# The Impact of Mounting Parameters Variation on Vibration Isolation Effectiveness

In a parametric study the impact of varying the attributes of an isolated system (damping, stiffness, and mass) on its vibration isolation effectiveness is analyzed. The results of this parametric study are summarized in Table 1. The table shows the positive (enhancement), negative (deterioration), and zero (no effect) impacts of change in damping, stiffness, and adding mass (to the isolated object) on the vibration isolation effectiveness.

Table 1

| Parameter Variation | | Effect on: Shock Isolation | Low-Frequency Vibration Isolation | High-Frequency Vibration Isolation |
|---|---|---|---|---|
| Damping | increase | + | - | - |
| | decrease | - | + | + |
| Stiffness | increase | + | - | 0 |
| | decrease | - | + | 0 |
| Added Mass* | increase | + | + | + |

* Note that the benefits of added mass is realized at the expense of a large weight penalty

Evident from Table 1, no single solution enhances all the attributes of a mounting mechanism isolating an object subject to simultaneous vibration and shock loadings, e.g., a diesel-generator on-board a watercraft. For example, enhanced low-frequency vibration isolation performance of a system with low damping and low stiffness is normally achieved at the expense of excessive displacement of the mounted mass around the resonant frequency of the system diminishing its shock isolation effectiveness.

## Parametric Study

The spring-mass-dashpot system of Figure 1 is frequently used as a one degree of freedom representation of an isolation system. The goal is to isolate the base from the vibration of the mass caused by the excitation force F, i.e., lowering the force transmitted to the base Ft , while avoiding excessive vibration of the mass (bouncing) cause by shock perturbations. The spring dashpot pair, known as Voight model, is used to approximate the dynamics of commonly used viscoelastic isolators such as rubber mounts.

The effects of varying damping, stiffness are presented below.

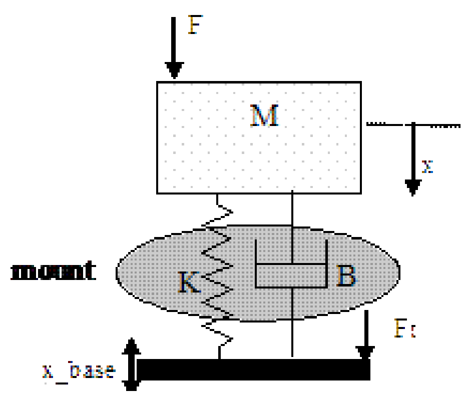


Figure 1 A simple mounting system

Figures 2(a), 3(a), and 5(a) depict the magnitudes of the frequency response functions (FRF) mapping the perturbations force (F) generated by the isolated machine and motion of the base (x_base) to the transmitted force Ft and mass displacement x, commonly known as transmissibility, for varying a) damping, b) stiffness, and c) mass (by adding a dead weight to the mass), respectively. Figures 2(b), 3(b), and 5(b) depict the magnitudes of the frequency response functions (FRF) mapping the vibration excitation force (F) to the mass displacement x, for the same parameter variations.

## Damping Variation

Clear from the transmissibility plot of Figure 2(a), lower damping does not affect neither the transmission of force from the vibrating mass (e.g. diesel generator) to the base nor the transmission of shock inputs at the base to the mass at low frequencies, increase them around the resonant frequency, and lower these transmissions at high frequencies. This is why in most isolation applications the mount is selected to be highly underdamped with the resonant frequency of the isolated system less than 70% of the lowest vibration and shock excitation frequencies of the system. Of course, when either or both excitations (vibration and shock) are broadband (like discontinuous shock excitation common in marine applications), the above stated guideline will not be effective; broadband excitation will set off the resonance of the mounted system causing the mass to bounce. Figure 2(b) indicates that variation in damping only influences the transmission of vibration force to the motion of the mass at resonance; the lower the damping the more severe the resonance.
The reason practitioners do not embrace the obvious solution to the resonance problem, i.e., adding damping, is that damping deteriorates the high-frequency vibration isolation effectiveness of the mount resulting in transmission of noise.

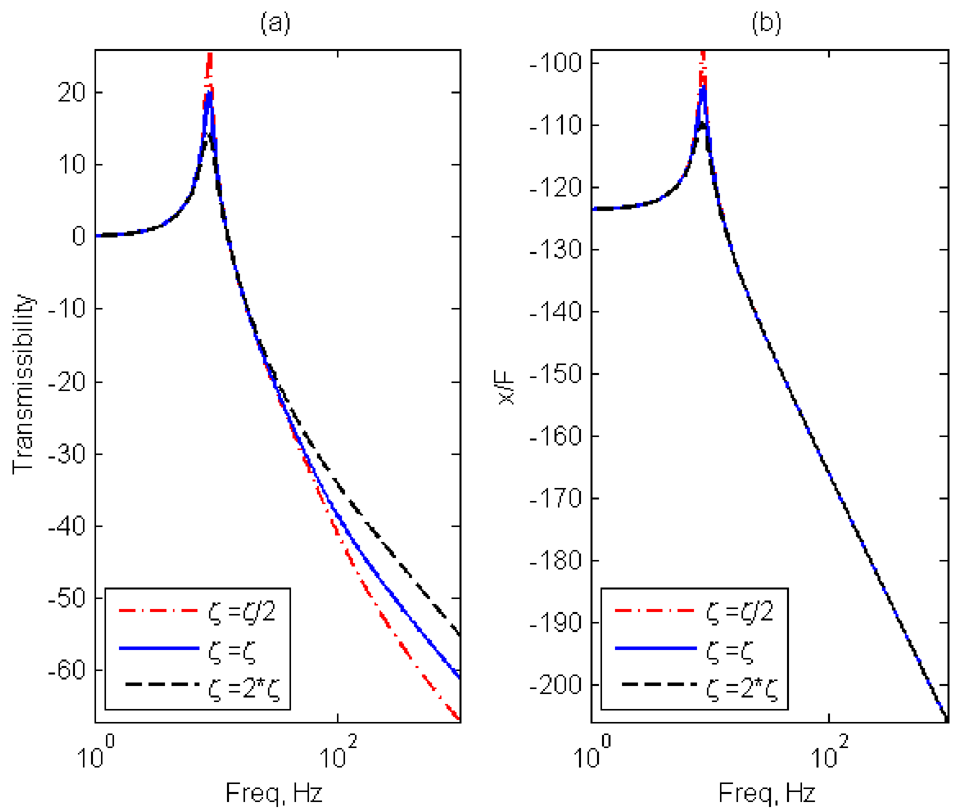


Figure 2 Transmissibility and displacement frequency response functions for different damping ratios.

## Stiffness Variation

Figure 3(a) indicates that softer mounts (mounts with lower stiffness) lower the transmission of force from the vibrating mass to the base and the transmission of shock inputs at the base to the mass. Figure 3(b) indicates that soft mounting adversely affects the transmission of vibration force to the motion of the mass at lower frequencies, including at resonance; the lower the stiffness the larger the low-frequency motion of the mass caused by vibration forces. This adverse effect of soft-mounting would be tolerable if a) the excitation is harmonic (not impulsive such as shock inputs) and b) the resonant frequency of the isolated system is below the lowest excitation frequencies of the system.

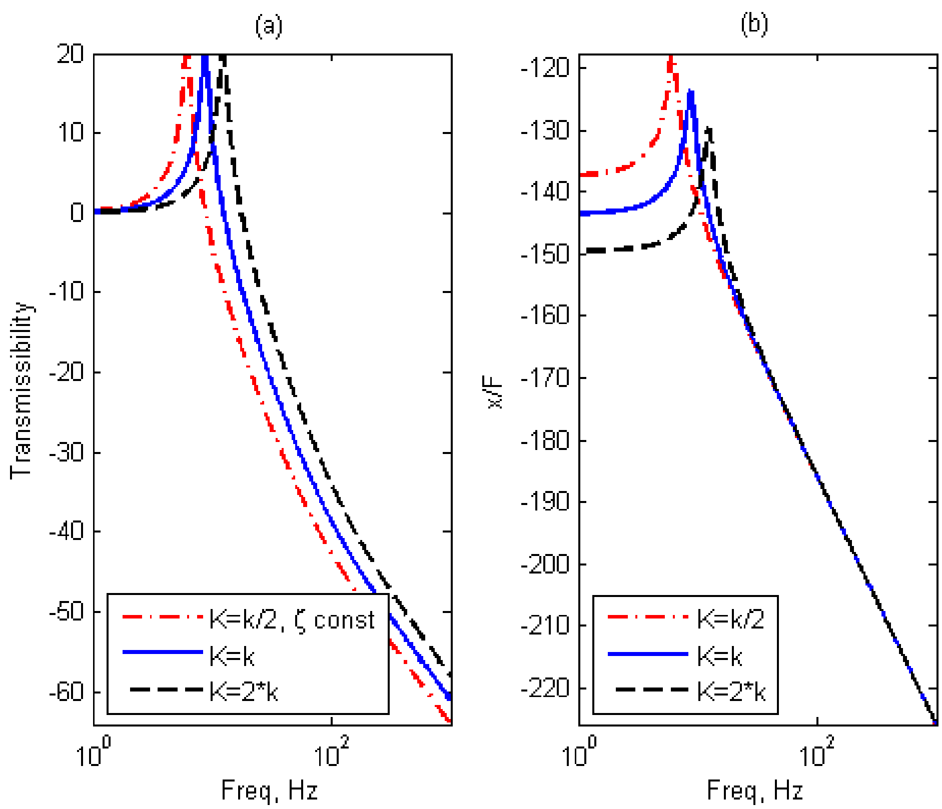


Figure 3 Transmissibility and displacement frequency response functions for different stiffnesses

Frequently, the isolated object is perturbed by both periodic vibration, e.g., the force of firing in a diesel engine, and impulsive shock, e.g., the base excitation in a marine diesel generator or any other vehicular engine applications. The base excitation has normally a broadband spectrum, so there is always some energy at the resonant frequency(ies) of the isolated system. If the mounts are highly underdamped, the vibration amplitude at this (these) frequency(ies) becomes excessive. This problem can be addressed by adding damping to the mount. But in vibration isolators, as stated earlier, broadband damping which results in high frequency vibration (and noise) transmission is avoided.

## Adding Mass

Increasing the mass of the isolated object by adding an auxiliary mass (also known as inertia mass) to it, in order to vary the attributes of the isolation system, is a common vibration isolation practice. One way of realizing this scheme is to install the vibrating machine on a massive concrete block and isolate the block from the foundation by rubber or neoprene isolators. A schematic of a simple mounted system with an inertial mass, m, added to the isolated body (machine/equipment), M, is shown in Figure 4.

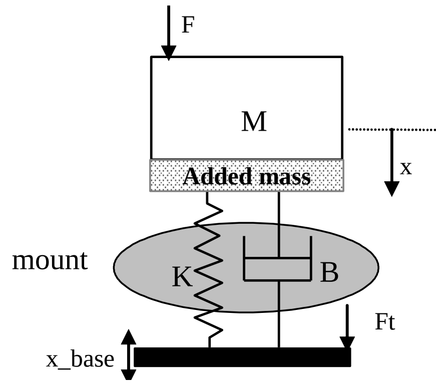


Figure 4 A simple mounting system with added mass (m)

Figure 5(a) indicates that the addition of extra mass to the mounted object, enhances the high-frequency shock and vibration isolation attributes.

Although the addition of mass does not change the low-frequency transmissibility of the system, but as shown in Figure 5(b) it lowers the low-frequency motion of the mass caused by vibration forces of the isolated machine. Note that as the added mass gets larger, the stiffness of the isolators should get larger, correspondingly; this keeps the natural frequency of the isolation system unchanged.

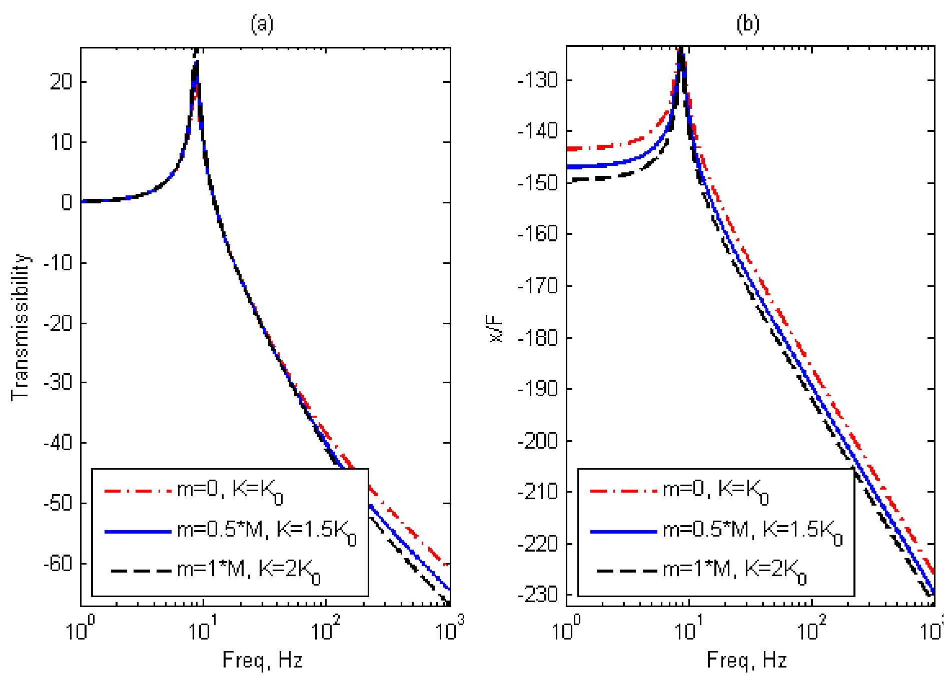


Figure 5 Transmissibility and displacement frequency response functions for varying the mass (m).

Clear from Figure 5, the high-frequency isolation enhancement associated with the added mass is realized when a sizeable amount of it (the added mass) is used. In some applications such as isolating a very sensitive lab equipment, the added mass is many times larger than the mass of the isolated equipment itself. That said, in mobile applications (e.g., onboard watercrafts) the considerable weight penalty associated with the use of large added mass is objectionable.