VIBRATION SIMULATION OF A SIMPLY-SUPPORTED BEAM WITH ATTACHED MULTIPLE ABSORBERS

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ABSTRACT

In general, the standard solutions to reduce the vibration and noise problems are to redesign and modify the system such as adding the thickness of wall panels, enhancing the elasticity of the structure and increase the damping mechanism of the structure. In this study, the concept of dynamic vibration absorber is used on a beam structure to reduce the vibration or amplitude. The methods employed in this study were analytical equations and finite element analysis. MATLAB® was used to transform analytical equations into graphs and at the same time to verify the finite element simulation of ANSYS®. Result shows that analytical equation and finite element simulation of a simply supported beam produce a similar outcome, which indicates a good agreement. The frequency range was studied between 5 Hz to 1000 Hz and there are four modes resulting shape. Further study was conducted by placing the absorber at different locations configuration. This was followed by adding a single absorber and multiple absorbers to see the average percentage reduction in vibration's amplitude. The overall reduction achieved for multiple absorbers is 88.6% compared to a single absorber which only achieved 8.21% reduction. Finally, it can be concluded that multiple vibration absorber can be reduce the global vibration of structure compared to a single vibration absorber. However, for the structures that concern about weight, adding more dynamic vibration absorber needs to be considered properly since excess weight will result in less fuel efficiency of vehicles, aerospace, automotive and machine systems.

ABSTRAK

Pada umumnya, penyelesaian bagi piawaian untuk memperbaiki masalah getaran dan kebisingan adalah dengan merekabentuk semula dan 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 rasuk, 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 rasuk yang disokong mudah adalah sama. Julat frekuensi adalah 5 Hz hingga 1000 Hz dan terdapat empat bentuk mod yang terhasil. 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 penyerap berganda adalah 88.6%% berbanding dengan penyerap tunggal yang hanya 8.21% 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.

CONTENTS

LIST OF FIGURES

LIST OF TABLES

LIST OF APPENDICES

CHAPTER 1

INTRODUCTION

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 vibration absorbers attached to a flexible beam 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

When a system begins to operate, the structure or machine experience vibration may exist and occur because of the dynamic effects on manufacturing, tolerance, rotation, friction or impact on the machine parts. Critical problems occur is increased number of dynamic systems, such as the mechanical equipment, machinery, bridges, vehicles and aircraft.

In this study, the problem of vibration of the beam structure will be reviewed by put vibration absorber configurations on different locations. This followed, using some of the shock absorber to reduce overall vibration in a simply supported beam structure.

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 beam without and with attached multiple absorbers. Based on the research, there are several objectives that need to achieve.

i. To determine the vibration reduction of a single vibration absorber attach to a beam

ii. To investigate the effect of changing the location (absorber), mass and damping on the absorber performance.

iii. To determine the effect of attaching multiple vibration absorbers to reduce vibration level of a simply supported beam.

iv. To provide guidelines for optimum numbers of passive vibration absorbers used, mass, stiffness, 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. Two mathematical models regarding simply–supported beam and when attached absorber will be derived.

iii. The dimension of a beam to be studied 1 x 0.2 x 0.02 m.

iv. The number of absorbers to be studied will be limited to 10.

v. Literature search will be carried out on finite element analysis (Ansys), analytical analysis (Matlab), vibration, a simply supported beam, vibration absorbers, natural frequency, damping and mode shape.

1.5 Expected Outcomes

There are several contributions to the body of knowledge presented in this research are: (1) 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, (2) provide guidelines for optimum numbers of passive vibration absorbers used and its placement in order to have substantial vibration attenuation.

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.

CHAPTER 2

LITERATURE REVIEW

This chapter explains about vibration theory, vibration control, Finite Element Analysis (FEA), MATLAB,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 Classification of Vibration

Vibrations can be classified into three categories: free, forced, and self-excited. Free vibration of a system is vibration that occurs in the absence of external force. An external force that acts on the system causes forced vibrations. In this case, the exciting force continuously supplies energy to the system. Forced vibrations may be either deterministic or random (see Figure 2.1 and 2.2). Self-excited vibrations are periodic and deterministic oscillation. Under certain conditions, the equilibrium state in such a vibration system becomes unstable, and any disturbance causes the perturbations to grow until some effect limits any further growth. In contrast to forced vibrations, the exciting force is independent of the vibrations and can still persist even when the system is prevented from vibrating [5].

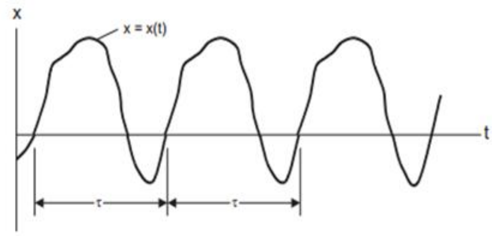

Figure 2.1: A deterministic (periodic) excitation

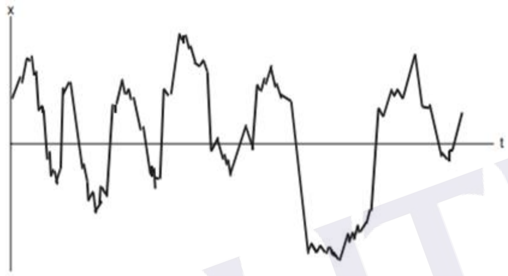

Figure 2.2: Random excitation

2.1.2 Elementary Parts of Vibrating System

In general, a vibrating system consists of a spring (a means for storing potential energy), a mass or inertia (a means for storing kinetic energy), and a damper (a means by which energy is gradually lost) as shown in Figure 2.3. An undamped vibrating system involves the transfer of its potential energy to kinetic energy and kinetic energy to potential energy, alternatively. In a damped vibrating system, some energy is dissipated in each cycle of vibration and should be replaced by an external source if a steady state of vibration is to be maintained [5].

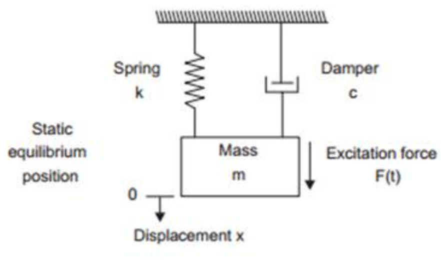

Figure 2.3: Elementary parts of vibrating systems

2.1.3 Periodic Motion

When the motion is repeated in equal intervals of time, it is known as periodic motion. Simple harmonic motion is the simplest form of periodic motion. If $x(t)$ represents the displacement of a mass in a vibratory system, the motion can be expressed by the equation

$$x = Acos\omega t = Acos2\pi\frac{t}{\tau} \tag{2.1}$$

where A is the amplitude of oscillation measured from the equilibrium of the mass. The repetition time, τ is called the period of the oscillation, and its reciprocal, $f = \frac{1}{\tau}$ is called the frequency. Any periodic motion satisfies the relationship

$$x(t) = x(t+\tau) \tag{2.2}$$

That is Period,

$$\tau = \frac{2\pi}{\omega} s/cycle \tag{2.3}$$

Frequency

$$f = \frac{1}{\tau} = \frac{\omega}{2\pi} cycles/s, or.Hz \tag{2.4}$$

ω is called the circular frequency measured in rad/sec.

The velocity and acceleration of a harmonic displacement are also harmonics of the same frequency, but lead the displacement by $\frac{\pi}{2}$ and π radians, respectively. When the acceleration Ẍ of a particle with rectilinear motion is always proportional to its displacement from a fixed point on the path and is directed towards the fixed point, the particle is said to have simple harmonic motion [5].

The motion of many vibrating systems in general is not harmonics. In many cases the vibrations are periodic as in the impact force generated by a forging hammer. If $x(t)$ is a periodic function with period, τ its Fourier series representation is given by

$$x(t) = \frac{a_0}{2} + \sum_{n=1}^{\infty}(a_n cosn\omega t + b_n sinn\omega t) \tag{2.5}$$

Where $\omega = \frac{2\pi}{\tau}$ is the fundamental frequency and a_0, a_1, a_2,..., b_1, b_2,... are constant coefficients, which are given by:

$$a_0 = \frac{2}{\tau}\int_0^{\tau} x(t)dt \tag{2.6}$$

$$a_n = \frac{2}{\tau}\int_0^{\tau} x(t)cos.n\omega tdt \tag{2.7}$$

$$b_n = \frac{2}{\tau}\int_0^{\tau} x(t)sin.n\omega tdt \tag{2.8}$$

The exponential form of $x(t)$ is given by:

$$x(t) = \sum_{n=-\infty}^{\infty} c_n e^{in\omega t} \tag{2.9}$$

The Fourier coefficients c_n can be determined, using

$$c_n = \frac{1}{\tau}\int_0^{\tau}(x)te^{-in\omega t}dt \tag{2.10}$$

The harmonic functions a_n cos $n\omega t$ or $b_n$sin $n\omega t$ are known as the harmonics of order n of the periodic function $x(t)$. The harmonic of order n has a period $\frac{\tau}{n}$. These harmonics can be plotted as vertical lines in a diagram of amplitude (a_n and b_n) versus frequency ($n\omega$) and is called frequency spectrum.

2.1.4 Discrete and Continuous System

Most of the mechanical and structural systems can be described using a finite number of degrees of freedom. However, there are some systems, especially those include continuous elastic members have an infinite number of degree of freedom. Most mechanical and structural systems have elastic (deformable) elements or components as members and hence have an infinite number of degrees of freedom. Systems which have a finite number of degrees of freedom are known as discrete or lumped parameter systems, and those systems with an infinite number of degrees of freedom are called continuous or distributed systems [5].

2.1.5 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, $\omega = v.\lambda$. 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, as shown in Eq.(2.11)

$$\omega_{d=\sqrt{1-\xi^2.\omega}} \tag{2.11}$$

2.1.6 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 [6].

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

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

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

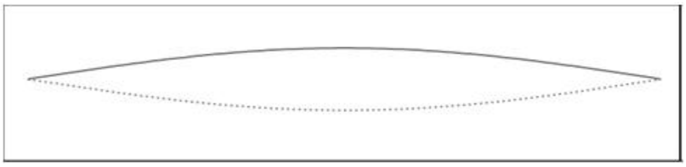

Figure 2.4: The motion in the centre of the string

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

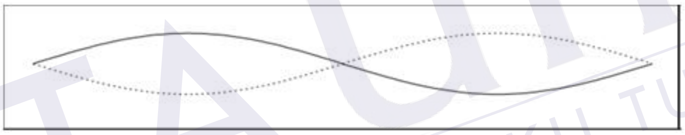

Figure 2.5: The two halves of the string vibrate in opposition each other

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

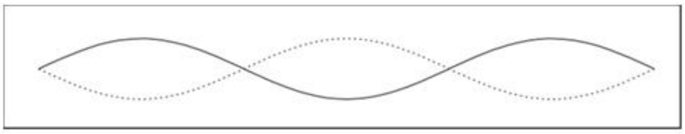

Figure 2.6: The string is divided into three equal length sections

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

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

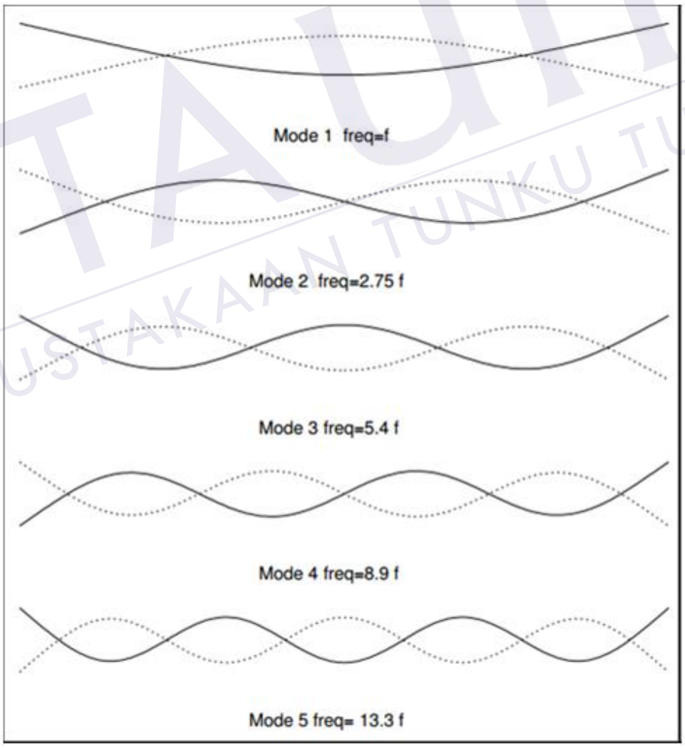

Figure 2.7: The string mode with difference frequencies

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

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

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

2.2 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 2.8 below. 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.

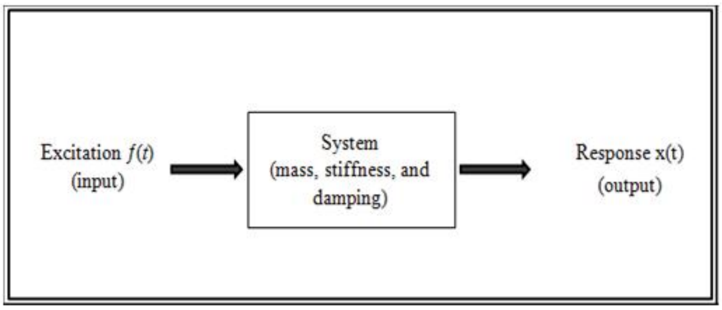

Figure 2.8: Input-output relationship of a vibratory system

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 Dynamic Vibration Analysis (DVA)

The basic physical principle of the dynamic vibration absorber is that of attaching to a vibrating structure a resonance system which counteracts the original vibrations. Ideally such a system would completely eliminate the vibration of the structure, by its own vibrations. Figure 2.9 illustrates these ideas. The mass, M, is here assumed to be the mass of a (rigid) machine structure producing the vibrating force, $P_0 sin(2\pi ft)$ [7].

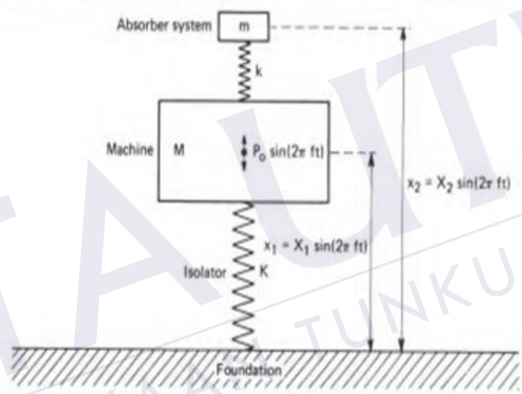

Figure 2.9: Illustration of the principle of the dynamic vibration absorber

The machine is mounted on a vibration isolator with are stiffness, K. Attached to the machine is a resonance (dynamic absorber) system consisting of the mass, m, and the spring element, k. It is now a simple matter to write down the equations of motion for the complete system:

$$M\frac{d^2x_1}{dt^2} + Kx_1 - k\left(x_2 - x_1\right) = P_0 sin\left(2\pi ft\right) \tag{2.12}$$

$$m\frac{d^2x_2}{dt^2} + k\left(x_2 - x_1\right) = 0 \tag{2.13}$$

Assuming that the stationary solutions to these equations can be written (where X_1 and X_2can be either positive or negative)

$$x_1 = X_1 sin(2\pi f t) \tag{2.14}$$

and

$$x_2 = X_2 sin(2\pi f t) \tag{2.15}$$

then

$$\left(1 + \frac{k}{K} - M\frac{(2\pi f)^2}{K}\right) X_1 - \frac{k}{K} X_2 = \frac{P_0}{K} \tag{2.16}$$

and

$$X_1 = \left[1 - (\frac{f}{f_a})^2 X_2\right] \tag{2.17}$$

where $f_a = \frac{1}{2\pi}\sqrt{\frac{k}{m}}$= resonant frequency of the attached (absorber) system by setting $1-(\frac{f}{f_a})^2 = 0$ i.e $f_a = f$ motion, X_1 , of the machine will be zero, i.e. the machine will not vibrate at all. The maximum amplitude of the mass, m, is in this case:

$$-\frac{k}{K} X_2 = \frac{P_0}{K} i.e. X_2 = -\frac{P_0}{K} \tag{2.18}$$

This again means that by tuning the absorber system resonant frequency to equal the "disturbing” frequency, the vibration of the machine can be eliminated.

Actually, in practical cases the "disturbing" frequency region often covers the resonant frequency of the machine-isolator system, and both the absorber and the isolation system contain some mechanical damping. The equations of motion for the complete system then become considerably more complex, and so do their solutions.

Figures 2.10, 2.11 and 2.12 illustrate the effects upon the vibration transmissibility of a machine/isolator system when the machine is supplied with a dynamic vibration absorber.

From Figure 2.10 it is seen that when the complete system contains no damping at all and the absorber system is tuned to the resonant frequency of the machine/isolator system the transmissibility at this frequency is zero, in conformity with the above statements and mathematical derivations. However, on both "sides" of the resonant frequency two, theoretically infinitely high, transmissibility "peaks" are found. The shape of the curve is caused by the dynamic coupling between the machine/isolator system and the absorber system. Coupling effects of this sort are quite common in many branches of physics.

If the absorber damping is infinite, the absorber mass is virtually clamped to the machine and the absorber system does not function at all. Figure 2.11. In practice,

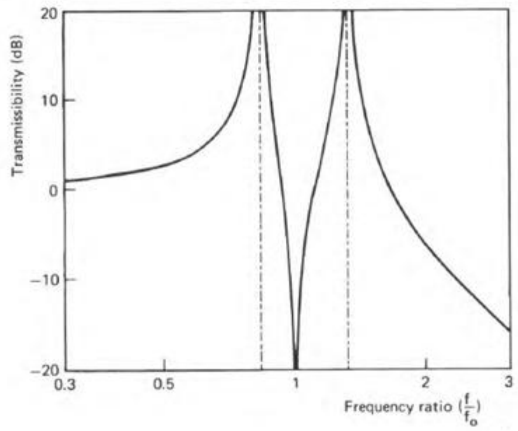

Figure 2.10: Theoretical transmissibility curves for a vibration isolated system supplied with an undamped dynamic vibration absorber

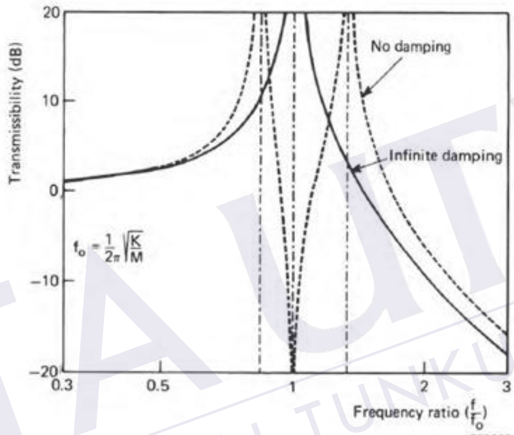

Figure 2.11: Effect of extreme absorber damping upon the transmissibility ratio of an undamped machine/isolator system

when a damped vibration absorber is applied to a machine/isolator system the transmissibility curve must lie between the two extremes sketched in Figure 2.11. This is illustrated in Figure 2.12 for various values of absorber damping ratio.

Theory has shown that when damping is added to the absorber the "optimum" performance conditions*) are, in general, no longer obtained by tuning the resonant frequency of the absorber system to equal the resonant frequency of the machine/isolator system. Actually the most favourable tuning depends upon the ratio between the absorber mass and the mass of the machine i.e. m /M. It has been found that when the damping is of the viscous type then the ratio between the absorber resonant frequency, f_a, and the machine/ isolator resonant frequency, f_0, should be:

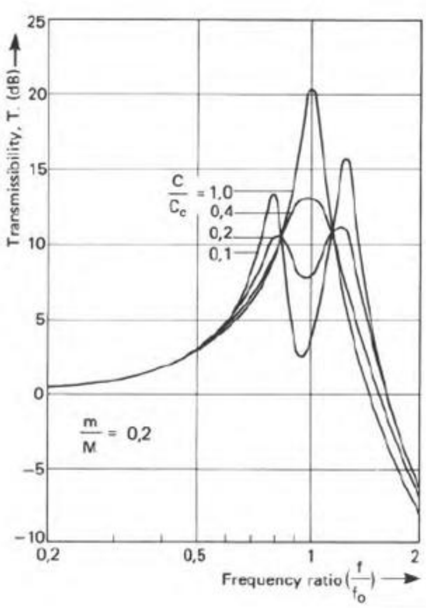

Figure 2.12: Transmissibility of a machine/isolator system when the machine is supplied with a damped vibration absorber. The degree of damping is indicated on the curves

$$\frac{f_a}{f_0} = \sqrt{\frac{M}{m+M}} = \sqrt{\frac{1}{1+\frac{m}{M}}} \tag{2.19}$$

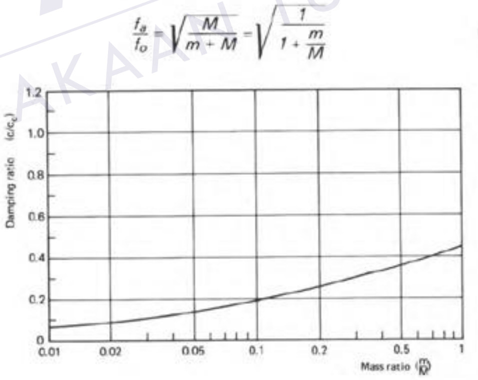

Figure 2.13: Curve showing "optimum" viscous damping factor as a function of the mass ratio

From this equation it is noted that when m/M is small the difference between the two resonant frequencies is negligible, while for an increasing mass-ratio the "detuning" of the absorber may become very significant. Also the "optimum" viscous damping factor depends upon the mass-ratio, see Figure 2.13. Finally, Figure 2.14 shows some theoretical transmissibility curves calculated for various mass-ratios and "optimum damping. Note the decrease in resonant amplification with increasing mass-ratios.

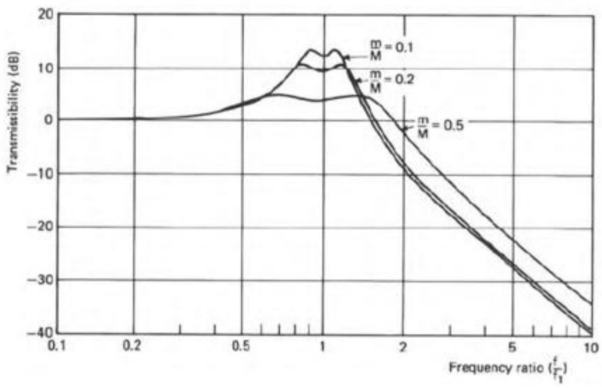

Figure 2.14: Theoretical transmissibility curves for a system of the type shown in Figure 2.9 supplied with a viscously damped dynamic vibration absorber. Optimum absorber tuning and damping for mass ratios of $\frac{m}{M} = 0.1$, $\frac{m}{M} = 0.2$, $\frac{m}{M} = 0.5$

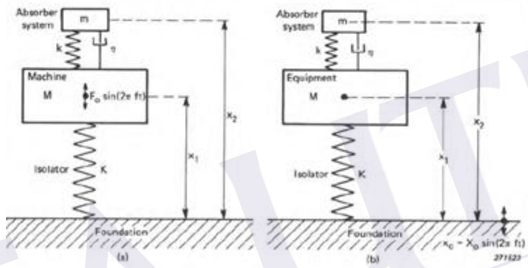

Figure 2.15: Dynamic vibration absorber applied to: a) Machine (source), b) Equipment

The theoretical treatment of the vibration transmissibility from a vibrating source (machine) to its foundation, and that of the vibration transmissibility from a vibrating foundation to mounted equipment is more or less identical. This, of course, also applies with respect to the use of dynamic vibration absorbers see, Figure 2.15.

2.4 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 sub domains, 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, defence, 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. Nonlinear 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 thermo-properties in the material that yield linear heat diffusion [9][10].

2.5 MATLAB

MATLAB (short for MATrix LABoratory) is a special-purpose computer program optimized to perform engineering and scientific calculations. It started life as a program designed to perform matrix mathematics, but over the years it has grown into a flexible computing system capable of solving essentially any technical problem.

The MATLAB program implements the MATLAB programming language and provides a very extensive library of predefined functions to make technical programming tasks easier and more efficient. MATLAB is a huge program with an incredibly rich variety of functions. Even the basic version of MATLAB without any toolkits is much richer than other technical programming languages. There are more than 1000 functions in the basic MATLAB product alone, and the toolkits extend this capability with many more functions in various specialties [11].

2.5.1 The Advantages of MATLAB

MATLAB has many advantages compared with conventional computer languages for technical problem solving. These include

1. Ease of Use

MATLAB is an interpreted language, like many versions of Basic. Like Basic, it is very easy to use. The program can be used as a scratch pad to evaluate expressions typed at the command line, or it can be used to execute large prewritten programs. Programs may be easily written and modified with the built-in integrated development environment, and debugged with the MATLAB debugger. Because the language is so easy to use, it is ideal for the rapid prototyping of new programs. Many program development tools are provided to make the program easy to use. They include an integrated editor/debugger, on-line documentation and manuals, a workspace browser, and extensive demos [11].

2. Platform Independence

MATLAB is supported on many different computer systems, providing a large measure of platform independence. Programs written on any platform will run on all of the other platforms, and data files written on any platform may be read transparently on any other platform. As a result, programs written in MATLAB can migrate to new platforms when the needs of the user change [11].

3. Predefined Functions

MATLAB comes complete with an extensive library of predefined functions that provide tested and pre-packaged solutions to many basic technical tasks. For example, suppose that you are writing a program that must calculate the statistics associated with an input data set. In most languages, you would need to write your own subroutines or functions to implement calculations such as the arithmetic mean, standard deviation, median, and so on. These and hundreds of other functions are built right into the MATLAB language, making your job much easier.

In addition to the large library of functions built into the basic MATLAB language, there are many special-purpose toolboxes available to help solve complex problems in specific areas. For example, a user can buy standard toolboxes to solve problems in signal processing, control systems, communications, image processing, and neural networks, among many others. There is also an extensive collection of free user-contributed MATLAB programs that are shared through the MATLAB Web site [11].

4. Device-Independent Plotting

Unlike most other computer languages, MATLAB has many integral plotting and imaging commands. The plots and images can be displayed on any graphical output device supported by the computer on which MATLAB is running. This capability makes MATLAB an outstanding tool for visualizing technical data.

5. Graphical User Interface

MATLAB includes tools that allow a programmer to interactively construct a Graphical User Interface (GUI) for his or her program. With this capability, the programmer can design sophisticated data-analysis programs that can be operated by relatively inexperienced users [11].

6. MATLAB Compiler

MATLAB's flexibility and platform independence is achieved by compiling MATLAB programs into a device-independent p-code and then interpreting the p-code instructions at runtime. This approach is similar to that used by Microsoft's Visual Basic language. Unfortunately, the resulting programs can sometimes execute slowly because the MATLAB code is interpreted rather than compiled. We will point out features that tend to slow program execution when we encounter them.

A separate MATLAB compiler is available. This compiler can compile a MATLAB program into a true executable that runs faster than the interpreted code. It is a great way to convert a prototype MATLAB program into an executable suitable for sale and distribution to users [11].

2.6 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 [12], Hunt [13], and Korenev and Reznikov [14]. 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[15][16]. There have history designing of vibration absorber long ago. First vibration absorber proposed by Herman Farhm [17] in year 1909, that consists of a second mass-spring device attached to the main device, also modelled as a as mass-spring system, which prevents it from vibrating at the frequency of the sinusoidal forcing acting on the main device. According to Den Hartog [12] 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. EsrefEskinat et.al. [18] 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 [19], he stated that DVA sometimes referred as tuned mass damper. In classical form, it's 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.

CHAPTER 3

METHODOLOGY

The research methodology is important for the scope of work to perform this study correctly. All steps of the research work carried out will be described in this chapter. The methodology of this study consists of the specifications of the beam, the beam supported and vibration absorber and the use of ANSYS software and MATLAB for simulation studies. The main part of this study is to identify the vibration of the structure and the location to reduce vibration using a vibration absorber. Such as the design of a vibration absorber that reduces the vibrations of the beam. Before starting work simulation, first set the specification, material, weight and size of the beam. Type of the beam is simply supported beam studied. Next, determine the location of the vibration absorber on any part of the beam and increase the number of vibration absorber if necessary to reduce the vibration of the beam. Then, using the software of ANSYS and MATLAB to find the characteristics of vibration and the effects and also double shock absorber and absorber on the beam.

3.1 Research Flow Chart

Figure 3.1 illustrated the methodology flow chart of research study that will be carried out throughout a year.

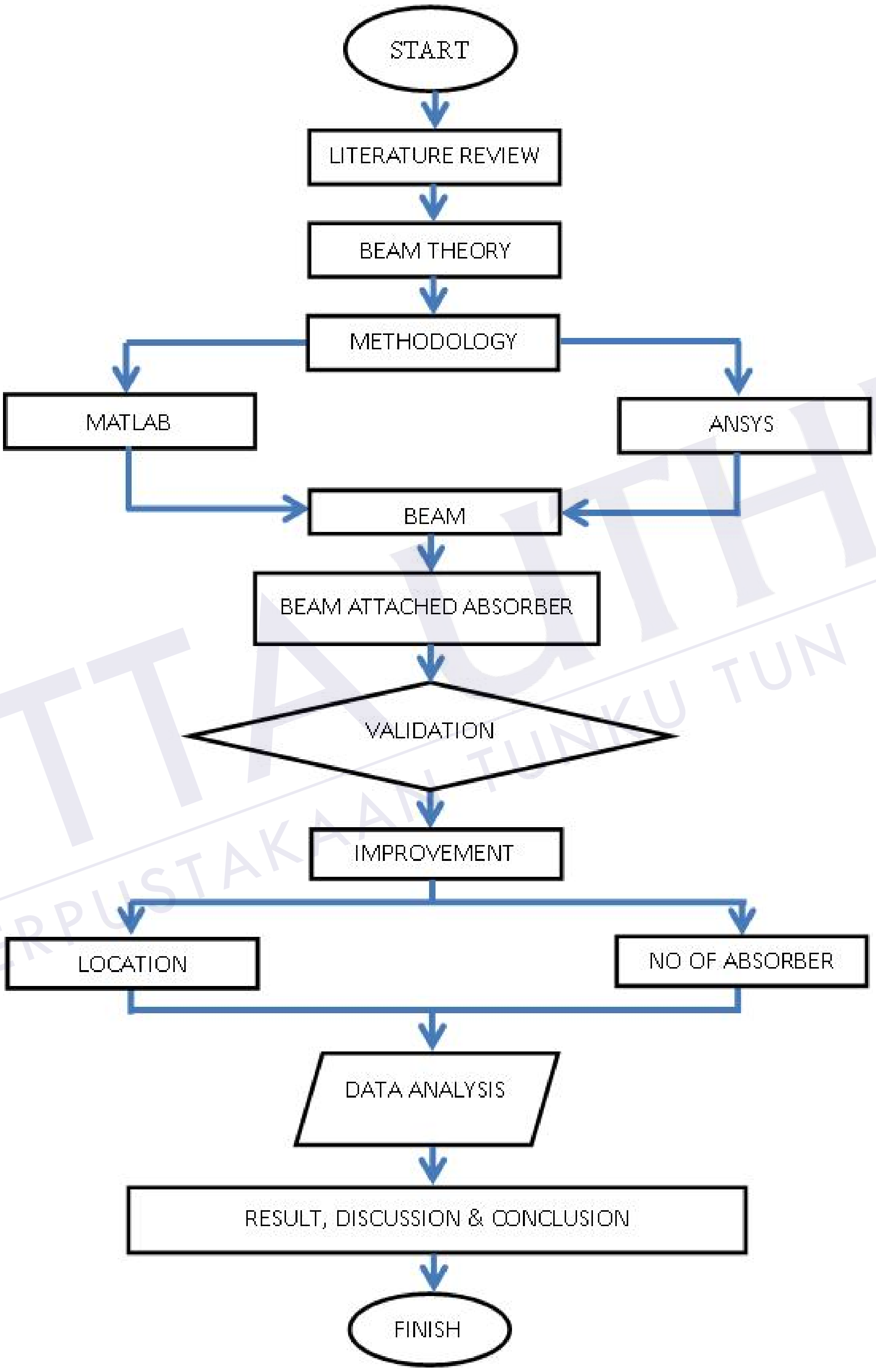

Figure 3.1: Research flow chart

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