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Good Vibrations

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FEATURES

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Editor's Message

Welcome to our Good Vibrations Issue!

Sunit Girdhar,
Westside Acoustics and Vibration Engineering

To celebrate NoiseCon2026, the theme of this issue is the same as the theme of the conference where we focus on vibration and how it affects us. Vibration is fundamental not only to humans, but to all living things. From the microscopic hairs on a caterpillar's body to the structural rumble beneath a busy pier, vibration is everywhere, and understanding it remains as relevant as ever.

We open with something that might just be our most delightful contribution yet: a look at how caterpillars sense their world not through ears, but through hair-like flow-sensing receptors that detect vibration. Nature, it turns out, has been solving vibration problems far longer than we have.

We then present an article of vibration criteria that guide engineers in assessing whether a structure, machine, or environment meets acceptable performance standards. Complementing this, we present a case study of vibration monitoring for the replacement project at the iconic Santa Monica Pier.

We then provide a practical reference on vibration data for mechanical equipment, a resource that practitioners will find valuable in day-to-day assessment work. The issue then turns to design and mitigation. Our article on floating floor design explores how important it can be to properly vent the design.

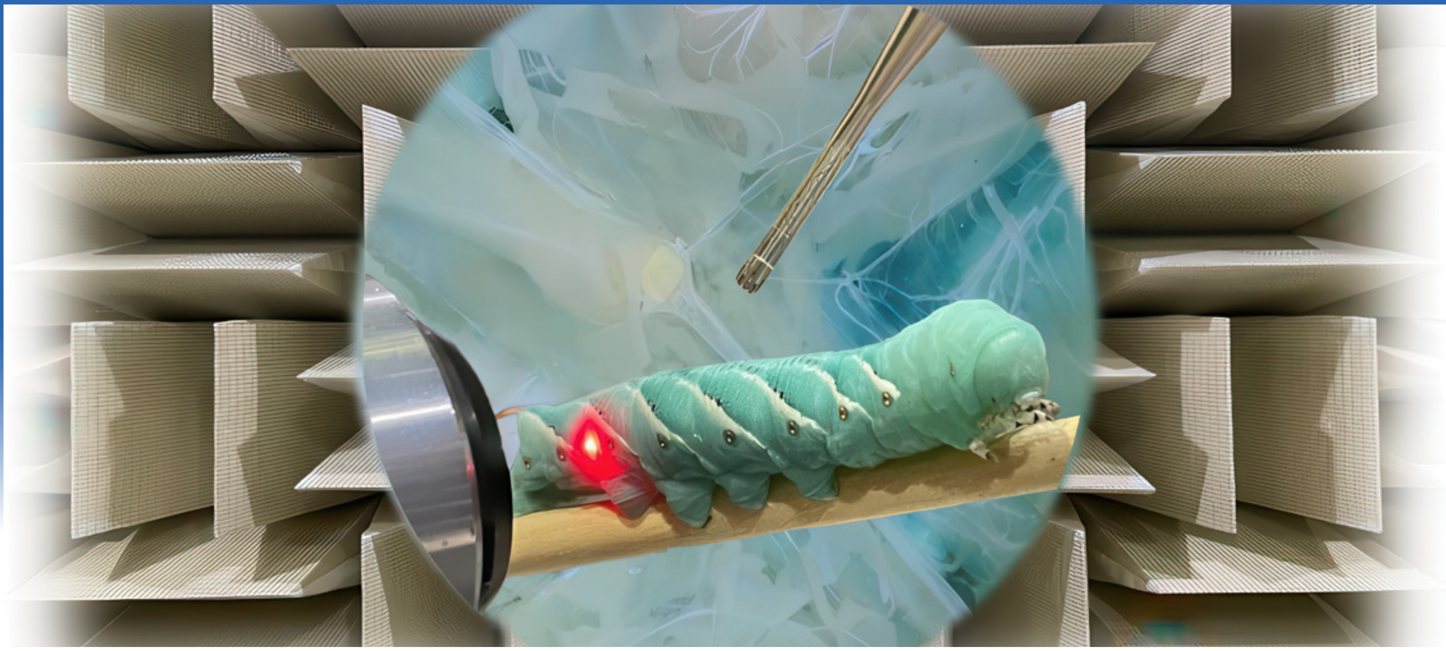
We round out the issue with some compelling case studies. We present a detective case study on tire problem that was solved using vibration analysis (watch out Sherlock Holmes, we have a new detective in the house!). We then present a monumental staircase vibration case study that illustrates how even seemingly simple structures can present complex dynamic challenges.

We also present some real-world examples where vibration may lead to noise concern and how to design for that. Finally, we present a case study of where the same part printed by different FDM printers is compared for their material properties.

Whether you are a seasoned practitioner or a curious newcomer to the field, we hope this issue gives you a renewed appreciation for the science of vibration, and perhaps a few new tools for your work.

Also, see you at NoiseCon 2026! We are working on some exciting 'games' for you so look out for that.

Do you have an article or topic in mind for a future issue? We love to hear from our readers. Share your feedback with us and follow us on LinkedIn for updates. ■



Manduca sexta caterpillars hear using hairs as flow-sensing receptors

Sara Aghazadeh[1]& Aishwarya Sriram[2], Carol Miles[2], Ronald Miles[1]
[1]Mechanical Engineering Department, [2]Biological Science Department, Binghamton University

It is about 150 years since Hermann von Helmholtz established the physical basis of human hearing, describing how acoustic pressure is converted into mechanical motion in the ear. Shortly after that, inventors such as Emile Berliner and Thomas Edison developed early microphones for telephony, translating similar pressure variations into electrical signals. Today, there are billions of microphones manufactured each year for various applications in communications, entertainment, human health diagnostics, and transportation nearly all based on the detection of sound pressure, inspired by the mechanisms employed in human hearing.

We hear the sound pressure created when a sound wave arrives in our ears. But in that traveling sound wave, the air particles are actually moving back and forth as the sound pressure fluctuates in time and space. In a great many animals, like insects, they hear that sound by detecting the air-flow rather than the fluctuating pressure detected by our eardrums. Detecting a flow is another way to hear rather than detecting pressure.

In this research, we explore how insects perceive sound-induced airflow. An aim of this work is to learn how to create MEMS microphones that are designed to detect this flow rather than detect sound pressure, as is done in essentially all microphone designs since the time of Edison's telephone. We investigate the hearing mechanism of *Manduca sexta* caterpillars- a common garden pest, the tobacco hornworm- to examine whether their hearing is due either to the detection of airborne sound using any tympanal membrane to sense the fluctuating pressure of a sound wave, or due to sensing the flow of the air particle motion using tiny hairs on their body. We also examine whether this animal is able to detect sound due to the sound-induced vibration of the surface (such as a leaf or plant stem) that they grasp with their feet.

We examined the caterpillars' behavioral responses to sound using two key targeted frequencies, pure tones of a low frequency at 150 Hz, and a high frequency of 2000 Hz. Previous studies have found strong behavioral responses at 150 Hz in tuning curve experiments at 80-90 dB. By using laser vibrometry, we measured the sound-induced motion of a thoracic hair on the caterpillar's body, and we observed a natural resonance of the hair at 2000 Hz. While we don't normally expect insect hairs to be effective sound detectors at such high frequencies, this observation motivated further examination to look for behavioral responses at this frequency.

We monitored caterpillars' behavioral responses to vibrations of the substrate that the caterpillars were holding on to, and to air-borne sound while we recorded the amplitude of the substrate vibration using an accelerometer. The caterpillar responses were classified into three categories: jump startle– a jerky reaction, lifting of thoracic and anterior abdominal segments; twitches– localized movements in any segment, freezing– cessation any movement. The stimulus level that was just high enough to elicit a behavioral response was considered as a detection threshold. You can find their reaction to sound here: https://sites.google.com/binghamton.edu/natures-microphones/home

The results revealed that the caterpillars were 10-100 times more responsive to airborne sound than sound-induced vibration of the substrate detected by their feet; this confirms that they perceive airborne sound at a low-frequency of 150 Hz and a high-frequency of 2000 Hz.

Figure 1. Manduca sexta caterpillars on tobacco plants

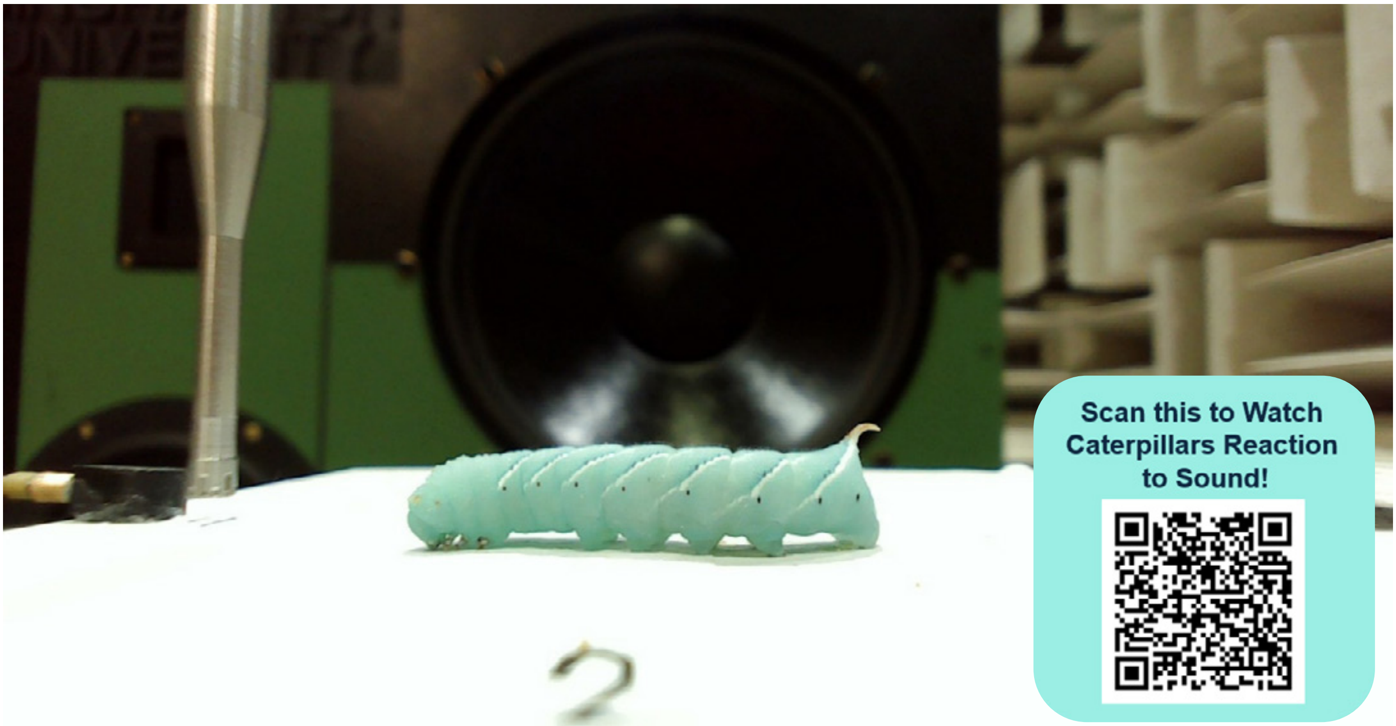


Figure 2. Sound Exposure Experimental setup; Play sound and observe caterpillar behavioral response at 150 Hz in the anechoic chamber.

Figure 3. Credited by Binghamton University; Team photograph - Greg Schuter & John Brhel

Also, *M. sexta* caterpillars display graded responses to sound stimuli corresponding to the intensity of the stimuli.

This raised a question: if they hear, where are their ears? To address this question, we performed the ablation method, removing targeted hairs on their thorax and abdomen segments on their body. The result of the behavioral response comparisons before and after removal of the hairs shows a greatly reduced ability of the caterpillars to detect sounds without the hairs. We found that thoracic hairs are primarily sensitive to higher frequencies, around 2000 Hz, while long abdominal hairs are associated with lower frequency detection, around 150 Hz. This suggests that different hair receptors contribute to detecting different components of the acoustic signal, and provides evidence of non-tympanal sound detection in these caterpillars for these specific frequencies.

Furthermore, we developed a mechanical setup to figure out whether these caterpillars respond to the air particle velocity or a sound pressure. A loudspeaker was used as a dipole source, producing distinct acoustic fields: A pressure-dominated field in front of the speaker, and a particle velocity–dominated field to the side.

We first determined the behavioral thresholds (SPL in dB) in the velocity-dominated field, where caterpillars exhibited a jump startle response. Then, we examined caterpillars' behavioral response in the field dominated by pressure with the same SPL. The result showed that much higher SPL is needed to elicit response in the pressure field than when the sound field is dominated by particle velocity.

The ratio of the pressure to the velocity in a sound field is the acoustic impedance. Using measured impedance values as a function of distance from the speaker, we estimate that the pressure required to elicit a response in the velocity field (side) is substantially lower than in the pressure field (front). This supports the hypothesis that *Manduca* caterpillars are more sensitive to particle velocity (air motion) than to sound pressure. The results suggest that these caterpillars use sound velocity as a cue to estimate the distance of an approaching parasitoid wasp. They freeze when the threat is distant, twitch as it approaches, and jump startle when it is very close. ■

Good Vibration Measurements, Criteria, and Standards

Ethan Brush, Acentech Incorporated, ebrush@acentech.com

We experience vibrations everywhere in our world. To say that something is vibrating means that it has an oscillating motion about a fixed reference position. The word comes from Latin vibrationem, with the closest translation meaning "shaking" or "trembling". Vibration is a concern to an enormous number of applications, so the ability to measure and quantify it is important. There are a multitude of quantitative ways to describe the motion of what is being measured. We can measure the movement as either a displacement, velocity, or acceleration; but sometimes as a jerk (derivative of acceleration) or even more obscurely as a jounce (second derivative of acceleration). When we know the frequency content of the measured vibration, usually via a spectrum analysis, conversion between all these levels is straightforward.

The most common way to measure vibration is with a piezoelectric accelerometer sensor, which when moved outputs a voltage proportional to acceleration. The SI units for acceleration are m/s^2, however, the unit "g" is often used to distinguish acceleration relative to free fall on earth. A quiet basement floor ambient vibration environment in a research laboratory could be expected to measure on the order of a few tenths of micro-g's, and shock events on aerospace components could reach several tens of thousands of g's. This tremendous range is on par with the world of acoustics in terms of the orders of magnitude between the quietest and loudest observable sound pressure levels.

Building Vibration Criteria

Let's focus on the huge range of vibration criteria one might encounter in the building industry. Vibrations in buildings can be bothersome or disruptive to occupants and activities. Whether they are residents trying to sleep, people who are distracted while working at their desks, or researchers trying to use a powerful microscope, the presence of unwanted vibration can be debilitating. The amount of vibration that is disruptive to a facility can vary by many orders of magnitude. The figure below presents an infographic that shows typical vibration criteria across a variety of building uses in velocity units of micro-inches/second. Research and semiconductor facilities are at the lowest level and most sensitive end of the spectrum because they can house very precise and sensitive instrumentation, often observing things at an atomic level. Above that, there are hospital spaces that may house surgery or magnetic resonance imaging (MRI) suites. Concert halls and recording studios are considered sensitive because vibrations above this range have the potential to cause intrusive structureborne noise. Animal facilities are sensitive because the animals in holding rooms have different thresholds of perception than humans, are always present (cannot be moved easily) and there can be significant consequences if the animals become stressed. An accepted average human threshold of perception to vibration is a velocity level of 8,000 micro-in/sec and is labeled on the chart (ANSI S3.18-1979). The criteria for occupants of residential and office buildings straddle this threshold. Notice how many spaces can be affected by vibration levels that we can't even feel.

Data centers and museums are higher up on the sensitivity spectrum. Given their mission critical function (data centers) and housing of potentially irreplaceable items (museums), vibration control is still a top priority. At the very top of the range lie vibrations that may be damaging to the building structure itself.

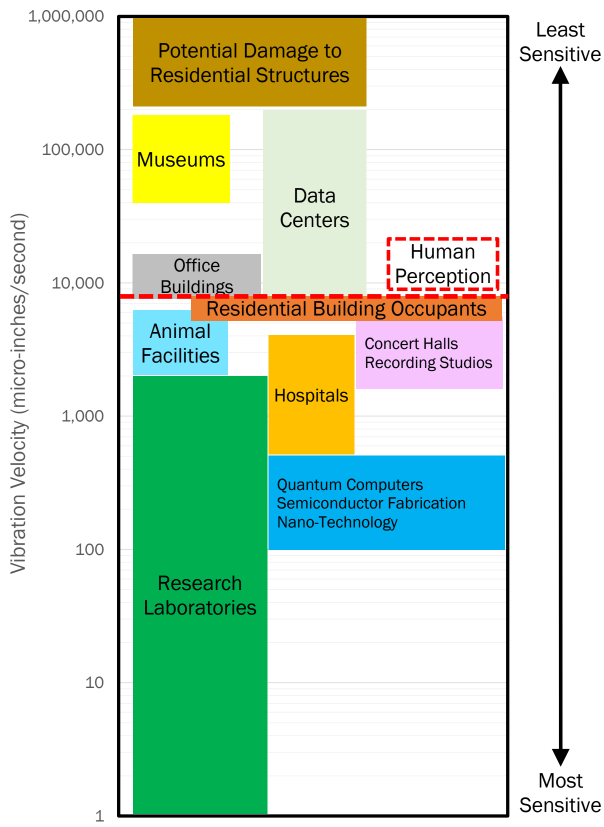


Figure 1: Typical Building Vibration Criteria

Comparing Apples to Apples

How we describe the amplitude of vibration matters a great deal. The vibration limits on the chart above are all in terms of micro-inches/second, but even when choosing consistent units there are a multitude of ways the amplitudes with the same units can be described. We describe the amplitude of sinusoidal signals mostly as root mean square (RMS), zero-to-peak, or peak-to-peak. Choose this one wrong, and your result may be off be as much as a factor of two. Next comes the various signal processing choices one makes to arrive at a frequency spectrum such as window length, data windowing function, and frame overlap percentage. Then, one must choose the amplitudes between narrow band, 1/3 octave, octave, or even power spectral density. If any of these choices are different for two people making the same set of measurements, then the results aren't fully comparable. To complicate things even further, some criteria are based on the peak waveform values. For a transient signal the difference between an RMS amplitude and the peak waveform value, called the crest factor, can be well over a factor of 10.

Because there are so many ways to describe a vibration amplitude, the presentation of data should include sufficient information indicating how the results were obtained so that someone else could reproduce the same processing steps. Equipment vibration specifications should also spell out exactly how to arrive at the limit values given in their documentation. Some equipment vendors get this right, and others do not.

Footfall Vibration Measurement Standard

It is often necessary to evaluate the floor vibration performance of existing buildings due to people walking. A common approach is to measure floor vibrations induced by a person walking and assess whether the floor is suitable for its intended use (e.g., residential, office, research laboratory). Other than a few generic guidelines there is currently no published standard for conducting footfall vibration testing in buildings. As a result, practitioners rely on their own methods for testing, data analysis, and reporting of results. This can lead to disparate results for all the reasons stated above and more.

The ASTM committee E33 on "Building and Environmental Acoustics" has created a new working group with the challenge to develop recommendations for conducting footfall vibration measurements in buildings.

The hope is that whether the group produces a guide or a full standard, the building industry can have a document to follow that allows for consistent footfall vibration tests. This will be of great benefit for mass timber buildings, where the knowledge of their vibration performance and design guides is less mature than for steel and concrete buildings. Indeed, the entire building community will benefit from more consistent empirical data to further inform and validate prediction and design methods. Please reach out if you have opinions about what should be included in the ASTM recommendations for footfall vibration measurements.

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Author Bio

Ethan Brush is a principal consultant at Acentech Incorporated in Cambridge, MA. He has 20 years of engineering and project management experience directing teams in the areas of noise and vibration testing, analysis and design of vibration mitigations. ■

Vibration and Noise Monitoring at the Santa Monica Pier Bridge Replacement Project

Christian Fogstad, Austyn Crites, Kush Parashivamurthy, and Marcus Pacheco, Sigicom Inc.

Background

Construction in tight, public environments has a way of turning “small” impacts into big problems. A few seconds of heavy equipment vibration can feel like a major event to someone standing nearby, and a brief spike in noise can spark complaints, delays, or questions about damage - especially when the work is happening next to older infrastructure, structures or buildings. That is why vibration and noise monitoring has moved from a niche precaution to a practical tool: it replaces guesswork with data and helps teams manage risk before conflicts arise. “Saying something did or did not happen, is not the same as proving something did or did not happen” quote Dr. C.H Dowding.

Currently one of California’s most recognizable coastal landmarks, The Santa Monica Pier Bridge Replacement Project is a good example of this predicament. Construction is taking place alongside aging structures, active pedestrian areas, utilities, and commercial activity. Demolition, heavy equipment operation, compaction, and material placement all have the potential to generate vibration and noise; the question is not if the public will be affected, but how the project team can prove clearly and credibly that such impacts are being appropriately managed.

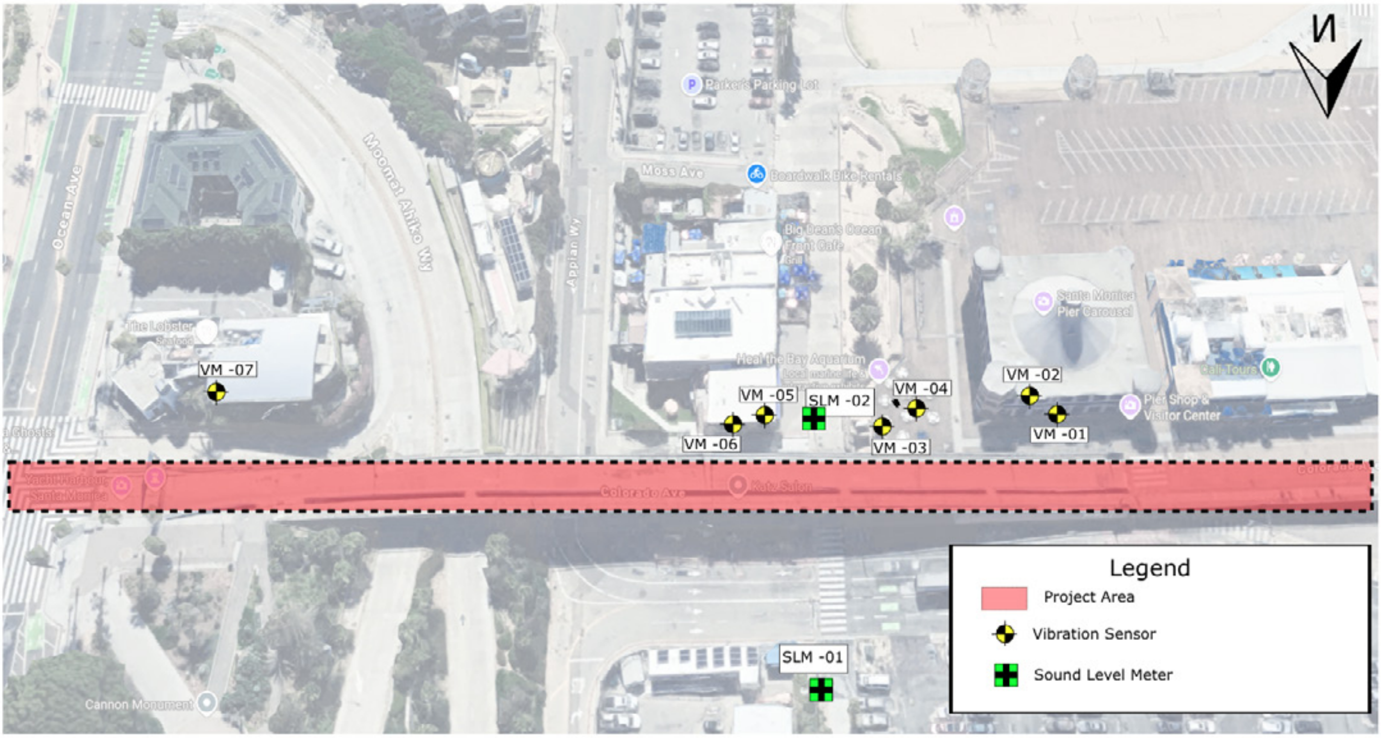
Thus, this project’s management team required a monitoring program with three clear objectives:

1. Protecting existing surrounding structures and assets.
2. Maintain compliance with applicable standards while monitoring according to project-specific pre-defined thresholds.
3. Provide reliable, near real-time information for quick decision making.

This article shares lessons learned from how the monitoring was planned and is currently deployed and interpreted.

Background

Typically, any landmark site comes with a wide stakeholder circle. On The Santa Monica Pier Bridge Replacement Project (hereafter the Pier), this team includes the project owner, municipal representatives,



Monitoring locations at the Pier

general contractors, subcontractors, monitoring specialists, and regulatory authorities. Equally important are the adjacent operators and the surrounding community, because comfort and perception are integral parts of the overall risk.

One early takeaway has been clear: monitoring serves as much as a community-outreach platform as it does an engineering control and evaluation tool. When expectations are not aligned before work commences, parameters such as what is being measured, where measurements occur, which thresholds apply, and how exceedances will be handled, may quickly escalate into a dispute once unexpected noise or vibration happen.

At the Pier, such concerns are not theoretical as aged timber and concrete elements may exhibit lower tolerance for repeated energy input and fatigue exposure. Added to these structural concerns are the human comfort impact of publicly occupied areas and the fact that municipal projects carry higher taxpayer accountability. This combination makes vibration and noise monitoring essential.

Approach

Carefully considered monitoring projects starts with the basics: understanding the environment and establishing a baseline. Baseline work is especially valuable because it provides the “as-is” or current conditions of a proposed construction site. No site is 100% “quiet” and free of noise and vibration. At the Pier, pedestrian, nearby vehicular traffic, and ocean activity create their own vibration and noise signatures. Capturing ambient levels early makes it easier to answer, “Was that construction related, or was that just the Pier being the Pier?”. Based on the ambient readings the overall construction monitoring scope was established as listed below.

1. Protecting nearby buildings and infrastructure
2. Safeguarding community health and comfort
3. Complying with legal and regulatory requirements

   The city of Santa Monica established the threshold based on the structure type and age. Two vibration thresholds were established:

   - 0.2 in./sec PPV - established under Coastal Development Permit Special Condition #7 - as the warning level.
   - 0.5 in./sec PPV - the damage-potential threshold for transient vibration sources.

   Technically, the monitoring specifications were consistent with the USBM RI 8507 standard. For noise, the threshold was 90 dBA, which reflects +5 dBA above ambient conditions.

4. Guiding construction planning and equipment selection
5. Managing stakeholder expectations
6. Reducing project risk and liability

Monitoring locations were deliberately selected to capture construction-induced vibration and noise, not simply placed where installation was deemed convenient. For vibration monitoring, the project team evaluated structural behavior, load paths, proximity to anticipated activities, and areas where stakeholders were most sensitive or concerned. With a limited number of systems available, the real work lay in choosing positions that best represented how the structure would respond and then maintaining stable, well-coupled installations throughout the project.

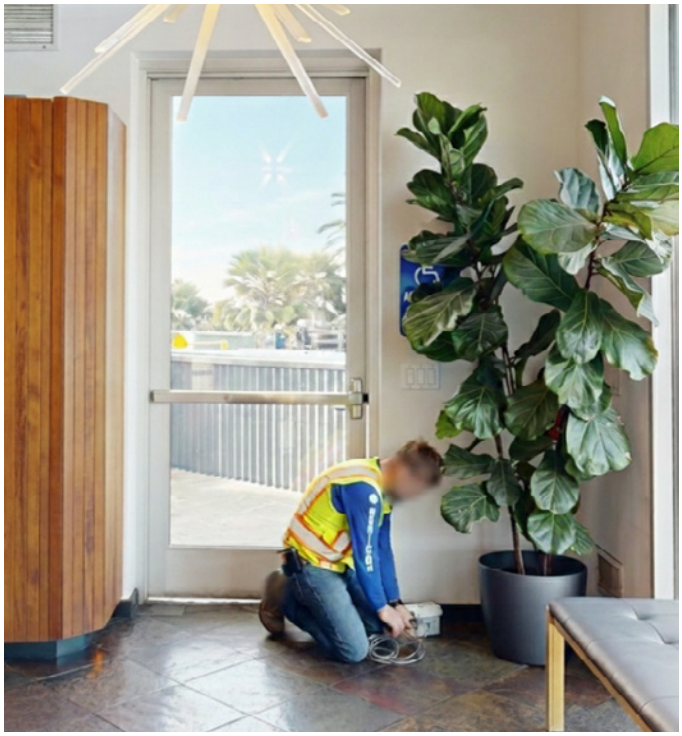
Similarly, noise monitor placement required thoughtful consideration to ensure measurements reflected conditions at the nearest sensitive receptors, avoided or properly accounted for localized noise sources, and remained reliable under prevailing wind and weather patterns. The overview below provides selected monitoring locations.

Deployment in a coastal, public setting introduces its own set of challenges. Vibration sensors require protection from salt air and moisture, along with secure, stable coupling to ensure that transmitted energy is captured accurately. Proper orientation is equally critical, allowing the system to evaluate true directional three-axis motion rather than random data.

Power and communications matter more than people expect; a monitoring program that drops offline at the wrong time undermines confidence. Access windows are constrained by public use and construction sequencing, so installations and checks are coordinated around low-traffic periods.

Selecting systems which provide near real-time data access was a deliberate choice. On a high-visibility job, waiting days to review logs and data is not sufficient. When levels rise, the team needs to quickly ascertain approaching thresholds and whether the activity should be adjusted.

Collecting accurate data is only the beginning. The real value of monitoring emerges through thoughtful analysis and interpretation by transforming raw measurements into insights that guide decisions. This exercise validates performance and ultimately protect people and projects.



Typical vibration monitoring placement

In the United States, peak particle velocity (PPV) is most commonly used when evaluating vibration, while dBA provides a familiar reference point for communicating noise levels. Experienced reviewers also consider duration, axis dominance, and, when available, frequency content to understand what likely produced an event and whether it is meaningful or related to construction activity. Not every spike represents a risk; informed judgment is what prevents both alarm fatigue and complacency.

Typical sources include heavy equipment operation, demolition activities, compaction, and traffic operations. When vibration or noise levels approach predefined thresholds, the project team can review methods and sequencing to adjust equipment use or change the timing of certain activities. The benefit is fewer unnecessary shutdowns. Instead of reacting to perception, the team can respond to measured conditions, with a clear record of what happens and why.

Reporting is tailored to support decisions. Time-stamped event summaries, trend graphs, comparisons to thresholds, and short interpretive notes help keep everyone informed using the same data set. The value chain is straightforward: monitoring leads to analysis, analysis leads to reporting, and reporting supports data driven decision-making.

The outcomes are what project teams seek on a public, complex site: compliance with established criteria, no documented vibration-related damage attributable to construction, and sustained stakeholder confidence. History shows that monitoring improves stakeholder conversations. When questions arise, the team has objective records rather than anecdotes. When adjustments are needed, they can be targeted and proportionate.

Five lessons have emerged.

1. Baseline conditions matter most in dynamic public environments.
2. Monitoring system reliability is the priority when you have a small number of measuring points.
3. Action levels must be paired with clear response steps.
4. Expert interpretation is what turns “data” into “decisions”.
5. Communication is part of the scope: credibility depends on clarity as much as measurement.

Thoughtful vibration and noise monitoring are not just damage prevention. At places like the Santa Monica Pier, it is about enabling construction to move forward while balancing progress with preservation and doing it in a way that is technically credible and publicly defensible.

Done well, monitoring becomes the calm voice in the room (or on the Pier) when tensions rise.

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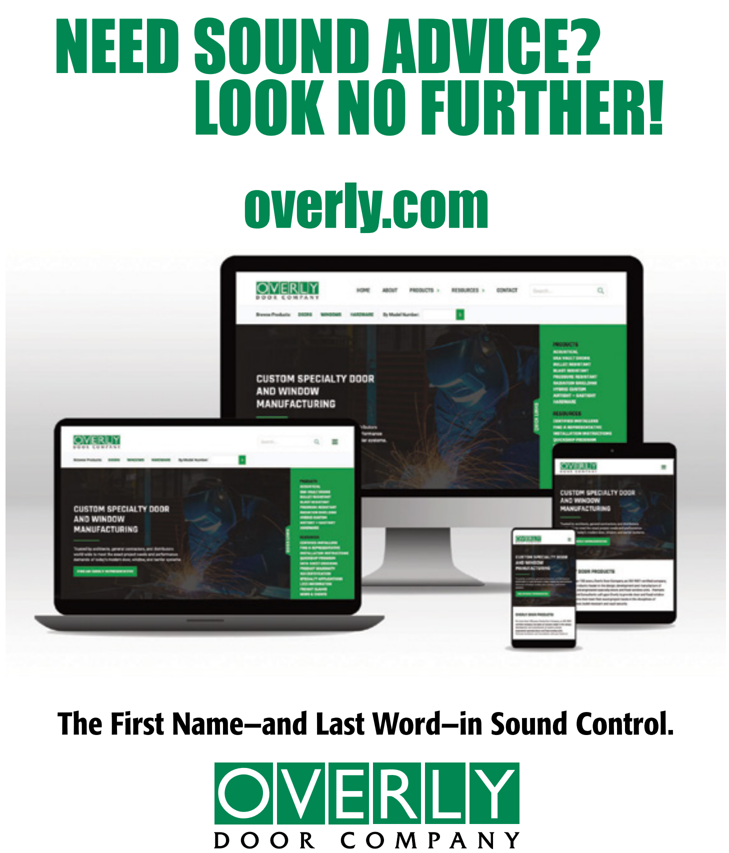
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Vibration Data for Mechanical Equipment: Shake Up the Paradigm

Brandon Cudequest, Acentech, bcudequest@acentech.com

Vibration isolation guidelines are based in industry best practice, owing much of their current form to the seminal work done by vibration isolator manufacturers in the 1960s (Mason, 1966). Engineers like Norm Mason understood that vibration isolation is a multi-degree of freedom problem and cannot be solved through single degree of freedom analysis. In addition to the floor deflection, engineers need to consider vibration severity: a large fan is capable of greater force than a smaller fan (think Newton's second law). Isolator deflection requirements need to consider these factors and cannot be based on the equipment rotational speed (i.e., forcing frequency) alone.

These isolation guidelines were adopted into the "Noise and Vibration" chapter of the *Handbook—HVAC Applications* published by the American Society of Heating, Refrigerating, and Air Conditioning Engineers (ASHRAE). They became ubiquitous and are the basis for many mechanical engineers' specifications. Over the years, these recommendations have marginally changed as new equipment technologies come to market and ASHRAE adapts to reflect contemporary practices.

The vibration isolation section of the "Noise and Vibration" chapter is undergoing revision for the 2027 edition of *ASHRAE Handbook—HVAC Applications*. Chief among the changes is a prescriptive and performance-based approach for isolator selection henceforth called "Simplified Engineering" and "Advanced Engineering" respectively. The Simplified Engineering approach will look familiar: an engineer selects vibration isolation measures from a table based on best practices. The Advanced Engineering method allows an engineer to select vibration isolation measures based on detailed analysis of the vibration source, sensitivity of adjacencies, and the structure type.

ASHRAE's Committee on Sound and Vibration will continue to help engineers and consultants understand the nuances of these approaches. For now, this article is fundamentally concerned with the question, "what does the industry need to make the Advanced Engineering method a success"? Is there a feasible source-path-receiver model that works for vibration? Short of Finite Element Analysis (FEA) or extensive field testing, what pieces of this puzzle are achievable on most projects?

Type of Equipment		Equipment Location	
		Noncritical	Critical
		Static Deflection, Inches	
Fans: Centrifugal, Axial, Package Air Handlers Below 40-inch Wheel			
	Above 1000 rpm	0.25	1.0
	600-1000 rpm	1.0	1.0
	400-600 rpm	1.0	1.0
	300-400 rpm	1.0	2.0
Above 40-inch Wheel			
	Above 1000 rpm	0.25	1.0
	600-1000 rpm	1.0	1.0
	400-600 rpm	1.0	2.0
	300-400 rpm	1.0	2.0
	175-300 rpm	2.0	2.0

1973 ASHRAE Systems Handbook → Present ASHRAE Applications Handbook

Type of Equipment			Equipment Location					
			Slab on Grade			30 to 40 ft span		
	Horsepower	RPM	Base Type	Isolator Type	Min. Defl., in.	Base Type	Isolator Type	Min. Defl., in.
Fans: Axial, Plenum, Cabinet, Centrifugal Inline								
Up to 22-inch diameter	All	All	A	2	0.25	C	3	0.75
24-inch diameter and up	<=2 in. SP	Up to 300	B	3	2.5	C	3	3.5
		301 to 500	B	3	0.75	C	3	2.5
		501 and up	B	3	0.75	B	3	1.5
	>=2.1 in. SP	Up to 300	C	3	2.5	C	3	3.5
		301 to 500	C	3	1.5	C	3	2.5
		501 and up	C	3	0.75	C	3	2.5

Base Types:

A. No base, isolators attached directly to equipment
B. Structural steel rails or base
C. Concrete inertia base

Figure 1 – Comparison of ASHRAE fan vibration isolation guidelines from 1973 and 2023.

To get from force to velocity, we need to characterize at least three components: the equipment vibration levels, the dynamic performance of the isolation mount, and the frequency response function of the floor structure. It's relatively straightforward to understand these elements separately and at discrete frequencies, it's another to know the application-specific values and possible mode coupling across a broad range of frequencies.

When I presented this content at the Acoustical Society of America (ASA) Honolulu meeting, I offered a live poll to attendees. The question was "when designing vibration isolation, I am least sure of…" The options were "Equipment Force Input," "Isolator Insertion Loss," "Structural Response," and "All of The Above." Based on 16 respondents here are the results:

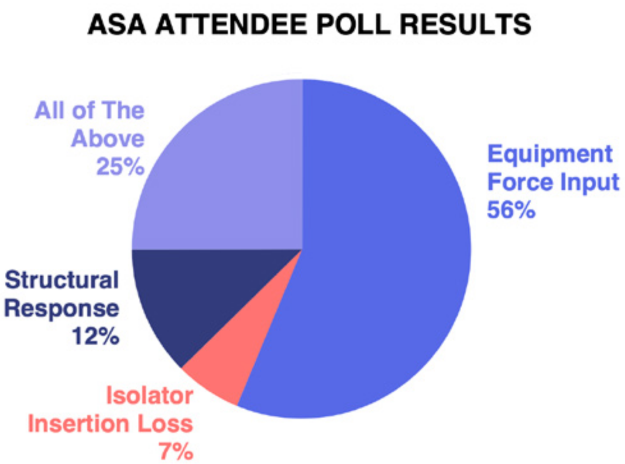

Figure 2 – Results from ASA attendee poll

"Equipment Force Input" was the overwhelming first choice at 56%. If "Equipment Force Input" and "All of the Above" are combined, there is a significant uncertainty around equipment vibration. Let's break down these topics into their fundamental components and explore the uncertainty around each element.

For equipment vibration, two major sources are unbalance and misalignment forces (see Figure 3 – Typical vibration forces in rotating and reciprocating equipment.).

Unbalance and misalignment forces have known causes and frequency relationships related to the equipment rotational speed. We'll ignore any bearing-related vibration forces, which are more indicative of the age and condition of the equipment. For certain equipment types, there are industry guidelines for balance quality and vibration severity. For fans, this is AMCA 204. Similar standards exist for pumps (ANSI/Hydraulic Institute Standard 9.6.4) and cooling towers (CTI 163).

In subsequent conversations with consultants and manufacturers following the ASA meeting, there is significant uncertainty in translating these values to actual field conditions. For example, AMCA 204 is a lab rating, likely at one operating point. How does this change as the fan is mounted to a unit chassis and operating at non-peak conditions? There are relatively few test standards for equipment vibration, such as ISO 12354-5:2023, but the test rigs are limited to lighter weight equipment. Useful for small heat pumps but not practical for commercial air handling units.

Next, we have to consider isolators. Consultants often oversimplify elastomers as springs and use static deflection as a guarantor of natural frequency. Ultimately, engineers need the dynamic stiffness of the isolator as this raises the natural frequency of the isolator well above theoretical models (see Figure 4 – Comparison of natural frequency of a theoretical spring and other

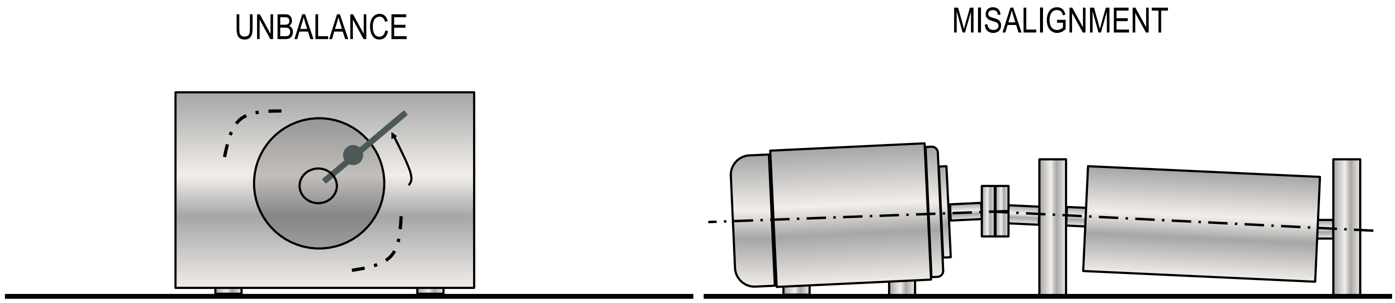

Figure 3 – Typical vibration forces in rotating and reciprocating equipment.

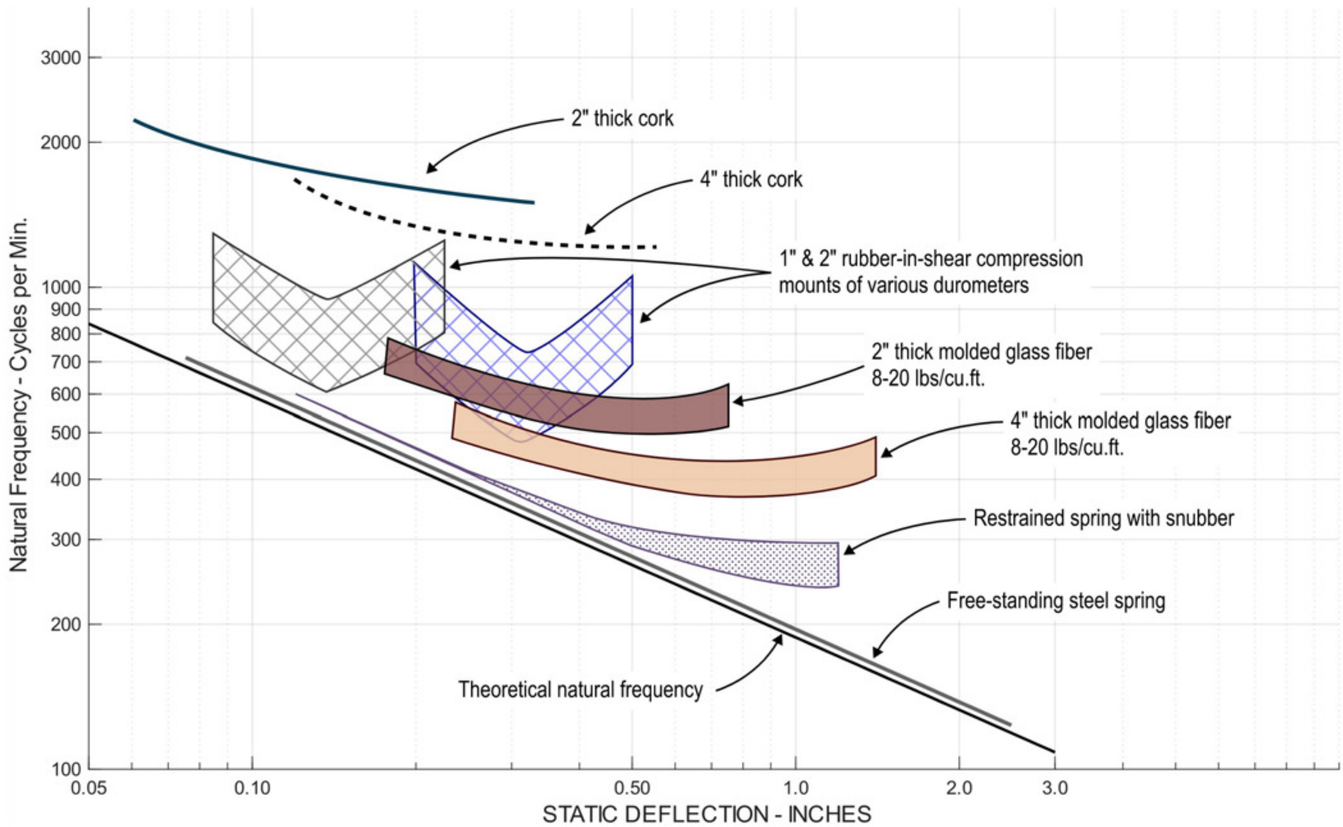

Figure 4 – Comparison of natural frequency of a theoretical spring and other isolator types. (Eberhart, 1966)

isolator types. (Eberhart, 1966)). The natural frequency of a spring isolator is easier to model, but high-frequency surge frequencies are difficult to predict for different load conditions (Ungar, 2007).

Several groups are pushing the state of the art in isolator insertion loss understanding. The University of Kentucky Vibro-Acoustics Consortium has explored the feasibility of the ISO 10846-1 test rig and an impedance matrix approach to quantifying isolator performance (Sun, 2015). Jerry Lilly has created a simple but effective rig and presented insertion loss values for springs and waffle pad-style isolators (Lilly, 2024). We're getting close to understanding isolators in ways that go beyond the simple static deflection curves.

Finally, engineers have to know the response of the floor structure and the resulting velocity. This requires close collaboration with the structural engineer during design. At a base level, the engineer and consultant should know the max acceleration caused by the marginal deflection of equipment. It's tempting to focus solely on the structural stiffness but if we recall our basic transmissibility curve, damping plays a significant role in the amplification response at resonance. Both stiffness and damping need to be optimized. The level of modeling and analysis should be scaled to the complexity and severity of the vibration problem and the sensitivity of the receiver.

A refined analysis is possible if engineers have access to the structure, which can be the case in renovation applications or if the consultant has scope that facilitates mockups and/or in-progress site verification testing. If the building does not yet exist, it is possible to use FEA to predict future structural responses, but this is less certain than actual field measurements. The Papadimos Group recently presented an approach for predicting mechanical equipment floor vibration using mobility and blocked forces (Young, Wowk, & Solheim, 2025). This level of analysis is typically performed by seasoned vibration experts and may be outside the realm of consultants focused on general building acoustics.

In conclusion, vibration isolation design is often based on the misalignment of equipment driving frequency and amplitude, natural frequency/dynamic stiffness of the isolator, and forced response of the structure. There is some uncertainty in isolator performance and structural response functions; however, the biggest knowledge gap lies in equipment force data. The path forward is to develop test standards, but we need consensus, expert opinions, and manufacturer buy-in on an approach. Have an interest in pushing the state of the art through research or standards? Contact me! Through my roles in both ASHRAE and ASA, I can put you in touch with other interested parties.

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Floating Floor Design in 2026

Wilson Byrick, VP Engineering Services, Pliteq Inc.

Floated floors on discrete isolators, such as rubber mounts, foam pucks, or springs, are common in sound and vibration isolation design. Whether the floating floor is being used for isolation of airborne sound (performing arts), vibration (mechanical and lab equipment) or impacts (fitness and sports) there a few aspects of the design that should be considered.

The traditional method of determining the natural frequency of a spring element is to consider the deflection under load and use Hooke's law. However, in a compressible non-isovolumic elastomer this relationship of load and natural frequency is non-linear, and the material properties must be measured.

If the floated mass layer is installed without sufficient venting, the entrapped air between the isolators will introduce its own stiffness, contributing to the overall natural frequency of the system. If this stiffness is neglected in design, the natural frequency of the system can be significantly higher than predicted, potentially leading to unexpected field performance. This is a critical issue when floated slabs are employed to isolate specific driving frequencies.

This problem is further complicated when a floated floor is not slab-on-grade. When a floated slab is supported by a suspended structural slab, the system no longer behaves as a single-degree-of-freedom (SDOF) system but rather as a two-degree-of-freedom (2DOF) system, due to the dynamic nature of the supporting floor. If the stiffness of the suspended structural slab is not accounted for in the natural frequency analysis, the overall system will not perform as predicted.

Air stiffness and venting radius

When both discrete isolators and an unvented air cavity are present, the air and isolators will act as 'springs in parallel', meaning their stiffnesses must be combined to calculate the overall natural frequency (fn) of the floated floor. This evaluates the single-degree-of-freedom (SDOF) frequency, which can be used in slab-on-grade applications where the structural slab can be considered 'infinitely rigid'.

In 1975, Eric Ungar derived the venting radius criteria based on the thermodynamic behaviour of air. This establishes requirements for the venting area necessary to avoid air stiffness contributions in a floated floor. He states that: "A floated slab may be considered as adequately vented if vents of area $2\pi R_L/h$ for each πR^2_L floor area are provided." To calculate this, using $2h/R_L$ as a percentage of the total floor area will determine the

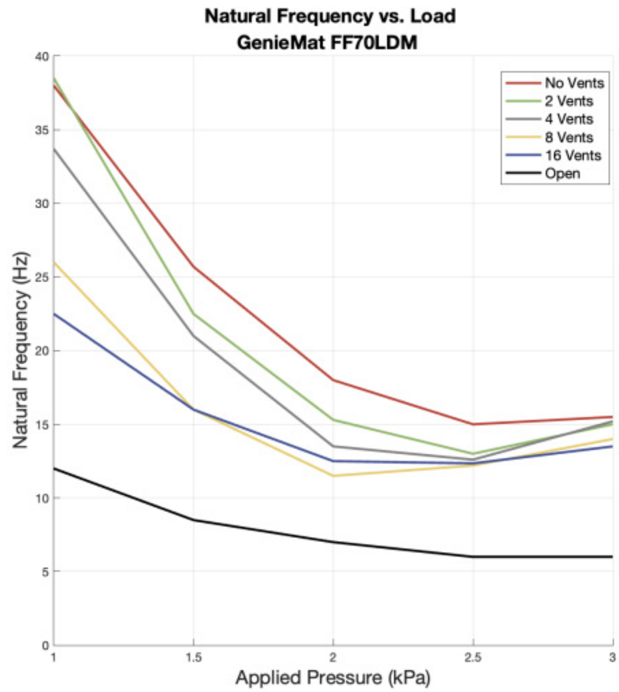


Figure 1

vented area required. What this does not consider, is 'partial venting'. In practice, an installed floating floor likely has some vented area, but perhaps not enough to consider the floor fully vented. This is due to architectural or structural constraints. For example, a rooftop swimming pool or basketball court with discrete isolators would need to be vented through the structural slab, but this would increase any airborne noise transmission through the assembly into the space below.

For the 2025 ICA/ASA conference in New Orleans, we published research into partial venting. A small-scale laboratory mock-up was built using a sealed box and a GenieMat™ FF70LDM panel – a 590 mm x 590 mm panel with 9 pre-adhered isolators and mineral wool insulation. The isolators are 50 mm thick, and the mineral wool is 35 mm thick. The panel was placed in a box to allow a 6.4 mm air gap along the perimeter. A 20 mm concrete panel on top was added to increase the loading and test a more representative load range. This assembly assumed complete venting, as there was sufficient perimeter gap for air to flow. It was placed onto a test frame where a frequency sweep at discrete loading intervals was performed.

To simulate the unvented situation, a plastic sheet was used to cover the top of the box. The sheet was pulled taught and taped to seal the box, ensuring that there were no gaps for air leakage. The same frequency sweep was performed with the unvented assembly. Holes in the plastic sheet along the perimeter were punctured, of 5 mm diameter each, to simulate partial venting. With 2, 4, 8, and 16 vents, a frequency sweep was performed again for each of these conditions at the same loads. This data is shown in the figure 1.

By using this data, we can better predict the effect of partially vented floating floors based on the vented floor area as a percent of total area. Further research showed significantly less change in overall system natural frequency when continuous elastomeric floating floor elements were used.

Considering the second degree of freedom

When floating floors are located on elevated structural slab the dynamic response of the structural slab needs to be considered. The concrete slab needs to be considered as both a mass and a spring and not infinitely more rigid than the single degree of freedom floating floor above it. When the structural slab is considered in this way, the single degree of freedom natural frequency shifts to the left and a new second harmonic appears to the right of the structural slab natural frequency. The amplitude and width of the second harmonic is difficult to predict and varies significantly with the venting condition.

In a paper presented at Inter-Noise in 2019, Zhuang Li from McNeese State University discusses how the use of SDOF models is insufficient for evaluating the natural frequency of floating floor systems on suspended structural slabs. Where a suspended structural slab introduces its own stiffness contribution, a two-degree-of-freedom (2DOF) model is more suitable. Practical examples include rooftop mechanical rooms, rooftop swimming pools, elevator shaft mechanical rooms, and gymnasium floors that incorporate floating slabs. Although 2DOF models themselves are well established in vibration theory, it was Li who derived the theoretical formula for damped 2DOF vibration transmissibility.

Using this model and laboratory measured material properties Pliteq engineers developed a calculator tool which can output two degree of freedom transmissibility data when inputs are provided. Engineers have access to this tool on Echo One™. Input variables include the floating floor load, the selected isolation material, load of the structural floor and the natural frequency of the structural floor.

Predictive modelling vs. field measurements

In May 2025, field tested measurements were collected by engineers from Brown Strachan Associates in Vancouver, Canada. The application was a 70 m^2 gym being installed on the 2nd floor of an existing mixed-use building. Vibration measurements were taken on the top of the slab before and after the installation of both the isolators and the 100 mm concrete topping. Two impact locations and 11 accelerometer locations were chosen. To excite the structure but not cause damage to the structure, a 24 kg weight was dropped on an 8 mm rubber sheet. The concrete topping was installed over the perimeter isolation to the walls, so the air within the system was sealed.

The isolator natural frequency based on simple SDOF analysis and loading was 5.83Hz. When air stiffness and the structural stiffness were included in a full system 2DOF analysis, Echo One™ predicted a low natural frequency of 11.8Hz and high natural frequency of 25.7Hz. The prediction and field testing matched as a resonance at 26Hz was measured onsite. It is imperative that manufacturers and designers consider a more detailed model in the design of floating floors if they want to accurately predict in situ performance.

For more information on the literature or research discussed in the above article please contact Wil Byrick at wbyrick@pliteq.com. ■

The Noise Control Engineering Journal

Volume 74, Issue 2, June – March 2026

If you haven't seen the recent June 2026 issue of NCEJ, check out this table of contents:

A Detective Story Solved by Vibration Analysis

Jim Thompson, JKT Enterprises

Many product problem diagnoses are like detective stories. There can be a lot of clues, and someone must determine how they relate. The search for answers can be difficult and time consuming. This column is about a tire problem from years ago that was a real mystery and finally understood using dynamic analysis.

In the 1990s, I was working for a tire company, and we got a call from an original equipment manufacturer (OEM), auto company, concerned about high pressures in their tires on the dealer lots. Some checks showed pressures over 100 psi (689 kPa). This was alarming since the normal pressure was around 32 psi (221 kPa). In addition, customers were very unhappy with the harsh ride and vibration with such over inflated tires. I was asked to help understand why this was happening.

To understand the problem, you need to know how tires are inflated on a vehicle assembly line. Air is blown past the tire bead on the wheel until the tire is mounted. The tire valve is not used. Instead of taking a few minutes to mount and inflate a tire, this process accomplishes it in a few seconds. There is a video of this process at https://www.youtube.com/watch?v=mXA-Z73X37Q. This video shows a truck tire being mounted, but the case I was working on was a passenger tire.

The assembly plant was having trouble mounting the tires, so they increased the air flow past the bead of the tire. This resulted in the high inflations pressures. This was a high-performance car with the lowest profile tires the company made. In addition, the OEM was seeking outstanding handling performance necessitating a very stiff sidewall. We knew this would make the tire hard to mount, but not this difficult.

The first thing we did was dismount some tires and visually inspect them. We found the bead was twisted and for tires that had been used, the rubber around the bead was worn to the point of exposing the cords. The basic tire construction and how it is mounted on a wheel is shown in Figure 1 to help you understand the situation.

In this case the bead toe was worn away. It was clear that something more than just high pressure was occurring. We then x-rayed a tire mounted on the wheel directly from the OEM. The bead was sitting on the toe and barely in contact with the edge of the wheel rim. It was clear

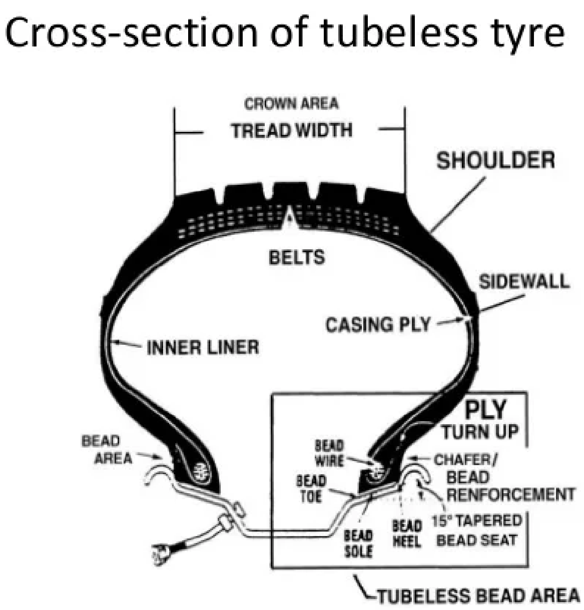


Figure 1 – Tire and Wheel Cross Section

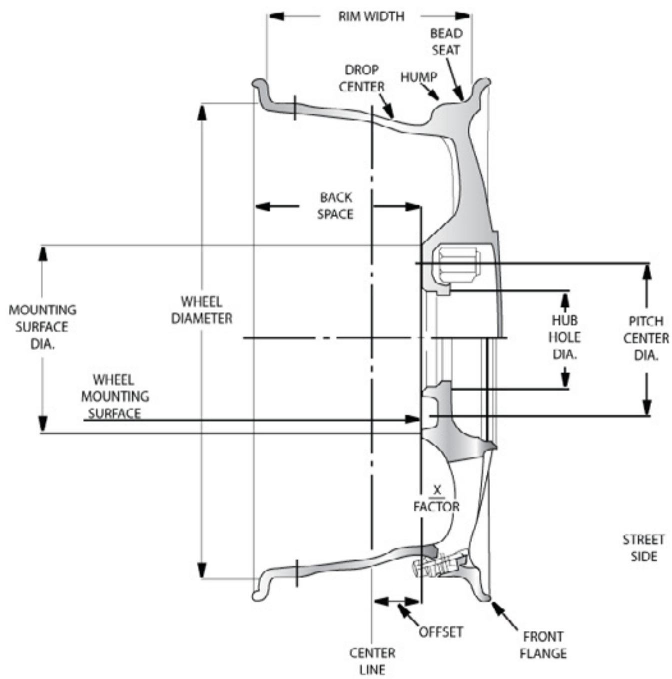


Figure 2 – Wheel Cross Section

the tires were not safe, and this problem had to be fixed. Using regular service mounting the tires were replaced with no difficulty. However, this was only a temporary solution.

Understanding the Problem

There were as many theories about the cause of the problem as there were people involved. My team set out to try to model what was happening during the inflation process. Using FEA with an explicit solver, we attempted to simulate the mounting process in small time steps to see if we could replicate what was happening. This proved to be a difficult task, and once the model was working, the analysts did not believe the results. Sitting with the analysts to examine the results, I saw why they were confused. The model showed the tire from mid-sidewall down to the bead. As it inflated it stalled for a bit getting over the wheel safety hump – please see the wheel diagram in Figure 2. Then there was sufficient pressure to slam the bead into the proper seating position. Following this there was motion in the sidewall and suddenly the bead rotated to sit almost on the toe. This sidewall motion and toe rotation was baffling the analysts. What they were modeling was a wave the went up the sidewall and was reflected backdown the sidewall twisting the bead. The wave was probably always there in a regular mounting situation, but at normal pressures there was not enough energy to overcome the friction of the bead against the wheel – the bead stayed in it original position.

Problem Solution

The immediate question we got when showing these results to the team was what could be done to prevent the sidewall dynamics and the bead twist. We could reduce the sidewall stiffness, but this would adversely affect handling. It was suggested that damping could be added around the bead or in the sidewall. Higher hysteresis rubber around the bead or in the sidewall might increase damping, but again these changes would have adverse effects.

The ultimate solution was much simpler. Based on some investigations in the OEM plant, we found that the lubricant "soap" used in the mounting process was part of the problem. While it worked fine for many tires, in this case with high and prolonged air flows it became tacky and acted to prevent the bead from sliding over the wheel safety hump. To make matters worse, when the problem first occurred the plant decided that adding more "soap" would solve the problem. So, what became an adhesive was even thicker and more uniform on this specific tire mounting machine line. After some experimentation, we found a better "soap" that allowed the bead to slide over the safety hump with less friction. Also, a check of tire pressure was added at the end of the production line.

So, the vibration analysis was not the solution to the problem, but it was part of the detective story. It helped to understand the dynamics of the process. This allowed the team to have confidence that the tire construction or design was not the source of the problem. This allowed us to go back to the mounting process and work with the assembly plant and resolve the problem with their cooperation.

Like many detective stories we had to prove who was not the "murderer" to identify the real culprit. Maybe this was not a classic "who done it". There were no locked rooms or smoking guns. However, there were a lot of people scratching their heads trying to understand what was going on. At the same time there was a very irate customer calling meetings and making phone calls accusing us of providing defective tires. In the end the car was well received and successful. The motoring press raved about the vehicle's handling, not knowing the difficulties we had experienced.

I am sure many of you have interesting detective stories, probably better than this one. We would like to hear them. ■

Specifically focused on all aspects of noise control and acoustics, NOISE-CON 2026 is the premiere conference for professionals and students in this field.

Discover the good vibrations at **NOISE-CON 2026**, in the thriving downtown district of Long Beach, California. Please join us for three days (**July 9–11, 2026**) of educational programming and networking, Connect with noise control leaders and get the tools you need to enhance your professional development.

The conference venue, the Westin Long Beach, is overlooking the marina and tranquility of the Pacific Ocean and very close to popular area attractions including the Aquarium of the Pacific and the Queen Mary. In addition, the hotel/venue is one block from a light rail line connecting to LAX, near two other airports, and is walkable to numerous restaurants, entertainment, and recreational activities — fun for the whole family!

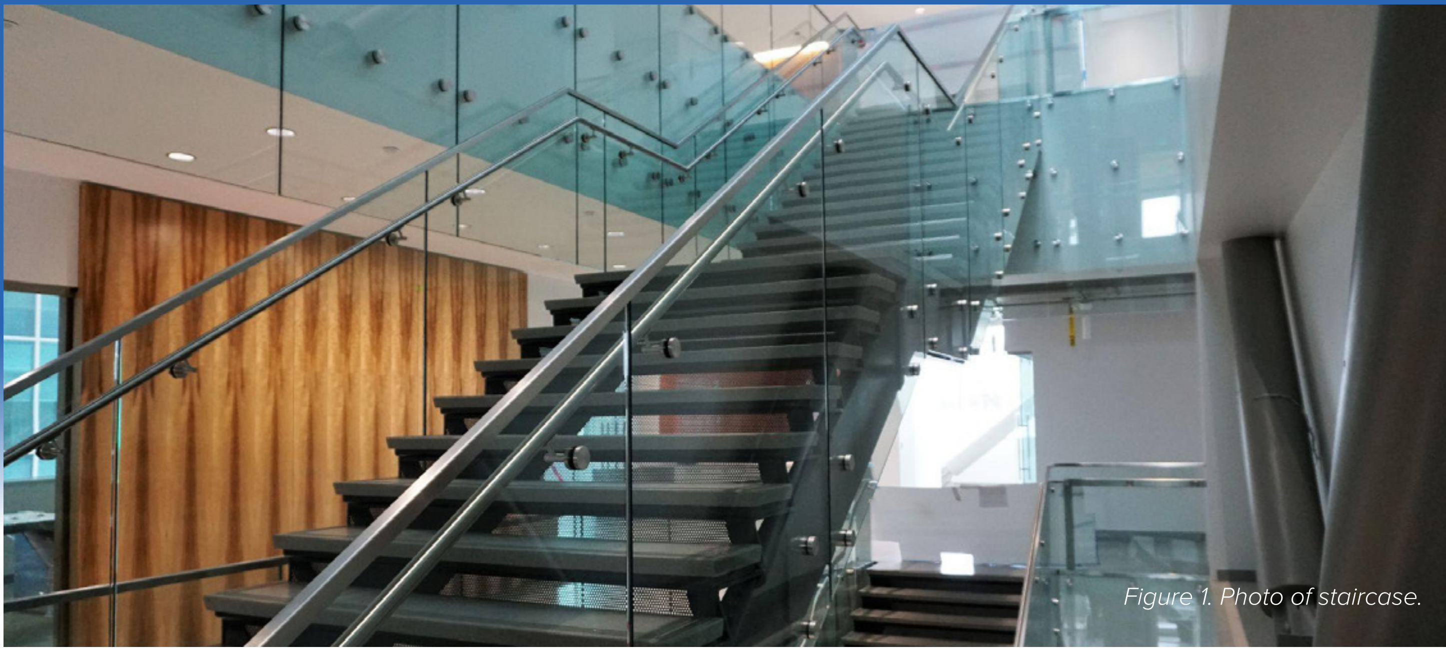


Figure 1. Photo of staircase.

Monumental Staircase Vibration: Finite Element Analysis and Post-Construction Testing

Roman Wowk, Principal, Papadimos Group

Monumental staircases have become a focal point in modern architecture. These three-dimensional sculptures, often associated with an atrium, improve occupant flow between floors and entice people to skip the elevator. Some of the larger or more elaborate staircases even function as gathering zones for casual collaboration, all-hands meetings or to provide spillover space during crowded events.

Presented below is an example of a simple but common monumental staircase configuration consisting of a linear span across an atrium with an intermediate landing. Finite element analysis (FEA) was used to evaluate vertical vibration under walking excitation and develop upgrades for vibration control. After construction was complete, vibration testing was done to verify that the design objectives were achieved and to check the accuracy of the FEA vibration predictions.

Vibration Criteria

ISO Standard 2631-2 defines a base curve for human perception of vertical vibration in terms of Root Mean Square (RMS) acceleration, with values of approximately 0.05% gravity in the 4 through 8 Hz ⅓-octave bands and gradually increasing values outside those bands. That standard also defines multiplying factors to this curve of 4 to 128 for office environments depending on whether the vibration is continuous or transient in nature. Design Guide 11 by the American Institute of Steel Construction specifically addresses vibration due to footfall excitation and recommends a multiplier of 10 to the ISO base curve for offices and a multiplier of 30 for indoor footbridges. The higher limit for indoor footbridges takes into account that people are more tolerant of vibration when moving rather than sitting or standing still.

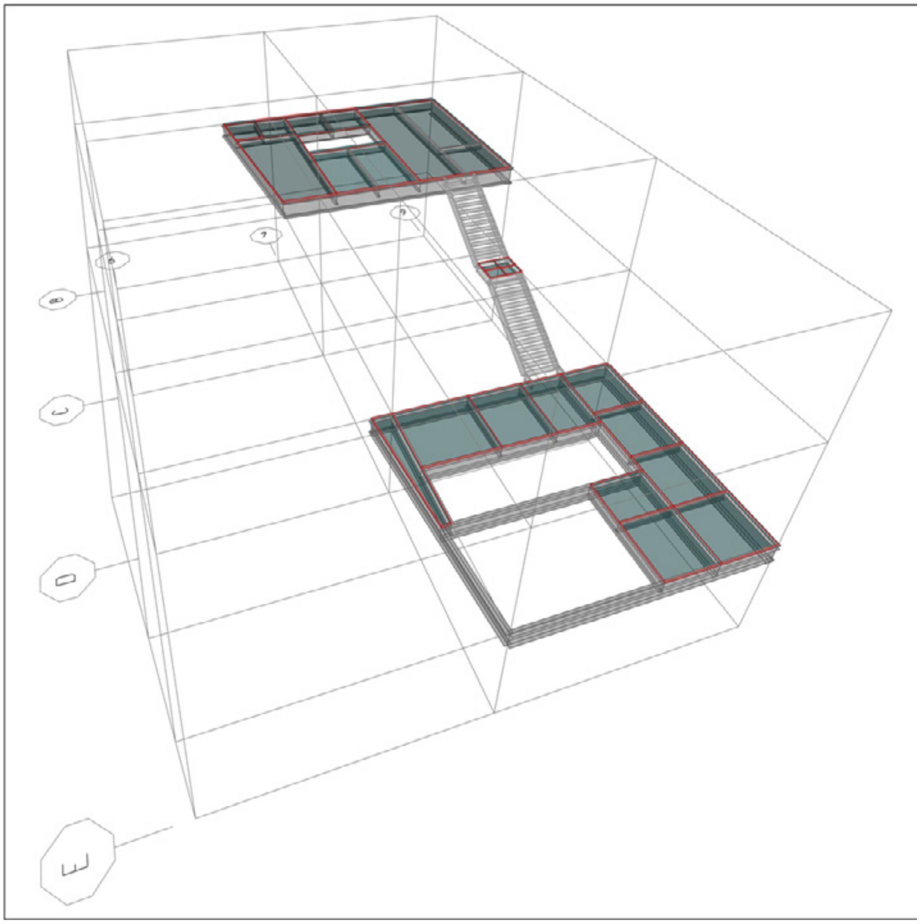


Figure 2. Screenshot of finite element analysis model (using SAP2000).

For the design of this staircase, the office criterion was targeted as a goal while the indoor footbridge criterion was set as an upper limit. The reasoning behind this approach was that people may stop at the intermediate landing for short periods but will otherwise be in motion most of the time. Other larger and more elaborate staircases where people may linger for long periods of time or gather to watch presentations may require more stringent design criteria.

Finite Element Analysis

A finite element model was developed using SAP2000 that included the stair stringers, treads and one structural bay on the top and bottom of the staircase (see Fig. 2). In addition to the basic material properties of steel and concrete, the following assumptions were made regarding the initial structural design:

- **Connections between stringer sections:** translational and rotational degrees of freedom connected ("full-moment" connection).
- **Connections between top of stringers and base structure:** translational degrees of freedom connected but rotational degrees of freedom released ("pinned" connection). Since the actual condition would be somewhat fixed and not truly free rotationally, this is a conservative assumption.
- **Connections between bottom of stringers and base structure:** translational degrees of freedom connected vertically and laterally perpendicular to stringers but all other degrees of freedom released ("slip" connection) between bottom of stringers and base structure.
- **Live loads:** up to 1,000 pounds of live load distributed across the stair span (in addition to the self-weight of the structure) was initially considered. However, since initial tests showed that this did not significantly reduce the frequency of the dominant mode and only improved vibration, zero load was ultimately used as a worst-case assumption.

Time history analysis was then done by appropriately meshing the model and solving for an adequate number of modes to capture vibration up to about 50 Hz with a vertical force function of a walker at the middle of the intermediate landing. This location is an obvious

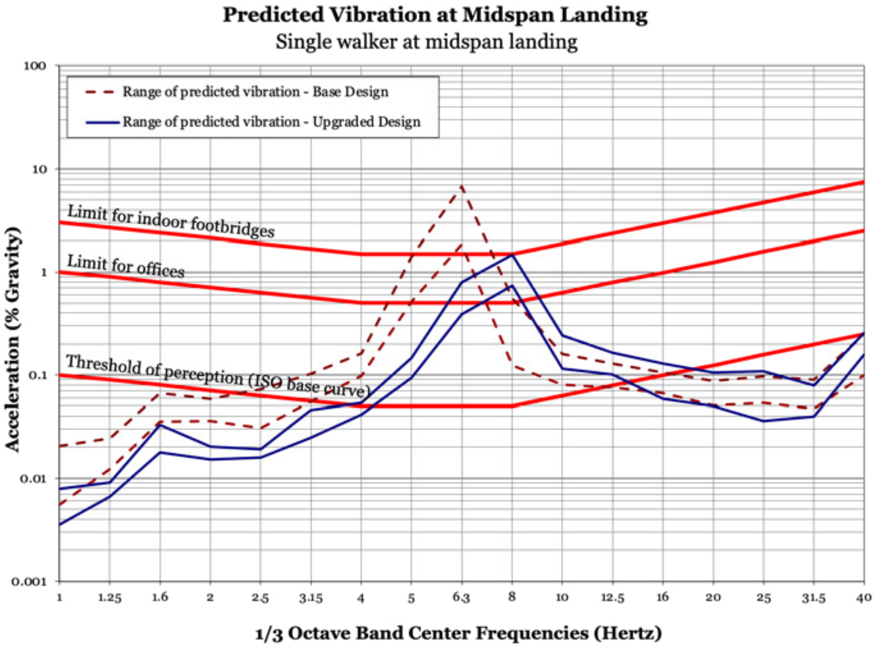


Figure 3. Vibration predictions for initial vs. final design

choice for where vibration would be highest, which was confirmed by reviewing mode shapes and frequencies prior to running time history analysis.

The forcing function used is based on typical brisk walking speeds in the range of 90 to 100 paces/min. However, some margin was also applied to account for real-world conditions such as higher forces when descending stairs rather than walking on a level surface, multiple walkers in-step and potentially higher walking speeds.

Predicted vibration for the base design as described above was up to a factor of five above the desired range (see Fig. 3) and exhibited a vertical dominant mode (i.e. "natural frequency") at 6 Hz. Rather than upgrading the stair stringers, which would increase steel quantities, a full-moment rather than pinned connection at the top of the stringers was considered. This relatively easy design change, which takes advantage of the stiffness of the base structure, increased the natural frequency to 7.3 Hz and reduced predicted vibration to within acceptable design limits (see Fig. 3).

Vibration Testing

After the staircase was completed, vibration was measured by placing an accelerometer at the middle of the intermediate landing (see Fig. 4) while ambient conditions and a variety of footfall inputs were tested. Measured vibration agreed closely with the predictions in terms of both spectral shape and amplitude, and confirmed that the design goals were met (see Fig. 5). Running was the only tested condition that produced vibration above the upper design limit (i.e. for indoor footbridges); however, this is not a regular use case and the exceedance was marginal. Typical walking produced vibration levels close to the lower design limit (i.e. for offices) and a "quick" descent produced vibration levels about halfway between the office and indoor footbridge limit.

Conclusions

This case study illustrates the power of Finite Element Analysis (FEA) in streamlining the design process and adding confidence to design decisions. With proper validation based on testing from previous projects, it can be successfully used to tackle complex problems, develop design options and visually communicate results and aid in the decision making process. Yet it can also be deceivingly wrong. Caution must be exercised to catch and eliminate errors in input assumptions, bugs in models or over-simplifications that completely mask a problematic condition. This is best done by continuously questioning design assumptions and conventional thinking, breaking down problems into their most basic elements and by checking predictions against real-world data.

Prior to relying on FEA for critical design decisions, measured data and test models of similar structures should be used to understand the degree of uncertainty in the predictions and to understand which input assumptions most contribute to errors. The lessons learned and refinements gained from that process can then inform design decisions that offer significant cost benefits and/or performance commitments. On this particular project, the sizing of the stair stringers was minimized and the architectural design intent was preserved through confidence in properly-validated FEA predictions. Similar cost savings and architectural innovation are possible on a wide array of projects that face potential issues from footfall excitation and other vibration sources.

Acknowledgements

Thank you to the entire Papadimos Group team for supporting this article and building our vibration practice by asking tough questions, running towards challenges, "earning the data" with difficult field measurement assignments, and bringing unique perspectives to each project. ■

Figure 4. Vibration testing with accelerometer at intermediate landing.

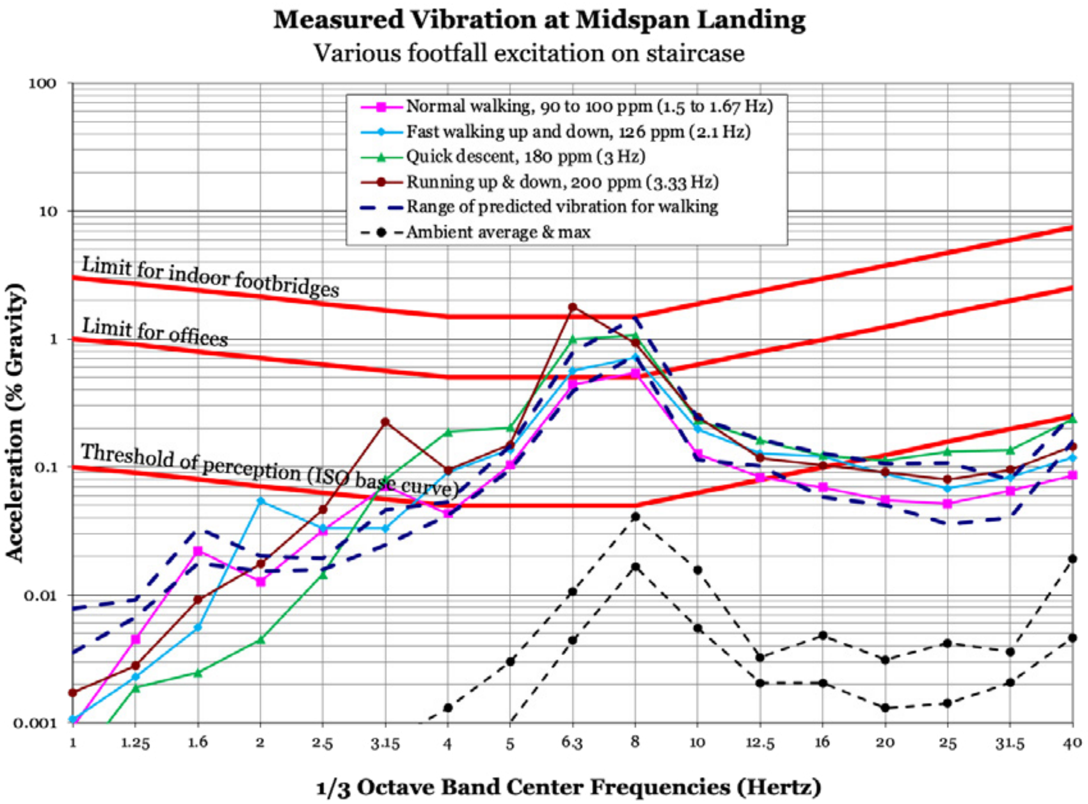


Figure 5. Measured vs. predicted vibration for final design

Predict, Don’t Repair – Prediction-Based Noise Control in Industrial Halls

Alfons Geltinger, M.Eng., Senior Software Consultant, DataKustik GmbH

Introduction

In many industrial sectors (e.g., bottling/packaging, metalworking, woodworking, logistics), noise at machine workstations often reaches critical ranges. In many of these applications, sound is generated by mechanical excitation and vibration-related radiation, for example when bottles collide and vibrate in bottling lines or when machine surfaces and moving components radiate sound. Besides direct sound from dominant sources, industrial halls typically exhibit a highly reverberant, partly diffuse sound field. Without absorption, elevated and relatively uniform levels may develop across large areas, with additional hot spots close to individual machines. Noise control addresses different parts of this problem: baffle ceilings/wall absorbers reduce the diffuse component, enclosures reduce source group emission, and barriers provide local shielding at workstations. However, the most effective combination and the resulting change in workplace exposure cannot usually be derived from measurements alone. Prediction-based methods enable systematic variant comparisons in design and retrofit (“Predict, don’t repair”).

Workplace Noise Targets

Across countries, occupational noise control aims to assess and document exposure and derive effective measures. While limits and action levels differ by region, decisions are typically based on time-averaged exposure (e.g., 8 h) and the consideration of peak/impulsive events. Table 1 provides widely used reference values (EU/USA) for orientation.

Why Prediction Matters

Baseline measurements are indispensable, but they only describe the current situation. Without prediction, it often remains unclear which measure will actually be effective—frequently leading in practice to over-dimensioned solutions, uncertainty about the achievable reduction in workplace exposure, and retrofit concepts that can only be implemented to a limited extent due to structural constraints. Especially in existing plants, measures may no longer be feasible or only at considerable effort, for example when baffle ceilings cannot be integrated because of existing structures, pipework/cable routing, or crane runways, making costly

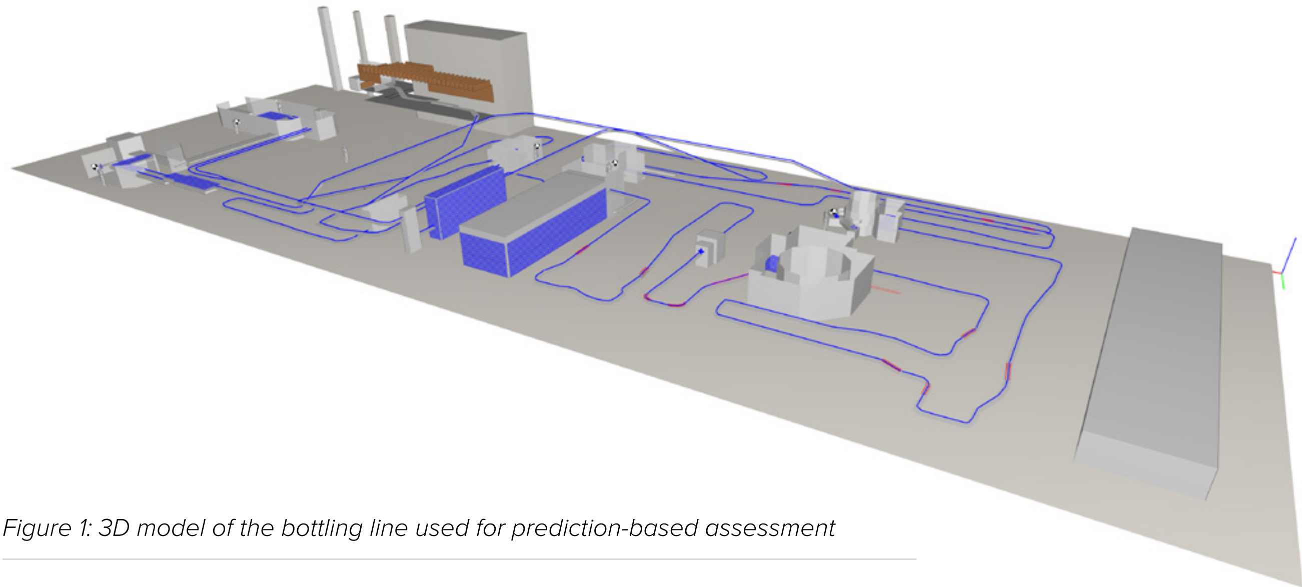


Figure 1: 3D model of the bottling line used for prediction-based assessment

modifications necessary. Prediction provides planning certainty, particularly for **(1)** planning new plants or layout changes and **(2)** optimising existing plants through systematic variant comparison—supported by robust input data and calibrated machine/source group models for reproducible assessments.

For the prediction calculations, CadnaR [4] was used as a software tool quality-assured for this application according to DIN 38457:2025-05 [5].

Case Study 1: Planning a Bottling Line

Prediction supports early identification of noise-intensive areas, workstation placement, and variant comparison before investments are made. In bottling lines, such noise-intensive areas can result from mechanically excited product flows, for example where bottles collide, accumulate, or are guided through regulating sections. In the example shown, a **baffle ceiling** is evaluated using baseline, variant and difference maps. Evaluation metric: The predictions are evaluated using A-weighted workplace sound pressure levels at selected operator positions and grid-based level maps to visualise the spatial distribution and the effect of the measure.

Case Study 2: Retrofitting a Cleaning Station

In existing plants, noise control is often driven by measured exceedances at specific workstations. While measurements are essential to characterise the current situation, they do not directly indicate which retrofit measure will be most effective once installed. Prediction enables a targeted variant comparison under real geometric and operational constraints, helping to avoid over-dimensioned solutions and costly rework. In the example presented here, a local noise barrier at a cleaning machine is used to modify radiation and provide shielding at nearby operator positions. Results are reported as A-weighted workplace sound pressure levels (LpA) at the operator position.

Table 1: Occupational noise exposure reference thresholds (EU/USA)

Region / standard	Reference / criterion	Permissible exposure (8 h)	Peak / impulse
USA (OSHA 29 CFR 1910.95) [1]	85 dB(A) TWA* (Action Level for Hearing Conservation)	90 dB(A) TWA (PEL**)	140 dB peak (impulse/ impact noise should not be exceeded)
USA (NIOSH recommendation) [2]	85 dB(A) TWA (REL***)	85 dB(A) TWA (REL, 3-dB exchange rate)	140 dB peak (ceiling recommendation)
EU (Directive 2003/10/EC) [3]	80 dB(A) (Lower Exposure Action Value) / 85 dB(A) (Upper Exposure Action Value)	87 dB(A) (Exposure Limit Value, taking hearing protection attenuation into account)	135 / 137 / 140 dB(C) peak (Lower/Upper Action Value / Limit Value)

* TWA = Time Weighted Average; ** PEL = Permissible Exposure Limit; *** REL = Recommended Exposure Limit

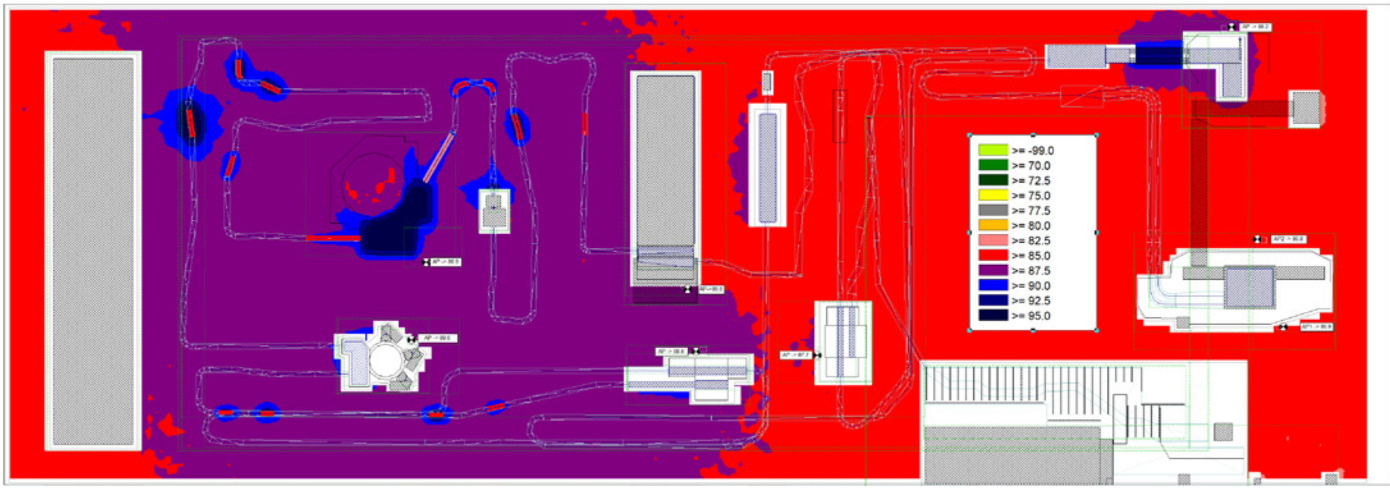


Figure 2: Baseline A-weighted level map of the bottling line (no acoustic treatment), highlighting high-level zones and local hot spots near dominant machines.

Figure 3: Variant A-weighted level map with baffle ceiling, showing reduced levels over larger areas due to a lower diffuse sound field component.

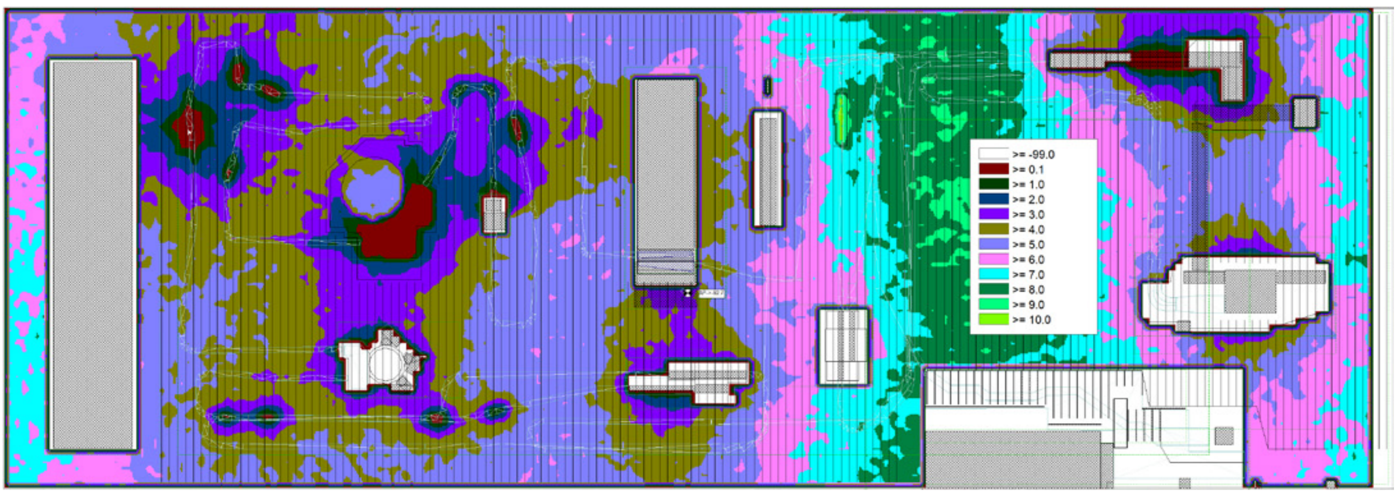


Figure 4: Difference map (baseline – baffle ceiling variant), visualising the spatial effect of the ceiling treatment (ΔLp, grid arithmetic).

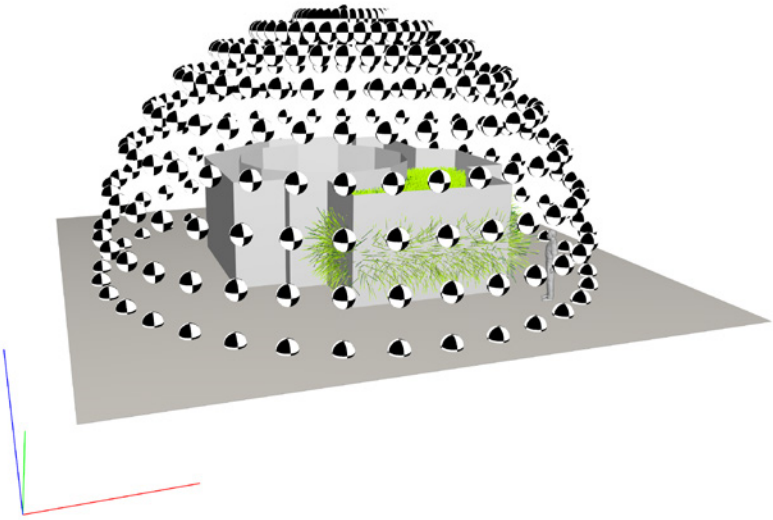


Figure 5: Enveloping surface around a filling machine for determining the sound power radiated by the source group after source modelling and calibration (in combination with free-field simulation) [6].

Noise barrier at the cleaning machine (sliding barrier open vs. closed)

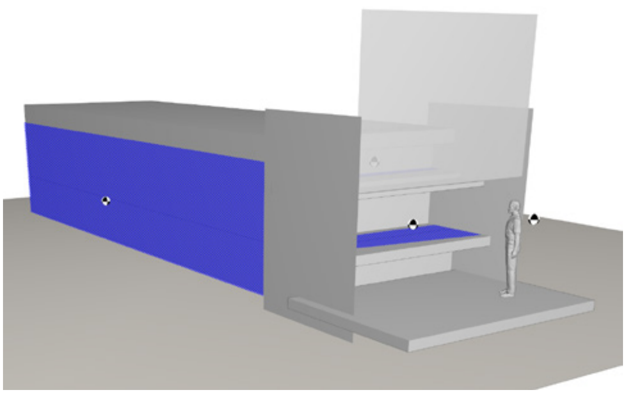
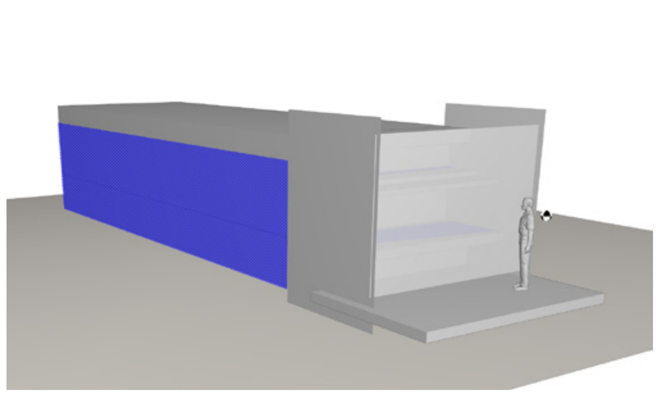
Near-field measurements were used to calibrate a source model of the cleaning machine. A local barrier was evaluated in a free-field setup and then assessed in the full plant model (including room response and other sources). The barrier is open during manual intervention when bottles are aligned into the cleaning machine infeed lanes.

In addition to barriers, which mainly influence the directionality of the radiated sound energy, complete enclosures can also be modelled, including their sound insulation performance (transmission loss) and the absorption inside the enclosure. In this case, the measure does not only change the directivity: the enclosure material and internal lining also absorb sound energy, further reducing the emitted noise.

Input Data and Source Modelling

Depending on machine size, geometry, and radiation characteristics, sound sources should be represented using different levels of detail to capture both emitted sound power and directivity. Typical input data, such as sound power level, workstation sound pressure levels (LpA) and geometric information (including relevant material properties such as absorption and transmission), are obtained from manufacturer data, literature sources, and measurements in existing plants. Because true free-field conditions rarely exist around installed machines, near-field measurements combined with free-field simulation can support practical source modelling and calibration (Fig. 5).

Table 2: Case Study 2 – barrier open vs. closed (free-field and in-plant). In-plant reduction at operator position: >5 dB.

Barrier open	Barrier closed
	
Free-field simulation: LpA = 90.3 dB(A) In-plant simulation: LpA = 91.0 dB(A)	Free-field simulation: LpA = 82.1 dB(A) In-plant simulation: LpA = 85.6 dB(A)

Ideally, machine suppliers provide an acoustic model of the equipment, including sources and geometry. Such detailed source models can be created and stored as separate files and then imported directly into prediction models of complete industrial halls.

Conclusion

Measurements are essential to document the existing situation in industrial halls, but they often cannot reliably predict how effective a retrofit measure will be once installed under real geometric and operational constraints. Prediction-based methods close this gap by enabling transparent, reproducible variant comparisons—both in early design and in existing plants—helping to avoid over-dimensioned solutions and costly rework. The examples shown illustrate how typical measures such as baffle ceilings, local barriers, and enclosures can be evaluated not only locally at workstations, but across the entire facility. In this sense, "Predict, Don't Repair" makes noise control plannable and verifiable, turning the acoustic effects of vibrating machinery and industrial processes into a measurable, design-driven outcome and supporting better, safer and truly "good vibrations" at the workplace.

Literature:

[1] OSHA, *29 CFR 1910.95 Occupational noise exposure* (U.S. Department of Labor).

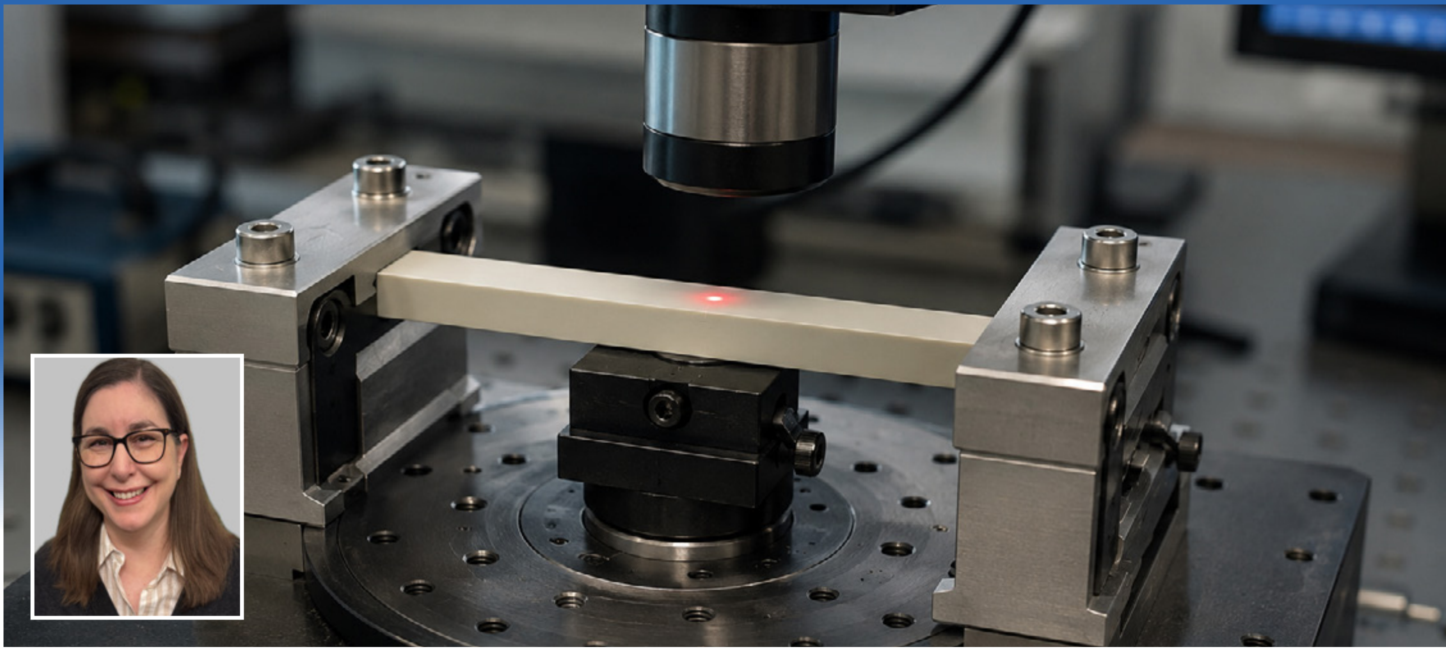
[2] NIOSH Pub. 98-126

[3] EU, *Directive 2003/10/EC* (Noise at work), 2003.

[4] DataKustik – CadnaR – Industry Acoustics (accessed: 12 May 2026)

[5] DIN 38457:2025-05; Acoustics - Software for the calculation for workspaces - Quality assurance of software-implemented methods

[6] DIN EN ISO 3744:2011-02; Acoustics - Determination of sound power levels and sound energy levels of noise sources using sound pressure - Engineering methods for an essentially free field over a reflecting plane. ■



Quantifying Vibration Variability in Additively Manufactured Materials

Christina J. Naify, Applied Research Laboratories, University of Texas at Austin

The rapid growth of research in architected acoustic materials has been enabled by a contemporary growth in additive manufacturing (AM) advancement and accessibility. Extensive research over the last two decades has demonstrated the ability to tailor sub-wavelength structure and control wave phenomena with exotic properties such as stop bands, wave focusing and steering, and non-reciprocity. However, despite the large number of studies conducted which exploit additive processes to produce complex designs, only a handful of studies account for manufacturing variability on the acoustic or elastic response of AM components [Katch 2023; Qin 2025]. The majority of structures studied in the literature include experimental data from a single sample or small set of samples for a given geometry. While the layer-by-layer nature of additive processes is known to lead to variability in properties depending on print settings, sample orientation, or sample size the information supplied by base material vendors does not provide adequate bounds for material properties.

The impact of limited study on manufacturing processes is uncertainty in as-printed material elastic properties and material losses in the design process. In order to reliably and repeatably utilize AM for acoustic and vibration applications it is increasingly important to understand the effect of print-to-print variation of samples and to have accurate material properties as an input to design models. This article highlights recent efforts to systematically investigate variability resulting from the print process on vibration response. Specifically, two related efforts quantified material properties of printed components across a range of printers and isolated influence of the variability of different printers compared to print settings.

Interlaboratory study and sample characterization

Interlaboratory studies are widely used to quantify and isolate the effects of, for example manufacturing [Fusaro 2023; Zieliński 2020], modeling or test processes [Horoshenkov 2007]. In this study, variation in the elastic response of fused deposition modeling (FDM)-printed materials was quantified using a non-destructive method of vibration analysis of slender beams. Test samples were printed on different printers at different research

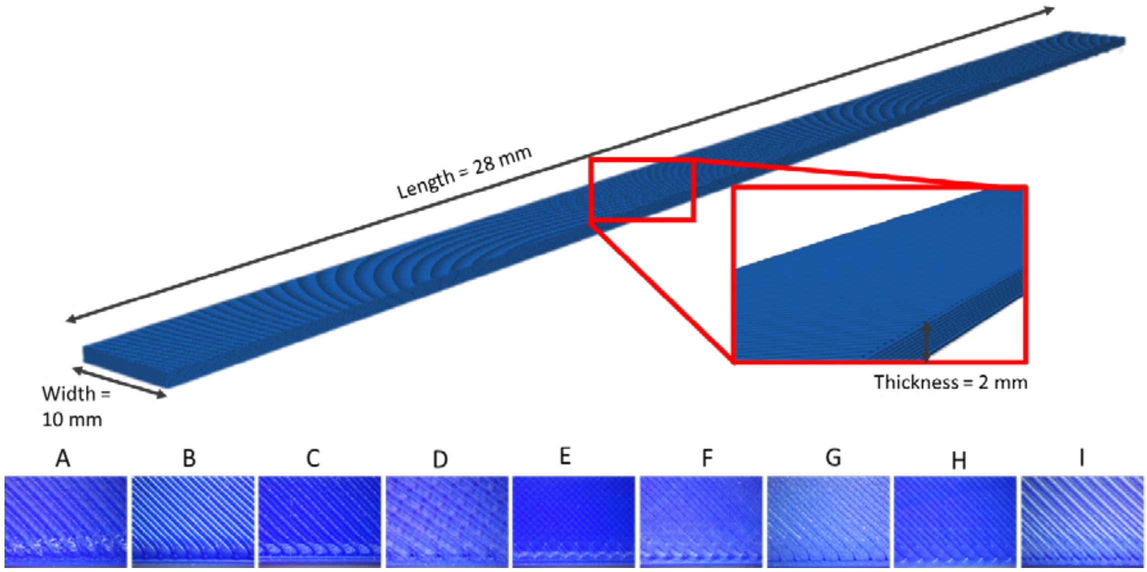


Figure 1: (Top) Slender beams were used as the test specimen of an interlaboratory study. (Bottom)An initial study collected 3 samples from 9 printers provided by 6 institutions. Each institution was instructed to use a common geometry file and optimal settings for their printer. Photographs of the tops of the samples provide qualitative comparison of the different samples. Complex moduli were extracted using experimental data and an inversion algorithm and showed variability of up to 20 percent in moduli for this study.

institutions and tested at a single location in order to isolate variability of the printer rather than the test process. A uniform geometry file, select printer settings, and base material were utilized in order to exploit any printer-based differences. The chosen geometry for the study was selected to meet key criteria. First, the design should be easily printable using the method of the study, in this case FDM. The dimensions of the part should enable printing on a wide range of printers in order to fabricate the study test articles on as many printers as possible within the participating institutions. An example of a test article used the study was a slender beam with embedded asymmetric resonator located at the beam's midpoint.

Planning for the study began with discussions among study participants to ensure that each participant had at least one printer able to produce the study samples and slicer settings capable of accommodating the required print parameters. Factors such as printer bed size, material processing, and customizable settings were discussed. It was decided that the sample material to be evaluated in this study would be a commercially available polylactic acid (PLA) blend. Each participant was provided with a common standard triangle language (STL) file generated from a 3D model of the part created in SolidWorks and were encouraged to use whichever printers they had available to print three copies of the sample. All samples were collected at a test location and evaluated using a process involving mounting the slender beams in a vibration shaker and measuring frequency-dependent velocity using a laser vibrometer. Uniform boundary conditions were enforced by clamping the ends of the beams with a set value torque wrench. The direction of excitation of the beams is in the smallest dimension marked thickness in Fig 1.

Interlaboratory study results

Results of the interlaboratory study were compiled from sample sets printed on nine individual printers from six institutions [Naify 2024]. The dimensions and mass of all samples was carefully measuring using digital calipers and a digital balance respectively. Variation in length, width, and thickness was within 1, 5, and 20 percent respectively relative to the as-designed values. Variation in mass was found to vary by 20 percent. Density was calculated for each sample from measured mass and volume and was within 10 percent of the value provided by the manufacturer of 1.2 g/cc. Most of the samples had average densities lower than the expected value.

The filament manufacturer also provides a single value for Young's modulus which was 3.07 GPa, though no information is provided about print settings for that

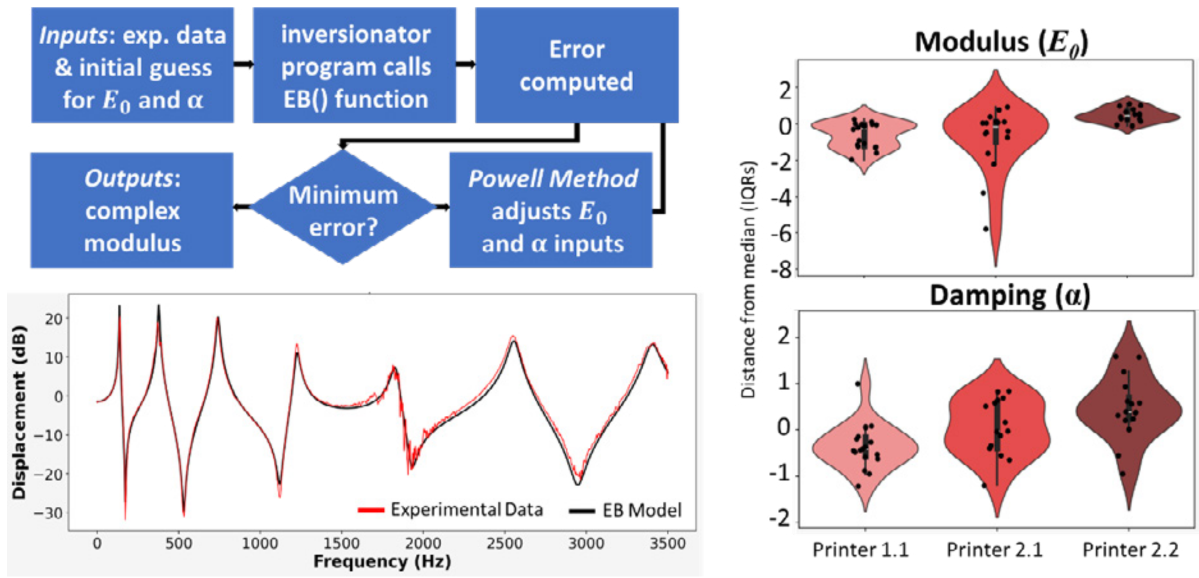


Figure 2: (Left) An inversion algorithm which treated the beams as Euler Bernoulli (EB) beams was used to identify material properties for each experimentally measured data set (bottom left). Moduli and damping values collected from a second set of samples indicated that outliers and a bimodal distribution can further increase the bounds on expected material properties and that both printer and print settings impact results significantly.

value. Determination of material properties from the vibration response of the slender beams utilized an inversion process based on treating the beams as Euler-Bernoulli (EB) beams. The EB inversion process utilized the mathematical equation for vibration of a slender beam to fit to identify best-fit complex Young's modulus to experimental data via a minimization algorithm. A flow chart of the inversion process is shown in Fig. 2 above a snapshot of measured frequency response function and the EB predicted best-fit for that data set. Resulting modulii ranged from 2.4 GPa to 3.2 GPa and loss factors ranged from .032 to .065. The sample-to-sample variation within each printer set had a maximum value of around 0.1 GPa showing good repeatability from each printer.

Expanded study results

While the interlaboratory study provide useful metrics for printer-to-printer variation across a large set of printers, the small number of samples from each printer and variation in settings used at each institution resulted in lingering questions about both the sample-to-sample variation and the effects of print settings on properties. To address these shortcomings a follow-on study was conducted which involved a larger data set of 20 samples per printer and controlled settings across printers. A control data set was printed using manufacturer-recommended settings on an FDM printer (settings on Printer 1.1). The second sample set was printed on a separate printer using the same printer settings (settings on Printer 2.1). Finally, a sample set was printed using different settings on the second printer (settings on Printer 2.2). In all cases, material from a single spool of base material was used.

Violin plots of the data show the distribution of dimensions and mass as shown in Fig. 3. The violin plots are distributed to scale the variables across each dataset for fair comparisons. Normalization was set relative to the interquartile range (IQR) where IQR is the range between the 25th and 75th percentiles, to help reduce the influence of outliers. All normalized values represent the distance (in IQRs) from the median of 0. Following the inversion process used on the interlaboratory study, complex moduli for each data set were calculated as shown in Fig 2 (right).

Similar distributions of values were found for the length and width of samples from all three data sets examined. For measured thickness, which is the driving dimension for the vibration of thin beams examined in this study, the set from Printer 2.2 showed a significantly tighter distribution than the other two sets. The implication of collecting a larger data set is highlighted in the occurrence of outlier values, especially notable in the thickness values from Set 2.1. Similarly, the mass distributions revealed a bimodal behavior and samples were printed in two groups.

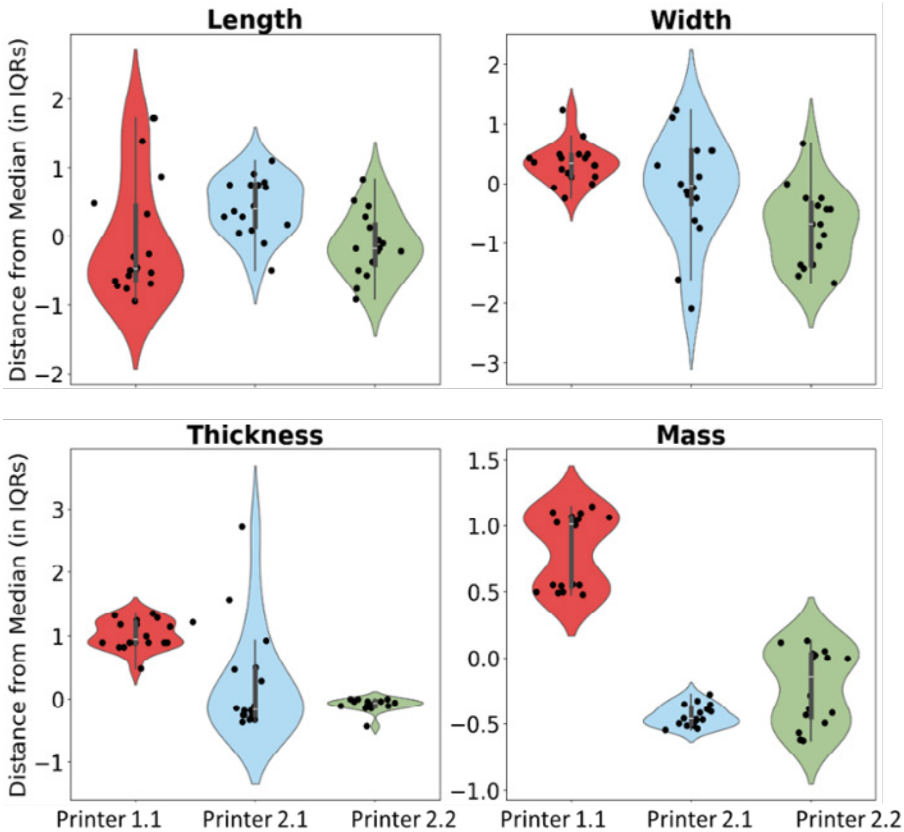


Figure 3: A follow-on study addressed relatively small sample size in the initial study which left lingering questions about sample-to-sample variability. Digital calipers were used to measure dimensions of sets of 20 samples printed on two printers using two sets of settings.

Conclusions

The studies described here provide insight into variability in the vibration response of printed structures across printers, across printer settings, and from sample-to-sample within a single printer and setting set. Key takeaways from the study include the need to assume variability of around 20 percent in material properties in component design and the need to assess large enough data sets to provide further bounds on expected response.

Acknowledgements

Thank you to Nathan Geib, Vinh Pham and Gael Nuno who contributed to the data collection in this article.

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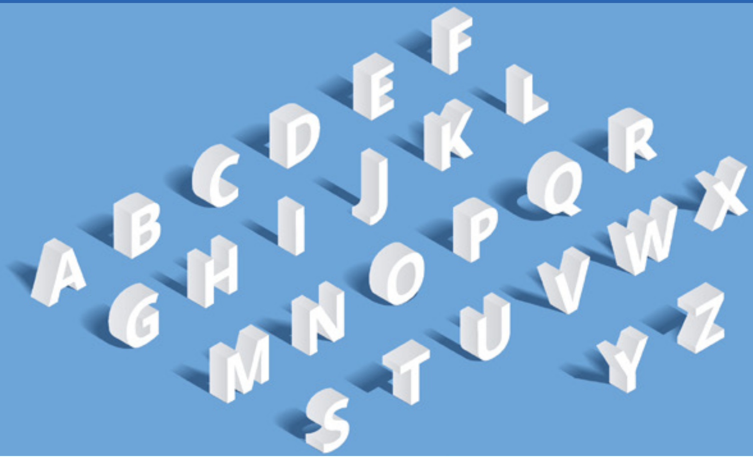
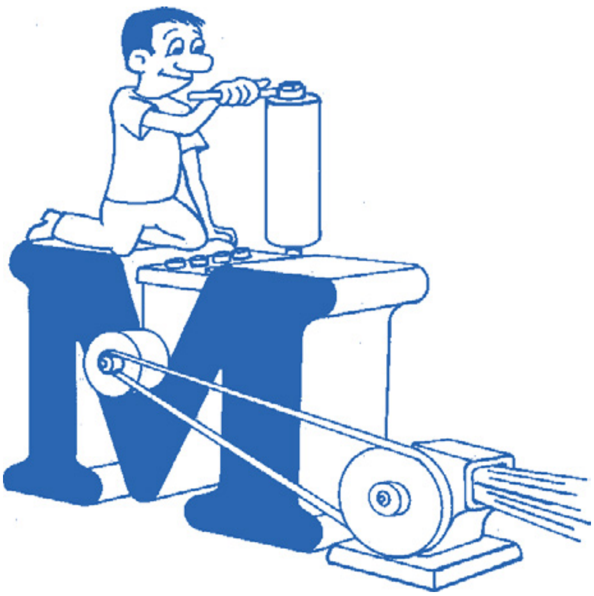


Editor's Note: We are republishing Eric Ungar's Acoustics from A to Z and Stig Ingemansson's Noise Control: Principles and Practice from previous NNI issues as part of our initiative to include more educational articles. Their lessons are just as valid today. Look for more in future issues and on noisenewsinternational.net.

Acoustics from A to Z

Eric E. Ungar



MACHINERY of any kind
Brings two main types of noise to mind.
One's from surface radiation
Due to structural vibration.
The other comes from pressured air
As air is moved from here to there.

Intake and exhaust noise usually predominates in engines or compressors, because intakes and exhausts are acoustic monopole sources that radiate sound efficiently. What is left when intakes and exhausts are quieted often is called "casing noise" – that is, noise radiated from the machine's structural envelope as the result of its vibrations. In machinery whose internal components do not communicate directly with the ambient air, casing noise is all there is.

Noise-radiating casing vibrations may result, for example, from internal pressure pulses, from hydraulic systems, from imbalance of rotating parts, from reciprocating elements, and from impacts and other interactions of mechanical components, such as those of bearings and gears. The latter have some particularly interesting aspects.

At a small lunchtime conference some time ago, one of my colleagues pulled a ball taken from a ball bearing from his pocket, rolled it across the table and asked why this shiny, smooth ball should make so much broadband noise as it rolls across a polished wood surface. Try it; you'll be surprised how noisy it is! When we later repeated the same experiment on a flat glass mirror, we again observed considerable noise. I don't know whether any research has been done on this problem, but I conjecture that the tiny asperities on the surfaces interact, possibly producing local surface deformations and causing the surfaces to vibrate and thus to radiate sound. We didn't try lubricated balls or balls with resilient covers, but I bet these would have produced a lot less noise.

Combustion noise, which is responsible for the roar of furnaces and the "core noise" of jet engines, is due to nonuniform combustion, where there in effect occur local hot spots that behave as acoustic monopoles and thus radiate sound well. Temperature and density inhomogeneities behave as dipoles when accelerated in non-uniform flow. Some flow-related acoustic phenomena involve feedback, such as those associated with edge tones and also some whistles. There also may occur thermal-acoustic feedback phenomena exemplified by the Rijke tube, which first was reported in 1859. As Lord Rayleigh describes it: "When a piece of fine metallic gauze, stretched across the lower part of a tube open at both ends and held vertically, is heated by a gas flame placed under it, a sound of considerable power and lasting for several seconds is observed almost immediately after removal of the flame." As he goes on to explain, the air column in the tube is driven at resonance by periodic transfer of heat from the gauze to the air, with appropriate phasing resulting from the combination of convection with the acoustic pressure oscillations. This phenomenon differs from that of "singing flames," in which acoustic pressures interact with the combustion process.

And, why do transformers make noise, where these have no intakes or exhausts, nor internal moving parts? The answer is magnetostriction – slight changes in the dimensions of iron or steel components resulting from changes in the magnetic fields acting on these components. Practical and economic constraints make it difficult to reduce the noise produced by large power transformers at the source. But I'm glad that power companies have not let this get in the way of providing our homes with electricity; otherwise, we would have to watch TV in the dark.

A noisy NOISE annoys an oyster
And quiet noise annoys a cloister.
Annoyance from a sound, it's true
Depends on what one wants to do.
Shaped noise can be used to mask
A sound that complicates a task.

I must admit that I have no idea whether oysters can perceive any sound at all; I was just carried away by the cadence of the words. But I do remember hearing a paper concerned with sound perception by fleas that was presented at an Animal Bioacoustics session of the Acoustical Society of America a few years ago. At that time ultrasonic flea collars for dogs were being advertised aggressively and a study was carried out to determine their efficacy. This study, which was not sponsored by manufacturers of flea collars, found that: (1) fleas cannot perceive sound; (2) ultrasound emitted by the flea collar would be blocked and absorbed by the dog's fur so that little sound would reach any fleas; and (3) in a comparison investigation, dogs wearing ultrasonic flea collars harbored a somewhat greater number of fleas than dogs without such collars. I don't recall whether anyone concluded that dogs might be driven to distraction by the ultrasound.

According to a paper presented by Douglas Barret of HMMH at the 1999 Summer Meeting of the Transportation Research Board, nuns objected to construction of a highway near their convent, insisting that quiet and serenity were essential to their work. They protested, even though the noise at the site was predicted to increase by a mere 10 dBA above the present 45 dBA. They may not have realized how valid their objections were. Highway noise levels typically are stated in terms of the energy-average levels observed during a day's loudest one hour period – and changes in this noise level clearly do not account for the greater interruption of the evening quiet by short-duration loud noise intrusions from passing trucks.

Quite a different situation exists in the "Land of the Rising Decibels," as described in a recent newspaper article. According to this article, the "Japanese are subjected to a variety of clatter that is perhaps unlike anywhere else in the world." Not only do their vending and ATM machines talk to customers with electronic voices and escalators tell them to watch your step, but there also are demonstrators with bullhorns everywhere. Even in rural towns one can hardly escape from the ubiquitous public address systems which spew forth messages at all hours of the day and night. Some public address proclamations quoted in the article include, "Children, go home, it's getting dark." "Don't use too much water, it hasn't rained in recent days." "Make sure the stove is off before you go to bed." On trains, passengers are instructed to turn off their cell phones, with announcements that are much louder and more annoying than the telephones themselves.

Although there is much quiet objection (pun intended) to this noise pollution, a citizens' group organized about a decade ago to fight this pollution reportedly has had little success, largely because some of their cultural attitudes prevent the Japanese from expressing their discontent publicly.

We've all heard that one person's music is another person's noise. But quiet may not be the optimum situation and what is noise to one person may be music to another. I've been in noisy offices of plant managers who were happy to hear the production machinery; they felt that they were making money as long as everything was running, and quiet was an indication of trouble. Some plant personnel could even identify problems from changes in the noise they heard in their offices and they objected strongly to any proposal to give them more quiet.

In a recent survey of workers in cubicles in open-plan offices, about 70% reported that noise was the number one distraction, with conversational noise and the lack of acoustical privacy as the leading cause of acoustical dissatisfaction and stress. The most practical solution here consists of making more noise – adding a 'masking' noise to reduce the information content of the total noise perceived by a listener. The installation of sound masking systems has become more prevalent in recent years and studies have shown that use of such systems has resulted in significant productivity improvements. ■

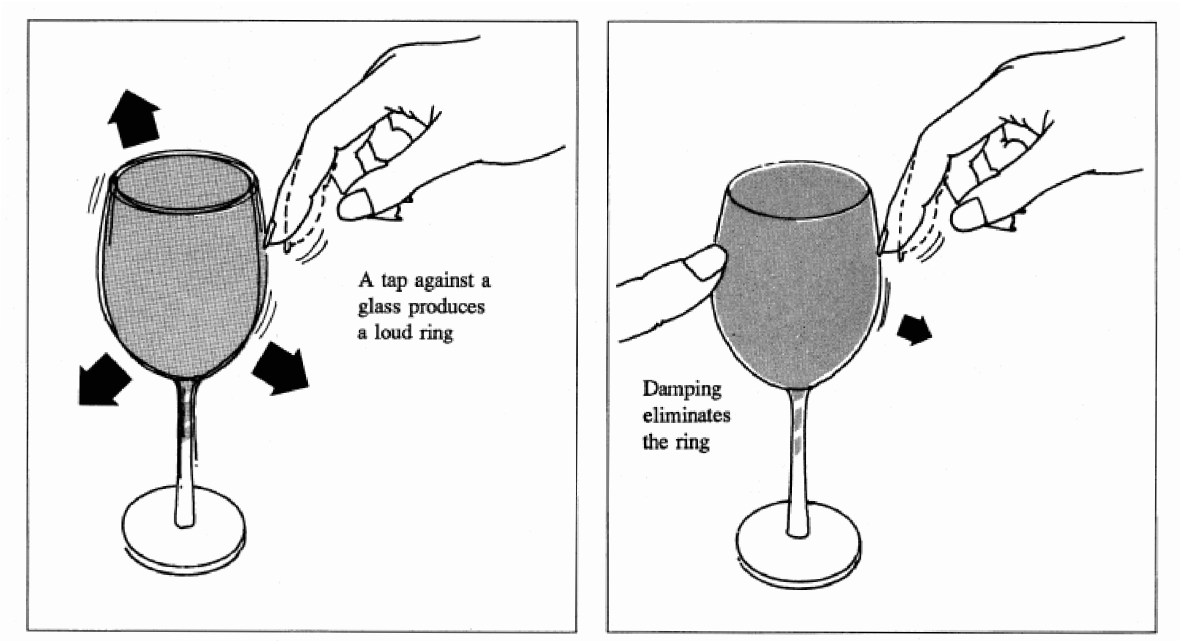
Stig Ingamansson's Noise Control: Principles and Practice

B7 – Sound from Vibrating Plates – Resonance

RESONANCE AMPLIFIES NOISE BUT IT CAN BE DAMPED

Resonance greatly increases noise from a vibrating plate, but it can be suppressed or prevented by damping the plate. It may often be sufficient to damp only part of the surface, and, in some rare cases, damping of a single point is effective.

Principle

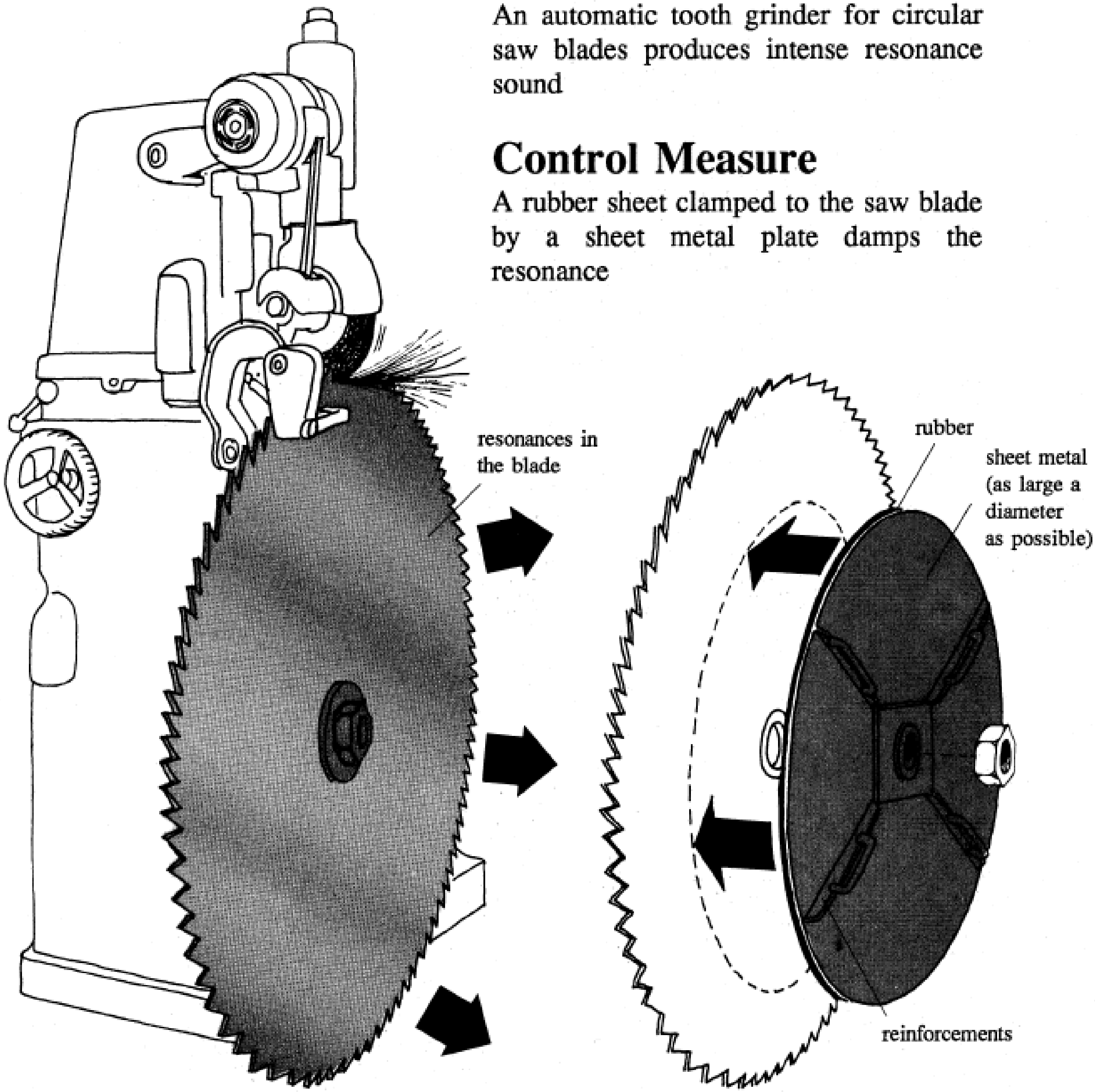


Example

An automatic tooth grinder for circular saw blades produces intense resonance sound

Control Measure

A rubber sheet clamped to the saw blade by a sheet metal plate damps the resonance

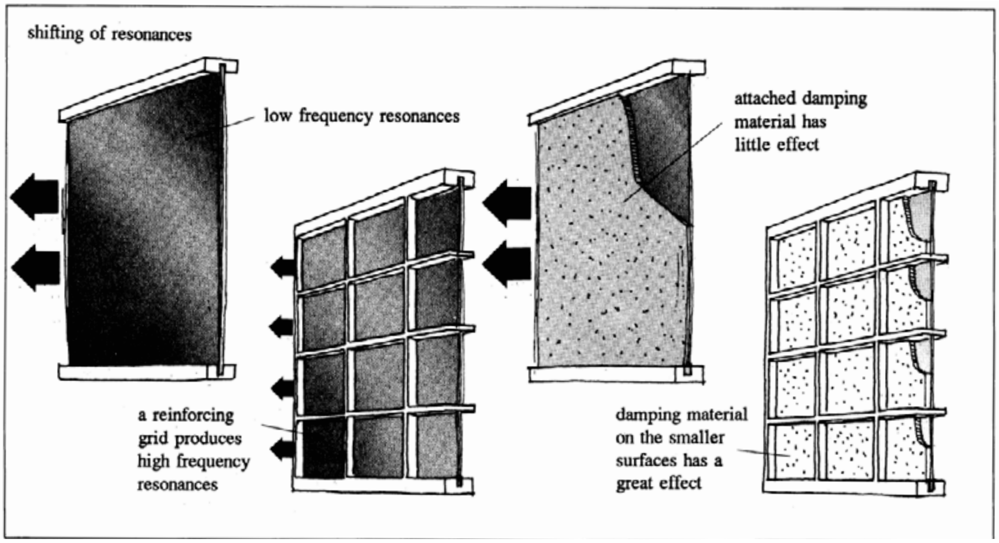


B8 – Sound from Vibrating Plates – Resonance

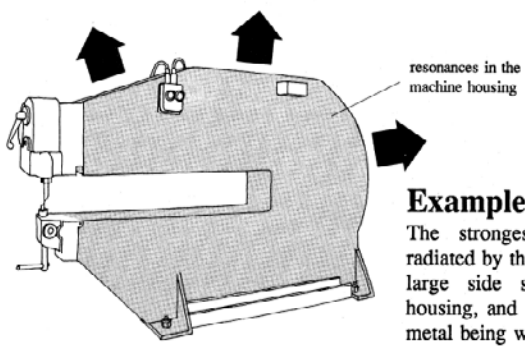
RESONANCE SHIFTED TO HIGHER FREQUENCY IS MORE EASILY DAMPED

Large vibrating plates often have low frequency resonances which can be difficult to damp. If a plate is stiffened, the resonances are shifted to higher frequencies which can be more easily damped.

Principle



Application to a machine housing

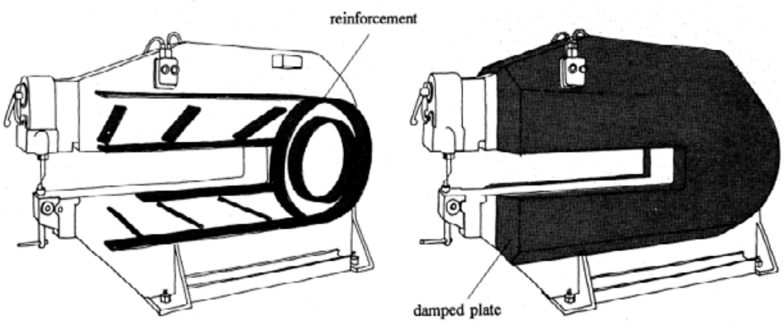


Example

The strongest low frequency sound radiated by this machine comes from the large side surfaces of the machine housing, and not, as expected, from the metal being worked.

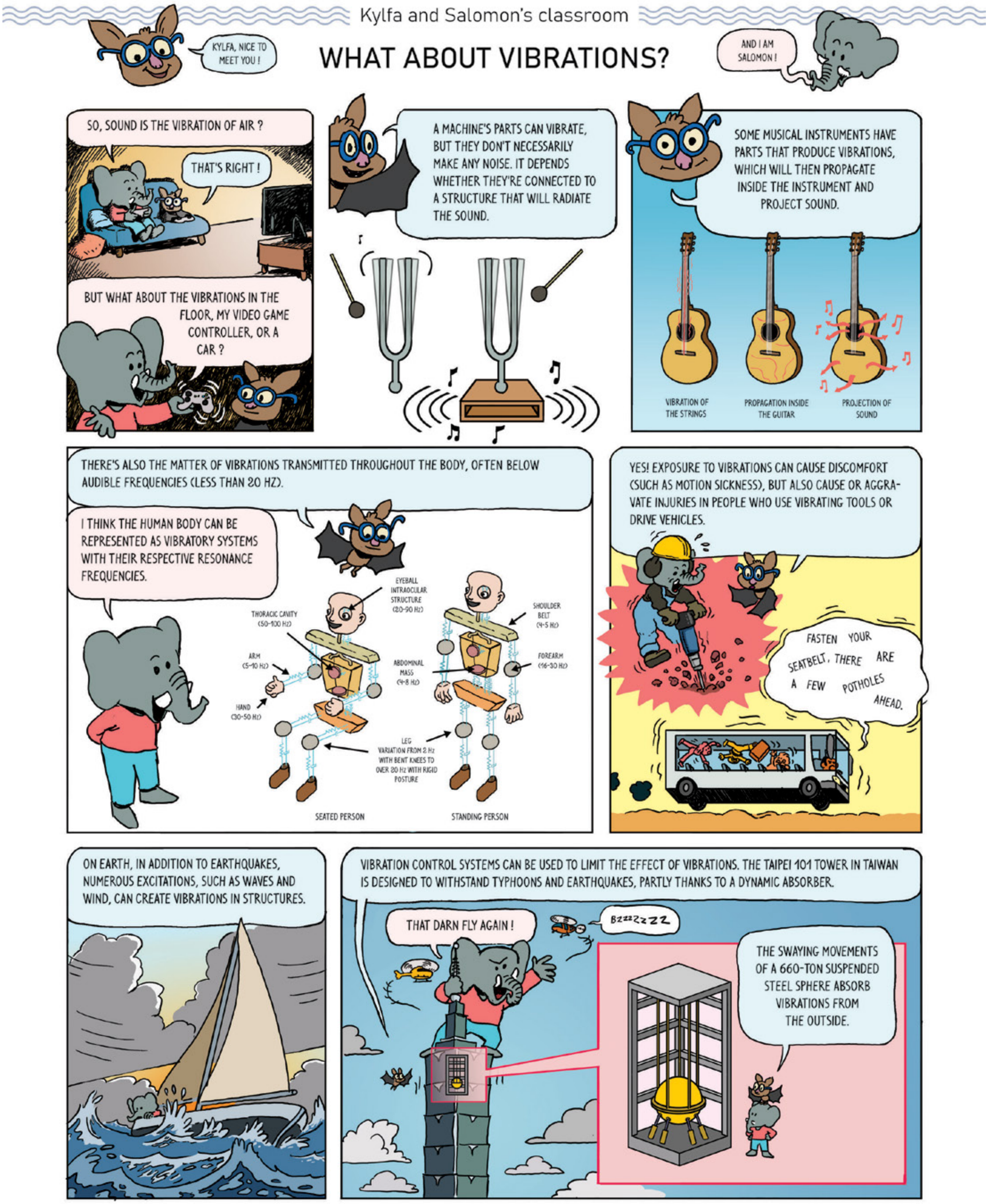
Control Measure

The sheet metal sides of the housing are stiffened with metal straps. A damped plate (total thickness 2 mm) is installed over the straps.



The **"Fantastic Acoustics"** series is a unique blend of comics and science featuring Kylfa, a bat specialized in acoustics, and Solomon, a funny elephant. It is a collaboration between artists and research students from four Québec universities - Université de Sherbrooke, École de Technologie Supérieure, McGill and Université du Québec à Rimouski. To learn more about this series, visit https://doi.org/10.1121/2.0002008.

Downloadable PDFs in French and English are available at https://en.fantastiqueacoustique.net/home



Sound as if you WERE THERE

On a typical Friday evening, you're watching a good movie on your TV. As you're engrossed in the action, you realize that what you're hearing is coming from two speakers. If there were more of them, would you be better immersed in the moment ?

This is where Wave-Field Synthesis comes into play: by placing different loudspeakers in strategic positions, we can reproduce the sound of certain places as if we were actually there.

But how many loudspeakers are needed to achieve this?

And how do several loudspeakers work together to reproduce these sound environments?

These are just some of the questions scientists and engineers have been asking themselves. Fortunately, they have found solutions to meet the needs of science, but also those of the arts and entertainment industry.

Follow us to the cinema to find out more !

Team member

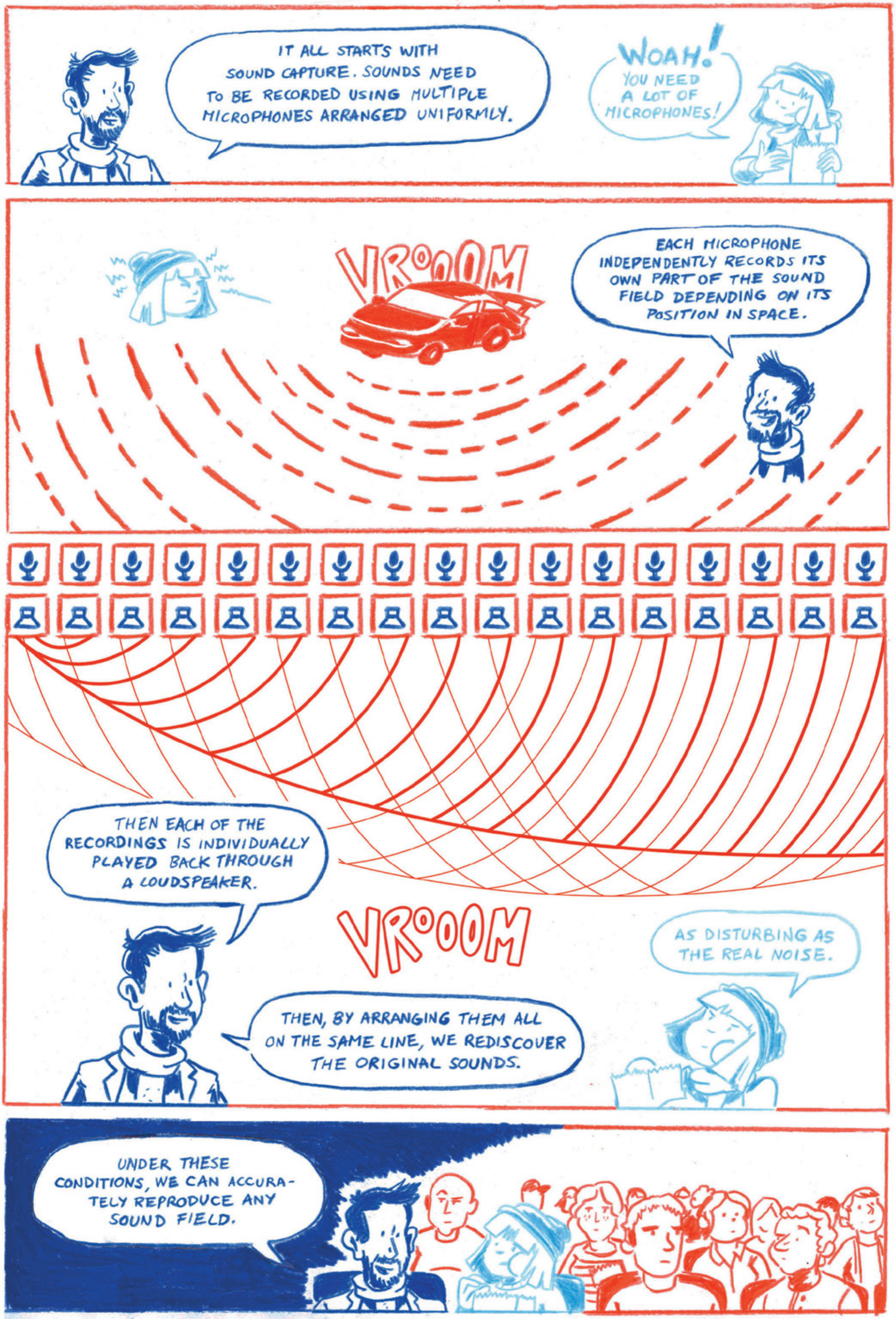
Student at Université de Sherbrooke

- François Proulx

Artist/Cartoonist

- Jordanne Maynard


GOOD SEE
IT'S REALLY NOISY, YOU CAN BARELY HEAR YOURSELF THINK
I CAN'T BELIEVE YOU HAVEN'T BEEN TO THE MOVIES IN ALL THIS TIME, YOU'VE GOT TO GET OUT, MY FRIEND!
POPCORN
POP
POP
YOU KNOW THAT WHAT YOU HEAR CORRESPONDS TO PRESSURE VARIATIONS ON OUR EARDRUMS, WHICH WE CALL ACOUSTIC PRESSURE
THESE PRESSURE VARIATIONS ARE CAUSED BY NOISE SOURCES PRESENT IN THE SPACE AROUND US AND...
BY THE DISTANCE, I SUPPOSE?
YES! THE ACOUSTIC PRESSURE WILL BE WEAKER THE FARTHER WE ARE FROM A SOURCE. IN THE CASE OF MULTIPLE SOURCES AROUND US, THE ACOUSTIC PRESSURE ON OUR EARS IS THEREFORE A COMBINATION OF ALL THE SOURCES.
NEXT
AND IT WILL BE DIFFERENT AS WE MOVE THROUGH THE SPACE AROUND US, ACROSS THE SOUND FIELD.
POP
POP
POP
POP
WHAT'S A SOUND FIELD?
IMAGINE DROPPING SEVERAL ROCKS INTO A POOL. THE ROCKS CORRESPOND TO NOISE SOURCES, THE WAVES REPRESENT THE SOUND TRAVELING, AND THE POOL REPRESENTS THE SOUND FIELD.
SOME ENGINEERS ARE WORKING TO ARTIFICIALLY RECREATE SOUND FIELDS THROUGH WAVE FIELD SYNTHESIS, OR WFS.
THEY'RE NOT REPRODUCING THE NOISE OF A LINE-UP AT THE MOVIES, ARE THEY?
HA! HA! NOT AT ALL!
CHHHHF
HOWEVER, THERE ARE STILL MOVIE THEATERS WHERE YOU CAN EXPERIENCE WFS.
10
?!
THEATER 14
MISSION NOT POSSIBLE


IT ALL STARTS WITH SOUND CAPTURE. SOUNDS NEED TO BE RECORDED USING MULTIPLE MICROPHONES ARRANGED UNIFORMLY.
WOAH! YOU NEED A LOT OF MICROPHONES!
VROOOM
EACH MICROPHONE INDEPENDENTLY RECORDS ITS OWN PART OF THE SOUND FIELD DEPENDING ON ITS POSITION IN SPACE.
THEN EACH OF THE RECORDINGS IS INDIVIDUALLY PLAYED BACK THROUGH A LOUDSPEAKER.
VROOOM
AS DISTURBING AS THE REAL NOISE.
THEN, BY ARRANGING THEM ALL ON THE SAME LINE, WE REDISCOVER THE ORIGINAL SOUNDS.
UNDER THESE CONDITIONS, WE CAN ACCURATELY REPRODUCE ANY SOUND FIELD.


WFS SEEMS COMPLEX TO CARRY OUT. IN WHAT CONTEXT IS IT USED?
EVEN THOUGH THE TECHNOLOGY IS EXPENSIVE AND REQUIRES A SUITABLE LISTENING SPACE, IT IS USED IN SEVERAL SITUATIONS, FOR EXAMPLE...
TO CREATE A SOUND INSTALLATION,
OR TO REPRODUCE A SOUND ENVIRONMENT.
BIP BIP BIP
TOC TOC TOC TOC TOC TOC TOC TOC TOC
AND NOW, IT IS EVEN POSSIBLE TO SIMULATE SOURCES WITHOUT RECORDING THEM.
WFS THEREFORE ALLOWS ENGINEERS AND ARTISTS TO TEST THE RENDERING AND IMPACT OF SOUND SOURCES IN A CONTROLLED ENVIRONMENT, SO IT'S ADAPTABLE TO ALL SITUATIONS.
SO CAN WE ADD AND REMOVE ALL SOUNDS, AS WE WISH?
YES, WAVE FIELD SYNTHESIS IS...
MISSION NOT POSSIBLE
MODULA...
HEY! WE'RE NOT IN A SYNTHESIZED SOUND FIELD HERE! I CAN'T MUTE YOU!
HE STILL MUTED US, SYNTHESIZED OR NOT...
MAI 23

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- **JULY 9–11, 2026**
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- **AUGUST 9–12, 2026**
  INTER-NOISE 2026
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