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Minimization of the residual vibrations of ultra-precision manufacturing machines via optimal placement of vibration isolators

Chinedum E. Okwudire*, Jihyun Lee

Mechatronics and Sustainability Research Laboratory, Department of Mechanical Engineering, University of Michigan, 2350 Hayward, Ann Arbor, MI 48109, United States

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ABSTRACT

Ultra-precision manufacturing (UPM) machines are used to fabricate and measure complex parts having micrometer-level features and nanometer-level tolerances/surface finishes. Consequently, low-frequency residual vibrations that occur during the motion of the machines' axes must be mitigated. A long-standing rule of thumb in vibration isolation system design is to locate the isolators in such a way that all vibration modes are decoupled. This paper uses the 2D dynamics of a passively isolated system to show that coupling the vibration modes of the isolated system by altering the location of the isolators provides conditions which allow for the drastic reduction of residual vibrations. An objective function which minimizes residual vibration energy is defined. Perturbation analyses of the objective function reveal that the recommended practice of decoupling the vibration modes more often than not leads to sub-optimal results in terms of residual vibration reduction. The analyses also provide guidelines for correctly locating the isolators so as to reduce residual vibrations. Simulations and experiments conducted on a passively isolated ultra-precision machine tool are used to validate the findings of the paper; a 5-fold reduction of the dominant residual vibrations of the machine tool is achieved without sacrificing vibration isolation quality (i.e., transmissibility).

1. Introduction

Ultra-precision manufacturing (UPM) machines are designed to fabricate and measure complex parts having micrometer-level features and nanometer-level tolerances/surface finishes [1]. They enable the production of micro devices that are largely responsible for the advances in the electronics, biomedical, communications and other cutting-edge industries. Examples of UPM machines include ultra-precision machine tools, wafer steppers and micro CMMs, to name a few.

Due to stringent accuracy requirements, UPM machines must be properly isolated from floor vibrations. Passive isolators provide an energy-neutral, reliable and cost-effective means for vibration isolation [2–4]. A major problem with the soft mounting provided by passive vibration isolators is that it causes undesirable residual vibrations. Typically, residual vibrations occur in the form of low-frequency rocking motions of the isolated machine due to internal and external excitations – most prominent of which are the inertial reactions induced by moving machine components [2,3,5]. Residual vibrations must be minimized because they degrade the achievable accuracy and speed of UPM machines [2–8].

A long-standing rule of thumb in isolation system design, widely promoted in academic literature and industrial practice, is to decouple all vibration modes by aligning the isolator mounting locations with the center of gravity of the isolated machine [4,9–13]. There are two important reasons for this rule. First, decoupling ensures that vertical ground motions are not transmitted to the typically more-sensitive horizontal axes of the machine. This is because, generally, vertical ground motions are about twice as severe as horizontal ground motions [3,4]. Secondly, decoupling ensures that two or more resonance peaks are not created in the transmissibility response of the machine, thereby reducing the rate of attenuation after the first resonance [12]. Rivin [3] however points out that the proper selection of the isolator mounting points as a means of reducing residual vibrations is an issue often ignored by designers. DeBra [2] makes a similar observation based on an ultra-precision diamond turning machine whose residual vibrations were seen to reduce after its center of gravity was raised relative to its isolator support point. However, neither DeBra [2] nor Rivin [3] provide analytical explanations for the effect of isolator locations in reducing residual vibrations.

This paper demonstrates analytically that coupling the vibration modes of passively isolated machines, by altering the location of the isolators, provides conditions that allow for the drastic reduction of residual vibrations compared to the decoupled system. Furthermore, it proffers analysis-based guidelines for selecting the location

* Corresponding author. Tel.: +1 734 647 1531; fax: +1 734 647 9379.
E-mail address: okwudire@umich.edu (C.E. Okwudire).

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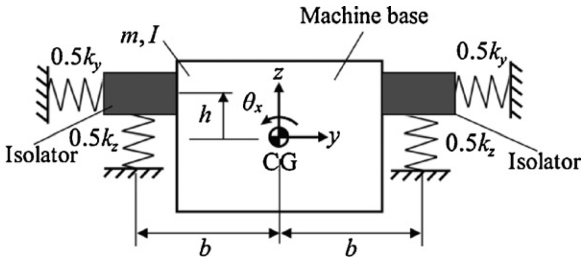

Fig. 1. Planar model of isolated machine.

of the isolators such that residual vibrations are reduced using mode coupling. The outline of the paper is as follows: In Section 2, the effects of isolator location on the planar (2D) dynamics of a simple isolated system are studied using a mathematical model. An objective function which minimizes residual vibration energy is then defined in Section 3 and used, in conjunction with a perturbation method, to analytically demonstrate the sub-optimality of decoupling vibration modes, in most situations. Moreover, practical design guidelines for judiciously placing vibration isolators to achieve residual vibration reduction are deduced. Finally, in Section 4, simulations and experiments are conducted on an ultra-precision five-axis machine tool, followed by discussions and conclusions.

2. Modeling and analysis

2.1. Modeling

Fig. 1 shows a 2D model of an isolated machine. m and I are respectively the mass and centroidal moment of inertia of the machine base about the x-axis. k_y and k_z are the combined stiffness of the isolators in the y and z directions, respectively. b is the half-span of the isolators while h is the vertical height of the isolator mounting point, measured from the center of gravity (CG) of the base. The vibrations of the machine base are assumed to occur only in the y–z plane. Such planar analyses can be applied, for instance, when the machine's structure is symmetrical in the x-direction. In a general sense, the dynamics of passive (typically pneumatic) isolators is nonlinear [14,15]. However, for small vibratory motions, linear models are adequate [14].

Assuming, for theoretical convenience, that the system in Fig. 1 is proportionally damped, its equation of motion is given by

$$\mathrm{M}\ddot{\mathrm{u}} + \mathrm{K}\mathrm{u} = 0 \tag{1}$$

M, **K** and **u** are respectively the mass matrix, stiffness matrix and vector of displacements of the system. They are given by

$$\mathrm{M} = \begin{bmatrix} m & 0 & 0 \\ 0 & m & 0 \\ 0 & 0 & I \end{bmatrix}; \quad \mathbf{K} = \begin{bmatrix} k_y & 0 & -k_y h \\ 0 & k_z & 0 \\ -k_y h & 0 & k_\theta + k_y h^2 \end{bmatrix}; \quad \mathbf{u} = \begin{Bmatrix} y \\ z \\ \theta_x \end{Bmatrix} \tag{2}$$

where $k_\theta = b^2 k_z$. Note that in Eq. (1), the damping of the system is not explicitly considered because a proportionally damped system can first be analyzed as an undamped system after which damping can be directly introduced into the modes [16].

As can be inferred from Eq. (2), only the dynamics in the y and θ_x directions are coupled as a result of h. Since the main purpose of this paper is to study the effects of h on the dynamics of the isolated system, we focus on the coupled subsystem,

$$\mathrm{M} = \begin{bmatrix} m & 0 \\ 0 & I \end{bmatrix}; \quad \mathbf{K} = \begin{bmatrix} k_y & -k_y h \\ -k_y h & k_\theta + k_y h^2 \end{bmatrix}; \quad \mathbf{u} = \begin{Bmatrix} y \\ \theta_x \end{Bmatrix} \tag{3}$$

Eq. (3) gives rise to two vibration modes, described by the coordinates y and θ_x, whose behaviors as h is varied need to be analyzed.

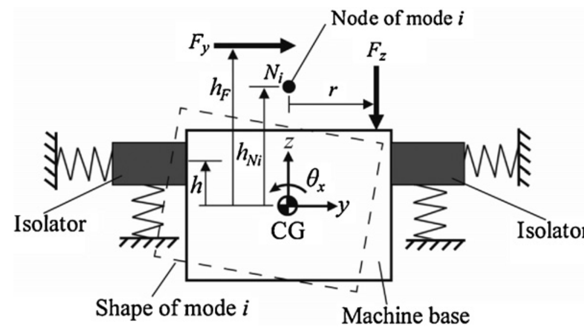

Fig. 2. Mode shape of isolated machine.

2.2. Analysis

Fig. 2 shows the shape of a rocking vibration mode i (where $i = 1, 2$) of the isolated machine about its node N_i. The node represents the point about which the isolated system rotates when vibrating in a particular mode. F_y and F_z are the inertial forces due to masses moving on the isolated system in the y and z directions, respectively. h_F and r respectively indicate the perpendicular distances of F_y and F_z from the CG. h_{Ni} is the height of N_i relative to the CG. As observed from the figure, the residual (rocking) vibrations of the isolated machine can be caused by the moments created by F_y or F_z about any node N_i of the isolated system. The influence of F_y depends on its moment arm $h_F - h_{Ni}$ about N_i, and can be represented by the frequency response function (FRF) between F_y and the angular acceleration $\ddot{\theta}_x$ of the base; i.e.

$$H_y(\omega) = \frac{-\omega^2\theta_x}{F_y} = \sum_{i=1}^{2} \frac{h_F - h_{Ni}}{I_{qi}} \frac{\omega^2}{-\omega^2 + 2j\omega\zeta_i\omega_{ni} + \omega_{ni}^2} \tag{4}$$

Similarly, the residual vibrations caused by F_z can be described by the FRF

$$H_z(\omega) = \frac{-\omega^2\theta_x}{F_z} = \sum_{i=1}^{2} \frac{r}{I_{qi}} \frac{\omega^2}{-\omega^2 + 2j\omega\zeta_i\omega_{ni} + \omega_{ni}^2} \tag{5}$$

In Eqs. (4) and (5), ω_{ni} and I_{qi} respectively denote the natural frequency and modal inertia of each vibration mode i, while ω represents the excitation frequencies; ζ_i is the modal damping added to account for the proportional damping of the isolated system which was ignored in Eq. (1) and j is the unit imaginary number. To understand how H_y and H_z are affected by mode coupling, the variation of ω_{ni}, h_{Ni} and I_{qi} as functions of h is studied. Ideally, ζ_i would also change as a function of h but we assume that the proportional damping is defined such that the modal damping ratios remain constant as h is varied.

To facilitate the analysis, a non-dimensional natural frequency for mode i (i.e., $\tilde{\omega}_{ni}$) is defined as

$$\tilde{\omega}_{ni} \triangleq \frac{\omega_{ni}}{\sqrt{k_y/m}} = \sqrt{\frac{1 + \varepsilon^2 + \tilde{h}^2 \mp \sqrt{(1 + \varepsilon^2 + \tilde{h}^2)^2 - 4\varepsilon^2}}{2}} \tag{6}$$

where $\tilde{h}$ is the non-dimensional height corresponding to h and ε is a non-dimensional variable defined as

$$\tilde{h} \triangleq \frac{h}{\rho}; \quad \varepsilon \triangleq \sqrt{\frac{k_\theta m}{k_y I}}; \quad \rho \triangleq \sqrt{\frac{I}{m}} \tag{7}$$

Similarly, $\tilde{h}_{Ni}$ and $\tilde{I}_{qi}$, the non-dimensional node height and modal inertia are defined as

$$\tilde{h}_{Ni} \triangleq \frac{h_{Ni}}{\rho} = \frac{1}{\tilde{h}}(\varepsilon^2 + \tilde{h}^2 - \tilde{\omega}_{ni}^2); \quad \tilde{I}_{qi} \triangleq \frac{I_{qi}}{I} = \tilde{h}_{Ni}^2 + 1 \tag{8}$$