EDITORIAL

When Only the Best Will Do

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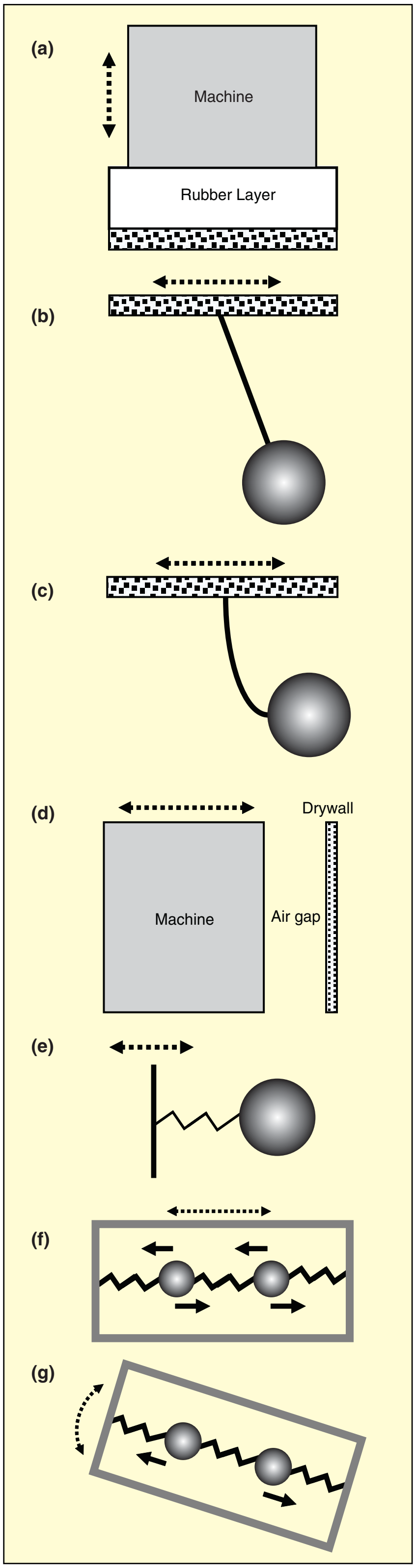
As distinct from engineers, politicians make "high-flying" mistakes, not only in their papers and implementations, but also when they talk to a broad audience. In Russia, former Prime Minister Victor Chernomyrdin was "famous" for his publicly outspoken "idioms" with unintentional comic effect. One of such expressions was uttered after a highly unsuccessful monetary exchange performed by the Russian Central Bank: "We wanted the best, but it turned out as always."

This idiom could be just slightly transformed to become a good motto for acoustical and vibration engineers: "We want the best, and it must turn out best." Is such a goal always achievable? The formula for success is simple: be professional and make no mistakes. However, its implementation is not easy, particularly for new designs or operational conditions, because generally, theoretical or experimental modeling is an approximation.

Indeed, it is impossible to segregate all the sources of discrepancy and therefore almost any simulation exhibits some error. Niels Bohr (one of the greatest physicists of the 20th century) used to joke: "Prediction is always difficult, especially if it's about the future." A more feasible goal is to make no obvious mistakes that can be predicted based on simplified mechanical models or analogies. However, despite Albert Einstein's warning "Make everything as simple as possible, but not simpler," even experienced engineers often develop unfeasible recommendations because of oversimplifying real-life phenomena.

I believe that *S&V* magazine is a fine place for practical engineers to share their experience on actual engineering problems. Let's consider some mistakes in order to avoid them in future.

Vibration Isolation with a Resilient Pad. It is a common knowledge that flexible mountings isolate machine vibrations. Very often, manufacturing engineers implement such solutions without consulting vibration engineers. As a result, many times the vibration is not reduced enough and sometimes even increased because vibration created by the operating machine at frequencies lower than its mounted resonant frequency is not isolated at all. Vibration at a resonance frequency may be amplified. The resonant frequency of the machine on its resilient mountings must always be well below the frequencies of the vibration to be isolated. If the mountings are steel springs, the resonant frequency is typically low, and the chances for success are commonly higher. With a rubber pad under the machine (Figure 1a), the resonant frequency may be relatively high and can coincide with one of the main vibration frequencies (in particular, with the motor speed of rotating machinery). I have seen quite a few such failures. In some cases, the machines were big and heavy, so the remounting procedure was costly.



Vibration Isolation of a Pendulum. This mistake is not as obvious and was made by a specialist invited to isolate a sensitive device from the environmental vibration at frequencies over 5 Hz. The specialist suspended the device from a lightweight thread to form a traditional pendulum (Figure 1b). The resonant frequency of such a pendulum can be calculated as:

$$f_{pend} = \frac{\sqrt{g / L}}{2\pi} \approx 0.5 \text{ Hz}$$

This is at least 10 times below the important frequencies. (Here, g = 9.81 m/s^2 as the acceleration of gravity and the thread length L was 1 m). However, the environmental vibration still badly affected operation of the device. In his theoretical analysis, the specialist neglected the mass of the thread, and the thread could be simulated as a uniform flexible string stretched by the tension force $T = Mg$. M is the mass of the device suspended from the thread. As a result, the wave motion efficiently transmitted the environmental vibration from the pivot to the device at the fundamental resonant frequency of the string (Figure 1c). If the total mass of the thread is m, this fundamental resonant frequency is given by:

$$f_{spr} \approx \frac{\sqrt{TL / m}}{2L} = \pi \sqrt{\frac{M}{m}} f_{pend}$$

The ratio $M/m \approx 160$, so the fundamental resonant spring frequency f_{spr} exceeded 40 times the resonant pendulum f_{pend} and was equal to 20 Hz, well within the operational frequency range.

Sound Insulating Wall. A large fan in the utility room of a commercial building created a terrifying low-frequency noise that propagated to a nearby office via the adjacent wall. To reduce the noise, an acoustical consultant recommended erecting an extra single drywall partition 0.6 m (2 feet) away from the fan (Figure 1d). However, the manufacturing engineer reduced this distance to 0.05 m (2 inches) to save the utility room area for new equipment. He thought that with the same wall, the recommendation should work anyway. Post-construction reality proved harsh – the drywall "roared" more loudly than the fan itself. The acoustical consultant returned, looked at the design, and quickly explained what happened. If the air gap between the drywall and fan is narrow, a 1-DOF vibration system is created where the air gap works as a spring and the drywall plays the role of a lumped mass (Figure 1e). The resonant frequency of this system is given by equation:

$$f_{res} \approx \frac{\sqrt{\rho c^2 / M_s d}}{2\pi} \approx 85 \text{ Hz}$$

where the surface density of the drywall $M_s \approx 10$ kg/m^2, the gap thickness $d = 0.05$ m, air density $\rho = 1.3$ kg/m^3, and sound speed in air $c = 330$ m/s. Such a low-frequency resonance amplified the noise of the fan at the same frequency.

Fatigue Failure of MEMS Design. A MEMS design was tested on a single-axis shaker to predict vibration fatigue failures (Figure 1f). Mechanically, this device could be approximated by a system of two similar masses attached with identical springs to opposite walls of a rigid container. The masses were connected to each other with a more compliant spring. After a relatively long test with no failures, the MEMS device was approved for use but it structurally failed in the real environment. The compliant element, approximated as the middle spring, broke in all the specimens. The design engineer did not know that a coincidence of the excitation frequency with the resonant frequency is not enough to create a resonance in a multi-DOF system; the axes of the excitation force are also important. The 2-DOF system shown here has two natural modes of vibration: (1) with the masses moving "in phase" in the same direction; and (2) with the masses moving in the opposite directions at the same time.

When tested on a single-axis shaker, both masses were driven by similar inertial forces, which would only excite mode 1. In this case, the masses move "in phase" and therefore the middle spring is not deformed at all. Under actual operational conditions, the MEMS device was firmly attached to the vehicle that performed both translational and rotational vibration. The rotational vibration occurred mostly around the perpendicular axis as shown in Figure 1g. Therefore, the masses were excited by alternating centrifugal forces similar in magnitude and oppositely directed. As a result, mode 2 – which deforms the middle spring twice as much as the edge springs – was excited in the real environment. A relatively high stress caused by this deformation destroyed the middle element of the MEMS device. S/V

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