Low Frequency Shock and Vibration Isolation for Precision Engineering and Nanotechnology

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Abstract:

Progress in precision engineering and nanotechnology demands greater shock and vibration attenuation of cultural and induced base mass or ground vibratory inputs. This document is a condensed version of existing approaches and transcends to the topic of state-of-the-art techniques by using modified current technology and novel solutions. The 0.4 Hz to 2.0 Hz frequency range will be examined since nanotechnology progress demands quieter environments to allow futuristic engineering to continue. Discussions also include soil and foundation vibration isolation trade offs for earthquakes with additional mention of shock and vibration isolation in a vacuum. Recent applications are reviewed with emphasis on finding the proper solution prior to any implementation since "after the fact" solutions could be compromising and are much costlier with no time line.

Low Frequency Vibration Isolation

Protecting your equipment from undesirable environmental effects such as vibration, shock and acoustic inputs, the degree and magnitude must be completely defined. To aid in this process a thorough analysis must be accomplished to define the problem. Techniques such as vibration survey/studies and performing a Finite Element Analysis (FEM) will greatly aid in the determination process. Once the vibration problem is fully understood, the establishment of the vibration control performance requirement can then be defined. Some equipment considerations such as system facility, structural integrity, fatigue and tolerances with external environmental effects are extremely critical.

Pneumatics

Pneumatics are chosen for the type of isolator to be used. Only low frequency vibrations are considered since disrupting frequencies greater than the 10 Hz to 15 Hz range may be solved by more conventional methods such as metal springs, elastomers in compression/ shear or fluidics. Achieving low natural frequency isolation, pneumatic non-deflecting elements or in combination with other elements will be reviewed using proven solutions. The classic single degree of freedom damped system with 6 db/octave roll off is shown in Figure 1. This can be expanded to a sprung damped system with a roll off rate of 12 db/octave shown in Figure 2, giving rise to an order magnitude greater attenuation at the same natural frequency by using the "series" technique.

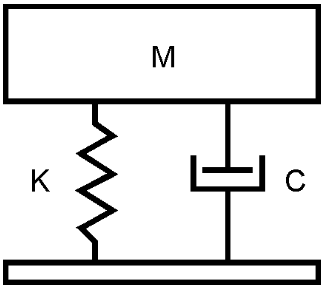

Figure 1

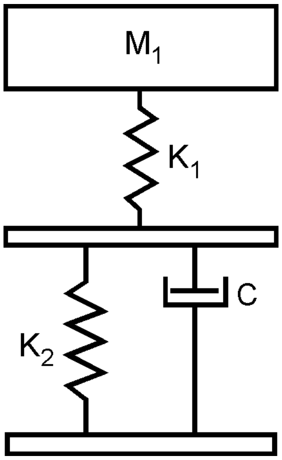

Figure 2

Vertical Isolation

This type of pneumatic isolator (Figure 3) uses dual chambers, primarily the upper chamber as the spring chamber V_S (K_1) and the lower chamber V_D acting as the K_2 with Damping C. The damping chamber is separate from the spring chamber and connected by pneumatic tubing. External damping adjustments may also be employed for field adjustment. This non passive system controls the spring chamber piston by using a servo controlled valve with gain adjustments, allowing the pressurized gas to flow into or out of the isolator to maintain the "no spring deflection" feature. This type of system may take on different shapes since only the isolator volume is critical as shown below in equation Ref (1).

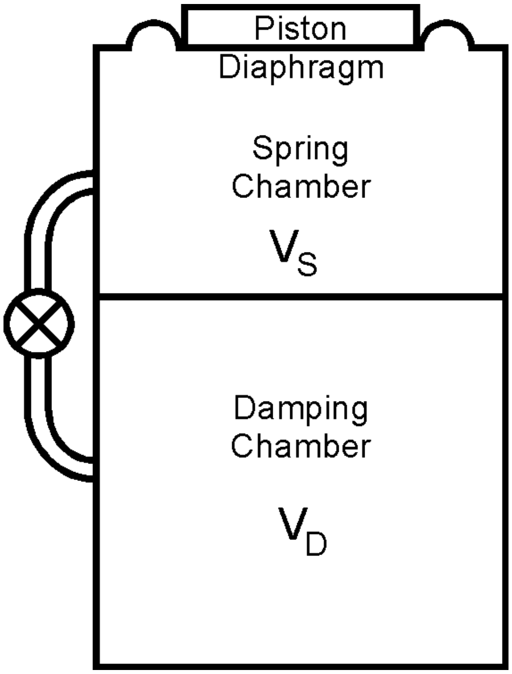

Figure 3

$$f_n = \frac{1}{2\pi}\left(\frac{\eta P A^2 g}{V_{eff}\, W}\right)^{\frac{1}{2}} \qquad \text{Ref (1)}$$

f_n = natural frequency

g = gravity

n = rate of specific heat of gas at constant pressure and volume

W = weight of supported load

A = effective area of diaphragm

V_{eff} = effective volume ratio

P = psig

This method and variations of same have successfully been used to support nanocenter technology throughout the world. Isolator natural frequencies have been as low as the 0.4 Hz to 0.5 Hz range. Isolators of this type can now support up to 70 metric tons per isolator, thus offering the most economical solution of all for the large/heavy requirements by using less isolators.

To make effective use of this isolator feature, a rigid payload base must be used. If a soft structure/payload is to be isolated, it must be mounted on a rigid mass/base to eliminate the "wet noodle" effect of some payloads since the wet noodle will result in an indeterminate faulty solution. This design criteria is best left to the experts in vibration isolation of same.

One of the primary design goals of any payload/base design is to eliminate coupling effects that induce turning moments or rotational inputs due to geometrical conditions leading to a total composite center of gravity of the payload(s) and rigid mass/base not being at the elastic plane of the isolator. Linear reactions are more controllable subsequently offering a manageable solution when the total composite center of gravity (TCG) is at the elastic plane of the isolator.

When using an elastomeric diaphragm, certain hysteresis effects become apparent at very low operation pressures such as 2 bar. As the pressure reduces, these effects eliminate the effectiveness of the pneumatic contribution of the isolator, therefore much higher pressures are used. Note in the equation (Ref 1) the pressure is proportional to the load (W) thus maintaining a constant natural frequency when the load changes while employing a servo valves to maintain level. Elastomeric diaphragms should be less than 2 mm thick since thicker diaphragms, bladders or air bags transmit vibrations with low G inputs.

Horizontal Isolation

When considering horizontal motions the diaphragm closely follows the empirical equation described by:

$$F = 1.25\, dP\delta^{0.77}$$

F = Horizontal force (pounds)
d = effective diaphragm area (inches)
P = absolute isolator pressure (psi)
δ = piston deflection (inches)
1.25 = developed constant

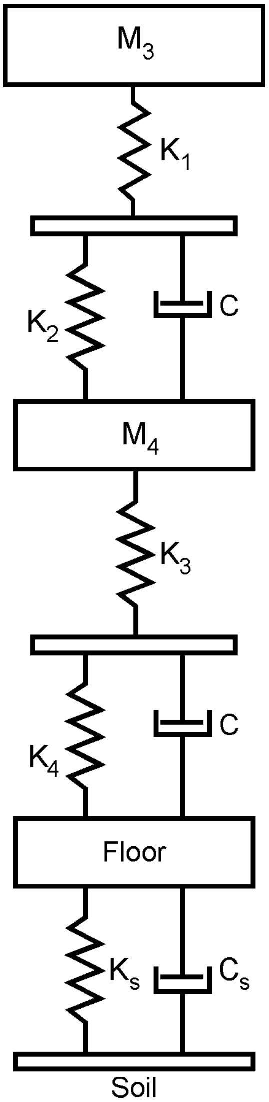

Figure 4

A lower horizontal natural frequency can be developed. When resulting natural frequencies are approximately 2 Hz specially designed pendulums are adapted to the isolator to achieve $f_n < 0.4$ as a function of space. The use of pendulums may appear to be a simple approach to lower the isolator natural frequency. However, one must consider type(s) which may be selected versus the negatives of the specific choice. Bifilar, multi-coupling, rod and cables may be some of the obvious choices. Since the equation for a pendulum is nonlinear, the use of a pendulum should be limited to a displacement of a maximum of 8E. Additionally, any restoring force, if any, other than gravity must also be considered to achieve a complete solution. Specially designed elastomeric diaphragms may also reduce the horizontal natural frequency to approach 1 Hz.

Two Degrees of Freedom

Extend design philosophy into Figure 4 and design by using the series method. This provides a 24 db/octave roll off but with a lower overall fundamental frequency. The primary natural frequency (f_{n1}) and a secondary natural frequency (f_{n2}) where $f_{n2} > f_{n1}$ for M_3 or $f_{n3} >> f_{n1}$ for M_4 is preferred, depending on the spring rate ratio K_1 / K_3, K_2 / K_4 and the mass ratio M_3 / M_4 where $M_4 > M_3$.

Using this approach isolation at frequencies beyond f_2 offer a superior roll off rate required for extremely vibration sensitive equipment such as calibration of laser gyros. This type of use of pneumatic isolators offers four (4) times more isolation than a standard isolator shown in Figure 1.

The sensitivity of any isolator is offset by the level of force being isolated. An example of a standard rubber in compression mount with a low “g” input will have a “Q” (amplification at resonance) lower than having a high “G” input. Additionally, with a low “Q” the roll off becomes less than 6 db/octave while at a high “G” input the roll off is approximately 6 db/octave with a high “Q” factor. Pneumatics does not present such a severe picture because the isolation contribution is largely pneumatic and the thin elastomeric membrane has little contribution at pressures of 4 bar and larger.

Earthquakes

Shock isolation of precision equipment such as CMM's, diamond turning machines and similar precision processes are primarily isolated from the floor/foundation to eliminate damage to the machine during an event. As an example, on Monday, January 18, 1994 an earthquake with a magnitude of 6.7 struck Southern California. About 25 miles from the epicenter at Hughes Aircraft (El Segundo, CA), two large upward motions followed by continuous horizontal oscillating motions were felt. This powerful series of large ground motions were the largest in Los Angeles modern history.

A special dynamically tuned large concrete seismic bench (385,000 pounds) supported an optical payload. The key ingredient to the survivability of this special program is the seismic shock and vibration isolation system which supports the seismic bench. Consisting of ten (10) pneumatic isolators which are designed specifically for zone 4 earthquake areas (zone 4 being the most damaging on a scale of 1 to 4) the system significantly decoupled the earthquake ground motion so that when the event ended, no damage, payload/seismic bench displacement, or drift effect occurred. This active system reinitialized itself and is currently in the same position as pre-earthquake alignment even though the supply of air pressure was interrupted during the event.

Vibration Isolation in a Vacuum Chamber

Vibration isolation can be internal or external to a vacuum chamber. Optimum isolation occurs when the payload is isolated internally to the chamber when space permits. When large isolators are required, the use of sealed flexible penetrations attached to the surface of a large isolated seismic mass under the chamber would be an alternate solution. Both are proven techniques and are operational around the world. Soft flexible special elastomeric vacuum seals are used at each penetration.

References

[1] Dankowski, Jack.: "Trade Off Study and Math Model of Pneumatic Isolators," Measurement Analysis Corporation, 1980. (Rev. 1)