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How to Select Vibration Isolators for OEM Machinery & Equipment

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Design engineers must consider several basic factors when utilizing isolators as integral components in OEM machinery and equipment. A previous article has described the selection of machinery mounts used to isolate complete machines.[1] This article, however, will deal with the physical significance of the concepts involved in the selection of vibration isolators which are designed, as components, into OEM machinery and equipment. It is hoped that the reader will be able to utilize this information to better understand how to specify and select component isolators.

Dynamic Functions Of Isolators

From the standpoint of dynamic performance, isolators can have one or more of three basic functions:

- protection against vibration
- protection against shock
- protection against structureborne noise transmission

Vibration, shock and structureborne noise transmission are different ways in which energy can travel through contiguous elements in a structure. When it is necessary to inhibit the transmission of energy from one point to another within a structure, an isolator is designed into the structure between those points.

These three forms of structureborne energy transmission, although similar, have numerous different characteristics, as have been described in exhaustive studies previously published. This article will, therefore, not attempt to cover theoretical considerations in great detail, but will rather state and describe the meaning of phenomena as they apply to selection of devices to inhibit structureborne energy transmission.

Vibration Isolation

Vibration isolators generally perform in accordance with the relationships shown in Figure 1, which plots transmissibility as a function of the ratio of forcing frequency to natural frequency for different damping ratios. Except for certain kinds of servo-controlled pneumatic isolators, unit isolators act in accordance with this family of curves.

Precise definitions of "damping ratio" and "natural frequency" terms have been well described in previous articles. For purposes of a handy reference, however, the following definitions will aid the reader in following the concepts presented in this article.

The effectiveness of an isolator is commonly described in two ways: transmissibility and isolation efficiency. Transmissibility is defined as the dimensionless ratio of the dynamic output to the dynamic input. Isolation efficiency, on the other hand, is defined as force input minus force output, all divided by force input. The two concepts are, of course, different ways of saying the same thing, an isolation efficiency of 90% being equivalent to a transmissibility of 0.1, while a 75% isolation efficiency is the same as a transmissibility of 0.25.

Note that the familiar transmissibility curves shown in Figure 1 are specified as single-degree-of-freedom curves. In essence, this means that all isolators under a given component act in concert, having the same natural frequency, stability and damping characteristics. Multiple-degree-of-freedom-systems should be evaluated only by experts.

Damping Ratio

The primary purpose of adding damping is to reduce amplification at and near resonance (f/f_n equal to one). Damping ratio is a measure of the degree of damping in a particular material or isolator. The higher the damping ratio, the more heavily damped the isolator. As seen in Figure 1, isolators with high damping ratios have lower value of transmissibility at resonance than isolators with low damping ratios. The figure also shows that in the region (f/f_n, greater than 1.414) where isolation is provided, isolators with a lower damping ratio provide a lower transmissibility (greater isolation efficiency) at any given frequency ratio.

Damping can take several forms in OEM equipment. It can, of course, be introduced merely by selecting isolators made of a highly damped elastomer such as silicone. Another way of adding damping is to incorporate mechanical or other damping devices into a dynamic system. Such devices either take the form of friction or fluid dampers built into isolator housings, or damper devices installed separately from the isolators themselves. Most automotive "shock absorbers" are merely fluid dampers utilized to augment a coil spring and/or leaf spring suspension systerns. These so-called "shock absorbers" literally damp out excessive motion under operating conditions which result in transmissibilities greater than one.

Natural Frequency

Natural frequency is technically defined as the number of oscillations per unit time that a system will carry out if displaced from an equilibrium position and allowed to vibrate freely. In other words, the natural frequency of a system is the frequency at which the system "prefers" to oscillate. The "forcing frequency" in Figure 1, on the other hand, is the rate at which input energy is applied to the isolator. This can be thought of as the rate at which the input forces "want" the system to vibrate.

The natural frequency characteristics of a given isolator are a function of the isolator configuration and materials. Both design and material establish the stiffness of the isolator in terms of the force increment required to result in a unit deflection increment. This stiffness is usually expressed in terms of pounds of force per inch deflection and is a constant throughout the usage range of most isolators, except as described later in this paper,

The natural frequency of an isolator can be shown to be proportional to the inverse of the square root of static

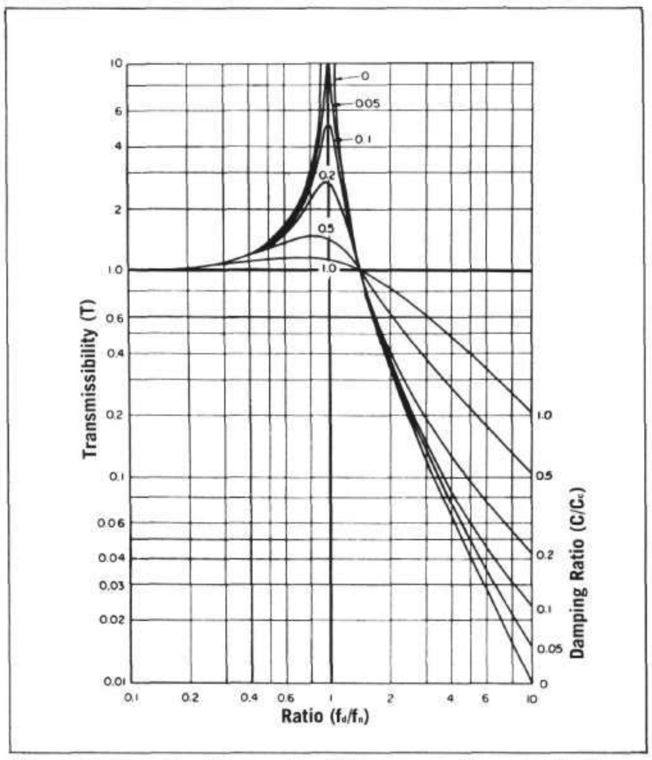


Figure 1 – Family of transmissability curves for a single-degree-of-freedom system.

deflection. Since static deflection is proportional to static weight, natural frequency is also proportional to the inverse of the static load on the isolator.

In general, the static stiffness of an isolator may not be the same as the dynamic stiffness of the same isolator. For this reason, the natural frequency of an isolator under vibratory operating conditions may not necessarily be accurately determined by using static load vs. deflection data and the familiar equation which indicates natural frequency to be 3.13 times the square root of the inverse of static deflection. In addition, some isolators are amplitude sensitive in that the dynamic load vs. natural frequency characteristics differ with different amplitude inputs.

Accordingly, manufacturers of isolator mounts often provide separate load vs. natural frequency and load vs. deflection charts, with the former directly applicable to determining dynamic natural frequency as a function of load per isolator. In the same manner, ratios of lateral to axial stiffness are properly published specifically as dynamic data. In cases where a mount may be amplitude sensitive, this fact should be stated in conjunction with the published load vs. frequency data. The important factor here is that the natural frequency of an isolator is *not* affected by the forcing frequency of the vibrational input.

The selection of an isolator, on the other hand, *is* based upon obtaining the natural frequency which achieves the §Jf_n ratio necessary to provide the desired isolation efficiency from Figure 1. In situations where the isolator may be exposed to several forcing frequencies due to a multiplicity of sources or operating speeds for the same source, it is usually necessary to determine which of the several frequencies results in the greatest disturbance. It would, of course, seem expeditious to merely select natural frequency at some predetermined fraction of the lowest forcing frequency, but this approach will not necessarily result in optimizing the design, particularly if the lowest forcing frequency represents vibratory inputs which are either not the most predominant, or not predominant enough to influence isolator selection.

Another facet of isolators designed for several forcing frequencies is the avoidance of problems at resonance. The proper combination of damping and natural frequency, along with the determination of "acceptable" resonance levels, must be worked out for each application.

One caution here is that isolators should not be loaded beyond the range indicated by the manufacturer. If loaded excessively, a mount may become much stiffer than it is within the recommended operating range and it may even be permanently deformed.

Sizing

Usually, all mounts supporting a given component are designed to have the same natural frequency and the same axial profile height. This rule of thumb insures that all support points have the same stiffness under load and that the component is installed parallel to its support base. Although there may be instances where one of these guidelines may be inappropriate, such instances are extremely rare. As a result, use of different stiffness and/or profile mounts in a single application should be referred to experts.

To facilitate selection of mounts in the manner described above, many isolators available to OEM design engineers are manufactured in a series of sizes with overlapping load capacities, all with the same profile height and all in the same natural frequency range.

Off-Axis Inputs

The concepts presented above all refer to situations in which the energy input is colinear to the axis of the isolators. Many practical isolator applications, however, involve isolated equipment which is excited by forces at some angle to the isolator axis, as well as by forces colinear to the axis.

As an example, a horizontally excited isolation system will exhibit two natural frequencies if the dynamic center of the isolators is not in line with the center of gravity of the mounted equipment, as shown in Figure 2. These frequencies, known as "rocking modes," are: a mode at which the equipment rocks about a point in the vicinity of the center of gravity and a lower mode in which the equipment oscillates at a point well below the elastic center of the isolators. If the equipment is rotated 90° in the horizontal plane with respect to the exciting forces, two other natural frequencies will occur if symmetry is different in each direction.

Dealing with this situation should usually be left to experts, but for the simple case of a solid, uniform mass with the isolators attached under the extreme comers, Figure 3 can be used to approximate rocking mode frequencies as a function of isolator stiffness rate *K* (in lbs. per inch), as a function of the ratio of lateral to vertical stiffness, and as a function of the vertical natural frequency of the isolators at the indicated load.

Note that this figure shows that the higher frequency rocking mode can be eliminated if the width of the mounted equipment is more than approximately five times the height. Under such conditions the equipment does not rock, but rather translates in a horizontal mode. If it is not possible to avoid rocking and the center of gravity is above the elastic center of the mounts, motion may be restricted to translation by relocating the mounting points so that the

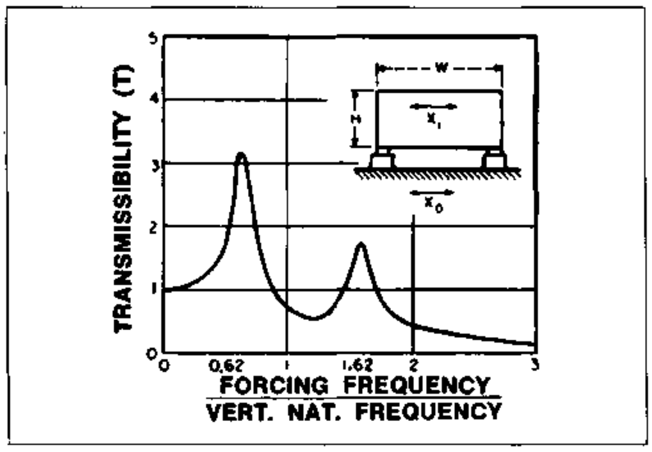


Figure 2 – Typical vibration test curves of a system excited in the horizontal direction.

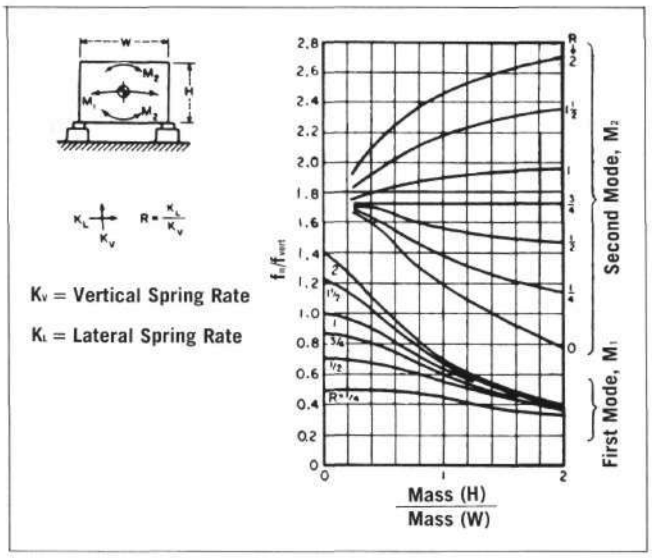


Figure 3 – Horizontal natural frequencies of a uniformly distributed mass mounted upon linear, undamped isolators acting at the edge of the mass.

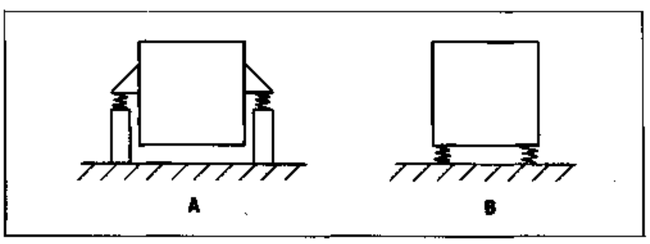


Figure 4 – Installation with mounts located: (A) at the same plane as the equipment center of gravity, (B) at the bottom corners of the equipment.

isolators are installed in the same plane as the center of gravity, Figure 4a, rather than at the bottom corners of the equipment, as shown in Figure 4b. A similar effect is often accomplished by using a heavy mass beneath the equipment to lower the center of gravity.

Stability

Vibration and shock mounts invariably incorporate the concept that in order to inhibit structureborne energy transmission, a mount must deflect and that the force consumed in causing that deflection is thereby not transmitted through the isolator but rather absorbed by it. Isolators can be described as being "stiff" or "soft." These descriptions cannot be quantified, but the "softer" the isolator, the easier it deflects under a given load. Accordingly, the "softer" the isolator, the lower its natural frequency.

When a very soft system must be designed into OEM machinery or equipment, engineers must make certain that the low natural frequency required for isolation in one plane does not result in low stability in another. If, as an example, a component requires an isolator with a 3 Hz vertical natural frequency, one must also know the lateral stiffness requirements. If subjected to excessive lateral loads, the isolated equipment may, depending upon the application, require less, or even greater lateral stiffness than vertical stiffness. If lateral stiffness is too low, the system may exhibit a tendency to topple over when subjected to lateral loads. If, on the other hand, lateral stiffness is too high, the isolator may not provide adequate horizontal isolation.

If lateral stability is a design problem in any given situation, there are basically two ways in which engineers can evaluate particular isolators. One is to select isolators which have the desired vertical and lateral requirements. For this purpose, isolator mounts are usually rated in terms of the dimensionless ratio, *R*, of lateral to vertical dynamic spring rates. Using this system, an *R* of unity implies that vertical and horizontal stiffness is equal and that the dynamic characteristics of the mount are unaffected by the direction in which the mount is loaded. An *R* value less than unity, on the other hand, indicates that the mount is a specified degree less stable radially (horizontally) than axially (vertically), whereas an *R* value greater than one indicates the mount to be a specified degree stiffer horizontally than vertically.

The other way in which an isolator can be stabilized is to design a housing or lateral restraint which will physically limit horizontal motion. Such systems are often used, but primarily as a means for preventing toppling or excessive sway. One potential difficulty with such housings is that they may increase lateral stiffness or permit low lateral stiffness only within a predetermined deflection limit. As that limit is reached, the lateral stiffness would increase rather suddenly as the isolator makes physical contact with the restraint. For obvious reasons, it is usually better to select an isolator with stability "built-in" as a result of material selection and design.

One other aspect of stability is that of sway allowance. As the mounts deflect to provide isolation, equipment will translate accordingly and space must be provided accordingly. When selecting isolators which may be subjected to severe loads, manufacturer's data on mount sway should be obtained or experimentally determined.

Shock

The isolation of shock, is, like that of vibration, also expressed in terms of transmissibility characteristics which are a function of damping ratio, forcing frequency, and natural frequency. The general characteristics of shock are, however, somewhat different. In an idealized case, vibration is a steady state phenomena in which force amplitudes regularly rise and decay in a sinusoidal manner. Shock, however, is defined as a motion in which there is a very sudden change in velocity. Shock pulse inputs are, therefore, usually characterized by a rapid acceleration and deceleration in a very short time period. Accordingly, the shock isolator "sees" input energy which has a very steep wave front with respect to time. The isolator accepts this input energy on a virtually instantaneous basis and releases it at a rate equivalent to the natural frequency of the isolated system.

The isolation of shock is dependent upon the form of the input pulse, as described in terms of peak amplitude (expressed as multiples of gravitational acceleration), duration of the pulse, and overall shape of the time history of the pulse. Depending on the type of mechanical shock involved in a particular shock wave, different time history patterns will occur. These patterns range from a simple rectangular acceleration (acceleration instantaneously increases to a peak value, remains constant for a very brief time period, and instantaneously decreases to its original value) to a random acceleration or force impulse. For each of these different patterns, the forces generated and transmitted due to a given change in velocity will be different. Figures 5a and 5b, for instance, show the shock transmissibility curves for rectangular and half-sine pulses, respectively.

Note that rather than utilizing the ratio of forcing frequency to natural frequency, as was the case with Figure 1 on vibration isolation, the shock curves use the ratio of shock pulse time duration to the natural period of the system. The natural period of the system is, of course, the inverse of the natural frequency, so the abscissa of the shock transmissibility curve can also be stated as the product of system natural frequency and shock pulse time duration. In order to minimize transmissibility for these pulse patterns, it is desirable to select an isolator with a low enoxigh natural frequency to result in an abscissa value significantly less than approximately 0.2. If, for example, one were isolating a rectangular pulse of 100 milliseconds (0.1 seconds) duration, a natural frequency of 1 Hz would be required to achieve a transmissibility of 0.5. Similarly, if the pulse period were only 20 milliseconds (0.020 seconds), a 5 Hz isolator would achieve the same result.

The natural frequency required of a shock isolator may be determined as follows:

$$f_n = \frac{61.4\,Gt}{V} \text{ and } \Delta\rho = \frac{V}{2\pi f_n} \qquad (1)$$

where

f_n = isolator natural frequency in direction of shock pulse
Gt = shock transmitted by the isolator
V = change in velocity as a result of shock
$\Delta\rho$ = dynamic deflection of isolator under shock

For information on calculation of V for different types of input pulses, the reader is referred to reference 2 at the conclusion of this article.

Two-Stage Mounts

Quite often a design engineer may encounter a problem which may seem to require two completely different mounts in order to provide acceptable isolation under significantly different dynamic inputs. A common example is equipment subjected, at different times, to both low frequency, steady state vibration and high impact shock loads. An isolator selected strictly on the basis of vibration considerations would, of course, be selected on the basis of determining the natural frequency and damping characteristics required to achieve a specific transmissibility. When subjected to lateral load and/or high impact shock conditions, however, conventional mounting systems pose

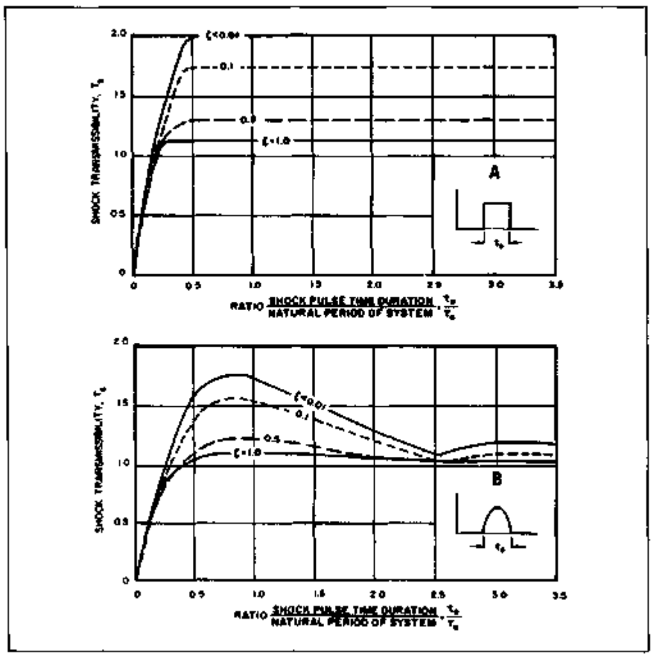


Figure 5 – Shock transmissibility curves for an isolation system subjected to: (A) rectangular shock pulse, (B) half-sine shock pulse.

difficulties for the engineer. A low frequency conventional coil spring, for instance, if designed to have a unity lateral to axial stiffness ratio, would have a static deflection under load of approximately eight-tenths the working height of the mount. Under shock loads, the mount will, of course, have to deflect even more, so steps must be taken to prevent the mount from "bottoming out" under impact. With coil springs, this usually results in a rather large working height and coil diameter. Similar problems exist with conventional air-bag isolators of the type which are basically pressurized elastomeric bladders or sleeves affixed between metal end caps. Both types of systems also often require separate snubbers to absorb impact shock.

To overcome the difficulties posed by these conventional mounting systems, a "two-stage" mount is often the ideal solution. This type of mount is designed to exhibit two sets of dynamic natural frequency vs. load characteristics. Under small dynamic deflections, the mount may be designed to exhibit a low natural frequency in the range of 3-5 Hz, while under impact shock forces the mount would have an 8-10 Hz natural frequency. Essentially, such a mount will exhibit the low natural frequency vibration isolation characteristics up to a certain deflection. Beyond that deflection, the mount becomes somewhat stiffer, acting as a high deflection shock mount.

The use of a two-stage mount enables one isolator to do two jobs for the same application. An excellent example of a two-stage mount is the pneumatic rubber mount described later on in this article.

Noise Control

All sounds, except those resulting from fluid turbulence, are caused by mechanical vibrations. A vibrating structure causes the air around it to vibrate, resulting in very small fluctuations in air pressure. These air pressure fluctuations in turn cause portions of our inner ear to vibrate, creating the sensation of sound. When that sound becomes unacceptably loud at frequencies to which the human ear is sensitive, it becomes classified as noise.

When the half wavelength of the forcing vibration is much larger than the thickness of the material through which it travels, the isolation of structureborne energy is accomplished in accordance with the concepts indicated above for single-degree-of-freedom isolation. When dealing with situations in which the half wavelength is much smaller than the material thickness, however, another concept becomes applicable; that of Specific Acoustical Impedance. This concept becomes important when dealing with structureborne energy frequency components in the 2-5 kHz region, the frequency range most important to the control of noise.

Table I[3] shows the acoustical properties of various materials which commonly form portions of structureborne noise transmission paths. Analysis of this table clearly demonstrates that elastomers, for instance, when subjected to frequencies above 2000 Hz, are in the operating regime affected by Specific Acoustical Impedance if the elastomer thickness is in the range of approximately *Vi* to 2 inches. This range is the approximate range of thicknesses encountered in elastomeric isolators or mounts with elastomeric pads. For this reason, virtually all elastomeric isolators are effective attenuators of structureborne noise.

The ability of a material to conduct sound is directly related to its Specific Acoustical Impedance in that materials with greater impedance values are better conductors of structureborne sound. The importance of Specific Acoustical Impedance is not, however, limited to the properties of any particular material but is rather based on the fact that structureborne noise reduction requires a "mismatch" of impedance between different segments of the structure through which the energy is transmitted.[4]

The acoustical impedance of steel, for example, is approximately 100 to 1000 times that of elastomeric materials such as rubber. Rubber, in turn, has impedance levels which are approximately 100 to 1000 times that of air. Accordingly, steel conducts sound better than rubber, which in turn conducts sound better than air. Structureborne noise traveling, for instance, from a steel motor housing to a steel support plate, "sees" no interruption in the path if the housing is hard-mounted to the plate. If an elastomeric mounting is used, however, the wave "sees" two mismatches in the sound path, creating a very effective and efficient means of reducing the level of noise radiated by the support.

In order to take advantage of these properties, elastomeric materials are often used in attachment and mounting devices in order to provide an impedance mismatch between elements of a structure, Figure 6. Use of such devices has been demonstrated to reduce structurally transmitted noise by more than 20 decibels compared to a structure fastened together by conventional bolting techniques, Figure 7.[5]

Table I — Acoustical properties of materials commonly encountered in noise transmission paths.

Material	Density (gm/cm³)	Velocity of sound (cm/sec)	Specific Acoustical Impedance
Steel	7.8	501×10^3	391×10^4
Brass	8.4	275×10^3	232×10^4
Lead	11.4	71.8×10^3	82×10^4
Wood (Fir)	0.45	443×10^3	20×10^4
Water	1.0	140×10^3	14×10^4
Rubber	1.91	3.5×10^3	0.35×10^4
(various types)	1.06	5.0×10^3	0.53×10^4
	1.11	6.4×10^3	0.71×10^4
	1.18	10.0×10^3	1.18×10^4
	1.25	23.0×10^3	2.87×10^4
Air (0°C)	0.00129	33×10^3	0.0042×10^4
Cork	0.25	50×10^3	1.2×10^4
Concrete	2 6	310×10^3	81×10^4

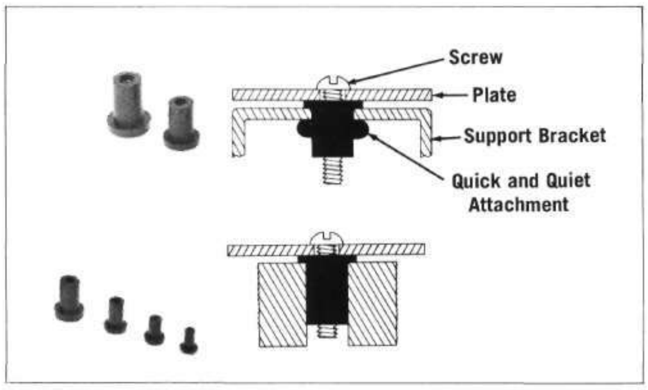


Figure 6 – One-piece "Quiet" attachment device.

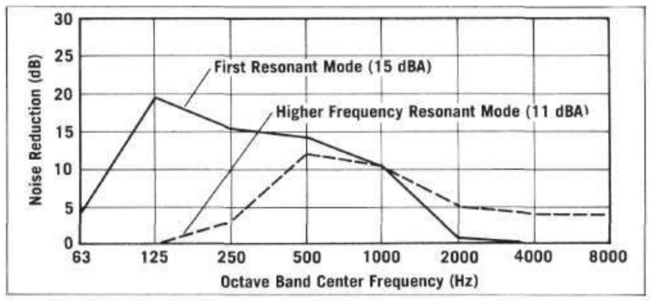


Figure 7 – Noise reduction effectiveness of "Quiet" attachment device in reducing noise created by resonant energy inputs.

Environmental Considerations

An important consideration in selection of an isolator is the environment in which the isolator must function. In order to adequately categorize the ability of certain isolators to withstand different types of environments, one must understand the basic differences between the different types of resilient media used for isolation.

Elastomeric Isolators

Perhaps the most commonly used type of isolator is the elastomeric isolator. Elastomers can be molded in many different configurations of many different materials, including natural rubber, neoprene, butyl, silicone, and a number of combinations of each. In addition, 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 also is often quite useful in preventing problems at resonance that would be difficult to restrain if coil springs were used. In addition, the relatively low Specific-Acoustical Impedance of elastomeric materials provide excellent resistance to the transmission of noise through structures where acoustic considerations are an important factor.

From a standpoint of shock isolation, 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 systems.

From the standpoint of users, natural rubber is a material which exhibits good strength and damping characteristics, but has the disadvantage of offering relatively poor resistance to oil, ozone and heat aging. Neoprene, however, is well suited to ozone, fuels, and oils, in addition to offering good resistance to heat aging. Neoprene is probably the best all-around elastomer, as evidenced by its rather widespread use in industry. Silicone, on the other hand, is one of the most costly elastomers, but offers the ability to provide effective isolation at temperatures from —65°F. to +350°F., whereas neoprene is limited to a range between -40°F. and +200°F.

In spite of the few limitations of elastomeric isolators, the advantages far outweigh the disadvantages and make elastomers the most highly desirable type of resilient media for the great majority of OEM applications.

Coil Springs

Coil springs have some very desirable characteristics as isolators in that they can be readily designed to exhibit relatively low natural frequency characteristics. In this regard, helical coil springs with natural frequencies down to 2-3 Hz or less are not uncommon.

The static deflection inherent in a 2 Hz spring, however, is approximately 2.4 inches under load. Unless the spring has adequate lateral stability under such conditions, instability may exist and mounted equipment may tend to topple or fall sideways. This can be a particular problem unless all forces on the spring are coaxial with the axis of the spring.

Although engineers can vary the degree of lateral stability in coil springs, such variations may involve either increasing the ratio of coil diameter to working height for a given natural frequency or adding an external housing around the spring. Such housings, or other structural restraints, are often used to physically restrict lateral motion of coil springs in order to provide stability. In load ranges beyond approximately 40 lbs. however, such devices are often impractical as machine components.

Another disadvantage of coil springs is that they have extremely low damping ratios, resulting in resonant transmissibilities of 100 or more. Coil springs also are good transmitters of high frequency energy and consequently are poor isolators for high frequency surges and for structureborne noise paths.

Metal Mesh

Stainless steel wires, knitted into a fabric-like mesh screen, have been successfully compressed into resilient cushions in a variety of OEM applications. Mesh isolators are made in many different configurations and can be fabricated to meet a wide range of natural frequency, damping, and radial-to-axial stiffness properties. In general, metal mesh cushions can be designed to exhibit the same dynamic characteristics as elastomeric isolators.

The advantages of metal mesh are that stainless steels can be used beyond the —65°F. to +350°F. range to which elastomers are limited and that corrosive environments which adversely affect elastomers, have virtually no effect upon stainless steel.

The dynamic advantages and disadvantages of metal mesh isolators are similar to those of elastomers, except that mesh may be more costly if extreme environmental conditions dictate use of a special stainless or other steel alloy. If isolators are to be exposed to extreme temperatures and corrosive environments, however, metal mesh may be the best solution.

Pneumatic Isolators

Pneumatic isolators offer a number of benefits in applications requiring low frequency isolation. Pneumatic isolators are designed to exhibit natural frequencies in the range of 0.1 Hz to 5 Hz, with corresponding isolation efficiencies as high as 99%. Pneumatic isolators are, therefore, used for many of the same applications as coil springs, but can be designed to avoid some of the disadvantages of coil springs. As an example, pneumatic isolators can be readily designed to require less height than coil springs. This advantage becomes significant when one calculates that a conventional damped spring isolator designed for a 1 Hz natural frequency would normally have a loaded height of as much as 30 inches, whereas a pneumatic isolator would require less than half that height. Pneumatic isolation systems are also generally not susceptible to high frequency surge and hysteresis, both of which are problems often associated with conventional coil springs.

Pneumatic isolators also lend themselves to being incorporated into "active" mounting systems which use servo-controlled valving systems to sense variations in dynamic input and automatically hold position of the mounted equipment while maintaining a very high isolation efficiency.

Other features of pneumatic isolators are that they are relatively well damped compared to coil springs and can be designed to effectively isolate sensitive elements from very small disturbances down into the "micro-g" range, where conventional coil springs are of questionable value.

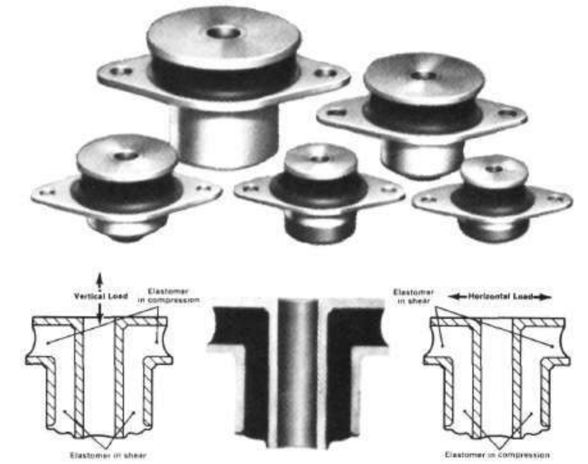
Pneumatic isolators are available to design engineers in the form of several types, each of which has significantly different characteristics. While all pneumatic mounts are basically low frequency isolators, some have relatively poor lateral stability, while others have equal vertical and horizontal dynamic stiffness. Also, some pneumatic mounts are ideal for high deflection shock applications, while others are not. The disadvantages of pneumatic isolators are primarily associated with the particular type of mount being considered. As an example, use of a pneumatic mount with poor lateral stability may require structural restraints or an external housing around the mount in order to prevent the mounted equipment from listing or tipping sideways. In general, pneumatic isolators are limited to loads down to approximately 100 lbs. per mount.

Other Resilient Media

Elastomers and metal mesh are not the only materials which have been successfully used as isolator media. In this regard, materials such as wool and felt have been used as resilient media with varying degrees of success. Unless highly compressed (and, therefore, stiffer and less resilient), however, these materials may be susceptible to absorbing oil, water and other industrial chemicals, causing deterioration of either the material or its dynamic properties.

Specific Isolator Types

As shown by the description of resilient media, different materials have inherently different isolation capabilities. The properties of any given isolator, however, are dependent not only on the material of which it is fabricated, but also of shape, configuration and overall construction with respect to structural material used within the body of the isolator. Accordingly, two mounts of the same material may be designed to demonstrate markedly different isolation characteristics.



Features & Benefits
Wide load range — 50 lbs. to 2700 lbs.
High capacity for compact size
Low natural frequency — 10 Hz
Inherently fail-safe
Equal stiffness axially and radially
Built-in snubbing both directions

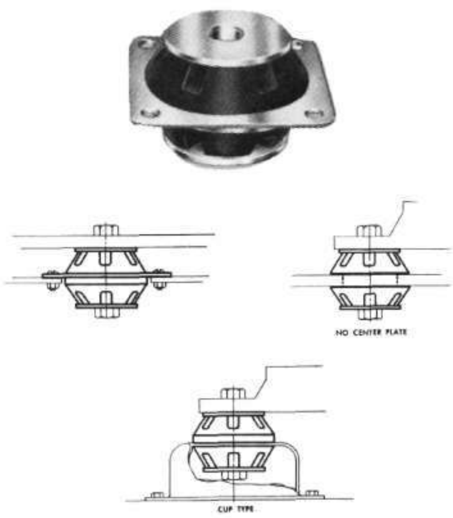
Typical Applications
Farm Equipment, Trucks, Construction Equipment: engines, transmissions, cabs, air-conditioners, radiators, pumps
Recreational Vehicles: engine generators, motorcycle engine mounts
Rapid Transit Vehicles: generators, motors, air-conditioners, pumps
Materials Handling & Industrial Equipment: engines, cabs, motors, generators, compressors
General Marine: power plants, pumps, generators

Figure 8 – Shouldered Core Platemount.

Features & Benefits
Bonded metal center cylinder
Built-in preload
All attitude load capacity
Natural Frequency down to 12 Hz
Rebound protection against severe shock inputs
Load ranges of 15 to 1600 lbs.
Low stress

Typical Applications
Farm Equipment, Trucks & Construction Equipment: engines, transmissions, cabs, radiators, battery boxes, commercial equipment, air-conditioners, pumps
Recreational Vehicles: power plants
Business Machines: memory drums, electronic components, motors, compressors
General Marine: power plants, pumps, generators
Rapid Transit Vehicles: pumps, motors, generators, bolsters

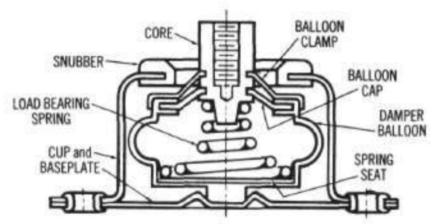
Figure 9 – Bonded Ring and Bushing Mount.



Features & Benefits
Good shock protection in 4 to 30 lb. load range
Good vibration isolation above 40 Hz
Fail-safe — good for vehicular or stationary applications
Compact, low profile
Attachment metals
Natural frequency 15 - 30 Hz
Radial to axial dynamic stiffness ratio of 0.6
Usable in different configurations with center plate or cup base, etc.

Typical Applications
Business Machines: memory drums, electronic components, motors, fans, pumps
Farm Equipment, Trucks, Construction Vehicles: communication equipment, air-conditioners, refrigeration units
Recreational Vehicles: power plants
General Marine: communication equipment, sonar, radar

Figure 10 – Tubular Core Mount.



Features & Benefits
Low natural frequency — down to 7 Hz
High damping — maximum transmissibility of 3.5 at resonance
Load range from 0.3 to 35 lbs.
Wide temperature range -65°F to +300°F.
Good high frequency isolation
Radial to axial dynamic stiffness ratio of 0.3
Captive metal cup

Typical Applications
Delicate, optical, processing and computer equipment.

Figure 11 – Air-Damped Spring Mount.

There are many different specific types of unit isolators available for application to OEM machinery and equipment. Each type fills a different range of loads, sizes, stability, damping, and isolation efficiency requirements. Some of the more commonly used mount types are described in Figures 8-16, with the performance characteristics summarized in each case.

It is, of course, difficult to establish "cookbook" application recommendations for general types of applications because desired isolation efficiency, desired size, stability, desired damping, and so on vary from application to application. Depending upon the specific performance and mount features pertinent to a particular application, Table II should help engineers "zero-in" on the types of mounts best suited to each application.

Designer's Check List

The following tabulation should be checked by engineers prior to finalizing the selection of a component

Features & Benefits
Low natural frequency — down to 7 Hz
Load range 60 lbs. to 250 lbs.
Attachment metals bonded
Horizontal to vertical dynamic stiffness ratio of 0.4
Used primarily for stationary applications

Typical Applications
Business Machines: memory drums, motors, electronic components, fans, pumps
Industrial Machinery: portable generators, air-conditioners, pumps, centrifuges, presses, shears, compressors

Figure 12 – General Purpose Elastomeric Compression Mount.

Features & Benefits
Low natural frequency — 10 Hz
Fail-safe captive design
Built-in snubbing in all directions
Self-captivating hex built into inner member
Load range 3 lbs. to 30 lbs.
Horizontal to vertical dynamic stiffness ratio of 0.4

Typical Applications
Business Machines: memory drums, electronic components, motors, fans
Farm & Construction Equipment: instruments, controls
Recreational Vehicles: engines, seats
Rapid Transit Vehicles: instruments, pumps, motors, generators
General Marine: navigation gear, communication gear, instruments and gages

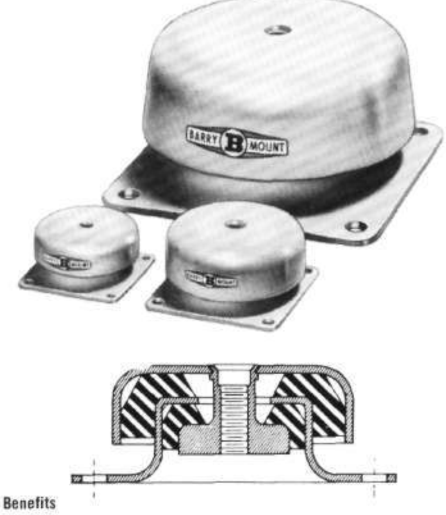
Figure 13 – Captive Elastomeric Mount.

isolator mount. Some of these items have been discussed previously in this article, but deserve attention here as design factors which are often critical:

a. **Is the direction of either the vibrational inputs or the static loading liable to vary, as might be the** case **in vehicular or shipboard applications? If** so, consider a so-called "all angle" mount. Such mounts have lateral to axial stiffness ratio near unity and can be loaded in any direction with no change in performance.

b. Will the mount be loaded in compression, tension, or shear? Regardless of stiffness ratio, certain mounts may be designed to be used in compression, but not in tension. Similarly, some mounts may not be recommended for lateral shear loads. Low stiffness ratio usually indicates that a mount is not designed for high lateral shear.

c. Has shock rebound protection been adequately



Features & Benefits
Excellent high shock protection
All attitude protection — can be used in tension, compression and shear with equal performance
Low profile, compact design
Rugged, fail-safe design
Load ratings from 8 lbs. to 1800 lbs.
Snubbing in all directions
Radial to axial dynamic stiffness ratio approximately equal to 1.

Typical Applications
Construction Equipment and Machinery, Farm Equipment Trucks: refrigeration units, air-conditioners, auxiliary motors, pumps, hydraulic equipment, pneumatic equipment
Business Machines: chassis supports
Industrial Equipment: motor generators, pumps, fans and blowers, motors
General Marine: power plants, generators

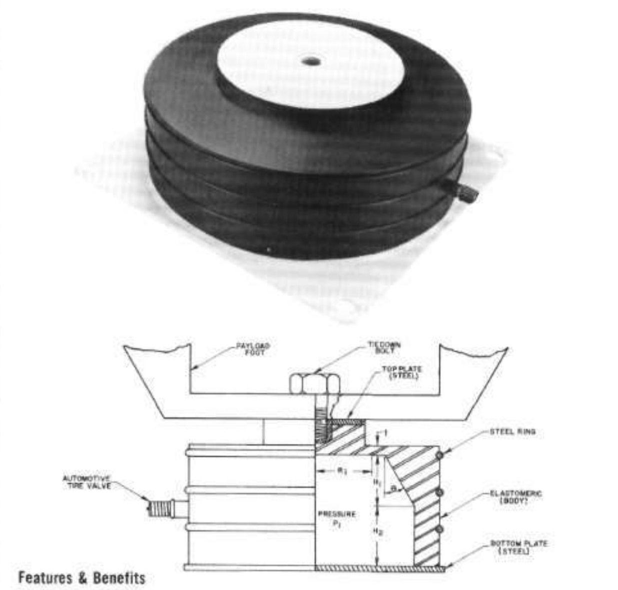
Figure 14 – Cupmounts.

considered? Shock impact will cause not only deflection in the direction of the applied shock, but also rebound in the other direction. For such applications, isolators either must have built-in rebound capacity, rebound cushions, or a dual system must be utilized in which isolators are used both above and below the mounted equipment, providing protection for both shock impact as well as isolator rebound.

d. Have adequate allowances been made for sway **space?** Space between mounted equipment and adjacent guards or housings must be adequate to enable the equipment to deflect, sway or rebound without collision.

e. **Are stabilizing mounts necessary?** Even to Figure 17, if the *alb* ratio *{a* being the distance to the center of gravity) is greater than 0.4, stabilizing mounts are recommended to prevent excessive lateral translation. In order to select stabilizing mounts, refer to Figure 18. The base mounts are selected as having to support the entire weight of the mounted equipment. The stabilizing mounts are then selected by taking moments about point 2 to find the total load at point 3. This total load is then divided by the desired number of stabilizing mounts to arrive at load per mount. When stabilizing mounts are selected from load vs. natural frequency curves, the natural frequency of the base isolators and stabilizer isolators should be the same. This usually means that all mounts are selected from the same family of isolators.

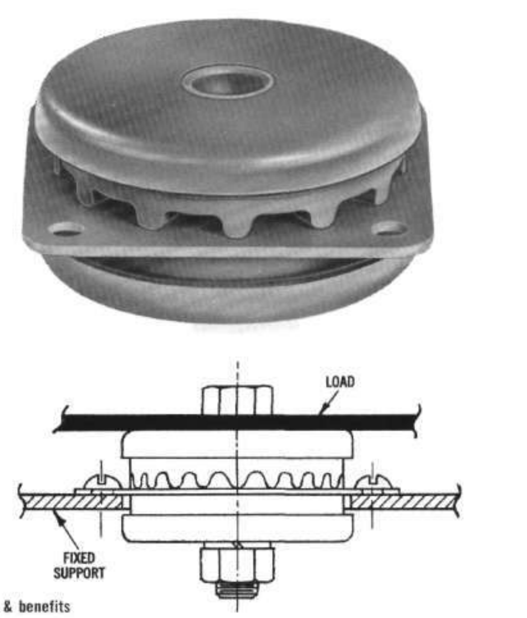
f. Is "fail-safe" construction necessary? One must always keep in mind that isolators are designed to be resilient, but are nevertheless counted upon to be integral components in a structure. Resilient materials, however, do not have the structural strength of steel and if subjected to extreme adverse loadings, could rupture or become damaged. In order not to depend upon the structural properties of resi-



Features & Benefits
Natural frequency as low as 3 Hz
Virtually constant natural frequency over load range of 75 to 19,200 lbs.
Zero static deflection under load
Radial to axial dynamic stiffness ratio of unity
Load carrying capacity with no air pressure
Built-in snubbing
Good lateral stability
Large shock deflection capability
Two-stage vibration and shock isolation characteristics

Typical Applications
Industrial Equipment: motor generators, centrifuges, vibratory feeders, vacuum pumps, refrigeration equipment, fans, industrial washing machines, compressors, pumps, rooftop air-conditioners
Trucks & Construction Equipment: cab mounts, seat mounts
Rapid Transit: car suspensions, air-conditioning
Shipping Container: delicate equipment

Figure 15 – Pneumatic Rubber Mount.



Features & benefits
Low cost
Low profile
Fail-safe design
Natural frequency down to 12 Hz
Three sizes — load range from 4 to 60 lbs.
Built-in snubbing all directions
Lateral to axial dynamic stiffness ratio of 0.4

Typical Applications
Business Machines: memory drums, electronic components, copiers, motors, fans
Trucks, Construction Equipment, Farm Equipment: communication equipment, instruments
Industrial Equipment: portable generators, pumps, fans, motors, air-conditioners
General Marine: navigation gear

Figure 16 – Plate Type Mount.

lient media, many commercially available isolators incorporate either metal core sections or captive metal housings so that if resilient sections are damaged, mounted equipment will not come loose. Examples of this type construction is shown in Figures 8 and 11. Another means of providing structural integrity into isolators is to incorporate very high overload capacity into the basic mount design. The Pneumatic Rubber mount shown in Figure 15, for

Table II — OEM isolator application index.

Equipment	Shouldered Core Platemount	Bonded Ring and Bushing Mount	Plate Type Mount	Tubular Core Mount	Air Damped Spring Mount	Gen. Purpose Elastomeric Mount	Captive Elastomeric Mount	Cupmount	Pneumatic Rubber Mount	"Quiet" Attachment Devices
Fans and Blowers	☐	☐	☐		☐	☐	☐		☐	☐
Vehicular Air-Conditioners	☐	☐	☐				☐	☐	☐	☐
Air-Conditioning Units	☐	☐	☐		☐	☐	☐	☐	☐	☐
Computers	☐		☐	☐	☐	☐	☐	☐		☐
Electronic Printers			☐	☐						☐
Range Hoods										☐
Construction Equipment & Vehicles	☐	☐		☐			☐	☐	☐	
Farm Equipment & Vehicles	☐	☐		☐			☐	☐		☐
Recreational Vehicles	☐	☐					☐			☐
Rapid Transit Vehicles	☐	☐		☐			☐	☐	☐	☐
Centrifugal Machinery	☐					☐			☐	
Centrifugal Extractors									☐	
Washer-Extractors									☐	
Aircraft Indicators			☐	☐						☐
Air-Conditioners & Refrigeration Compressors	☐	☐			☐	☐			☐	
Centrifugal & Reciprocal Compressors	☐	☐			☐	☐			☐	
Motor-Generator Sets	☐	☐		☐		☐	☐	☐		
Vibratory Feeders, Conveyors	☐					☐	☐	☐	☐	
Heavy Duty Trucks	☐	☐		☐				☐	☐	☐
Pumps	☐	☐				☐	☐		☐	
Marine	☐						☐	☐		
Electronic Cabinets		☐	☐			☐	☐	☐	☐	☐

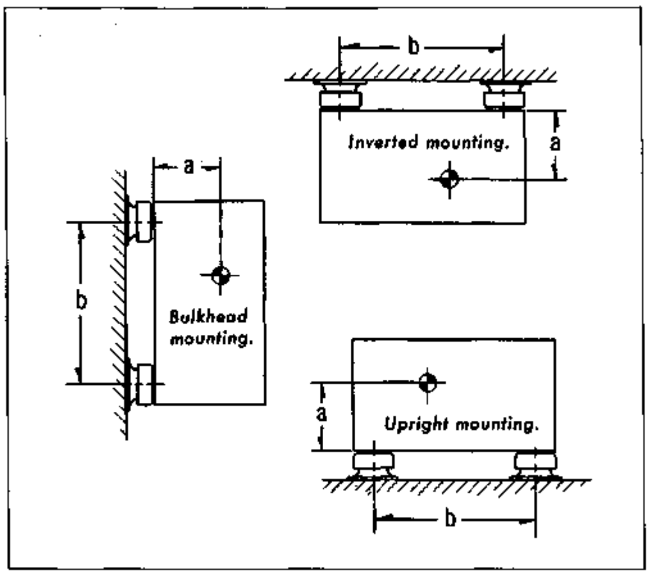


Figure 17 – Aspect ratio determination.

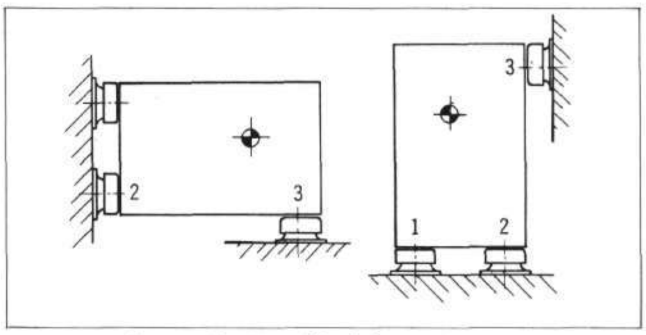


Figure 18 – Use of Stabilizing Mounts.

instance, has a shock transient overload capacity of up to 100 times rated static load. As can be seen, most isolators described in this article have some type of "captive" or "fail-safe" construction feature.

Summary and Recommendations

This article has been intended to serve as a guide to help the reader better understand the meaning of some of the more important characteristics of isolators used in integral components in OEM machinery and equipment. If the reader has a good, basic knowledge of shock, vibration, and noise control theory, this article will be a helpful aid in the process of selecting and applying component isolators.

In all cases, however, it is recommended that mount selections be confirmed with the manufacturers of the isolators. Experience in mounting machines is invaluable and a simple check of selection prior to purchase will not only prevent needless problems if errors have been made in the selection process, but may also avoid having to reinvent a solution that already exists.

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