Vibration protection of laptop hard disk drives in harsh environmental conditions

Vibration protection of laptop hard disk drives in harsh environmental conditions

Ashkan Haji Hosseinloo

Massachusetts Institute of Technology, 77 Massachusetts Avenue, Cambridge MA

E-mail: ashkanhh@mit.edu

Abstract. Ultra-portability and compact design of laptop computers have made them more vulnerable to harsh environments. Hard disk drives (HDDs) in particular, are critical components in laptop computers whose read/write performance is severely affected by excessive vibrations. Here we take a system-level approach to design an optimal vibration isolator so as to minimize the transmitted vibration to the HDD while the laptop chassis is confined within an allowable vibration travel. The laptop is modeled as a 3-dof lumped-parameter system and the base excitation is assumed Gaussian random vibration with zero mean and uniform power spectral density over the frequency range [0 2000] Hz. The problem is cast as a constrained optimization problem with two decision variables, namely isolation frequency and damping. A combination of analytical and numerical approaches is utilized to solve the constrained optimization problem. It is shown that the optimized isolation system could reduce the transmitted root-mean-square acceleration to the HDD by a factor of over four compared to a rigidly-mounted laptop. Furthermore, the methodology presented here is not case-specific and could be applied to the isolation system design of a wide range of systems.

Keywords: hard disk drive, laptop computer, random vibration, vibration isolation, optimization

1. Introduction

In the last few years, laptop computers have undergone some massive changes, starting with the sea-change shift to slimmer, more portable designs. Ultra-portability and compact designs of laptop computers and electronics in general, make them more vulnerable to harsh environments. The portability feature of these devices is severely compromised if their critical components fail to function properly in the rugged conditions.

Hard Disk Drives (HDDs) are of critical components in laptop computers whose read/write performance is severely affected by excessive vibrations. Laptops used in land, air, and sea transportation like those used in trains, cars, trucks, ships, and planes in addition to those used in industrial sites and factories among other harsh environments are subjected to different types of vibration excitation; and hence, they need to be accordingly ruggedized for a smooth performance.

Passive vibration isolation which is typically comprised of a set of tuned spring and damper, is an effective technique to isolate the object to be protected from the source excitation/disturbance so as to mitigate the undesirable effects of shock and vibration. Although design of passive vibration isolation system for a rigid object is a textbook problem [1], designing such a system for more sophisticated systems with sensitive internal components is not a trivial problem and need to be attended in a special manner. Examples could be drawn from different applications such as designing isolation system for printed circuit boards[2, 3], infrared equipment[4, 5, 6], hand-held percussion systems[7, 8, 9], and jet impingement cooling systems[10, 11, 12]. In all of the said references, sensitive internal components play a crucial role in designing their isolation systems.

In the last decade, a large body of research has focused on dynamic characteristics of HDDs and the techniques to improve them. However, they have mainly investigated this from a component-level perspective. Jayson et al.[13] studied the effect of air bearing stiffness on the HDD response to the external shock and vibration while Lim et al.[14] investigated the effect of design parameters of a 2.5 inch HDD including base, cover, and ramp stiffness, and the e-block, and disk thickness on the operational shock performance. Zhu et al. [15] used topological optimization to improve HDD suspension dynamic characteristics while Matusa et al. [16] devised, prototyped, and evaluated a flexible support mechanism for HDD to decrease the vibration disturbance at HDDs. Wang et al.[17] investigated the design and optimization of a collocated dual stage suspensions to improve the lateral deflection of the read/write element while keeping the resonance frequency of the design as high as possible. Harmoko et al. [18] studied the design of an isolation system for a HDD subjected to shock and random vibration based on military standards of MIL-STD-810E.

More recently, researchers started to investigate the vibration protection of HDDs from a system-level perspective. The system-level modeling and analysis enables one to consider the effects of dynamic coupling between the HDD and the other components of a computer such as the computer chassis, and hence, results in more realistic and effective vibration protection designs. Lim et al. [19] and Park et al. [20] took a 2-dof lumped model representing a laptop with its HDD with a nonlinear stiffening rubber mount for the HDD in addition to a stiff bumper. They used six different rubber mounts with different damping and stiffness characteristics to protect the HDD against shock and vibration. More recently, Lai et al. [21] developed a 4-dof model of a laptop computer to study the vibration transmission from the laptop speaker to the HDD in order to isolate the external disturbance through optimization of chassis structure. They conducted a rigorous analytical study backed up by finite element simulations. In another study, Alavi et al. [22] took a 2-dof model of a laptop-HDD and optimized stiffness and damping of the laptop and HDD mounts to minimize the HDD acceleration in response to random vibration. They used modified constrained steepest descent algorithm for optimization.

In a similar study, Rahmati et al. [23] used genetic algorithm to optimize the stiffness and damping of the laptop and HDD mounts in order to concurrently minimize two objective functions, namely the root-mean-square acceleration transmitted to the HDD during random excitation and the peak acceleration experienced by the HDD during shock excitation.

In this study, we adopt a 3-dof lumped model representing a flexible laptop chassis containing an HDD, and we study the vibration transmission from the base to the HDD. The problem is then formulated as optimizing the laptop isolation mounts to minimize the root-mean-square (RMS) acceleration of the HDD constrained by a given vibration travel (rattle space) of the laptop chassis when the whole system is based-excited by random vibration.

2. Mathematical modeling

2.1. 3-dof lumped model of a laptop computer

In order to investigate the problem from a system-level perspective for a more realistic analysis, we consider a 3-dof lumped model of a laptop as illustrated in Fig.1. This model is adopted and modified from Ref.[21]. The chassis is modeled as a flexible structure with two degrees of freedom x_{c_1} and x_{c_2} with corresponding masses of m_{c_1} and m_{c_2}, respectively. The structural stiffness and damping coefficient are denoted by k_c and c_c, respectively. HDD is characterized with the mass m_h and its mounting stiffness and damping coefficient of k_h and c_h, taking the third degree of freedom x_h. Both parts of the notebook chassis are supported by identical isolation mounts with stiffness k_m and damping coefficient c_m. The laptop chassis is based-excited which is designated by x_b.

Governing dynamic equations of the system in Fig. 1 could be written in a compact matrix form as follows:

$$\mathbf{M}\ddot{\mathbf{x}} + \mathbf{C}\dot{\mathbf{x}} + \mathbf{K}\mathbf{x} = \mathbf{F}(t), \tag{1}$$

where, $\mathbf{x} = [x_{c_1}, x_{c_2}, x_h]^T$ is the displacement vector, $\mathbf{F}(t) = [c_m\dot{x}_b + k_m x_b, c_m\dot{x}_b + k_m x_b, 0]^T$ is the excitation force vector and $\mathbf{M} = \text{diag}[m_{c_1}, m_{c_2}, m_h]$ is the diagonal mass matrix. The damping $\mathbf{C}$, and stiffness $\mathbf{K}$ matrices are defined as:

$$\mathbf{C} = \begin{bmatrix} c_m + c_c & -c_c & 0 \\ -c_c & c_m + c_c + c_h & -c_h \\ 0 & -c_h & c_h \end{bmatrix}, \quad \mathbf{K} = \begin{bmatrix} k_m + k_c & -k_c & 0 \\ -k_c & k_m + k_c + k_h & -k_h \\ 0 & -k_h & k_h \end{bmatrix} \tag{2}$$

By applying the Laplace transform to Eq.1 and introducing the following frequency, damping and mass ratio parameters:

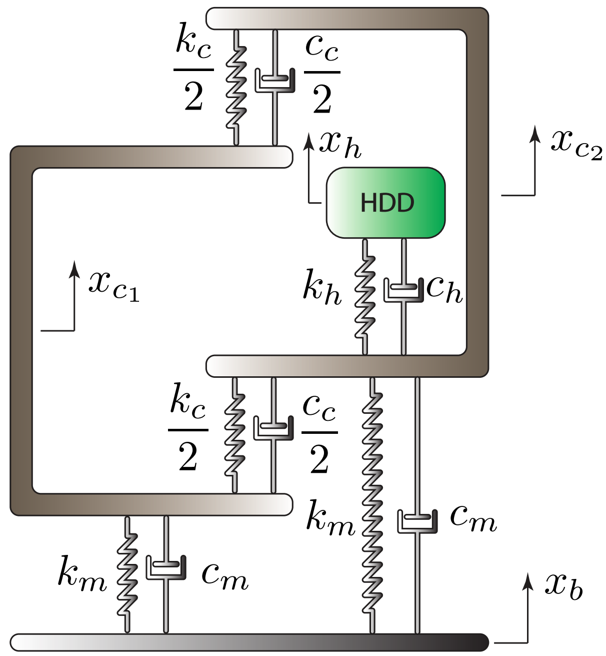

Figure 1. The 3-dof lumped-parameter model of the laptop computer

$$
\begin{aligned}
\omega_m^2 &= \frac{2k_m}{m} & \omega_c^2 &= \frac{k_c}{m_{c_2}} & \omega_h^2 &= \frac{k_h}{m_h} \\
\zeta_m &= \frac{c_m}{\sqrt{2k_m m}} & \zeta_c &= \frac{c_c}{2\sqrt{k_c m_{c_2}}} & \zeta_h &= \frac{c_h}{2\sqrt{k_h m_h}} \\
\mu_{c_1} &= \frac{m}{m_{c_1}} & \mu_{c_2} &= \frac{m}{m_{c_2}} & \mu_h &= \frac{m}{m_h},
\end{aligned}
\tag{3}
$$

where, $m = m_{c_1} + m_{c_2} + m_h$ is the total mass of the laptop, the transfer (complex transmissibility) functions from the base (x_b) to any of the three degrees of freedom could be derived by solving three scalar linear equations in the Laplace or frequency domain. The transfer functions from the base to the HDD ($H_{x_b}^{x_h}$) and to the chassis ($H_{x_b}^{x_{c_1}}$ and $H_{x_b}^{x_{c_2}}$) are derived as:

$$
\begin{aligned}
H_{x_b}^{x_h}(s) &= \frac{X_h(s)}{X_b(s)} = \frac{I(s)\Big(C(s)E(s) + A(s)G(s)\Big)}{A(s)\Big(D(s)H(s) - F(s)I(s)\Big) - B(s)H(s)E(s)} \\
H_{x_b}^{x_{c_2}}(s) &= \frac{X_{c_2}(s)}{X_b(s)} = \frac{H(s)\Big(C(s)E(s) + A(s)G(s)\Big)}{\Big(A(s)D(s) - B(s)E(s)\Big)H(s) - A(s)F(s)I(s)} \\
H_{x_b}^{x_{c_1}}(s) &= \frac{X_{c_1}(s)}{X_b(s)} = \frac{C(s) + B(s)H_{x_b}^{x_{c_2}}(s)}{A(s)},
\end{aligned}
\tag{4}
$$

where, $X_h(s)$, $X_{c_1}(s)$, and $X_{c_2}(s)$ are the Laplace transforms of $x_h(t)$, $x_{c_1}(t)$, and $x_{c_2}(t)$, respectively. The other functions are defined below:

$$\begin{aligned}
A(s) &= s^2 + \mu_{c_1}\left(\zeta_m\omega_m + \frac{2}{\mu_{c_2}}\zeta_c\omega_c\right)s + \mu_{c_1}\left(\frac{1}{2}\omega_m^2 + \frac{1}{\mu_{c_2}}\omega_c^2\right) \\
B(s) &= \frac{\mu_{c_1}}{\mu_{c_2}}\omega_c(2\zeta_c s + \omega_c), \quad C(s) = \mu_{c_1}\omega_m\left(\zeta_m s + \frac{\omega_m}{2}\right) \\
D(s) &= s^2 + \left(\mu_{c_2}\zeta_m\omega_m + 2\zeta_c\omega_c + 2\frac{\mu_{c_2}}{\mu_h}\zeta_h\omega_h\right)s + \mu_{c_2}\left(\frac{1}{2}\omega_m^2 + \omega_c^2 + \frac{1}{\mu_h}\omega_h^2\right) \qquad (5) \\
E(s) &= 2\zeta_c\omega_c s + \omega_c^2, \qquad F(s) = \frac{\mu_{c_2}}{\mu_h}\omega_h(2\zeta_h + \omega_h) \\
G(s) &= \mu_{c_2}\omega_m\left(\zeta_m s + \frac{\omega_m}{2}\right), \quad H(s) = s^2 + 2\zeta_h\omega_h s + \omega_h^2, \quad I(s) = 2\zeta_h\omega_h s + \omega_h^2.
\end{aligned}$$

In the above equations s denotes the Laplace parameter and could be replaced by $j\omega$ where $j = \sqrt{-1}$ to get the expressions in the frequency domain.

2.2. Design of optimal vibration isolation system for HDD protection

This study aims to design optimal laptop mounts to protect the HDD in harsh dynamic environment, in particular when subjected to random vibration excitation. Here, we assume the HDD mount characteristics (k_h and c_h) are fixed and are not to be optimized in this study. Nowadays, HDDs along with their mounts are purchased and integrated into the laptop chassis by the laptop manufacturers as we trend towards customizable laptops; hence, the fixed HDD mount characteristics is a valid assumption. With that said, the design analysis presented in this paper could be similarly applied to optimize the HDD mounts.

We assume that the random excitation is zero-mean, Gaussian, and stationary and has a uniform power spectral density (PSD) $S_{\ddot{x}_b} = S_0$ over the frequency range of [0 2000] Hz. This frequency range covers excitation frequency experienced in almost all applications from car transportation to random vibration experienced in maneuvering and cruising of a fighter .

The HDD performance associated with the data transfer rate between the head and the disk is characterized by the so-called position error signal (PES). PES is also a good prediction index for the track mis-registration (TMR) level [20]. It is also known that PES is directly correlated with the transmitted acceleration to the HDD [20]; therefore, here the HDD protection is based on the transmitted acceleration to the HDD as a measure of its performance.

A very soft mount usually results in good isolation but it brings about large vibration travel at the same time. Thus we would like to optimize the laptop mounts to minimize the root-mean-square (rms) acceleration of the HDD subjected to vibration travel of the laptop chassis. The optimization problem is then formulated as:

- **Optimization Problem (isolation system design:)** *Find the optimal isolation system characteristics ω_m and ζ_m (or equivalently k_m and c_m) so as to minimize* *the transmitted rms acceleration to the HDD ($\sigma_{\ddot{x}_h}$) subjected to the restrain imposed on the maximum relative deflection of the laptop chassis ($z_c^{\max}$):*

$$\min_{\omega_m,\,\zeta_m} \sigma_{\ddot{x}_h} \quad \text{subjected to} \quad z_c^{\max} \leq \Delta, \tag{6}$$

where, Δ is the maximum allowable vibration travel of the laptop chassis. Since the chassis has two parts in this model, the maximum relative deflection of the laptop chassis ($z_c^{\max}$) is the larger maximum deflection of the two, that is, $z_c^{\max} = \max\{z_{c1}^{\max}$ and $z_{c2}^{\max}\}$. $z_{c1}^{\max}$ and $z_{c1}^{\max}$ are the maximum relative defections of the two parts of the chassis i.e. $z_{c_i}^{\max} = \max\{|z_{c_i}| = |x_{c_i} - x_b|\}$ where $i = 1, 2$.

Given the input PSD, $S_{\ddot{x}_b}(\omega)$, and in view of the transfer functions for the HDD acceleration, $H_{x_b}^{x_h}(\omega)$ and the relative chassis displacement, $H_{\ddot{x}_b}^{z_{c_i}}(\omega)$, their corresponding PSD functions could be calculated as:

$$S_{\ddot{x}_h}(\omega) = \left|H_{x_b}^{x_h}(j\omega)\right|^2 S_{\ddot{x}_b}(\omega), \quad S_{z_{c_i}}(\omega) = \left|H_{\ddot{x}_b}^{z_{c_i}}(j\omega)\right|^2 S_{\ddot{x}_b}(\omega) = \frac{1}{\omega^4}\left|H_{x_b}^{x_{c_i}}(j\omega) - 1\right|^2 S_{\ddot{x}_b}(\omega). \tag{7}$$

Once the PSD functions above are calculated, the rms acceleration of the HDD and rms displacement of the relative laptop chassis deflection could be calculated as[24]:

$$\sigma_{\ddot{x}_h} = \sqrt{\int_0^\infty S_{\ddot{x}_h}(\omega)\,\mathrm{d}\omega}, \quad \sigma_{z_{c_i}} = \sqrt{\int_0^\infty S_{z_{c_i}}(\omega)\,\mathrm{d}\omega}. \tag{8}$$

For the normally-distributed random process the maximum value of the random variable could be estimated with the probability of 99.7% using the 3σ rule [3]. The maximum deflection of the chassis could be calculated in view of this rule as:

$$z_{c_i}^{\max} = 3\sigma_{z_{c_i}}. \tag{9}$$

Now optimization problem in Eq.6 could be numerically done using the Eqs.7-9. Although this could be done numerically, the problem could be further simplified by reducing the model from 3-dof to single-degree-of-freedom (sdof) only for the purpose of calculating the chassis deflection. Reducing the model order to an sdof system is a valid assumption here for two reasons. First, the HDD has much smaller mass than the chassis, and hence, its dynamics does not affect that of the chassis significantly. Secondly, the natural frequencies pertaining to the HDD and to the deflection of the two parts of the chassis relative to each other are both higher than the isolation frequency. And since the deflection is primarily dominated by the first mode (this could be seen by the quartic power of the frequency in the denominator in Eq.7), the isolation mode will be the dominant one. This will be numerically proven in the next section.

For the above-mentioned reasons, only for the purpose of the maximum vibration travel of the chassis, we model the laptop as an sdof system characterized by the laptop total mass m, and isolation stiffness and damping coefficients of $2k_m$ and $2c_m$, respectively. The chassis absolute displacement is denoted by x_c and the base excitation is designated by x_b as before. The transfer function for the relative deflection of this system could be written as:

$$H_{\ddot{x}_b}^{z_c}(s) = \frac{X_c(s) - X_b(s)}{\ddot{X}_b(s)} = \frac{-1}{\omega^2}\left(\frac{X_c(s)}{X_b(s)} - 1\right) = \frac{1}{s^2 + 2\zeta_m\omega_m s + \omega_m^2}, \tag{10}$$

where, ω_m and ζ_m are defined as in Eq.3. The rms relative deflection of the chassis could then be calculated as:

$$\sigma_{z_c} = \sqrt{\int_0^\infty S_{z_c}(\omega)\, \mathrm{d}\omega} = \sqrt{\int_0^\infty \left|H_{\ddot{x}_b}^{z_c}(j\omega)\right|^2 S_{\ddot{x}_b}(\omega)} = \sqrt{\frac{\pi S_0}{4\omega_m^3\zeta_m}}. \tag{11}$$

To derive the final expression in Eq.11 we have assumed a constant PSD for the input acceleration i.e. $S_{\ddot{x}_b}(\omega) = S_0$. The maximum vibration travel of the laptop chassis could then be calculated using the 3σ rule as:

$$z_c^{\max} = 3\sigma_{z_c}. \tag{12}$$

Combining Eqs. 11 and 12 and equating $z_c^{\max}$ with the maximum allowable vibration travel Δ, one could derive ω_m in terms of the input PSD, Δ, and the isolation damping ratio ζ_m as:

$$\omega_m = \sqrt[3]{\frac{9\pi S_0}{4\zeta_m\Delta^2}}. \tag{13}$$

Choosing ω_m based on Eq.13 guarantees that we meet the displacement constraint and it makes the optimization problem in Eq.6 substantially easier. It not only relaxes the displacement constraint in Eq.6 but it also reduces the dimension of the feasible set of the optimization from two to only one which is the damping ratio ζ_m.

3. Results and discussion

In this section, we first show that the sdof model of the laptop is sufficient for the purpose of calculating the vibration travel of the laptop for reasonable values of laptop isolation frequency. Then we present the optimization results of the isolation mount characteristics for two cases of flexible and stiff laptop chassis. We also compare the results to the case where there is no constraint on the chassis displacement.

Figure 2 depicts transmissibility curves from the base acceleration to the relative displacement of the chassis and to the absolute acceleration of the HDD for the laptop with the flexible chassis, and with arbitrary but reasonable values of the isolation system ($\omega_m = 140\pi$(rad/s) and $\zeta_m = 0.3$). All the other parameters are listed in Table 1. Based on these transmissibility curves, which are the bases for calculating the vibration travel of the chassis and acceleration of the HDD, the sdof model is sufficiently good for modeling the displacement of the chassis but does not properly model the acceleration of the HDD. This justifies the assumption of the sdof model for calculating the vibration travel of the chassis and using the 3-dof model for the acceleration of the HDD. Figure 3

Table 1: Parameters for different components of the laptop computer

Parameter	Value
k_h	$3 \times 10^5\ \frac{\mathrm{N}}{\mathrm{m}}$
c_h	$10\ \frac{\mathrm{Ns}}{\mathrm{m}}$
m_h	0.12 kg
k_c (flexible chassis)	$2.3 \times 10^6\ \frac{\mathrm{N}}{\mathrm{m}}$
k_c (stiff chassis)	$2.3 \times 10^{16}\ \frac{\mathrm{N}}{\mathrm{m}}$
c_c	$50\ \frac{\mathrm{Ns}}{\mathrm{m}}$
m_{c_1}	1.125 kg
m	1.5 kg

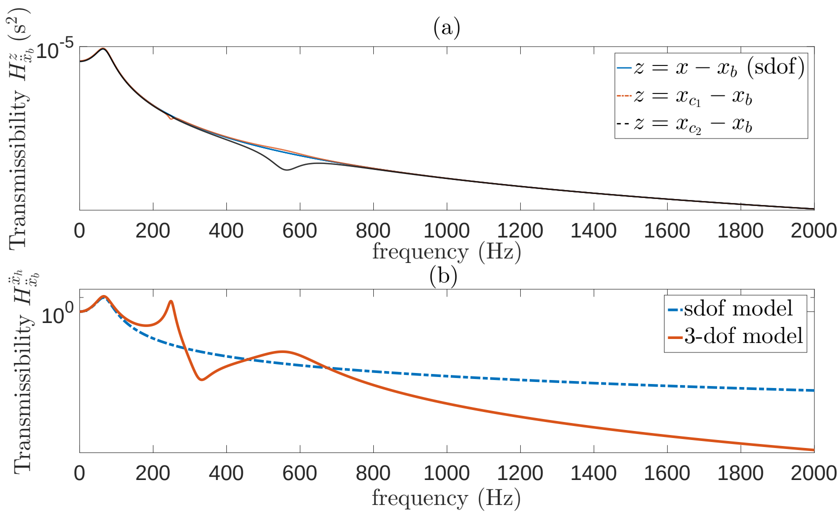

Figure 2. Transmissibility curves for (a) vibration travel of the chassis, and (b) absolute acceleration of the HDD. Flexible chassis ($k_c = 2.3 \times 10^6$) and isolation parameters $\omega_m = 140\pi$(rad/s) and $\zeta_m = 0.3$ are used here.

illustrates dependence of the HDD acceleration level on isolation system parameters ω_m and ζ_m when there is no displacement constraint on the vibration travel of the chassis (Eq.8). It shows that in general, increasing ω_m increases the acceleration level, and that for a given ω_m, there is an optimum ζ_m that minimizes the acceleration of the HDD. The unit used here for the acceleration is grms that is the root-mean-square acceleration in unit of $g = 9.81\mathrm{m/s}^2$.

Reduced-order optimization is depicted in Fig.4. As discussed in the previous section, the constrained optimization with two decision variables ω_m and ζ_m is reduced to an unconstrained optimization in only ζ_m, in view of Eq.13. Figure 4 shows the dependence of the HDD acceleration level on the isolation damping ratio ζ_m, in presence of the chassis displacement constraint. The optimum damping ratio for the flexible and stiff chassis is 0.50, and 0.38, respectively. The corresponding optimum isolation frequencies $\omega_m/2\pi$ are then calculated by Eq.13 as 70.42 Hz, and 77.5 Hz. The transmitted acceleration to the HDD for the flexible and the stiff chassis will be 6.23

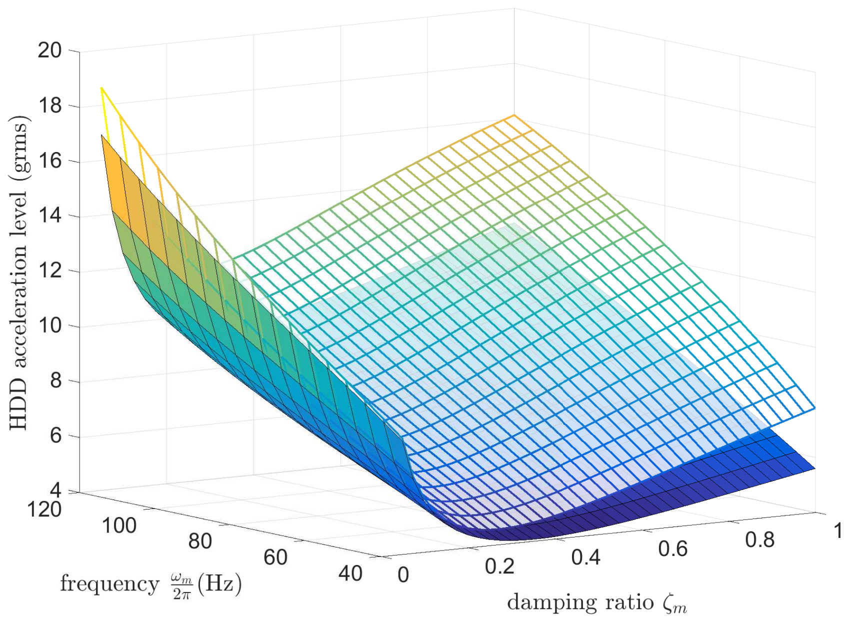

Figure 3. Dependence of the HDD rms acceleration on the isolation frequency ω_m and damping ratio ζ_m. The transparent wire-frame shows this dependence for the stiff chassis while the solid surface beneath shows it for the flexible chassis.

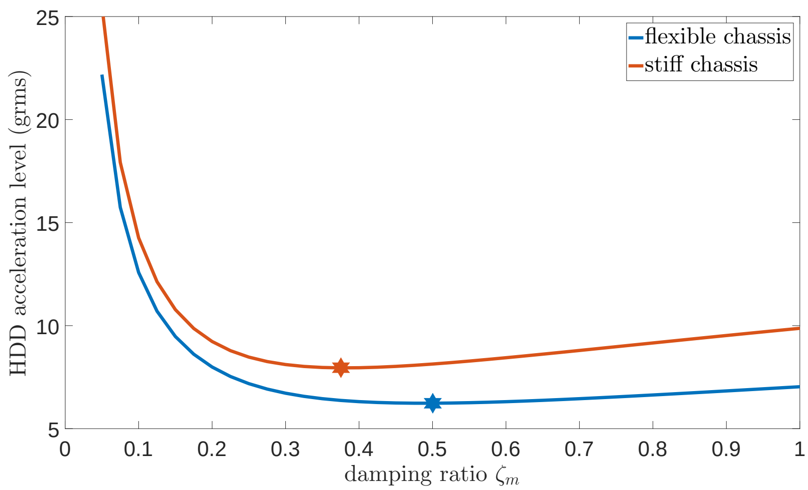

Figure 4. Dependence of the HDD acceleration level on isolation damping ratio ζ_m with optimal isolation frequency ω_m and in presence of the chassis displacement limit. Optimal points are marked with hexagrams.

grms and 7.95 grms with their optimal isolation system. This acceleration would be 27.42 grms if the laptop was rigidly mounted to the base with no isolation system. Figure 5 shows the same dependence as in Fig.4 but in the absence of the displacement constraint and for a fixed isolation frequency ($\omega_m/2\pi$) of 80 Hz. Compared to Fig.4, the HDD acceleration is more sensitive to the isolation damping in the absence of the

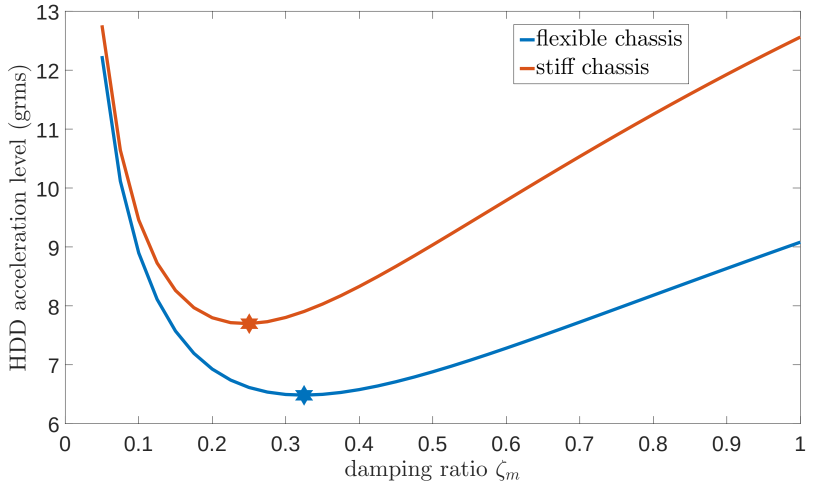

Figure 5. Dependence of the HDD acceleration level on the isolation damping ratio ζ_m at isolation frequency $(\omega_m/2\pi)$ of 80 Hz and in the absence of the chassis displacement limit. Optimal points are marked with hexagrams.

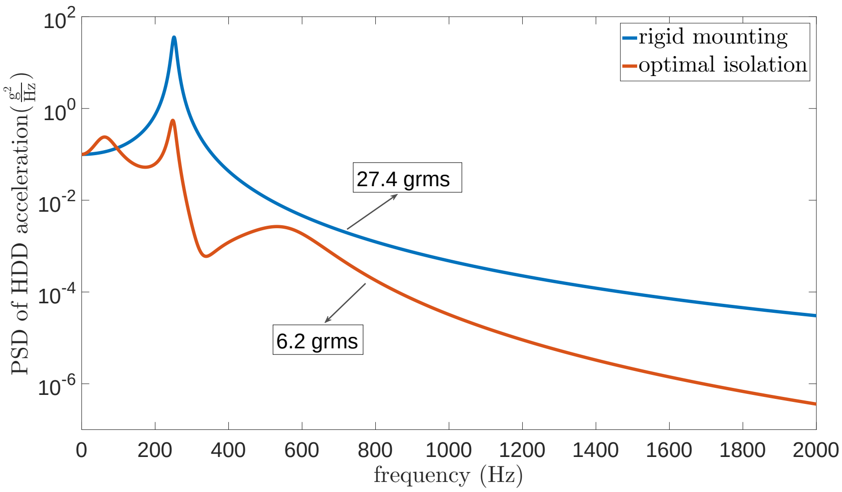

Figure 6. PSD of the HDD acceleration for optimally-isolated (with displacement constraint) and rigidly-mounted laptops. Flexible laptop chassis is considered here.

displacement constraint. To see how the isolation system helps to reduce the acceleration at the HDD, PSD of the HDD acceleration is depicted in Fig.6 for optimally-isolated (with displacement constraint) and rigidly-mounted laptops. It could be seen that the PSD level in the vicinity of and after the mode pertaining to the HDD vibration (≈ 250 Hz) is significantly suppressed at the expense of a very well-damped isolation resonance at about 70 Hz. This consequently results in an attenuation factor of about 4.4.

4. Conclusions

In this study, we investigate the design and analysis of passive isolation system for laptop computers. We consider a 3-dof lumped-parameter model of a laptop computer and cast the the isolation system design as an optimization problem. The problem is defined as optimizing the isolation frequency and damping such that the rms acceleration of the HDD is minimized while the vibration travel of the laptop chassis is confined within a given displacement limit. The excitation is assumed to be zero-mean, stationary, and Gaussian random base vibration with flat PSD level of 0.1 g^2/Hz in the wide frequency range of [0 2000] Hz. Also, both flexible and relatively stiff chassis were considered.

It is shown that for the purpose of vibration travel of the chassis an sdof model will be sufficient while a more sophisticated model (a 3-dof model here) is necessary for modeling the transmitted acceleration to the HDD. In view of the displacement constraint and the sdof model, an analytical expression is derived relating the isolation frequency to the maximum allowable vibration travel and the isolation damping. Hence, the constrained optimization problem with two decision variables is turned into an unconstrained optimization with only one decision variable which is then numerically solved.

It was shown that for the flexible and the stiff chassis, there is optimum damping ratio of 0.50, and 0.38, with corresponding optimum isolation frequencies of 70.42 Hz and 77.5 Hz, respectively, that minimize the transmitted rms acceleration to the HDD while confining the laptop chassis within the allowable space. With the optimum isolation system, the transmitted acceleration to the HDD is reduced by a factor of over four when compared to a rigidly-mounted laptop. Deviation from the optimum isolation damping, in particular decreasing it, could severely deteriorate the vibration isolation quality. The isolation quality is even more sensitive to the isolation damping if the displacement constraint is neglected i.e. for a fixed isolation frequency. Last but not least, the methodology presented here is not case-specific and could be applied to the isolation system design of a wide range of systems.

References

[1] Harris C M and Piersol A G 2002 *Harris' shock and vibration handbook* vol 5 (McGraw-Hill New York)
[2] Veprik A and Babitsky V 2000 *Journal of sound and vibration* **238** 19–30
[3] Veprik A 2003 *Journal of Sound and Vibration* **259** 161–175
[4] Veprik A, Babitsky V, Pundak N and Riabzev S 2000 *Shock and Vibration* **7** 363–379
[5] Veprik A, Babitsky V, Pundak N and Riabzev S 2001 *Shock and Vibration* **8** 55–69
[6] Veprik A, Djerassy S and Babitsky V 2008 *Proceedings of the Institution of Mechanical Engineers, Part C: Journal of Mechanical Engineering Science* **222** 1885–1897
[7] Golycheva E, Babitsky V and Veprik A 2003 *Journal of sound and vibration* **259** 829–843
[8] Golysheva E, Babitsky V and Veprik A 2004 *Journal of Sound and Vibration* **274** 351–367
[9] Sokolov I, Babitsky V and Halliwell N 2007 *Journal of sound and vibration* **306** 59–73
[10] Hosseinloo A H, Tan S P, Yap F F and Toh K C 2014 *Applied Thermal Engineering* **73** 1076–1086
[11] Hosseinloo A H, Yap F F and Chua E T 2014 *Aerospace Science and Technology* **35** 29–38

[12] Hosseinloo A H, Yap F F and Lim L Y 2015 *Journal of Vibration and Control* **21** 468–482
[13] Jayson E M, Murphy J, Smith P and Talke F E 2003 *Journal of tribology* **125** 343–349
[14] Lim G, Park K S, Park N C, Park Y P, Lee Y, Chang Y B and Kim C S 2015 *Microsystem Technologies* **21** 2705–2715
[15] Zhu D l, Wang A l and Jiang T 2006 *Structural and Multidisciplinary Optimization* **31** 497–503
[16] Matsuda Y, Nakamura S, Sega M and Katou Y 2009 *IEEE Transactions on Magnetics* **45** 5108–5111
[17] Wang Y, Li L, Morris K, Hogan J and Talke F E 2015 *Microsystem Technologies* **21** 2657–2662
[18] Harmoko H, Yap F F, Vahdati N and Li C 2009 *Shock and Vibration* **16** 143–154
[19] Lim S, Chang Y B, Park N C and Park Y P 2006 Optimal design of rubber mounts supporting notebook hdd for shock and vibration isolation *Asia-Pacific Magnetic Recording Conference, 2006* (IEEE) pp 1–2
[20] Park K S, Lim S, Park Y P, Chang Y B and Park N C 2012 *Microsystem technologies* **18** 1559–1566
[21] Lai F, Mou J, See I and Lin W 2013 *International Journal of Mechanical Sciences* **76** 60–69
[22] Alavi S R, Rahmati M and Ziaei-Rad S 2016 *Microsystem Technologies* 1–10
[23] Rahmati M, Alavi S R and Ziaei-Rad S 2016 *Microsystem Technologies* 1–15
[24] Hosseinloo A H, Yap F F and Vahdati N 2013 *International Journal of Structural Stability and Dynamics* **13** 1250062