Engineering Guide

Introduction

Mechanical vibrations and shock are present in peoples everyday life. These disturbances can range from small office vibrations to heavy ballistic shock and the adverse effect of this disturbance depend on the fragility level of the equipment.

These vibration environments can range in levels from simple foot traffic in an office environment or heavy seismic disturbances that can effect sensitive equipment such microchip equipment etching, where any adverse vibration can effect the accuracy of the machine.

Other external vibration produced by vehicles, trucks, trains, air conditioners, generators, pumps, etc., can cause adverse responses in sensitive machinery or equipment and produce negative or erroneous results.

In the office environment disturbances from fans and AC units can transmit noise and vibrations to the surrounding structure and produce an unhealthy level of noise and fatigue in the work environment .

Non stationary products subject equipment to much higher shock and vibration that stationary applications. Vibrations from engines, pumps and equipment are present in air, sea and on the road as well as shock and vibration effects from the medium they travel on.

Some of the disturbances from rough roads, impart severe transients shock and vibration to the vehicles traveling on them. In addition to rough seas, naval ships are also subjected to very severe mechanical shock from depth charges and other explosions.

Disturbance elimination techniques from shock and vibration isolators have been designed to provide protection to all types of equipment.

Three main elements:

1. The equipment that needs to be isolated.
2. The support structure that connect the isolator to the equipment.
3. The resilient member or the vibration and shock isolator.

If the equipment is the source of the vibration or shock, the isolator should be designed to reduce the force transmitted from the equipment to the support structure. This is illustrated in **Figure 1**, where M represents the mass of equipment, which is the vibrating source, and K is the spring or isolator, which is located between the mass and the support structure,

If the support structure is the source of the vibration or shock, the purpose of the isolator is to reduce the disturbance transmitted from the support structure to the equipment. An example would be protecting delicate measuring instruments from vibrating floors. This illustrated in **Figure 2**, where M represents the mass of a instrument which is protected by the isolator K from a vibrating floor.

In each case, the principle of isolation is the same. The isolator, a resilient element, stores the incoming energy like a spring based on a discreet time interval like a (dashpot) which reduces the disturbance to the equipment or support structure.

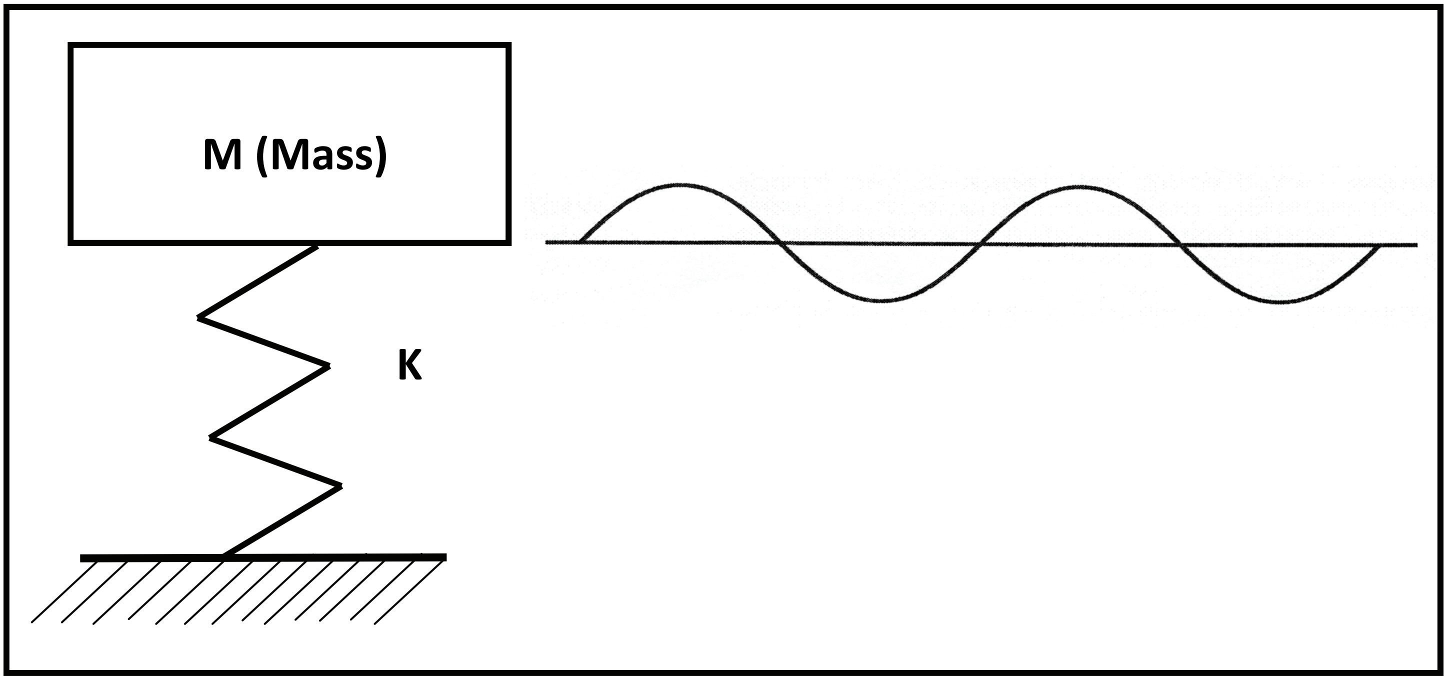


Figure 1. *Schematic of a single degree of freedom dynamic system where the mass, M, is the vibratory source.*

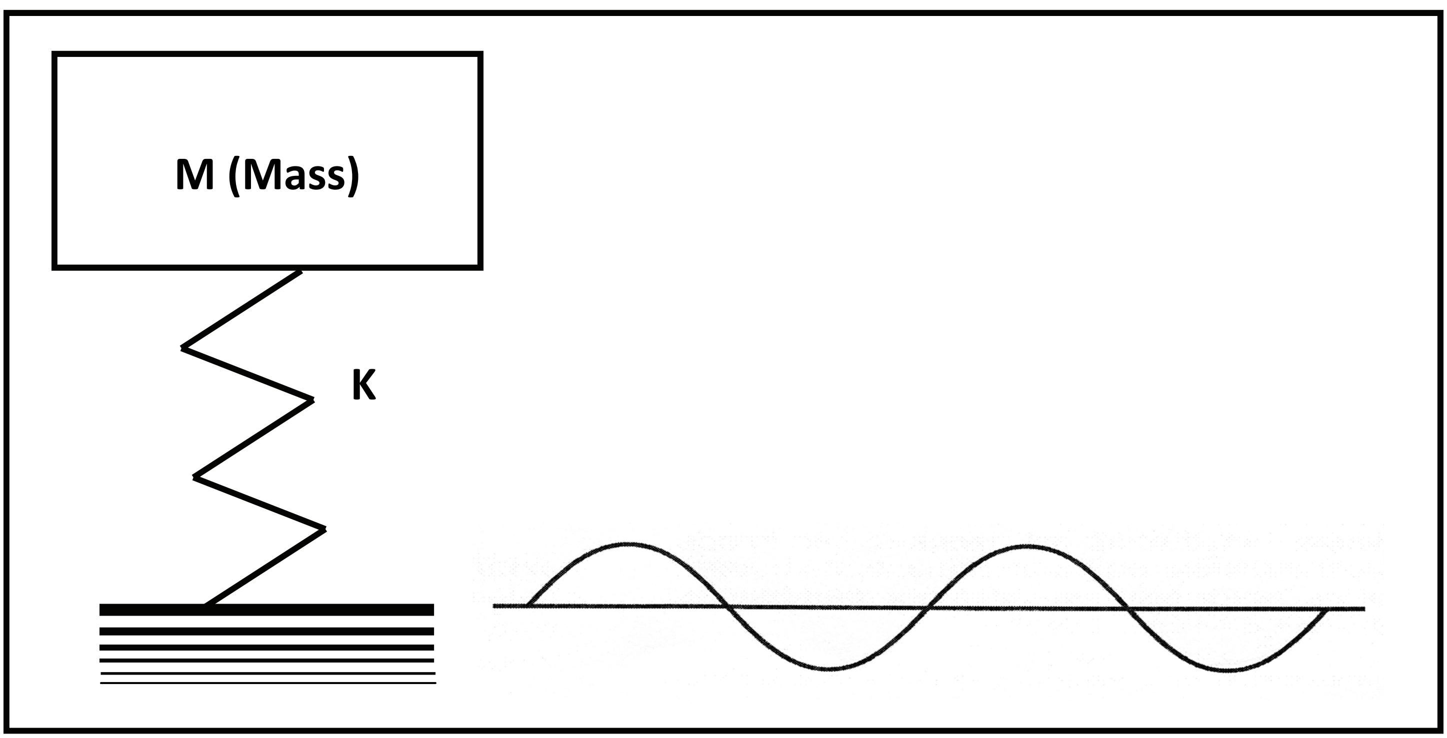


Figure 2. *Schematic of a single degree of freedom dynamic system where the floor is the vibratory source.*

TERMS AND DEFINITIONS

In order to fully understand vibration and shock theory there are a number of terms and definitions that should be understood. The terms and definitions are basic and easy understood to maximize the effectiveness of this engineering guide.

A vibration isolator and a shock isolator are not always mutually exclusive, but for the purposes of analysis must be treated separately. For all practical purposes the environmental conditions will dictate the design of the isolator . A heavy shock will more than likely have a different design than an office vibration environment.

Before any selection of a vibration or shock isolator can be made, the engineer should have a basic understanding of the following definitions, terms and equations:

VIBRATION

A magnitude (force, displacement, or acceleration) which oscillates about some specified reference where the magnitude of the force, displacement, or acceleration is alternately smaller and greater than the reference. Vibration is commonly expressed in terms of frequency (cycles per second or Hz) and amplitude, which is the magnitude of the force, displacement, or acceleration. The relationship of these terms is illustrated in **Figure 3**.

FREQUENCY

Frequency may be defined as the number of complete cycles of oscillations which occur per unit of time.

$$\text{Frequency} = f = \frac{\text{cycles}}{\text{second}}(\text{cps}) = \text{Hertz (Hz)}$$

PERIOD

The time required to complete one cycle of vibration.

$$\text{Period} = \lambda = \frac{1}{f}$$

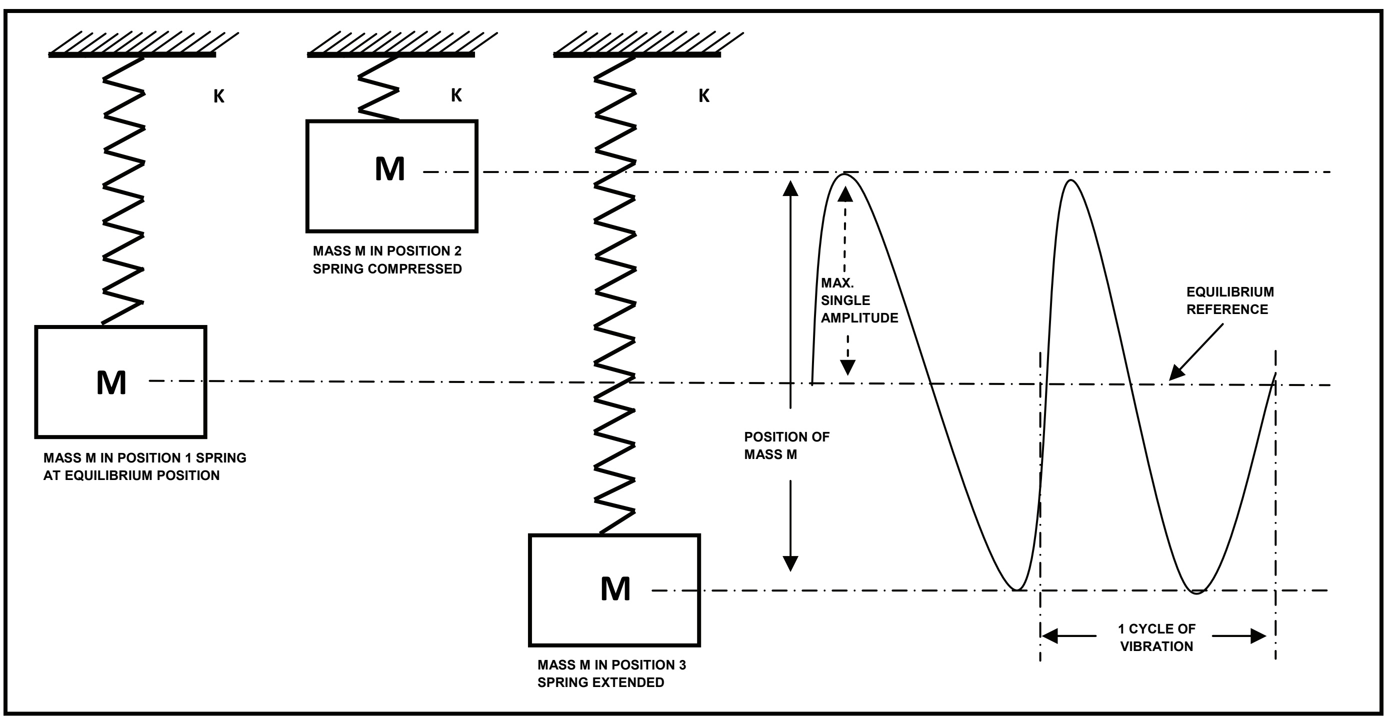


Figure 3. *A Schematic of oscillating spring mass system and vibratory responses.*

FORCING FREQUENCY

The number of oscillations per unit time of a force or displacement applied to a system.

Forcing Frequency = f_d

NATURAL FREQUENCY

Natural frequency may be defined as the number of oscillations that a system will carry out per unit time if its equilibrium position and allowed to vibrate freely. Where K is the spring stiffness and M is the mass and W is the weight. **(See Figure 3)**

$$f_n = \frac{1}{2\pi}\sqrt{\frac{K}{M}} \quad \text{Eq.1}$$

$$f_n = \frac{1}{2\pi}\sqrt{\frac{Kg}{W}} \quad \text{Eq.2}$$

$$f_n = 3.13\sqrt{\frac{K}{W}} \quad \text{Eq.3}$$

Natural frequency in terms of static deflection ΔS:

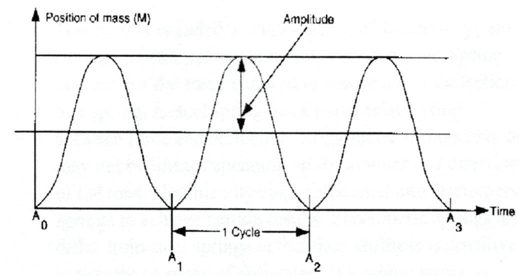
$$f_n = 3.13\sqrt{\frac{1}{\Delta s}} \quad \text{Eq.4}$$

Equations 1 through 4 all neglect the effects of damping. When damping is considered, Equation 2 becomes where C/CC is damping ratio which is specific to the material or structure of damping being used:

$$f_n = \frac{1}{2\pi}\sqrt{\frac{Kg}{W}\left[1-\left(\frac{C}{C_c}\right)^2\right]} \quad \text{Eq.5}$$

AMPLITUDE

The amplitude of a sinusoidal vibration as displacement, velocity, or acceleration is the zero to peak value corresponding to the maximum value of a vibration time-history. **(See Figure 3)**.



DISPLACEMENT

Displacement is a amount movement that specifies the change of the position of a body to an equilibrium position.

VELOCITY

Velocity is a time rate of change of displacement with respect to a frame of reference.

ACCELERATION

Acceleration is a time rate of change of velocity with respect to a frame of reference. These values of acceleration changed with both latitude and elevation based on the point of reference. These valves are measure in G's =386 In/Sec^2, 32 Ft/Sec^ or 9.8 M/Sec^2 which is the standard measured used for the acceleration due to gravity.

DEFLECTION

Deflection is the distance an elastic body or spring will move when subjected to a force, F.

SPRING STIFFNESS

The ratio of the force applied divided by the distance or deflection traveled.

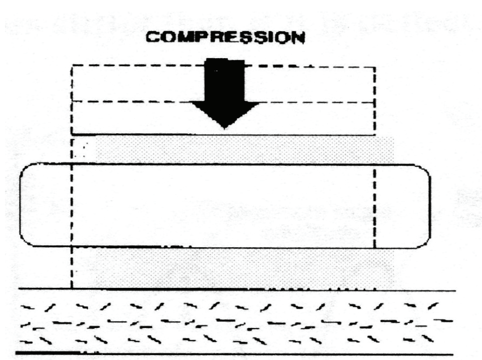
$$K = \frac{\text{Force}}{\text{Deflection}} = \frac{\text{lb}}{\text{in}} \quad \text{Eq.6}$$

ELASTIC CENTER

The elastic center is defined as a single point at which the stiffness of an isolator or series of isolators can be represented by a single stiffness value.

COMPRESSION

A deformation caused by squeezing the layers of an object in a direction perpendicular to the layers.



DAMPING

Damping is when energy is dissipated in a vibratory system. There are three types of damping generally encountered: coulomb, hysteresis and viscous.

COULOMB DAMPING

If the damping force is constant and is independent of the velocity of the system, the system is said to have coulomb damping.

HYSTERESIS (Material) DAMPING

Damping which results from the molecular motion of the structure of a material when that material is subjected to a velocity is referred to as hysteresis damping. Elastomers are examples this type of damping.

VISCOUS DAMPING

If a particle encounters a force which its magnitude is proportional to the magnitude of the velocity of that particle in an opposite direction, the particle is said to be viscously damped. This is the easiest type of damping to model mathematically. All of the equations in this text books via a dashpot are based on use of a viscous damping coefficient.

DAMPING COEFFICIENT

Damping for material is expressed by its damping coefficient.

$$\text{Damping coeff.} = C = \frac{\text{lb.sec}}{\text{in}}$$

CRITICAL DAMPING

A system is critically damped when it is displaced from its original position and returns to its initial static position without any rebounding. The damping coefficient for critical damping can be calculated using:

$$C_c = 2\sqrt{KM} \qquad \text{Eq.7}$$

DAMPING FACTOR

The non-dimensionless ratio which defines the amount of damping in a system.

$$\text{Damping factor} = \frac{C}{C_c} = \zeta$$

DUROMETER (HARDNESS)

A numerical value which measures the resistance to the penetration of the durometer meter indenter point; value may be taken immediately or after a very short specified time.

FRAGILITY

Is the highest level vibration or shock that a system can stand without any equipment failure.

"G" LEVEL

Is a dimensionless ratio of the shock acceleration level divided the acceleration due to gravity.

ISOLATION

The protection of equipment from vibration or shock. The percentage of isolation required is a function of the fragility of the equipment.

LOAD DEFLECTION CURVE

The measured and recorded displacement of a mounting plotted versus an applied load.

RANDOM VIBRATION

Non-sinusoidal vibration characterized by the excitation of a broad band of frequencies at random levels simultaneously.

RESONANCE

When the forcing frequency equals the natural frequency of a system, this condition is known as resonance.

SET

Is the amount of permanent deformation that is never recovered after removal of a load. It may be in shear or compression.

SHEAR

A deformation caused by sliding layers of an object past each other in a direction parallel to the layers.

TRANSMISSIBILITY

Defined as the ratio of the dynamic output to the dynamic input.

$$T = \sqrt{\frac{1 + \left(2\frac{f_d}{f_n} \cdot \frac{C}{C_c}\right)^2}{\left(1 - \frac{f_d^2}{f_n^2}\right)^2 + \left(2\frac{f_d}{f_n} \cdot \frac{C}{C_c}\right)^2}} \qquad \text{Eq.8}$$

For negligible damping (C/C_C=0), T becomes:

$$T = \left| \frac{1}{1 - \left(\frac{f_d}{f_n}\right)^2} \right| \qquad \text{Eq.9}$$

When resonance occurs, f_d/f_n= 1 and C / C_C= any value, T is at its max and Equation 8 becomes:

$$T_{max} = \frac{1}{2\frac{C}{C_c}} \qquad \text{Eq.10}$$

SHOCK

Movement in which there is a sharp and abrupt change in velocity. Examples of this are an explosion or a package falling to the ground.

SHOCK PULSE

A shock pulse is the transmission of kinetic energy to a system which happens in a very short time. This pulse is then followed by a natural decay in motion. Shock pulses are normally displayed graphically as acceleration vs. time curves. **See Figure 11**

SHOCK TRANSMISSION

This can be calculated with the following equation:

$$\text{Shock transmitted} = G_T$$

$$G_T = \frac{V\left(2\pi f_n\right)}{386} = \frac{V\left(f_n\right)}{61.4} \qquad \text{Eq.11}$$

In this equation, V is the instantaneous velocity of the shock and fn is the natural frequency of the system.

The dynamic linear deflection of an isolator under the shock pulse can be determined by the use of the following equation:

$$\Delta_D = \frac{V}{2\pi f_n} \qquad \text{Eq.12}$$

Design Considerations

VERTICAL VIBRATION

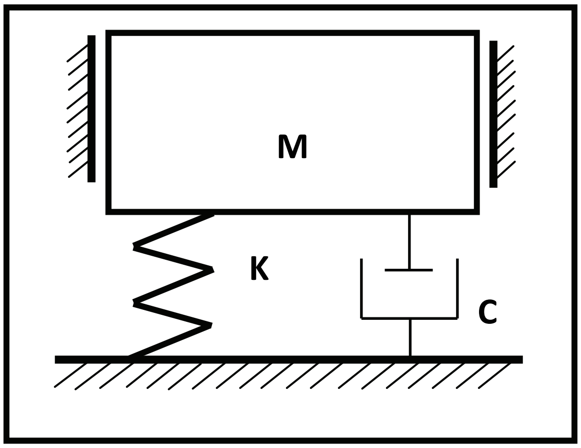


Figure 4. *Schematic of the simplest form of an isolator, a spring, K, and a viscous damper, C, supporting the equipment mass, M.*

The isolator may be best understood by first reducing it to its simplest form. The system of **Figure 4** includes a rigid body mass *M* supported by a spring *K* that is constrained to move only in vertical direction without any rotation. A dashpot *C* is arranged in parallel with the spring between the support and the mass. The mounted equipment is represented by the mass M , while the spring and dashpot represent the visco-elastic properties of a conventional isolator. The simple system shown in Figure 4 is said to be a single-degree-of-freedom system because it can only move in a positive or negative vertical direction.

Isolation is maintained by the proper relationship between the disturbing frequency and the system's natural frequency. The natural frequency, or more properly, the natural frequency of the system consists of isolator and mounted equipment.

$$f_n = \frac{1}{2\pi}\sqrt{\frac{Kg}{W}\left[1-\left(\frac{C}{Cc}\right)^2\right]} \qquad (Eq.6)$$

A critical damped system returns to it original position without any oscillation if displaced; it has no natural frequency , $C=C_C$ in **Equation 6**.

In most real life circumstances the value of the damping coefficient C is relatively small. The influence of damping on the natural frequency may then be neglected. Setting the damping coefficient *C* equal to zero, the system becomes an undamped single-degree-of-freedom system, and the undamped natural frequency given by:

$$f_n = \frac{1}{2\pi}\sqrt{\frac{Kg}{W}} \qquad (Eq.2)$$

Static deflection often is used to define the characteristics of an isolator. Static deflection is the deflection of the isolator under the static load of the mounted equipment.

Referring to **Equation 2** and substituting g =386 in/sec^2, $W / K = \Delta_s$, the following expression is obtained for natural frequency in terms of static deflection:

$$f_n = 3.13\sqrt{\frac{1}{\Delta_s}} \qquad (\ Eq.4\ )$$

A graphic portrayal of **Equation 4** is given in **Figure 5**. It thus appears possible to determine the natural frequency of a single-degree-of-freedom system by measuring only the static deflection.

This is true under two circumstances

1) The spring must have a linear load vs deflection curve.

2) The static to dynamic stiffing factor must be one. The isolator must have the same stiffness statically as it does dynamically.

The dynamic modulus of elasticity of elastomeric materials is higher than the static modulus. Since the modulus is higher dynamically the natural frequency of the isolator is thus somewhat greater than that calculated on the basis of the static deflection alone.

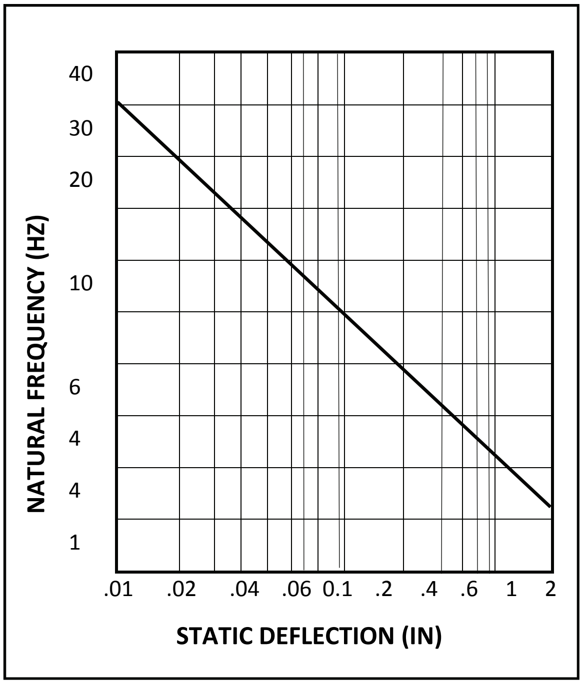


Figure 5*. Graph of the natural frequency and static deflection of a linear, single-degree-of-freedom system.*

The dynamic stiffness and natural frequency may be determined when the isolator is vibrated based on a known load and calculating the dynamic stiffness from **Equation 2**.

The efficiency of isolators in reducing vibration is indicated by the transmissibility of the system. **Figure 6** illustrates a typical transmissibility curve for an equipment of weight *W* supported on an isolator with stiffness *K* and damping coefficient *C* which is subjected to a vibration disturbance of frequency f_d. When the system is excited at its natural frequency, the system will be in resonance and the disturbance forces will be amplified rather than reduced. Therefore, it is very desirable to select the proper isolator so that its natural frequency will be excited as little as possible in service and will not coincide with any critical frequencies of the equipment.

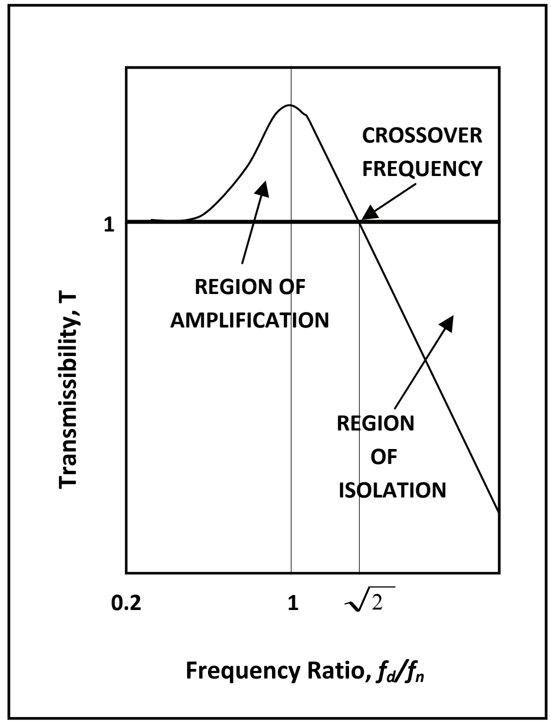


Figure 6. *Transmissibility curve for an isolated system where f_d = disturbance frequency and f_n = system natural frequency.*

In **Figure 6**, when the ratio of the disturbing frequency f_d over the natural frequency f_n is less √2 than the transmissibility is greater than 1, or the equipment experiences amplification .

$$f_d / f_n \leq \sqrt{2}\ , T \geq 1$$

isolation begins when:

$$f_d / f_n = \sqrt{2} \ \text{(at this point } T = 1\text{)}$$

$$f_d / f_n > \sqrt{2}\ , T < 1$$

DAMPING

The majority of isolators have varying degrees of damping levels depending on the material used and the construction of the isolator. **Table 1** shows the various levels of damping factor C/CC in different materials. Damping is important when the isolation system is operating near resonance because it helps to reduce transmissibility. An air compressor mounted on steel springs which possess very little damping, upon start up and shut down the disturbing frequency of the compressor will at some point correspond with the natural frequency of the spring-mass system. With lightly damped system, the forces from the compressor to the support will be very large and the transmissibility will be very high. An elastomeric isolator which has a higher degree of damping, the amplification at resonance would be much less, but there are always trade offs.

The correlation between a high damped and a lightly damped system is shown in **Figure 8**. This figure shows that as damping is increased, isolation efficiency is reduced in the isolation region. While high values of damping cause significant reduction of transmissibility at resonance, its effect in the isolation region is only a small increase in transmissibility.

Material	Approx. Damping Factor C/C_c	T_{max} (approx.)
Steel Spring	0.005	100
Elastomers:	-	-
Natural Rubber	0.05	10
Neoprene	0.05	10
Butyl	0.12	4.0
Hi Damp Silicone	0.15	3.5
Polybutadiene	0.11	4.5
SBR	0.08	6.0
Friction Damped Springs	0.33	1.5
Metal Mesh	0.12	4.0
Air Damping	0.17	3.0
Felt and Cork	0.06	8.0

Table 1. *Damping factors for materials commonly used for isolators*

The curves which relate f_n, f_d, transmissibility and damping are shown in **Figure 8**.

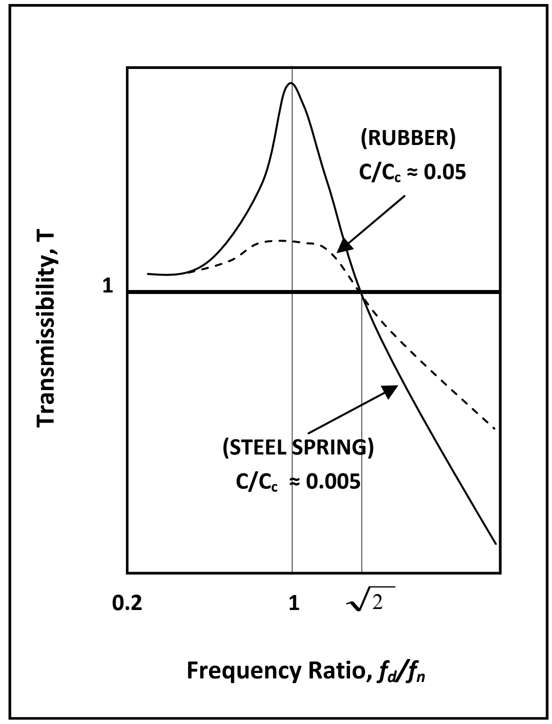


Figure 7. *Typical transmissibility curves for highly and lightly damped systems.*

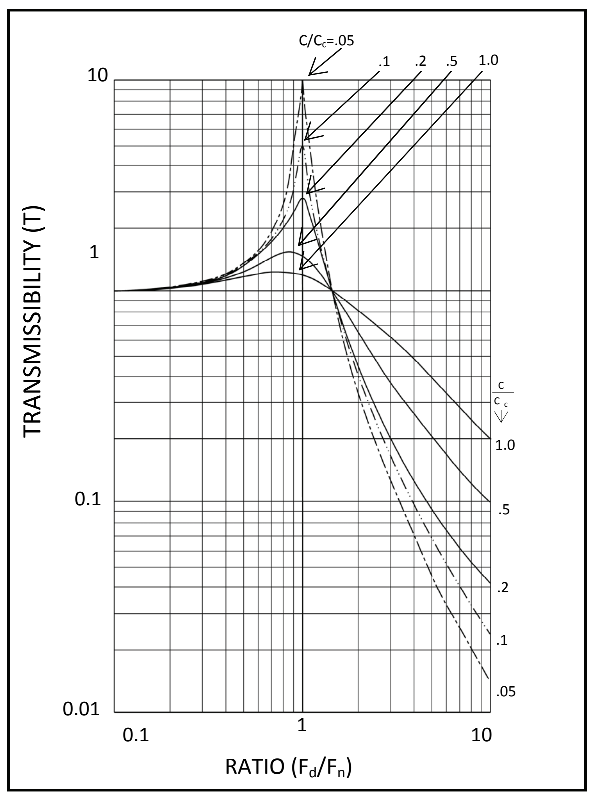


Figure 8. *Family of transmissibility curves for a single degree of freedom system.*

SHOCK

Shock is normally a transient event while vibration is a steady-state condition. A shock input is normally characterized by its peak amplitude in g's and a period that is normally expressed in milliseconds. The shock pulse will typically have a prescribed shape to it that is normally in the form of a half sine, sawtooth, etc., random shaped waveforms are shown in **Figure 9**.

There are a number of different types of shock pulses encountered in the real world. There are different shock tests that are associated with the environment that the equipment will encounter during its lifetime. Equipment installed in aircraft and helicopters is normally tested on a free-fall shock machine which will generate either a half-sine or sawtooth shaped pulse with a typical period of 11 milliseconds at 15 G input. Large shock pulses due to explosive shocks are 6-millisecond sawtooth at 100 g's. Navy vessels the normal test will be the hammer blow specified in MIL-S-901, which exhibits a velocity shock of approximately 100 in./sec. Transit cases or shipping containers are normally tested by dropping the container on a concrete floor. These drop tests can be done to simulate dropping the equipment on edge, flat side and corner drops. These types of tests simulate the shock pulse which will be encountered in the environment of the equipment.

The attenuation of shock inputs is very different from that of a vibration input. The shock isolator is characterized as an energy absorbing device, with a very steep wave front that is absorbed by the isolator. This energy is stored in the isolator and released at the natural frequency of the spring-mass damper system.

The most common equations for predicting shock isolation are in **Figure 9**, for determining the velocity, and **Equation 11**, for calculating transmitted accelerations.

The two methods for solving shock problems are valid as long as two criteria are met:

1) The shock pulse is fully defined, acceleration levels, the time history and the shape of the curve; and
2) The isolation system must respond to the shock event in the linear portion of the load verse deflection curve.

STRUCTURE-BORNE NOISE

By today's standard equipment and products are required to run faster and produce more at a very high rate. These higher rates cause higher noise and vibration to occur and must be dealt with to reduce overall fatigue of the components. High frequency vibrations can occur by this rapid movement of these mechanical or electromechanical components and cause structures that they are mounted to vibrate and produce noise. The best way to reduce the noise and vibration is to de-couple the vibration from the structure by using an elastomeric material. The elatomeric material will absorb the mechanical vibration forces to the structure and therefore reduce the overall noise. Constrained Layer Damping products can be applied to structures to reduce the overall noise signature very effectively. CLDM and other damping products have high levels of inherent damping and can be produced in various shapes.

PROPERTIES	NATURAL RUBBER	NEOPRENE	HI-DAMP ®SILICONE	POLYBUTADIENE
Adhesion to Metal	Excellent	Excellent	Good	Very Good
Tensile Strength	Excellent	Excellent	Good	Excellent
Tear Resistance	Good	Good	Fair	Good
Compression Set Resistance	Good	Fair	Fair	Good
Damping Factor C/C_C (approx.)	0.05	0.05	0.15	0.12
Operating Temperature (max)	200F	200F	300F	200F
Stiffness Increase (approx.)@ -65F	10X	10X	<2X	2X
Oil Resistance	Poor	Good	Fair	Fair
Ozone Resistance	Poor	Good	Excellent	Fair
Resistance to Sunlight Aging	Poor	Very Good	Excellent	Good
Resistance to Heat Aging	Fair	Good	Excellent	Good
Cost	Low	Low	High	Moderate

Table 2. *Relative properties of elastomers used as the resilient media for isolators.*

NONLINEAR ISOLATORS

Up to now we have assumed that all the isolation systems have a linear response, thus there load verse deflection curves are linear in shape. The theory is sound, most systems have steady state vibrations were amplitudes are small. Non linear systems are used where the reverse is true high transit vibration levels or high shock loads are present and space constraints are a premium. The level of isolation is proportional to the isolators ability to accommodate the required deflection due to a heavy static load with an imposed high transit shock. Linear isolators require a high level of deflection to absorb the same transit condition than non linear isolators and this space may not be available in the application. There are a couple of different techniques that can be applied to produce a non linear isolation system.

1) The first is to increase the stiffness of the isolator in proportion as the deflection increases. The amount of deflection will be limited, and produce a higher G level imparted to the equipment.

2) The second method is to design the isolator to buckle as shown in **Figure 10.** The isolator is stiff as the linear portion of the curve and then has a relatively constant load over the higher deflection ranges. Since isolators are an energy absorbing device the style isolator can store more energy for a given deflection based on the area under the load verse deflection curve.

Isolators and Materials

Isolators can be produced from a variety of materials that are both elastomeric and also combinations of spring with other mediums like air or friction. Each selection of materials should be chosen by the designer in accordance with the application and environmental conditions.

ELASTOMERIC ISOLATORS

Elastomers make excellent shock absorbing isolators because the damping level can be tailored to the application and they have high energy absorbing capacity. Elastomers can be molded using a typical rubber molding technique and constructed in numerous shapes to achieve both linear and non linear isolators to achieve the appropriate shock isolation.

Drift or Creep is a negative occurrence with all elastomeric isolators and must be taken into account when designing isolators. The maximum static strain varies widely, but it may be taken as a conservative limitation that elastomers should not be continuously strained more than 10 to 15% in compression, nor more than 25 to 50% in shear. These rules of thumb are often used to determine the maximum load capacity of a given isolator.

In spite of the limitations of elastomeric materials used in isolators, the overall advantages far outweigh the disadvantages and make elastomers the most highly desirable type of resilient media for isolators.

SPRINGS

Metal springs can be used as vibration isolators. In some cases, these types of isolators work well. Springs lack damping and experience extremely violent motions that occur at resonance. (see “Damping” section and **Figure 8**).

SPRING-FRICTION DAMPER

To add damping in coil springs, friction dampers can be designed in parallel with the load-carrying spring. These types of isolators are widely used in practice. An example of this illustrated in **Figure 11.**

The friction damped spring setup is composed on a set of split circular shoes that have a spring that separates the shoes at a fixed normal force that is applied against the wall of a round aluminum housing. There are two sets of springs that are used to support the static load. The normal force is provided by the weight of the equipment, and damping results from the sliding during horizontal excitations. Transmissibility values of about 2 to2.5:1 are exhibited by using this type of spring/damper combination.

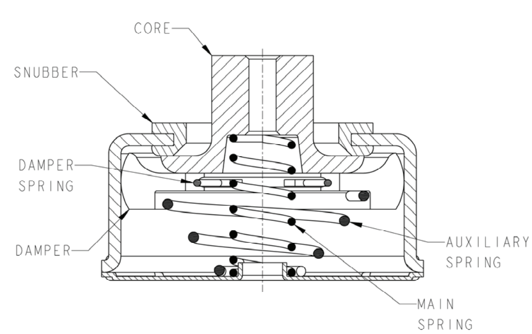


Figure 11. *Isolator using friction damped spring.*

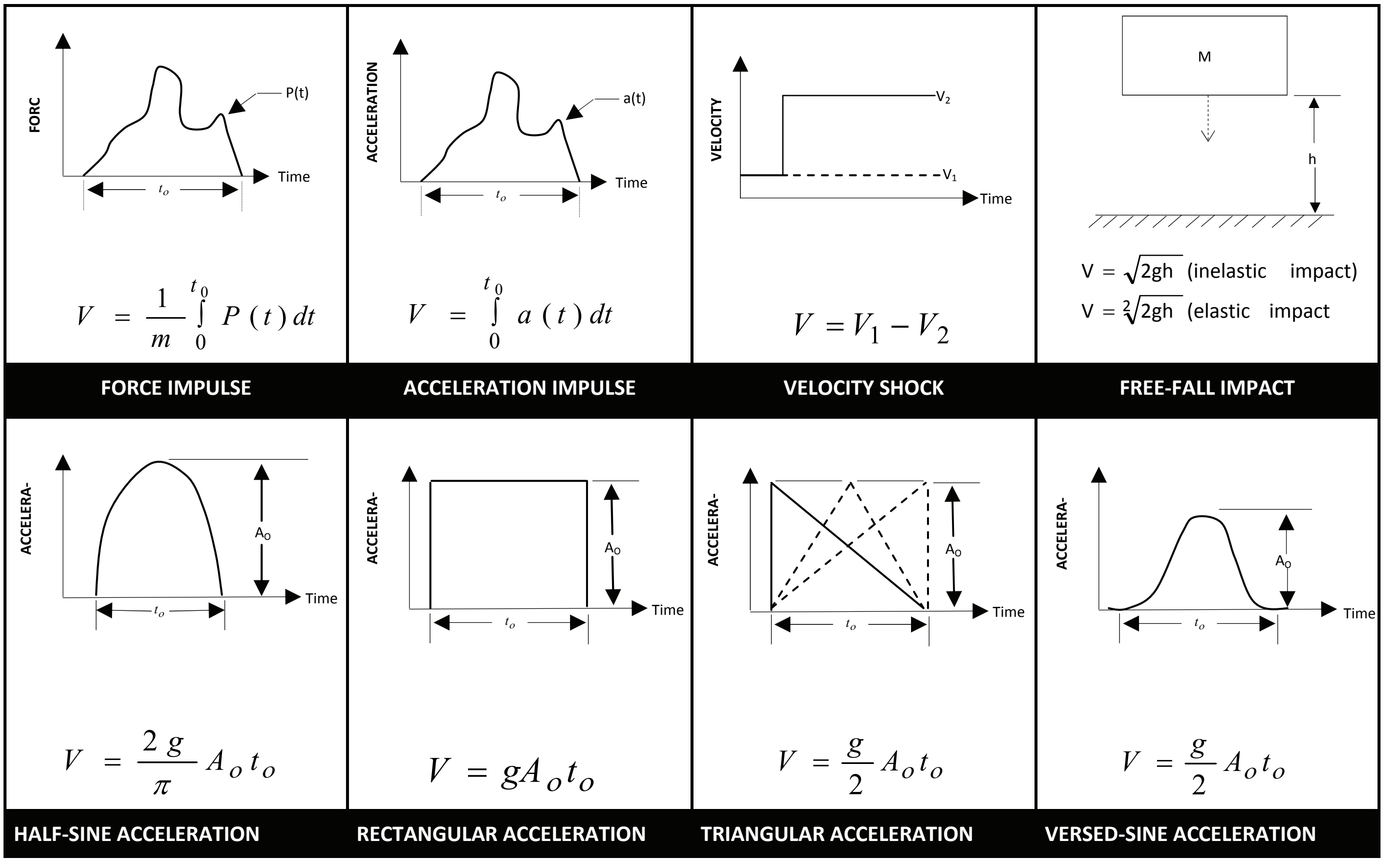


Figure 9. *Shock excitation and the velocity change, V, associated with each shock pulse*

SPRINGS WITH AIR DAMPING

An additional method of adding damping to a spring is by use of an air chamber or balloon with an orifice that meters the air flow. An example of this type of isolator is illustrated in **Figure 12**. A spring is located within the interior of an elastomeric balloon. The air chamber formed at the top of the balloon with a cap which contains an orifice or the force flow metering.

Under excitations the air volume in the balloon passes through a predetermined sized orifice by which damping is closely controlled. Transmissibilities generally under 3.5-4:1 result with this type of design.

Air-damped springs have some significant advantages over similar friction damped designs with respect to isolating low-level inputs.

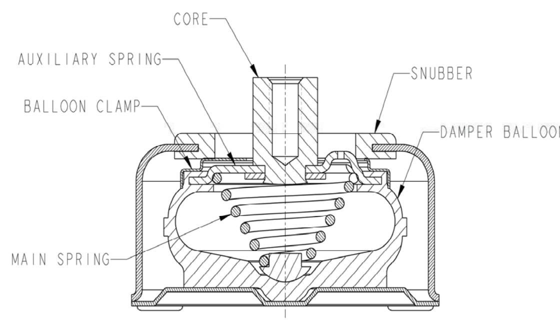


Figure 12. *Isolator using air damped spring.*

With friction damping, the friction force is constant. The damping ratio is effectively increased when the input levels are decreased. Referring to **Figure 8**, increasing the damping ratio decreases the level of isolation.

Friction Damped isolators are suited well for higher vibration levels were air damped systems are well suited for low level vibrations.

WIRE MESH DAMPING

Metal mesh isolators used when there is a high temperature extremes or other environmental factors, damping can be added to a load carrying spring by use of metal mesh inserts. When dynamic loads are applied, the wire mesh strands rub on each other and create friction and thus heat energy is dissipated creating damping. Transmissibilities under 6:1 are generally exhibited by the wire mesh damper.

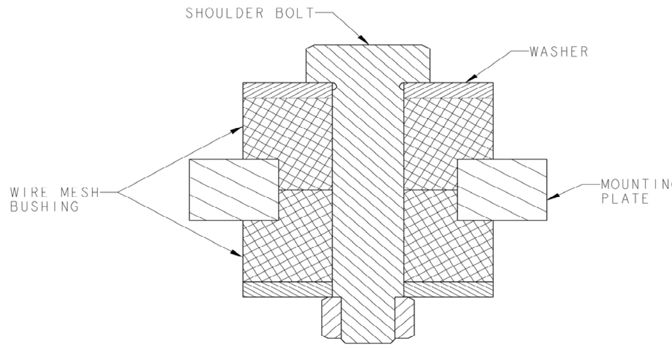
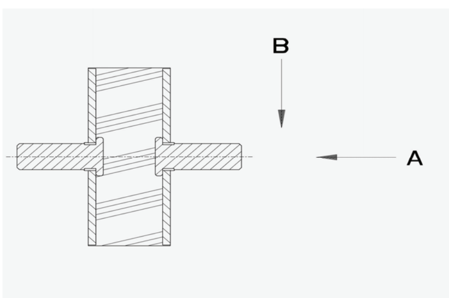
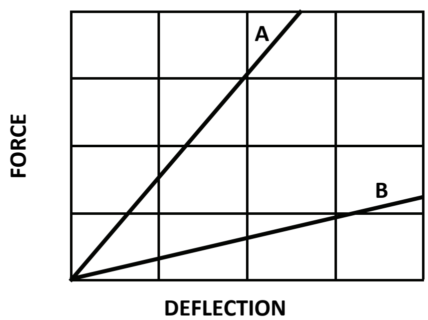
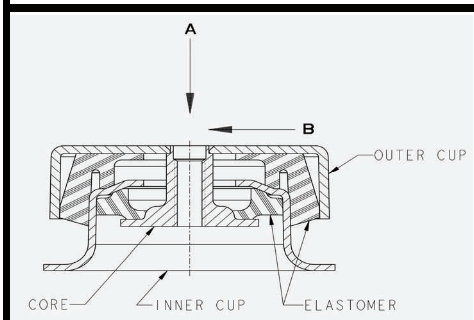


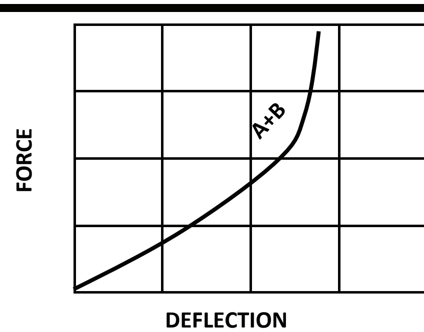
Figure 13. *Isolator with wire mesh load carrying pad.*



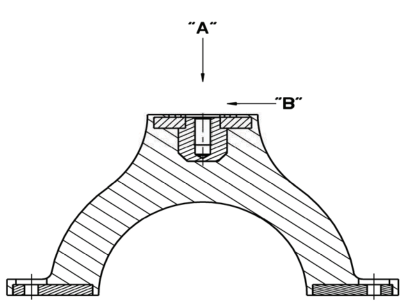


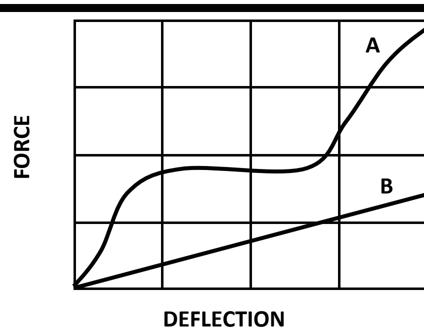
With this type of isolator, the elastomer is strained in compression when the load is applied along "A" direction. Stiffness in any direction perpendicular to the "A" direction, such as the "B" direction, is a function of the shear modulus of the elastomer, and is relatively low compared to compressive stiffness.





This isolator is symmetrical with respect to its central axis. The elastomer sections and cup shapes are designed such that the stiffness is approximately the same in every direction. This is done by loading equal amounts of elastomer in compression in all directions. Also, the mount tends to exhibit stiffening after a certain range of linear deflection.





This isolator supports the static load in the "A" direction. When additional force is applied in this direction, the cylindrical walls of elastomer buckle. Buckling creates the shape of curve "A", which is very effective at attenuating shock forces. Forces in the B direction strain the walls in shear.

Figure 10. *Force vs. Deflection curves for some typical elastomeric isolators.*