

**VIBRATION SIMULATION OF PLATE WITH ATTACHED MULTIPLE
ABSORBERS**

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For my beloved mother and father



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In the name of Allah, The Beneficent, The Merciful, none of this would have been possible.

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Nadia Nurnajihah binti Mohamad Nasir, Parit Raja

ABSTRACT

In general, solutions for standards to improve the vibration and noise problems is 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. So in this study, the concept of Dynamic Vibration Absorbers (DVA) are used to reduce vibration or amplitude. On a Simply Supported Plate (SSP) structure, the method used in this study were of analytical equations and finite element analysis. MATLAB ® is used to solve the equations of analytical theory, as well as simulation results verify the ANSYS ®. The study found that the analytical and simulation equation for SSP are same. The frequency range used are 10 Hz to 500 Hz and there are a number of modes show. In addition, the mass and damping in the absorber is changed. It aims to identify the mass and damping are most suitable for use in absorber system. Once seen, the mass of 1.0 kg more suitable for use as a percentage reduction or vibration amplitude is the highest of 40.99%. The damping values shows that the percentage reduction in the amplitude of the vibration or spring, 10 N / m is 44.64%. Further, this study was undertaken by placing the absorber at different locations configuration. This was followed by the addition of a single absorber and multiple absorber to see the average percentage reduction in vibration or amplitude. Overall global reduction of the four absorber is 68.07% compared to the single absorber only 33.31% reduction. Finally, it can be concluded that multiple vibration absorber can reduce the overall global vibration compared to a single absorber. However, for structures that take into account the weight of the main things to consider increasing the number of absorbers as excessive weight of the structure will result in a reduction in vehicle fuel efficiency, aerospace, automotive and machine systems.

ABSTRAK

Pada umumnya, penyelesaian bagi piawaian 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. Maka dalam kajian ini, konsep penyerap getaran dinamik digunakan untuk mengurangkan getaran atau amplitud. Pada sesuatu struktur plat, kaedah yang digunakan dalam kajian ini adalah persamaan analitikal dan analisis unsur terhingga. MATLAB® digunakan untuk menyelesaikan teori persamaan analitikal, disamping mengesahkan keputusan simulasi ANSYS®. Keputusan kajian mendapati persamaan analitikal dan simulasi plat adalah sama. Julat frekuensi yang digunakan adalah 10 Hz hingga 500 Hz dan terdapat beberapa bentuk mod yang terhasil. Sebagai tambahan, nilai jisim dan redaman pada penyerap diubah. Ia bertujuan untuk melihat jisim dan redaman yang paling sesuai untuk digunakan dalam sistem penyerap. Setelah dilihat, nilai jisim 1.0 kg lebih sesuai digunakan kerana peratusan pengurangan getaran atau amplitud adalah yang paling tinggi iaitu 40.99%. Manakala nilai redaman pula menunjukkan bahawa peratusan pengurangan getaran atau amplitud bagi nilai pegas, 10 Ns/m adalah 44.64%. Seterusnya, kajian ini dijalankan dengan meletakkan penyerap pada lokasi yang berlainan konfigurasi. Ini diikuti dengan penambahan penyerap tunggal dan penyerap berganda untuk melihat peratusan purata pengurangan getaran atau amplitud. Pengurangan global keseluruhan bagi empat penyerap adalah 68.07% berbanding dengan penyerap tunggal yang hanya 33.31% pengurangan. Akhirnya, dapat disimpulkan bahawa penyerap getaran berganda dapat mengurangkan keseluruhan getaran global berbanding dengan penyerap tunggal. Walaubagaimanapun, bagi struktur yang mengambil kira berat sebagai perkara utama penambahan bilangan penyerap perlu dipertimbangkan kerana penambahan berat yang berlebihan pada struktur akan mengakibatkan pengurangan kecekapan bahan api pada kenderaan, aeroangkasa, sistem automotif dan mesin.

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PTTA UTHM
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LIST OF SYMBOLS

ν	Poisson Ratio
ρ	Density
ζ	Viscous Damping
a	Length
b	Width
c	Damping Coefficient
E	Young's Modulus
h	Thickness
I	Arae moment of inersia
k	Spring Stiffness
m	Mass
m	Mode Number
n	Mode Number

CHAPTER 1

INTRODUCTION

Nowadays, vibration has become a major concern in most engineering fields and it is one of the most common aspects of life. It has turned out to be a critical and growing problem in the society. Many natural phenomena, as well as man-made devices, involve periodic motion of some sort. In man-made devices vibration is often less impressive, but it can be a symptom of malfunctioning and is often a signal of danger.

Vibration can be put to work for many useful purposes like vibrating sieves, mixers, and tool are the most obvious examples. Vibrating machines also find applications in medicine, curing human disease. Another useful aspect of vibration is that it conveys a quantity of useful information about the machine producing it.

In this century, the increase of the number of dynamic systems, such as in mechanical tools, machinery, automobiles and aircraft, environmental vibration are expected to increase in the future and affect more people in their everyday life. Over the past 20 years, there are numerous research works concerning on excessive vibration through the thin-walled structures such as plate. This is a problem commonly found in automobiles, in aircraft and the fairings of rocket launch vehicles.

1.1 Research Background

Vibration is the mechanical oscillations, that produced by the regular or irregular motion of a particle or a body or systems that connected bodies displaced from a position

of equilibrium. Vibration can be a source of problem at an engineering level because resulting in damage and loss of control of equipment and thus reducing the efficiency of operation in machines. They are produce increased stresses, energy losses, because added wear, increase hearing loads, induce fatigue, create passenger discomfort in vehicles and absorb energy from the system. Vibration can also cause discomfort at a low-high level that can be risk to the person safety [1, 2].

Each vibration structure has tendency to oscillate with large amplitude at certain frequencies. These frequencies are known as resonance frequencies or natural frequencies of the structure. At these resonance frequencies, even a small periodic driving force can result in large amplitude vibration. When resonance occurs, the structure will start to vibrate excessively.

The primary method of eliminating vibration is at a source by designing the equipment and ensuring control over the manufacturing tolerances. Others method that can reduce the vibrations that generated by machinery is by modifying the system so that the natural frequencies are not close to the operating speed, to prevent large responses by including damping, install vibration isolating devices between adjacent sub-systems and the other way is include auxiliary mass into the equipment to reduce the response and absorb vibration. [3]

This research aims to develop a new control strategy using multiple passive vibration absorbers attached to a flexible thin plate and tuned to operational frequency, in such a way to counter the vibrating force across the structure globally. The properties of the absorbers are also adapted in order to minimize the vibration level of the structure at the maximum performance.

1.2 Problem Statement

In this research are concerns in excessive vibration through the thin-walled structure such as plate. It is a problem commonly found in automobiles, in aircraft and the fairings of rocket launch vehicles.

Generally, the standard solution to ameliorate this vibration problem is to re-design and modify the system, which is done by combination of increasing the thickness of the panel wall, stiffening the structure, and adding damping. In the past, many studies have been devoted to develop a feasible method 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 isolating devices between adjacent sub-systems, and (iv) adding discrete masses into equipment to reduce the response and absorb vibration.

However, all these solutions are difficult to apply and adding excessive weight to the host structure that eventually has disadvantage on the fuel efficiency of vehicle, in particular aerospace and automotive systems. In fact, the first three aforementioned methods are hard to be implemented and not really effective because of its design complexity, costly, and unfeasible at lower frequency. The latter approach using discrete masses or known as passive vibration absorbers, however, are sounder because they do not contribute significant additional vibrational energy to the structure, besides proven to yield substantial attenuation in structural vibration.

1.3 Objectives of Study

The objective of this research is to study and simulate the vibration characteristics of a vibration of a simply-supported plate without and with attached multiple absorbers by using ANSYS® and MATLAB®. Based on the research, there are several objectives that need to achieve.

- i. To determine the vibration characteristics of a simply supported plate.
- ii. To determine the vibration reduction of a single vibration absorber attached to a plate.
- iii. To investigate the effect of mass and damping as well as absorber location on the absorber's performance.
- iv. To determine the effect of attaching multiple vibration absorbers to reduce vibration level of a simply supported plate.
- v. To provide guidelines for optimum numbers of passive vibration absorbers used, mass, damping properties and its placement in order to have substantial vibration attenuation.

1.4 Scopes of Study

The research is limited according to the scopes below:

- i. Two approach will be carried out with
 - numerical simulation works by ANSYS®
 - analytical analysis by MATLAB®
- ii. The element type used for the numerical simulation in ANSYS® is hexahedral 8 nodes.
- iii. Mathematical models regarding simply-supported plate with and without attached absorber will be derived.
- iv. The dimension of a plate to be studied 450 x 450 x 2 mm.
- v. The number of absorbers to be studied will be limited to 5.

1.5 Expected Outcomes

There are several contributions to the body of knowledge presented in this research and it will, (i) identify the vibration characteristics of a simply supported plate, (ii) with the developments of multiple passive vibration absorbers which are used to target wide frequency range are able the reduce the global vibration of the structure, (iii) provide guidelines for optimum numbers of passive vibration absorbers used and its placement in order to have substantial vibration attenuation, and (iv) with using several software, time and cost consuming will be reduced significantly, while at the same time design and modification can be done repeatedly without any hassle.

1.6 Significant of Study

In engineering history, excessive vibration has been a common problem in causing the fatigue life of structures shorter and the performance of machines reduces. 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 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 absorber due to the needs of vibration protection itself. As a result, it gave an idea on how to produce an effective absorber. The knowledge gained from this research can be used to minimize the vibration amplitude of a structures and machines, increasing their life-span simultaneously.

A complete understanding of vibration is needed involves in the analysis and design of a vibration absorber 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 fundamental in terms of identifying and deriving theoretical and mathematical model in development of dynamic vibration absorber for multi degree freedom systems. The other benefit comes from this research in specific or potential application aspect is it could control vibration in building or bridge structure and airplane wing flutter control. Therefore, it is judge to be important for doing this research.



PT TAA
PERPUSTAKAAN TUNKU AMINAH

CHAPTER 2

LITERATURE REVIEW

This chapter explains about vibration theory, vibration control, introduction of thin-walled structure such as plate, dynamic vibration absorber (DVA) and previous research on the dynamic absorber.

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 [4].

2.1.1 Natural Frequency

The natural frequency is the rate at which an object vibrates when it is not disturbed by an outside force. 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, $\omega = v \cdot \lambda$. 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, $\omega_d = \sqrt{1 - \zeta^2} \cdot \omega_n$.

2.1.2 Modes 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 [5].

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 2.1.

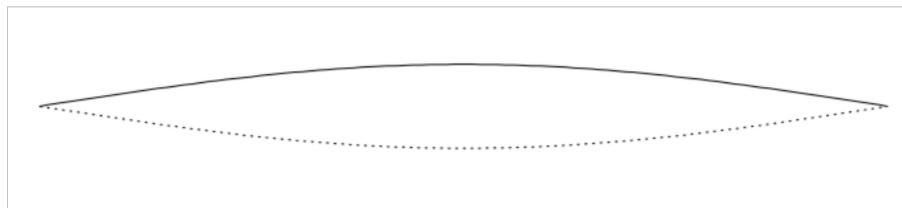


Figure 2.1: The motion in the centre of the string.

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 2.2. As one half vibrates up, the other moves down, and are vice versa.

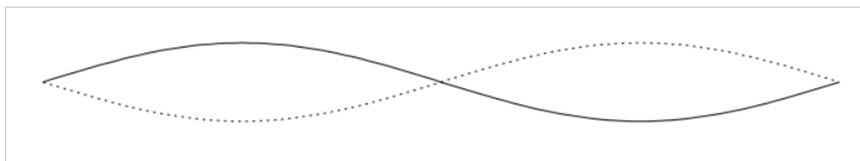


Figure 2.2: The two halves of the string vibrate in opposition each other.

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 2.3 shows the string is divided into three equal length sections, each vibrating in opposition to the adjacent piece.

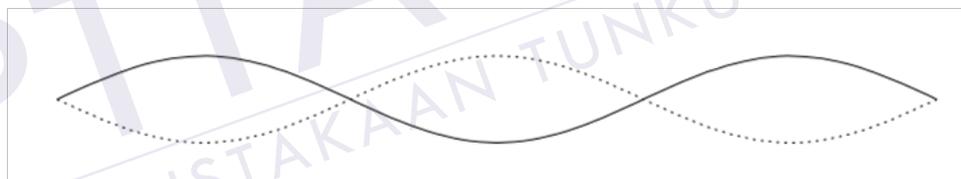


Figure 2.3: The string is divided into three equal length section.

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 2.4, 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.

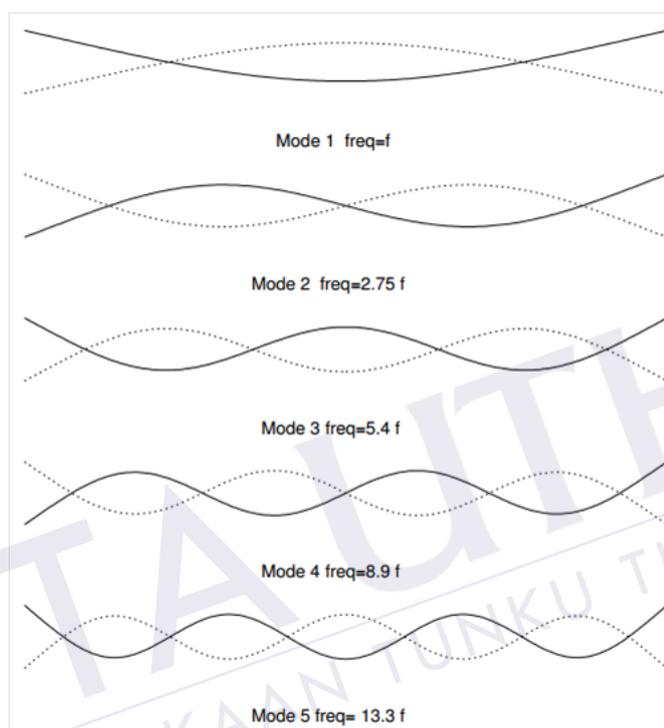


Figure 2.4: The string mode with difference frequencies.

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 its centre, 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.

2.1.3 Mass-Spring-Damper

The vibration analysis can be understood by studying the simple mass–spring–damper model. Indeed, even a complex structure such as an automobile body can be modeled as a "summation" of simple mass–spring–damper models. The mass–spring–damper model is an example of a simple harmonic oscillator in Figure 2.5. The mathematics used to describe its behavior is identical to other simple harmonic oscillators.

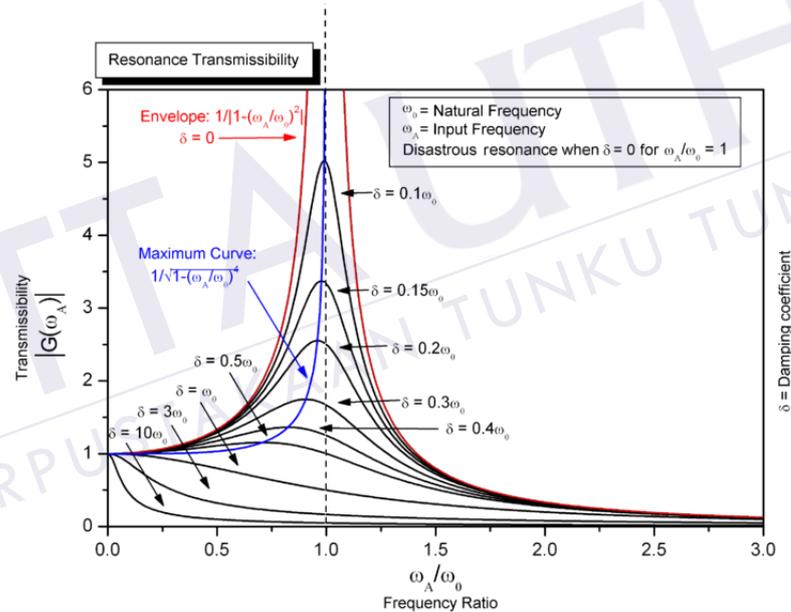


Figure 2.5: A simple harmonic oscillator.

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

Vibrations are classified into the following types:

1. Undamped and damped vibration: There is no loss or dissipation of energy due to friction or other resistance during vibration of a system, the system is said to be undamped. If there is energy loss due to the presence of damping, the system is called damped.
2. Free and forced vibration: A system vibrates due to an initial disturbance (with no external force applied after time zero), the system is said to undergo free vibration. On the other hand, if the system vibrates due to the application of an external force, the system is said to be under forced vibration.
3. Linear and nonlinear vibration: All the basic components of a vibrating system (e.g., mass, the spring, and the damper) behave linearly, the resulting vibration is called linear vibration. However, if any of the basic components of a vibrating system behave nonlinearly, the resulting vibration is called nonlinear vibration.

The physical explanation of the phenomena of vibration concerns the interplay between potential energy and kinetic energy. A vibrating system must have a component that stores potential energy and releases it as kinetic energy in the form of motion (vibration) of a mass and it is shown in Figure 2.6. The motion of the mass then gives up kinetic energy to the potential energy storing device. Vibration can occur in many directions and can be the result of the interaction from many objects.

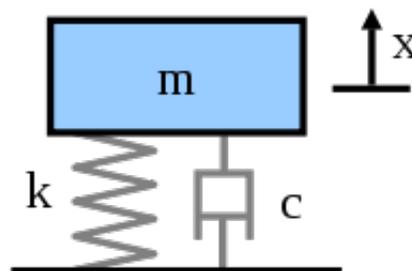


Figure 2.6: Elements of vibratory system where: m -mass (stores kinetic energy); k -spring (stores potential energy, support load) and c -damper (dissipates energy, cannot support load).

Vibration can be caused by many types of excitation. These include:

1. Fluid flow
2. Reciprocating machinery
3. Rotating unbalanced machinery
4. Motion induced in vehicles travelling over uneven surfaces.
5. Ground motion caused by earthquake.

2.2 Vibration Analysis

A vibratory system is a dynamic system for which the response (output) depends on the excitations (input) and the characteristic of the system (e.g; mass stiffness and damping) as indicated in Figure 2.7. The excitation and response of the system are both time dependent. Vibration analysis of a given system involves determination of the response for the excitation specified. The analysis usually involves mathematical modelling, derivation of the governing equation of motion, solution of the equations of motion, and interpretation of the response results.

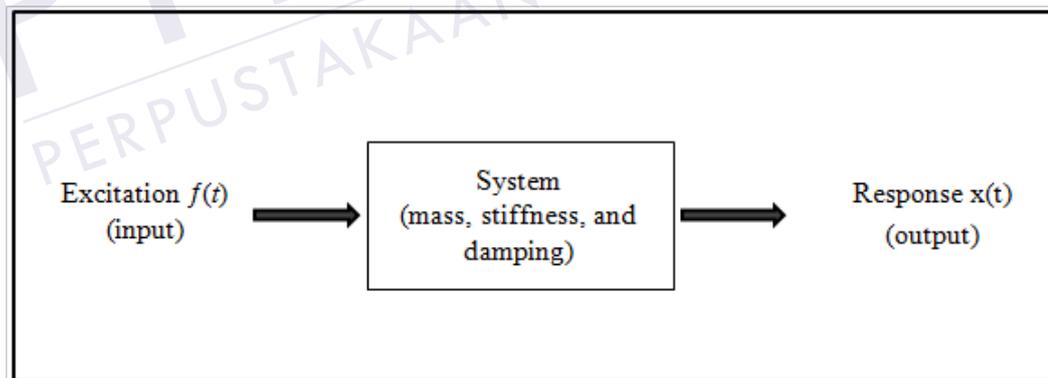


Figure 2.7: Input-output relationship of a vibratory system.

The purpose of mathematical modelling is to represent all at the important characteristic of a system for the purpose of deriving mathematical equations that govern the behaviour of the system. The mathematical model may be linear or nonlinear, depending on the nature of the system characteristic. Once the mathematical model is selected, the principles of dynamics are used to derive the equations of motion of the vibrating system. For this, the free-body diagrams of masses, indication all externally applied forces (excitations), reaction forces, and inertia forces, can be used.

2.3 Plate Structure

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 of the plate. Assume that the plate thickness is small compared with other characteristic dimensions of the faces (length, width, diameter, etc.).

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. So, thin plates combine light weight and form efficiency with high load-carrying capacity, economy, 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 2.8 shows typical application of plate in industry.

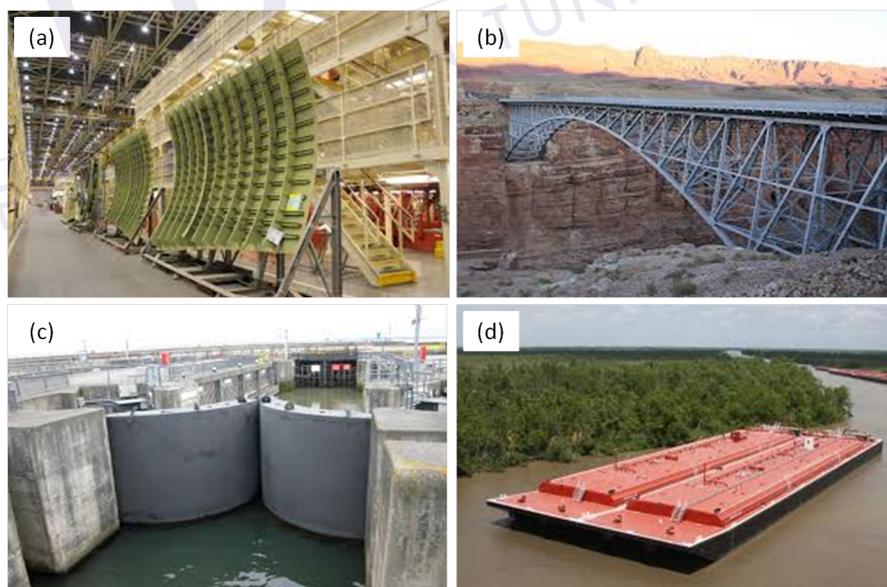


Figure 2.8: Example of thin plate in engineering fields, (a) Cell of Wing Skin, (b) Bridge Deck, (c) Lock-Gate, and (d) Oil Barge

A flat plate is typically thicker than a membrane, and has significant bending stiffness. Thus, like the beam, the equation of motion is now of fourth order. But, like membrane, two coordinates are needed to locate each point on the midplane of

the plate. Inplane vibrations would be the same as for the membrane, and seldom of interest.

Plate are commonly divide the thickness h into equal halves by a plane parallel to its faces. This plane is called the middle plane (or simply, the midplane) of the plate in Figure 2.9. Being subjected to transverse loads, an initially flat plate deforms and the midplane passes into some curvilinear surface, which is referred to as the middle surface. For such plates, the shape of a plate is adequately defined by describing the geometry of its middle plane. Depending on the shape of this midplane, we will distinguish between rectangular, circular, elliptic, or each plate.

A plate resists transverse loads by means of bending, exclusively. 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. These group are

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- it 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 [6].

2.3.1 General Behaviour of Plates

Consider a load-free plate, shown in Figure 2.9, in which the xy -plane coincides with the plate's midplane and the z coordinate is perpendicular to it and is directed downwards. The fundamental assumptions of the linear, elastic, small-deflection theory of bending for thin plates may be stated as follows:

1. The material of the plate is elastic, homogeneous, and isotropic.
2. The plate is initially flat.
3. The deflection (the normal component of the displacement vector) of the midplane is small compared with the thickness of the plate. The slope of the deflected surface is therefore very small and square of the slope is negligible quantity in comparison with unity.
4. The straight lines, initially normal to the middle plane before bending, remain straight and normal to the middle surface during the deformation, and the length of such elements is not altered. This means that the vertical shear strains γ_{xy} and γ_{yz} are negligible and the normal strain ϵ_z may also be omitted. This assumption is referred to as the 'hypothesis of straight normal.'
5. The stress normal to the middle plane, σ_z , is small compared with the other stress components and may be neglected in the stress-strain relations.
6. Since the displacements of a plate are small, it is assumed that the middle surface remains unstrained after bending.

Many of these assumptions, known as Kirchhoff's hypothesis, are analogous to those associated with the simple bending theory of beams. These assumptions result in the reduction of three-dimensional plate problem to a two-dimensional one. Consequently, the governing plate equation can be derived in a concise and straightforward manner. The plate bending theory based on the above assumptions is referred to as the classical or Kirchhoff's plate theory [7].

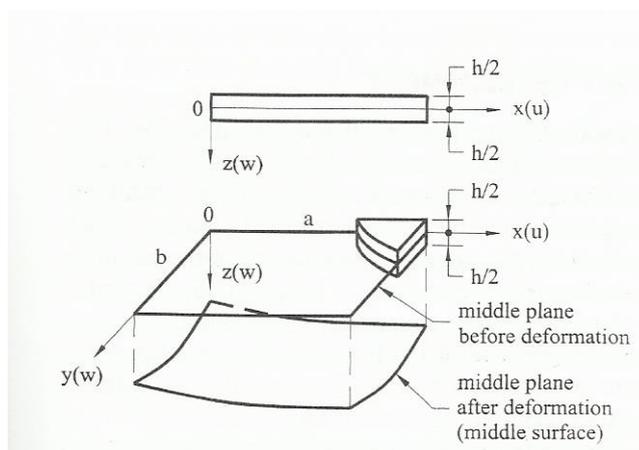


Figure 2.9: The middle plane of plate

2.4 Dynamic Vibration Absorber

Traditional treatment methods that involve structural modifications are often time consuming and expensive. Blocking the problem frequencies in the variable frequency drive limits the use of the system by the user. One possible solution is an installation of a dynamic vibration absorber.

A dynamic vibration absorber 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 absorber 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 absorber can be designed and tested before installation. In many scenarios, this offer an economical vibration reduction solution.

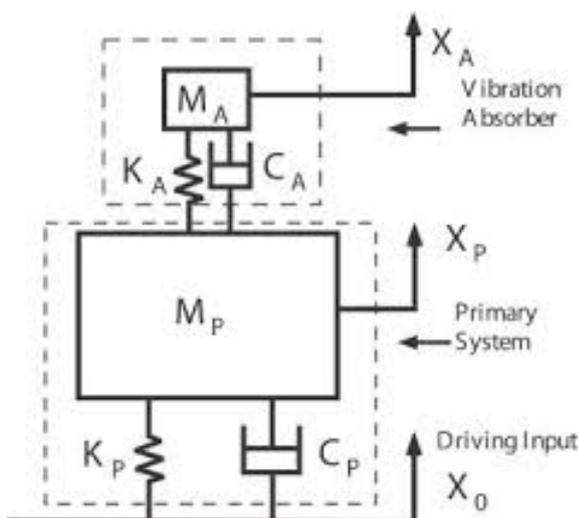


Figure 2.10: Example of Dynamic Vibration Absorber.

Figure 2.10 depicts a dual dynamic vibration absorber mounted on identical primary systems. M_A and M_P are the corresponding mass, K_A and K_P are the corresponding 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[5].

A 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 absorber mass has two degrees of freedom, and thus the new system has two natural frequencies.

If the absorber 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 absorber 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 absorber'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 absorber's mass and stiffness. Thus the resulting motion absorber will be large.



2.5 Finite Element Analysis (FEA)

Finite element method (FEM) is a numerical technique for finding approximate solutions to boundary value problems for differential equations. It uses variational methods (the calculus of variations) to minimize an error function and produce a stable solution. Analogous to the idea that connecting many tiny straight lines can approximate a larger circle, FEM encompasses all the methods for connecting many simple element equations over many small subdomains, named finite elements, to approximate a more complex equation over a larger domain.

Finite Element Analysis (FEA) was first developed in 1943 by R. Courant, who utilized the Ritz method of numerical analysis and minimization of variational calculus to obtain approximate solutions to vibration systems. Shortly thereafter, a paper published in 1956 by M. J. Turner, R. W. Clough, H. C. Martin, and L. J. Topp established a broader definition of numerical analysis. The paper centered on the "stiffness and deflection of complex structures".

By the early 70's, FEA was limited to expensive mainframe computers generally owned by the aeronautics, automotive, defense, and nuclear industries. Since the rapid decline in the cost of computers and the phenomenal increase in computing power, FEA has been developed to an incredible precision. Present day supercomputers are now able to produce accurate results for all kinds of parameters [8].

Engineering analysis of mechanical systems have been addressed by deriving differential equations relating the variables of through basic physical principles such as equilibrium, conservation of energy, conservation of mass, the laws of thermodynamics, Maxwell's equations and Newton's laws of motion. However, once formulated, solving the resulting mathematical models is often impossible, especially when the resulting models are nonlinear partial differential equations. Only very simple problems of regular geometry such as a rectangular of a circle with the simplest boundary conditions were tractable. There are several type of engineering analysis in finite element, (i) structural analysis consists of linear and non-linear models. Linear models use simple parameters and assume that the material is not plastically deformed. Non-linear models consist of stressing the material past its elastic capabilities. The stresses in the material then vary with the amount of deformation. (ii) Vibrational analysis is used to test a material against random vibrations, shock, and impact. Each of these incidences may act on the natural vibrational frequency of the material which, in turn, may cause resonance and subsequent failure. (iii) Fatigue analysis helps designers to predict the life of a material or structure by showing the effects of cyclic loading on

the specimen. Such analysis can show the areas where crack propagation is most likely to occur. Failure due to fatigue may also show the damage tolerance of the material. (iv) Heat Transfer analysis models the conductivity or thermal fluid dynamics of the material or structure. This may consist of a steady-state or transient transfer. Steady-state transfer refers to constant thermoproperties in the material that yield linear heat diffusion [9, 10].

The finite element method (FEM) is the dominant discretization technique in structural mechanics. The basic concept in the physical interpretation of the FEM is the subdivision of the mathematical model into disjoint (non-overlapping) components of simple geometry called finite elements or elements for short. The response of each element is expressed in terms of a finite number of degrees of freedom characterized as the value of an unknown function, or functions, at a set of nodal points.

The response of the mathematical model is then considered to be approximated by that of the discrete model obtained by connecting or assembling the collection of all elements. The disconnection-assembly concept occurs naturally when examining many artificial and natural systems. For example, it is easy to visualize an engine, bridge, building, airplane, or skeleton as fabricated from simpler components. Unlike finite difference models, finite elements do not overlap in space. There are some objectives using the FEM:

1. Understand the fundamental ideas of the FEM.
2. Know the behaviour and usage of each type of elements.
3. Be able to prepare a suitable FE model for structural mechanical analysis problems.
4. Can interpret and evaluate the quality of the results (know the physics of the problems).
5. Be aware of the limitations of the FEM (don't misuse the FEM - a numerical tool).

A variety of specializations under the umbrella of the mechanical engineering discipline (such as aeronautical, biomechanical, and automotive industries) commonly use integrated FEM in design and development of their products. Several modern FEM packages include specific components such as thermal, electromagnetic, fluid, and structural working environments. In a structural simulation, FEM helps tremendously

in producing stiffness and strength visualizations and also in minimizing weight, materials, and costs.

FEM allows detailed visualization of where structures bend or twist, and indicates the distribution of stresses and displacements. FEM software provides a wide range of simulation options for controlling the complexity of both modeling and analysis of a system. Similarly, the desired level of accuracy required and associated computational time requirements can be managed simultaneously to address most engineering applications. FEM allows entire designs to be constructed, refined, and optimized before the design is manufactured.

This powerful design tool has significantly improved both the standard of engineering designs and the methodology of the design process in many industrial applications. The introduction of FEM has substantially decreased the time to take products from concept to the production line. It is primarily through improved initial prototype designs using FEM that testing and development have been accelerated [10]. In summary, benefits of FEM include increased accuracy, enhanced design and better insight into critical design parameters, virtual prototyping, fewer hardware prototypes, a faster and less expensive design cycle, increased productivity, and increased revenue.

2.6 MATLAB®

MATLAB® is an on-line system providing machine aid for the mechanical symbolic processes encountered in analysis. It is capable of performing, automatically and symbolically, such common procedures as simplification, substitution, differentiation, polynomial factorization, indefinite integration, direct and inverse Laplace transforms, the solution of linear differential equations with constant coefficients, the solution of simultaneous linear equations, and the inversion of matrices. It also supplies fairly elaborate bookkeeping facilities appropriate to its on-line operation.

MATLAB® is a high performance language for technical computing. It integrates computation, visualization, and programming environment. Furthermore, MATLAB® is a modern programming language environment, which has sophisticated data structures, contains built-in editing and debugging tools, and supports object-oriented programming. These factors make MATLAB® an excellent tool for teaching and research.

MATLAB® has many advantages compared to conventional computer lan-

languages, example as C, FORTRAN, for solving the technical problems. MATLAB® is an interactive system whose basic data element is an array that does not require dimensioning. The software package has been commercially available since 1984 and is now considered as a standard tool at most universities and industries worldwide.

It has powerful built-in routines that enable a very wide variety of computations. It also has easy to use graphics commands that make the visualization of results immediately available. Specific applications are collected in packages referred to as toolbox. There are toolboxes for signal processing, symbolic computation, control theory, simulation, optimization, and several other fields of applied science and engineering.

2.7 Previous Study

The study of the dynamic and vibrations of mechanical systems is one of the important problems in industry. The suppression of unwanted vibrations is an important goal in many applications such as machines, tall buildings, bridges, pipelines and aircraft cabins. The application of a DVA to linear systems has been investigated by many authors, for example, Den Hartog [11], Hunt [12], and Korenev and Reznikov [13]. A significant amount of work has been devoted to search for a suitable solution to reduce the vibration level in these applications. The different concepts had been developed and employed in this research area. One of the concepts is using vibration absorber. Vibration absorber is a mechanical device, basically known mainly of mass, spring and damper, designed to have a natural frequency equal to the frequency of the unwanted vibration of the primary system [14, 15].

There have history designing of vibration absorber long ago. First vibration absorber proposed by Herman Farhm [16] in year 1909, that consists of a second mass-spring device attached to the main device, also modelled as a mass-spring system, which prevents it from vibrating at the frequency of the sinusoidal forcing acting on the main device. According to Den Hartog [11] in his book, the classical problem of damped vibration absorber that consists of a mass, spring and viscous damper attached to an undamped single degree of freedom system of which the mass is subject to harmonic forcing, has a well known solution. Esref Eskinat et al. [17] has said if damping is added to the absorber, the vibration amplitude of the main mass cannot be made zero at the forcing frequency but the sensitivity of the system to variations in the forcing frequency decreases. Also the vibration amplitude of the absorber mass decreases considerably with a damped absorber. In the literature, the term 'vibration absorber' is

used for passive devices attached to the vibrating structure.

In the research of Yuri Khazanov [18], he stated that DVA sometimes referred as tuned mass damper. In classical form, its a natural frequency is tuned to match the natural frequency of the structure it is installed on. Because of this tuning, a DVA exerts a force on the main system that equal and opposite to excitation force, cancelling vibration at the resonant frequency. In the modern applications, the goal is to assure the performance within specifications over a wide frequency range while minimizing the size of the device.

2.7.1 Dynamic Vibration Absorber as Controller

A dynamic vibration absorber (DVA) is a device consisting of an auxiliary mass–spring system which tends to neutralise the vibration of a structure to which it is attached. First patented by Harman Frahm [16] in early century is a classic technique to neutralize a harmonic disturbance. A DVA is essentially a flexible substructure having a resonant frequency close to that of the disturbance. The original DVA was lightly damped and the system might be only marginally stable as a refinement, a damped vibration absorber was introduced.

Dynamic vibration absorber (DVA) have been successfully used to attenuate the vibration of many structures. The DVA usually consists of a mass attached to the structures to be controlled through a spring mass-damper system. It is usually used to suppress a harmonic excitation at given frequency. Many researchers have considered the topic of vibration control in the last century but Ormondroyd and Den Hartog published the first mathematical treatment of passive system of vibration control 1928.

The dynamic vibration absorber was first invented in 1909 by Hermann Frahm, and since then it has been successfully used to suppress wind-induced vibration and seismic response in buildings. The characteristic of DVA were studied in depth by Den Hartog [8]. The application of a DVA has been investigated by many authors, for example Sun et al. [19] in their research in finding a simple and effective vibration control of multi-frequency harmonic excitation by the comparisons of different DVAs and frequency tuning methods. The comparisons they did is between single DVA, two DVA and state-switched absorber (SSA). In the research, they found that SSA and dual DVAs have better performance than single DVA. Another DVA usage by Sun et al. [19], where they did implemented mathematical models of floating raft system with or without DVAs by assembling the mobility matrices of the subsystem. They

investigated the vibration reduction performance of this passive DVA by numerical simulation based on vibration energy transmission and the power flow transmission characteristic of the floating raft system.

Other authors that interested in DVA research are Bonsel, Fey and Nijmeijer [6] and Wong et.al. [20]. Damped and undamped DVA have been used By Bonsel et al. [6] to suppress first resonance of piecewise linear system. They did investigate the possibility of reducing the vibrations of an archetype piecewise linear system using a linear DVA. If we compare the efficient of undamped DVA or damped DVA in vibration absorption, the undamped absorber can be use by tuning the both mass of it. For the damped absorber, we can adjust the position or distance of mass from the centre of the vibration absorber.

Aside from work mentioned above, an author named Farhad S. Samani with his research partner Francesco Pellicano [21] conduct a research on structure excited by moving loads using a single DVA. The effectiveness and performance of the DVA being compared with the classical damper. Their result prove that are capable of reducing the vibration amplitude in the presence of excitations due to moving loads.

In addition to their analysis, Hsueh et al. [22] did study an analytical and closed-form vibration transmissibility of a general unidirectional multi degree of freedom system with multiple dynamic absorbers. Analytical expressions for both force and displacement transmissibility has been derived by them and did discover that it is easy to apply the method in designing the dynamic absorber for vibration reduction of mechanical system. A study that can be considered as an earlier in this area did by Kojima [23] when he investigated the forced vibrations of a structure with a non- linear DVA.

The traditional treatment methods that involve structural modifications are often time consuming and expensive. Blocking the problem frequencies in the variable frequency drive limits the use of the system by the user. One possible solution is an installation of a dynamic vibration absorber (DVA). As we know, the DVA has certain advantages over other methods of vibration suppression. It is external to the machine structure, so no re-installation of equipment is necessary. Unlike with structural modifications, when the final effect is unknown until mass-elastic properties of the machine components have been modified, a DVA can be designed and tested before installation. It can be adjusted in the lab environment with predictable field results.

CHAPTER 3

METHODOLOGY

This chapter discusses about the flow and guideline to conduct this research study and the the step to design and develop the passive vibration absorber by using MATLAB® and ANSYS®. In this study area, the right sequence and procedure is crucially important in order to lead the project run smoothly and produce accurate result. Generally, the research study was conducted based on the designed research methodology flow chart as shown in Figure 3.1.

This method plays an important role to implement the research study in according to the designed plan. Initially, theoretical equation of simply supported plate will be derived and calculated based on existing plate's theory, which later will plotted in MATLAB®. This is followed by derivation of simply supported plate with attached vibration absorber system. Later, a simulation finite element analysis will be carried out using ANSYS® to validate the derived theoretical model. Once the finite element model is validated, further research will be carried out to determine the effect of mass, spring and damper parameters of vibration absorber, including the effect of its location. The end stage of study will investigate the effectiveness of adding multiple vibration absorbers to reduce the vibration level of a simply supported plate, and this result will compare with single absorber attachment.

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