# Vibration and Shock Isolation

Vibration and Shock isolation systems lower the transmission of vibration and shock between two interconnected objects. Such systems are commonly realized by placing a set of resilient elements such as elastomeric (rubber), steel, or air springs between the two objects isolated from each other (e.g., a piece of equipment and its support structure/base).

In addition to load-supporting (resilience), an isolation scheme has energy dissipating attributes. In elastomeric isolators, made of natural or synthetic rubber, the load-supporting and energy-dissipating tasks are commonly performed by a single element, i.e. the material itself.

If an isolator has the resilience but lacks sufficient energy-dissipating characteristics, e.g., metal springs; then separate energy-dissipating means (e.g., viscous dampers) are paired with the resilient element.

The spring-mass-damper system of Figure 1 is commonly used as the one degree of freedom representation of an isolated machine/equipment. The mass *M* resembles a machine or equipment being isolated and the pair of spring *K* and damper *C* resemble the isolator.

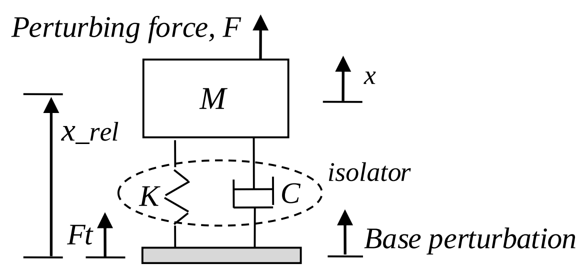


Figure 1 Schematic of an isolated system

## VIBRATION ISOLATION

The goal of a vibration isolation system is to isolate the support structure (base) from the vibration of the mass caused by the perturbation force *F*, i.e., lowering the force transmitted to the base *Ft*, while avoiding excessive motion of the mass, *x*. In addition, the vibration isolation system is to isolate the mass (isolated machine/equipment) from the perturbing motion of the base.

The effectiveness of a vibration isolation system intended to reduce the transmission of the perturbation force (*F*) generated by the machine/equipment to the base (*Ft*) is evaluated by transmissibility. The design goal of an isolation system is to reduce the magnitude of transmissibility, over the frequencies of interest, without inducing too much motion into the machine/equipment, itself; in other words, reducing the magnitudes of **transmissibility *Ft/F*** and **motion response *x/F*** (or its dimensionless representation, *x/(F/K)* ).

> **Transmissibility,** defined by ***Ft/F,*** is a measure of the reduction in a) transmitted force (from the equipment to the base), provided by an isolator.

The effectiveness of a vibration isolation system used to reduce the vibratory motion transmitted from a vibrating base (*x_base*) to the machine/equipment (*x_rel* ) is characterized by **Relative transmissibility *x_rel/x_base*** . Note that *x_rel* is the motion of the mass/equipment relative to the perturbing motion of the base.

**Relative transmissibility,** defined by ***x_rel/x_base*** is the ratio of the relative motion of the isolated machine/equipment with respect to the base to the displacement of the base. Note that the relative motion *x_rel* is also the 'deflection' of the isolator; it is a measure of the working space required for the isolator.

## The Impact of Damping on Vibration Isolation

The impact of isolator damping of performance of a vibration isolation system is illustrated using the isolation scheme shown in Figure 1. The isolation system is to isolate the base from the vibratory perturbing force of the equipment with frequencies starting from 12.5 Hz. The natural frequency of the isolated system is selected to be 60% (less than 70%) of the lowest perturbing frequency.

The performance is defined in terms of absolute transmissibility, relative transmissibility, and motion response of the vibration isolation system. Figure 2 shows the variation of these metrics over various frequencies, for a low and a high levels of isolator damping.

Figure 2 indicates that increasing damping tames the resonance of the isolated system; the higher the damping the less severe the resonance. High damping would have been a positive attribute had it not adversely impacted the transmission of perturbing force from the isolated mass to the base presented by transmissibility. The transmissibility plot of Figure 2(b) shows that the isolator with more damping transmit more of the vibratory perturbing force from the equipment to the base, over the frequencies of interest. This is why in most isolation applications the mount is selected to be highly underdamped.

Damping does not impact the motion response (shown in Figure 2(a)) and relative transmissibility (shown in Figure 2(c)) of the system, over the frequencies of interest.

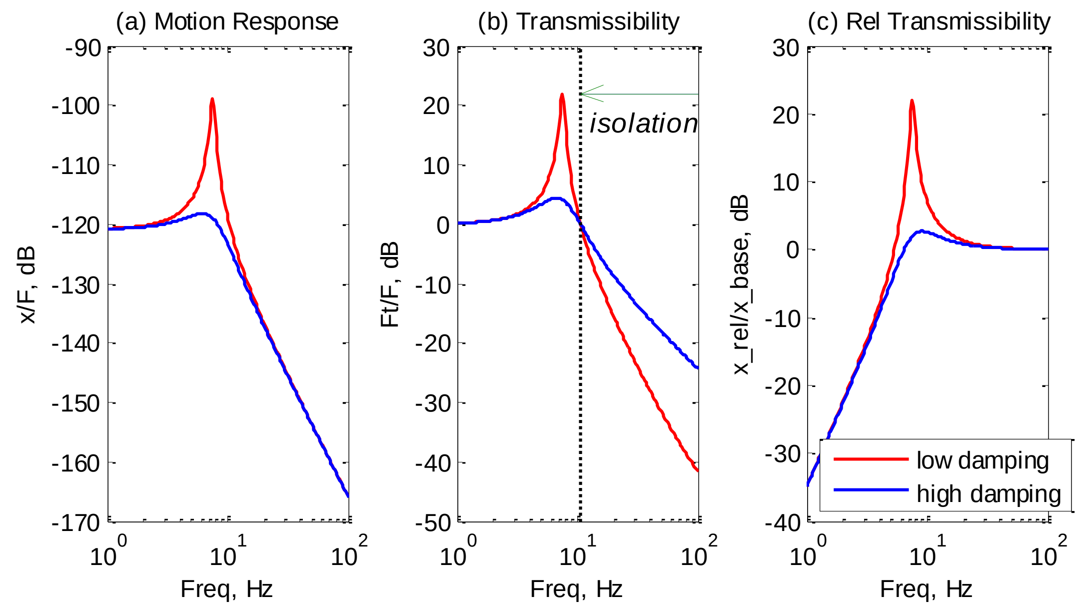


Figure 2 Motion response (a) transmissibility (b) and relative transmissibility (c) frequency response functions for an isolation system with low damping (red) and high damping (blue)

## The Impact of Isolator Stiffness on Vibration Isolation

The isolation scheme shown in Figure 1 is used to illustrate the impact of isolator stiffness on the performance of a vibration isolation system intended to isolate the base from the vibratory perturbing force of the equipment with frequencies starting from 12.5 Hz. A rather stiff isolator placing the natural frequency of the isolated system at 60% and a softer isolator placing that natural frequency at 30% of the lowest perturbing frequency are considered in this study. In both cases, the extent damping was kept the same at 4%.

Evident by the transmissibility plot of Figures 3(b) and relative transmissibility plot of Figure 3(c) the softer isolator lowers both transmission of perturbing force from the vibrating equipment to the base and transmission of vibratory perturbing motion from the base to the equipment, over the frequencies of interest (frequencies including and higher than the 'lowest perturbation frequency').

The motion response of Figure 3(a) indicates that although softer isolator does not have significant impact on the vibration isolation effectiveness over the frequencies of interest, but could adversely affect the motion of the equipment 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 on a vibration isolation system need to be paid attention to if the perturbing frequency becomes lower than the lowest design frequency of the isolation system (e.g., as an engine goes thru its start up or shut down), or the perturbation becomes random or impulsive.

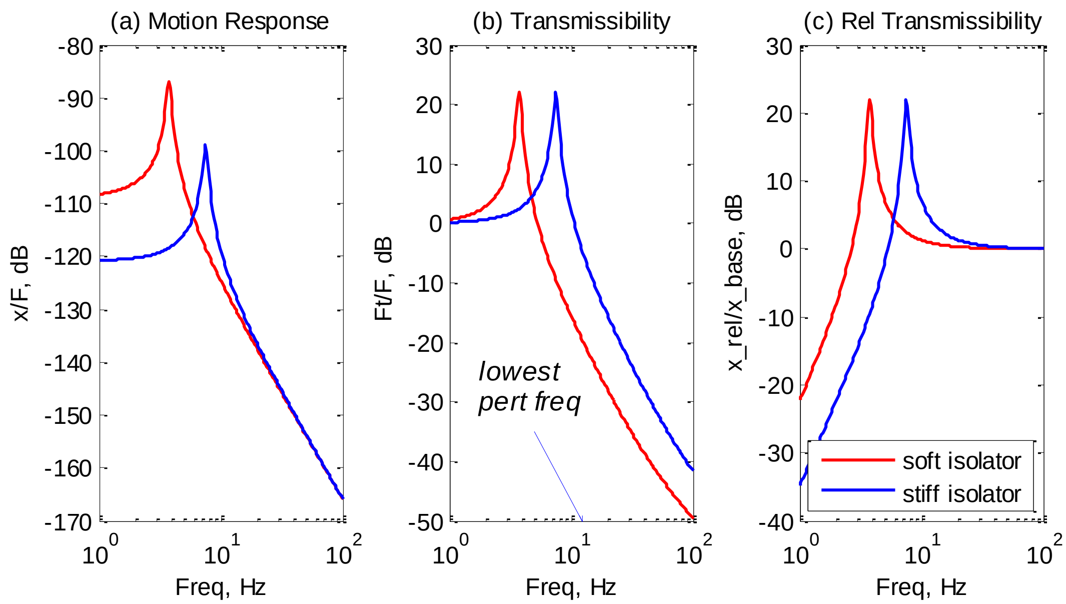


Figure 3 Motion response (a), transmissibility (b) and relative transmissibility (c) frequency response functions for an isolation system with low stiffness (red) and high stiffness (blue)

The common practice in designing a vibration isolation system is to select the resilience of the isolator (*K*) such that the natural frequency it creates in conjunction with the mass of the machine (*M*) is set substantially lower than the principal perturbing frequency.

**The Impact of Added Mass on Vibration Isolation**

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.

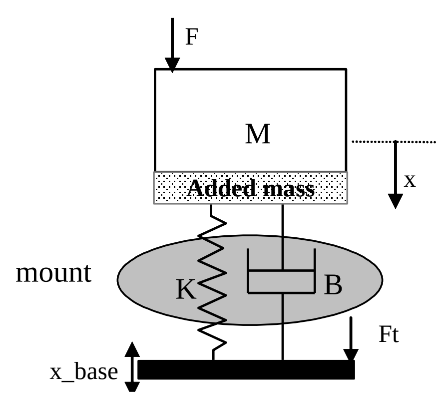


Figure 4 A simple mounting system with added mass (m)

Figure 5(a) indicates that the addition of extra mass to the mounted object, lowers the high-frequency transmissibility.

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

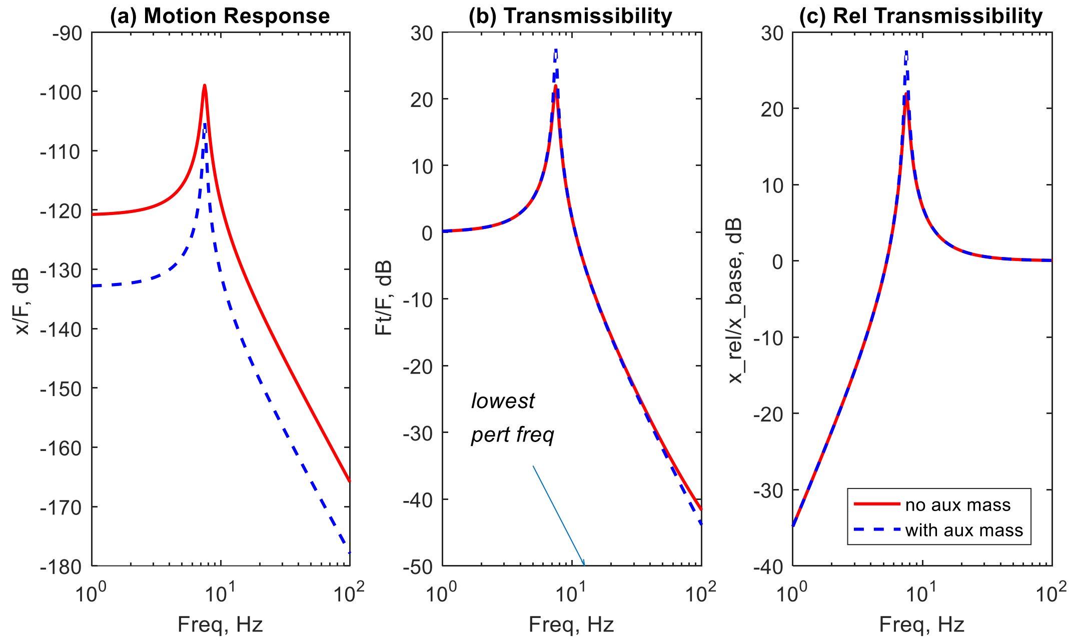


Figure 5 Motion response (a), transmissibility (b) and relative transmissibility (c) frequency response functions for an isolation system without (red/solid line) and with (blue/dashed line) added mass

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.

## SHOCK ISOLATION

Contrary to vibration which is for the most part a steady state phenomenon, shock is defined as a transient condition occurring as abrupt perturbations (containing a pulse of energy of short duration and large intensity). The parameters characterizing a shock input pulse are: its peak acceleration amplitude commonly expressed in percentages of acceleration of gravity (g), duration, and overall shape which can take such forms as half-sine, triangular, rectangular, etc., all of which influence the extent of energy in the shock and thus its severity. Shock perturbation, with its abrupt nature, excites all resonances in an isolated system. The transmission of shock input to a machine/equipment can be reduced by proper shock isolation.

The effectiveness of a shock isolator is judged by its ability to reduce the transmission of the base acceleration to the machine/equipment. Acceleration transmissibility is commonly used to characterize the performance of a shock isolation system.

**Acceleration transmissibility,** defined as the ratio of the acceleration experienced by the machine/equipment to the perturbing acceleration of the base.

The mechanism for shock isolation is the storage of shock energy in the resilient element of the isolators and its subsequent release into a "smoother" vibratory process, over a longer period of time (at the natural frequency of the spring-mass isolated system) and dissipation of a portion of that energy, in every cycle of oscillation, by the dissipative element of the isolators. However, the energy storage process takes place by a rather large deflection of the isolator itself, resulting in excessive motion of the isolated machine/equipment.

Alternatively, a shock isolation system can be viewed as a mechanical low pass filter with a low corner frequency, i.e., having low stiffness. Such low pass filter lowers the transmission of high frequency components of the shock perturbation to the isolated mass; this is despite the misconception that a good shock isolation system must be mechanically “stiff”.

As in vibration isolation systems, the resilience (stiffness) and the extent of damping built into an isolator are the main parameters affecting the performance of a shock isolation system.

### The Impact of Isolator Parameters on Shock Isolation

The two traces in Figure 6 show the displacement (a) and the acceleration (b) responses of a 500 Kg mass to a ½ sine shock perturbation with the peak amplitude of 10 g. The mass is isolated using a soft isolation system with the natural period of 0.32 sec (red trace) and a stiff isolation system using the natural period of 0.16 sec (blue trace). The shock pulse duration is 16 msec (i.e., 1/10 of the natural period of the stiffer isolation system). *Note that the response to a shock perturbation is normally the transient oscillation of mass at the isolation system’s natural frequency.*

Comparison of the two traces shown in Figures 6(a) and 1(b) indicates that while soft isolation (having a natural period many times larger than the duration of shock pulse) reduces the transmitted *acceleration* more than the stiff isolation does, but it increases the motion of the mass (deflection of the isolator).

It should be noted that if the isolator runs out of permissible deflection and bottoms out, the acceleration (*g* level) transmitted to the isolated mass (equipment) becomes excessive. As such, the system must be allowed to oscillate freely, and not come to a 'hard stop', once it has been exposed to a shock disturbance.

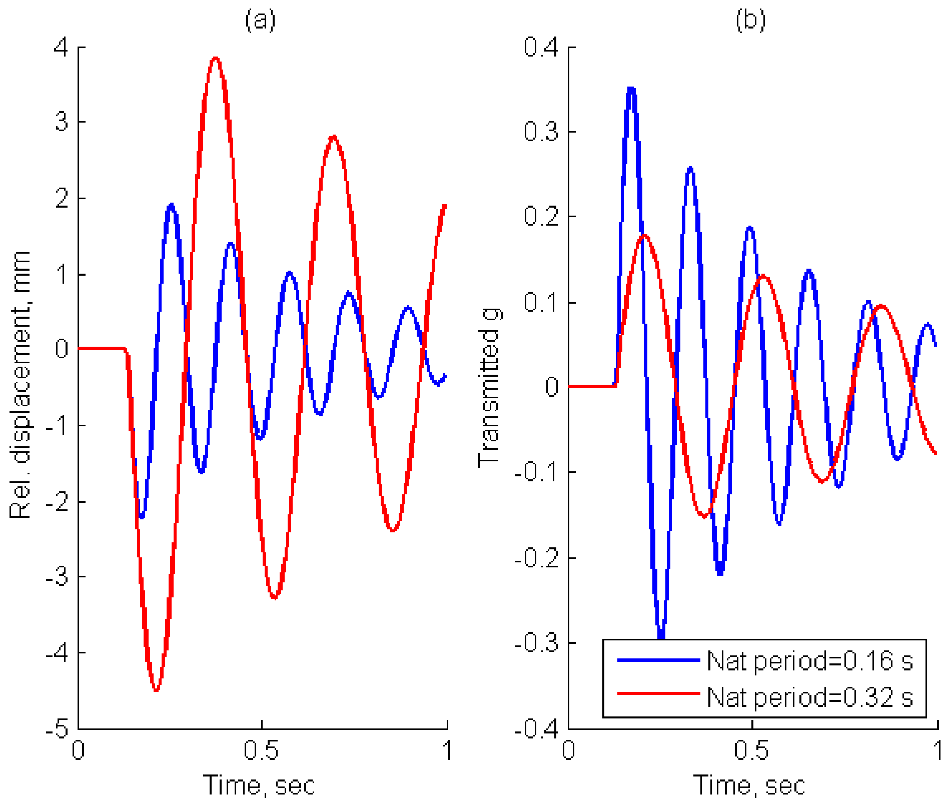


Figure 6 Shock induced relative displacement (a) and transmitted acceleration (b) of an isolated object using soft (red traces) and stiff (blue traces) isolators with negligible damping

To lower the excessive isolator deflection, a shock isolation system needs to dissipate a considerable amounts of energy in a minimal amount of time which can be done by incorporating a sizeable amount of damping into the isolation system. As shown in Figure 7, introduction of more damping in the isolation system will dissipate more of the vibration energy resulting in lower peak transmitted *g* to and lower peak displacement of the isolated mass, followed by substantially lower ringing (lingering of oscillation) of both.

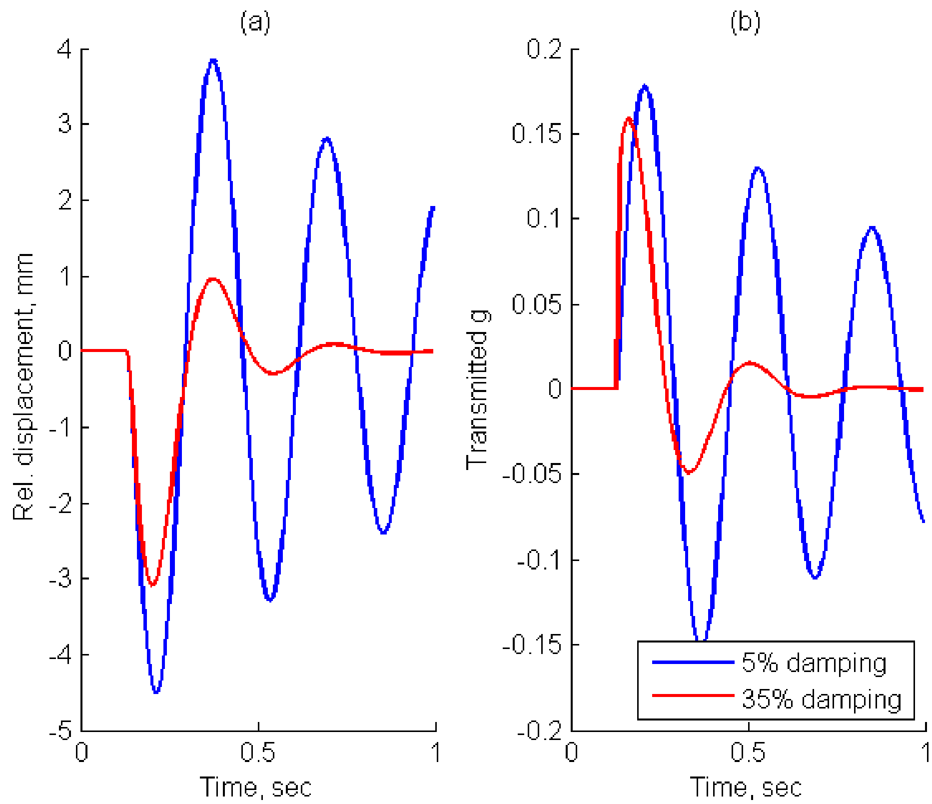


Figure 7 Shock induced relative displacement (a) and transmitted acceleration (b) of an isolated object using soft isolator with sizeable (red traces) and negligible (blue traces) damping

To enhance the shock isolation effectiveness, it is desirable for the isolation system to be soft and heavily damped.

**Shock Induced Motion Reduction**

As discussed above and shown in Figures 6 and 7 soft isolation reduces the shock-induced *acceleration* but it increases the motion of the mass (deflection of the isolator). In some applications, the concern is not so much abating the transmitted shock-induced acceleration of the isolated equipment. The main concern is lowering the shock-induced relative displacement of the isolated object. A Diesel generator installed in a watercraft is a good example of such application. Such machine can certainly take the harsh acceleration it experiences when the watercraft is perturbed by a massive wave, but what is cannot tolerate is too much motion; excessive motion endangers the structural integrity of the installation. In such applications, the equipment needs to be isolated using a stiff isolator.

**VIBRATION AND SHOCK ISOLATION**

No single set of parameters enhances all the attributes of an isolation system isolating an object subject to simultaneous vibration and shock loadings.

- Low stiffness provided by soft isolators, such as air springs, enhances both shock and vibration isolation effectiveness but causes excessive motion of the isolated equipment
- Large damping while detrimental to force transmissibility under vibration inputs, is beneficial in a shock isolation system.

Stiffness (resilience) and damping (energy dissipation) of an isolation system dictate its vibration isolation effectiveness measured by lowering force and displacement transmissibilities as well as its shock isolation effectiveness measured by lowering its acceleration transmissibility.

The design of an isolation system has traditionally been addressed by a compromise meeting all the vibration and shock isolation requirements.

With the use of air isolators and the advancements in low-cost and reliable sensors and computer technology, DEICON's computer controlled air isolation system addresses all the conflicting requirements of an isolation system, with no compromise. It provides light damping and low stiffness in vibration isolation mode, but automatically upon sensing a shock event supplies a) high damping while maintaining low stiffness for lowering the shock induced acceleration and b) high damping and high stiffness for lowering the shock induced displacement of the isolated equipment.