

**ATTENUATION OF GRASS TRIMMER HANDLE
VIBRATION USING IMPOSING NODE
TECHNIQUE**

KO YING HAO

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**ATTENUATION OF GRASS TRIMMER HANDLE VIBRATION USING
IMPOSING NODE TECHNIQUE**

by

KO YING HAO

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requirements for the degree
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TABLE OF CONTENTS

	Page
ACKNOWLEDGEMENTS	ii
TABLE OF CONTENTS	iii
LIST OF TABLES	vii
LIST OF FIGURES	viii
LIST OF NOMENCLATURE	xi
LIST OF APPENDICES	xvi
LIST OF PUBLICATIONS	xvi
ABSTRAK	xvii
ABSTRACT	xix
CHAPTER ONE : INTRODUCTION	1
1.0 Overview	1
1.1 Brief introduction	1
1.2 Objectives	5
1.3 Contributions	6
1.4 Thesis scope and outline	6
CHAPTER TWO : LITERATURE REVIEW	8
2.0 Overview	8
2.1 Clinical features of hand arm vibration syndrome	8
2.2 Epidemiological study of hand arm vibration	10
2.3 Standards for measurement and evaluation of hand arm vibration	13
2.4 Existing form of handle vibration confinement	15

2.4.1	Anti-vibration gloves	16
2.4.2	Vibration isolator	18
2.4.3	Tuned vibration absorber (TVA)	21
2.5	Imposing node technique	27
2.6	Recent studies on the effort to reduce grass trimmer handle vibration	29
2.7	Summary	32
CHAPTER THREE : METHODOLOGY		35
3.0	Overview	35
3.1	Grass trimmer description	35
3.2	Identify handle vibration of grass trimmer	37
3.2.1	Frequency analysis of grass trimmer handle vibration	37
3.2.2	Handle acceleration transmissibility measurement	39
3.2.3	Experimental modal analysis of grass trimmer	40
3.2.4	Operational deflection shape (ODS)	43
3.3	Implementation of imposing node technique on grass trimmer	45
3.3.1	Grass trimmer modeling	45
3.3.2	Governing transverse deflection equations	46
3.3.3	Obtaining transverse deflection from experiment	49
3.3.4	Parameter estimation	50
3.3.5	Imposing node at loop handle location using TVAs	51
3.4	Experimental test and design of the TVAs	55
3.4.1	Design of practical TVAs	55
3.4.2	Experimental verification	58
3.5	Field test and subjective evaluation	59
3.6	Section summary	62

CHAPTER FOUR : RESULTS AND DISCUSSIONS	64
4.0 Overview	64
4.1 Handle vibration identification of grass trimmer	64
4.1.1 Frequency analysis of handle vibration	64
4.1.2 Frequency-weighted rms acceleration	68
4.1.3 Acceleration transmissibility of loop handle	70
4.1.4 Experimental modal analysis of grass trimmer	71
4.1.5 Operating deflection shape	74
4.1.6 Summary	76
4.2 Implementation of imposing node technique	77
4.2.1 Transverse deflection	77
4.2.2 Parameters estimation	78
4.2.3 Imposing node using TVAs	79
4.3 Tuning of TVAs	82
4.4 Experimental validation	83
4.4.1 Experimental Transverse Displacement (with TVAs)	83
4.4.2 Vibration reduction subject to variation frequencies	85
4.4.3 Operating deflection shape of grass trimmer with proposed TVAs	88
4.4.4 Experimental modal analysis of grass trimmer with proposed TVAs	90
4.5 Effectiveness of TVAs	92
4.5.1 Frequency-weighted rms acceleration and vibration total value	92
4.5.2 Reduction of vibration	94
4.5.3 Subjective rating	97
CHAPTER FIVE : CONCLUSION	100

5.0 Conclusion	100
5.1 Recommendations for future research	101
BIBLIOGRAPHY	103
APPENDICES	
Appendix A: Coordinate system for hand-arm vibration measurement	117
Appendix B: Experimental modal analysis of the shaft of grass trimmer	118
Appendix C: Logarithmic decrement technique	122
PUBLICATIONS	124

LIST OF TABLES

		Page
Table 3.1	Important specification of grass trimmed used in present study	37
Table 3.2	Physical characteristics of the operators participated in the field test	61
Table 4.1	Modal parameters identified from global optimization	78
Table 4.2	Frequency-weighted rms acceleration and standard deviation (loop handle)	93
Table 4.3	Frequency-weighted rms acceleration and standard deviation (rear handle)	94
Table 4.4	The t-test values for the reduction of frequency-weighted acceleration in different conditions (loop handle)	96
Table 4.5	The t-test values for the reduction of frequency-weighted acceleration in different conditions (rear handle)	97
Table B.1	First five natural frequencies and its corresponding damping ratio of the shaft	120
Table C.1	Subjective Rating of perception of vibration based on Borg CR 10	122

LIST OF FIGURES

		Page
Figure 1.1	Typical grass trimmer usage	2
Figure 2.1	Vibration white finger (Mueller, 2008)	9
Figure 2.2	Carpal Tunnel Syndrome (Branchhardin, 2011)	10
Figure 2.3	Frequency weighting filter, W_h for hand arm vibration with band-limiting included (ISO 5349-1, 2001)	14
Figure 2.4	A simple hand-glove-handle coupling model (Dong, 2005b)	16
Figure 2.5	Schematic diagrams of vibration isolation systems	19
Figure 2.6	Theoretical transmissibility for an isolator	20
Figure 2.7	Concept of tuned vibration absorber	22
Figure 2.8	The narrow band reduction of undamped TVA	23
Figure 2.9	Ergonomic handle (Tudor, 1996)	30
Figure 2.10	Grass trimmer equipped with elastomeric material (Rajbhandary et al., 2011)	31
Figure 3.1	Grass trimmer (Tanaka SUM 328 E)	36
Figure 3.2	Instruments for frequency analysis measurement	38
Figure 3.3	Three orthogonal axes (X_h , Y_h and Z_h) of the handles	39
Figure 3.4	Acceleration transmissibility measurements on loop handle	40
Figure 3.5	Geometry of grass trimmer system for the experimental modal analysis	41
Figure 3.6	Experimental modal analysis of grass trimmer system	43
Figure 3.7	Operating deflection shape measurement of grass trimmer	44
Figure 3.8	Schematic representation of grass trimmer constrained beam model	45

Figure 3.9	Experimental setup for transverse deflection measurement	50
Figure 3.10	Representative of grass trimmer with attachment of TVAs	52
Figure 3.11	Schematic of TVA of dual cantilevered mass design; (a) cantilever beam, (b) cantilever beam with secondary mass attached at the end	57
Figure 3.12	Experimental setup for tuning the natural frequency of TVA	58
Figure 3.13	Transverse deflection measurement of grass trimmer with TVAs	59
Figure 3.14	Field test	60
Figure 3.15	Flow chart of the procedures for carrying out imposing node technique	63
Figure 4.1	Input spectrum of loop handle	66
Figure 4.2	Input spectrum of rear handle	67
Figure 4.3	Frequency spectra of loop handle in one-third octave frequency band	69
Figure 4.4	Frequency spectra of rear handle in one-third octave frequency band	69
Figure 4.5	Acceleration transmissibility of loop handle in different axes	70
Figure 4.6	Summation of FRF curves of grass trimmer	71
Figure 4.7	Mode shape of grass trimmer	73
Figure 4.8	Transmissibility	75
Figure 4.9	Auto power spectrum	75
Figure 4.10	Deflection shape of ODS at (a) 80 Hz ; (b) 100 Hz	76
Figure 4.11	The transverse deflections along the shaft of grass trimmer at (a) 80 Hz and (b) 100Hz	78
Figure 4.12	Comparison of computed transverse deflection with that of the measured data	79
Figure 4.13	The transverse deflections of the grass trimmer system	80

Figure 4.14	The required tuning frequencies for different attachment location	82
Figure 4.15	Frequency response functions for TVAs	83
Figure 4.16	Experimental transverse deflection of a controlled and uncontrolled grass trimmer at 80 Hz	84
Figure 4.17	Experimental transverse deflection of a controlled and uncontrolled grass trimmer at 100 Hz	85
Figure 4.18	Variation of transverse deflection at 4800 ± 500 rpm	86
Figure 4.19	Variation of transverse deflection at 6000 ± 500 rpm	88
Figure 4.20	Auto power spectrum of grass trimmer carrying proposed TVAs	89
Figure 4.21	Operating deflection shape of grass trimmer carrying TVAs at (a) 77.5 Hz ; (b) 99.5 Hz	90
Figure 4.22	Mode shape of grass trimmer carrying TVAs	91
Figure 4.23	Reduction of vibration for loop handle	95
Figure 4.24	Reduction of vibration for rear handle	97
Figure 4.25	Subjective rating, (a) loop handle, (b) rear handle	99
Figure A.1	Hand grip position (ISO 5349-1, 2001)	117
Figure B.1	Experimental modal analysis of the shaft	119
Figure B.2	Summation of FRF curves of shaft	120
Figure B.3	Mode shape of the shaft in Y_h axis	121

LIST OF NOMENCLATURE

Notations	Description
A(8)	Daily vibration exposure
CTS	Carpal Tunnel syndrome
EAV	Exposure action value
ELV	Exposure limit value
EU	European Union
FRF	Frequency response function
HAV	Hand-arm vibration
HAVS	Hand-arm vibration syndrome
ISO	International Standard Organization
MDOF	Multiple-degrees-of freedom
ODS	Operating deflection shape
rms	Root mean square
SDOF	Single-degree-of-freedom
TVA	Tuned vibration absorber
TVA	Tuned vibration absorbers
VWF	Vibration white finger
a_{hj}	rms acceleration for the one-third octave band j
a_{hv}	Vibration total value
a_{hw}	Frequency-weighted rms acceleration in single axis
a_{hwx}	Frequency-weighted rms acceleration in X_h axis
a_{hwy}	Frequency-weighted rms acceleration in Y_h axis

a_{hwz}	Frequency-weighted rms acceleration in Z_h axis
a_k	Variables at k^{th} mode
b_k	Variables at k^{th} mode
c_k	Variables at k^{th} mode
c_p	Damper of primary system
c_s	Damper of secondary SDOF system
d_k	Variables at k^{th} mode
E	Young's modulus
$f(t)$	Harmonic excitation
F_i	Forcing amplitude at the i th harmonic frequency
I	Second moment of inertia of the cantilever beam
m_a	Mass of undamped SDOF absorber
m_b	Mass of cantilever beam
m_{ca}	Mass of brass mass
m_p	Mass of primary system
m_s	Mass of secondary SDOF system
N	Number of modes
j	$\sqrt{-1}$
k_a	Stiffness of undamped SDOF absorber
k_p	Spring of primary system
k_s	Spring of secondary SDOF system
L	Length of the shaft
l_a	Length from the fixed center to secondary mass

l_b	Length of the cantilever beam
P	Number of harmonic excitations
r	Frequency ratio
s	Number of locations along the shaft
T	Total daily duration
T_a	Vibration transmissibility of a glove
T_0	Reference duration of 8 hours
W_h	Frequency weighting filter
W_{hj}	Weighting factor for the one-third octave band j
$w(x, t)$	Transverse deflection
$w_{mj}(x_i)$	The corresponding measured transverse displacement at location x_i
$w_j(x_i)$	The transverse deflection of harmonic frequency j derived from the model at a location x_i
$w_{rva}(x, t)$	The transverse deflection of the grass trimmer with attachment of TVAs
$w_{total}(x, t)$	The total transverse deflection of grass trimmer with TVAs
$w_{80}(x, t)$	The transverse deflection of grass trimmer with TVAs subject to harmonic frequency 80Hz
$w_{100}(x, t)$	The transverse deflection of grass trimmer with TVAs subject to harmonic frequency 100Hz
\ddot{z}_h	Accelerations due to vibration measured at the glove-hand interface
\ddot{z}_0	Accelerations due to vibration measured at tool handle surface
$z_1(t)$	The vibration amplitude of TVAs
$z_2(t)$	The vibration amplitude of TVAs
ω	Excitation frequency

ω_{bk}	k^{th} natural frequency of unloaded shaft
ω_{cb}	Natural frequency of the TVA
ω_{cb1}	Natural frequency of the cantilever beam
ω_{cb2}	Natural frequency of cantilever beam with secondary mass
ω_i	Forcing frequency at the i^{th} harmonic frequency
ω_p	Natural frequency of primary system
ω_s	Natural frequency of secondary system
ω_n	Natural frequency
X	Amplitude of the mass
$x(t)$	Response of the combined isolator system
x_{a1}	Attachment location of TVA1
x_{a2}	Attachment location of TVA2
x_f	Location where the force is applied
x_i	Location along grass trimmer
x_n	Node location
$y(t)$	Steady-state sinusoidal vibration
ζ	Damping ratio
ζ_k	k^{th} damping ratio of unloaded shaft
α	Frequency ratio
μ	Mass ratio
$\eta_k(t)$	Generalized coordinates
$\phi_k(x)$	Eigenfunctions
λ	Eigenvalue
λ_k	Natural frequencies of the constrained grass trimmer system

$E(\chi)$	Optimization or error function to be minimized
χ	Vector of model parameters
ρ_{steel}	Density of steel
$\{V\}$	Eigenvector
$[C]$	Damping matrix
$[C_b]$	Damping of the unloaded grass trimmer
$[C_{tva}]$	Damping matrix of grass trimmer system with attachment of TVAs
$[I]$	Identity matrix
$[K]$	Stiffness matrix
$[K_a]$	The stiffness matrix describe the effects of the attached TVAs
$[K_{tva}]$	Stiffness matrix of grass trimmer system with attachment of TVAs
$[M]$	Mass matrix
$[\hat{M}_a]$	The mass matrix describe the effects of the attached TVAs
$[M_t]$	Inertia coupling matrix caused by the effects of the attached masses
$[M_{tva}]$	Mass matrix of grass trimmer system with attachment of TVAs
$[\Omega_b]$	Stiffness of the unloaded grass trimmer

LIST OF APPENDICES

	Page	
A	Coordinate system for hand-arm vibration measurement	117
B	Experimental modal analysis of the shaft of grass trimmer	118
C	Subjective evaluation	122

LIST OF PUBLICATIONS

		Page
1	Tuned vibration absorber for suppression of hand-arm vibration in electric grass trimmer	125
2	The design and development of suspended handles for reducing hand-arm vibration in petrol driven grass trimmer	126
3	Analysis of dynamic vibration absorber to attenuate hand arm vibration	127

PENGURANGAN GETARAN PEMEGANG MESIN RUMPUT DENGAN TEKNIK PAKSAAN NOD

ABSTRAK

Pengguna mesin rumput terdedah kepada getaran tangan. Pendedahan yang melampau terhadap getaran tersebut akan mendedahkan pengguna kepada risiko sindrom getaran tangan. Teknik paksaan nod telah dikaji demi mencapai getaran yang rendah atau nod di bahagian pemegang mesin rumput. Kajian ini menunjukkan bagi sistem mesin rumput yang boleh dimodelkan sebagai rasuk, getaran pada bahagian mesin rumput dapat dikurangkan. Model matematik untuk sistem mesin rumput telah dibentuk dan kriteria pelaksanaan pengurangan getaran dengan menggunakan teknik paksaan nod telah dibangunkan. Frekuensi talaan yang optimum bagi penyerap getaran bertala yang dipasang pada lokasi $0.74L$ and $0.85L$ sepanjang aci mesin rumput telah ditentukan dengan menggunakan teknik ini. Untuk mengesahkan teori yang dicadangkan dan memaparkan pelaksanaan secara praktikal, eksperimen pesongan melintang, eksperimen analisis ragam dan analisis mod pesongan operasi telah dilakukan. Keputusan menunjukkan bahawa nod telah dipaksa pada sekitar bahagian pemegang mesin rumput ($0.79L$). Selain itu, getaran di sepanjang segmen aci mesin rumput ($0.70L-0.94L$) juga dikurangkan dengan dramatik. Penyerap getaran bertala didapati mempunyai prestasi yang terbaik dengan pengurangan pecutan punca purata kuasadua (ppkd) sebanyak 71% pada pemegang gelung dan 72% pada pemegang belakang. Keputusan daripada eksperimen analisis ragam dan analisis mod pesongan operasi menunjukkan bahawa penyerap getaran bertala telah mengurangkan pesongan secara ketara di mana nodnya telah beralih hampir kepada lokasi pemegang mesin rumput. Keberkesanan penyerap getaran

bertala juga dinilai semasa operasi pemotongan rumput dan taksiran subjektif. Keputusan menunjukkan bahawa pengurangan purata pecutan rms semasa operasi pemotongan rumput adalah sebanyak 25%, 69%, 17%, 58% bagi paksi X_h , Y_h , Z_h dan a_{hv} . Daripada ujian operasi tersebut, taksiran subjektif terhadap persepsi getaran untuk mesin rumput terkawal juga lebih baik.

ATTENUATION OF GRASS TRIMMER HANDLE VIBRATION USING IMPOSING NODE TECHNIQUE

ABSTRACT

Prolonged use of grass trimmer exposes the user to the risk of hand-arm vibration syndrome. Imposing node technique was investigated in order to achieve low vibration (node) at the grass trimmer handle location. It is established that, for the grass trimmer system which can be modelled as a beam, the vibration at the handle location can be minimized. The mathematical model of grass trimmer system is presented, and a criterion for implementing the vibration attenuation by imposing node technique is developed. The optimum tuning frequencies of tuned vibration absorbers (TVAs) attached at the points of $0.74L$ and $0.85L$ along the shaft of grass trimmer were determined using this approach. Transverse deflection, experimental modal analysis, and operating deflection shape analysis of the grass trimmer were carried out and TVAs were designed and fabricated to validate the proposed theory and display its practicality. The results indicated that node was induced in the vicinity of the loop handle location ($0.79L$). Moreover, the vibration along the segment of the shaft ($0.70L$ - $0.94L$) was also found to have relatively small amplitude. The TVAs were found to have best performance with 71% reduction on the frequency-weighted rms acceleration at loop handle and 72% for rear handle. The results from modal analysis and operating deflection shape revealed that the presence of TVAs has successfully reduced the large deformations of the loop and rear handle where the node was shifted nearer to the handle location. The effect of TVAs was also evaluated during the field test involving grass trimming operation and subjective rating. The results indicated that average reduction of frequency-weighted rms acceleration was by 25%, 69%, 17%, 58% in X_h , Y_h , Z_h -axes and a_{hv} respectively

during the cutting operation. From the field test, subjective rating of vibration perception consistently rate better for controlled grass trimmer.

CHAPTER 1

INTRODUCTION

1.0 Overview

In this chapter, a brief introduction of the thesis is presented. The chapter also discusses the motivation behind the research, contributions and objectives. Finally, the chapter will describe the thesis outline.

1.1 Brief introduction

Prolonged use of hand-held power tools increases the risks of hand-arm vibration syndrome (HAVS). Owing to this, the European Union (EU) has adopted a directive in 2005, which provides guidance for making health risk assessments. The EU (2005) has set an exposure action value (EAV) of 2.5 m/s^2 and an exposure limit value (ELV) of 5.0 m/s^2 for daily vibration exposure A(8). Even for lower vibration level it can produce some level of numbness under a prolonged exposure to vibration. Therefore, it is important to find ways to reduce the vibration level of the machines used by the workers.

In Malaysia, a study in 2005 showed that there are about 2.3 million workers who use vibrating tools in agriculture, forestry, construction, mining and quarrying sectors (Su, et al., 2011). Petrol driven grass trimmer is widely used in Malaysia for maintenance of grass areas. A typical public open grass compound requires monthly grass trimming operation (Figure 1.1). This has created a service industry with large numbers of workers using grass trimmer to maintain open grass areas. These workers are exposed to hand-arm vibration (HAV) when operating the grass trimmer. Due to

prolonged use of this type of hand-held machine, serious muscle, nerve and bone problem, collectively name as HAVS could appear. However, the workers involved in these tasks are generally contract labourers with little awareness of HAVS.



Figure 1.1: Typical grass trimmer usage

The BS EN ISO 11806 (2008) has listed that the vibration total value (a_{hv}) at grass trimmer handle is expected to be below 15 m/s^2 for machines with engine displacement of less than 35 cc. Early work on the vibration of grass trimmer handle has reported frequency-weighted rms acceleration to reach the level of 23.94 m/s^2 for an unbalanced single string grass trimmer (Tudor, 1996). Our earlier measurement of a commercial grass trimmer with 32cc engine has recorded a vibration total value (a_{hv}) of 11.30 m/s^2 . It is clear that the vibration total value (a_{hv}) listed above exceeded the ELV of A(8) (based on 4 hours working period) which prevented the safe usage of grass trimmer. Hence a study in greater depth is needed to make this type of machine safer to use.

Various techniques are available for controlling vibration of machine tools. One feasible technique is to absorb the vibration energy using tuned vibration absorber (TVA). The TVA is an additional mass-spring system, when appropriately chosen can suppress the steady state force acting on particular degree of freedom system. Recent studies have shown that the vibration of a location or segment on beam can be made stationary (node) by properly choosing the TVA parameters (Cha and Pierre, 1999; Cha, 2002; Cha, 2005; Cha and Ren, 2006; Cha and Zhou, 2006; Cha and Chan, 2009; Foda and Bassam, 2006; Wong et al., 2007). Assumed-modes method has been implemented together with the use of a chain of undamped TVA as passive mean device to impose nodes for the normal modes of undamped beam during free vibration (Cha and Pierre, 1999; Cha, 2002). The desired node either can coincide with the TVA (collocated) or it can be located elsewhere (non-collocated). As long as the TVA's parameters were carefully chosen such that the natural frequency is similar to the combined natural frequencies of the beam, then a node can be imposed at any desired location of the beam in normal mode for collocated case. However, not all the normal modes can be made to have a node at the desired location when the TVA and the node are not collocated. The imposing nodes technique can also be implemented for an arbitrary supported, linear elastic structure carrying a series of TVA (Cha and Ren, 2006). The constraint of the maximum allowable amplitude of the TVA should be included in the procedure for tuning the TVA (Cha, 2005). Since physical inputs seldom consist of single harmonic excitation, the idea of imposing nodes to suppress vibration for a structure is further developed for conditions of general dynamic loads (Cha and Ren, 2006). For beam-like structure a method to impose points of zero displacement and zero slope can be achieved by attaching properly tuned translational and rotational TVA (Cha and

Zhou, 2006). The case for both point and distributed loading on beam was also considered (Wong, et al., 2007). Dynamic Green function can be used to determine the optimum values of the masses and springs and their locations for imposing node (Foda and Albassam, 2006) and further confinement of a region of beam has also been demonstrated (Alsaif and Foda, 2002). The application of imposing node technique depends on the ability to physically come up with springs of precise stiffness which is challenging to manufacture; to overcome this the imposing node technique is further developed to rely on mass alone (Cha and Chen, 2011).

The application of the imposing node technique is very suitable for machine components or structures that have beam-like behaviour. One such a machine is the grass trimmer where the rotating cutting head is driven by the long shaft housed inside a hollow aluminium shaft supported by several intermediate rubber sleeves and the handle is mounted on the hollow shaft. The vibration of the handle can be harmful to the operator. This handle vibration can potentially be minimized if the node is enforced on the handle location using the imposing node technique. Other conventional vibration attenuation techniques have been applied to the grass trimmer with varying degree of success and may all be effective, provided that certain conditions are satisfied for their performance (E.Roland, 1990; Tudor, 1996; Rajbhandar et al, 2011). However, handle vibration control is still a problem that cannot be solved satisfactorily where most of them require some form of modification to the handle. The imposing node technique on the other hand does not require any modification to the handle; the TVA can be added to the structure without any modification to it. Furthermore, the handle of grass trimmer is the main part that transmits vibration to hand. Therefore, it is important to eliminate vibration

from the handle of the grass trimmer more than other. To the best of the author's knowledge no practical application of the imposing node technique to a specific machine has been reported in the literature and this is the first time the imposing node technique is applied to a practical machine vibration problem. This work proposes the use of imposing node technique to find the optimum tuning frequencies of TVA applied to a grass trimmer system in order to achieve low vibration at the grass trimmer handle. The main advantage of the technique is that the vibration of the point where the handle is mounted can now be passively brought to the theoretically lowest possible level (node) without resorting to handle modification or active vibration control.

1.2 Objective

This thesis investigates a new type of vibration control technique for grass trimmer that involves imposing node to reduce the excessive vibration. The technique proposed here is a unique one. No published results exist in the literature to control vibration in grass trimmer by imposing node technique. Thus, the present research objectives are:

- To develop the procedure of imposing node technique for actual beam-like machine (grass trimmer system) subjected to varying excitations.
- To investigate the effectiveness of proposed TVAs in suppressing handle vibration subjected to varying excitations.
- To evaluate the operator judgment in terms of vibration perception during the field test operation.

1.3 Contributions

This thesis investigates the implementation of imposing node technique to control grass trimmer handle vibration. The imposing node technique is a well-developed technique however to date, there is no report of its application on actual machine. In this case the imposing node technique is being used to enforce the node to be at the vicinity of the handle which will result in very low HAV because the response of a multi-degree of freedom system at the node will be practically very low. The second major contribution is to overcome the narrow-band effectiveness of the TVA in absorbing frequency over a wider range of frequency. The TVA is known to be very effective in attenuating vibration within a narrow frequency range. However in this case, the use of the TVA is in the imposing of the node at the handle since the response of the system at the node over a relatively wider range of frequency is practically very low. The effective frequency range is determined by the mode closest to the excitation frequency. This is to fulfil the gap of knowledge of imposing node technique where this technique can now be apply to actual beam like structure where its excitation frequencies has the tendency to change with the speed of the engine .

1.4 Thesis scope and outline

The scope of this research deals with the implementation of the imposing node technique to achieve low vibration (node) at the grass trimmer handle location. The mathematical model of grass trimmer system will be presented, and a criterion for implementing the vibration attenuation by imposing node technique will be developed. Experimental modal analysis, experimental transverse deflection, operating deflection shape and field test will be carried out to validate the proposed

theory and demonstrate its practicality and to what extent the imposing node technique can be applied to solve structural vibration problem.

The presented thesis is delivered in five primary chapters which commenced with the introduction followed by a literature review, methodology, results and discussions, and conclusions. The first chapter has briefly introduced the issue of HAV arising from conventional grass trimmer and available vibration control approaches.

In chapter two, literature on HAVS risk, several control approaches for hand-held tools and their applications are reviewed. The benefits gained with the use of imposing node technique against traditional vibration approached are also discussed in this chapter.

In chapter three, the flow of research is presented. A grass trimmer model is presented along with simulation results. Finally, the experimental test procedure is discussed at the end of this chapter.

In the chapter four, the outcomes from the experiments are presented and explained. The evaluation and discussion of the results are also embedded in this chapter.

At the end of thesis, chapter five summarises the work which has been completed in this study.

CHAPTER 2

LITERATURE REVIEW

2.0 Overview

Some fundamental knowledge and some key results found in the literature survey related to vibration control are presented here. The survey covers topics for six main scopes:

- Clinical features of hand arm vibration syndrome
- Epidemiological study of hand arm vibration
- Standards for measurement and evaluation of hand arm vibration
- Existing form of handle vibration confinement
- Imposing node technique
- Recent studies on the effort to reduce grass trimmer handle vibration

2.1 Clinical features of hand-arm vibration syndrome

HAVS is a complex disorder which consists of vascular and nonvascular syndromes, associated with the exposure of hand-held vibrating machine tools (Bovenzi, 1998; Mansfield, 2005; Heaver et al., 2011). Giovanni Loriga in 1911 was the first to document the relationship between the exposure of HAV and HAVS (Bylund, 2004). Later in 1918, Leake described a condition called dead fingers and noted that workers using pneumatic tools were likely to incur vascular disturbances in the form of a limited type of local anemia of the fingers (Leake, 1918).

The vascular aspect on HAVS is characterized by Secondary Raynaud's Disease (Bovenzi, 1998; Fridén, 2001; Mansfield, 2005). It is commonly referred as

dead finger or vibration white finger (VWF) (Figure 2.1). In the early stage of VWF, the tips of one or more fingers blanch. With time and further exposures of HAV, severity of the blanching increases. In worse cases, fingers can appear cyanotic and develop trophic changes (Heaver et al., 2011).



Figure 2.1: Vibration white finger (Mueller, 2008)

The nonvascular aspects of HAVS are represented by neurological and musculoskeletal disorders. Results from the neurological disorder are tingling, sensory loss, numbness, decreased dexterity and Carpal tunnel syndrome (CTS). CTS is the commonest reported entrapment neuropathy in the upper limb (Burt et al., 2011; Herbert et al., 2000; Kouyoumdjian et al., 2000). This is due to the compression of median nerve at the hand wrist and thickened ligaments and repetitive stress in the carpal tunnel (Bland, 2007) (Figure 2.2). Musculoskeletal manifestations predominantly present as osteoporosis of the wrist, elbow, joint as well as hand bone cysts (Hagberg, M., 2002).

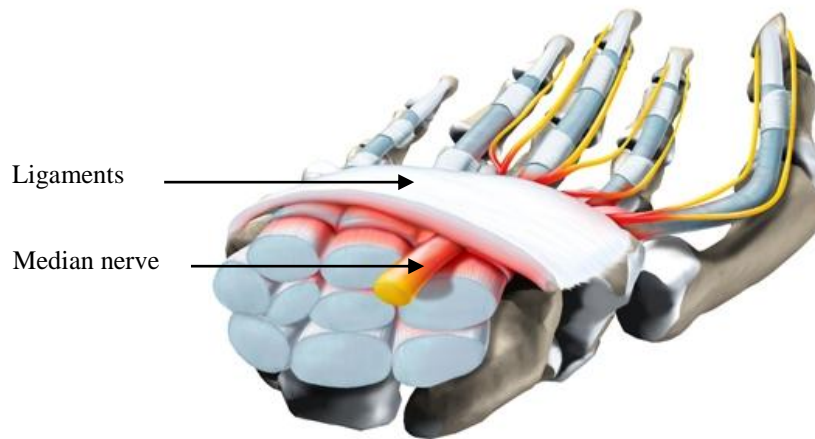


Figure 2.2: Carpal tunnel syndrome (Branchhardin, 2011)

2.2 Epidemiological study of hand arm vibration

The frequency of vibration, duration of exposure, latency and level of exposure are the factors that will affect the development of HAVS (Lin et al., 2005). Different frequency of vibration affects different zones within the hand-arm system (Pyykkö et al., 1976; Reynolds and Angevine, 1977). High frequency (>100 Hz) hand-held tools, such as impact drills and wrenches are associated with a high incidence of wrist symptoms. The low frequency (< 40Hz) hand-held tools, such as rock drills and concrete breakers are associated with more proximal joint symptoms (Dong et al., 2005c, 2005d; Heaver et al., 2011). Poor posture, power grip, repetitive movement and manual labour are among the risk factors for the development of musculoskeletal disorders (Hagberg, 2002).

The duration of exposure to vibration needed to produce HAVS cannot be exactly defined. This is due to different individual susceptibilities to vibration and the different physical characteristics of the vibration exposure (Pelmeur and Leong, 2000). The prevalence of vascular symptoms in workers using vibrating tools can be as high as 71% with average exposure duration of 11.3 years (Letz et al., 1992).

However, short term duration (30 minutes) exposure to HAV is enough to result in the increase of the vibration perception thresholds and the development of numbness (Malchaire et al., 1998).

The latency period of HAVS is extremely difficult to predict (Gemne, 1997). The period of latency before the appearance of the VWF symptoms can be as earlier as one year. Nilsson et al. (1989) have shown that the prevalence of VWF's phenomenon was 8% within four years of exposure, 84% after five to nine years, 94% after ten years and 92% after 40 years in a study of 89 platers using vibrating machinery. With less than 2000 hours' vibration exposure, the latency period for the HAVS symptoms typically is not easily predicted (Miyashita et al., 1983). The latency period varies considerably between individuals due to differences in the type of exposure and individual susceptibility. However, a higher risk of HAVS is generally expected in a work environment where there is a shorter mean latency period (Fridén, 2001).

Most of the studies showed there is a positive association between high level exposure to HAV and the development of HAVS. An epidemiological study in South Africa gold mines showed that 15% of the rock drill workers have been diagnosed with HAVS. Among these 15% workers, 8% had both the vascular and neurological syndromes, 5% had only neurological syndrome and 2% had only vascular disorder. This is due to high vibration levels (up to 31 m/s^2) have been measured in association with rock drills (Nyantumbu et al., 2007). Bovenzi et al. (2005) had investigated the prevalence of HAVS among the female workers using orbital sander. With the weighted rms acceleration of orbital sander averaged from 3.7 m/s^2 to 7.3 m/s^2 . There

are 4% of 100 female workers had the vascular disorder while 19% of them have been diagnosed with CTS. Recent epidemiological study from Malaysia showed that the prevalence of HAVS among 243 construction workers on a Kuala Lumpur construction site exposed to hand arm vibration was 18%. Vibration total values for concrete breakers (10.02 m/s^2), impact drills (7.72 m/s^2) and grinders (5.29 m/s^2) have been reported in the same study (Su, et al., 2011).

Apart from that, studies on the contribution of environmental variables (Scheffer et al., 1989; Yamamoto, 2002) and gender differences (Bylund, 2004; Neely et al., 2006) to the development of HAVS have been reported. An overview of epidemiological studies shows that neurological and musculoskeletal signs of HAVS are more common than vascular symptoms in tropical countries because the critical ambient temperature for the provocation of VWF is around 15°C (Su et al., 2011). Neely et al. (2006) have concluded that there are no differences between male and female subjects for threshold measurement, while Bylund (2004) stated that both female and male are receiving the same power absorption.

The labour force survey report carried out by the Department of Statistics, Malaysia in 2011 showed that about 2.7 million workers were employed in construction, mining, quarrying, agriculture, forestry and mining (Department of Statistics Malaysia, 2011). There are 161 compensation claims under musculoskeletal disorders in 2009 (Social Security Organization (SOCSSO) Malaysia, 2009). Although there are limited report regarding the prevalence of HAVS in Malaysia, the occupation diseases causes by vibration has been monitored by Ministry of Human Resources (Department of Occupational Safety and Healthy,

2010). Recent study by Su et al. (2011) showed that HAVS is a recognisable problem in Malaysia. The study identified clinical symptoms and signs of HAVS among 243 workers working in a construction site, Kuala Lumpur with 18% HAVS prevalence reported.

2.3 Standards for measurement and evaluation of hand arm vibration

In order to make the exposure level of HAV more comparable, these are measured using several standard procedures. International Standard Organization (ISO) drew up the first guideline in 1986 for the measurement and assessment of hand arm vibration. However, this standard was replaced by the new version of ISO 5349 in 2001 (Mansfield, 2005). The ISO 5349 (2001) is divided into two parts. The first part of the ISO 5349 (2001) provides the general requirement for measuring and reporting hand arm vibration exposures while the second part highlights the practical guidance for measurement at the workplace.

ISO 5349-1 (2001) defines a frequency weighting filter, W_h which is the combination of band limiting and weighting filter to allow uniform comparison of level of vibration (Figure 2.3). This filter forecast adverse effect of hand arm vibration over the frequency range by the octave bands from 8 to 1000 Hz. It has been found that low frequency components are most injurious; therefore the highest relative weightings (between 0.873 and 1.00) are concentrated between frequencies components centred from 8 Hz - 16 Hz.

The magnitude of vibration is measured by means of the frequency-weighted root-mean-square (rms) acceleration, expressed in m/s^2 :

$$a_{hw} = \sqrt{\sum_j (W_{hj} a_{hj})^2} \quad (2.1)$$

where a_{hw} is the value of the frequency-weighted rms acceleration, W_{hj} is the weighting factor for the one-third octave band j and a_{hj} is the rms acceleration for the one-third octave band j .

The vibration total value, a_{hv} is established by root-sum-of squares of frequency-weighted rms acceleration measured in three orthogonal axes, written as (ISO 5349-1, 2001):

$$a_{hv} = \sqrt{a_{hwX}^2 + a_{hwY}^2 + a_{hwZ}^2} \quad (2.2)$$

where a_{hwX} , a_{hwY} , a_{hwZ} are the value of a_{hw} (frequency-weighted rms acceleration in single axis) in metres per second squared, m/s^2 , for the X_h , Y_h and Z_h axes respectively.

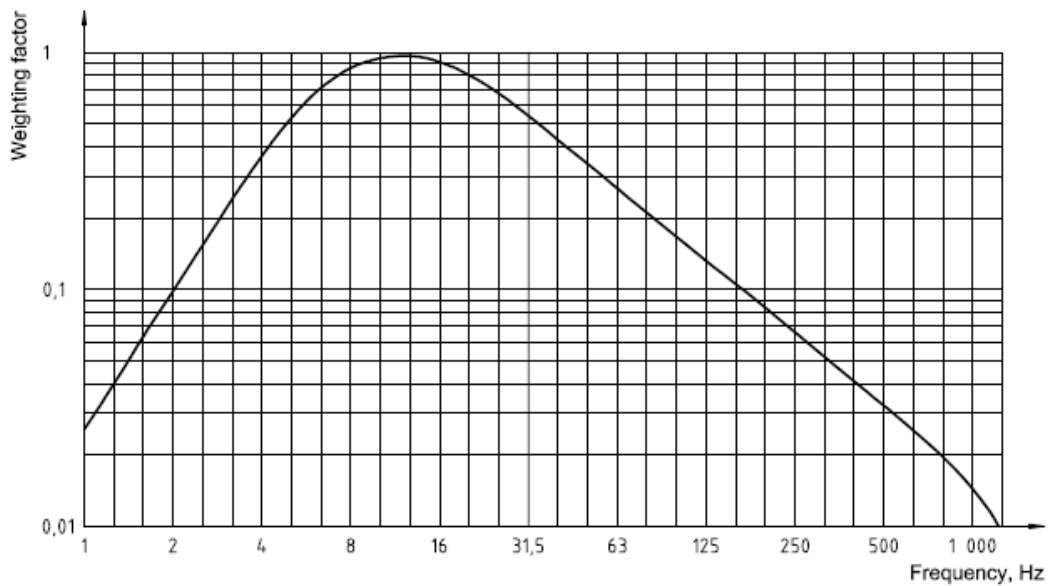


Figure 2.3: Frequency weighting filter, W_h for hand arm vibration with band-limiting included (ISO 5349-1, 2001)

Although the ISO 5349 has standardized the measurements, it does not provide the dose-exposure relationship of hand arm vibration. The European Union (EU) has adopted a directive in 2005 which provides guidance for making risk assessments based on ISO 5349 (2001).

EU (2005) is the replacement of the earlier version of the standard which was established in 2002 (Nelson and Brereton, 2005; EU, 2005). The EU (2005) has set an exposure action value (EAV) of 2.5 m/s^2 and an exposure limit value (ELV) of 5.0 m/s^2 for daily vibration exposure, A(8). The A(8) is derived from vibration total value, a_{hv} and daily exposure duration, given in (ISO 5349-1, 2001):

$$A(8) = a_{\text{hv}} \sqrt{\frac{T}{T_0}} \quad (2.3)$$

where T is the total daily duration and T_0 is the reference duration of 8 hours. The employers are required to take action to control the hand-arm vibration risk of employee's workplace if the A(8) is above the EAV. Meanwhile, the ELV is the limit above which employees should not be exposed to.

2.4 Existing form of handle vibration confinement

Various techniques are available for controlling vibration at the handle. These include isolate the hand from the vibrating handle with the use of anti-vibration gloves (Brown, 1990; Muralidhar et al., 1999; Voss, 1996), isolate the tool handle from the vibrating source by using isolators (Sam and Kathirvel, 2009; Tewari and Dewangan, 2009); mounting a tuned vibration absorber (TVA) on the source of vibration.

2.4.1 Anti-vibration gloves

One of the ways to reduce the risk of HAVS among hand-held workers is to equip them with the appropriate anti-vibration gloves (ScienceDaily, 2011). Anti-vibration gloves are usually lined with special vibration attenuating materials that are intended to decouple the hand-arm system from the tool handle (Sampson and Van Niekerk, 2003). The principal of anti-vibration glove can be represented as a simple hand–glove–handle coupling model (Dong, 2005b), shown in Figure 2.4.

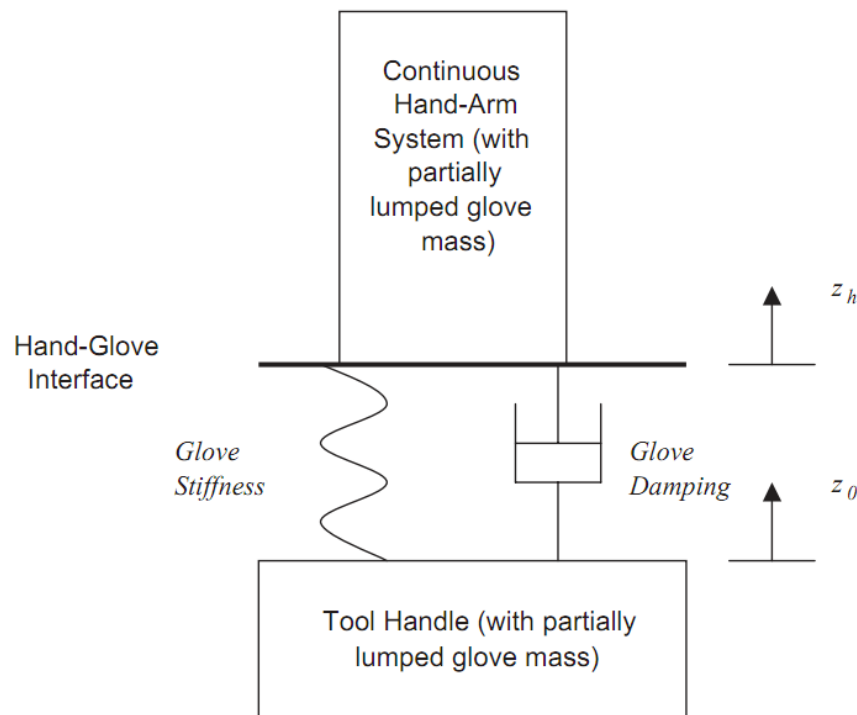


Figure 2.4: A simple hand-glove-handle coupling model (Dong, 2005b)

The glove material is represented as a mass-less spring–damper system, assuming that the mass of the material is partially lumped to the tool handle and the hand-arm system. Commercial anti-vibration gloves are usually made from Gelfôm®, Sylomer®, Viscolas™, etc (Sampson and Van Niekerk, 2003). The effectiveness of

the anti-vibration glove is measured by vibration transmissibility of a glove (T_a), the ratio of the vibration at the glove-hand interface to the handle vibration:

$$T_a = \frac{\ddot{z}_h(j\omega)}{\ddot{z}_0(j\omega)} \quad (2.4)$$

where \ddot{z}_h and \ddot{z}_0 are the accelerations due to vibration measured at the glove-hand interface and the tool handle surface, respectively, corresponding to excitation frequency ω and $j = \sqrt{-1}$.

The use of anti-vibration gloves as a mean of attenuation has been extensively studied (Chang et al., 1999; Dong et al., 2003), which include the effect of vibration reduction on the anti-vibration gloves made of different glove materials (Yun et al., 2011); vibration isolation characteristic for a gloved hand using a laser-based vibration sensor (Gurram et al., 1994); development of a method for assessment of effectiveness of anti-vibration glove (Dong et al., 2003); the evaluation of the effect of wearing anti-vibration gloves on the grip strength applied to cylindrical handles (Wimer et al., 2010).

However, most of the commercially available anti-vibration gloves are not efficient for attenuating vibration below 100Hz (Sampson and Van Niekerk, 2003) and only lessen high frequency vibration (Dong et al., 2009; Smutz et al., 2002). It is important to note that different hand tools will have different influence on the effectiveness of the anti-vibration glove since it is tool or excitation spectrum specific (Rakheja et al., 2002).

Dong et al. (2005a) found a strong linear correlation between effectiveness of anti-vibration gloves with biodynamic characteristic of human hand-arm system in the frequency range of 40-200Hz and the anti-vibration gloves were less effective in the middle frequency range (50-100Hz) for people with larger hand size. Furthermore, the effectiveness of anti-vibration gloves are location specific since vibration perceptions at different locations on the hand-arm system could vary (Dong et al., 2010). It significantly reduces vibration level at the palm but not at the fingers (Dong et al., 2005e). The use of anti-vibration gloves may also lead to reduce dexterity, restricted blood circulation, allergy and lowering of performance rate (Brown, 1990, Muralidhar et al., 1999).

2.4.2 Vibration isolator

Isolator is defined as “support, usually resilient, designed to attenuate the transmission of shock and/or vibration” (ISO 2041, 2009). The concept of vibration isolation is well established and illustrated by considering a single degree-of-freedom system shown in Figure 2.5. This system consists of a mass supported by spring and damper connected to a foundation by an isolator having resilience and energy-dissipating means. The effectiveness of an isolator can be evaluated by the characteristic of the response of the combined isolator system $x(t)$ to steady-state sinusoidal vibration $y(t) = Y \sin \omega t$. This characteristic is referred as the steady state displacement transmissibility, X/Y given as:

$$\frac{X}{Y} = \left[\frac{1+(2\zeta r)^2}{(1-r^2)^2+(2\zeta r)^2} \right]^{1/2} \quad (2.5)$$

where X is the amplitude of the mass, ζ is the relative damping, r is the frequency ratio (ω/ω_n), ω is the excitation frequency and ω_n is the natural frequency.

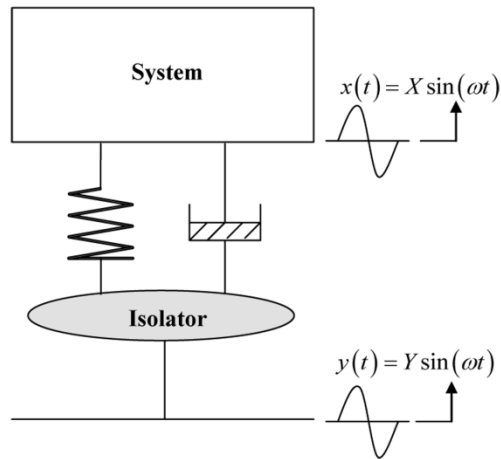


Figure 2.5: Schematic diagrams of vibration isolation systems

The skill in selecting the natural frequency of the isolator is to be well below the forcing frequency. Generally, isolator becomes effective when the frequency ratio exceeds 2 (Fig. 2.6). If the ratio is less than 2, the vibration may easily be amplified (Eugene, 2003).

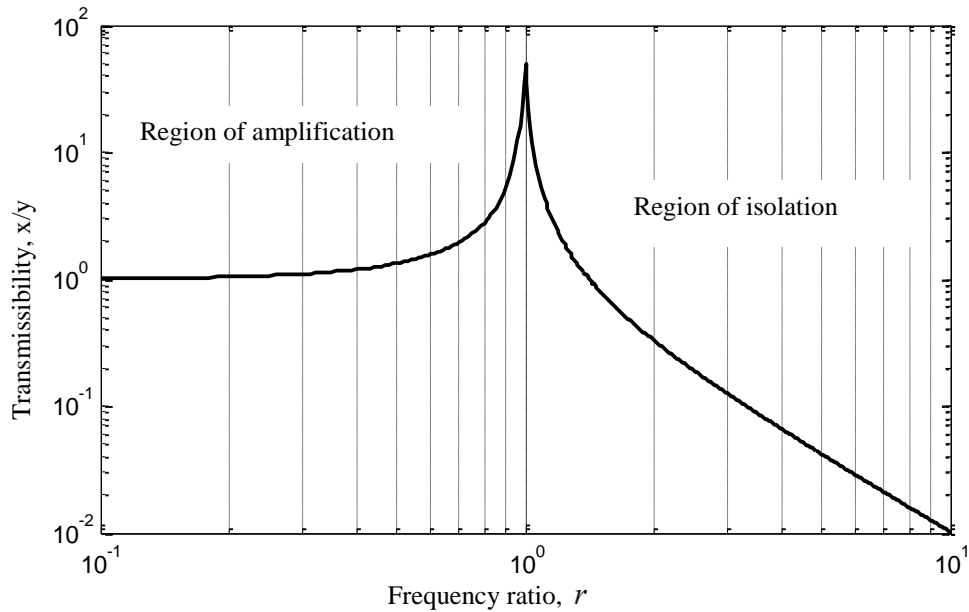


Figure 2.6: Theoretical transmissibility for an isolator

It is widely understood that reducing the stiffness of isolators/mounts and increasing the mass of the part to be isolated are two ways to increase the efficiency of isolator. Rubber mounts have been used as isolator for engine vibration reduction purposes (Choi et al., 2008; Tewari and Dewangan, 2009) and can be made to have improved adhesive and extended temperature range (Miller and Ahmadian, 1992).

The performance of isolator/rubber mount is determined by its dynamic properties which exhibit nonlinear behaviour. Experimental methods have been developed to estimate rubber mount dynamic properties (Dickens and Norwood, 2001; Mallik et al., 1999; Ooi and Ripin, 2010; Ooi and Ripin, 2011; Tian et al., 2005). The biggest challenge to design rubber mounts is to make them statically stiff and dynamically soft (Ahn et al., 2003) However, rubber mounts that are too soft will cause large static displacement (Yunhe et al., 2001). This can be solved by providing mechanical stop or by installing rubber mounts in parallel.

Andersson (1990) invented a vibration-damping handle (Andersson, 1986) comprising of both damping and rubber elements showed that the handle effectively reduced the vibration transmitted to the hand. Nagashima & Akira (1990) investigated a cylindrical vibration-isolating attachment means for a handle of a chain saw. However, this design principle is not suitable for the handgrip and the rubber element that cannot separate with enough space to provide an effective damping along the hand grip. Tewari and Dewangan (2009) have shown that the frequency-unweighted and frequency-weighted rms accelerations were reduced by 50.9% and 29.8% respectively by installing vibration isolators on the engine and the handle of hand tractor. Subjective rating on work-related body pain indicated the reduction in the range of 32% to 61% for different parts of the hand-arm. Sam and Kathirvel (2009) investigated the effect of installing three stage vibration isolators for engine, handle bar and handle of hand tractor, resulted in 50-60% reduction of handle vibration in stationary condition. However, both studies did not show how the isolators affected the handle-isolator transmissibility since vibration transmissibility is one of the most important aspects to characterize the effectiveness of vibration isolation system (Tu and Zheng, 2007).

2.4.3 Tuned vibration absorber (TVA)

Another feasible technique is to channel the vibration energy with tuned vibration absorber (TVA). The TVA is an additional mass-spring system, which is appropriately chosen to suppress the steady state force acting on particular degree of freedom system. Consider the single-degree-of-freedom (SDOF) primary system (Figure 2.7a) consisting mass (m_p), spring (k_p), damper (c_p) subjected to harmonic excitation $f(t) = F_0 \sin \omega t$. The objective is to eliminate the steady-state vibration.

This is implemented by attaching additional SDOF system with mass (m_s), spring (k_s), damper (c_s) to the primary system (Fig. 2.7b).

The first TVA has no damper ($c_s=0$) which was originally invented by Frahm. He registered a US patent in 1909 for a ‘Device of Damping Vibration of Body’. It was used to control vibration at single frequency, predominant at the resonance frequency of the undamped primary system ($c_p=0$) (Yang et al., 2011).

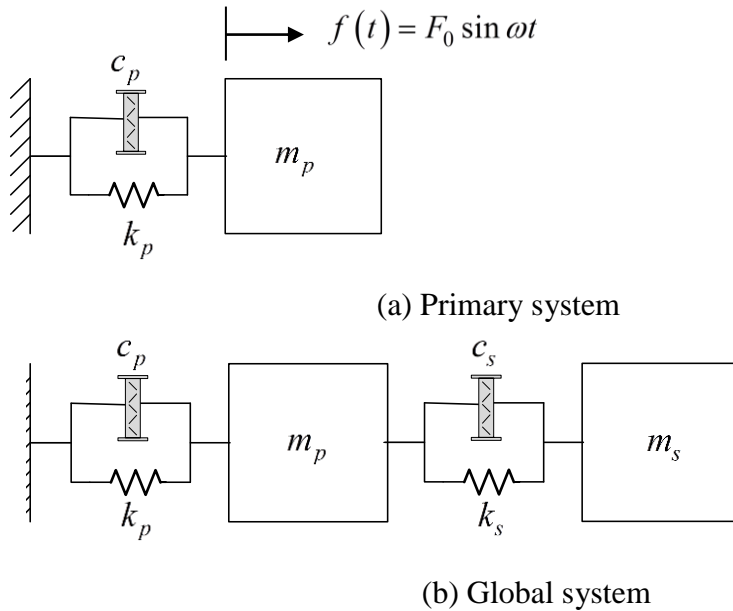


Figure 2.7: Concept of tuned vibration absorber

In the design of traditional undamped TVA, the natural frequency of TVA is selected as $\omega_s = \omega$ or $\alpha = r$ such that the steady-state response of the primary system (Eq. 2.6) becomes zero at the excitation frequency, ω (Figure 2.8). However, this traditional has a narrow operation region and the performance deteriorates significantly when the forcing frequency varies.

$$\left| \frac{X_p}{F_0/k_p} \right| = \left| \frac{\alpha^2 - r^2}{(1-r^2)(\alpha^2 - r^2) - \mu\alpha^2 r^2} \right| \quad (2.6)$$

where

Forced frequency ratio, $r = \omega/\omega_p$

Frequency ratio, $\alpha = \omega_s/\omega_p$

Mass ratio, $\mu = m_s/m_p$

Natural frequency of primary system, $\omega_p = \sqrt{k_p/m_p}$

Natural frequency of secondary system, $\omega_s = \sqrt{k_s/m_s}$

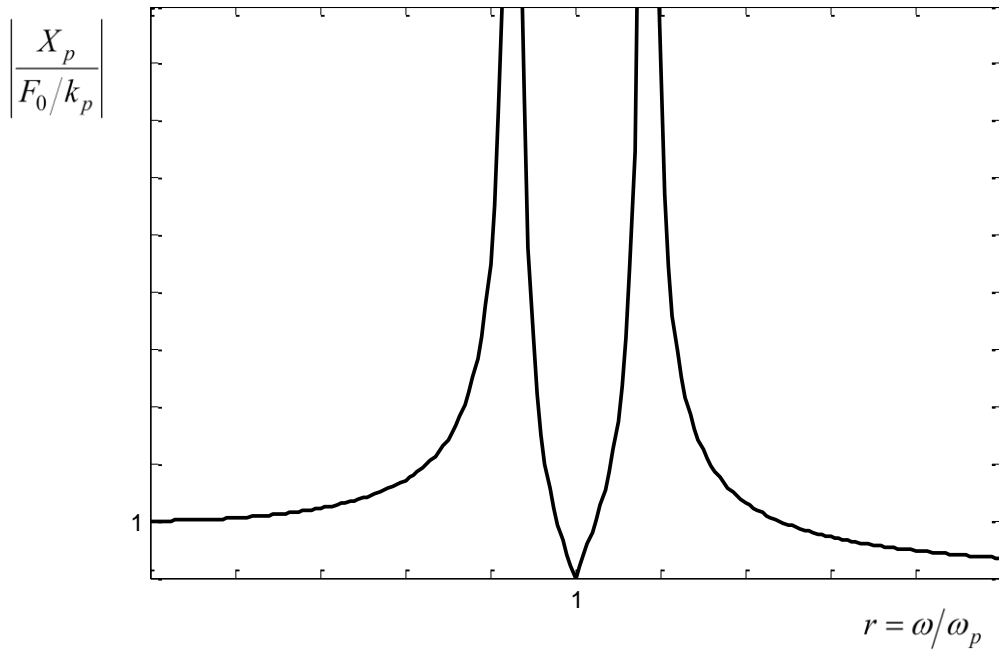


Figure 2.8: The narrow band reduction of undamped TVA

The narrow band performance can be improved by introducing a damper in TVA design. The first classical paper on the damped TVA was presented by Ormondroyd and Den Hartog (1928). They pointed out that the damping of the TVA had an optimum value for the minimization of the resonance amplitude magnification factor of a SDOF undamped primary system. Later, Brock (1946) derived the optimum damping of the traditional TVA. This optimum design method of the TVA is based on the well-known fixed-points theory (Den Hartog, 1947). In this

circumstance, a good trade-off between the suppressed original peak and the two newly emerged coupled peaks induced by the insertion of the TVA is essential to obtain a global vibration reduction within the frequency band of interest.

Since this pioneering approach, variety of TVA optimization work was appeared to suit different applications. Asami et al. (2002) demonstrated an analytical series solution for optimum TVA when attached to damped SDOF system. Ren (2001) designed a variant design of TVA which reported having better vibration suppression performance. Many papers model the primary system as damped SDOF system (Asami et al., 2002; Ren, 2001; Wong and Cheung, 2008). However for practical purposes, the primary structures would always have to be treated as multiple-degrees-of freedom (MDOF) systems or continuous models that have multiple vibration modes and resonant frequencies.

In order to deal with the multiple modes vibration, many studies have focused on the concept of TVA mounted on the MDOF system or continuous model (Brennan and Dayou, 2000; Cheung and Wong, 2008; Dayou, 2006; Thompson, 2007; Zuo and Nayfeh, 2004). For MDOF or continuous systems, the tuning process becomes more delicate as the TVA is coupled with all structural modes of the host system (Yang et al., 2011). Young was the first to utilize the TVA in the vibration control at the resonance frequency of a continuous system in 1952 (Yang et al., 2011). Snowdon (1966) derived optimum value of TVA tuning and damping depending upon the location of TVA and the beam resonance of its concern. Kitis et al. (1983) employed a numerical optimization method for delivering the optimal parameters of a damped TVA attached to a 22 DOF primary systems. Esmailzadeh