INDUSTRIAL APPLICATIONS AND MOUNTING CONSIDERATIONS

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SELECTION OF STANDARD BARRY ISOLATORS FOR IOEM-TYPE APPLICATIONS

General Industrial Equipment

I Business Machines:

Typical Equipment Requiring Isolation

Delicate electronic components
Winchester Disk drive
Computer peripheral equipment
Electronic printers
Copiers
Chassis support for all types

Typical Sources of Disturbance:

Electric motors
Moving mechanical devices
Fans and blowers
Air compressors
Pumps

Desired Isolator Characteristics:

Low natural frequency (5 to 20 Hz)
Structure borne noise reduction
Easy installation
Low cost

Recommended Isolators:

6300-6550
S-mount
6800 Series
7800 Series
FM 300 & FM 450 Fan Mts.
EPF-100 & EPF 500 Equipment Feet
FS & ME 100 Series
FS & ME 500 Series

II. Industrial Machinery and Equipment

Typical Equipment Requiring Isolation:

Motors- Electric and Internal Combustion
Generators
Pumps
Centrifuges

Refrigeration Equipment
Fans and Blowers
Industrial washing machines
Compressors
Air conditioners
Range hoods
Centrifugal extractors
Vibratory feeders

Typical Source of Disturbance:

Mass Unbalance of:
Motors and Engines
Generators
Pumps
Fans and Blowers
Compressors

Desired Isolator Characteristics:

Low natural frequency (5–15 Hz)
Wide load range
Low profile
Easy installation
Low cost

Recommended Isolators

633A
670 & 297
661
990 & 915
Cupmounts
22000 series
500 series
Stabl-levl
Special designs

Vehicles

I. Construction Equipment, Farm Equipment, and Trucks-- Loaders, Scrapers, Tractors, Mowers, Backhoes, Skidders, Crawlers, Heavy-Duty Trucks, Forklifts

Typical Equipment Requiring Isolation:

Cabs
Air conditioners
Radiators

Battery boxes
Seats
Communications equipment
Compressors
Engines

Typical Source of Disturbances

Engines
Transmissions
Air conditioners
Pumps
Hydraulic and pneumatic equipment
Off–road inputs
Rough–road terrain
Normal road inputs

Desired Isolator Characteristics:

Low natural frequency (5 to 20 Hz)
Built–in snubbing for shock inputs (2 to 7 g's)
Wide load range
Compact size
Failsafe
Radial Stability

Recommended Isolators:

Cupmounts
*22000 series
*500 series
5200 series
Stabl–levl
Special designs

*most commonly recommended

II Rapid Transit Vehicles

Typical Equipment Enclosed

Car suspensions
Air conditioners
Pumps
Motors
Generators
Bolsters
communications Equipment

Typical Sources of Disturbances:

Structural resonances
Reciprocating equipment
Road beds or rails

Desired Isolator characteristics:

Natural frequency 15 to 30 Hz
Isolate high-frequency transients
Isolate high-frequency, steady-state vibration
High-deflection shock attenuation

Recommended Isolators:

990 & 915
Cupmount series
22000 series
500 series
Stabl-levl
Special designs

III. Recreational Vehicles - Motor Bikes and Motorcycles, Campers, Snowmobiles, Boats and Golf Carts

Typical equipment requiring Isolation:

Seats
Horns
Footrests
Generators
Communication equipment
Vehicle structure

Typical Source of Disturbance:

Terrain
Power plants
Motors/Generators

Desired Isolator Characteristics:

Low natural frequency (10 to 20 Hz)
Easy installation
Built-in snubbing for shock inputs (8 to 10 g's)
Low cost

Recommend Isolators:

990 & 915
22000 series

500 series
6300-6550 series
5200 series
Plate type
Special designs

IV. Packaging (Shipping Containers)

Typical Equipment Isolated:

Computers
Gyroscopes
Printers
Copiers
Electronics
Laboratory equipment
Optical equipment
Business machines in general

Typical Source of Disturbance

Handling
Air, sea, truck, and train transport

Desired Isolator Characteristics:

High shock deflection (1 to 3 inches)
Low natural frequency (7 to 10 Hz)
Stability
Low stress on bonds

Recommended Isolators

Cupmount
Stabl-levl
Shipping container mounts
VHC series
2Kl, 2K2 series
Special designs

500 SERIES

MAXIMUM LOADS ON BARRY 500 SERIES MOUNTS

We recommend that the maximum load applied to the mounts (dynamic and static load) not exceed 10 times the maximum rated load for a particular size. Using this formula, the maximum dynamic and static load in the vertical down and lateral directions for each size is as follows:

505 - 900 lbs
506 - 1,500 lbs
507 - 3,300 lbs
508 - 5,700 lbs
510 - 10,200 lbs
512 - 15,000 lbs
516 - 27,000 lbs

These are not the loads at which metal deformation takes place or even elastomer failures. The loads at which this occurs are higher, but, since in the above stated directions elastomer failure will occur before metal deformation, we feel that deformation loads are not relevant. This is not the case in the vertical up direction. Permanent deformation of the hold down ears occurs before elastomer failure at the following loads:

505 - 900 lbs
506 - 1,000 lbs
507 - 2,600 lbs
508 - 3,300 lbs
510 - 3,300 lbs
512 - 3,800 lbs
516 - 9,000 lbs

These loads are based on tests on the 516 and 510 series, and calculated values for the rest.

500 AND 2200 SERIES

22000 AND 500 SERIES MAXIMUM TIGHTENING TORQUES

BARRY P/N	BOLT DIA.	TORQUE (DRY)	TORQUE (LUBRICATED OR PLATED)
22001	.375	12	9
22002	.500	120*	90*
22003	.625	240*	180*
22004	.875	660*	500*
22005	1.000	1000*	740*
505	.313	25*	20*
506	.375	45*	35*
507	.438	80*	60*
508	.500	120*	90*
510	.625	240*	180*
512	.750	120	90
516	1.000	365	275

* - GRADE 8 BOLT TORQUE

If higher bolt torques are anticipated for applications such as construction equipment cab systems, special heat treated metal components can be employed to accomodate the higher loads. For now, applications requiring special material components are to be handled as nonstandard, and price quotations should be requested.

Practical Considerations

A. INTERACTION WITH THE SUPPORT STRUCTURE

Regardless of how carefully the stiffness characteristic and isolator locations are selected, the whole exercise can be for nought if the interactions with the support structure is not considered. In the usual analysis, the following assumptions are made:

1. The supported structure is a compact, rigid mass
2. The isolator is an ideal massless spring.
3. The foundation is considered as infinitely rigid.

Now, in fact, not one of these assumptions is true. The supported structure is not a concentrated mass but has many resonances. Particularly are the mounting brackets that attach the unit to the isolator likely to have an appreciable degree of flexibility. The isolator (assumed to be of the elastomeric type) is not an ideal massless spring but will have what is called standing wave resonances.The foundation is not and cannot be infinitely stiff because if it were it would not respond to a vibratory force. Since the foundation does respond to the force, its response must have an effect on the components mounted on it. In reality the mounting bracket, isolator, and support structure are springs in series.

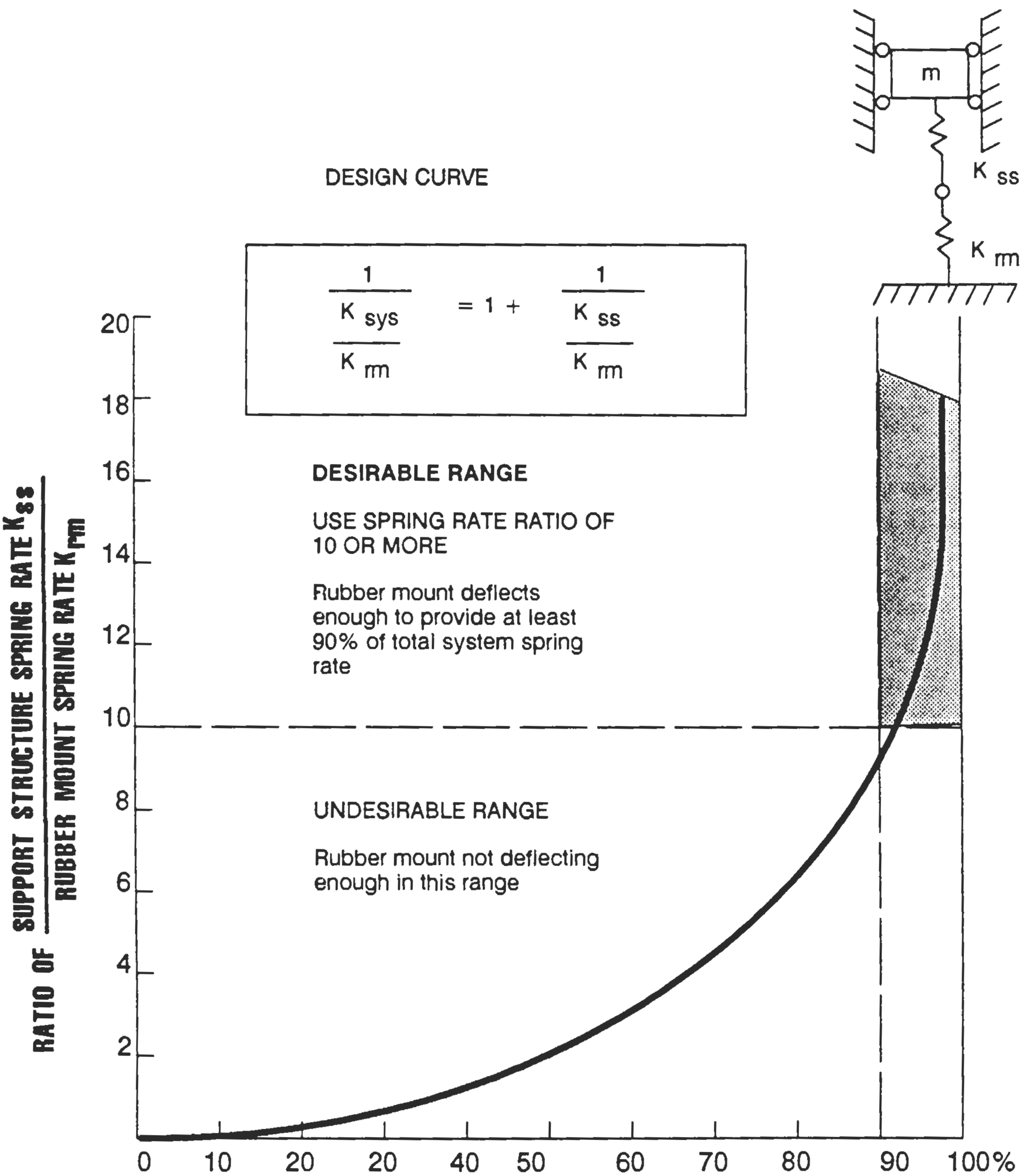
A relatively simple analysis shows that the transmitted force across this series of springs can be represented by

$$T = \frac{\text{M bracket} + \text{M support structure}}{\text{M bracket} + \text{M isolator} + \text{M support structure}}$$

Where:

M is the mobility of each element, T is the ratio of the absolute value of the sinusoidial force acting on the engine and the resulting force on the support structure. From this we see that the isolator is relatively ineffective unless its mobility is high in comparison to both the engine bracket or support structure; i.e., these elements must be considerably stiffer that the isolator. Practice has shown that the stiffness of the engine bracket and support structure should be at least 10 times greater than the isolator. See figure I. The importance of having the proper structure stiffness cannot be over emphasized. This is a very real problem because designers, in an attempt to make their equipment lighter and to reduce manufacturing costs are using structures that do not meet this criterion with an adverse effect on the isolation system performance.

FIGURE 1



$\frac{\text{TOTAL SYSTEM SPRING RATE } (K_{sys})}{\text{RUBBER MOUNT SPRING RATE } (K_{rm})}$

B. NOISE REDUCTION

Today designers are as much concerned in noise reduction as they are in vibration control. Structureborne disturbances can be grouped into two bands:

mechanical vibration 0 – 50 H_z
noise 50 –20K H_z

Thus noise is merely high frequency vibration and therefore elastomeric isolators, which provide excellent low frequency isolation, should provide excellent isolation for noise as well. Under ideal conditions such isolators provide 9 – 12 db per octive attenuation.

However, in many cases, while the mechanical vibration problem is not diminished and in some rare cases actually increases.

The reasons for this are:

1. Airborne and structureborne noise both exist and since the isolators can reduce only the structureborne noise, the airborne noise which probably predominates is not affected.
2. Structureborn noise attenuation does not depend solely on the isolator performance characteristics. The ideal conditions mentioned above with regards to isolator attenuation assume infinitely rigid mass and support structure, conditions, as noted earlier, that cannot exist. Therefore, the farther one deviates from the ideal condition, the less attenuation can be expected. Thus the most common cause for poor system performance in noise reduction is the lack of structural rigidity in the supporting frame. This explains why for example, a particular power plant and isolation system may perform well in one vehicle, while the same power plant and isolation system performs inadequately in a different vehicle from the standpoint of noise reduction.

Because the noise problem is so complex it is not possible to analyze or predict isolator performance in advance without a complete knowledge of the impedance characteristics of the supported mass and supporting structure.This type of information is virtually impossible to obtain from the manufacturers of industrial equipment.

On occasion the phenomena called “standing waves” is blamed for poor performance of elastomeric isolation at frequencies in the noise range. A “standing wave” is an internal mount resonance that is caused by the wave length of sound coinciding with two times (2x) a major isolator dimension. Prediction of the first standing wave for a simple elastomeric shape is relatively easy but this is not the case with the complicated shapes do occur, they are not a problem unless they coincide with a structural resonance, This is a rare occurrence. The point bears repeating: poor structural noise attenuation is primarily due to a support foundation that is too flexible.

C. BONDED VS. UNBONDED ISOLATORS

Figure 2 illustrates some bonded and unbonded isolators. In a bonded isolator, metal inserts are bonded to the elastomer on all load carrying surfaces(see the lower right hand isolator in Figure 2).
Is there a difference besides cost? (The fully bonded units are usually more expensive than unbonded isolators.) Yes, some important differences.

One important difference results from the fact that a more constant stress distribution is developed in the elastomer body by bonding the metal inserts to the load bearing surfaces. Unbonded isolators usually suffer from relatively high stress concentrations that shorten the service life by causing premature cutting or tearing of the elastomer. Often excessive abrasion occurs at the rubber to structure interface. Greater care to avoid sharp corners from contacting the elastomer is required with unbonded units and all edges must be properly radiused. So while the initial cost is low, the cost for extra machining of the support structure and the reduced service life often make unbonded isolators a poor bargain.

Another important difference relates to how elastomer behaves under load. Rubber has approximately the same compressibility as water and therefore, for all practical purposes, is incompressible.Thus where an elastomer pad deflects under load its volume remains constant, only its shape is changed. The rubber is said to "bulge" under load. By controlling the degree or ability to bulge we control the load-deflection characteristics of the isolator.

An isolator with metal elements bonded to the load bearing surfaces has a fixed degree of bulge because the elastomer along the bond line cannot move and therefore, remains in a fixed position regardless of the load or environmental conditions. Not so with an unbonded unit. Its ability to bulge depends on the condition of the rubber to metal interface. When a piece of equipment is new and all the surfaces are clean and dry, the difference between the bonded and unbonded unit's ability to bulge is negligible. But if oil or sand works its way into the rubber to metal interface of the unbonded unit, the ability of the elastomer to bulge is greatly increased and therefore the load deflection characteristics originally assumed in the selection of the isolator no longer exist. The isolator can now exhibit load deflection characteristics that are 50% less than when the equipment was new, which, in many cases, can cause the isolation system to malfunction. Thus where consistent load-deflection characteristics are required for the life of the equipment, bonded isolators must be used.

D. FATIGUE LIFE CONSIDERATIONS

Regardless of geometry, both metals and elastomers exhibit some basic fatigue failure properties resulting from repeated cyclic loadings.

In many publications, the lower law equation, shown below, has been suggested for representing the fatigue life of elastomers and metals.

$$N = \left(\frac{K}{\epsilon}\right)^b$$

Where: N = number of cycles to failure

ϵ = maximum shear strain resulting from cyclic load

K, b = empirical coefficients for each type of elastomer

The empirical coefficient K depends somewhat on the shape factor of the isolator. For neoprene K can be as high as 1000, while for natural rubber, 1250 for high-shape factors. For both, "b" equals 5 as noted above. From this equation we can appreciate the important effect dynamic strain has on service life.

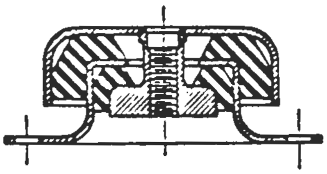
The S-N curve of aluminum is typical of that for elastomers in that the damage done by cyclical loads, regardless of their magnitude, is cumulative. Therefore, to determine fatigue life, a spectrum of all cyclic loads, their magnitude, frequency, and duration, is required. With this information we can calculate the expected service life using Minor's linear cumulative damage law to sum up the damages for each fatigue spectrum condition and to establish the overall damage coefficient.

The damage coefficient is defined as:

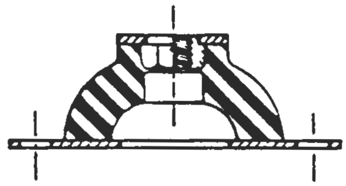
$$D = \sum_{i=1}^{I} \frac{\eta_i}{N_i}$$

Where η_i = number of cycles at strain ϵ_i
N_i = number of cycles to failure at strain ϵ_i
I = number of different cyclic loading conditions

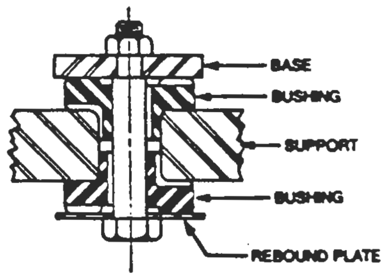
FIGURE 2



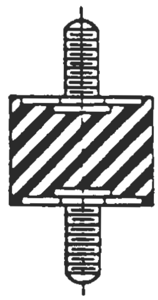
Typical compression isolator handles compression loads in all directions. Preloaded, unbonded rubber elements have load range from 12 to 1800 lb/mount.



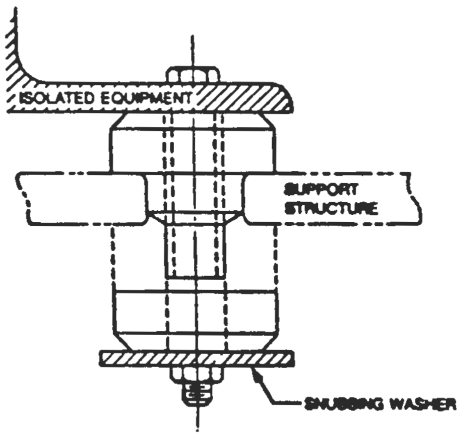
Buckling column Isolator uses fully bonded element for load range from 60 to 260 lb/mount.



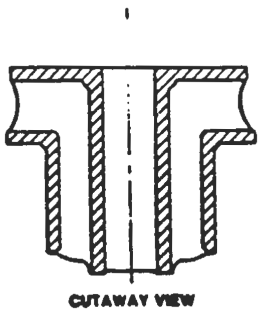
Unbonded ring and bushing isolators have ribbed protrusions for gradual snubbing under shock. Load range, 1 to 350 lb/mount.



Bonded shear or compression isolator has relatively high stiffness in compression compared to shear. Load range, 3 to 260 lb/mount in compression.



Bonded bushing with rebound ring is preloaded in axial and radial directions, can be loaded in all directions. Load range, 40 to 1600 lb/mount.



Shouldered core design has fully bonded element, can be loaded in any direction. load range, 50 to 2700 lb/mount.

Some types of bonded and unbonded isolators

The number of cycles η_i can be written as:

$$\eta_i = \left(\frac{P_i}{100}\right)(T) \quad (f)$$

Where: P_i = percent of time for each cyclic load

T = total time of service life

f = frequency of each cyclic load

Substituting the above equation and the power law equation into the damage law equation results in:

$$D = \frac{(T)(f)}{100\,(K)^b} \sum_{i=1}^{I} (P_i)\,(\epsilon_i)^b$$

Since "D" represents the amount of cumulative damage attained after a given time T (according to the hypothesis, when D = 1, the part fails), the smaller the value of "D," the longer the service life.

It should be noted that unlike metals elastomers do not experience catastrophic–type fatigue failures. The failure begins as a tear at the point of highest cyclic shear strain which is generally on the outer extremity (and, therefore, in many cases visible) and gradually propagates through the body of the elastomer. The result is a gradual reduction in stiffness that usually becomes apparent before total failure is achieved.

Another factor contributing either to the success or to the premature failure of an isolation system is the effect of the mean strain on fatigue life. For elastomers which crystallize under high strains (such as neoprene and natural rubber), the effect of mean strain is to greatly increase life if the minimum cyclic stress is always either plus or minus and never passes through zero.

Proper preloading of the isolator is an efficient way to prevent zero stress reversals.

From the above it should be appreciated that isolators of sufficient size to achieve lower strain levels are the wisest choice.

E. DAMPING REQUIREMENTS

Often times the terms "Damping" and "Isolation" are used as being synonymous. However, they are distinctly different. Isolation is the reduction of the magnitude of force or motion transmitted across the isolator. Damping is the ability of elastomeric isolators to convert mechanical energy into heat and thus dissipate that energy.

Damping is advantageous in an isolation system when the system has to function at or near resonance since damping controls the response at resonance or the transmissibility of the system. This situation can occur if the system is subject to a random vibratory input or the supported equipment passes through resonance relatively slowly on startup or slowdown.

However, damping reduces the efficiency of the suspension system in the isolation range as shown in figure 3. Therefore, only enough damping as needed should be incorporated in the isolator design. The moderate damping of Natural and Neoprene elastomers is sufficient for most applications. Where more damping is needed, Polybutadiene, Butyl, and Silicone elastomers are considered.

F. BELT POWER TAKE-OFF

When the power take-off is by belt, serious isolation problems develop if the design is such that the belt is routed across the mounting system. Isolators flexible enough to provide isolation will not maintain pulley alignment or belt tension. Isolators stiff enough to provide this are usually too stiff to achieve the desire isolation. A compromise with reduced isolation performance is always the result.

Vehicle designers should be encouraged to avoid this situation where possible by mounting the engine and driven unit on a common base which can be isolated from the main frame. If this is not possible, then the only thing the designer can do to lessen the problem is to locate the isolators, on whichever end of the engine has the drive pulley, as close to the plane of the pulley as possible. This reduces the loads on the isolators due to the pulley offset moment to a minimum value.

The most expedient solution to this type of problem is trial and error until the isolators that provide the best possible performance are found. The belt is in effect a spring in parallel to the isolation system and since virtually no data on the dynamics of such a spring are available, a meaningful dynamic analysis to determine the proper spring rates of the isolators is impossible.

TRANSPORATION VIBRATION SPECTRA