The Development of Economy Mechanism for Low Frequency Vibration Isolation: The Characteristic of Stiffness and Natural Frequency

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Abstract

This research describes the design and the construction of a low cost mechanism for low vibration isolation used to study and develop the method to reduce seismic vibration disturbed the scanning tunneling microscope. The prototype mechanism is designed based on principle of mass-spring-damper system in which it consists of 3 pillars and a top plate. Inside each pillar, a low cost motorcycle's inner tube and tire are employed as the air spring and internal enclosure, respectively. Evaluation of the characteristic of the developed mechanism using tested mass from 35 to 105 kg when applied air pressure of 0.20, 0.25 and 0.30 bar into the inner tubes over a period of 10 minute illustrates that the employed inner tubes have satisfied lift up level. The maximum relative displacements of the top plate for a tested mass of 105 kg while applying air pressure of 0.20, 0.25 and 0.30 bar into the inner tubes are measured of 0, 12.45 and 25.89 mm, respectively. The displacements are used to calculate the stiffness then the natural frequency of the mechanism. The results show that the average natural frequency for the mass starting from 35 to 105 kg when applied air pressure of 0.25 and 0.30 bar are of 4.14 and 3.00 Hz, respectively, are achieved.

1. Introduction

The scanning tunneling microscope (STM) [1] has several advantages such as simple structure for construction, can be used to image the sample surface in air [2] and need not complicated sample preparation so that it becomes a popular tool and widely used in the fields of surface engineering and nanometrology [3]. However, the ambient environmental conditions such as temperature and vibration during STM imaging affect directly the quality of STM image [4-6]. The fluctuation of an ambient temperature in the order of sub degree Celsius causes the enormous distortion in STM image which can yield unrecognizable atomic image. The low frequency seismic vibration passed through the base of instrument and high frequency acoustic vibration via the atmosphere also direct disturb the tunneling gap so that the quality of an STM image is reverse to the amplitude of the disturbed vibration. Air spring is becoming a popular element and can be applied in variety research fields because its performance can be adapted easier than coil spring [7-9]. However, the commercial imported air spring is still expensive and has limited model to select to meet the requirement in the design of local developed instrument. To study the feasibility and construct a mechanism for low frequency vibration isolation using the local and low cost material, a prototype mechanism using motorcycle's inner tubes as air springs is developed and reported.

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2. Design of mechanism for vibration isolation

2.1 Mathematical modeling

The mechanism in this research is designed based on mass-spring-damper system whose the mathematical model can be represented [10] as

$$m\frac{d^2y}{dt^2}+b\left(\frac{dy}{dt}-\frac{dx}{dt}\right)+k(y-x)=0 \qquad (1)$$

where

m is mass that desired to isolate the vibration (kg),

b is friction of the mechanical parts (N·s/m),

k is spring stiffness used in the mechanic (N/m),

x is input displacement which proportional to the amplitude of disturbed vibration (m),

y is displacement measured at the mass (m).

The natural angular frequency (ω_n) and damping ratio (ζ) can be expressed from (1) as

$$\omega_n=\sqrt{\frac{k}{m}} \qquad (2)$$

$$\zeta=\frac{b}{\sqrt{4km}} \qquad (3)$$

Eq. (2) illustrates that the design of a mechanism for low frequency vibration isolation to accomplish the elimination of the amplitude of the disturbed vibration, the natural angular frequency ω_n of the system should be reduced as low as possible using the smallest spring stiffness k. Eq. (3) shows that if the friction b of the vibration isolation mechanism is adjusted suitably, the mechanism will suppress the resonant displacement when the frequency of the disturbed vibration equal the natural angular frequency ω_n. Therefore, the study of the stiffness of the mechanism provides the estimation of parameters in (2) and (3).

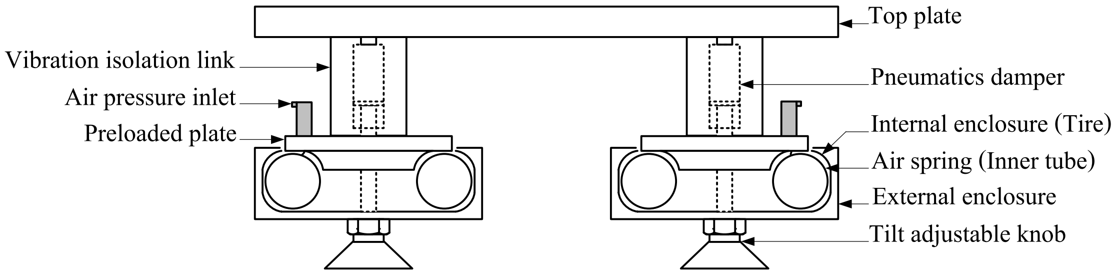

Figure 1 The schematic diagram of a prototype mechanism for vibration isolation (the third leg is not shown in the picture).

2.2 Structure of the economy mechanism

As the schematic diagram illustrated in Fig. (1), the prototype mechanism for low frequency vibration isolation is designed using CATIA V5 R16. It consists of tree legs and a top plate. The top plate is removable for future use with the closed surface capsule we have developed [11, 12] in previous report. Each leg consists of a tilt adjustable knob, an external enclosure, an internal enclosure, an air spring, a pneumatics damper, a preloaded plate, an air pressure inlet and a vibration isolation link. The isolation links are used to support the top plate in which some experimental systems will be set up over this plate. In this study, the pneumatics dampers are disassembled. The motorcycle's inner tube DEESTONE 300/350-8 and tire DEESTONE 3.50-8 4P.R. NYLON are employed as air springs and internal enclosures, respectively. Each tire and each inner tube are coaxial installed within an external enclosure layer by layer. To reduce the deformation of the modified internal enclosure, the metal sheet with sufficient thickness is employed to construct an external enclosure with precise dimension to use with the internal enclosure. All materials and parts used in this research can be found at the local hardware shop in Thailand, especially the inner tube which much cheaper than the commercial imported air springs we have used [9].

3. Evaluation of the economy mechanism

To verify the feasibility of using the motorcycle's inner tubes as the low cost air springs and investigate the characteristic of the developed mechanism for low

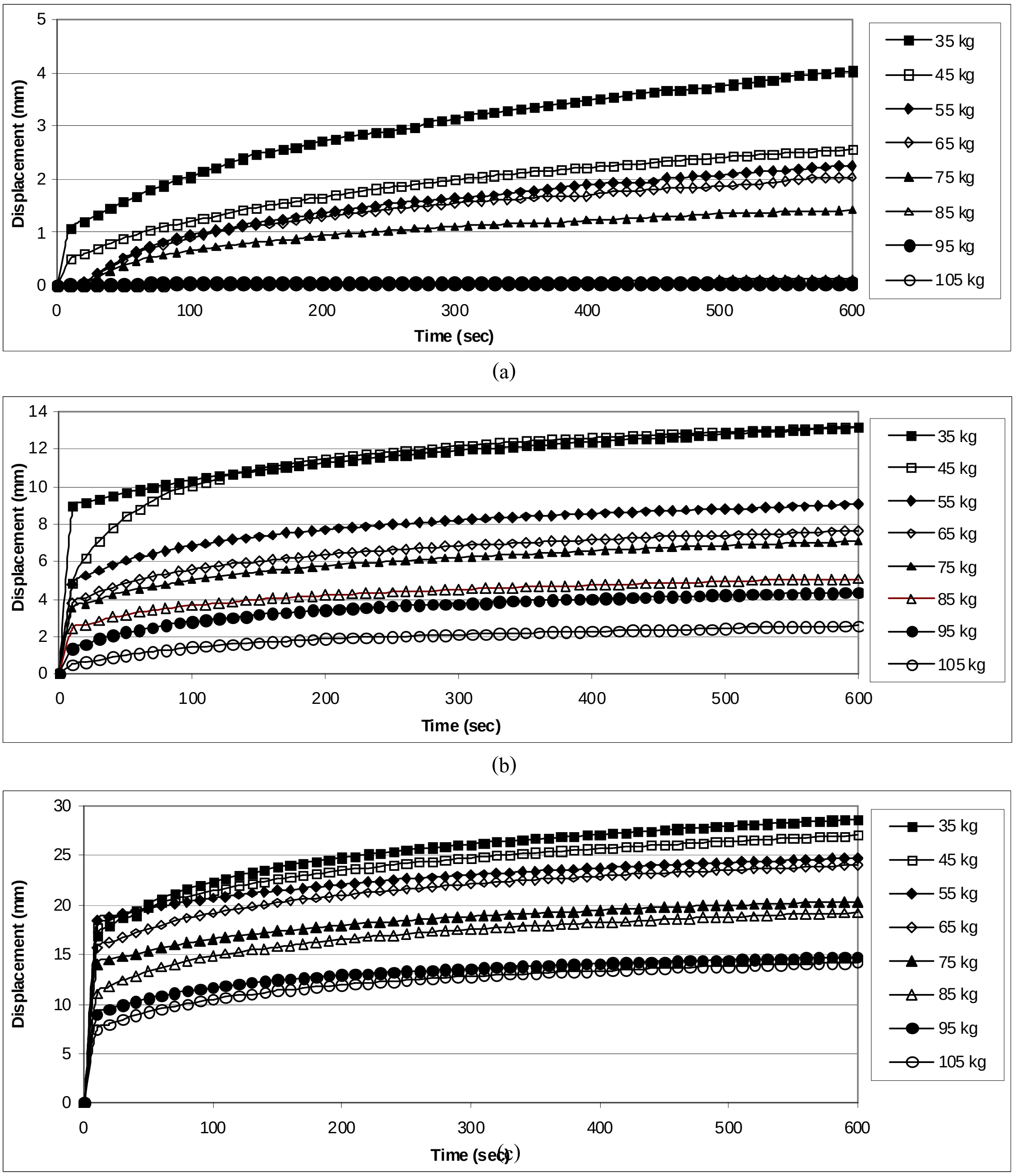

Figure 2 The displacement of the top plate with respect to the tested mass when air pressure of (a) 0.20 bar, (b) 0.25 bar and (c) 0.30 bar, respectively, are applied into the inner tube over a period of 10 min.

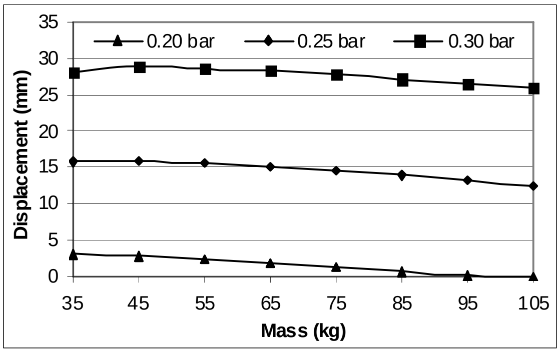

Figure 3 The relative displacement of the top plate with respect to graduate tested mass for applied air pressure of 0.20 , 0.25 and 0.30 bar.

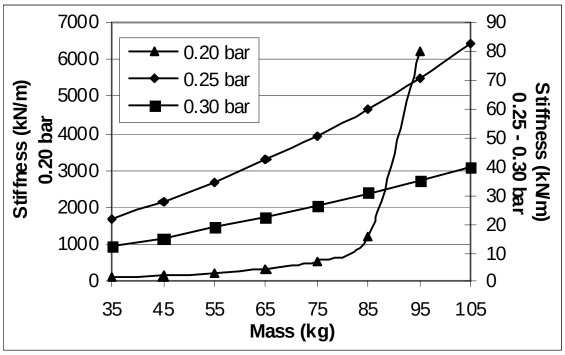

Figure 4 The stiffness of inner tubes with respect to the graduate tested mass for applied air pressure of 0.20 (left axis), 0.25 and 0.30 bar (right axis).

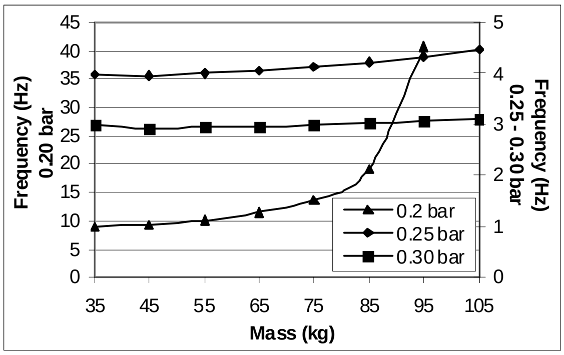

Figure 5 The natural frequency of the vibration isolation mechanism for applied air pressure of 0.20 bar (left axis), 0.25 and 0.30 bar (right axis).

frequency vibration i.e. displacement, stiffness and natural frequency, the laser triangular displacement sensor is installed underneath the top plate to study the relation between the applied air pressure, the tested mass and the displacement of the top plate lifted up by pushing force of air springs. The preliminary experimental results show that the developed mechanism with selected inner tubes work properly for the tested mass between 35 to 105 kg when the applied the air pressure is in the range of 0.20 to 0.30 bar. Fig. 2 show the displacement of the top plate when the tested mass is placed over a period of 10 min during air pressure is applied into the inner tubes. Fig. 2(a) indicates that the applied air pressure of 0.20 bar almost has no sufficient pushing force to lift up the tested mass higher than 75 kg. Fig. 2(b) and (c) show that the top plate has satisfied displacement level when air pressure of 0.25 and 0.30 bar, respectively, is applied to the inner tubes. To avoid the different initial level of the top plate, the height in which there is no air pressure applied into the inner tubes is used as a reference level in the displacement measurement for stiffness evaluation. The accumulative tested mass method at a fixed applied air pressure is performed. After the tested mass is placed on the top plate for a period of 10 min, the displacement is measured then compared to the reference level to calculate the relative displacement. The measurement procedure is repeated by increasing the tested mass for 10 kg for a period of 10 min then measure the displacement. Fig. 3 shows the relative displacements with respect to the graduate tested mass starting from 35 to 105 kg at fixed applied air pressure. For the sake of simplicity of the stiffness evaluation in sealed-spring condition, the spring stiffness resulting from change in spring effective area with change in spring height and the adiabatic spring stiffness are combined as single stiffness. The stiffness calculated using the tested mass and the relative displacement are shown in Fig. 4. For the air pressure of 0.2 bar, the stiffness at the tested mass of 105 kg can not be calculated due to there is no relative displacement. As shown in Fig. 5, the evaluated stiffness of the inner tubes is used to calculate the natural frequency of the prototype mechanism for vibration isolation using Eq. (2). The minimum natural frequency for the tested mass between 35 to 105 kg at fixed applied air pressure of 0.20, 0.25 and 0.30 bar are of 8.97, 3.97 and 2.98 Hz, respectively. The minimum average natural frequency found at applied air pressure 0.30 bar of 3.00 Hz.

4. Conclusion

An economy mechanism for low frequency vibration isolation was designed based on a principle of mass-spring-damper with 3 pillars structure. In each pillar, a low cost motorcycle's inner tube, used as air spring, and a motorcycle's tire, used as internal enclosure, were coaxial installed with each other to support the top plate. The preliminary results show that the developed mechanism using the selected inner tubes as air springs has satisfied lifted up level of the top plate for the tested mass from 35 to 105 kg, especially when the applied air pressure were of 0.25 and 0.03 bar. To evaluate the stiffness of the selected inner tubes used in the prototype mechanism over the air pressure range, the accumulative tested mass method, with the same initial height as the reference level, was used to measure the displacement of the top plate. The maximum relative displacement when the tested mass of 105 kg at the applied air pressure of 0.20, 0.25 and 0.30 bar were of 0, 12.45 and 25.89 mm, respectively. The experimental results of relative displacement measurement were used to calculate the stiffness of the inner tubes. The stiffness of the prototype mechanism for the tested mass of 105 kg at the applied air pressure 0.25 and 0.30 bar were of 82.73 and 39.79 $kN \cdot m^{-1}$, respectively. Therefore, the natural frequency for tested mass of 105 kg calculated using the stiffness at applied air pressure 0.25 and 0.30 bar were of 4.47 Hz and 3.10 Hz, respectively. We can conclude that the developed economy mechanism has satisfied performance i.e. relative displacement, stiffness and natural frequency. It can be used to combine with the closed surface capsule and the temperature control system to perform the STM imaging. The characteristic of pneumatics dampers and damping ratio of the mechanism will be investigated and reported in the near future.

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