SUPERVISOR DECLARATION

"I hereby declare that I have read this thesis and in my opinion this report is sufficient in terms of scope and quality for the award of the degree of Bachelor of Mechanical engineering (Thermal-Fluids)"

Signature: ....

Supervisor: Dr. Azma Putra

Date: 21/6/2013

DESIGN OF A DYNAMIC VIBRATION ABSORBER

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DECLARATION

"I hereby declare that the work in this report s my own except for summaries and quotations which have been duly acknowledged."

Signature: ....

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ABSTRACT

Dynamic vibration absorbers have been widely used in all sorts of application especially in engineering field to attenuate vibration. The present work is to model the amplitude of a vibrating beam with a dynamic auxiliary beam absorber to reduce the vibration. Mathematical model is presented here to simulate the effect of the dynamic absorber parameters such as the damping ratio, mass ratio and layer stiffness on the vibration response. Experiment with double steel beams was conducted where the elastic layer used the rubber eraser and the helical spring. The experiment shows the same trend as the finding in the theory where the auxiliary beam suppresses the first mode of the main beam.

ABSTRAK

"Penyerap Getaran Dinamik" digunakan secara meluas untuk mengurangkan getaran terutamanya dalam bidang kejuruteraan. Projek ini adalah untuk model amplitud dari struktur bergetar dengan menggunakan rasuk penyerap dinamik untuk mengurangkan getaran. Model matematik telah diwujudkan untuk mensimulasikan kesan dari parameter penyerap gentaran dinamik, sebagai contoh kesan nisbah redaman, nisbah berat dan lapisan kekenyalan terhadap getaran. Eksperimen telah dijalankan terhadap kedua-dua rasuk keluli dengan menggunakan pemadam getah dan spring heliks sebagai lapisan anjal. Walau bagaimanapun, keputusan daripada eksperimen adalah dalam trend yang sama dengan keputusan yang diperoleh daripada teori, di mana amplitud daripada rasuk utama telah disekat oleh rasuk tambahan.

TABLE OF CONTENT

LIST OF FIGURE

LIST OF SYMBOLS

DVA	=	Dynamic vibration absorber
STVA	=	Semi tuned vibration absorber
PSM	=	Final year project
M	=	Mass of mild steel beam (kg)
k_1	=	Stiffness of mild steel beam (N/m)
c_1	=	Damping factor of mild steel beam (Ns/m)
w_1	=	Vertical displacement of mild steel beam (m)
F	=	Excitation force (N)
q	=	Generalized time function of amplitude (m)
x	=	Position coordinate (m)
σ_n	=	Eigenvalue of the mode shape function
X	=	Mode shape function
L	=	Length of mild steel beam (m)
Q	=	Complex displacement amplitude (m)
ω	=	Radial frequency (rad/s)
E	=	Modulus of elasticity of mild steel beam (N/m^2)
I	=	Area moment of inertia of mild steel beam (m^4)
m_a	=	Mass of dynamic vibration absorber (kg)
c_2	=	Damping factor of dynamic vibration absorber (Ns/m)
k	=	Stiffness of spring (N/m)
w_2	=	Vertical displacement of dynamic vibration absorber (m)
t	=	Time (s)
ζ	=	Damping ratio
n	=	Mode of vibration

LIST OF APPENDICES

CHAPTER 1

INTRODUCTION

1.1 OVERVIEW

Beams are the elements that have been used widely in engineering field especially in machines, buildings and bridges (Stanisław Kukla, 2005). There are many types of beam design such as I-beam, flitch beam and cantilever beam. I-beam is the most common beam that straight in shape. The various support patterns of I-beam such as L, V, H and W shapes has enabled I-beam to be widely used in floors, roofs and walls support (Cyprus, 2012). Flitch beam consists of multi layers of wood and steel. Therefore, the flitch beam is much more lightly compared to pure steel beam. Flitch beam is widely used as vertical extra support in wood structure. Cantilever beam is known as one end supported beam which load is distributed on the entire beam. The load distribution on the beam has enabled it to support balconies and bridges safely.

Most of the structure beam has the presence of vibration. There are several natural frequencies that occurred in every structure beam. When there are any forces excited at any of these natural frequencies the structure beam will react particularly strongly and yield to large amplitude (Dennis G. Zill, 2007). The vibration existed in structure beam might affect the performance and damaged on the structure beam. Thus the lifetime of the structure beam might be decreases due to the vibration. However, dynamic vibration absorber (DVA) can be used to eliminate the structure beam from vibrating.

Dynamic vibration absorber is a simple spring-mass system that is widely used in attenuating vibration of vibrating system in mechanical field. The vibration of a vibrating system can be controlled in few ways such as passive control, semi active control and active control. Passive control reduces the vibration by redesign the structure of passive dynamic vibration absorber including springs and damper. While active control can be retuned via software to reduce the vibration (BARBARA TISEO, 2010). The first passive dynamic absorber (DVA) was introduced by Hermann Frahm. The passive dynamic absorber is only effective at the narrow range frequency that near to the resonance frequency of the primary system (Yan-Ying Zhao, 2011). Semi active dynamic vibration absorber is not suitable to be applied in vibration suppression due to its instability. However, semi active dynamic vibration absorber is effective over wider frequency range (Yan-Ying Zhao, 2011). An active vibration absorber is designed to reduce the vibration of the primary system to zero (T. Mizuno, 1995).

Dynamic vibration absorber (DVA) has been applied in most of structure and vehicles to reduce the vibration of the system. The incorporate ability into the structure after design stage is one of the advantages of the DVA. The incorporate ability of DVA enabled it to be mounted in the structure without affecting the structure and supporting foundation. Besides of incorporate ability, the other advantages of DVA are low cost, stable, and simplicity of implementation (Ting-Kong, 1999).

1.2 PROBLEM STATEMENT

Beam shape structure is widely used in constructing building. Every beam shape structure has its own natural frequencies which controls its dynamic behavior. Resonance occurred when the natural frequencies of the vibrating structure coincides with the excitation frequency of the external dynamic force. At the resonance frequency, the vibrating beam structure vibrates at maximum amplitude. This phenomenon happened due to the excessive of permissible limit of mass and elasticity in the vibrating beam structure (Rao, 2009). Since the vibration of the beam will propagate the vibration to the neighboring structures as well as affects the beam's life time. Thus a vibration absorber must be installed to reduce the vibration of the beam.

1.3 OBJECTIVE

This study embarks with the following objective:

1. To model the amplitude of the vibrating beam.
2. To design a dynamic vibration absorber to reduce vibration of the beam.

1.4 SCOPE

1. Development of mathematical model to represent the vibration behavior of beam with vibration absorber.
2. Fabrication of a simple structure with vibration absorber and experimental validation.

CHAPTER 2

LITERATURE REVIEW

2.1 FIXED POINT THEORY VIBRATION

Vibration neutralizer is the device that attenuated the vibration by tuned the device natural frequency to the excitation frequency of the primary system (Michele Zilletti, 2012).Fixed point theory was used in vibration control of a continuous structure. The kinetic energy of the continuous structure is dominated in narrow band near to the natural frequency. At the narrow band vicinity of natural frequency, the kinetic energy is reduced with respect to the increased in neutralizer's mass ratio. With the presence of high neutralizer's mass ratio, the kinetic energy resonance of structure is reduced and it is more effective at lower frequency region. The neutralizer has lowest kinetic energy at the location with higher modal amplitude. However, the kinetic energy increases as the neutralizer is approaching nodal point (Dayou, 2006).

According to Figure 2.1, it is well shown that the kinetic energy is lower as the neutralizer's mass ratio is higher. Although the kinetic energy is affected by the damping ratio of neutralizer, however the effect is much lesser compared to effect from mass ratio of neutralizer.

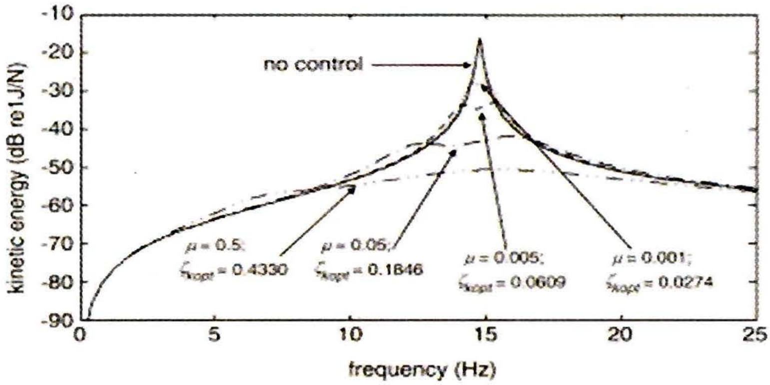

Figure 2.1: Graph of Kinetic Energy against Frequency (Dayou, 2006).

2.2 ABSORBERS ON CONTINUOUS STRUCTURE

Beam is one of the continuous structure types. Figure 2.2 shows that there are three mode shape occurred in a simply supported beam. For beam 1, three vibration absorber are placed at each antinodes respectively. While for beam 2, all the three vibration absorbers are placed together at the middle antinode of the beam.

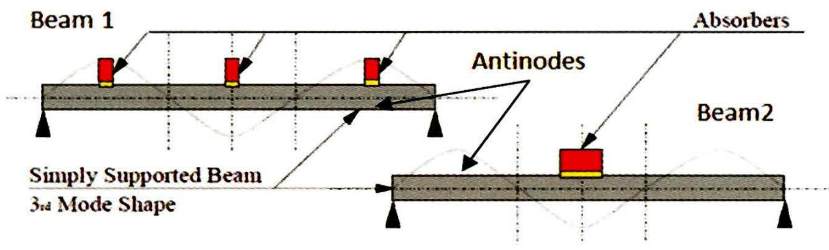

Figure 2.2: Mode Shape of Simply Supported Beam (Cambau, 1998).

According to Figure 2.3, it is clearly shown that with placing 3 vibration absorbers at each antinodes reduced maximum amplitude of the simply supported beam. However, the vibration absorber only able to reduced the amplitude on the antinode.

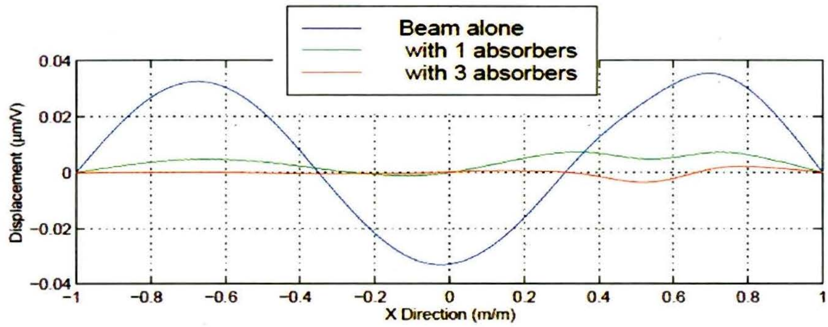

Figure 2.3: Graph of Displacement against X Direction (Cambau, 1998).

2.3 DYNAMIC PERFORMANCE ANALYSIS OF STVA

Semi tuned vibration absorber used when the parameter of a vibrating system is changing over time. There are some control algorithms that used to control the damping of vibration absorber such as by magnetorheological fluid damper and electrorheological fluid damper (La, 2012). Magnetorheologoical fluid damper as the controllable damper is used to control damping of the semi active dynamic vibration absorber. Both the peak transmissibility decreases when the damping ratio of STVA is tuned to a higher value. This phenomenon indicates that the STVA and the structure mass decoupled at the tuned frequency. However, the damping ratio of the semi active dynamic vibration absorber should be tuned to an optimum value to offer the maximum performance gains (Jeong-Hoi Kooa, 2008).

Figure 2.4 shows that the transmissibility curves become flatter as the damping ratio of the STVA increases. Besides that, the frequency ranges become larger as the damping ratio of STVA increases.

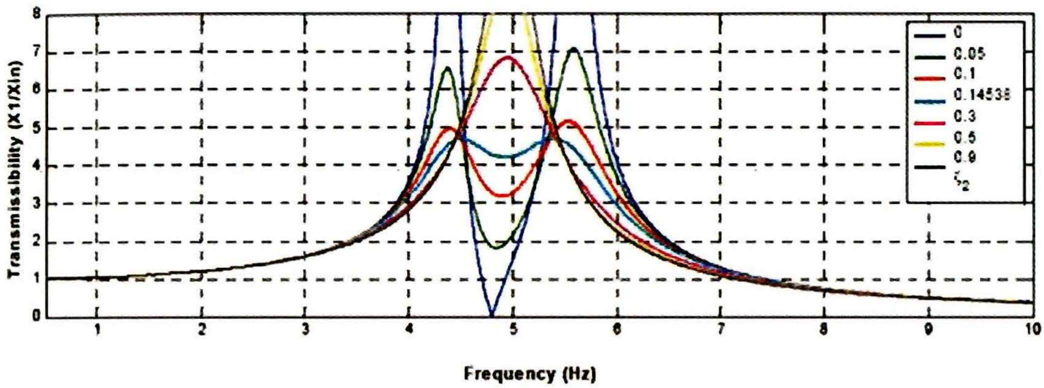

Figure 2.4: Graph of Transmissibility against Frequency (Jeong-Hoi Kooa, 2008).

2.4 APPLICATION OF DVA IN ELECTRIC GRASS TRIMMER

Electric grass trimmer has high vibration level due to the unbalanced rotationally. The dynamic behavior of the hand-arm can be changed by attaching a tunable vibration absorber to the shaft of electric grass trimmer. The hand-arm vibration can be reduced by installing the tunable vibration absorber at the optimum location (Ko Ying Hao, 2011). However, the optimum location of tunable vibration absorber is determined by imposing nodal technique. Besides that, the acceleration level of the hand-arm can be reduced by attaching the tunable vibration absorber at optimum point (Ko Ying Hao Z. M., 2012).

Figure 2.5 shows that at the same operating frequency, the amplitude of hand-arm varies with location. The maximum reduced amplitude occurred at location of 0.025L.

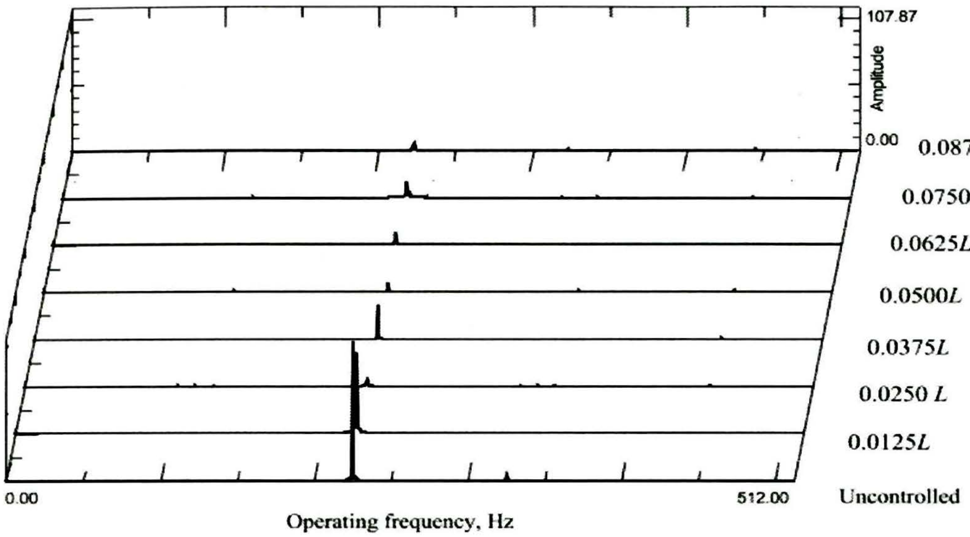

Figure 2.5: Graph of Amplitude and Location versus Operating Frequency (Ko Ying Hao L. X., 2011).

2.5 PIECEWISE BEAM LINEAR SYSTEM

A damped and undamped dynamic vibration absorber is attached separately to the piecewise beam linear system to reduce the vibration amplitude. Based on Figure 2.6 the piecewise beam linear system consists of both ends pinned and supported by at spring at the middle of beam.

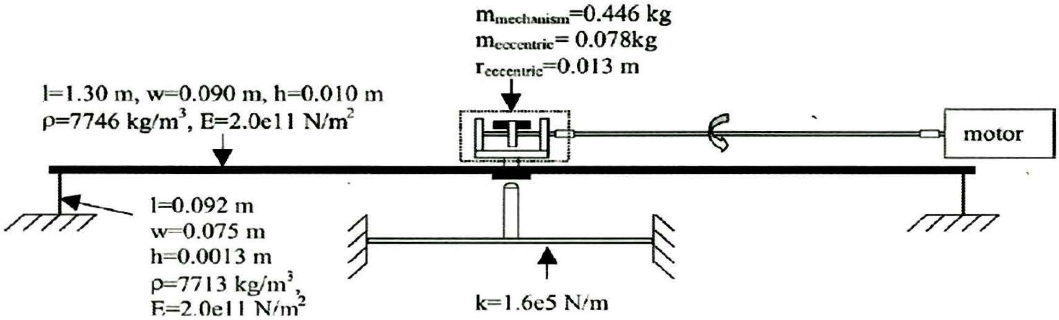

Figure 2.6: Schematic Diagram of Piecewise Linear Beam System (J. H. BONSEL, 2004).

The solid line in Figure 2.7 shows the amplitude ratio of the beam when undamped dynamic vibration absorber is used to reduce the amplitude of the piecewise beam. By using undamped dynamic vibration absorber, the amplitude ratio of the piecewise beam is reduced to zero at resonant frequency. Thus, it can be conclude that undamped vibration absorber perform most effective at resonant frequency. However, the frequency band which over the reduced amplitude of the system remains constant.

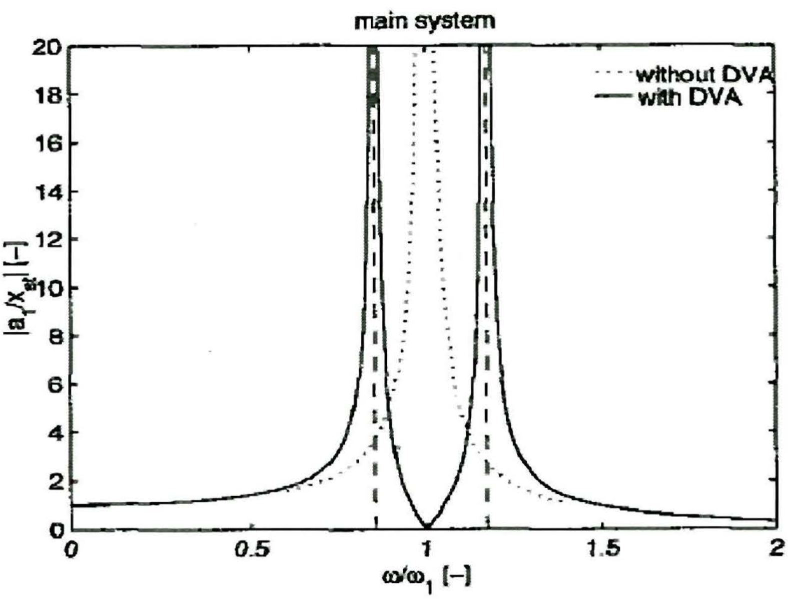

Figure 2.7: Graph of Amplitude Ratio against Frequency Ratio (Undamped DVA) (Bonsel, 2003).

The solid line in Figure 2.8 indicates piecewise beam linear system with damped dynamic vibration absorber. The amplitude ratio of the piecewise beam does not decreased to zero with the present of damped dynamic vibration absorber. However, the frequency band over which the reduced amplitude ratio is increased. Therefore, damped dynamic vibration absorber is suitable to be used to suppress the amplitude over larger frequency band.

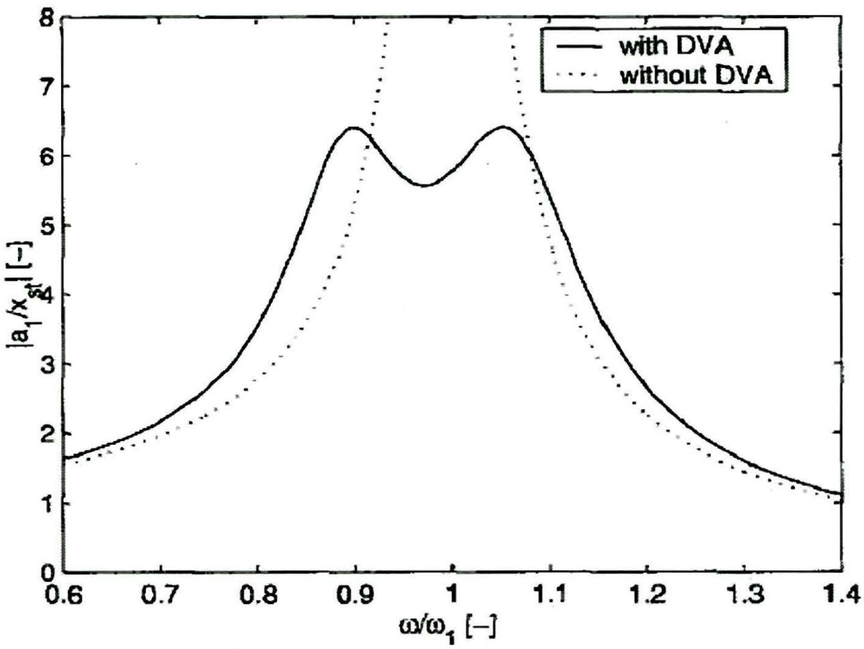

Figure 2.8: Graph of Amplitude Ratio against Frequency Ratio (Damped DVA) (Bonsel, 2003).

2.6 HYBRID VIBRATION ABSORBER

Hybrid vibration absorber is constructed by active element such as actuator with a passive dynamic vibration absorber. Hybrid vibration absorber is used to reduce the resonant and mean square vibration amplitude of the vibrating structure (M.H. Tso, 2012).

Case A shows the vibration control of the simply supported beam by using hybrid vibration absorber. While Case B shows the vibration control of simply supported beam by using passive vibration absorber.

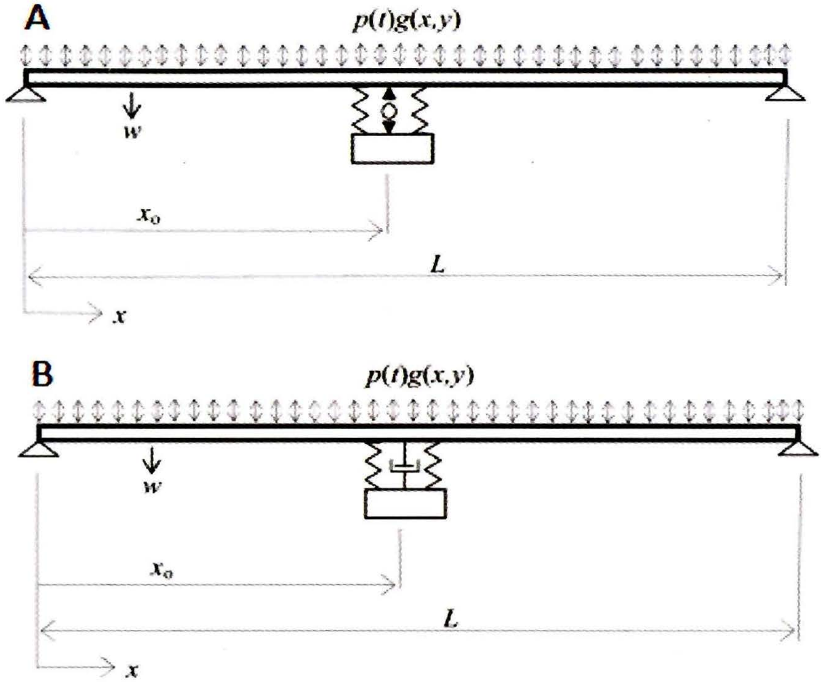

Figure 2.9: Schematic diagram of Case A and Case B (Y.L. Cheung, 2012).

The dash-line in Figure 2.10 shows the mean square vibration amplitudes of Case B, while the solid line in Figure 2.10 indicate the mean square vibration amplitudes of Case A. From Figure 2.10, it is clearly shown that the mean square vibration amplitude of the vibrating structure is reduced more by using hybrid vibration absorber compared to passive vibration absorber.

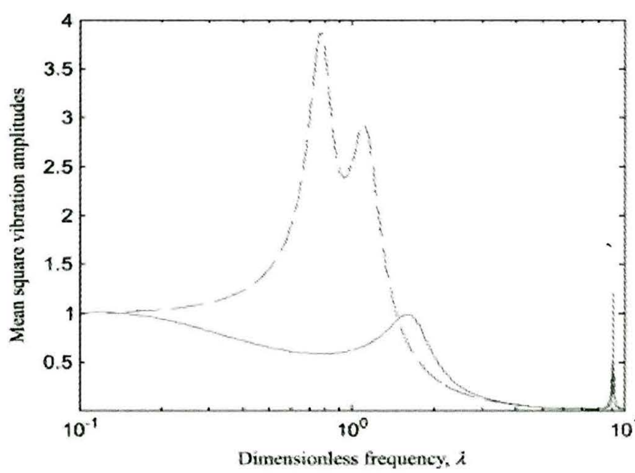

Figure 2.10: Graph of Mean Square Vibration Amplitudes against Dimensionless Frequency (Y.L. Cheung, 2012).

CHAPTER 3

METHODOLOGY

3.1 OVERVIEW

A research procedures or processes were carried out in systematic way by methodology. Gantt charts are generated to provide tasks visualization of the final year project. Every tasks stated in Gantt Charts are served together with the planning duration of task completion. PSM is divided into 2 parts such as PSM 1 and PSM 2. In PSM 1 schedule, journals related to dynamic vibration absorber is reviewed. Besides that, mathematical and MATLAB model have been developed to represent the relationship between amplitudes, frequencies and damping ratio. The lumped mass system was designed and the fabrication materials were prepared. A model constructed of two mild steel beams with dimension 60cm length, 4cm width and 0.2cm thickness were used in fabricating the lumped mass system. Besides that, hollow section of 0.2cm thickness was used to construct the frame and stand for the double beam system. However, the mild steel beams are only used to validate the derived mathematical model and provide visualization about the effect of dynamic vibration absorber in attenuating vibration. In PSM 2 schedule, the experimental model is fabricated and the experiment has been set up. Measurement on the amplitude of the mild steel beam will be carried out by the accelerometer. Then the effectiveness of the beam vibration absorber will be analyzed.