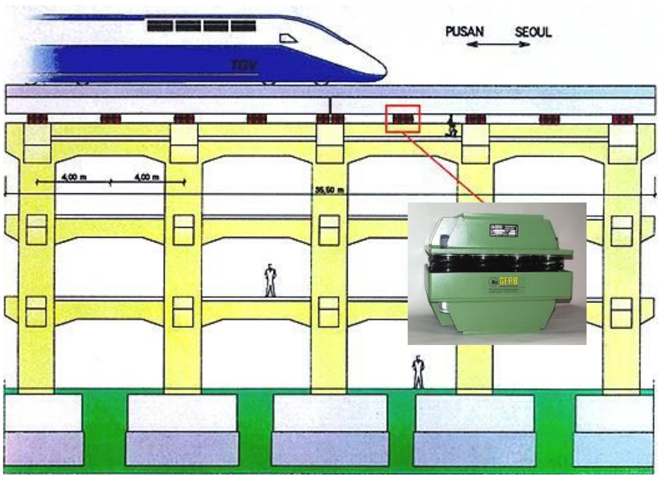
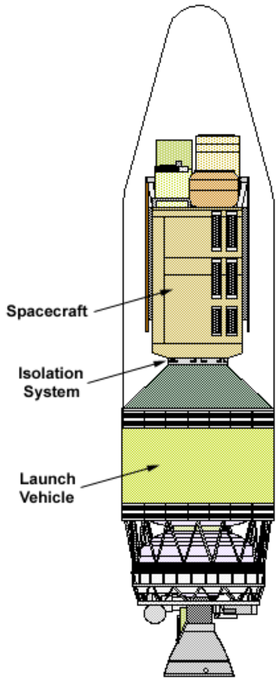
Vibration Isolation
MODULE: MMME2046 DYNAMICS & CONTROL

Vibration isolators (also known as "anti-vibration mounts") are used for reducing the vibration transmitted from a source. They work by introducing flexibility between a device and its support.

Case (a) In some cases, the source of vibration is within the device and **the objective is to minimise the force transmitted to the support**. Examples are a car engine that can transmit vibration to the body shell and a passing train that can produce ground-borne vibration.

Case (b) In other cases, the source may be remote, but causes the support for a device to vibrate. Here, **the objective is to minimise the displacement transmitted to the device**. Examples are a satellite mounted in launch vehicle or the need to protect sensitive laser instruments from ground-borne vibration.





TYPES OF ISOLATOR

Elastomeric isolators

These are the most common type of isolator. Elastomers can be moulded from many different configurations of many different materials, including natural rubber, neoprene, butyl, silicon and a number of combinations of each. A typical mount made with these materials generally employs the elastomer in shear but many utilise compressive strain design also. The mounts may employ the elastomer in a manner that provides both shear and compressive loading for effective isolation performance in both the horizontal as well as the vertical direction. It is relatively easy to design various degrees of damping, shape, load-deflection characteristics and transmissibility characteristics into elastomeric isolators. The inherent damping of elastomers is useful in preventing problems at resonance that would be difficult to restrain if coil springs were used.

For isolation from shocks, elastomers offer some significant advantages because of the fact that they can generally absorb shock energy per unit weight to a greater extent than that attainable through other forms of isolator system.

Pneumatic vibration isolators

Pneumatic isolators are air filled, reinforced rubber bellows with mounting plates on top and bottom. Isolators such as these can provide very low natural frequencies with small static deflections. To provide a 1 Hz natural frequency, a steel coil spring isolator would need to be about 600 mm long and capable of deflecting about 250 mm. It would thus be difficult to install and would also present some lateral stability problems. Unlike most isolators, which are passive devices, pneumatic isolators are also used with position feedback in active control systems. This is used, for example, to maintain the height of a table for mounting optical equipment that is sensitive to the slightest movement.

Coil spring isolators

Springs may be loaded in tension but it is frequently more convenient to load them in compression. Coil springs are used primarily for the isolation of low frequency vibration. Consequently, they operate with a relatively large static deflection and lateral stability may be a problem. It can be shown that a coil spring will be stable if

$$\frac{\text{lateral stiffness}}{\text{axial stiffness}} > \frac{\text{static deflection}}{\text{working height}}$$

Coil springs possess practically no damping and the transmissibility at resonance is extremely high. This can be overcome by the addition of friction dampers in parallel with the load carrying spring and these types of isolators are widely used. Another method of adding damping to a spring is by the use of an air chamber with an orifice for metering the airflow. For applications where all metal isolators are desired because of temperature extremes or other environmental factors, damping can be added to a load carrying spring by the use of metal mesh inserts.

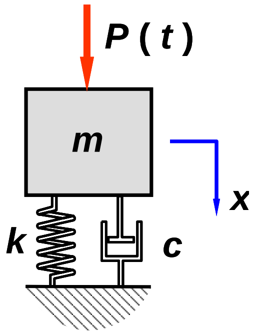
High frequency vibrations can be transmitted through the coils to the "isolated" unit. To overcome this, one or both ends of the spring can be fitted with elastomeric pads. Conventionally, the pad is attached to the bottom of the spring assembly, which has the added advantage of providing a non-slip surface that frequently eliminates the need to bolt the isolator to the floor.

The Vibrostop web site (http://www.vibrostop.it/) shows other examples of the wide range of different designs available.

TRANSMISSIBILITY ANALYSIS

The isolators are invariably very much more flexible than the device they support, so the first approximation is to use a single degree-of-freedom model in which the device to be isolated is treated as a rigid body and the isolators are represented by a spring-damper combination. The steady-state response to harmonic excitation provides a way of characterising the isolation performance at different frequencies.

Case (a) Source of vibration within a device transmitting vibration to the support



Assume that the device generates an excitation force, $P(t) = P\cos\omega t$

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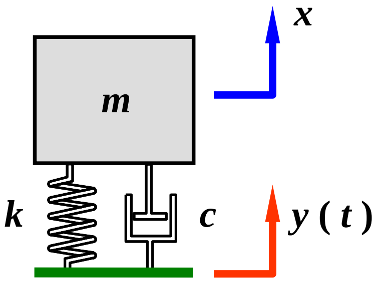
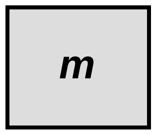
Case (b) Vibration from the support transmitted to the device





The support vibration is defined by the **displacement**, $y(t) = Y\cos\omega t$.

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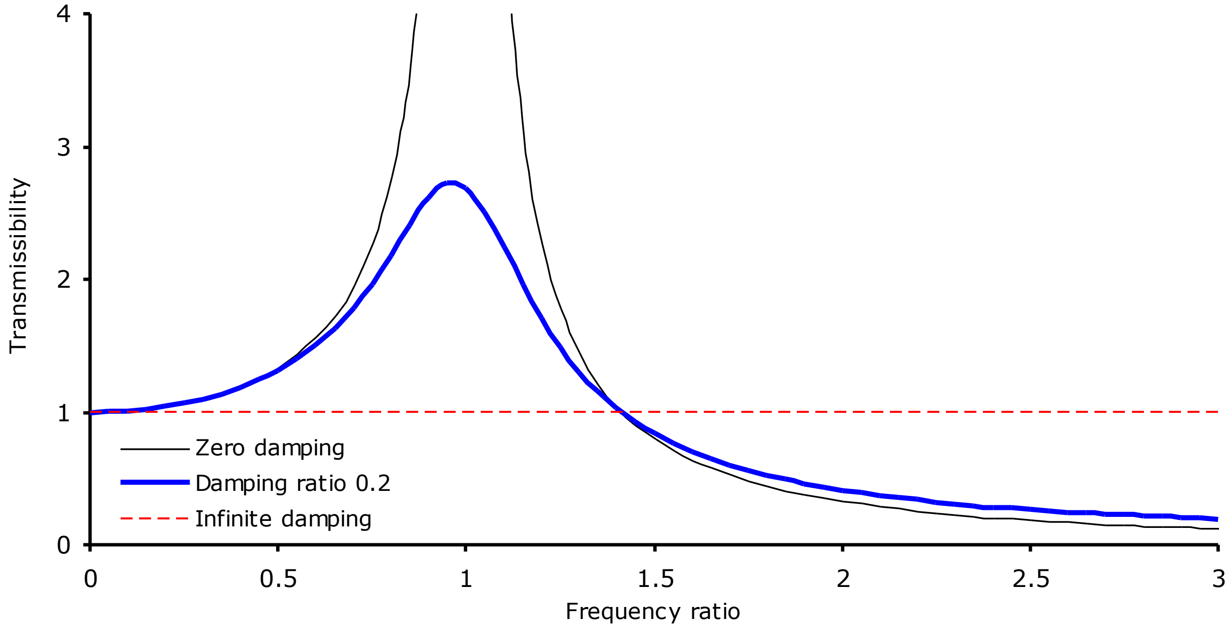
The transmissibility graphs at the top of page 4 are for the simple mass-spring-damper system above. Other systems may have different transmissibility expressions. Infinite damping means that no relative movement is possible, so it represents a rigid connection between the device and its support, which is the original situation without isolators.

Isolation efficiency = $\dfrac{\text{Reduction in displacement (or force)}}{\text{Original displacement (or force)}}$ x 100% = (1 – T) x 100%

Exercise: Show that T = 1 when $\omega/\omega_n = \sqrt{2}$

If $\omega/\omega_n < \sqrt{2}$, the transmitted displacement (or force) is higher than the input. In this case, it would be better to have no "isolators"!

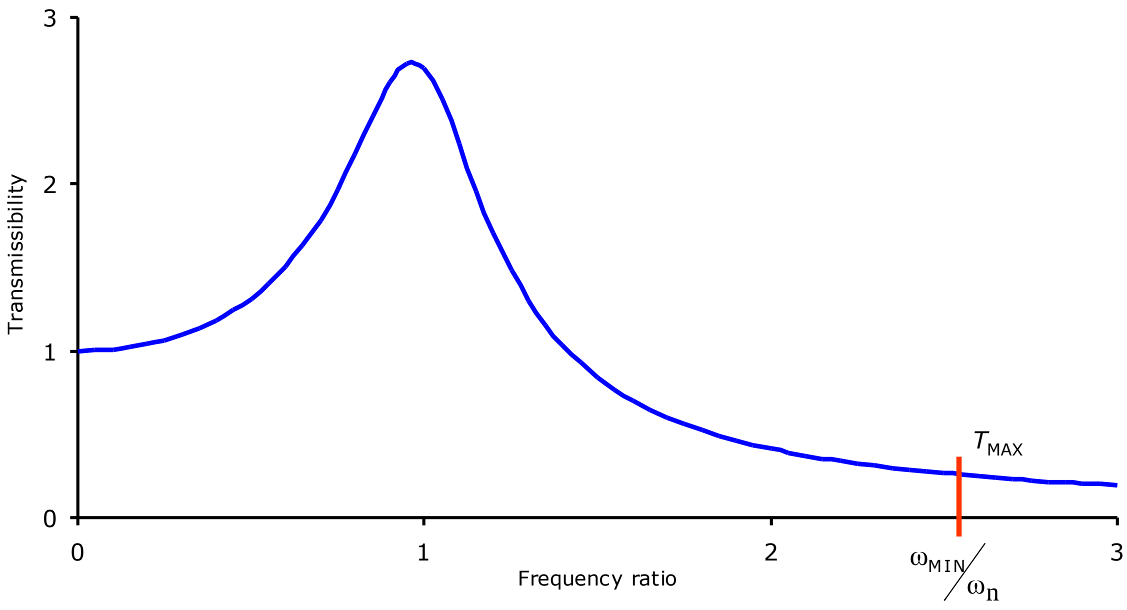
If $\omega/\omega_n > \sqrt{2}$, the transmissibility is less than 1.0, resulting in vibration reduction. The aim in selecting isolators is to ensure that the system operates in this "isolation region".



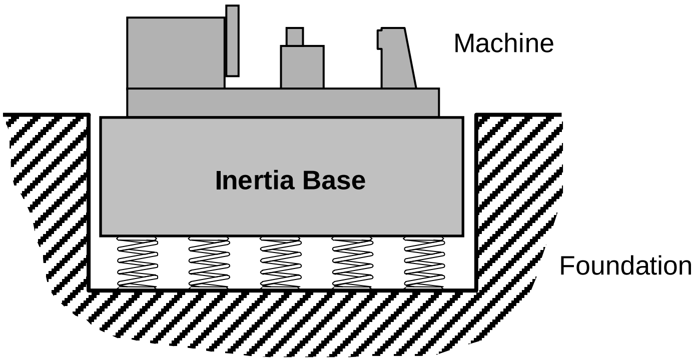
Visit http://physics.nad.ru/Physics/English/spri_txt.htm for some animated illustrations of how the transmitted vibration for case (b) (support excitation) changes as the frequency increases.

B. Design Approach for Isolator Selection

There are two constraints that must be considered when selecting suitable isolators. These are the lowest excitation frequency to be encountered, ω_{MIN}, and the maximum allowable transmissibility, T_{MAX}.



It is clear from the transmissibility diagram that ω_n must be very much less than ω_{MIN}. Three variables affect the system's dynamics; m, k and c. m, k together determine ω_n. The stiffness, k, is given by the set of isolators selected.



At first sight, the mass of the device may not seem to be a variable for this problem. However, m can be increased by mounting it on an inertia base. This will reduce ω_n.

In most vibration situations it is desirable to increase damping. Here, however, increasing c will increase the transmissibility in the isolation region so that low damping is desirable and, of course, easy to achieve. Most commercial isolators give a damping ratio, γ, which is less than 0.1.

It is normal to base design on the assumption of zero damping and, as can be judged from the transmissibility graph at the top of page 4, the error involved in the isolation region is small.

It is also normal to treat each isolator independently of the others. **In this case, *m* is the effective mass supported by the isolator in question.**

For the simple mass-spring model with zero damping,

$$T = \left| \frac{1}{1 - \frac{\omega^2}{\omega_n^2}} \right|$$

$$= \frac{1}{\frac{\omega^2}{\omega_n^2} - 1} \qquad \text{for the isolation region where } \omega > \omega_n$$

If $T = T_{\text{MAX}}$ at $\omega = \omega_{\text{MIN}}$,

$$\omega_n^2 = \frac{T_{\text{MAX}}\,\omega_{\text{MIN}}^2}{1 + T_{\text{MAX}}}$$

Since $\omega_n^2 = \frac{k}{m}$, the required isolator stiffness is

$$k = m\omega_n^2 = \frac{mT_{\text{MAX}}\,\omega_{\text{MIN}}^2}{1 + T_{\text{MAX}}} \qquad (1)$$

If selecting isolators from a catalogue, it is unlikely that one with precisely this stiffness will be found. The stiffness given by equation (1) is the ***maximum*** value consistent with the design constraints.

It might appear from this that any isolator would be suitable provided its stiffness was less than the above value. This is not the case, since there are also constraints imposed by static considerations.

In the case of coil spring isolators, there could be installation, coil bottoming or lateral stability problems if the static deflection is too large. With elastomeric isolators, there are strength limitations under static load.

Manufacturers often express these constraints by specifying a *maximum static deflection*. After selecting an isolator to satisfy the maximum stiffness limit, it is necessary to check that the static deflection limit is not exceeded. The actual static deflection, X_0, is given by

$$X_0 = \frac{m\,g}{k_{\text{isolator}}} \tag{2}$$

Alternatively, combining (1) and (2) gives

$$X_0 = \frac{g}{\omega^2_{\text{MIN}}}\left(1 + \frac{1}{T_{\text{MAX}}}\right) \tag{3}$$

This represents the *minimum* static deflection consistent with the design constraints.

C. Design Procedure

1. Find the centre of mass of the machine.
2. Select the number and position of attachment points for isolators.
3. Estimate the load supported by each isolator.
4. For each isolator position in turn,

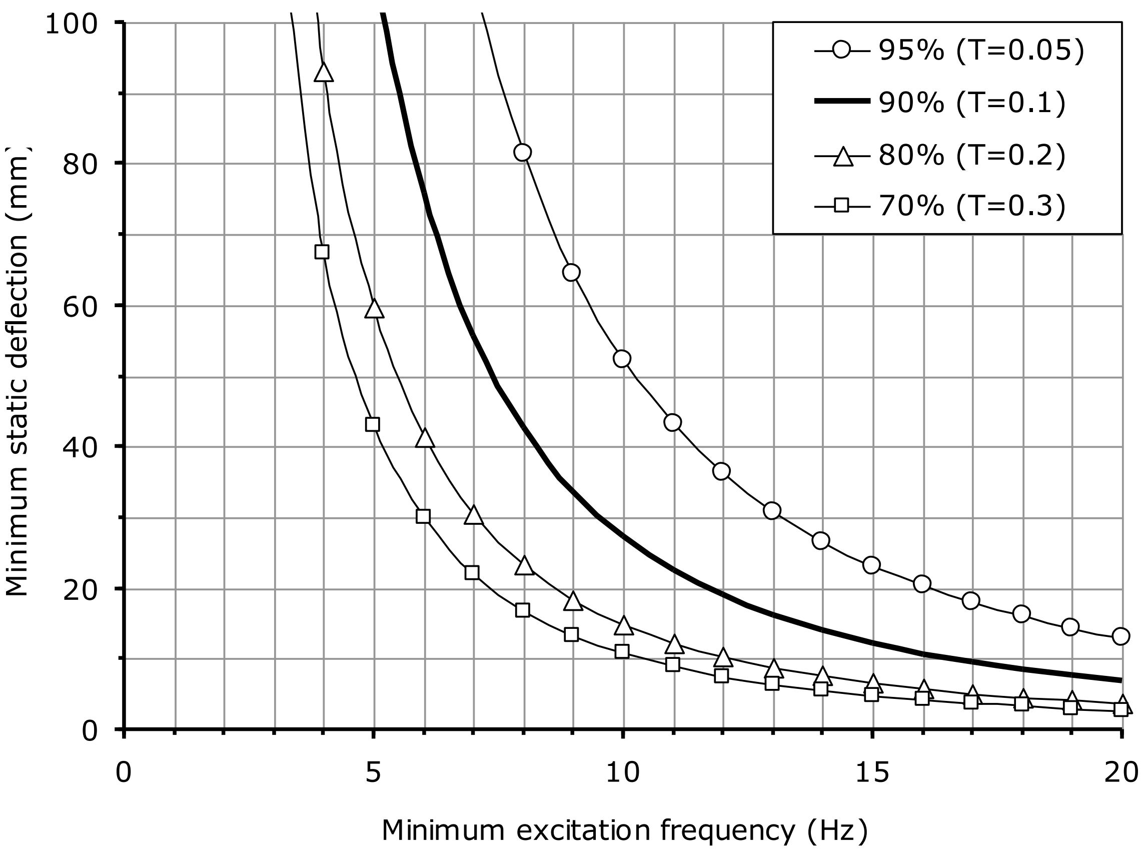
   4.1 Calculate the maximum stiffness from equation (1)

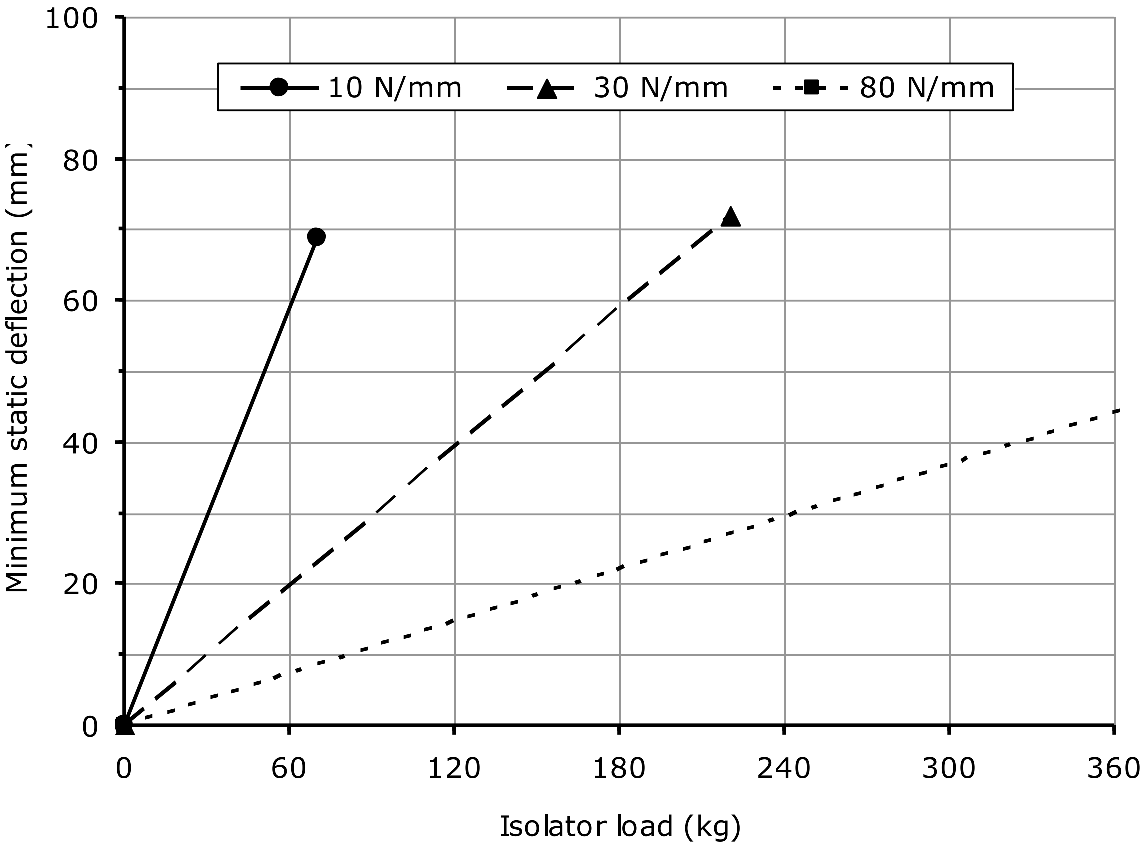
   4.2 Select an isolator with a lower stiffness

   4.3 Check that this does not exceed any static deflection limit using equation (2).

   4.4 Although this will give a satisfactory selection, it is often worth repeating 4.2 and 4.3 with other isolators having even lower stiffness. The lower the stiffness, the greater the isolation efficiency, so the limiting factor becomes the maximum allowable static deflection.

Some manufacturers provide design aids in the form of selection charts (see example on page 7). When using these charts, some modification of the above procedure may be necessary, depending on the nature of the charts provided. In the examples given, each curve on the deflection-frequency graph is obtained from equation (3) using different values of T_{MAX}. The deflection-load graph below it represents equation (2) for different isolators.





Deflection – Frequency Characteristics (Top)
Deflection – Load Characteristics (Bottom)