ISOLATOR SELECTION GUIDE

Introduction

Mechanical vibration and shock are present in varying degrees in all locations where equipment and people function. The adverse effect of these disturbances can range from negligible to catastrophic depending on the severity of the disturbances and the sensitivity of the equipment.

In one extreme, the vibration environment may consist of low-level seismic disturbances present everywhere on earth, which present operating problems to highly sensitive items such as delicate optical equipment. When other disturbances are superimposed on the seismic disturbances, a wide range of precision equipment is adversely affected.

These other disturbances are caused by such things as vehicular and foot traffic, passing trains, air conditioning systems, and nearby rotating and reciprocating machinery. They cause resolution problems in electron microscopes, disturb other optical systems, cause surface finish problems on precision grinders and jig borers, and hamper delicate work on microcircuitry.

Another concept is the detrimental effect of vibrating internal components of certain equipment such as motors. blowers, and fans in computer or similar systems. These components transmit noise and vibration to the surrounding structure resulting in fatigue, reduced reliability, and a "noisy" product.

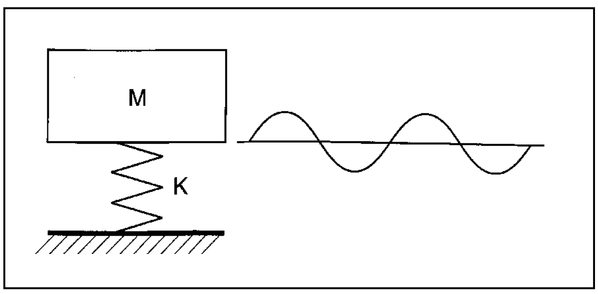
When compared to stationary applications, vehicular installations subject equipment to much more severe shock and vibration. Vibration from a propulsion engine is present in air, sea and road vehicles as well as shock and vibration effect from the media in which they travel.

Such common phenomena as air turbulence and rough roads impart severe dynamic transients to the vehicles travelling on them. In addition to rough seas, military ships are also subject to very severe mechanical shock when they encounter near-miss air and underwater explosions in combat.

Vibration-control techniques in the from of shock and vibration isolators have been devised to provide dynamic protection to all types of equipment.

In discussing vibration protection, it is useful to identify the three basic elements of dynamic systems:

1. The equipment (component, machine motor, instrument, part, etc. ..);
2. The support structure (floor, baseplate, concrete foundation, etc. ..);
3. The resilient member referred to as an isolator or mount (rubber pad, air column, spring, etc.) which is interposed between the equipment and support structure.



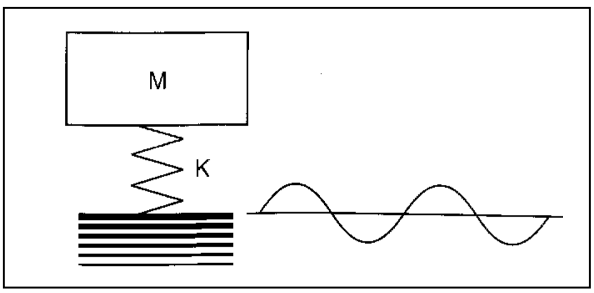


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Figure 1: Active isolation

Figure 2: Passive isolation

In the equipment is the source of the vibration and/or shock, the purpose of the isolator is to reduce the force transmitted from the equipment to the support structure. The direction of force transmission is from the equipment to the support structure. This is illustrated in Figure 1, where M represents the mass of a motor which is the vibrating source, and K, which is located between the motor and the support structure, represents the isolator. This kind of isolation is said to be active.

If the support structure is the source of the vibrating and/or shock, the purpose of the isolator is to reduce the dynamic disturbance transmitted from the support structure to the equipment. The direction of motion transmission is from the support structure to the equipment. This occurs, for instance, in protecting delicate measuring instruments from vibrating floors. This condition is illustrated in Figure 2, where M represents the mass of a delicate measuring instrument which is protected from a vibrating floor by an isolator signified as K. This kind of isolation is said to be passive.
In either case, the principle of isolation is the same. The isolator, being a resilient element, stores the incoming energy at a time interval which affords a reduction of the disturbance to the equipment or support structure.
The purpose of this Design Guide is to aid the design engineer in selecting the proper isolator to reduce the amount of vibration and/or shock that is transmitted to or from equipment.

Definitions

Although vibration isolator will provide some degree of shock isolation, and vice versa, the principles of isolation are different, and shock and vibration requirements should be analysed separately. In practical situations, the most potentially troublesome environment, whether it be vibration or shock, generally dictates the design of the isolator. In other applications, where both are potentially troublesome, a compromise solution is possible.
Before a selection of a vibration and/or shock isolator can be made, the engineer should have a basic understanding of the following definitions, symbols, and terms:

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 (cycled 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.

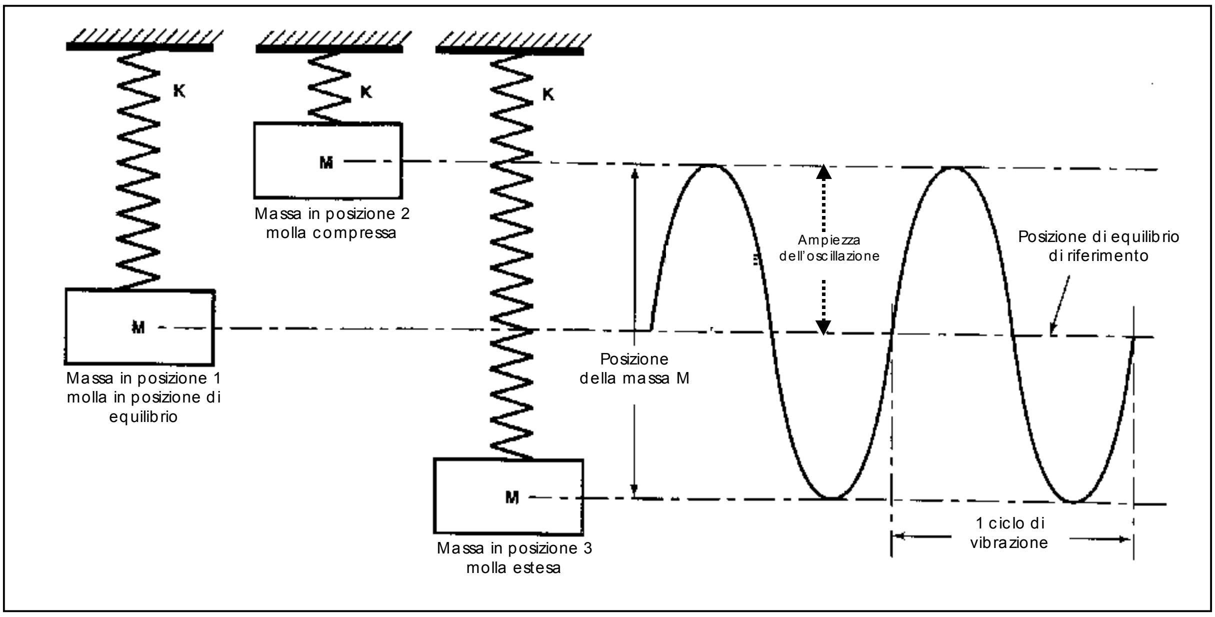


Figure 3: Schematic of free oscillating spring mass system and graphical representation of vibratory responses (no damping).

Frequency

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

$$Frequency = f = \frac{cicles}{\text{second}}(cps) = Hertz(Hz) \quad 1$$

Period

The time required to complete one cycled of vibration.

$$Period = T = \frac{1}{f} \qquad 2$$

Forcing frequency

Defined as the number of oscillations per unit time of an external 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 in unit time (sec) if displaced from its equilibrium position and allowed to vibrate freely (figure 3).

$$f_n = \frac{1}{2\pi}\sqrt{\frac{k}{M}} \qquad 3$$

Natural frequency in terms of static deflection:

$$f_n = \frac{1}{2\pi}\sqrt{\frac{g}{\delta_s}} \qquad 4$$

In previous equations effect of damping has been neglected. When damping is considered, equation 3 becomes:

$$f_n = \frac{1}{2\pi}\sqrt{\frac{k}{M}\left[1-\left(\frac{C}{C_c}\right)^2\right]} \qquad 5$$

Amplitude
The amplitude of a harmonic vibration such as displacement, velocity, or acceleration is the zero to peak value corresponding to the maximum magnitude of a harmonic vibration time-history (figure 3).

Displacement
Displacement is a vector quantity that specifies the change of the position of a body or particle and is usually measured from the mean position or equilibrium position. In general it can be represented by a translation or rotation vector or both (figure 3).

$$displacement = X = mm, cm, ...$$

Velocity
Velocity is a vector that specifies the time rate change of displacement with respect to a frame of reference.

$$velocity = V = \dot{X} = \frac{m}{\sec}$$

Acceleration
Acceleration is a vector that specifies the time rate of change of velocity with respect to a frame of reference. The acceleration produced by the force of gravity, which varies with the latitude and elevation of the point of observation, is given by g=9.81 m/sec^2, which has been chosen as a standard acceleration due to gravity.

$$acceleration = a = \ddot{X} = \frac{m}{\sec^2}$$

$$g = 9.81 \frac{m}{\sec^2}$$

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

$$deflection = \delta = mm$$

Spring stiffness
Described as a constant which is the ratio of a force increment to a corresponding deflection increment of the spring.

$$Spring\ Stiffness = K = \frac{F}{\delta} \qquad 7$$

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

Damping
Damping is the phenomenon by which energy is dissipated in a vibratory system. Three types of damping generally encountered are: coulomb, hysteresis and viscous.

Coulomb damping
If the damping force in a vibratory system is constant and independent of the position or velocity of the system, the system is said to have coulomb or dry friction damping.

Hysteresis damping
Damping which result from the molecular structure of a material when that material is subjected to motion is referred to as hysteresis damping. Elastomers are good examples of materials which posses this type of damping.

Viscous damping
If any particle in a vibrating body encounters a force which has a magnitude proportional to the magnitude of the velocity of the particle in a direction opposite to the direction of the velocity of the particle, 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 are based on use of a viscous damping coefficient. Although most isolators do not use viscous damping, equivalent viscous damping usually yields excellent results when modelling system.

Damping coefficient
Damping (viscous) for a material is expressed by its damping coefficient.

$$Damping\ \ coeff. = C = \frac{N \cdot \sec}{m}$$

Critical damping
A system is said to be critically damped when it is displaced from its static position and most quickly returns to this initial static position without any over-oscillation. The damping coefficient required for critical damping can be calculated using:

$$C_C = 2\sqrt{KM} \tag{8}$$

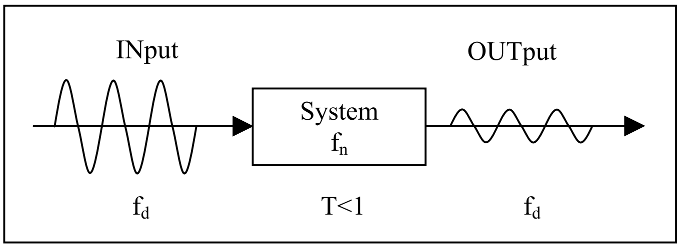
Damping factor
The dimensionless ratio which defines the amount of damping in a system.

$$\zeta = \frac{C}{C_C} \tag{9}$$

Resonance
When the forcing frequency coincides with the natural frequency of a suspension system, this condition is known as resonance.

Transmissibility
Defined as the radio of the dynamic output to the dynamic input of displacements, velocities, accelerations, etc., of the vibrating system:

$$Transmissibility = \frac{OUTput}{INput}$$



Transmissibility in attenuation

$$T = \sqrt{\frac{1+\left(2\frac{f_d}{f_n}\frac{C}{C_C}\right)^2}{\left(1-\frac{f_d^2}{f_n^2}\right)^2+\left(2\frac{f_d}{f_n}\frac{C}{C_C}\right)^2}} \qquad 10$$

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

$$T = \left|\frac{1}{1-\frac{f_d^2}{f_n^2}}\right| \qquad 11$$

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

$$T_{\max} = \frac{1}{2\frac{C}{C_C}} \qquad 12$$

Shock
Defined as a motion in which there is a sharp, nearly sudden change in velocity. Examples of this are a hammer blow on an anvil or a package falling to the ground. Shock may be expressed mathematically as a motion in which the velocity changes very suddenly.

Shock pulse
Shock pulse is a primary disturbance characterised by a rise and decay of acceleration from a constant value in a very short period of time. Shock pulses are normally displayed graphically as acceleration vs. time curves.

Shock transmission
Shock transmitted to the object through the isolator can be calculated with the following equation:

$$Shock\ transmitted = G_T$$

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$$G_T = \frac{V(2\pi f_n)}{9.81}$$

In this equation, V represent an instantaneous velocity shock. Most shock input can be approximated by an instantaneous velocity shock.
The associated dynamic linear deflection of an isolator under shock can be determined by the use of the following equation:

$$\delta_D = \frac{V}{2\pi f_n} \qquad 14$$

Design Considerations

Vertical vibration

In the introduction of this guide, it was pointed out that vibration and shock can have gross detrimental effects on the performance and reliability of a particular product. The vibration which a unit transmit to a supporting structure or the vibration which a unit feels when it is being excited by a vibrating structure can be reduced or attenuated by an isolator if properly selected.

The function of an isolator may be understood by first reducing it to its simplest form, as illustrated in Figure 4. The system of the figure includes a rigid mass *M* supported by a spring *K* and constrained by guides to move only in vertical translation without rotation about a vertical axis. A damper C is arranged in parallel with the spring between the support and the mass. The mounted equipment is simulated by the mass while the spring and damper taken together simulate the elasticity and damping of the conventional isolator. The system shown in Figure 4 is said to be a single-degree-of-freedom system because its configuration at any time may be specified by a single coordinate; *e.g.*, by the height of the mass *M* with respect to the fixed support.

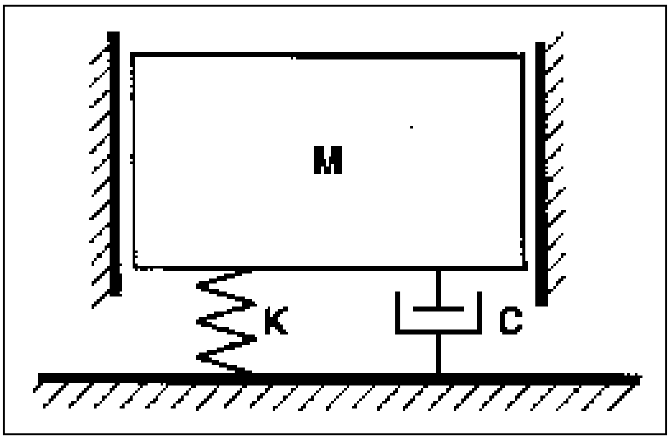

Figure 4: Schematic of the simplest system mass-spring-damper, one degree of freedom.

Isolation is attained primarily by maintaining the proper relationship between the disturbing frequency and the system’s natural frequency. The characteristics of the isolator include its natural frequency, or more properly, the natural frequency of the system consisting of isolator and mounted equipment. In general, a system has a natural frequency for each degree of freedom; the single-degree-of-freedom system illustrated in figure thus has one natural frequency. The expression for the damped natural frequency of the system illustrated in figure 4, expressed in cycles per second (Hz), is:

$$f_n = \frac{1}{2\pi}\sqrt{\frac{k}{M}\left[1-\left(\frac{C}{C_c}\right)^2\right]} \qquad 5$$

A critical damped system returns without oscillation to equilibrium if displaced; it has no natural frequency of oscillation, as indicated by the substitution of $C=C_C$ in equation 5.

In most circumstances the value of the damping coefficient 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:

$$f_n = \frac{1}{2\pi}\sqrt{\frac{k}{M}} \qquad 3$$

This expression is sufficiently accurate for calculating the actual natural frequency in most instances.

The concept of static deflection often is used to define the characteristics of an isolator. Static deflection is the deflection of the isolator under the static or deadweight load of the mounted equipment. Referring to Equation 3 and substituting for K the equation

$$K = \frac{Weight}{\delta_S} = \frac{Mg}{\delta_S} \qquad 7$$

the following expression is obtained for natural frequency in terms of static deflection:

$$f_n = \frac{1}{2\pi}\sqrt{\frac{g}{\delta_s}} \qquad 4$$

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 with certain qualification. First, the spring must be linear, i.e. its force vs. deflection curve must be a straight line. Second, the resilient must have the same type of elasticity under both static and dynamic conditions.

Metallic springs generally meet this latter requirement, but many organic materials used in isolators do not. The dynamic modulus of elasticity of these materials is higher than the static modulus; the natural frequency of the isolator is thus somewhat greater than calculated on the basis of static deflection alone.

Dynamic stiffness may be obtained indirectly by determining the natural frequency when the isolator is vibrated with a known load and calculating the dynamic stiffness from equation 3.

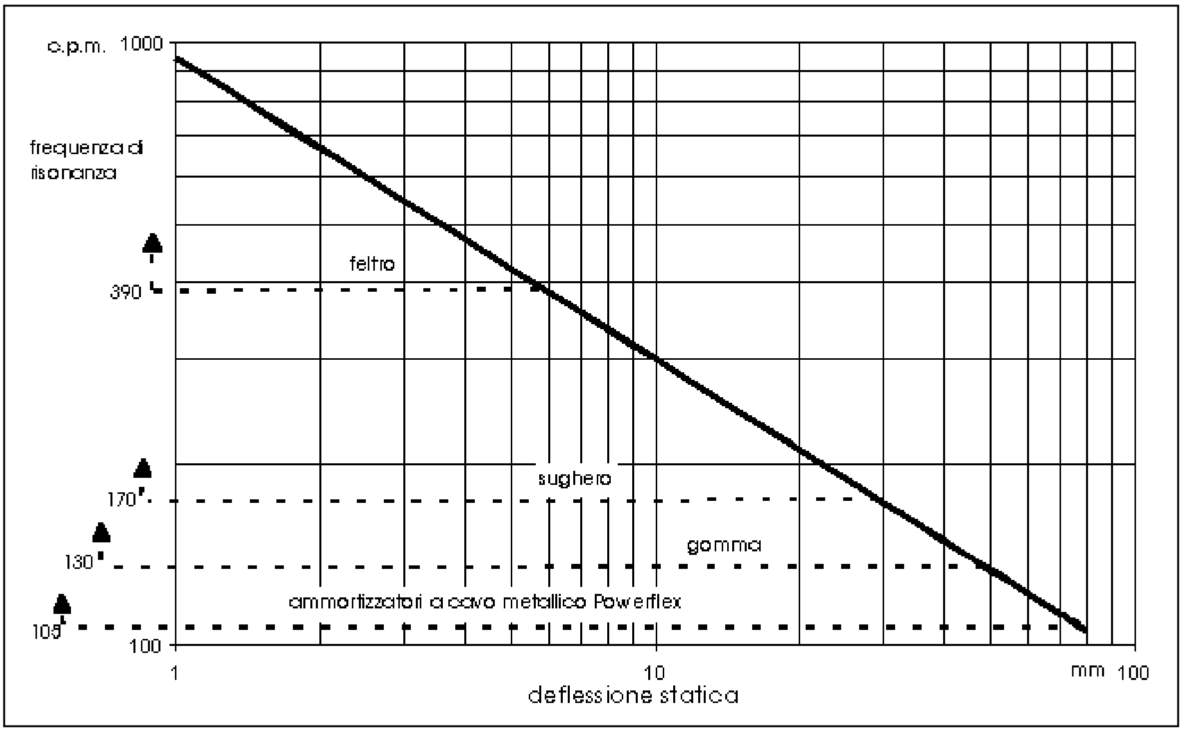

Figure 5: Relation of natural frequency (CPM) and static deflection of a linear, single degree of freedom system.

Effectiveness of isolators in reducing vibration is indicated by the transmissibility of the system. Figure 6 illustrates a typical transmissibility curve for an equipment of mass M 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 (i.e. $f_d=f_n$), 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 non coincide with any critical frequencies of the equipment.
Referring to figure 6, it can be seem that when the ratio of the disturbing frequency f_d over the natural frequency f_n is less than $\sqrt{2}$, or 1.4, the transmissibility is greater than 1, or the equipment experiences amplification of the input. Simply expressed, when:

$$\frac{f_d}{f_n} \leq \sqrt{2} \quad \Rightarrow \quad T \geq 1$$

theoretically, isolation begins when:

$$\frac{f_d}{f_n} = \sqrt{2} \quad (for\ this\ value \quad T = 1)$$

Also it can be seen that when:

$$\frac{f_d}{f_n} \geq \sqrt{2} \quad \Rightarrow \quad T < 1$$

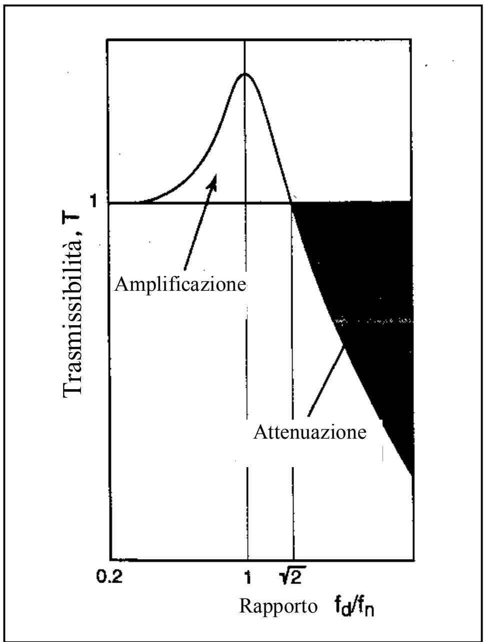

Figure 6: Transmissibility curve for a one d.o.f. system.

the mounted unit is said to be isolated; i.e., the output X_0 in less that input X_1.

DAMPING

The majority of isolators posses damping in varying degrees. A convenient reference illustrating damping factor C/C_C for various materials is shown in Table 1. Damping is advantageous when the mounted system is operating at or near its natural frequency because it helps to reduce transmissibility. For example, consider an internal combustion engine mounted on steel spring which possess very little damping (see Table 1). Upon start up of the engine and as the engine RPM increases, the disturbing frequency of the engine will at some point correspond with the natural frequency of the spring-mass system. With light damping, the build-up of forces from the engine to the support will be very large, that is, the transmissibility will be very high. If the idle RPM of the engines falls in the range of the natural frequency of the spring-mass system, serious damage may result to the engine r the support chassis. If, on the other hand, the designer selects an elastomeric isolator which possesses a higher degree of damping, amplification at resonance would de much less.

Table 1: Damping factors in materials used for isolators

MATERIAL	**C/C_C** (approx.)	**TRANSMISS. MAX** (approx.)
Steel spring	0.005	100
Elastomers:		
Natural rubber	0.05	10
Neoprene	0.05	10
Butyl	0.12	4.0
Friction damped springs	0.33	1.5
Air damping	0.17	3.5
Metal mesh	0.12	4.0
Felt and Cork	0.06	8.0
POWERFLEX dampers	***0.16 ÷ 0.25***	***3.0 ÷ 2.0***

The relationship between a highly damped and a lightly damped system is illustrated in figure 7. This figure shows that as damping is increased, isolation efficiency is somewhat 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. A family of curves which relate f_n , f_d, transmissibility and damping are shown in Figure 8. This family of curves was derived by use of equation 10.

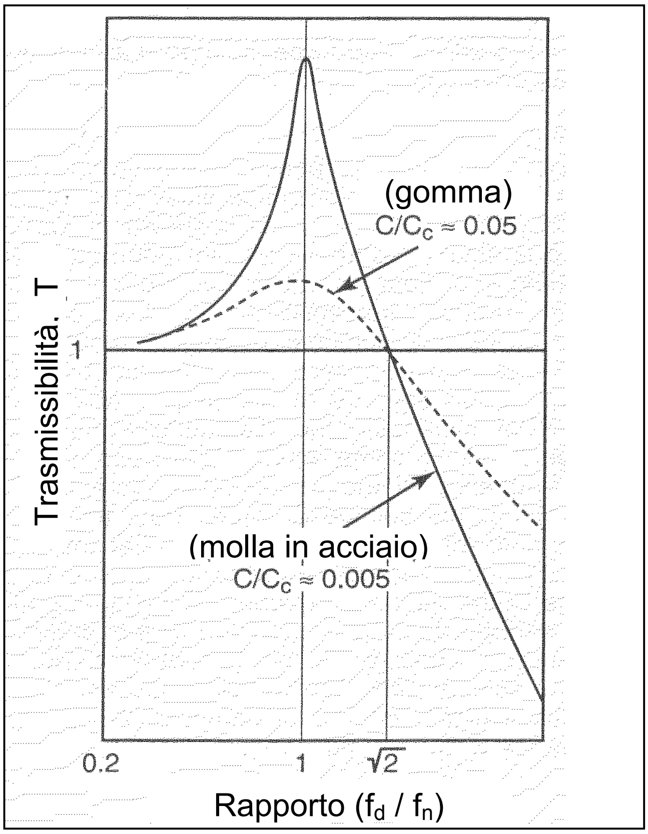

Figure 7: Transmissibility curves for a rubber isolator and a steel spring.

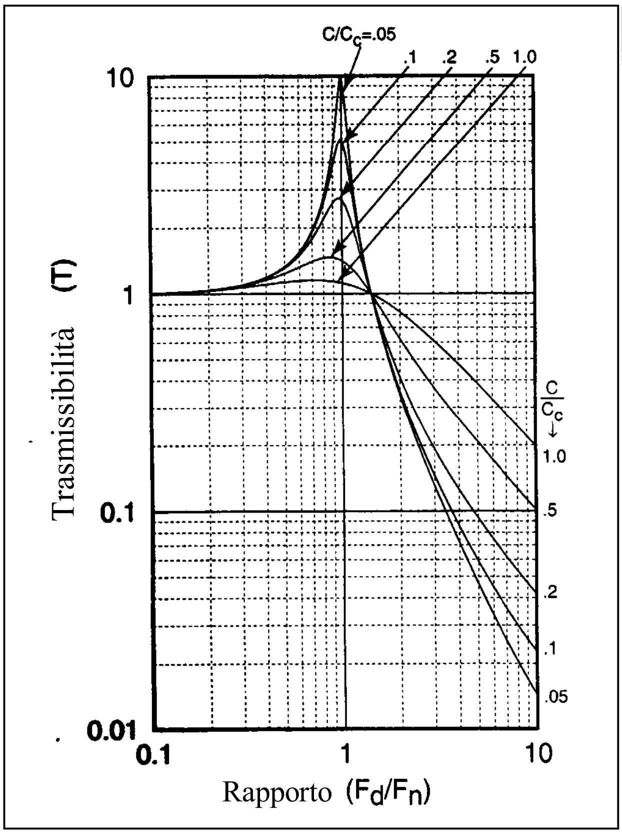

Figure 8: Transmissibility varying with damping factor, 1 d.o.f.

HORIZONTAL VIBRATION

When an isolation system is excited horizontally, two natural frequencies result if the centre of gravity of the unit is not in line with the elastic centre of the isolators. A typical transmissibility curve illustrating this horizontal vibration output illustrated in figure 9. The two natural frequencies which are involved include a lower mode wherein the equipment rocks about a point well below the elastic centre of the isolators and higher mode where the equipment oscillates about a point in the vicinity of the centre of gravity. Two other natural frequencies will occur if the equipment is rotated 90 degrees in the horizontal plane with respect to the exciting force.
Figure 10 can be used to determinate the approximate frequencies of these modes as a function of spring stiffness and equipment dimensions. These curves assume that the equipment is solid, of uniform mass, and that the equipment are attached at the extreme corners. Under horizontal excitation the equipment may be made to translate only by lining up the centre of gravity of the equipment with the elastic centre of the isolators instead of installing the isolators at

the bottom corners of the equipment. In this case, Figure 10 may be applied by letting *H/W*= 0, which result in only one mode of vibration, that of translation. A second mode can only be excited by torsional excitation.

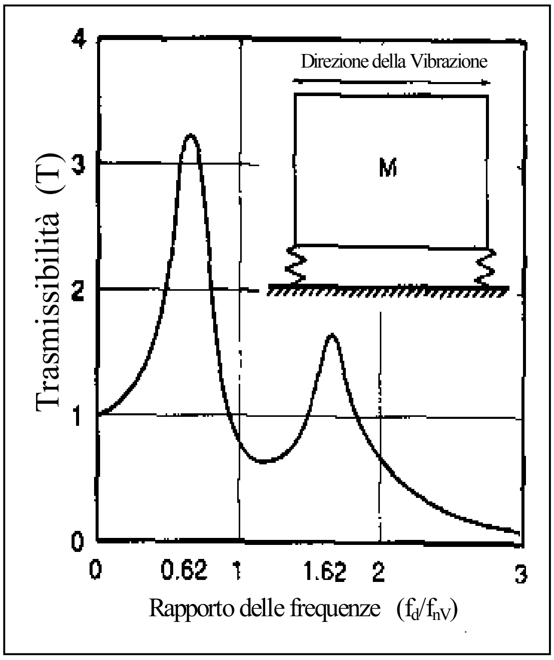

Figure 9: Example of transmissibility curve for horizontal vibrations.

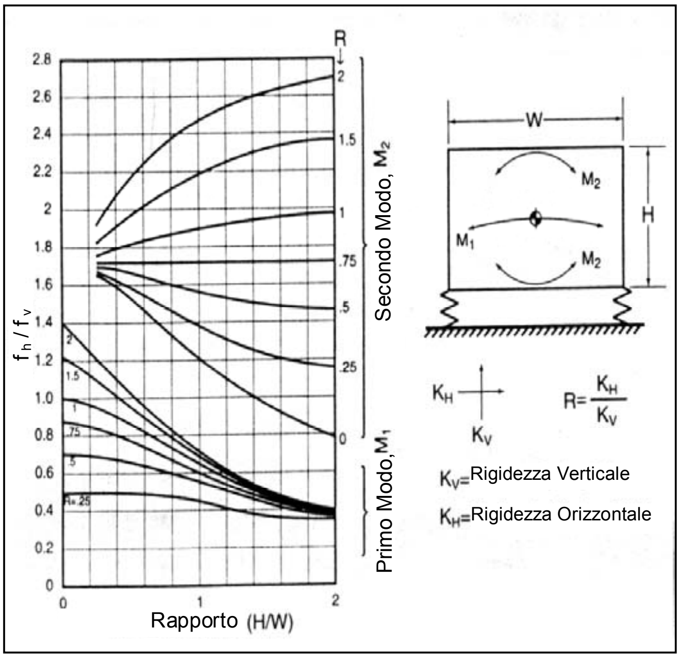

Figure 10: Horizontal natural frequencies for a homogeneous solid mounted on linear, undamped springs at edge of mass.

SHOCK

Shock is normally classified as a transient phenomenon, while a typical vibration input is classified as a steady-state phenomenon. A shock input pulse is normally described by its peak amplitude A expressed in g's, by its duration t normally expressed in milliseconds, and its overall shape, with can take such forms as half-sine, triangular, (initial peak saw-tooth, symmetrical and terminal peak- saw-tooth), versed sine, rectangular, and the form most likely to occur in nature, a more or less random shaped complex waveform force and acceleration impulse as shown in figure 11.

Since there are many types of shock pulses encountered in nature, there are many types of shock test specified for testing a piece of equipment. The different shock test are normally associated with the environment that the equipment will encounter during its lifetime. Equipment installed an aircraft is normally tested on a free fall shock machine which will generate either a half-sine or terminal peak saw-tooth form. A typical test is an 11-millisecond half-sine waveform with a peak acceleration of 15 g's. For components in some areas of missiles where large shock pulses will be felt due to explosive separation of stages, a 6-millisecond saw-tooth at 100 g's may be specified. If a piece of equipment is going on board a Navy vessel, the normal test will be the hammer blow specified in MIL-S-901, which exhibits a velocity shock of approximately 3 m/sec. Shipping containers are normally tested by dropping the container on a

concrete floor, or by suspending it by some suitable support mechanism and letting it swing against a concrete abutment. Other test pertaining to shipment are edge and corner drops from various drop heights. All of these mentioned attempt to simulate the shock pulse which will be encountered in the normal environment of the equipment. These era generally called out by the specific contractual requirements either in a specification or in a work requirement.
The isolation of shock inputs is considerably different from that of a vibration input. The shock isolator is characterised as a storage device wherein the input energy, usually with a very steep wave front, is instantaneously absorbed by the isolator. This energy is stored in the isolator and released at the natural frequency of the spring-mass-system.

The most common procedure for predicting shock isolation is a mathematical approach utilising equations in figure 11, for determining the velocity, and equation 13, for calculating transmitted accelerations.
Another means is through the use of shock transmissibility curves. Shock transmissibility curves are not included in this Guide.
These two methods are valid for solving shock problems provided that the shock pulse is thoroughly defined, and that the isolation system responds in its linear region.

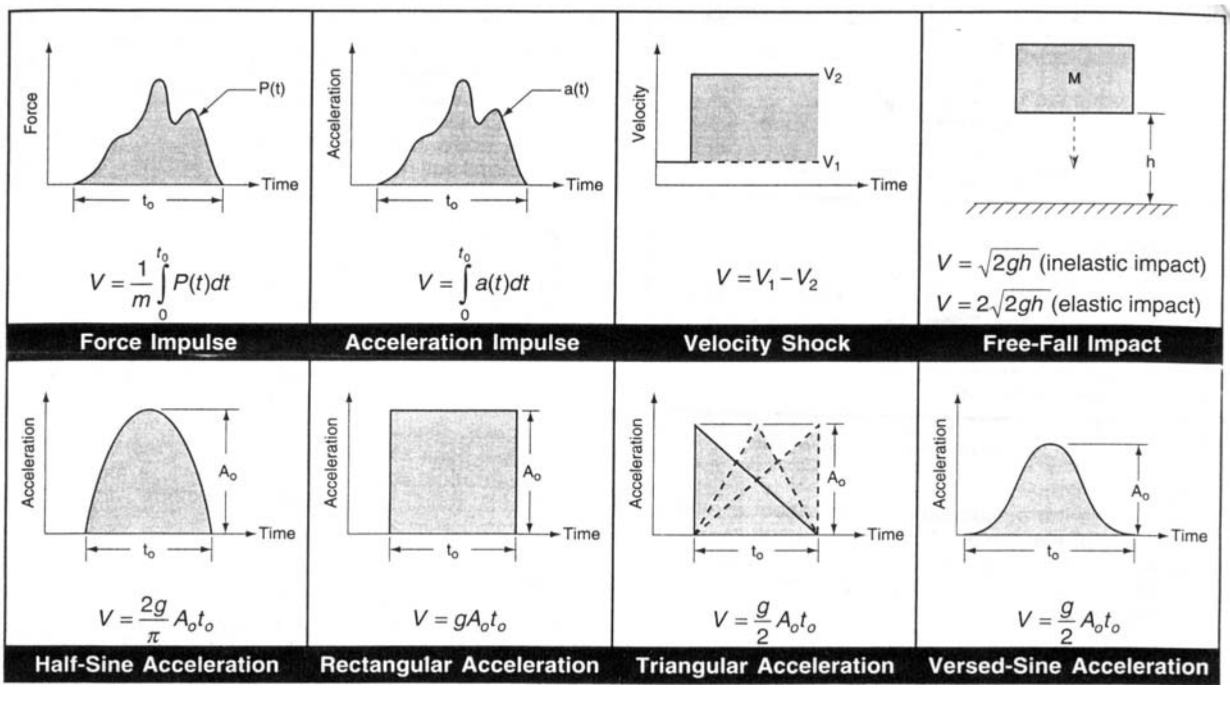

Figure 11: Idealised shapes of shock excitation and related velocity variation, V, associated with each shock pulse.

NONLINEAR ISOLATORS
The preceding discussion of vibration and shock isolation presumes that he isolator is linear, the force-deflection curve for the isolator is a straight line. This simplified analysis is entirely adequate for many purposes. In the isolation of steady-state vibration, displacement amplitude is usually small, and nonlinearity of the isolator tends to be unimportant except where deflection resulting from the static load is relatively great. In the isolation of shock, nonlinearity tends to be more important because large deflections prevail. The

degree of isolation may then be substantially affected by the ability, or lack thereof, of the isolator to accommodate the required deflection.
In many applications of shock isolation, sufficient space is not available to allow for full travel of a linear isolator to reach the desired g out. Therefore, a nonlinear isolator is necessary. There are two types of isolators that can be designed to help solve the problem of insufficient space.
The first solution is to make an isolator that gets stiffer as deflection increases. This will limit the amount of motion, but will increase the acceleration level (g) imparted on the equipment.
The second is to use an isolator that is stiff at small deflection, but gets softer at higher deflections. This is referred to as a buckling isolator, and is shown in figure 12. This allows the isolator to store more energy in the same amount of deflection. A shock isolator is basically an energy storage device; it stores high g-level, short-duration shock and releases them as low g-level, longer-duration shock.

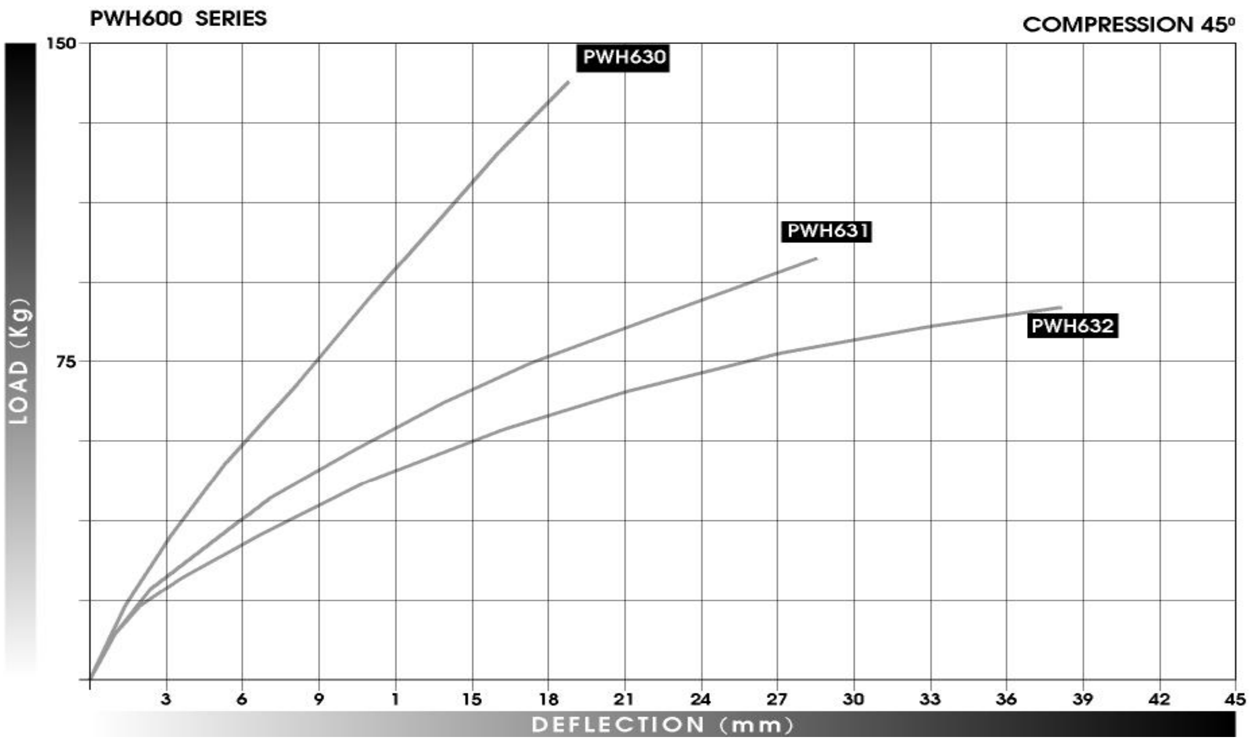

Figure 12: Non linear force vs. deflections curves for POWERFLEX wire cable dampers of the PWH600 series.

STEP-BY-STEP ISOLATOR SELECTION

Step 1: Determinate the frequency of the disturbing vibration, often called the disturbing frequency, f_d . There are a number of ways to determinate the disturbing frequency. For rotating equipment, the disturbing frequency is usually equal to the rotational speed of the equipment, expressed in relations per minute (RPM) or cycled per minute (CPM). If the speed is specified in RPM or CPM, it must be convert to cycles per second (Hz) by dividing by 60.
For other types of equipment, disturbing frequencies must be specified by the manufacturer or measured. Environmental vibrations can also be measured, or are sometimes specified in military or commercial specifications or test reports. There could be more than one disturbing frequency. In this case, one should first focus one the lowest frequency. If the lowest frequency is isolated, then all of other higher frequencies will also be isolated. The most important thing to remember about vibration isolation is that without knowing the frequency of the disturbing vibration, no analytical isolation predictions can be made. In many of these cases, Powerflex S.r.l. can recommend solutions that have worked well in similar past applications. Please contact our Engineering Department at +39.0823.362410, if you need help or advice on your applications.

Step 2: Determinate the max natural frequency, f_n , of the mass-isolators system that will provide isolation. This natural frequency can be calculated by using the following equation:

$$(f_n)_{\max} = \frac{f_d}{\sqrt{2}} \cong f_d \times 0.707 \qquad 15$$

If this f_n is exceeded, this isolation system will not perform properly, and it is quite possible that you will *amplify* the vibrations. Systems that have a f_n lower than that calculated in equation 15 will be isolated.

Step. 3: Determinate what system's natural frequency will provide the desired level of isolation. Step 2 has provided a quick way to determinate which mounts provide isolation, but does not provide any information of the level of isolation that will be achieved. Equation 11 can be used to calculate transmissibility:

$$T = \left| \frac{1}{1 - \frac{f_d^2}{f_n^2}} \right| \qquad 11$$

Equation 11 can be used to calculate the transmissibility of a known disturbing frequency through a mass-mount system with a known natural frequency. It can also be rearranged to the following form:

$$\text{for} \quad \frac{f_d}{f_n} > 1, \quad \Rightarrow \quad f_n = \frac{f_d}{\sqrt{1 + \frac{1}{T}}} \qquad 16$$

Equation 16 is valid only when $f_d/f_n>1$. This can be used to calculate the required natural frequency to achieve the desired level of isolation of a particular disturbing frequency.

Step 4: Select the appropriate isolator such that its stiffness will give the required natural frequency calculated in the previous steps. Starting by the known mass of the system, equation 3 can be used to compute K.

$$f_n = \frac{1}{2\pi}\sqrt{\frac{k}{M}} \qquad 3$$

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DESIGN EXAMPLES

This selection deals with the selection and application of vibration and shock isolators. For the proper selections of isolators, it is desirable to obtain, where possible, pertinent information relating to the equipment, input and output requirements, and the general environment. Examples to the type of information or data required are:

Relating to the equipment:
- Weight.
- Dimensions
- CG location.
- Number and location of isolators.
- Available space for isolators.
- Fragility level of the equipment

Relating to the dynamic input and output:
- Level of vibration.
- Level of shock.
- Space limitations.

Relating to general environment:
- Temperature.
- Humidity.
- Salt spray.
- Corrosive atmosphere.
- Altitude.

All of the above information is not always readily available nor is it always completely required in some applications. This will be further clarified in the following problem example.

Example 1 – Vertical Vibration
A centrifugal compressor is directly driven by a 1500 RPM motor is causing vibration disturbance to the floor on which it is mounted. The compressor, motor, and support base weigh 180kgf. There are 4 mounting points for the isolators. The required isolation is 90%.

1. Determinate f_n of the system required by using mathematical methods.
2. Determinate static deflection of isolators by using (a) mathematical method and (b) the static deflection vs. natural frequency curve in figure 5.
3. Determinate damping factor C/C_c to limit transmissibility at resonance to 3 by using (a) mathematical methods and (b) the transmissibility curve in figure 8.
4. Determinate the resilient media which could be used in the isolator selected to provide the C/C_c required.
5. Determinate the proper isolator to use for this application.

SOLUTION

Known facts:
- W = 180 kgf
- Weight per mounting point = 180/ 4 kgf
- Isolation required = 90%
- i.e. transmissibility = 0.10
- Disturbing frequency, f_d = 1500 RPM = 25 Hz

1. Using equation 16:

$$f_n = \frac{f_d}{\sqrt{1+\frac{1}{T}}} = \frac{25Hz}{\sqrt{1+\frac{1}{0.1}}} = 7.5Hz \qquad 16$$

2a. To find static deflection using mathematical approach use equation 4.

$$\delta_s = \frac{g}{(2\pi f_n)^2} = \frac{245}{f_n^2} = 4.3\ mm \qquad 4$$

2b. To find static deflection use the static deflection – natural frequency curve figure 5. The intersection of f_n of 60x7.5Hz = 450CPM and the solid diagonal line yields a δ_s of approximately 4.3 mm.

3a. To find C/C_c for a transmissibility of 3 at resonance by mathematical approach, use equation 12. Solving for C/C_c:

$$\frac{C}{C_C} = \frac{1}{2T} = \frac{1}{2 \cdot 3} = 0.17 \qquad 12$$

3b. To find C/C_c for a T of 3 by use of the transmissibility curve figure 8. This curve shown that for a transmissibility of 3, C/C_c = 0.17.

4. To find the correct resilient media which exhibits a C/C_c = 0.17, refer to Table 1. It can be seen that air dampers or POWERFLEX dampers would be the proper selection.

5. Referring to Powerflex data sheets of PWH series, an isolator of the PWH600 series which best fits the above solved parameters is the PWH632. By the table in the data sheet and the values of compression load (symbol " ▼ "), the application of load of 44kgf gives a static deflection (δ_s) of 5 mm, that gives a vertical natural frequency of 7Hz (equation 4, or figure 5).

Example 2 – Vertical and Horizontal Vibration

An electronic transmitter which weight 44 kgf, and has a height of 380 mm, a width of 510 mm and a length of 760 mm is to be mounted in a ground vehicle which imparts both vertical and horizontal vibratory input to the equipment. Four mounting points, one at each corner, are provided. It has been determined that the first critical frequency of the equipment is such that a system with12 Hz vertical natural frequency would be satisfactory. Select an appropriate isolator and determine the approximate horizontal rocking modes in the direction of the short axis of the equipment which would be excited.

SOLUTION

1. For vertical natural frequency:

Load per isolator 44/4=11 kgf.

Referring to a POWERFLEX isolator series PWH200, the PWH202 is suitable. From the load rating table in the data sheet information (compression load symbol " ") it would handle the 11 kgf load, giving a static deflection (δ_s) of 2 m, which gives a 11Hz vertical natural frequency (equation 4, or figure 5).

2. For horizontal rocking modes: the dynamic stiffness ratio of horizontal to vertical = 0.4 for the PWH202. This value can be calculated by the ratio between the longitudinal shear load (symbol " ") and the compression load at the same static deflection (2 mm in this case).
Referring to figure 10, and assuming that mass is homogeneous and isolators are at extreme corners, the following is found:

$$R = \frac{K_H}{K_V} = 0.4$$

$$\frac{H}{W} = \frac{380}{510} = 0.7$$

From the curves in figure 10 the ratios of f_{nH} /f_{nV} for first mode M_1, is 0.55 and for second mode , M_2, is 1.6.

$$I \text{ modo}, \qquad f_{nH\,I} = 11Hz \times 0.55 = 6.1Hz$$

$$II \text{ modo}, \qquad f_{nH\,II} = 11Hz \times 1.6 = 17.6Hz$$

it is seen that this procedure lends a ready solution to determining the horizontal rocking modes bases on the assumptions made. This solution is not exact but is generally satisfactory for a practical purposes.

Example 3 – Shock

A piece of electronic equipment is to be subjected to a 30G, 11 millisecond half-sine shock input. The equipment is mounted on a 10Hz natural frequency isolation system. Determinate maximum shock transmission and isolator deflection.

SOLUTION

1. From figure 11 the equation for shock velocity change for a half-sine pulse is:

$$V = \frac{2gA_0t_0}{\pi}$$

Where: A_0 = 30
g= 9.81 m/sec^2
t_0 = 11x10^{-3} sec

$$V = \frac{2 \times 9.81 \times 30 \times 0.011}{\pi} = 2.06\ m/\sec$$

Using equation 13 the maximum shock transmission is:

$$G_T = \frac{V(2\pi f_n)}{9.81} = 13.2g \qquad 13$$

Using equation 14 the isolator deflection required to attenuate this shock:

$$\delta_D = \frac{V}{2\pi f_n} = \frac{2.06}{2\pi \ 10} = 0.032\ m \qquad 14$$

This example could also be done in the “reserve” direction. If one knew the desired output, 13.2 g’s one could calculate the required natural frequency, 10Hz, to attenuate the input shock.

In either case, the deflection is calculated last, and used to determine 1) if the allowable sway is sufficient to accommodate the required deflection, and 2) if the selected isolator has enough linear deflection capability to withstand the shock.