THE DESIGN OF AN ADAPTIVE TUNED VIBRATION ABSORBER

AINATUL MARDIAH IBRAHIM

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

Faculty of Mechanical Engineering
Universiti Malaysia Pahang

NOVEMBER 2007

ABSTRACT

The harmonic vibration of a machine is an undesirable effect of rotating out-of balance mass within the system. It may influence the machine operation by create unwanted noise and resulting excess wear. However the vibration of the machine can be suppressed by attaching vibration absorber whose natural frequency is tuned to be equivalent to the excitation frequency of the machine. Almost vibration absorber is only effective at particular forcing frequency. A way of overcoming this problem is by designing an adaptive tuned vibration absorber which could adapt itself in response to changes in excitation frequency of machine. This could be realized by varying the stiffness of the spring since the vibration absorber can be simplified as a mass spring system. Due to its variable stiffness capability, it can now be operated at a wide range of frequency by applying a flexibility coefficient method. A practical method on varying the stiffness element is by designing a cantilever beam acting as a spring with a moveable end-mass. The linear movement of the end-mass is controlled by a linear actuator which is connected to the controller. The design requirement of the project is to design a small absorber which can be used in the laboratory and has a capability to track frequency drift approximately ± 50% of its centre frequency of 100 Hz.

ABSTRAK

Getaran merupakan fenomena yang dihasilkan akibat dari pergerakan jisim mesin semasa bekerja. Getaran ini juga akan menghasilkan bunyi yang tidak diingini dan boleh menyebabkan kelonggaran bahagian-bahagian, kerosakannya dan kegagalannya terus. Walaubagaimanapun ketidakinginan getaran ini boleh dikurangkan dengan meletakkan penyerap getaran di mana frekuensi tabiinya adalah sama dengan frekuensi ujaan yang diberikan kepada mesin. Kebanyakan mesin yang ada hanya berkesan pada satu frekuensi ujaan sahaja. Satu cara untuk mengatasi masalah ini adalah dengan mereka sebuah penyerap getaran yang boleh berfungsi pada kebanyakan frekuensi ujaan yang diberikan. Hal ini dapat direalisasikan dengan mempelbagaikan keanjalan pegas sistem memandangkan sebuah penyerap getaran boleh diringkaskan sebagai sistem jisim pegas. Berikutan keanjalan sistem dapat diubah, peyerap getaran yang direka dapat berfungsi pada frekuensi ujaan yang berbeza. Cara yang praktikal untuk mempelbagaikan keanjalan sistem ialah dengan mereka sebuah alang yang bertindak sebagai sistem pegas dengan satu jisim pemberat yang boleh bergerak sepanjang alang. Pergerakan pemberat yang selari pada alang adalah di kawal oleh satu motor lelurus. Penyerap getaran yang akan dihasilkan untuk projek ini adalah sebuah penyerap getaran kecil pada mesin yang boleh bekerja di dalam makmal dan dapat menyerap frekuensi ujaan sekitar 50 hingga 150 Hertz.

TABLE OF CONTENT

LIST OF TABLES

LIST OF FIGURES

LIST OF SYMBOLS

SYMBOL

A_1	Cross sectional area of shaft 1 (m^2)
A_2	Cross sectional area of shaft 2 (m^2)
C	Damping coefficient (Ns/m)
E_1	Elastic modulus of material 1 (GPa)
E_2	Elastic modulus of material 2 (GPa)
F	Excitation force (N)
h	Vertical spacing between shaft 1 and shaft 2 (mm)
I_1	Second moment area of cross section 1 (mm^4)
I_2	Second moment area of cross section 2 (mm^4)
j	$\sqrt{-1}$
k	Spring stiffness of the cantilever (kN/m)
k_1	Spring Stiffness of the Primary System (kN/m)
k_2	Spring Stiffness of the Secondary System T.V.A (kN/m)
L	Distance from centre of steel mass to the shaft root. (mm)
M	Steel Mass (g)
M_1	Primary System Mass (g)
M_2	Secondary System Mass (g)
r	Radius of the Shaft
y_1	Distance from the Neutral Axis to Centroid 1 (mm)
y_2	Distance from the Neutral Axis to Centroid 2 (mm)
$\overline{Y_1}$	Amplitude of the Primary System
$\overline{Y_2}$	Amplitude of the Secondary System

ω	Excitation Frequency in angular form (rad/s)
ω_2	Natural Angular Frequency of the Secondary System (rad/s)
ζ_2	Damping Ratio of the Secondary System
μ	Mass ratio
β	Frequency Ratio
T.V.A	Tuned Vibration Absorber
A.T.V.A	Adaptive Tuned Vibration Absorber

LIST OF APPENDICES

CHAPTER 1

INTRODUCTION

1.1 Introduction

The purpose of this project is to design an adaptive tuned vibration absorber which could tune itself in response to changes in excitation frequency. Since the vibration absorber can be simplified as a mass spring system, the natural frequency of the absorber can be tuned by varying the stiffness of the spring. The method of achieving variable stiffness in adaptive tuned vibration absorber is by moveable of the end mass position.

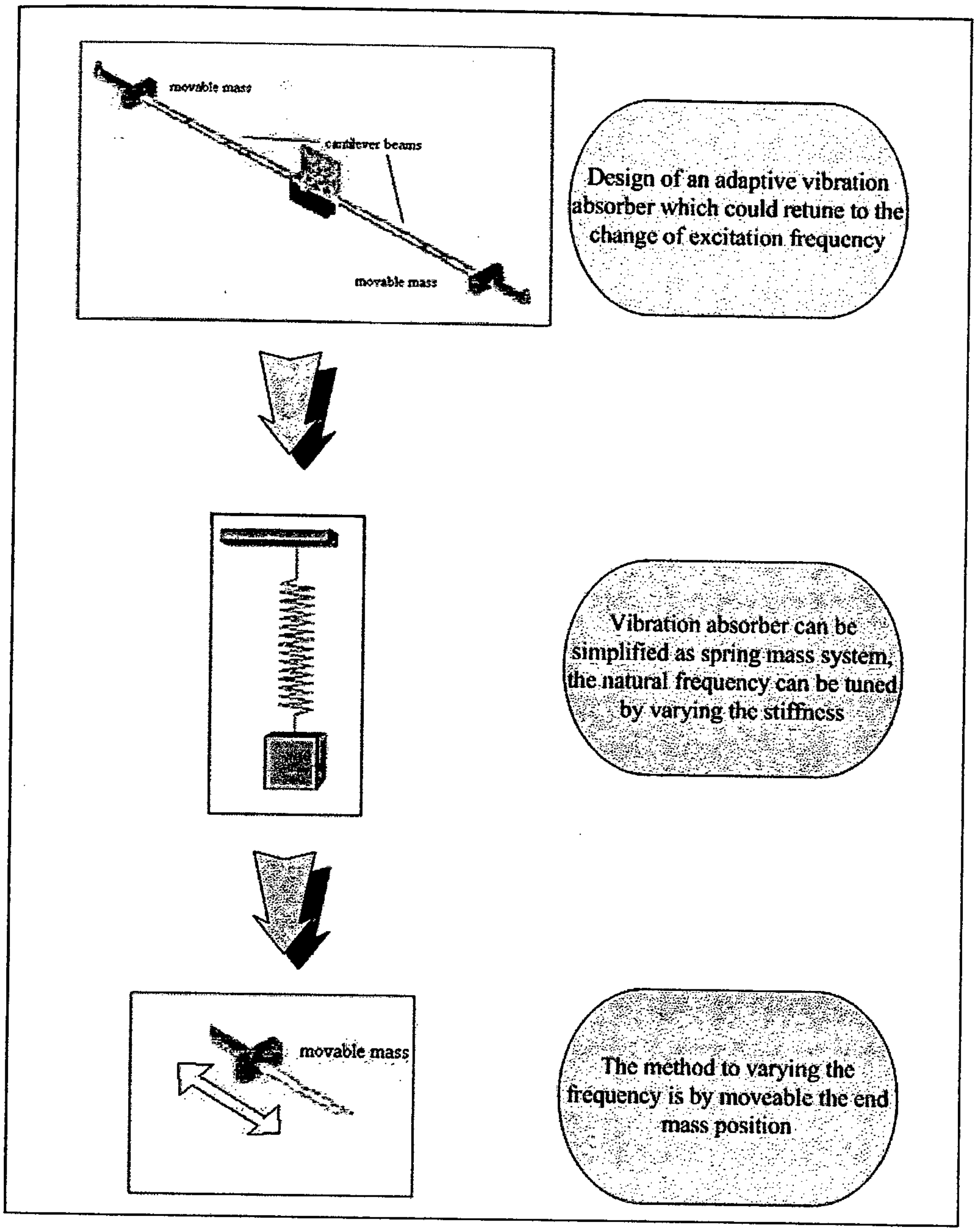

Figure 1.1 The concept of the project

Due to its variable stiffness capability, it is now capable of retuning itself in response to time-varying excitation frequency, enabling effective vibration control over a range of frequencies. A practical method on varying the stiffness element is by designing cantilevered beam acting as a spring with a moveable end mass. The tuned vibration absorber will then mounted to a vibratory primary system which is directory connected to the controller. The controller that move the end mass due to excitation frequency is not part of the project. The design requirement of this project is to design a small absorber which can be used in a laboratory and has a capability to track frequency drift approximately ±50% of its center frequency 100Hz.

1.2 Problem statement

The harmonic vibration of a machine is an undesirable effect of rotating out of balance mass within the system. However the vibration of the machine can be suppressed by attaching vibration absorber whose natural frequency is tuned to be equivalent to the excitation frequency of the machine. The conventional design of the vibration absorber is only effective at particular forcing frequency. A way of overcoming this problem is by designing an adaptive tuned vibration absorber which could retune itself in response to changes in excitation frequency.

1.3 Objectives

i) Design the cantilever beam (shaft) with the moveable end mass that can attenuate vibration of the host structure over greater frequency range

The limitation is:

The design tuned vibration absorber could practically be used in the laboratory and could attenuate the vibration with the variation of ±50% from the center frequency 100Hz of the primary system.

1.4 Scope

i. Suppress vibration due to harmonic excitation in structural (translational) system

ii. Achieving variable stiffness in adaptive T.V.A is by moveable of the end mass position.

iii. Using COSMOSWorks® software as c computational method in Finite Element Analysis.

iv. Using MATLAB® for solving mathematical equation

v. Using SOLIDWorks® to develop 3-dimensional model of ATVA.

CHAPTER 2

LITERATURE REVIEW

2.1. Overview of vibration

Vibration or oscillation can be defined as any motion that repeats itself after an interval of time. Vibration of rigid bodies can be translational, rotational or combination of two. Translational vibration refers to a point which path of vibration is straight line, and rotational vibrations refer to a rigid body whose vibration is angular for about some reference line. The vibration of a system involves the transfer of its potential energy to kinetic energy and kinetic energy to potential energy, alternately.

2.2 Description of vibration

When describing vibration, a waveform diagram is typically used. A waveform is a diagram or mathematical function that shows how the position of the vibrating point is changing with time. The most studied form of vibration is simple harmonic motion either sinusoidal or harmonic, in which the waveform is sine or cosine curve. The simplest example vibratory motion that will exist is the movement in one direction of a mass which is control led by a single spring. Such a mechanical system is called a single degree of freedom spring mass system. If the mass is displaced a certain distance from the equilibrium, but by then the mass will have some kinetic energy and will overshoot the rest position and deflect the spring in the opposite direction. It will then decelerate to stop at the other extreme of its displacement where the spring will again begin to return it toward s equilibrium. The same process repeats over and over with the energy sloshing back and forth between the spring and the mass, from kinetic energy in the mass to potential energy in the spring and back.

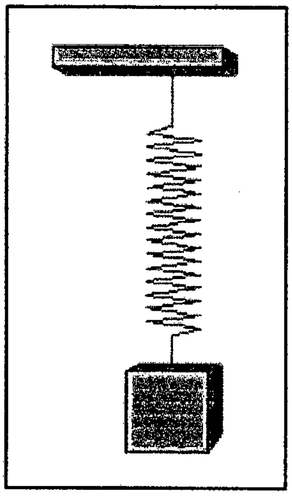

Figure 2.1 Simple harmonic motion

Mathematically, harmonic vibration of a single degree of freedom system can be given by the expression

$$u(t) = u_o \sin \omega t \tag{2.1}$$

Where;

$u(t)$ = the position of the point with respect to time t

u_o = the peak or maximum displacement (i.e. the amplitude) of the point from a datum line.

ω = the circular frequency

t =time

Graphically this waveform is represented in Figure 2.2. T is generally the notation given for the period. The period is defined as the length of time from a point on the waveform to the next point where the wave repeats itself, and the units of time typically in seconds. Its reciprocal is known as the cyclic frequency, f and is measured in cycles per second, or hertz. The relations between ω, f, T are given below:

$$T = \frac{2\pi}{\omega} \text{ sec} \tag{2.2}$$

$$f = \frac{\omega}{2\pi} \text{ cycles per second} \tag{2.3}$$

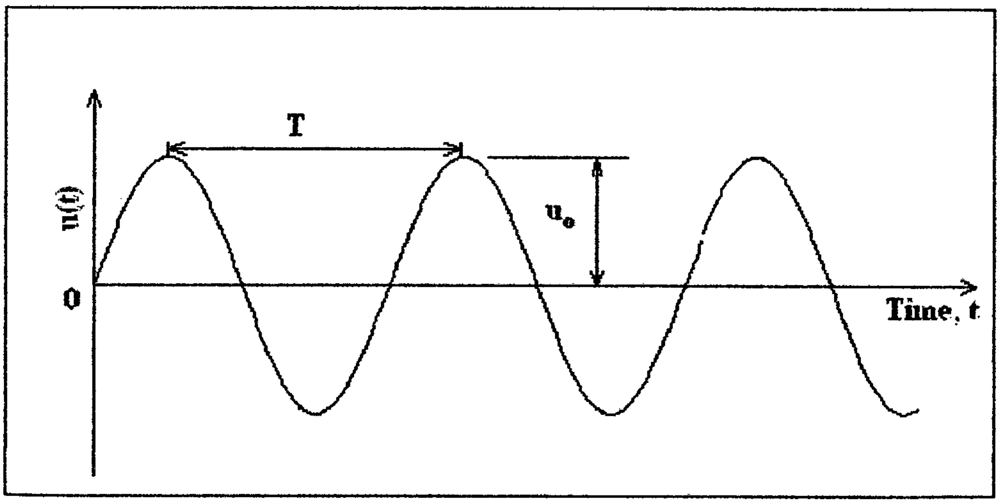

Figure 2.2 Waveform of simple harmonic motion

Expressions describing the velocity and acceleration of the vibrating point can be determined by taking the first and second derivatives of Equation (2.1) with respect to time.

Therefore;

$$\frac{du}{dt} = u_o \omega \cos \omega t \qquad \text{(Velocity)} \qquad (2.4)$$

$$\frac{d^2u}{dt^2} = -u_o \omega^2 \sin \omega t \qquad \text{(Acceleration)} \qquad (2.5)$$

From these equations, one can see that the displacement *u (t)* and the acceleration values are at a maximum when the velocity is equal to zero. This means the vibrating point momentarily comes to a stop at its maximum position from the reference datum, and then begins accelerating at its maximum rate. Harmonic motion is a form of periodic motion. Periodic motion is defined as a vibration whose waveform is repetitive. Many types of engines, compressors, pumps, and other machinery that run continuously generate a form of periodic vibration. If a motion is periodic, its velocity and acceleration are also periodic.

Another important parameter to discuss when describing vibration is damping. If a system is initially displaced a certain distance and then released, such as a pendulum, it will vibrate about a certain datum line for a finite amount of time before coming to rest. The amplitude of the motion decays, and the cause of the decay in motion, or dissipation of energy, is referred to as damping. It is present naturally, and if a system is not being forced to vibrate by an external source, its motion will eventually decay because of the intrinsic damping that is present. Damping can also be introduced into a system as a means of controlling the vibrations.

2.3 Vibration Absorber

All bodies or systems that posses both mass and elasticity are capable of undergoing vibration. People become interested to study vibration when the first musical instrument, probably whistles or drums were discovered. The Greek philosopher and mathematicians Pythagoras (582-507 B.C) is considered to be the first scientific investigation on musical instrument. Since then, people have applied critical investigation to study the phenomenon of vibration and it become the important issue especially in engineering field.

Vibration is occasionally desirable. For example the reed in a harmonica or the cone of a loudspeaker is desirable vibration, necessary for the correct functioning of the various devices. But more often, vibration is undesirable, wasting energy and creating unwanted sound. For example, the motions of engines, electric motors, or any mechanical device in operation are usually unwanted vibrations. Excessive vibration may cause undesirable effect on the host structure. A machine or system may experience excessive vibration if it is acted upon by a force whose excitation frequency nearly coincides with a natural frequency of the machine or system. In such cases, the vibration of the machine or the system can be reduced by using a vibration neutralizer or vibration absorber, which is simply another mass spring system. By adding an extra mass, spring and damper on the host structure and by calibrating its natural frequency to be equal with the excitation frequency of the host structure, the device can aid in vibration reduction. The device is called tuned vibration absorber (T.V.A).

2.4 Tunable Vibration Absorber

A tuned vibration absorber is a relatively small spring-mass oscillator that suppresses the response of a relatively large, primary spring mass oscillator at a particular frequency. The mass of the tuned vibration absorber is typically a few percent of the mass of the primary mass, but the motion of the tuned vibration absorber is allowed to be much greater than the expected motion of the primary mass. The natural frequency of the T.V.A is tuned to be the same as the frequency of excitation. T.V.A is particularly effective when the excitation frequency is close to the natural frequency of the primary system. Tuned vibration absorber has been discussed in literature since the early twentieth century. These devices are implemented to suppress the system's vibration by transferring energy to the absorber mass. The device does not work by absorbing the energy of the main structure. It rather creates a reactive force in response to the disturbance and therefore reduces the base motion at the resonance frequency. Frequently, TVA's are used to suppress running speed imbalance responses from rotating machinery. This device can be very effective solutions under two conditions:

i. The system dynamic are invariant
ii. The running speed is within the narrow frequency range.

The elastic tunable vibration absorber (TVA) is a vibration device often named after *J.P. den hartog* who first analytically described its behaviour in 1928. Den Hartog's works was preceded by *H.Frahm* who was used mass spring systems and also the oscillation of water between two tanks to counter the rolling of ships. In this case the water acted as an absorber which dissipate energy transferred from the motion of the ships. In 1968, *J.C Snowdon* presented the action of one and two TVA on the vibration of cantilever beams. However, after almost a century of development, the typical application remained the suppression of vibration in machinery. TVA has found much application for the construction of the bridges and earthquake proof buildings. In 1982, *GB Warburton* proposed a design procedure for absorber. His main interest was earthquake engineering. Recently TVA has been used more specifically for acoustic purposes and a new detuning approach was presented by *C.R Fuller et al.* in 1997. The main factor for a TVA to behave properly is to have it resonance frequency properly tuned (or "detuned") in respect to the frequency of vibration of the main structure. In general, there are two types of tuned vibration absorber which are passive and active.

2.5 Passive Tuned Vibration Absorber

The former type of passive tuned vibration is particularly suitable and very effective for a host structure which does not undergoes operational speed variations. Therefore by attaching passive T.V.A, it will only suppress the vibration at a single frequency where it has been tuned. Passive tuned vibration absorber maybe effective at attenuating low frequency noise, but are generally limited in range and effectiveness. Passive T.V.A include a suspended mass which is tuned such that the devices exhibits resonant natural frequency(fn) which generally cancels or absorbs vibration of the vibrating member at the point of thereto.

There are some disadvantageous of passive T.V.A which that;

i. There are only effective at a particular frequency or within a very narrow frequency range thereabouts.
ii. Therefore, passive TVA maybe ineffective if the primary frequency is changed and the TVA is not operating at its resonant frequency.
iii. Furthermore, passive device maybe unable to generate the proper magnitude and phasing of forces needed for effective vibration suppression and/or control.

General approach for the design of passive system has been under taken by *Juang* and *Phan,* 1992. Attempts have been made to create the absorber design problem as an output feedback problem in the context of LQ design *Stech*, 1994, or in the context of structural optimization, as a decentralized output feedback problem does not have a clear cut solution and the optimization problem posed by this formulation is a difficult one to solve. It is possible to overcome the performance limitations of the passive TVA by using active tuned vibration absorber.

2.6 Active Tuned Vibration Absorber

Active TVA, consisting of sensors, actuators and a controller offer more flexibility to the designer and have better performance, but there are more costly. Active TVA which is a stand alone active element (actuator), made to behave like a tuned absorber or an active element introduced to the make up of a traditional tuned absorber. For an example, a linear actuator placed in parallel with a spring, between the structure and the tuned mass in TVA.

A comparison of active and passive absorber was given by *Herzog*, 1994, pointing out the performance limitations of the passive absorber. *Stephen, Roach* and *Tewani* develop a theory using a damped dynamic vibration absorber with an active control element. Their work focuses on transfer function with various feedback laws. The control law implemented consisted of linear combination of the primary structure velocity and acceleration. *Seto, Sawatari* and *Takita* present work on the linear quadratic (LQ) optimal control theory applied to active TVA's. *Seto* and *Sawatari* proposed a design method for active dynamic absorber using the LQ optimum control theory. It is experimentally shown that the random response of the vibration controlled system is attenuated by five times of an uncontrolled. They use an electromagnetic voice coil actuator design, to provide additional internal forces.