Choosing the best WRI for the job

To select the right WRI for the job, it is best to take full advantage of Vibro/Dynamics' engineering support by supplying the data in the attached form (see last page in the catalog). A few guidelines are detailed below in order to proceed to a preselection in simple cases.

1. Define the mounting attitude

The isolator's mounting attitude is usually dictated by the equipment design. Common examples are shown below; for additional mounting attitudes consult Vibro/Dynamics.

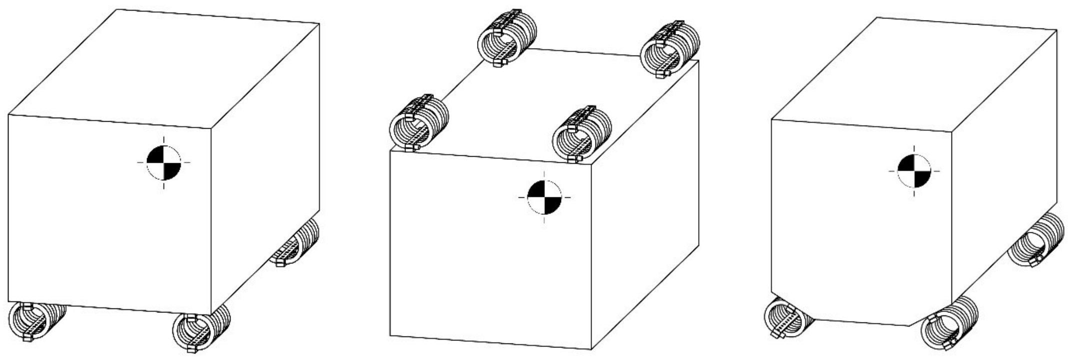


Figure 14. Mounting attitude examples: Compression, Tension, Compression/Roll 45°

2. Calculate the sprung mass per isolator

If the Center of Gravity (CG) is located at the geometric center of the sprung mass m, then n identical isolators can be used, and $m_{static} = m/n$ is the sprung static mass per isolator. If this isn't the case, then an individual calculation should be carried out for every type of isolator and/or loading. A good rule of thumb is to arrange all the isolators to have the same static deflection, in which case the system is then called "balanced".

3. Select for shock

Since shock is often the sizing factor, it makes sense to first consider shock, and then check vibration. There are two main ways of dealing with shock; either (a) shock by an instantaneous velocity change or (b) shock by an instantaneous displacement step.

(a) Velocity change

It is assumed that the shock is an impulse or impact, for example a free fall drop. This is regarded as an instantaneous velocity change between the sprung mass and the unsprung mass.

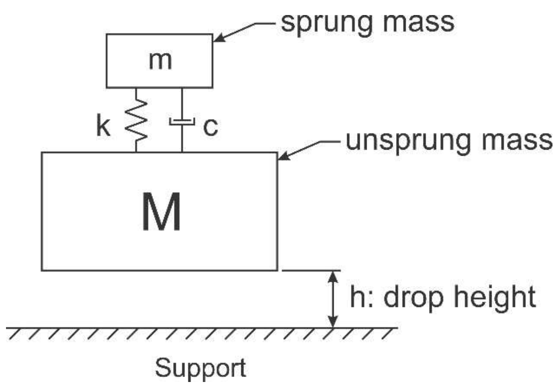


Figure 15. Simplified model used for a velocity shock

Example: A linear, 1-DOF system (W = 520 lbf), having its CG at its geometric center, is mounted in compression on 4 identical WRIs. The input to this system (Fig. 16) is a shock of a ½ sine pulse with a_0 = 45g amplitude and τ = 6 ms duration, or equivalently a drop height of 5.7 in.

Assuming that the duration of the pulse input is short compared to the duration of the response, the input can be characterized as a velocity change, as explained in Section 3.1.3:

$$\Delta v = \int_0^\tau a(t)dt = \frac{2a_0\tau}{\pi} = \frac{2 * 45 * 386.1 * 0.006}{\pi} = 66.4\ in/s$$

Recall in Section 3.1.3, the following equations were derived:

$$k_{shock} = \frac{F_{shock}}{d_{shock}}$$

$$x_{max} = \sqrt{\frac{\Delta v^2 m}{4k_{shock}}} \qquad (m = \frac{W}{386.1 in/s^2})$$

$$a_{max} = \frac{4k_{shock}x_{max}}{m}$$

Here in the example the solutions are:

- <u>FH56-2008:</u> *Per the corresponding datasheet in compression (p.53): F_{shock} = 3500 lbf and d_{shock} = 1.2 in. Therefore:*
  k_{shock} = F_{shock} / d_{shock} = 3500 / 1.2 = 2917 lbf/in
  x_{max} = √ [($\Delta v^2 m$) / ($4k_{shock}$)] = √ [(66.4^2 x 520 / 386.1) / (4 x 2917)] = ***0.71 in***
  a_{max} = $4k_{shock}x_{max}$ / m = (4 x 2917 x 0.71) / (520 / 386.1) = ***6151 in/s^2 (15.9g)***

- <u>FH56-4808:</u> *Per the corresponding datasheet in compression (p.53): F_{shock} = 860 lbf and d_{shock} = 4.0 in. Therefore:*
  k_{shock} = F_{shock} / d_{shock} = 860 / 4.0 = 215 lbf/in
  x_{max} = √ [($\Delta v^2 m$) / ($4k_{shock}$)] = √ [(66.4^2 x 520 / 386.1) / (4 x 215)] = ***2.6 in***
  a_{max} = $4k_{shock}x_{max}$ / m = (4 x 215 x 2.6) / (520 / 386.1) = ***1660 in/s^2 (4.3g)***

Another way to find the solution to this shock problem is through the use of the Shock Response Spectrum (SRS). Shown in Figure 17 is the SRS of the ½ sine pulse with 45g amplitude and 6 ms duration. To determine the resulting maximum acceleration, first the system's natural frequency is calculated using $f_n = \frac{1}{2\pi}\sqrt{\frac{4k_{shock}}{m}}$, then the corresponding maximum acceleration can be readily found from the SRS curve. For a linear system, if its maximum acceleration is known, then its maximum dynamic deflection can be determined using the following formula:

$$x_{max} = \frac{a_{max}}{(2\pi f_n)^2}$$

For this example, using the SRS curve:

- FH56-2008: f_n = 1 / (2π) x √ [$4k_{shock}$ / m] = 1 / (2π) x √ [(4 x 2917) / (520 / 386.1)] = 14.8 Hz
  From Figure 17 this results in: a_{max} = **15.3g**, and x_{max} = a_{max} / (2πf_n)^2 = 15.3 x 386.1 / (2π x 14.8)^2 = **0.68 in**

- FH56-4808: f_n = 1 / (2π) x √ [$4k_{shock}$ / m] = 1 / (2π) x √ [(4 x 215) / (520 / 386.1)] = 4.0 Hz
  From Figure 17 this results in: a_{max} = **4.1g,** and x_{max} = a_{max} / (2πf_n)^2 = 4.1 x 386.1 / (2π x 4.0)^2 = **2.5 in**

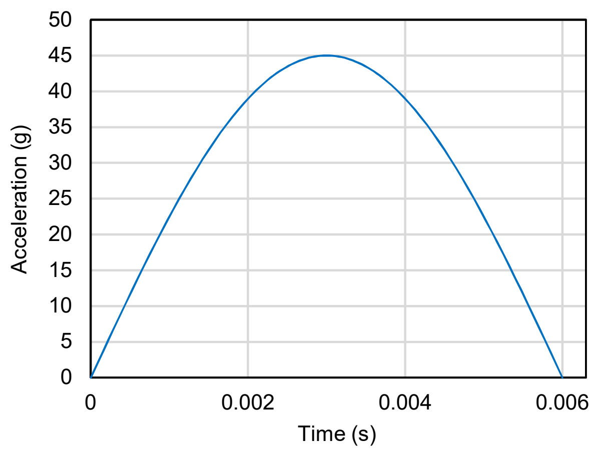


Figure 16. ½ sine pulse input of 45g amplitude and 6 ms duration

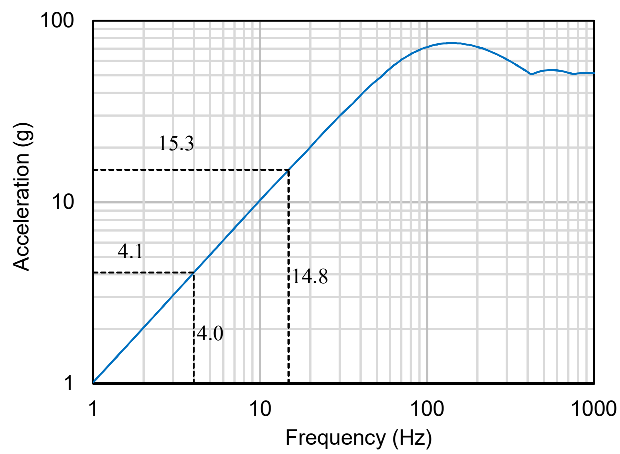


Figure 17. SRS of the ½ sine pulse input assuming a 15% damping ratio

Using the SRS method is advantageous compared to the energy method, since there is no need to make the assumption that the duration of the input is short compared to the response's duration. The SRS method works for any input, as long as the system is linear.

(b) Displacement step

It is assumed that the shock is an instantaneous displacement of the support foundation, which is transferred immediately to the isolators. For example, subsequent to a non-contact underwater explosion.

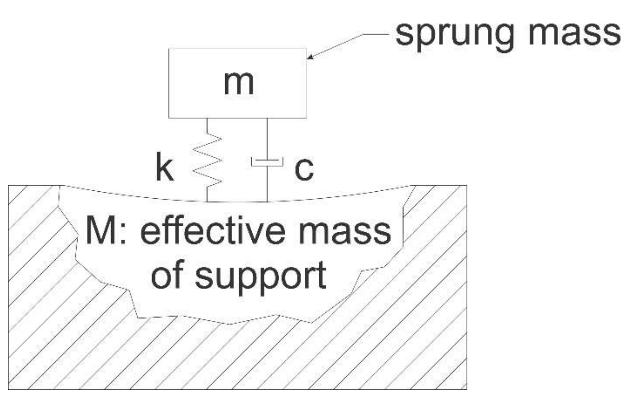


Figure 18. Simplified model of a displacement shock

Example: A linear, 1-DOF system (W = 520 lbf), having its CG at its geometric center, is mounted in compression on 4 identical WRIs. The input to this system is a displacement step of 2.5 in, as per typical naval shock response spectra. Figure 19 shows the input in terms of displacement, velocity, and acceleration vs. time.

For a soft enough system, the maximum dynamic deflection in the isolator is equal to the displacement step. Therefore, the corresponding acceleration can easily be deducted by first determining the average shock stiffness from the datasheets ($k_{shoc} = F_{shoc} / d_{shoc}$), then using Newton 2nd Law ($\sum F = ma$):

$$a_{max} = \frac{4k_{shock}\Delta d}{m}$$ (Δd: displacement step, and $m = \frac{W}{386.1 in/s^2}$)

Here in the example the solutions are:

- FH56-4408: *Per the corresponding datasheet in compression (p.53): F_{shock} = 1400 lbf and d_{shock} = 2.9 in. Therefore:*
  $k_{shock} = F_{shock} / d_{shock}$ = 1400 / 2.9 = 483 lbf/in
  $x_{max} = \Delta d$ = 2.5 in
  $a_{max} = 4k_{shock}\Delta d / m$ = (4 x 483 x 2.5) / (520 / 386.1) = ***3586 in/s² (9.3g)***

- FH56-4808: *Per the corresponding datasheet in compression (p.53): F_{shock} = 860 lbf and d_{shock} = 4.0 in. Therefore:*
  $k_{shock} = F_{shock} / d_{shock}$ = 860 / 4.0 = 215 lbf/in
  $x_{max} = \Delta d$ = 2.5 in
  $a_{max} = 4k_{shock}\Delta d / m$ = (4 x 215 x 2.5) / (520 / 386.1) = ***1596 in/s² (4.1g)***

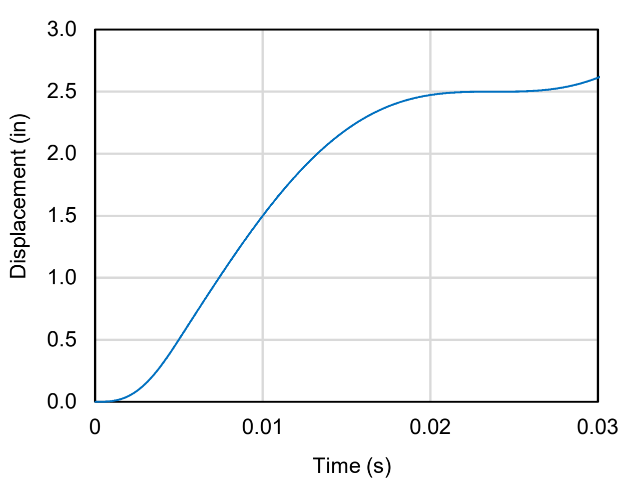
Another way to find the solution to this shock problem is through the use of the SRS. Shown in Figure 19 is the SRS of the 2.5 in displacement input. This way of representing the SRS as a tripartite logarithmic plot, with pseudo-velocity (relative velocity between the support and sprung mass) as the vertical axis, is typical of a naval design spectrum. Here again, finding the maximum response is determined by calculating the natural frequency using $f_n = \frac{1}{2\pi}\sqrt{\frac{4k_{shock}}{m}}$, then reading the corresponding acceleration and displacement from the tripartite logarithmic SRS curve.

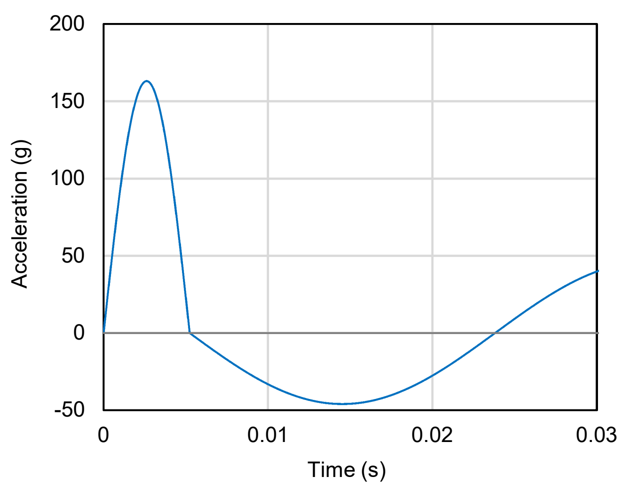
For this example, using the SRS curve:

- FH56-4408: *f_n = 1 / (2π) x √ [$4k_{shock} / m$] = 1 / (2π) x √ [(4 x 483) / (520 / 386.1)] = 6.0 Hz*
  From the SRS in Figure 19 this results in: x_{max} = 2.5 in and a_{max} = ***8.6g***

- FH56-4808: *f_n = 1 / (2π) x √ [$4k_{shock} / m$] = 1 / (2π) x √ [(4 x 215) / (520 / 386.1)] = 4.0 Hz*
  From the SRS in Figure 19 this results in: x_{max} = 2.5 in and a_{max} = ***4.3g***

The two chosen solutions above are in the SRS region of constant displacement, which is necessary for attenuation of shock input.





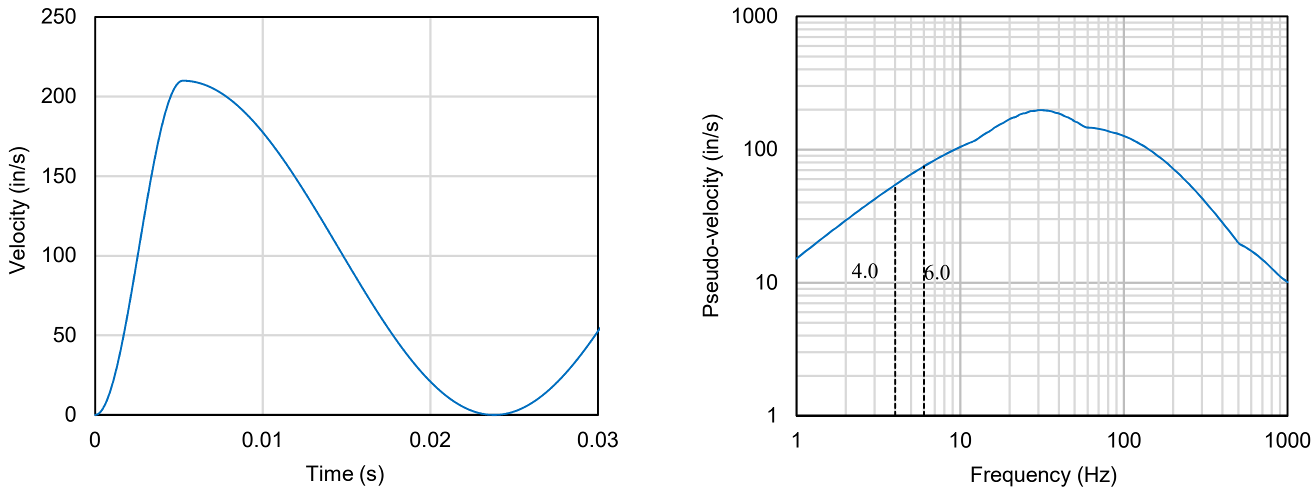


Figure 19. Displacement, velocity, acceleration and corresponding SRS of a 2.5 in displacement step input of 30 ms duration

(c) Shock testing machines

In naval applications, most standards define the design input for a system in the form of an acceleration time history or SRS, adapted to the size and type of vessel, dimensions and weight of the equipment, and location within the vessel. The previous examples described the process of selecting isolators when dealing with explicit, well defined inputs (e.g. ½ sine pulse acceleration or displacement step). However, in some naval applications test inputs are not as straightforward, as they are based on a complicated shock testing apparatus. Each equipment is tested under the specified MIL-S-901D, and will either pass or fail the qualification test.

The complexity of the test and its parameters makes a simple analytical approach very difficult. Combining this with the financial cost and time needed for testing, the design engineer will often oversize isolator selection to ensure passing the test the first time. This methodology is less than ideal, which is why the Socitec Group has developed a full model of the various shock machines (LWSM, MWSM, FSP, and soon DSSM) using the SYMOS software package. These models divide the machines into a number of rigid bodies connected by nonlinear springs and dampers to which the shock parameters are applied as initial conditions. The models even include the shock deck simulator at the appropriate frequency, as well as all test setup conditions such as height of hammer drop, table travel, total weight on anvil table, etc. The model parameters have been carefully verified and calibrated with data from physical tests. One example is shown below.

An electronics cabinet mounted vertically on VBFN45-130 elastomeric isolators was tested with the MWSM under the following test conditions:

- Height of hammer drop: 2.3 ft
- Table travel: 3 in

The measured response from the physical test was compared to the response calculated by the SYMOS model of the MWSM, as shown in the comparative time histories in Figure 23.

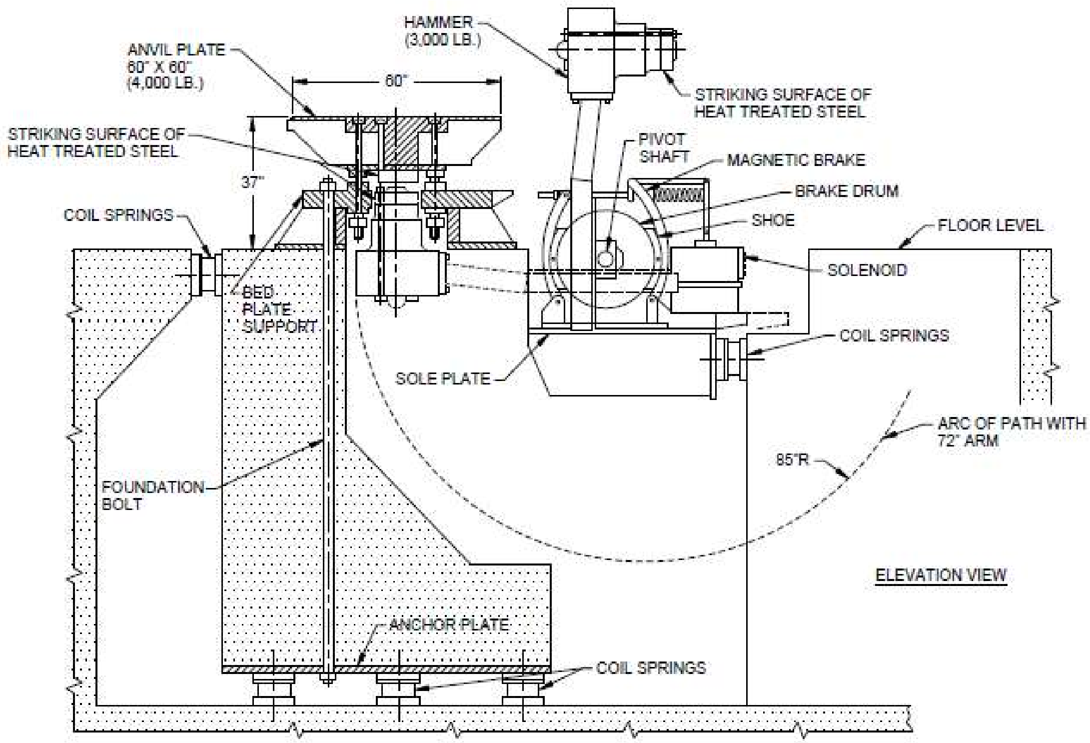


Figure 20. High impact shock testing machine for medium weight equipment per MIL-S-901D

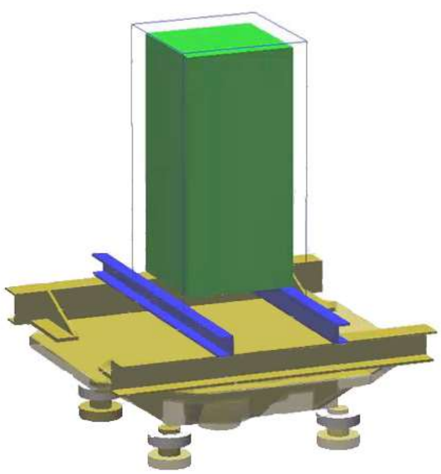


Figure 21. SYMOS model of the MIL-S-901D MWSM

Figure 22. Electronics cabinet mounted for testing

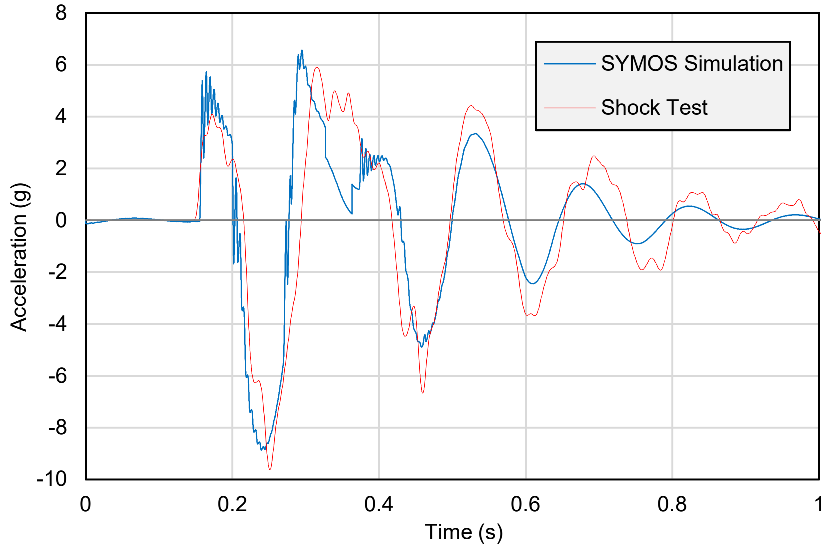


Figure 23. Comparative time histories of measured vs. calculated response with VBFN45-130 elastomeric isolators

While these SYMOS models do not replace the requirement for physical qualification tests, they provide the engineer with insight that can help avoid oversizing the isolators and potentially reduce the number of testing iterations.

As a rule of thumb, in order to achieve a response of 10–15g under the MIL-S-901, it is recommended for the isolators to meet the following:

- Static deflection of 0.2–0.4 in, depending on the application parameters
- Minimum dynamic deflection capability of 3 in

Usually, WRIs are smaller compared to elastomeric isolators with the same response characteristics. In most cases, the size of WRIs used is 5–6 in. Finally, even though these SYMOS models were designed for use with isolators, they can also be used for hard-mounted equipment.

(d) Additional considerations

When the unsprung mass is large relative to the sprung mass, the cable will go in tension. Since WRIs are significantly stiffer in tension than in compression, the sprung mass will experience a larger acceleration when the cable is in tension. For this reason, if the unsprung mass (container, etc.) is substantially larger than the sprung mass, it is advised to multiply the calculated accelerations above by an adjustment factor between one and two. Vibro/Dynamics can help determine the adjustment factor specific to the application.

Since the system in the examples was assumed linear with only 1-DOF, the results are typically used as initial approximations. This is far from what is encountered in real life, since it does not account for coupling. A better model is a nonlinear multiple degree of freedom system that Vibro/Dynamics has the experience and expertise to solve. It is important to use conservative safety margins as it is not unusual to have critical parameters such as mass and CG location change significantly throughout the project. The cost of slightly oversizing the isolators is minute compared to the cost of redesigning the whole system. Some guidelines are supplied in Table 2 for typical applications in terms of static deflection.

Table 2. Recommended static deflection per type of application

Application	Naval	Shipping Containers	Armored Vehicles	Industrial	Nuclear	Space MGSE
Recommended Static Deflection (in)	0.20–0.40	0.04–0.16	0.04–0.12	0.04–0.40	0.20	0.40–0.80

Per the datasheets, it should be ensured that the static and additional dynamic deflections are within the isolators' capabilities.

If the static load (F_{st}) is less than the maximum static load ($F_{st,max}$) provided in the datasheets, then Figure 24 can be used to determine the static deflection (d_{st}) in the WRI, taking into account its nonlinearity.

Example: A system (W = 520 lbf), having its CG at its geometric center, is mounted in compression on 4 identical WRIs. The static load per mount is: $F_{st} = 520 / 4 = 130$ *lbf.*

- FH56-2008: *Per the corresponding datasheet in compression (p.53):* $F_{st,max} = 1200$ *lbf and* $d_{st,max} = 0.23$ *in*
  $F_{st} / F_{st,max} = 130 / 1200 = 0.11$. *From Figure 24, this results in:* $d_{st} / d_{st,max} = 0.24$
  Therefore: $d_{st} = 0.24 \times d_{st,max} = 0.24 \times 0.23 =$ ***0.06 in***

- FH56-4808: *Per the corresponding datasheet in compression (p.53):* $F_{st,max} = 290$ *lbf and* $d_{st,max} = 0.75$ *in*
  $F_{st} / F_{st,max} = 130 / 290 = 0.45$. *From Figure 24, this results in:* $d_{st} / d_{st,max} = 0.59$
  Therefore: $d_{st} = 0.59 \times d_{st,max} = 0.59 \times 0.75 =$ ***0.44 in***

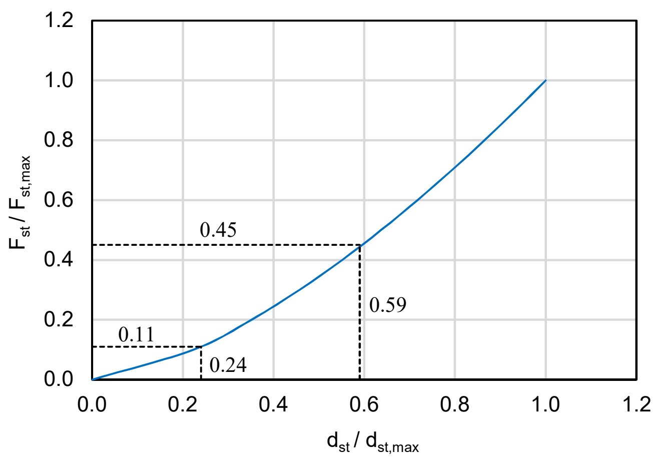


Figure 24. Curve used to determine the static deflection (d_{st}) in a WRI under a static load (F_{st})

4. Select for vibration

In practice, the most interesting parameter in vibration is the resonant frequency of the system. Given that WRIs are nonlinear, the resonant frequency depends on the displacement amplitude in the WRIs. For a 1-DOF system under a constant peak to peak displacement input, it is possible to assess its corrected resonant frequency (f_{res}) using the corresponding datasheets and Figure 25. From the datasheets, f is the uncoupled resonant frequency under maximum static loading and maximum peak to peak vibration input (2a).

Example: A 1-DOF system (W = 520 lbf), having its CG at its geometric center, is mounted in compression on 4 identical WRIs. This system is under a constant peak to peak displacement input: $2u_0 = 0.10$ in.

- FH56-2008: *Per the corresponding datasheet in compression (p.53): 2a = 0.14 in and f = 6.6 Hz*
  $2u_0 / 2a = 0.10 / 0.14 = 0.71$. *From Figure 25, this results in:* $f_{res} / f = 1.11$. *Therefore:*
  Corrected resonant frequency: $f_{res} = 1.11 \times f = 1.11 \times 6.6 =$ ***7.3 Hz***
  Maximum displacement in the isolators at resonance, assuming a Q factor of three: $x_{max} = Q \times u_0 = 3 \times 0.05 = 0.15$ *in*
  Maximum acceleration of the sprung mass: $a_{max} = (2\pi f_{res})^2 \times x_{max} = (2\pi \times 7.3)^2 \times 0.15 =$ ***316 in/s² (0.82g)***

- FH56-4808: *Per the corresponding datasheet in compression (p.53): 2a = 0.44 in and f = 3.6 Hz*
  $2u_0 / 2a = 0.10 / 0.44 = 0.23$. *From Figure 25, this results in:* $f_{res} / f = 1.53$. *Therefore:*
  Corrected resonant frequency: $f_{res} = 1.53 \times f = 1.53 \times 3.6 =$ ***5.5 Hz***
  Maximum displacement in the isolators at resonance, assuming a Q factor of three: $x_{max} = Q \times u_0 = 3 \times 0.05 = 0.15$ *in*
  Maximum acceleration of the sprung mass: $a_{max} = (2\pi f_{res})^2 \times x_{max} = (2\pi \times 5.5)^2 \times 0.15 =$ ***179 in/s² (0.46g)***

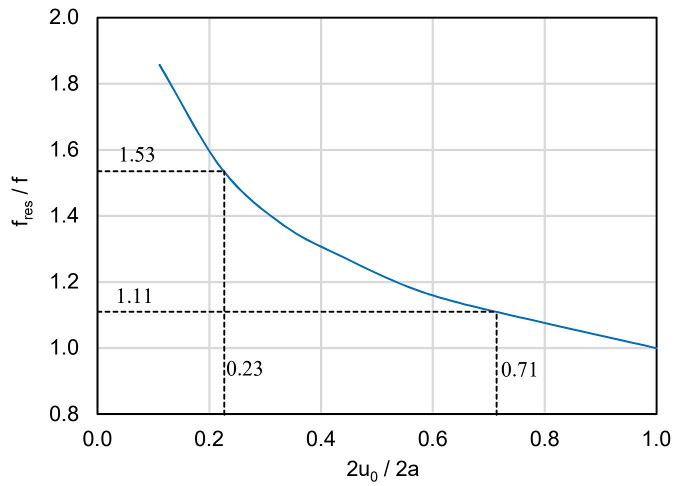


Figure 25. Curve used to determine the corrected resonant frequency (f_{res}) of a system under a constant peak to peak displacement input ($2u_0$)

If displacement input is unknown but only acceleration, velocity or PSD is known, then the equivalent displacement input is calculated assuming a resonant frequency. This process will require an iterative analysis best achieved with the proper simulation software.

Using the linearization process above yields an approximation useful for preselection. For critical and multiple degrees of freedom applications a more correct and realistic approach is to carry out a nonlinear calculation in terms of resonant frequency and transmissibility. This requires nonlinear simulation software such as Socitec Group's SYMOS.

Note: For high cycle applications a fatigue review may be required. Contact Vibro/Dynamics for assistance.

5. Select for both shock and vibration

All selected isolators should meet criteria explained in steps 3 and 4, keeping in mind that shock is usually the sizing parameter.

The above selection process is still valid for Socitec Group's elastomeric isolators, but is simplified because elastomers are less nonlinear and their average stiffness is supplied in the datasheets.

There is usually more than one isolator that will meet all requirements; however, it is not advised to select the smaller one.

Finally, for both WRIs and elastomeric isolators, the maximum shock and vibration values given in the datasheets must never be exceeded unless contacting Vibro/Dynamics.

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