Mechanical Design for Microphonic-Sensitive Electronics

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Abstract
Modern RF/microwave electronic systems and subsystems often rely on precision frequency sources that contain microphonic sensitive components, like DROs, crystal oscillators, etc. Designing enclosures and other mechanical structures for such systems and subsystems presents substantial challenges, especially when aimed at mobile applications. The requirement to reduce size, weight, and power (SWaP) makes this task even more challenging for mechanical engineers working in the field of electronic packaging design. This article describes considerations for finding the optimal balance for often conflicting requirements like minimizing size while providing adequate sway space for vibe isolated modules/components, maintaining high rigidity of the structure while minimizing weight, and other details. We will touch on the constraints imposed by SWaP reduction requirements and on vibe isolation system designs as well. However, a more detailed review would require a separate article.

Introduction
Microphonic-sensitive devices and/or components are often used in modern electronic systems and subsystems. When such a system or subsystem is intended for mobile applications, such as missiles, aircraft, or shipboard uses, the microphonic-sensitive device needs to be protected against shocks and vibrations to reduce degradation of electrical performance such as phase noise, spurious, etc. This can be achieved with passive or active vibe isolation systems. Active vibe isolation systems generally require significantly more space, are heavier than passive vibe isolation systems, and require power, which is usually at a premium in mobile systems and subsystems. Therefore, we will limit our discussion to applications where only passive, elastomer-based vibe isolation systems are utilized. However, most recommendations presented here will improve the performance of the systems/subsystems that employ active vibe isolation.

Background
All the mechanical structures of an electronic system/subsystem can be viewed as a mechanical oscillator, as the unsupported sections of it, between attachment points to the next higher level assembly, will deflect under load. If such an external load is cyclical in nature, like the vibration of an airframe or a ship hull, the structure will exhibit properties of an oscillator with its own natural frequency driven by rigidity of the structure and its mass. The more rigid the structure, the higher the natural frequency. The higher the mass, the lower the natural frequency. This relationship between the aforementioned properties of the structure, pertinent to a simple harmonic motion type of response, is best described using the following equations:

$$\omega = \sqrt{\frac{k}{m}} \text{ (rad/sec)}$$

Where $\omega$ is the radial frequency, $k$ is the spring constant, and $m$ is the mass of our system/subsystem. It is expressed in radians per second.

To convert this into cycles per second, or Hertz (Hz), we need to convert radians into full cycles:

$$f = \frac{1}{2\pi} \sqrt{\frac{k}{m}} \text{ (Hz)}$$

Structural Design
Structures intended for mounting vibe isolated, microphonic-sensitive devices usually serve as a chassis for the subsystem or system as a whole. To get the most performance out of a passive vibe isolation system, it is beneficial to design a chassis as rigid as possible. This will move the resonant frequency of the structure as far away from the resonant frequency of a vibe isolated dielectric resonator oscillator (DRO) as possible. In this case, when determining the chassis resonant frequency, one must include the masses and/or point-loads of all the modules mounted to it. At this stage, using finite element analysis (FEA) software for determining the natural frequency appears to be more practical.

Ideally, the natural frequency ($f_0$) of the whole structure should be higher than the operating (electrical) frequency of the suspended microphonic-sensitive device (for example, DRO). However, with modern devices operating in the GHz range, it's hardly possible. Still, it's advisable to push for the highest possible $f$, of the structure to minimize linear displacements of such structures under shock and vibration.

In the case of new designs, using multiple mounting points along each side of the structure helps keep $f$, reasonably high. However, it might be more difficult to achieve higher $f$, when one is working on a drop-in replacement design, where mounting points are often located far from each other, given the general tendency that each next-generation system/subsystem is smaller and lighter than the previous one. That requires spanning considerable distances, which corresponds with the lower natural frequencies of the structure, in general.
Using the Full Extent of the Envelope to Improve the Rigidity of the Structure

Quite often, the mechanical engineer working on packaging design for mobile applications is pressured to keep the lowest possible cover height or housing depth based on the height of the tallest electrical components in the volume to be encapsulated/covered. While it’s reasonable from the standpoint of minimizing the weight (and, often, cost of parts), it would be a bad trade-off for designs where a passive vibe isolation system is employed. In these cases, the cover height (or housing depth) is one of the most powerful contributors to the rigidity of the structure. Steiner’s theorem of parallel axis describes this relationship very well.

\[ I = I_{cm} + md^2 \]

Where \( I \) is a moment of inertia with respect to a given axis; \( I_{cm} \) is a moment of inertia with respect to the axis drawn through the center of gravity; \( m \) is mass; and \( d \) is the distance between the two aforementioned axes (axes are parallel to each other).

As deflection of the structure is reverse-proportional to the moment of inertia, increasing the distance between structural members is a very effective way to improve response of the structure exposed to shock/vibe environments.

Material Sets

While a mechanical engineer has very little leverage for choosing materials for printed circuit boards, microwave substrates, surface-mount components, or related items, where the major driving force is electrical performance regardless of structural properties (essentially, turning such parts of the design into dead weight from the structural standpoint), the materials used for enclosures and chassis can and need to be selected based on their structural properties.

Quite often, the choice is made in favor of the lowest density materials (aluminum and magnesium are rather popular from that standpoint), without taking into consideration other important properties, such as the modulus of elasticity (aka Young’s modulus) and Poisson’s ratio. The more appropriate approach, however, would be to use a quality one might call specific stiffness—a ratio between Young’s modulus and density. Namely, the elastic modulus per mass density of the material. From this standpoint, both aluminum and steel are about equally attractive as the specific stiffness is about the same for both. In one of our experiments, aluminum stiffener was replaced with steel stiffener for comparison. Both configurations performed well, but the one with steel stiffener produced higher Q; therefore, aluminum was chosen for the final design.

For extreme situations, there are some exotic materials available, like CE7, CE11, and aluminum-silicon-carbide (AlSiC). Some of them require extensive custom tooling, which translates into significantly longer lead times and cost. Others can be machined using conventional CNC milling machines (like CE7 and CE11), but require carbide tools with TiN coating to produce acceptable surface quality. The main advantage of such materials is the specific stiffness. These materials have a much higher Young’s modulus than aluminum, while having about the same density. Disadvantages include high cost and a limited number of suppliers for such materials—which makes cost reduction for designs employing such materials quite difficult to achieve.

Table 1. Material Constants.

<table>
<thead>
<tr>
<th>Material</th>
<th>Young’s Modulus (GPa)</th>
<th>Density (g/cm³)</th>
<th>Specific Stiffness</th>
</tr>
</thead>
<tbody>
<tr>
<td>Brass and Bronze</td>
<td>113</td>
<td>8.57</td>
<td>13.13</td>
</tr>
<tr>
<td>Steel</td>
<td>200</td>
<td>7.90</td>
<td>25.32</td>
</tr>
<tr>
<td>Magnesium</td>
<td>45</td>
<td>1.74</td>
<td>25.89</td>
</tr>
<tr>
<td>Aluminum</td>
<td>69</td>
<td>2.70</td>
<td>25.56</td>
</tr>
<tr>
<td>Titanium Alloys</td>
<td>113</td>
<td>4.50</td>
<td>25.00</td>
</tr>
<tr>
<td>AlSiC</td>
<td>125</td>
<td>2.80</td>
<td>44.64</td>
</tr>
<tr>
<td>CE7</td>
<td>125</td>
<td>2.50</td>
<td>50.00</td>
</tr>
<tr>
<td>Diamond</td>
<td>1220</td>
<td>3.53</td>
<td>345.61</td>
</tr>
</tbody>
</table>

The comparison in Table 1 shows why aluminum is the material of choice, given its affordability, ease of machining, well developed plating processes, and other advantages. Diamond is added as a reference point only. We don’t suggest using it as a structural material for systems or subsystems, although its specific stiffness value appears very attractive.

Location, Location, Location

Location is not as important in design as it is in real estate, but location is still very important for designing systems and subsystems that require vibe isolation of microphonic-sensitive modules. Vibe isolated modules located symmetrically with respect to mounting points of system’s or subsystem’s chassis will behave more predictably than the one located asymmetrically. In cases where two levels of vibe isolation are employed—one immediately around the sensitive component and the other at the next level assembly—symmetry needs to be maintained at both levels.

Placing the vibe isolated payload right in the geometrical center of the next higher level assembly is challenging—expect a rather strong pushback from electrical engineers and PCB designers as such placement would leave a less-than-optimal configuration of the real estate available for electrical component placement and PCB routing. However, this is a small price to pay compared with greater electrical performance degradation due to uneven loading of the vibe isolation system, resulting in a more complex and asymmetrical nature of mechanical vibration imposed on microphonic-sensitive components/modules.

![Figure 1. FEA model (ANSYS) of a carrier plate under vibration.](image)
Positive Side Effects of Designing for Microphonic-Sensitive Devices

Most structures for microphonic-sensitive devices are designed with the reduction of displacement under shock/vibe in mind. Therefore, they are overdesigned from the point of view of pure structural integrity. This gives mechanical engineers peace of mind when it comes to overloading such structures for accelerated life testing, concerns about excessive flexing under load leading to violation of customer envelope, and other similar circumstances.

“Architecture Is Frozen Music”
—Johann Wolfgang von Goethe

The most generic architecture of a subsystem or system consists of several microwave modules aimed at possessing adequate structural qualities suited to carry vibe isolated payload, as is depicted in Figure 4. The modules are mounted on both sides of a carrier plate, which also serves as a thermal conduction interface to the customer’s mounting surfaces (usually considered an infinite heatsink), in addition to structural service duties. The structural rigidity is ensured by adding top and bottom stiffening plates, which would be positioned in close proximity to the limits of the envelope defined by the customer’s specifications. Multiple linkages ensure a solid connection between both stiffening plates. Structures like this allow for maintaining significant structural rigidity over quite long spans between mounting points at the interface/airframe. In cases where room between the outer surfaces of the microwave modules and customer’s envelope is very limited, steel stiffening plates will be more effective than aluminum ones.

Bolted Interfaces as Mechanical Attenuators

In addition to a passive vibe isolation system, bolted interfaces between the airframe and vibe isolated payload act as vibration dampeners based on the micromovements they allow and the friction associated with those micromovements. Therefore, the more sequential bolted interfaces exist between the mounting tabs of the subsystem’s chassis and the attachment points for microphonic-sensitive device/module, the greater the attenuation of the vibrations propagating through the structure from the customer’s airframe to the module.

Frequency Plans Are Needed for Mechanical Design

Subsystem/system $f_n$ should be far away from the natural frequencies of every module mounted onto it, unless one of the modules serves as a chassis for the entire subsystem/system. In that case, the $f_n$ for the module shall be calculated/modeled for the configuration where all modules (mass loads) are attached. The aforementioned mass loads will lower/reduce the $f_n$ of the so-called chassis module.

Mixed Material Sets Requires Careful Analysis

Designs with mixed material sets for structural components need to be carefully analyzed for the possible consequences of CTE mismatches (for example, stainless steel 316 (16 ppm/°C), aluminum 6061 (23.6 ppm/°C). With the change of temperature, such mismatch may produce statically preloaded conditions leading to an increase of $f_n$ for structures comprised of parts with dissimilar CTE. If there are no other parts/sections of the structure whose $f_n$ is higher, but not close to the $f_n$ of the module/section in question, this shift is of no consequence. However, if there are structures with a rather small offset of its $f_n$ to the $f_n$ of the chassis module, then the two natural frequencies may end up closer to each other than desired, producing some degree of mutual amplification of shock/vibration—not quite reaching resonance yet, but an undesirable condition nevertheless.

If the next higher $f_n$ module is constructed from similarly form materials with different CTEs, its $f_n$ will increase as well, helping to maintain the delta $f_n$ in the safe region. This illustrates the positive effects of consistency in design practices, which reduces the likelihood of less than desirable outcomes at hardware testing.

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Conclusion

Designing enclosures and other mechanical structures for systems and subsystems containing microphonic-sensitive devices presents substantial challenges. However, there are certain specific considerations and design approaches that allow for a greater degree of first-pass success. Understanding the nature of the degradation of electrical performance due to mechanical impact is one of the most helpful qualities mechanical engineers working in the field of electronic packaging design can possess.

About the Author

Sergey Sokol is a principal mechanical engineer in the Aerospace and Defense Business Unit at Analog Devices. Prior to his role at Analog Devices, Sergey held RF and microwave focused engineering roles for over 30 years. He holds a patent for 70 GHz PCB edge launch. Sergey graduated from Moscow State Aviation Technological University in Moscow, Russia. He can be reached at sergey.sokol@analog.com.

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