to define the geometric domain of the problem and define element type. Then define material properties of the elements. After that, mesh the element to ensure connectivity between elements and nodes. Lastly apply model with physical constraints whereas boundary condition and loading (Bathe, 2006).

Second step is run the solution process. This step involves by computing the unknown values of the primary field variables, then the computed values used by substitution in order to compute additional variable such as reaction force, element stresses and heat flow. Last processed is post processing. Post processing stage contain sophisticated routines used for sorting, printing and plotting result from finite element solution (Morris, 2008).

2.6 Experimental Modal Analysis

In the past two decades, modal analysis has become a crucial method to determine, improve and optimize the dynamic characteristics of engineering structures. Modal analysis method not only has it been recognized in mechanical and aeronautical engineering, but modal analysis has also discovered profound applications for civil and building structures, biomechanical problems, space structures, acoustical instruments, transportation and nuclear plants (Dziedziech *et al.*, 2015; Heylen & Sas, 2006; Qu, 2004).

In general, the parameters that can be gathered from modal analysis include natural frequencies, damping ratios and mode shapes. These parameter are important not only to achieve desirable properties and performance but also can be used to prevent undesirable characteristics such as excessive noise, vibrations and catastrophic failures. Figure 2.11 shows that how the mode shape patterns extracted from the modal analysis when the excitation coincides with one of the natural frequencies of the system (Mohanty & Rixen, 2004).

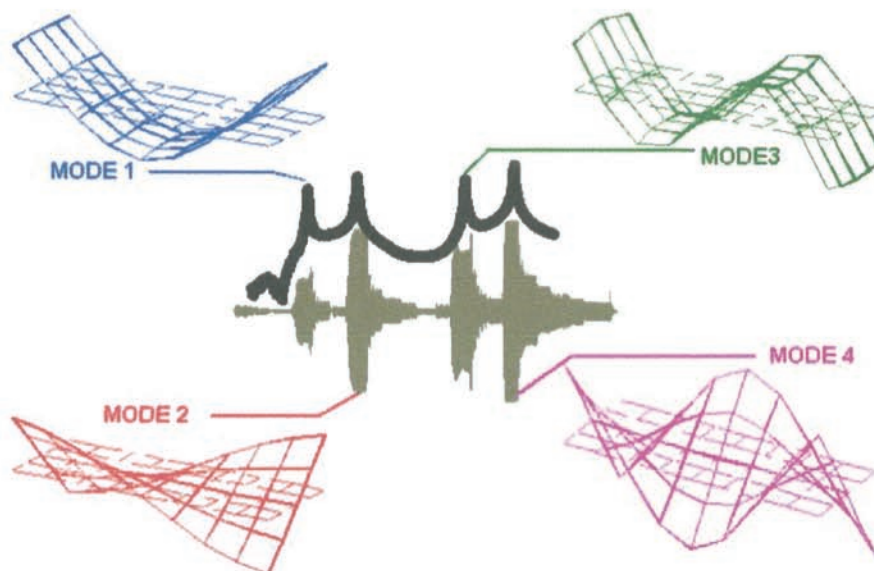


Figure 2.11: Modal analysis deformation pattern on the structure (Nieter & Singh, 1982)

Experimental modal analysis or commonly known as modal testing is an experimental technique used to derive the modal model of a linear time-invariant vibratory system. The theoretical basis of the technique is secured upon establishing the relationship between the vibration response at one location and excitation at the same or another location as a function of excitation frequency. This relationship, which is often

a complex mathematical function, is known as frequency response function, or FRF in short. Combinations of excitation and response at different locations lead to a complete set of frequency response functions (FRFs) which can be collectively represented by an FRF matrix of the system. This matrix is usually symmetric, reflecting the structural reciprocity of the system (Conciauro *et al.*, 2000).

Experimental modal analysis involves three constituent phases, test preparation, frequency response measurements and modal parameter identification. Test preparation involves: (i) selection of a structure's support, (ii) type of excitation forces, (iii) locations of excitation (iv) hardware to measure force, and (v) responses determination of a structural geometry model which consists of points of response to be measured and identification of mechanisms which could lead to inaccurate measurement. During the test, a set of FRF data is measured and stored which is then analyzed to identify modal parameters of the tested structure (Ewins, 2000).

2.6.1 Purpose of modal testing

Modal testing is useful in many application of design and development, production and quality assurance and qualification, utilization of a product. Depending on the outcome of a modal test, design modifications can be recommended for a preliminary design or a partial product. From preliminary design can be determined most desirable location in terms of minimal noise and vibration. Other than that, this method of testing can verify the performance of individual components of a complex system before the overall system is built and evaluate. On top of that, the tests are beneficial for designer and manufacturer in improving the quality of performance of the product (Rao & Ratnam, 2012).

Other than that, the purpose of this modal testing is to validate between the experimental and finite element results. The second is to quantify the extent of the differences or similarities between the two sets of data. Then third is to make adjustments or modifications to one or other set of results in order to bring them closer into line with each other through finite element modal updating approach (Ewins & Inman, 2001). In finite element model updating, experimental and analytical databases should be compared to assess the improvement in the modelled response. In this comparison, major problem arise because of some errors occurred during experimental testing and also the limitation in the finite element analysis.

2.6.2 Impact hammer

Impulse or impact hammer test commonly used in modal vibration testing. Impulse or impact hammer, which consists of a hammer with force transducer at its tips, balancing mass and handle sketched in Figure 2.12 shows the detail of impact hammer. Typical materials for the tip are rubber, plastic and steel. A piezoelectric or strain gauge type force sensor may be used. More sophisticated hammers have impedance heads in place of force sensors. An Impedance head measures force and acceleration simultaneously (Ewins, 2000).

Impact hammer does not need a signal generator and a power amplifier. The hammer by itself is the excitation mechanism and is used to impact the structure and thus excite a broad range of frequencies. On the other hand, as the impact hammer does not need a connecting device, its application avoids mass loading the test structure and it is faster than an exciter (Böhme & Kalthoff, 1982).

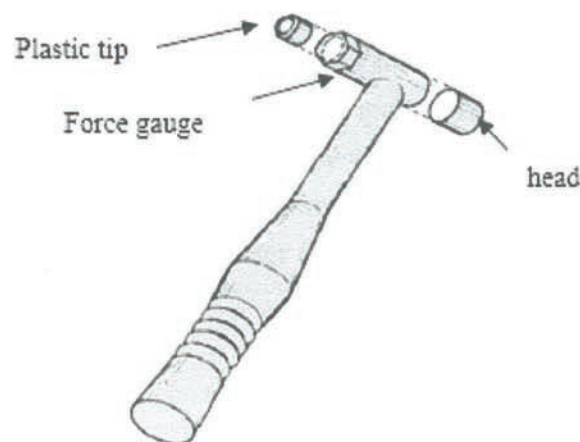


Figure 2.12: Impact hammer (Ewins, 2000)

2.6.3 Motor shaker

There are various hardware of excitation equipment that are able to excite the structure. The two most common ones are hammer and shaker such as the ones schematically represented in Figure 2.12 and Figure 2.13, respectively. Three basic types of vibration exciters are widely used whereas hydraulic shakers, inertial shakers and electromagnetic shakers driven by a power amplifier (Batel, 2002).

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