DESIGN OF AN ACTIVE VIBRATION ISOLATION UNIT FOR OPTICAL APPLICATIONS

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ABSTRACT

This paper outlines work undertaken in the design of an active isolation unit for optical applications. The paper contains an overview of the proposed active isolation system including all components to be used in the device.

A significant proportion of this project has been devoted to designing a magnetic passive spring to be used in the system. The field properties of complex magnetic geometries are not widely available. For this reason, the magnetic field properties of different magnetic configurations needed to be determined experimentally. The method and results of this experimental derivation are included in this paper.

Initial results from this project indicate that the proposed device has the potential to be commercially viable. Further development of the system will be undertaken through a scholarship provided by The University of Adelaide.

KEYWORDS: Active Vibration Control, Magnetic Spring

INTRODUCTION

The transmission of ground-borne vibrations can pose significant problems in sensitive optical applications such as interferometry, holography and laser research. These areas typically require accuracy to within a fraction of a wavelength of visible light. In order to achieve such accuracy, optical instrumentation generally requires some degree of vibration isolation [1].

Traditional passive vibration control is often unsuitable for optical applications because of its characteristic resonance properties. Passive springs generally resonate in an area of the frequency spectrum at which significant levels of ground-borne vibration occur [2]. For this reason, passive spring control methods often act to amplify rather than attenuate disturbance vibrations [3].

The aim of this project is to design and construct a low cost active vibration isolation unit for use in optical applications. The following sections summarise the steps undertaken in the design of such a system and include a discussion of all experimental results obtained.

SYSTEM DESIGN OVERVIEW

It is proposed that the active vibration isolation system use an adaptive feedback control arrangement as shown in Figure 1. Feedback control avoids the problems of vibrational coupling and inadequate measurement time associated with feedforward control [4].

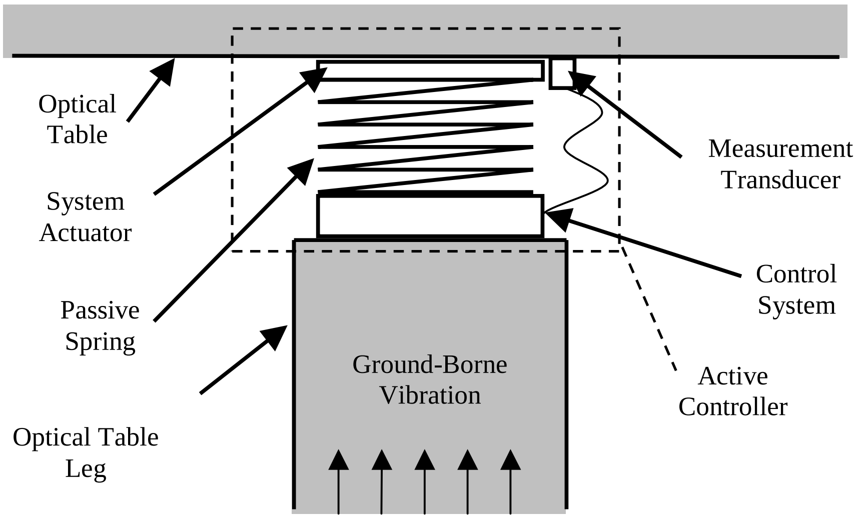

Figure 1 - Proposed Physical Layout of Control System

The system is to consist of the following components:

- An accelerometer to measure the vibration levels present in the table
- A shaker to provide an attenuating force
- A magnetic passive spring
- A control system consisting of:
    - An adaptive algorithm to allow the system to adapt to different mass loading conditions
    - A control algorithm to generate an appropriate control input signal for the shaker

The use of an appropriately designed magnetic passive spring would avoid the problems of unwanted horizontal and rotational vibration transmission associated with traditional types of passive springs (e.g. helical springs, air springs and rubber springs). This would give the spring highly idealised properties well suited to uni-directional active vibration suppression.

Magnetic springs are not currently in widespread use and literature on the equations governing their behaviour and the problems associated with their use is not readily available. There are no industrial manufacturers of magnetic spring devices and as a result the desire to use a magnetic spring has required a significant design effort and a lengthy and complicated testing period.

Preliminary design of a suitable control algorithm was undertaken using a second order model of a spring-mass-damper system in the software package *Matlab*. Practical testing and further development of the control algorithm has been undertaken on a d SPACE control board using the following components:

- An accelerometer as the measurement transducer
- A helical spring as the passive spring
- An Aura shaker as the system actuator
- A second Aura shaker as the input disturbance vibration

Initial testing used a simple PID feedback control algorithm which failed to provide any detectable attenuation. Further testing used a feedback control algorithm developed using pole placement techniques and the SISO tool in *Matlab* [5]. Simulating control with this algorithm in *Matlab* showed that the goal of 40 dB attenuation could be achieved over the required frequency range. However, practical problems arose due to the presence of anti-aliasing in the d SPACE control board. Now that this problem has been detected and methods for overcoming it identified, it is expected that further testing will allow the theoretical attenuation to be achieved in practice [4].

Stiffness and damping values for the isolator system are needed to model the active isolation unit so that a control algorithm can be properly developed. These values were obtained from analysis of a transfer function taken between the input disturbance vibration and the accelerometer. This transfer function was obtained using a frequency analyser.

EXPERIMENTAL WORK

The repulsive force between like magnetic poles can be used to remove any physical contact between two sides of a magnetic spring device. This lack of physical contact removes the direct transmission paths through which vibration could otherwise enter the system providing highly desirable vibration transmission properties. However, the field properties of complex magnetic geometries are not widely available. For this reason, the magnetic field properties of different magnetic configurations needed to be determined experimentally in order for a suitable magnetic spring device to be designed.

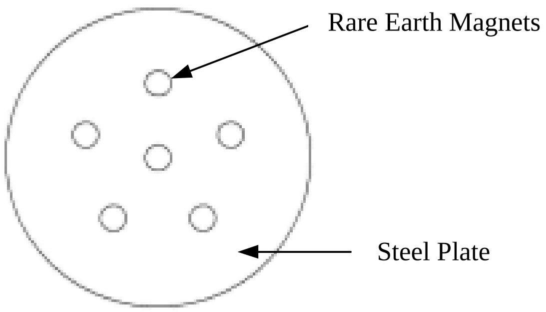

Figure 2 - Proposed Magnetic Configuration

Rare earth magnets are to be used in the passive spring due to their exceptionally high strength to size ratio and their relatively low cost [6]. It was initially proposed that a uniform magnetic field be produced across a circular plate by using a number of small rare earth magnets as shown in Figure 2. The decision to use small rare earth magnets was made because of their low cost and the belief that they would be capable of supporting the maximum required load of 375 kg (i.e. the distributed weight of the largest commercially available optical table [3]).

A magnetic field testing device (see Figure 3) was designed to allow the determination of the properties of the magnetic field produced by the proposed magnetic configuration. The magnetic properties measured were:

- The uniformity of the magnetic field produced by the magnetic configuration
- The relationship between magnetic field strength and separation distance

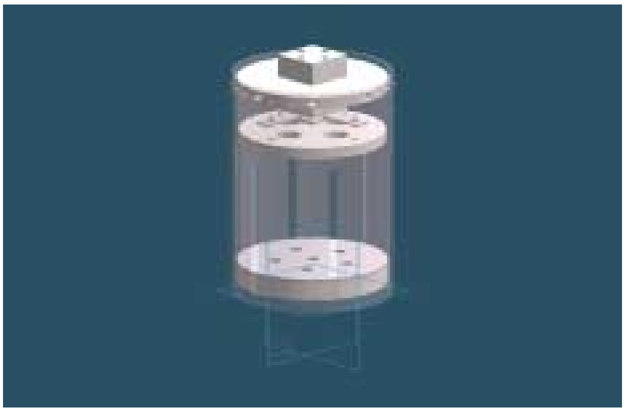

Figure 3 – Magnetic Field Testing Device

The testing device consists of two parallel circular plates which can be moved to alter their separation distance. The bottom plate houses the magnetic configuration to be tested. The upper plate, suspended by four cantilever arms, holds a small rare earth magnet (diameter 6 mm) that can be moved both radially and angularly to allow the magnetic field produced by the magnetic configuration to be tested at a variety of positions. The magnetic field strength is determined by measuring the strain in the four cantilever arms generated by the repulsive magnetic force between the magnetic configuration and the small rare earth magnet. The uniformity of the magnetic field produced by the magnetic configuration can be evaluated by comparing the strains measured in the four cantilever arms as the small rare earth magnet is moved across the suspended circular plate.

UNIFORMITY OF THE MAGNETIC FIELD PRODUCED BY THE MAGNETIC CONFIGURATION

Testing of the proposed rare earth magnetic concept revealed two problems that prohibited its use in the final passive spring design. Firstly, the magnetic field it produces is highly non-uniform, having characteristics similar to the field illustrated in Figure 4 (red colours represent north poles and deep blue colours south poles). Though the level of field non-uniformity decreased when the distance between adjacent rare earth magnets was reduced, the configuration was still unsuitable for the highly sensitive application of optical research. Secondly, the field produced by the magnetic configuration was not strong enough to support the required 375 kg load. The only solution to this problem is to increase the size of the magnets used in the configuration. It is now proposed that one large rare earth magnet (with 100 mm diameter and 40 mm thickness) be used in place of the configuration shown above. A magnet of this size should be capable of supporting the necessary load and should produce a substantially uniform magnetic field across its surface.

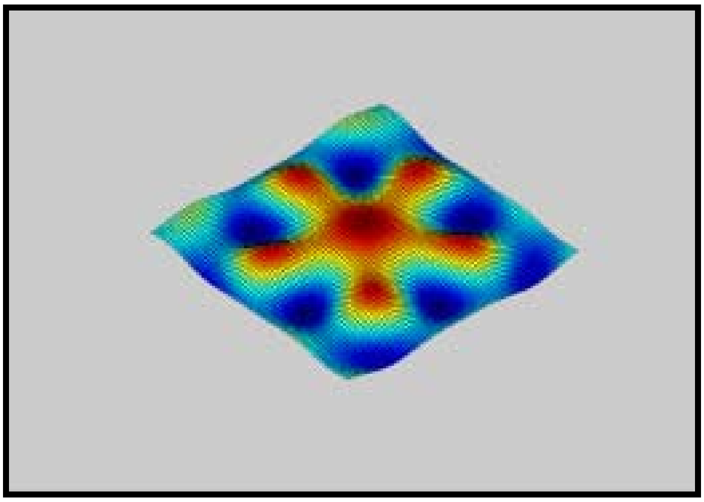

Figure 4 - Magnetic Field Produced by The Magnetic Configuration

RELATIONSHIP BETWEEN REPULSIVE FORCE AND SEPARATION DISTANCE

The testing device was used to compare the published relationship between the repulsive force and separation distance of two like magnetic poles to the actual relationship. The published relationship between magnetic force (*F*) and pole separation distance (*d*) is given by Equation 1 [7].

$$F \propto \frac{1}{d^4} \qquad (1)$$

Rare earth magnets with a diameter of 25 mm and a thickness of 12.7 mm were used to verify this relationship experimentally. Figure 5 shows the results collated from testing and a model of this relationship derived using the software package *Matlab*. The modelled relationship is given by Equation 2 below.

$$F = 1.0934 \times 10^7 \frac{1}{d^5} - 4.471 \times 10^6 \frac{1}{d^4} + 4.18 \times 10^5 \frac{1}{d^3} \qquad (2)$$

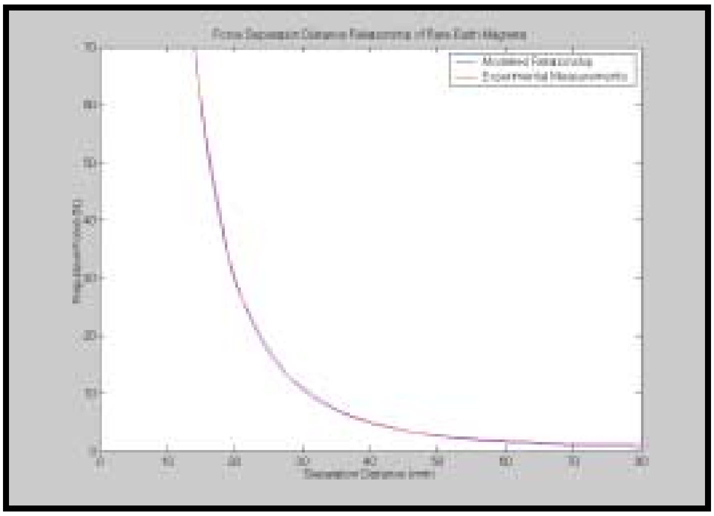

Figure 5 - Relationship Between Repulsive Force and Separation Distance for Rare Earth Magnets

The slope at any point on the curve gives the stiffness of the magnetic passive spring at this separation distance according to Hooke's Law [7]. A very soft passive spring is highly desirable as it ensures that the spring's resonance peak is at a very low frequency. This results in improved high frequency vibration attenuation and attenuation over a wider frequency band [8]. Consequently, very strong rare earth magnets need to be used such that the spring is operating in the area of the graph where the curve is virtually horizontal.

CONCLUSIONS

Uniform magnetic fields are difficult to produce using configurations of small magnets because of the complex interactions of their different fields. A uniform field will be more likely to result from the use of a single, large diameter magnet. The actual relationship between force and separation distance for two like magnetic poles has been identified experimentally allowing appropriate magnets to be selected such that a soft magnetic passive spring can be designed.

A simple PID control algorithm is unable to achieve the necessary attenuation over the required frequency range due to an insufficient number of poles and zeros. A control algorithm designed using the SISO pole placement tool in *Matlab* will allow the number of poles and zeros to be increased providing a more flexible control algorithm.

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FUTURE WORK

A further year will be devoted to the development of the active vibration attenuation device. This will be funded through a business start up scholarship provided by The University of Adelaide.

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