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Wire Rope vs. Elastomeric Isolators in Naval Applications

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Shock and Vibration have always been a concern in commercial and naval vessels, whether for the comfort of passengers or simply as a matter of survivability for the ship. While shipbuilding has benefited from many technological improvements over the years, the level of vibration and shock inputs to which onboard equipment aboard are subjected have not substantially decreased, due to the ever increasing capability of vessels. In addition, the widespread use of non-rugged commercial off the shelf (COTS) equipment has made shock and vibration isolation even more necessary.

A ship is exposed to many different inputs, such as shock from underwater explosions (UNDEX) or the vibration induced by its' propellers, main and auxiliary engines. An isolation system that supports equipment on the ship is required to protect against both types of input.

A shock isolator requires a certain amount of dynamic displacement capability in order to cope with the induced displacement step subsequent to the above mentioned type of shock, while a vibration isolator should exhibit a low dynamic stiffness in order to achieve the required amount of isolation.

Selection of a shock and vibration isolator typically requires a tradeoff between both, which makes it critical to completely understand all isolator characteristics during the selection process.

This paper will discuss two well-known technologies used on naval ships, wire rope and elastomer shock and vibration isolators. The benefits and drawbacks of each type will be detailed through an application case involving the medium weight shock testing machine of MIL-S-901D and its modeling with Socitec Group non-linear numerical software SYMOS.

Figure 1-Several Types of Wire Rope Isolators

Typical Characteristics of Wire Rope Isolators

- Heavy duty all metal construction
- Can be mounted in any attitude (e.g. compression, tension, or shear)
- Excellent viscous equivalent damping ratios approaching 20%
- Large deflection capability
- Strong non-linearity
- Modularity to allow for adaptation to any application
- Superior temperature range and resistant to environment
- Long product lifetime
- No creeping or aging
- Highly repeatable characteristics allow for reliable modeling

Figure 2-Several Types of Elastomeric Isolators

Typical Characteristics of Elastomeric Isolators

- Can be shaped to reach any desired characteristics
- Compression loading required
- Low stiffening coefficient
- Viscous equivalent damping ratios dependent on material
- Limited application temperature range
- Characteristics are sensitive to temperature
- Limited product lifetime
- Sensitive to chemical agents depending on material
- Manufacturing tolerances and temperature dependence make modeling less reliable

Vibration in Naval Applications

A system experiences forced vibration when it is subjected to steady-state excitations. In naval applications, a typical source of this would be the shipboard propellers and engines.

The system consisting of equipment supported by isolators can be represented by the following model:

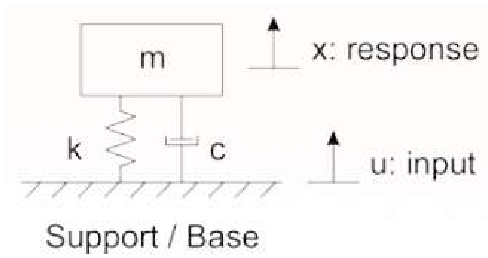

Figure 3-Mass-Spring-Damper Model

In the above system:

- k = stiffness of isolators
- c = damping of isolators
- m = sprung mass
- u = harmonic base excitation

For a system experiencing harmonic base excitation, correct implementation of a vibration isolator to the system will attain a condition wherein the acceleration transmitted to the sprung mass, m, is less than the input acceleration of the base. The performance of an isolator is best evaluated through the transmissibility (T), which is the ratio of the maximum acceleration of the sprung mass $(\ddot{x}_0)$ to the maximum acceleration of the base $(\ddot{u}_0)$:

$$T = \frac{\ddot{x}_0}{\ddot{u}_0} = \sqrt{\frac{1 + \left(2\xi \frac{\omega}{\omega_n}\right)^2}{\left(1 - \frac{\omega^2}{\omega_n^2}\right)^2 + \left(2\xi \frac{\omega}{\omega_n}\right)^2}} = \frac{F_T}{F_0}$$

$\omega_n = 2\pi f_n$: undamped natural frequency (rad/s), F_0: magnitude of the harmonic force input applied to the sprung mass, F_T: magnitude of the force transmitted to the base

The transmissibility equation is also expressed graphically in the following figure:

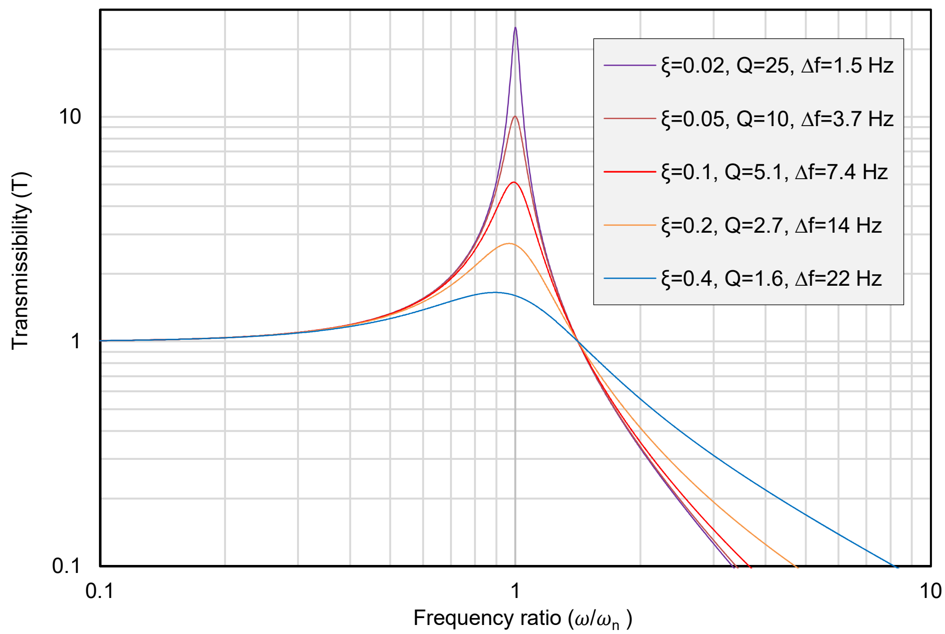

Figure 4- Transmissibility Curves for Mass-Spring-Damper Model Given Varioud Damping Ratios

The motion of the base and of the sprung mass may be expressed in terms of displacement, velocity, or acceleration, and the same transmissibility equation holds in each.

The Q factor is commonly used in vibration theory. It is defined as the transmissibility at resonance, and is inversely proportional to the amount of damping in the system. Mathematically, the Q factor is defined as:

$$Q = \frac{f_r}{\Delta f} = \sqrt{\frac{1 + 4\xi^2}{4\xi^2}}$$

$$Q \approx \frac{1}{2\xi}$$ (for small damping ratio)

f_r: resonant frequency (Hz), Δf: frequency bandwidth (Hz).

The frequency bandwidth is the range of frequencies over which the transmissibility value is $(Q/\sqrt{2})$ and above. Therefore, for a given system with a fixed resonant frequency, the higher the Q factor, the narrower the frequency bandwidth, as seen in Figure 4.

Given the provided equations and figures, it is clear that an optimal isolator for vibration isolation is one that possesses a low damping ratio (High Q factor), as increased levels of damping actually decrease isolation performance in the attenuation region of the transmissibility curve. However, if forced vibration is present in the resonance region, then it would be advantageous to utilize an isolator with more damping in order to minimize the transmissibility at resonance (Low Q factor).

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Shock in Naval Applications

The most typical source of shock in a naval application is an underwater explosion. This sudden, transient event imposes a displacement shock on the system, which then undergoes free vibration upon removal of the input. Free vibration is dependent on the stiffness and damping characteristics of the isolators supporting the equipment. The motion of a system after a shock input can be typically represented by the following equation.

$$x(t) = x_0 * e^{-\xi\omega_n t}\sin(\omega_d t - \varphi)$$

Where x_0 (amplitude) and φ (phase shift) both depend on the initial conditions given to the system.

From this equation it can be seen that the system will vibrate at its damped natural frequency, as denoted by the following equation:

$$(\omega_d = \omega_n\sqrt{1-\xi^2})$$

An example of the typical motion after a shock input is shown in the following figure:

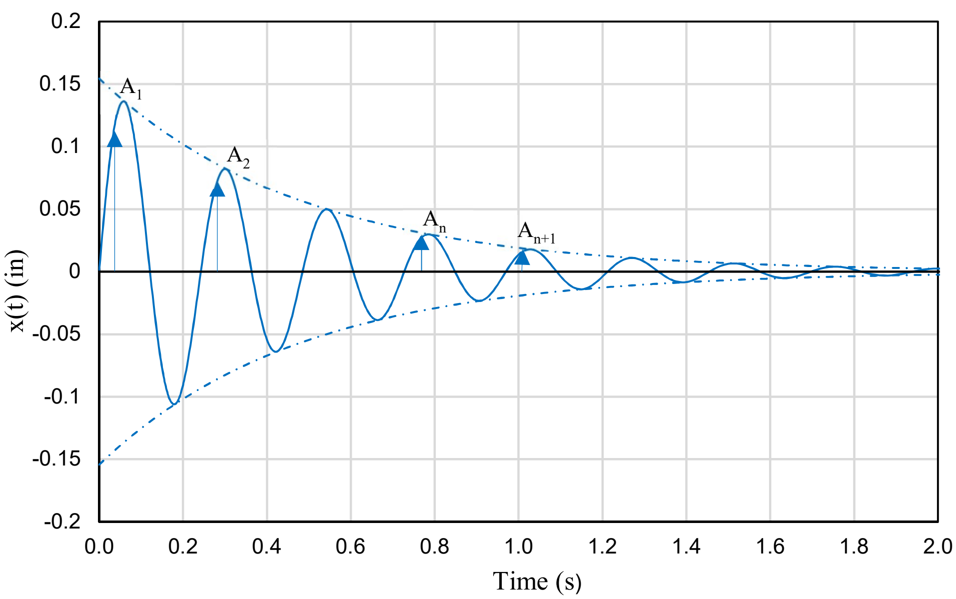

Figure 5-Post-shock free vibration of a typical underdamped linear-mass-spring-damper system

A useful quantity in free vibration is the logarithmic decrement. The logarithmic decrement is the rate at which motion of a system decays. Mathematically, it is defined as:

$$\delta = \frac{2\pi\xi}{\sqrt{1-\xi^2}} = \ln\left(\frac{A_n}{A_{n+1}}\right)$$

As can be seen in the equations and figure, the higher the damping in a system when vibrating freely, the faster the vibration decays to zero. For example, WRIs with about 17% damping ratio will decay in less than three oscillations.

Chloroprene rubber, with a damping ratio of around 10%, will decay in less than six oscillations. Natural rubber possesses an even lower damping ratio of about 4%, requiring over ten cycles before decaying to zero.

Case Study

Previous discussions treat isolation applications with a simple linear model and inputs. However, in some naval applications the application is not as straightforward, and the performance of an isolation solution is tested with complicated shock testing apparatuses. Each piece of 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 is often forced to oversize isolator selection to ensure passing the test the first time. This methodology is less than ideal, which makes an approach with a numerical software package such as SYMOS attractive. The SYMOS model divides the machine into a number of rigid bodies connected by nonlinear springs and dampers to which the shock parameters are applied as initial conditions. The model even includes 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 was mounted vertically on WRI and elastomeric isolators, and was tested with the MWSM under the following test conditions:

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

Additionally, sensors were placed in the cabinet to measure the response acceleration. The measured response from one of the sensors in the cabinet during the physical test was compared to the response calculated by the SYMOS model of the MWSM, as shown in the comparative time histories in Figures 9 and 10.

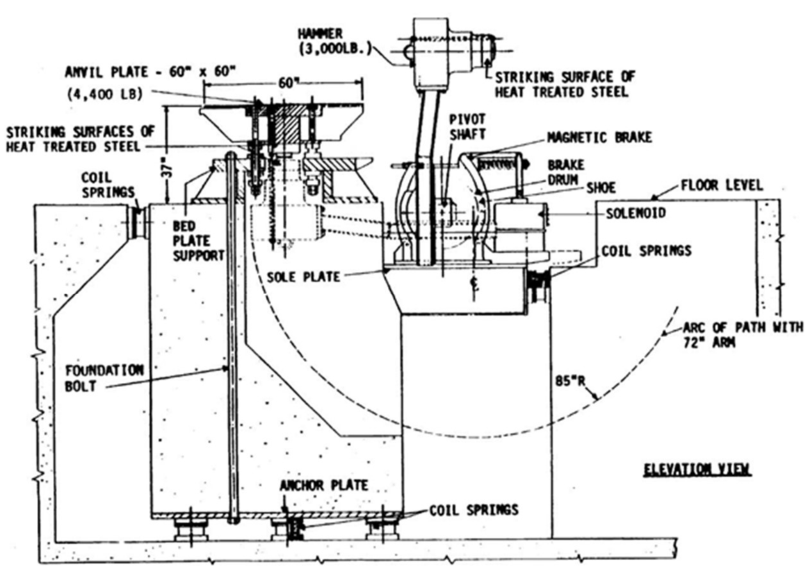

Figure 6-High Impact Shock Testing Machine for Medium Weight Equipment per MIL-S-901D (MWSM)

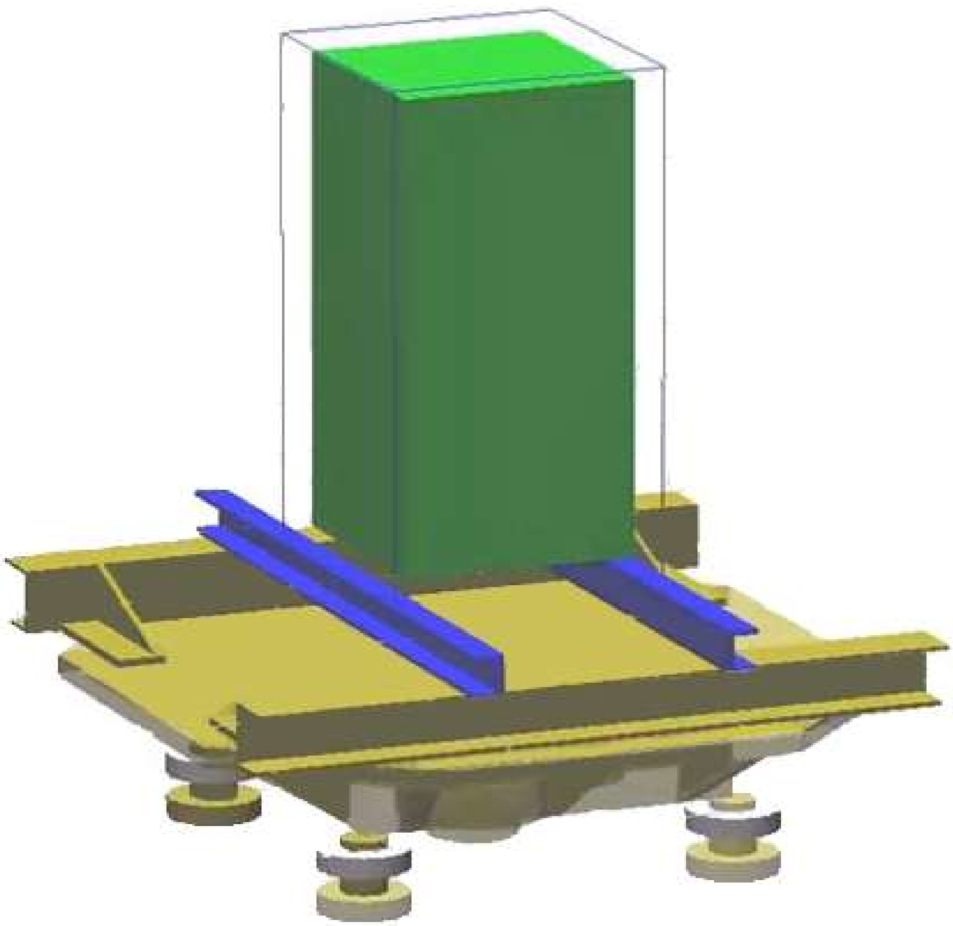

Figure 7-Image of SYMOS Model Constructed to Simulate MWSM

Figure 8-Test Setup on Wire Rope Isolators (Left) and Elastomeric Isolators (Right)

Figure 9-Inside of Cabinet Showing Sensor Measuring Acceleration

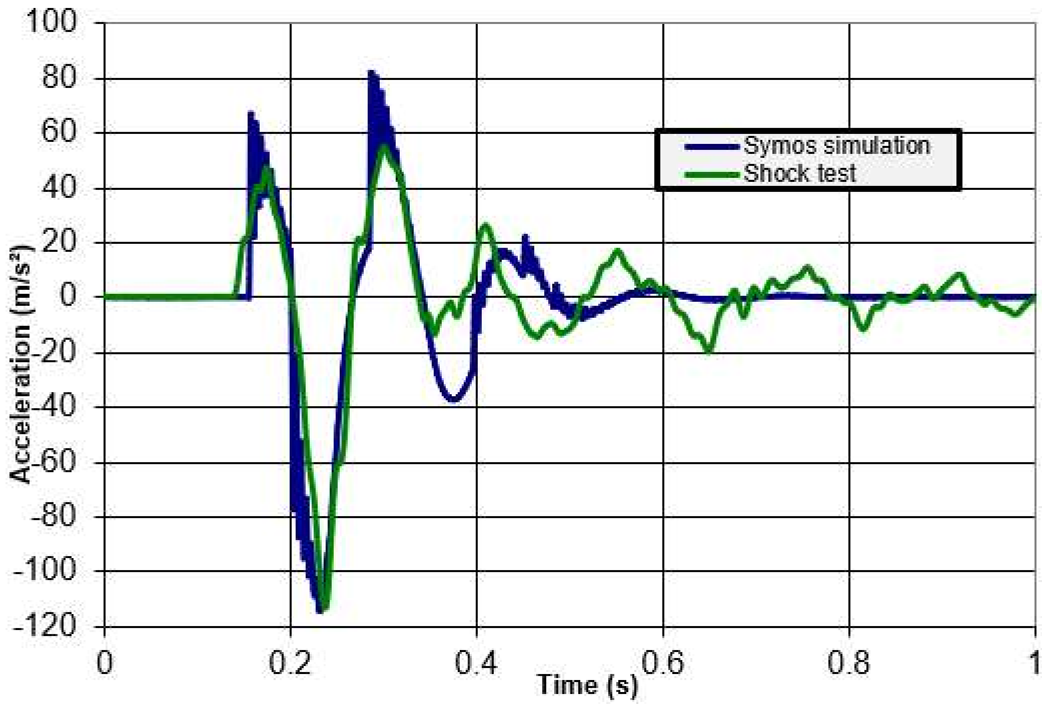

Figure 10- Comparative Time Histories of Measured vs. Calculated Response with WRI

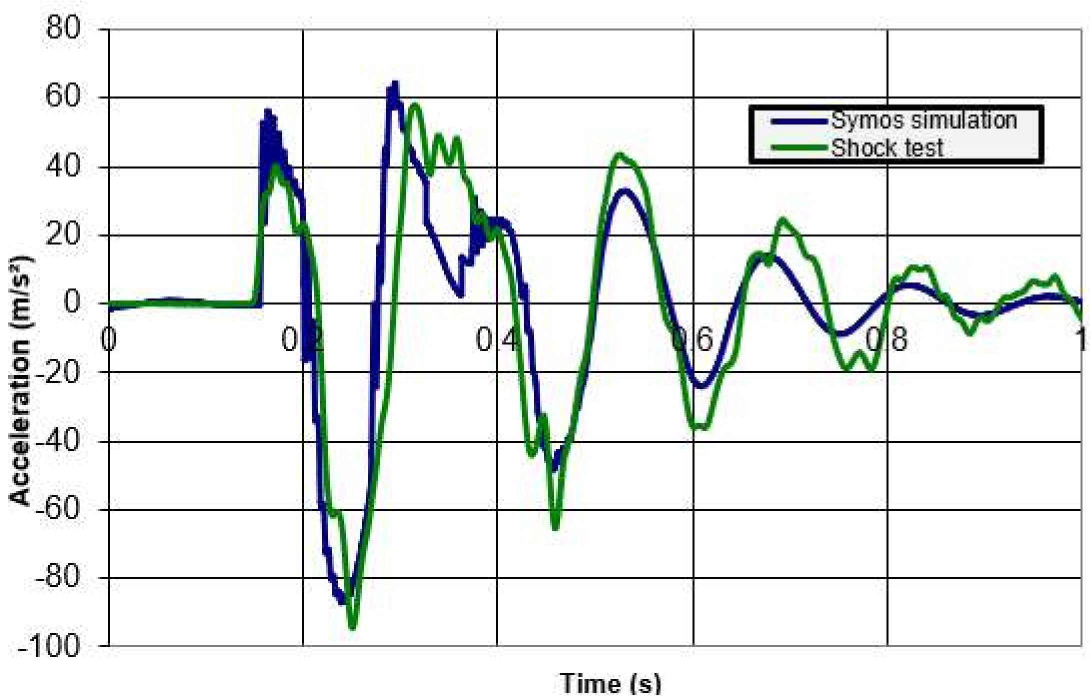

Figure 11-Comparative Time Histories of Measured vs. Calculated Response with Elastomeric Isolators

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The figures demonstrate the accuracy of the SYMOS model, as the predicted peak response in both cases is very close to the measured response during the test. The difference in damping properties of the two solutions is also evident in the charts as the WRI solution decays to levels near zero after only 0.5s while the elastomeric solution time to zero approaches 1s.

The implementation of isolators in naval applications requires the consideration of both shock and vibration inputs. Proper selection of isolators warrants that the engineer selecting isolators understand the equipment isolation requirements, as well as the properties of the various isolation system options. A solution that is optimal for shock isolation may not offer sufficient vibration isolation and a compromise must be made.

Wire Rope Isolators are versatile, long-lasting all metal isolators that can be used in virtually any application. Their excellent damping properties make them ideal for shock attenuation, but less effective for vibration isolation.

Elastomeric Isolators can be designed to fit any characteristic needed, but have a much narrower application range and shorter lifetime. The low stiffening coefficient allows for excellent vibration isolation, but decreases efficiency for attenuating shock.

The final selection will depend on the particular application and whether shock isolation or vibration isolation is more critical for the equipment.