Vibration and Shock Isolation

The principles involved in both vibration and shock isolation are similar. However, differences exist due to the steady state nature of vibration and the transient nature of shock.

This paper discusses the basic properties of vibration and shock isolators, the requirements for designing successful isolation systems and offers practical examples using these principles.

Vibration Isolation

Introduction

The purpose of vibration isolation is to control unwanted vibration so that its adverse effects are kept within acceptable limits.

Vibrations originating from machines or other sources are transmitted to a support structure such as a facility floor, causing a detrimental environment and unwanted levels of vibration.

If the equipment requiring isolation is the source of unwanted vibration, the purpose of isolation is to reduce the vibration transmitted from the source to the support structure.

Conversely, if the equipment requiring isolation is a recipient of unwanted vibration, the purpose of isolation is to reduce the vibration transmitted from the support structure to the recipient.

An isolator is a resilient support which decouples an object from steady state or forced vibration. To reduce the transmitted vibration, isolators in the form of springs are used. Common springs used are pneumatic, steel coil, rubber (elastomeric) and other pad materials.

Natural frequency and **damping** are the basic properties of an isolator which determine the **transmissibility** of a system designed to provide vibration and/or shock isolation. Additionally, other important factors must be considered in the selection of an isolator/isolation material.

Two such factors are:

- The source and type of the dynamic disturbance causing vibration/shock.
- The dynamic response of the isolator to the disturbance.

With an understanding of its properties, the type of isolator is then chosen for the load it will support and the dynamic conditions under which it will operate.

Natural Frequency (Spring Rate)

The simplest form of mechanical vibration to consider is based on a linear system. When applying linear theories, the values of displacement, velocity and acceleration have proportional relationships to the mechanical stiffness (spring rate) of the vibration isolation system.

Typically, to make design and analysis easier, the response of isolators which are not truly linear (such as elastomers, cork, felt) are approximated by using linear relationships.

Not all isolators whose isolation characteristics are based on mechanical deflection have a linear relationship between load and deflection. A common mistake is that Equation 1 can be used to calculate the natural frequency for all isolators if the spring rate (k) and supported mass (m) are known.

Equation 1

$$F_N = \frac{1}{2\pi}\sqrt{\frac{k}{m}}$$

If the spring rate (k) is not known, the equation can be rewritten (Equation 2) so that the calculated natural frequency of the isolator is a function of its static deflection. This results in a determination of the isolator's static natural frequency where (g) represents the gravitational constant.

Equation 2

$$F_N = \frac{1}{2\pi}\sqrt{\frac{g}{\delta_s}}$$

However, using the static, linear principles in Equations 1 and 2, the following mistakes are true:

1. Damping properties are neglected.
2. Only the static natural frequency is obtained.
3. The isolator is assumed to have a linear spring rate.

The static deflection principle can only be used to determine the natural frequency of an isolator if the isolator under consideration is both linear and elastic. For example: rubber, felt, fiberglass and composite materials tend to be non-linear and exhibit a dynamic spring rate which differs from the static spring rate.

Similarly the spring rate of a pneumatic isolator changes when undergoing a change from the static condition to a dynamic condition.

The natural frequency as calculated based on static load vs. deflection data will give inaccurately lower natural frequencies than will be realistically experienced during dynamic vibration.

Any isolator with a calculated natural frequency based on static deflections may not behave in the predicted way because the dynamic spring rate differs from the static spring rate. *It is the dynamic natural frequency which has to be used in isolation calculations rather than the static.*

Damping

The essential properties of an isolator are natural frequency (developed by the spring rate or stiffness) and an energy dissipating mechanism known as damping. In some types of isolators the stiffness or natural frequency and damping properties are contained in a single element such as elastomers, cork, rubber mats, etc. Other types of isolators may have separate means of providing stiffness and damping as is the case with air springs (pneumatic isolators) and coil steel springs which are relatively undamped until used in conjunction with auxiliary damping elements such as orifice flow restrictors and viscous dampers.

The purpose of damping in an isolator is to reduce or dissipate energy as rapidly as possible. Damping is also beneficial in reducing vibration amplitudes at resonance. Resonance occurs when the natural frequency of the isolator coincides with the frequency of the source vibration.

The ideal isolator would have as little damping as possible in its isolation region and as much as possible at the isolator's natural frequency to reduce amplification at resonance. Damping however can also lead to a loss of isolation efficiency. (See transmissibility on the following pages).

Transmissibility

The ratio of vibration transmitted after isolation to the disturbing vibration is described as "transmissibility" and is expressed in its basic form in Equation 3, where F_D is the disturbing frequency of vibration and F_N is the natural frequency of the isolator.

Equation 3

$$T = \frac{1}{(F_D / F_N)^2 - 1}$$

When considering damping, the equation is rewritten (Equation 4), where ζ represents the damping ratio of the isolator.

Equation 4

$$T = \sqrt{\frac{1 + (2\zeta F_D / F_N)^2}{(1-[F_D^2 / F_N^2])^2 = (2\zeta[F_D / F_N])^2}}$$

Maximum transmissibility of an isolator occurs at resonance when the ratio of the disturbing frequency to the natural frequency is equal to 1 (i.e. $F_D / F_N = 1$). At resonance the transmissibility is given by Equation 5. Note that the magnitude of an isolator's amplification at resonance is a function of that isolators damping.

Equation 5

$$T = \frac{1}{2\zeta}$$

Figure A graphically shows the transmissibility of an isolator as a function of the frequency ratio. Several percentages of critical damping are displayed to show the effect of damping in the isolation region and the amplification region, including the maximum amplification at resonance. ($F_D / F_N = 1$)

Note that as damping is increased, the curve of transmissibility is flattened, so that in the region near to resonance, the curve is reduced, but in the region where isolation is required, the curve is increased. The curves show that if there is a significant amount of damping in an isolator, its natural frequency has to be reduced to retain the desired degree of isolation at the frequency ratio of concern.

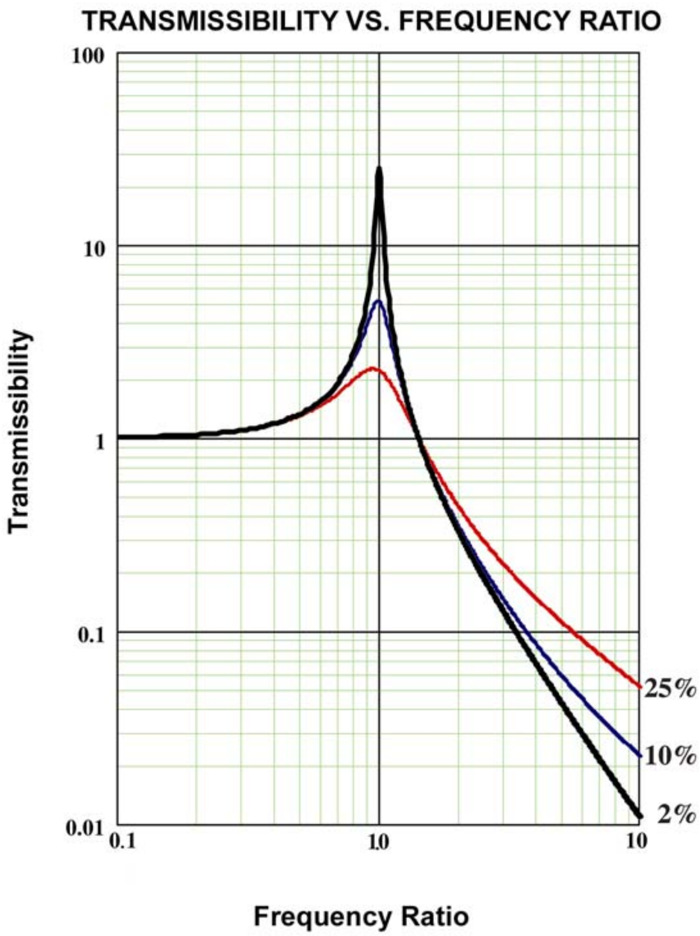

Figure A

To better illustrate transmissibility and how efficient an isolator is at isolating unwanted vibration, let's use the natural frequency and damping properties of the following example:

> The natural frequency of an isolator is 10.0 Hz (F_D / F_N = 1) with a damping rate of 7% (0.07).
>
> The transmissibility curve with these two values will take the shape of the blue curve in Figure B:

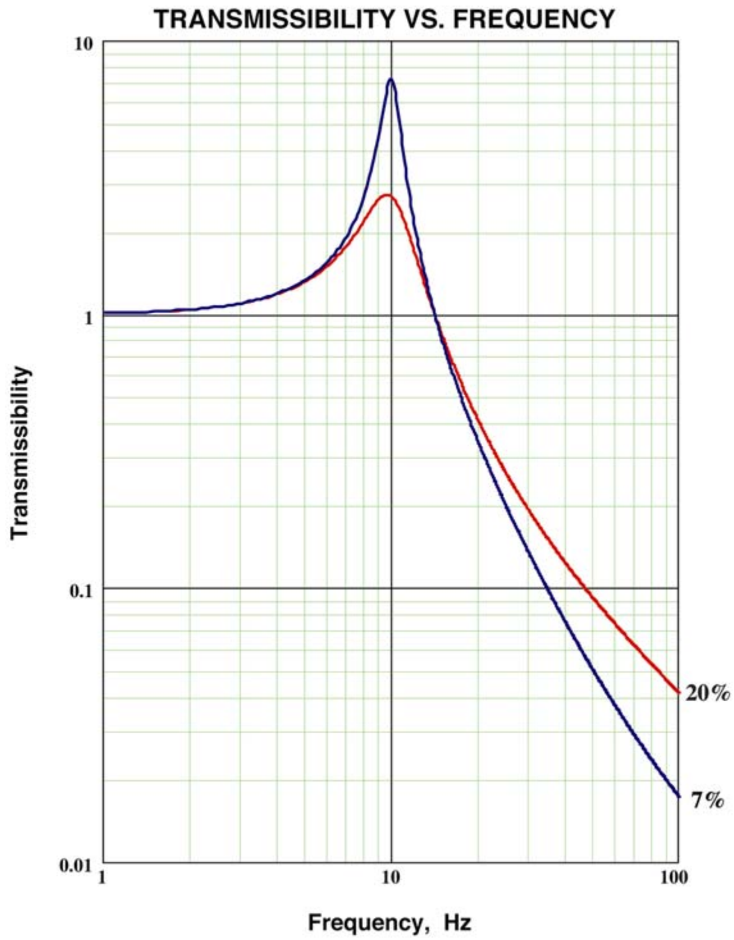

Figure B

Looking at the curve, we can see that the isolator will have an amplification at resonance (10.0 Hz) of 7.2 due to the 7% damping.

It can also be seen that the isolator will begin to isolate disturbing vibration at approximately 14.1 Hz.

As a reference, let's use the resultant transmissibility at 30 Hz (F_D / F_N = 3) which is 0.135 or 13.5%. This results in a reduction of vibration (isolation) at 30 Hz of 1 - 0.135 or 86.5%.

What happens if we add damping to the isolator? Let's change the damping rate to 20% and see how this effects the isolation and transmissibility. (See the red curve in Figure B.)

With a damping rate of 20%, transmissibility at resonance is now only 2.6. However, by adding damping, the transmissibility at 30 Hz is now 0.193 or 19.3% resulting in a reduction at 30 Hz of 80.7%.

Shock Isolation

Introduction

The purpose of shock isolation is to limit the forces transmitted to the surroundings of the equipment in which shock originates. Shock can be defined as a transient condition where a single impulse of energy produced by a force is transferred to a system in a short period of time and with large acceleration.

The reduction of shock is obtained by the use of isolators which results in the storage of the shock energy within the isolator and its release of the energy over a longer period of time. The energy storage takes place by the deflection of the isolator.

The effectiveness of a shock isolator is measured not by transmissibility (as with vibration isolators) but by transmitted force through the isolator and its corresponding deflection.

There are two classes of shock to consider:

1. Shock characterized by motion of a support where a shock isolator reduces the severity of the shock experienced by the equipment on the support.
2. Shock characterized by forces applied to, or originating within, a machine where an isolator reduces the severity of shock experienced by the support.

As mentioned in the introduction, a shock isolation system receives and releases energy over a period of time greater than what would have been observed had a resilient isolator not been applied.

The shock isolator releases the energy much more slowly than it receives and stores it. As a result, the output of the shock is not as high as the input. A simple example of an impact or transient condition is a rigid object of a given weight being dropped through a vertical distance onto a floor.

The maximum force that would be transmitted into the floor depends upon the deflection of the floor. The smaller the deflection in the floor, the larger the impact force created will be.

In order to determine this impact force, the kinetic energy of the object the instant before it contacts the floor must be calculated. This kinetic energy is equal to the weight of the object multiplied by the distance through which it falls. The floor must absorb this kinetic energy in bringing the object to rest after impact.

For a linear isolator, the allowable force transmitted after isolation can be calculated if the desired dynamic displacement is known and the spring rate of the isolator is known by using Equation 6.

Equation 6 $F=KX$

Conversely, if the allowable force is known, then the dynamic deflection of the isolator can be found by using various spring rates.

Now, we must check to see if a practical isolator exists given what we'd like to have for a dynamic displacement and/or transmitted force.

Assuming we have a known allowable transmitted force of 1,000 lbs after impact, and we want no more than 1" of deflection on a shock isolator, use Equation 6 (above) to solve for the isolator's spring rate (K), 1,000 lbs/in.

To solve for the required natural frequency, assume a velocity of 7 in/sec.

$$1/2KX^2 = 1/2MV^2$$

So:

$$\frac{K}{M} = \frac{V^2}{X^2}$$

If the velocity of the falling weight is 7 in/sec, then:

$$\frac{K}{M} = \frac{49}{1} = 49$$

$$F_N = \frac{1}{2\pi}\sqrt{\frac{K}{M}}$$

So:

$$F_N = \frac{1}{2\pi}\sqrt{49} = 1.12 \text{ Hz}$$

Again, this method assumes no damping and predicts the static natural frequency of a shock isolator being dynamically displaced having a displacement of 1" maximum.