Performance Loss In Stacked Pneumatic Vibration Isolation Systems

Wes Wigglesworth, BSEE

Product Manager, TMC-Ametek Ultra Precision Technologies

October, 2013

Abstract

Many instruments have built-in pneumatic vibration isolation systems. This note demonstrates that placing these systems on a secondary platform which is supported by a second set of pneumatic isolators both *degrades* isolation performance in the critical 0.5-5Hz frequency band and leads to systems with poor tilt stability. Improved isolation performance and much greater stability can be achieved by supporting such a payload with a 'hard-mount' active isolator such as TMCs *STACIS*^{tm} system.

1 Introduction

Passive pneumatic vibration isolation systems provide a highly cost effective means for protecting sensitive instruments from floor vibrations. Their automatic leveling systems adjust the pressure in the isolators to 'float' the payload to an accurately controlled height and also adjust the stiffness of the isolators to provide a consistent isolation performance defined by the isolator's resonant frequency, given by $\sqrt{k/m}$. Here 'k' is the isolator's spring constant and 'm' is the supported mass. Isolation starts at ∼1.2-1.5 times this frequency, depending on damping. Below this the isolators actually *amplify* ground motion. Passive damping can reduce this amplification but at the cost of high-frequency isolation. A typical compromise between these two considerations is to limit the damping to a *Quality factor* of about 10.

To increase attenuation of floor vibration the instrument can be placed on a secondary isolation platform, often as part of a cleanroom subfloor environment. This is referred to as a stacked isolation system. If that secondary system is based on pneumatic isolators, several significant problems may be encountered. First, the vibration isolation performance in the 0.5-5Hz band is actually degraded. Second, the system becomes much more sensitive to tilt instability.

2 Isolation degradation

The amount of degradation in isolation performance can be evaluated using a computer model of a stacked isolation system. Such a model is shown in Figure 1 below. The RMS motion (or acceleration) of the isolated payload is dominated by the resonant peaks of the isolators.[1]

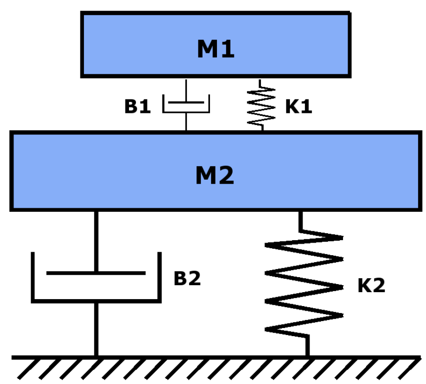

Figure 1: A simple model of a stacked pneumatic system

In stacked systems, the mass ratio (M2/M1) varies from about 1 to 30 and determines the level of degradation. The lower isolators (K2 & B2) are assumed to be medium-load isolators with a 1.8Hz resonance. The top isolators (K1 & B1) are assumed to be low-load isolators with a 2.0Hz resonance. Using this model the vibration transfer function of the stacked isolation system can be compared with a single isolator. That calculation is shown in Figure 2 below. The black curve is the single-stage isolator while the blue curves show the isolation performance for stacked systems with the mass ratios (MR) of 1, 2, 5, 10 and 30. It can be seen that the stacked systems actually have two 'normal mode' frequencies that are

[1] Ground vibration varies dramatically from site to site. The peaks dominate the motion if the acceleration spectrum has a slope of $(freq)^2$ or less. Our calculations assume a flat acceleration noise spectrum.

different from the isolators' frequencies.

One reason the performance degrades is that the effective bounce frequency of M2 is higher than the 1.8Hz nominal frequency for the lower isolators. This is because their air pressure (and therefore stiffness) is appropriate to lift the combined weight of M1 and M2 (the static load) while the resonance is dominated by the *dynamic* mass M2 only.

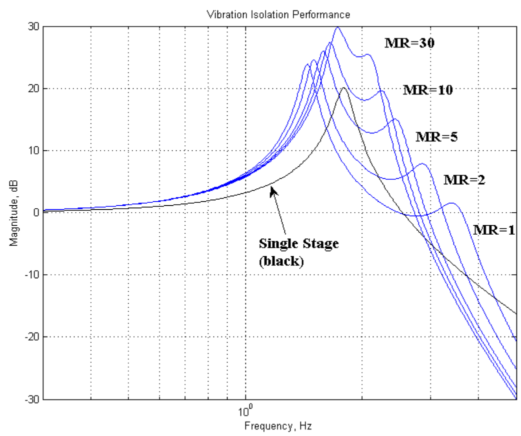

Figure 2: Predicted isolation vs mass ratio

One can use these curves to predict the degradation in the isolation performance. In Table 1 below, the RMS motion of the table is calculated relative to a single-stage isolator and the percentage increase in RMS noise is given. The RMS integration was performed between 0.3 and 5Hz.

Table 1: % Increase in RMS Noise vs Mass Ratio	
Mass Ratio	% Degradation
1	24
2	43
5	83
10	130
30(+)	225

3 Tilt Stability

Unlike vibration isolation performance, tilt stability of platforms depends on far too many parameters for modeling to be informative. Instead we introduce the concept of tilt stability and present a simple model to illustrate the problem.

Figure 3 shows a diagram of a platform resting on two isolators (represented as springs). There is a parabola above the table of height (H) and base width (W). If the payload's center-of-mass is inside this curve, the system will be tilt stable. (H) is proportional to the frequency of the isolators and the square of their separation (W). Because the effective support point of many pneumatic isolators is 4″ to 9″ below its support pad, (H) can be surprisingly low. This is particularly true for semiconductor tools that drive footprints to minimal dimensions.

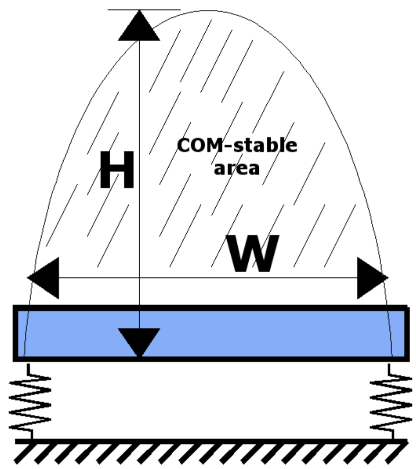

Figure 3: The region of COM stability

Figure 4 shows a toy model of what's happening. Payloads are represented by a circle at their center-of-mass (CM) with soft springs supporting them by an inverted "T" (A). Like a pencil standing on its point, a payload supported by soft springs wants to fall over due to the force of gravity (B). If a system is unstable, like the pencil, there is no perfect balance where it will stand in place - it will always fall. As the isolators get spread apart, however, their restoring force overcomes this and the system becomes stable.

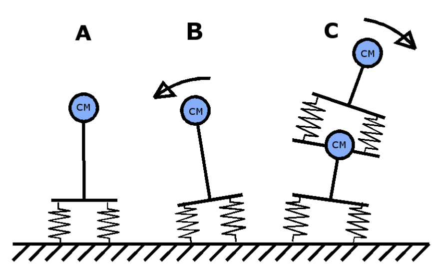

Figure 4: Toy model of tilt stability

The situation with stacked isolators is shown in Figure 4(C). It is obvious that (C) will have much greater tilt stability issues than (A). There are only three basic ways to overcome this instability. These are to increase the footprint of the platform, lower the center of gravity of the payloads and increase the stiffness of the isolators. The first two can be very difficult to implement without affecting the economy of the tool. The third option heavily compromises isolation performance in passive isolators. TMC's *STACIS*^{tm} active isolation system uses stiff passive isolators but compensates for the isolation loss with a high-performance active feedback system, as discussed below.

4 The Active Solution

TMC offers an alternate solution to address excessive vibration in the 0.5-5Hz band while immediately solving the stability problem. The *STACIS*^{tm} line of active piezo-electric vibration isolators can be placed under any passively isolated platform to increase isolation performance without introducing tilt instability. It is a "hard mount" vibration isolation system utilizing highly sensitive vibration sensors and piezoelectric actuators to provide vibration cancellation starting at 0.6Hz in all six degrees of freedom. "Hard mounting" means there are no soft air springs supporting the weight of the payload. This makes it inherently compatible with tools that incorporate internal pneumatic isolation systems. Outside of its active bandwidth, *STACIS*^{tm} employs specially developed 18Hz rubber isolators that do not affect the tilt stability of the payload's passive system.

STACIS^{tm} performance in the 0.5-5Hz band is so effective that it *effectively erases* the amplification caused by the passive system. It reduces horizontal and vertical vibration by over 50% at 1Hz and over 90% at frequencies above 2 Hz.

Developed specifically for high resolution metrology tools the system has proven itself time and time again in semiconductor fabrication facilities around the world. It allows tools to be installed in facilities which otherwise do not meet the tool's vibration requirements. *STACIS*^{tm} is thus a point-of-use solution that allows for greater flexibility and cost savings in the design and planning of fabs and lab facilities.

5 Conclusions

We have demonstrated that stacked pneumatic isolation systems dramatically increase the RMS motions of supported payloads, in some cases by over a factor of 3. Though harder to quantify, tilt stability is also a significant issue. To address the problem of excessive noise in the 0.5-5Hz band, the *STACIS*^{tm} active hard-mount systems offer an attractive alternative.

TMC-Ametek
15 Centennial Drive, Peabody, MA 01960
(978) 532-6330