

**SUPPRESSING STRUCTURAL VIBRATION USING MULTIPLE ARRAYS OF
LINEAR DYNAMIC VIBRATION ABSORBER (DVA)**

CHAN CHEN HUA

**A report submitted
in fulfillment of the requirements for the degree of Bachelor of Mechanical of Engineering**

Faculty of Mechanical Engineering

UNIVERSITI TEKNIKAL MALAYSIA MELAKA

2018

DECLARATION

I declare that this report entitled “Suppressing structural vibration using multiple arrays of linear dynamic vibration absorbers (DVA)” is the result of my own research except as cited in the references. The report has not been accepted for any degree and is not concurrently submitted in candidature of any other degree.

Signature :

Name :

Date :

APPROVAL

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 degree of Bachelor of Mechanical Engineering.

Signature :

Supervisor Name :

Date :

ABSTRACT

A fixed-free beam is a structural element supported at one side only and have no any constraint at other side. Exposure to vibration can cause excessive deflections and structure might failed. The aim of this research is to study the application of multi dynamic vibration absorber on a fixed- free beam strucure. For experimentation, two absorbers were fabricated to be installed to the beam and subjected to a force vibration frequency loading using shaker. The resonance frequencies of interest were 14Hz and 99.6Hz. The vibration level of beam is measured for comparing the effect with the presence of application vibration absorbers to see the reduction in its amplitudes. The results shows the reduction of amplitude with single vibration absorber. With application of multiple DVAs, the frequency bandwidth of suppression is increased. Configuration of vibration absorbers on location did not affect its performance in vibration suppression at second natural frequency. From these results, it showd that distributed vibration absorbers is the prefered method of configuration where the range of frequency bandwidth is greatest. The knowledge gained from this research can be used to minimize the vibration amplitdue of sturcutures and increase their life-span. This application can be apply in suppressing the vibration of machine, bridge, and fluttering of airplane wing.

ABSTRAK

Rasuk bebas tetap adalah elemen struktur yang disokong di satu pihak sahaja dan tidak mempunyai sebarang kekangan di sisi satu lagi. Pendedahan kepada getaran boleh menyebabkan pesongan berlebihan dan struktur mengalami kemungkinan gagal. Tujuan penyelidikan ini adalah untuk mengkaji penggunaan penyerap getaran pelbagai dinamik pada structure rasuk tetap bebas. Untuk eksperimen, dua penyerap telah direka untuk dipasang ke rasuk dan tertakluk kepada frekuensi getaran daya getaran menggunakan penggetar. Frekuensi resonans yang menjadi tumpuan kajian adalah 14Hz dan 99.6Hz. Tahap getaran terhasil pada rasuk diukur untuk membandingkan kesan dengan kehadiran penyerap getaran permohonan untuk melihat pengurangan dalam amplitudnya. Keputusan menunjukkan hasil pengurangan amplitud dengan penyerap getaran tunggal. Dengan menggunakan pelbagai DVA, frekuensi jalur lebar kekerapan penindasan meningkat. Konfigurasi penyerap getaran di lokasi tidak mempengaruhi prestasinya dalam penindihan getaran pada frekuensi resonans kedua. Hasil daripada kajian ini menunjukkan bahawa penyerap getaran teragih adalah kaedah konfigurasi pilihan di mana julat frekuensi jalur lebar adalah lebih besar. Pengetahuan yang diperolehi daripada penyelidikan ini boleh digunakan untuk meminimumkan getaran getaran daripada pelebaran dan meningkatkan jangka hayat mereka. Permohonan ini boleh digunakan pada getaran mesin, jambatan, dan pergerakan sayap pesawat.

ACKNOWLEDGEMENTS

First, I would like express my most sincere gratitude to my supervisor Associate Professor Dr. Roszaidi Ramlan from the Faculty of Mechanical Engineering Universiti Teknikal Malaysia Melaka (UTeM) for his essential supervision and guidance towards the completion of this thesis.

Particularly, I would also like to express my deepest gratitude to Mr. Johardi Bin Abdul Jabar, technician from Vibro laboratory Faculty of Mechanical Engineering, and Mr Muhammad Harith Mustaffer, a master student for their assistance and efforts in all the lab and analysis works.

Finally, I dedicate this thesis to my beloved family specially, a greatest appreciation thanks to my parents for their encouragement, guidance and moral support. I would like to use this opportunity to express my gratitude to my friend, Laura for her support.

TABLE OF CONTENTS

	PAGE
DECLARATION	
ABSTRACT	i
ABSTRAK	ii
ACKNOWLEDGEMENTS	iii
TABLE OF CONTENTS	iv-v
LIST OF TABLES	vi
LIST OF FIGURES	vii-x
LIST OF APPENDICES	xi-
LIST OF ABBREVIATIONS	xii
CHAPTER	
1. INTRODUCTION	1
1.1 Background	1-5
1.2 Problem Statement	5-6
1.3 Objective	6
1.4 Scope of Project	6
2. LITERATURE REVIEW	7
2.1 Vibration	7
2.1.1 Vibration Terminology	7
2.1.2 Elementary Parts of Vibrating System	7
2.1.3 Free Vibration and Force Vibration	7-8
2.1.4 Deterministic and Random Vibration	8
2.1.5 Discrete VS Continuous	8
2.2 Beam Structure As Continuous System	8-9
2.3 Mode Shape	9-10
2.4 Terminology of Dynamic Vibration Absorber	10-11
2.5 Active Tuned Vibration Absorber	11
2.6 Adaptive Tuned Vibration Absorber	12
2.6.1 Nonlinear Adaptive Vibration absorber based On Shaped Memory Alloy(SMA)	12-14
2.6.2 Cantilever Beam as Tuned Vibration Absorber	14-19
2.6.3 Implementation of Magnet in electromagnetic vibration absorber	19
2.6.4 Curved Beam Tuned Vibration Absorber	20-21
2.7 Passive Tuned Vibration Absorber	21
2.7.1 Nonlinear Tuned Vibration Absorber	21-24
2.7.2 Linear Tuned Vibration Absorber	24-28
3. METHODOLOGY	29
3.1 Overview	29-30
3.2 Theoretical Calculation of Cantilever Beam	31-32
3.2.1 Calculation of Natural Frequencies of Unloaded Beam	32-33
3.2.2 Calculation of Natural Frequencies of Loaded Beam	33-34

3.3	Excitation Technique of Beam	34-35
3.4	Operating Deflection Shape	36
3.5	Conceptual Design of DVA	37-39
3.6	Design and Fabrication of the Absorber clamp	40
3.7	Tuning of DVAs Using Impact Testing	40-42
3.8	Performance Testing of Fabricated DVAs	42
	3.8.1 Performance Testing of DVA at 1st Natural Frequencies	42-43
	3.8.2 Performance Testing of DVA at 2nd Natural Frequencies	44-47
4.	RESULT AND DISCUSSION	48
4.1	First and Second Natural Frequencies of loaded and unloaded beams	48-51
4.2	Operating Deflection Shape of Beam	51-53
4.3	Tuning of DVA	52-54
4.4	Performance of DVA at 1st Natural Frequency	53-55
	4.4.1 Performance of Single DVA at 1st Natural Frequency	55-57
	4.4.2 Performance of Multiples DVA at 1st Natural Frequency	57-60
4.5	Performance of DVA at 2nd Natural Frequency	60
	4.5.1 Performance of Single DVA at 2nd Natural Frequency	60-64
	4.5.2 Comparison of Performance of Multiple DVAs At 2nd Natural Frequency	64-68
	4.5.3 Comparison for Performance of Single DVA And Multiple DVAs at 2nd Natural Frequency	69-73
5.	CONCLUSION AND RECOMMENDATIONS FOR FUTURE RESEARCH	74
5.1	Conclusion	74
5.2	Recommendations	75
	REFERENCES	76-79
	APPENDICES	80

List of Tables

Table	Title	Page
3.1	Parameter of the beam	32
3.2	Information of each DVA	39
4.1	Percentage Difference of Natural Frequencies	51
4.2	Percentage Difference of Calculated Tuning Length and Actual Tuning Length	55
4.3	Summarization of Percentage of Vibration Amplitude and Range of Bandwidth for Single DVA	64
4.4	Summarization of Percentage of Vibration Amplitude and Range of Bandwidth for Multiple DVAs	68
4.5	Comparison of performance of Single DVAs and multiple DVAs at Point A	70
4.6	Comparison of performance of Single DVAs and multiple DVAs at Point B	73

List of Figures

Figure	Title	Page
1.1	Collapse of Tacoma Bridge	2
1.2	Broughton Suspended Bridge after Rebuilt	2
1.3	Tuned Mass Damper in Taipei 101	3
1.4	Millennium Bridge	4
1.5	Application of Stockbridge Damper on Transmission Cable	5
1.6	Frequency Response	6
2.1	Boundary Conditions of Beam	9
2.2a	First Mode of Vibration at Fundamental Frequency	10
2.2b	Second Mode of Vibration	10
2.2c	Third Mode of Vibration	10
2.3	Attachment of DVA on Primary Structure	11
2.4a	Configuration of SMA	13
2.4b	Attachment of SMA ATVA on Primary Structure	13
2.5	ATVA using wired made out of SMA and insert eddy current damping	14
2.6	Cantilever beam with attach mass as tuned vibration absorber	14
2.7a	Leaf Type Adaptive Vibration Absorber	15
2.7b	Nonlinear behavior of the system	16
2.8	Design of absorber for aircraft turboprop noises suppression	16
2.9a	Design of Piecewise Linear Stiffness	17
2.9b	Configuration of Piecewise Linear Stiffness Absorber	17
2.9c	Nonlinearity with increase of slider position	17
2.10	Leaf Spring Absorber	18

2.11	Design of Vibration Absorber	19
2.12	Design of Electromagnetic Vibration Absorber	19
2.13a	Schematic diagram of dual beam absorber	20
2.13b	Schematic Diagram shows the Crown Height of the beam	20
2.13c	Configuration of Beam Absorber	21
2.14	Comparison of Linear and Nonlinear Absorber	22
2.15	Attachment of Magnet on Metastructure Structure	23
2.16	Comparison between linear stiffness and nonlinear stiffness	23
2.17	Nonlinearities in Cubical Spring	24
2.18	Configuration of suspended multi-beam absorber for vibration suppression	25
2.19a	The Schematic Diagram of Experimental Set Up of Piecewise Linear Absorber	25
2.19b	The Design of the Absorber	26
2.20	Vibration absorber consists of leaf spring and helical spring	26
2.21	Attachment of spring on clamped-clamped beam	27
2.22	Response graph	28
3.1	Summarized Deployment Flowchart	30
3.2	Cantilever Beam	31
3.3	Mode shape of first mode of vibration	32
3.4	Mode shape of second mode of vibration	33
3.5	Experiment setup for determine natural frequency of loaded beam	35
3.6	Software interface of Data Physics Analyzer	35
3.7	Examples of defined Points on Beam Using Available Software	36
3.8	Two DVA mounting with total mass of 0.408kg	40
3.9	Schematic diagram of the configuration of the experiment	41
3.10	Impact Testing of Fabricated DVA	42
3.11a	Experiment Configuration of Single DVA	43
3.11b	Experiment Configuration of Multiple DVA	43
3.12a	Experiment configuration of single DVA at point A	44
3.12b	Experiment configuration of single DVA at point B	45

3.13a	Experiment configuration of multiple DVAs at point A	46
3.13b	Experiment configuration of multiple DVAs at point B	46
3.13c	DVAs are distributed where DVA3 is attached at point A and DVA 4 is attached at point B	47
4.1a	Schematic diagram of unloaded beam	48
4.1b	Natural frequencies (1 st and 2 nd) of the unloaded Beam	49
4.2a	Schematic Diagram of loaded Beam	50
4.2b	Natural Frequencies (1 st and 2 nd) of the loaded Beam	50
4.2c	Comparison of natural frequencies of loaded Beam and unloaded Bea	50
4.3a	ODS for first mode of vibrating beam	52
4.3b	Schematic diagram showing the point with maximum amplitude at first natural frequency	52
4.4a	ODS for second mode of vibrating beam	52
4.4b	Schematic diagram of points with maximum amplitude at second natural frequency	53
4.5a	Frequency Tuning of DVA1 and DVA2	54
4.5b	Frequency Tuning of DVA3 and DVA4	54
4.6a	Schematic Diagram of DVA1 attached to beam	56
4.6b	Frequency Response of Beam with Single DVA attached	57
4.6c	Phase Changes of Beam and DVA1	57
4.7	Schematic diagram of DVA1 and DVA2 attached to beam	58
4.8	Comparison of Performance in terms of bandwidth increase of Single DVA and Multiple DVAs on suppressing the beam vibration amplitude	59
4.9	Comparison of Performance in terms of vibration amplitude reduction ease of Single DVA and Multiple DVAs on suppressing the beam vibration amplitude	60
4.10	Vibration amplitude of Point A and Point B	61
4.11a	Schematic diagram of DVA3 attached to point A	62
4.11b	Schematic diagram of DVA3 attached to point B	62
4.11c	Frequency response of beam at point A with DVA3 attached	62

4.11d	Frequency response of ream at point B with DVA3 attached	63
4.11e	Bandwidth increase in Frequency response at point A	63
4.11f	Bandwidth increase in frequency response at point	63
4.12a	Schematic Diagram of concentrated DVAs attached to Point A	65
4.12b	Schematic Diagram of concentrated DVAs attached to Point B	65
4.12c	Schematic Diagram of distributed DVAs attached to Point A and DVA4 attached to point B	65
4.13	Comparison of Frequency Response at Point A with different location of attachment of multiple DVAs	67
4.14	Comparison of Frequency Response at Point B with different location of attachment of multiple DVAs	67
4.15	Comparison of Single DVA and Multiple DVAs at Point A	70
4.16a	Comparison of Frequency Bandwidth for Single DVA and multiples DVAs at Point B	71
4.16b	Comparison of vibration amplitude reduction for Single DVA and multiples DVAs at Point B	72

List of Appendices

Appendix	Title	Page
A	Presence of Noise at interval of 26Hz in Experiment	80

LIST OF ABBREVIATIONS

DVA	Dynamic Vibration Absorber
TVA	Tuned Vibration Absorber
TVD	Tuned Vibration Damper
ATVA	Adaptive Tuned Vibration Absorber
SMA	Shape Memory Alloy
ATMD	Adaptive Tuned Mass Damper
EVA	Electromagnetic Vibration Absorber
PTVA	Passive Tuned Vibration Absorber
VA	Vibration Absorber
ODS	Operating Deflection Shape
HVAC	Heating Ventilation Air Conditioning
ODS	Operating Deflection Shape
FRF	Frequency Response Function
FFT	Fast Fourier Transform

CHAPTER 1

INTRODUCTION

1.1 BACKGROUND

Structural vibration is basically described as repetitive motion that can be measured and observed in the structure which occurs due to present of external forces and internal forces. External forces are the force that acts from the outside of the structure such as where these external forces can be due to wind induction, seismic activity, rail system or jackhammer. Internal forces are forces that generated from vibrating structure and transmit to another structure such as force generated from rotating imbalance of a pump inside a plant and transmission to the nearby machine. In most cases, this vibration is undesired as it will results in fatigues in structure, reducing the performance and the life span of the structures where these symptoms is not visible. In production plant, vibration can resulted in increase of energy loss, efficiency reduction, and increase of maintenance cost. At this point, vibration can be a source of engineering problem that needed to be solved.

Resonance is a phenomenon that occurs in vibrating machines or structure, when the external excitation frequency coincides with the natural frequency of the vibrating object. In most cases, resonance is detrimental as it will results in large vibrating magnitude and this will lead devastating effects on machines or collapsing of a structure. One of effects of resonance are incident collapsing of Tacoma Narrows Bridge shown in Figure 1.1, a suspended bridge on 7 November 1940 due to wind-induced vibration. There is also case of bridge collapsing due to

induction of mechanical resonance such as collapse of Broughton Suspension Bridge due to marching steps by the troops. The Broughton Suspension Bridge is shown in Figure 1.2.



Figure 1.1: Collapse of Tacoma Bridge (Ellison, 2015).



Figure 1.2: Broughton Suspended Bridge after Rebuilt (Dan, 2016).

As Taiwan prone to natural phenomenon such as earthquake wind storm, The Taipei 101, one of the tallest buildings in the world had installed a 728 Ton pendulum suspended from 92th floor to 87th floor that acts as mass damper to cancel the movement of building induced by strong gust of wind. The Figure 1.4 shows that the location of installment of mass damper inside

the buildings. For Millennium Bridge shown in Figure 1.5, precautions were taken by engineers to install 37 fluid-viscous dampers and 52 tuned mass dampers to dampen the vibration caused by pedestrians' footsteps when crossing the bridge.



Figure 1.3: Tuned Mass Damper in Taipei 101 (Kaushik, 2014).



Figure 1.4 Millennium Bridge (McManus, 2015).

Stockbridge damper, is a dumbbell-shaped device that consists of two masses at the ends of a short length of cable or flexible rod is also used to suppress the oscillation of transmission line caused by induction of wind. The mass damper is usually clamped at its middle to the main cable due to the large amplitude at the middle of cable as it is clamped at both ends. The figure of application of Stockbridge Damper is shown in Figure 1.5.

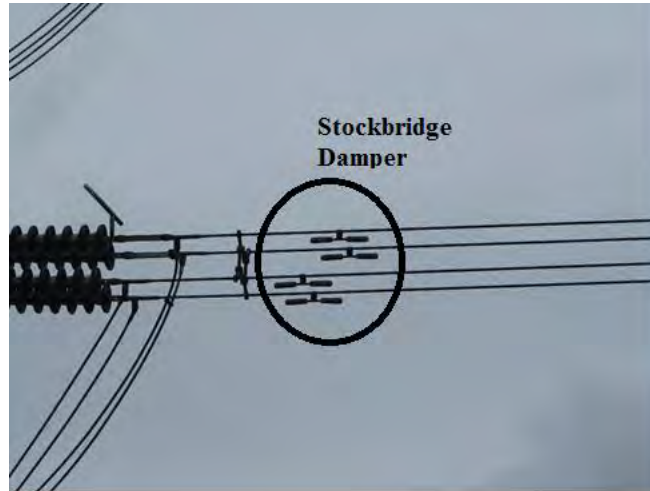


Figure 1.5: Application of Stockbridge Damper on Transmission Cable (Rao, 2015).

1.2 PROBLEM STATEMENT

Dynamic Vibration Absorber (DVA) is a designed mechanical devices that consists of spring-mass system that is able reduce unwanted vibration in machines or structure by exerting a counter force to couple the vibration force. However, DVA can only tuned to a single frequency and it's only effective for that particular resonance that is tuned. Figure 1.6 shows the graph of amplitude against frequency ratio. From Figure 1.6, there is one single point where amplitude is minimum at frequency ratio almost equal to one. The frequency ratio of one means that the DVA's frequency should be equal to the natural frequency of primary system. This indicates that any frequency miss-tuned of the dynamic vibration absorber's frequency result in larger vibration amplitude which will affect the main vibrating system.

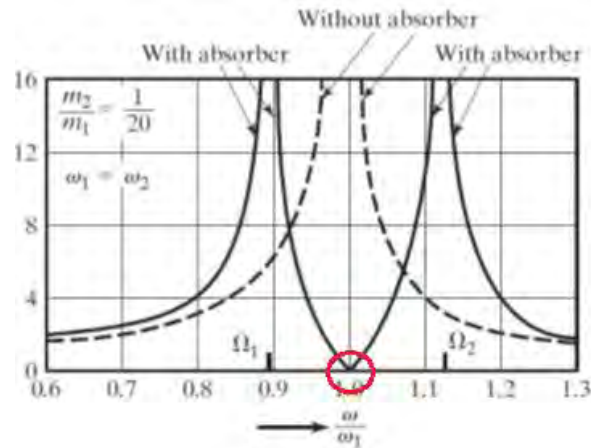


Figure 1.6: Frequency Response (Rao, 2015).

1.3 OBJECTIVE

The objectives of this project are as follows:

1. To characterize the property of a linear DVA.
2. To design and characterize DVA for main structure vibration suppression.
3. To investigate the performance of multiple arrays linear DVA to suppress structural vibration.

1.4 SCOPE OF PROJECT

1. The first and second natural frequency of beam will be considered.
2. Only two DVA will be used to suppress the vibration amplitude of the beam at the first and second natural frequency.

CHAPTER 2

LITERATURE REVIEW

2.1 Vibration

2.1.1 Vibration Terminology

Vibration is a periodic motion of particles of an elastic body or medium in alternately opposite from position of equilibrium where that equilibrium has been disturbed. In other words, vibration is the repetitive motion of after an interval of time, which also referred as oscillation.

2.1.2 Elementary Parts of Vibrating System

In a vibrating system, it consists of mass (inertia) for storing kinetic energy, spring (elasticity) for storing potential energy and damper for dissipating energy. The vibration of system involve the exchange of kinetic energy and potential energy and vice versa, in an alternate manner. Present of damper in vibrating system will dissipate energy for each cycles of vibration and eventually convert the energy of the vibrating system into other forms.

2.1.3 Free Vibration and Forced Vibration

For free vibration, the system is vibrating on its own without the influence of external force. For free vibration, the system is vibrating under external driving force

and if the excited frequency coincide with one of the natural frequency of system, resonance will occurs and resulted large vibration amplitude.

2.1.4 Deterministic VS Random Vibration

In deterministic vibration, the magnitude of the force excitation acting on vibrating system is known at any instant. In this case, the excitation is deterministic. Example for deterministic vibration is sinusoidal force produce by shaker where the signal is generated by computer controlled and amplified by amplifier. For random vibration, the value of the excitation at that particular instant not able to be determine. Examples of random vibration are wind induction, seismic activity, or a car moving on a rough surface road.

2.1.5 Discrete VS Continuous

Number of degree of freedoms refer to number of independent coordinates that required to determine completely the position of system at any of time. For discrete or lumped parameter system, the number of degree of freedom are finite. In contrary, continuous or distributed system have infinite number of degrees of freedom and mostly related to deformable structural and machine system with elastic properties.

2.2 Beam Structure as Continuous System

Euler-Bernoulli Theorem, a linear theory of elasticity which relate between beam's deflections and applied load is widely use to study the propagation of flexural waves by considering the boundary condition of the beam. The boundary conditions of beam include

pinned-pinned, free-free, fixed-fixed, fixed-free, fixed-pinned and pinned free. The Figure 2.1 shows the boundary conditions of the beam.

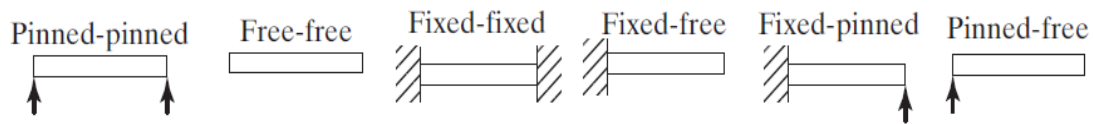


Figure 2.1: Boundary conditions of beam (Rao, 2011).

2.3 Mode Shape

Compared to single mass attached to a single spring, the vibration of complex body vibrate in more different ways. By changing the excitation frequency, the body will exhibit different vibrating pattern, where these patterns are termed as mode shape. Consider a string that is simply supported at both side. By exciting the lowest frequency, also known as the fundamental frequency at the string, the string will oscillates repeatedly as one– with the greatest motion in the center of the string. The Figure 2.2 shows the mode shapes of the string at the different frequency. When the frequency is increase twice the fundamental frequency, two halves of the spring vibrate in opposite manner as shown in Figure 2.3. The solid line is the maximum displacement at one instant and dotted line is the displacement at later instant. If the frequency is further increase to third mode as in Figure 2.4, vibration of string will be divided into three equal length section, each vibrating in opposite to adjacent pieces. Nodal point is point with zero displacement and antinode is point where displacement is maximum.