AN ELECTRO-MECHANICAL AUTOMATIC IMPACT HAMMER DESIGN

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ABSTRACT

Measuring mechanical admittance (velocity over force) is a relevant method for analysing the vibration characteristics of mechanical systems. This generally involves exciting the structure with a known force and measuring the resulting acceleration using suitable sensors. The vibration is usually excited either by a shaker using a sinusoidal input force (sine sweep) or by an impact hammer using a force pulse. Depending on the hammer size and the object being measured, it is very difficult to reproduce excitations by hand. Therefore, it has become standard practice to stabilise the impulse hammer by suspending it like a pendulum. If the excitation is also to be automated, the pendulum assembly can be expanded to include a stepper motor. Commercial systems tested by the authors are expensive and, due to the loudness of the drive motor, are not suitable for acoustic measurements. This paper presents a relatively simple method for implementing an automated impact hammer system using commercial measurement technology and audio components. In addition to design proposals, the paper also discusses first measurement results and the potential for further development.

1. CONTEXT

The method of mechanical input admittance measurement is often used to determine the vibration behavior of mechanical systems [1, 2]. Measurements by excitation using a impact hammer [3] offer the advantage that the measurement can usually be carried out very quickly and easily. Since the contact duration between the hammer tip and the structure to be excited is very short, this form of excitation does not alter the mass of the test object, unlike excitation using a shaker. With hand-held hammers, it is often difficult or impossible to hit exactly the same impact point. Furthermore, the angle of impact of the excitation force is also relevant. Experience with various impact hammers has also shown that it is almost impossible to reproduce the impact with very small and lightweight impact hammers.

Another disruptive factor is what is known as double strikes or bouncing of the hammer tip on the test object. Depending on the rigidity of the test object, it can be very difficult to stimulate the structure with a single, short impulse of force. A frequently used approach is to attach the hammer to the handle as a pendulum, which ensures reproducible excitation in terms of impact point and angle (Figure 1). If the rotation is controlled by a stepper motor, the impact can be controlled and automated by a software. A software-controlled system can then retract the hammer at the appropriate moment, thereby preventing multiple impacts. With all systems featuring a pendulum mechanism, it is often difficult to ensure that the hammer tip strikes the test object at a perpendicular angle. Furthermore, the hammer system is sometimes too bulky to be placed at a suitable location. The cost-effective design discussed here is intended to offer a quiet and compact alternative in addition to reproducible excitation.

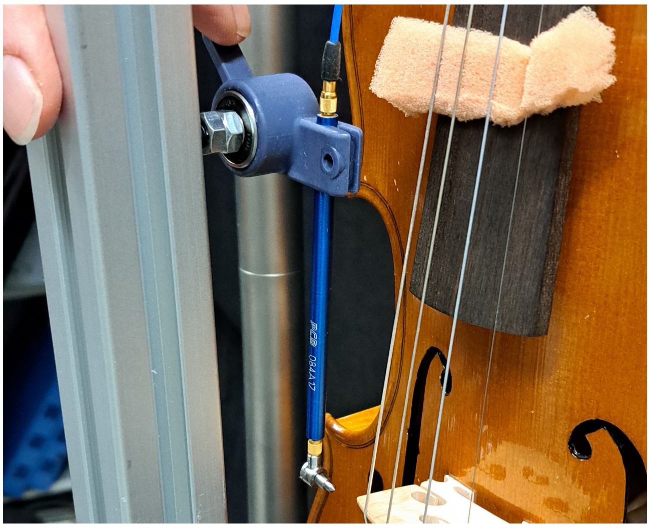

Figure 1: The PCB© impact hammer (Model 086E80 [4], aluminum handle assembly attached) is mounted as a pendulum using a custom-made 3D-printed lever and bearing device.

2. HARDWARE

Unlike pendulum suspensions, the head of the mini hammer from PCB, is mounted directly onto the tip or output of a shaker. Initial tests were successfully carried out on shakers for measurement technology purposes (PCB and B&K). However, as the aim was to find a cost-effective alternative, two different types of vibration loudspeakers were included in a suitability test. Vibration loudspeakers usually consist of a voice coil and a permanent magnet and differ little in design from a conventional electrodynamic loudspeaker. Depending on the design, instead of a diaphragm, there is either a flange (e.g. [5]) for direct coupling to a surface or a spring suspension (e.g. [6]). Mounted on vibrating surfaces, these shakers generate sound by exciting the surface and thus serve as loudspeakers. Although the possible deflection is rather small compared to the shakers used in measurement technology (in the mm range), objects with low vibration amplitude can be easily excited.

Until now, the head of the mini impulse hammer Model 086E80 [4] and an acceleration sensor Model 352C23 [7] from PCB have been used as force sensors. A laser Doppler vibrometer (LDV) from Polytec [8] was used for non-contact measurement of the deflection. The signal conditioner and pre-amplifier Model 483C15 [9] provides the constant current supply (ICP®) for the sensors and signal matching to the DAQ device. When mounting the force sensor, it is important to ensure that it is properly fixed and in line with the shaker axis (see Figure 2, top). For measurements where the hammer must be maneuvered into a confined geometry, the hammer with the aluminum handle assembly can be mounted off-axis from the shaker (see Figure 2, bottom). This configuration is less rigid compared to the on-axis mounting, potentially making it harder to prevent double strikes. A suitable 3D-printed part for both configurations can be found in the appendix.

For the tests taken, the DAQ device PCI-6251 data acquisition card [10] from NI was used as analog input and output converter. Unlike PC sound cards, this DAQ device is designed to process DC voltages. It can also achieve much higher sampling rates than usual audio devices. In principle, audio components can be used to both record the sensor signals and generate the excitation signal. However, in order to obtain measurements in SI units, the inputs and outputs of the sound card would then have to be calibrated. The commercially available audio amplifier module, t.amp, PM40C [11] was used as a power amplifier to drive the vibration speaker.

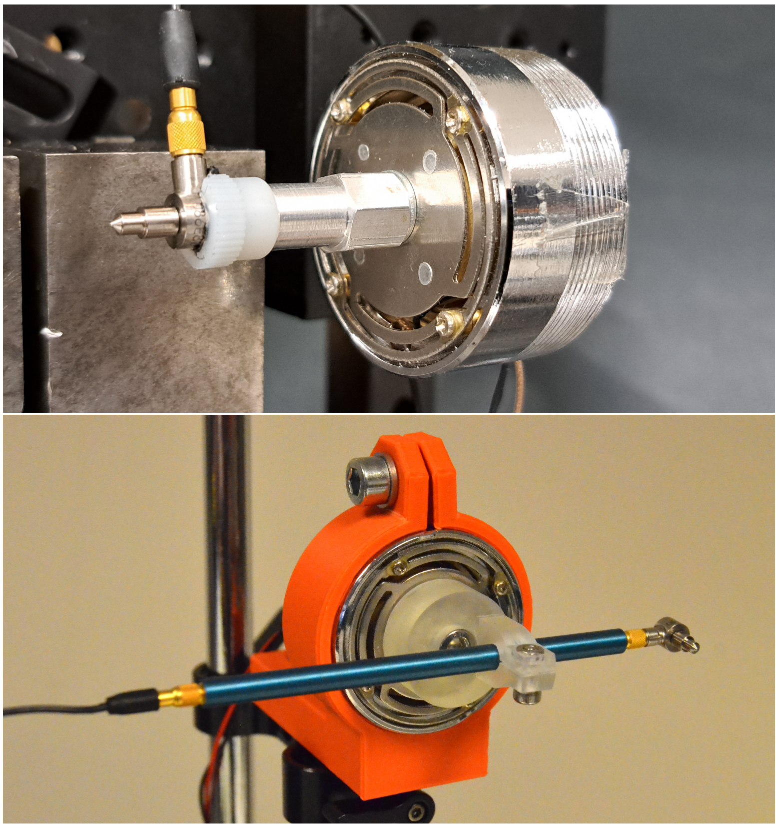

Figure 2: Top: The head of the PCB© impact hammer (Model 086E80 [4]) mounted in a test configuration using modified stinger components on a 44 mm diameter vibration speaker (e.g. [6]). Bottom: The impact hammer and aluminum handle assembly clamped in 3D-printed holder on a 44 mm diameter vibration speaker. A similar approach has already been proven in a measurement setup used in a master's thesis. [12].

3. EXCITATION SIGNAL

To drive the system in an impulsive manner and prevent double strikes, an excitation signal was designed to accelerate the speaker towards the object under test then retract the impact hammer back to its resting position. As the majority of audio amplifiers are ac-coupled, meaning low frequency content near dc is filtered out, the excitation signal takes a form similar to a square wave but with a titled trailing edge. A formula description of the excitation signal, $x(t)$, built from a raw pulse signal x_p and smoothing kernel w_s is given below:

$$x(t) = x_p(t) \star \frac{w_s(t)}{\|w_s(t)\|}, \tag{1}$$

$$x_p(t) = \begin{cases} 1, & \tau_0 \leq t < \tau_0 + \tau_w, \\ 1 - \frac{t-(\tau_0+\tau_w)}{\tau_t}, & \tau_0 + \tau_w \leq t < \tau_0 + \tau_w + \tau_t \\ 0, & t < \tau_0 \text{ and } t \geq \tau_0 + \tau_w + \tau_t \end{cases}, \tag{2}$$

$$w_s(t) = \exp\left(-\frac{1}{2}\frac{(t-\tau_s/2)^2}{\tau_s^2}\right). \tag{3}$$

The design parameters are τ_w, the pulse width, τ_t, the length of the tilted trailing edge, and τ_s, the length of the smoothing window. τ_0 represents the start of the excitation signal. The excitation response is shown in Figure 3(a), and the measured

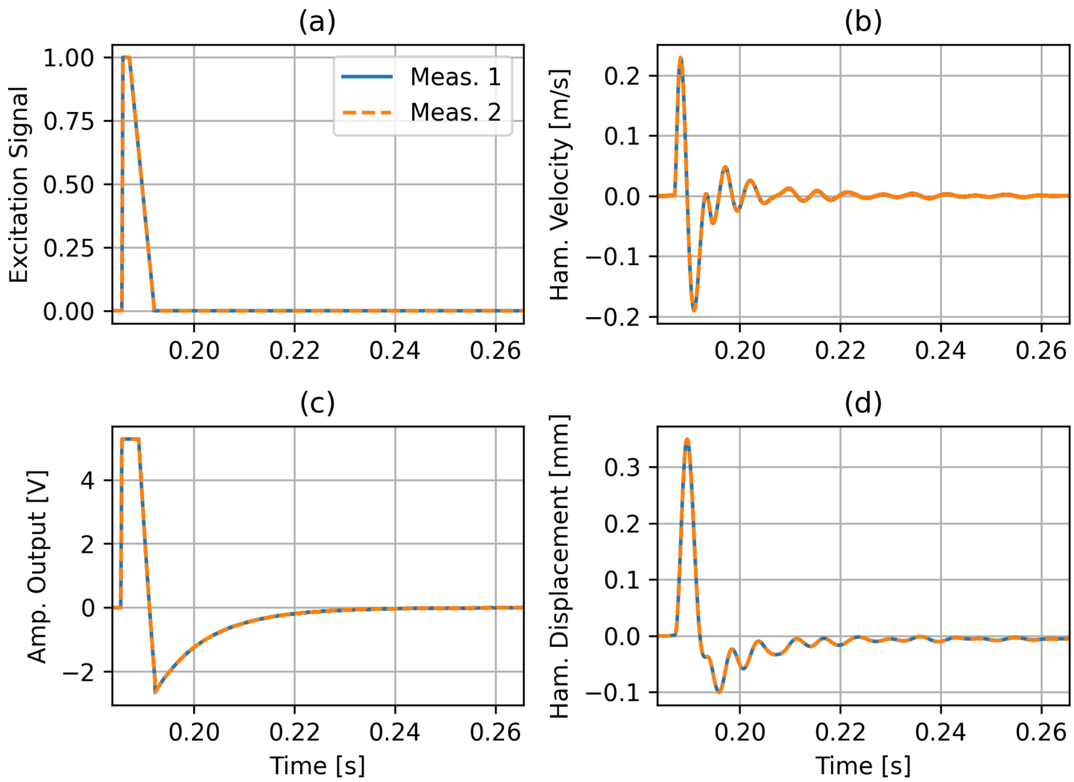

Figure 3: Two measurements of the tuned excitation signal. (a) The digital signal supplied to the audio amplifier, (c) the measured audio amplifier signal, (b) the velocity of the system as measured using an LDV, (d) the displacement computed via integration of the velocity signal.

audio amplified signal is shown in Figure 3(c). A sharp and square trailing edge could re-excite the hammer and generate an additional impact. To avoid this, the trailing edge is tilted to relax the speaker's movement back to its resting position.

The pulse width, τ_w, should be adjusted such that the trailing edge is timed to "catch" the backward movement of the speaker based on the system's vibrational frequency. The excitation signal results in a sudden jerk of the speaker assembly which can produce an undesirable non-smooth acceleration of the hammer. To prevent this, the raw excitation signal is smoothed via convolution with a 1.2 ms long Gaussian kernel, w_s. The velocity of the system driven by a tuned excitation signal and measured using an LDV is shown in Figure 3(b) with the integrated displacement shown in Figure 3(d). As can be seen in Figure 3(d), the system provides a sharp impulse before being relaxed back to its resting position.

4. SYSTEM ANALYSIS

To analyze the linearity of the system in response to sharp impulses, as used in the excitation signal, the system was driven with positive and negative edge signals and the maximum positive or negative displacement, respectively, was calculated via numerical integration of the measured velocity signal. The result is shown in Figure 4 demonstrating the roughly linear and symmetric behavior of the system.

4.1. System verification with a mass

Accurate calibration of an impact hammer and accelerometer is essential for ensuring reliable measurement and analysis in dynamic testing applications, such as modal analysis and vibration studies. The calibration process involves suspending a known mass freely to minimize external constraints and enable controlled impact testing. The impact hammer delivers a measured force to the mass, while the accelerometer records the resulting acceleration. By analyzing the relationship between the applied force and the measured acceleration, the sensitivity and linearity of both instruments can be verified and adjusted. Key parameters, such as the mass's natural frequency, damping characteristics, and the alignment of the sensors, are considered to ensure accuracy. This method provides a straightforward and repeatable procedure for calibrating impact hammers and accelerometers, enhancing measurement reliability in experimental setups. The calibration is based on the linear behavior of a rigid body. In theory a rigid mass has no resonance frequencies, so it has a linear frequency response function when comparing a measurement of acceleration over force [13].

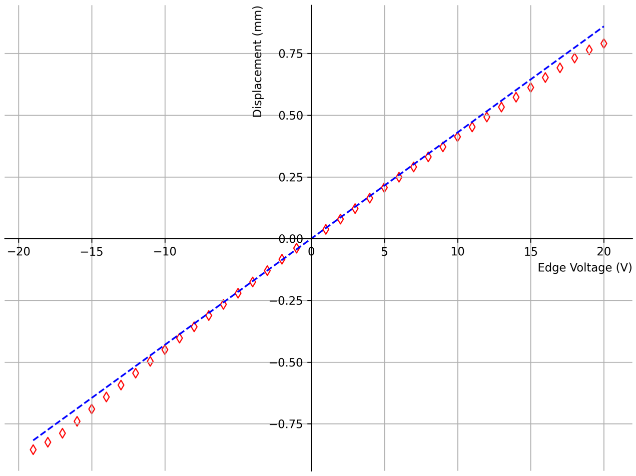

Figure 4: In red: maximum positive or negative displacement versus the voltage of a positive or negative edge signal. Dashed blue is a linear regression fit to the measurement data.

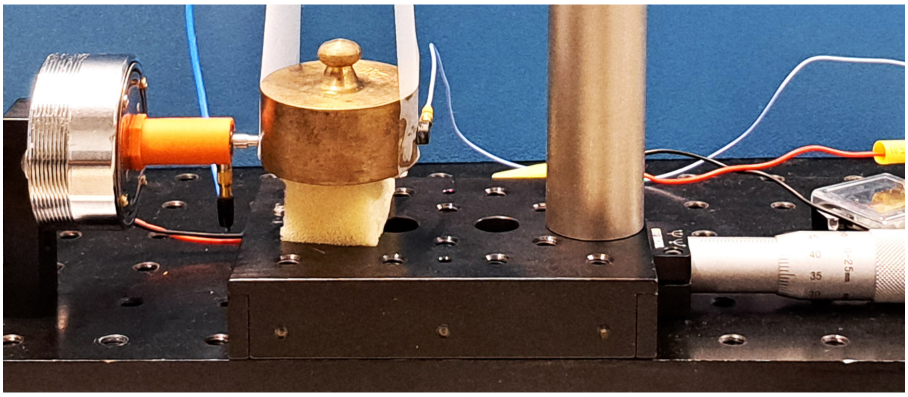

Figure 5: Test setup [13] for checking the systems calibration by exciting a freely suspended but damped mass of 0.2 kg. An accelerometer is located on the back of the mass.

In the setup used, the mass was slightly damped to prevent a swing back and bouncing onto the hammer tip (Figure 5). However, since the sensors used were delivered by the manufacturer already calibrated, this paper will only focus on examining the time signals of impact force and acceleration (see Figure 6). Except for a slight vibration component in the acceleration signal, the signal shapes are identical. When the values are substituted into Newton's law $m = \frac{F}{a}$, the calibration of the system can be determined to be correct.

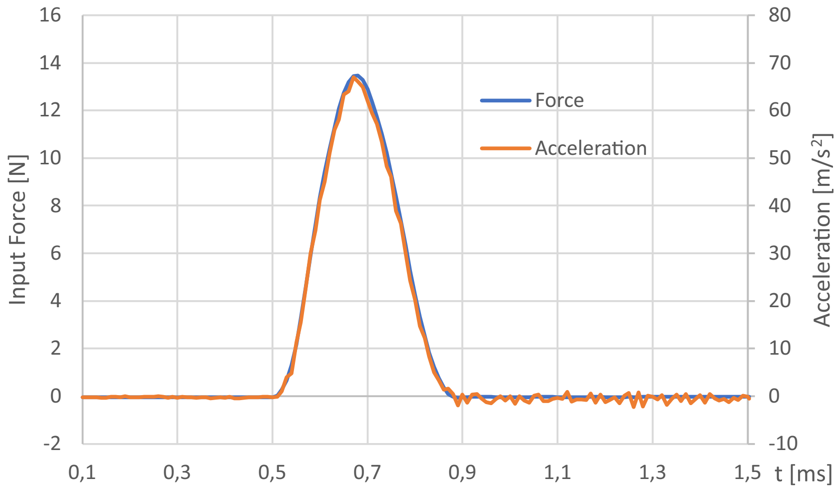

Figure 6: Measured time domain signal of the impact hammer force (blue) and the signal of the accelerometer (orange).

4.2. Test measurement of a violin

The input admittance of bowed string instruments is a critical parameter that gives valid information about some properties of the instrument [14]. This method involves exciting the instrument's structure by delivering a controlled force impulse at specific locations, such as the bridge, with a calibrated impact hammer. The resulting vibrational response is captured by an accelerometer that is commonly placed nearest to the point of impact. As this is difficult with some instruments, an accelerometer is often attached to the opposite side of the bridge. The input admittance is obtained by analyzing the ratio of the velocity response to the applied force in the frequency domain. However, since the resonances do not change significantly, it is often customary in musical acoustics to dispense with the integration calculation and instead represent the ratio of acceleration to force.

In the test setup, a violin was suspended freely on rubber bands. The impact hammer, this time mounted on the smaller shaker, was moved close to the instrument using an XYZ translation stage[1] (see Fig. 7). Fig. 8 illustrates the time signals of the force measurements on the left and the curves of the acceleration sensor signals on the right. Twelve excitation trials were recorded in order to check reproducibility. In contrast to the clean force curve presented in Fig. 6, however, the falling edge of the input force pulse appears here less symmetric. We attribute this to the fact that the test object is mounted in a vibrating but almost undamped manner, which can lead to slight bouncing. A more rigid suspension could prevent this effect, but it could also affect the violin's eigenfrequency behavior. Another approach would be to use a shaker with a larger stroke, which would increase the resting distance to the test object and prevent further contact between the test object and the hammer.

Figure 7: Excitation of a violin, using a 34 mm diameter vibration speaker (e.g. [5]) as driver. A 3D-printed centerpiece is glued on the plastic rim of the moving coil assembly to transfer the movement to the impact hammer. The shaker's small diameter allows it to fit in front of the violin body without making contact.

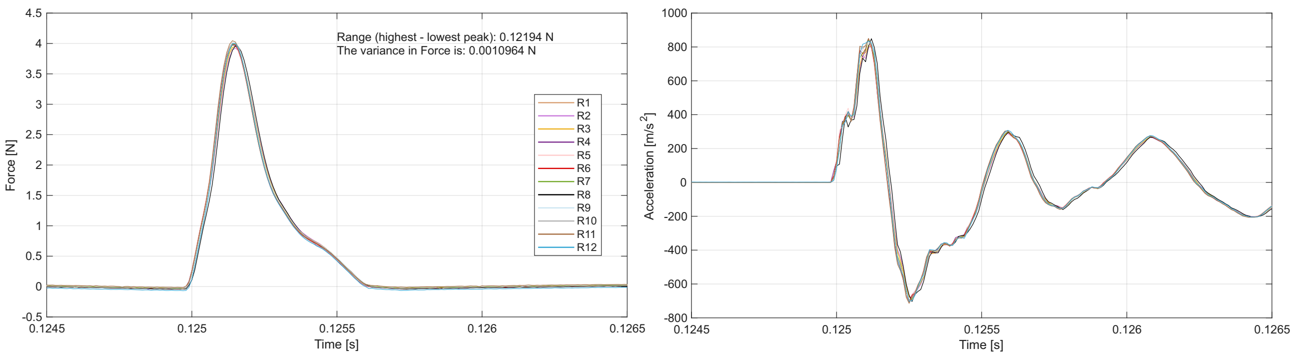

Figure 8: Time domain signals of the measured hammer input force (left) and the acceleration response signal measured on the opposite side of the violin bridge (right). The overlay of 12 measurements indicate a stable reproducibility. The shape of the slope of the force signal's decay suggests a faint double-hit.

The admittance curves are presented in Figure 9 (orange curve). The FFT of the time signals was performed unfiltered and with rectangular windowing with the length of the whole recording. The resonance properties are clearly visible, and the measurement system's reproducibility can also be rated as very good. The coherence function is also used to assess the quality of the data (blue curve in Figure 9), identifying how much of the output signal is related to the measured input

[1] e. g. XYZ translations stage PT3 from Thorlabs: www.thorlabs.com/1-inch-25-mm-travel-translation-stages?tabName=Overview

signal (see [15, 16]). To obtain this indicator, the power spectra $S_{xx}(f)$ of the input signal $x_k(f)$ and the power spectra $S_{yy}(f)$ of the output signal $y_k(f)$, both averaged across all trials, are calculated. Another component is the cross power spectrum $S_{xy}(f)$, also averaged across all trials.

$$S_{xx}^{(k)}(f) = \text{fft}(x_k(f) \cdot \text{conj}(\text{fft}(x_k(f)) \text{ and averaging across (K) trials: } S_{xx}(f) = \frac{1}{K}\sum_{k=1}^{K} S_{xx}^{(k)}(f) \tag{4}$$

$$S_{yy}^{(k)}(f) = \text{fft}(y_k(f) \cdot \text{conj}(\text{fft}(y_k(f)) \text{ and averaging across (K) trials: } S_{yy}(f) = \frac{1}{K}\sum_{k=1}^{K} S_{yy}^{(k)}(f) \tag{5}$$

$$S_{xy}^{(k)}(f) = \text{fft}(x_k(f) \cdot \text{conj}(\text{fft}(y_k(f)) \text{ and averaging across (K) trials: } S_{xy}(f) = \frac{1}{K}\sum_{k=1}^{K} S_{xy}^{(k)}(f) \tag{6}$$

Finally, the coherence $C_{xy}(f)$ is given by:

$$C_{xy}(f) = \frac{|S_{xy}(f)|}{\sqrt{S_{xx}(f)} \cdot \sqrt{S_{yy}(f)}} \tag{7}$$

Focusing on the coherence plot in conjunction with the admittance curve in Figure 9, a poor coherence at the lower frequency range between 10 Hz and 200 Hz is obvious. As the response of the instrument close to 100 Hz is low, low SNR levels may cause the low coherence factor. The ripples in the admittance curve below 100 Hz could have been caused by the slight bounce of the impact hammer. However, the coherence in the frequency range relevant for this type of instrument is stable and good. Only at frequencies above approx. 5000 Hz does the signal deteriorate. This is again due to the poor response of the instrument and low SNR in this range. In order to obtain more excitation energy in the lower frequency range, a softer hammer tip should also be tried.

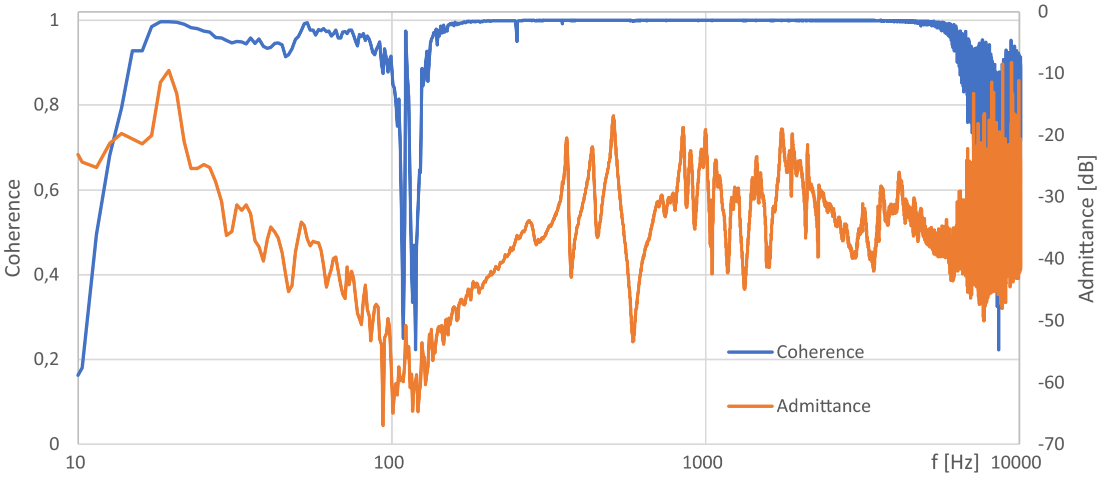

Figure 9: Blue: the calculated coherence indicates a good measurement almost along the whole frequency range of interest (ca. 200 Hz to 6 kHz), poor signal-quality can be observed at the admittance minima close above 100 Hz. Orange: unfiltered admittance curve of the violin.

4.3. Test measurements of an alto saxophone reed

Experimental modal analysis can also be used to determine the material parameters of various object such as alto saxophone reeds. Measuring the saxophone reeds in a manner similar to the violin is difficult due to their small size, thin geometry, and low mass relative to the sensors used in the measurements. For this reason, deflections are measured using an LDV to avoid mass loading from an attached accelerometer. The reed is easily excited, making it prone to double strikes and moving at velocities faster than what can be suitably measured by an LDV. These issues are mitigated by limiting the force pulses used to excite the reed to low amplitudes, around 1 N.

Figure 10 demonstrates the measurement setup for measuring reeds. The LDV is aimed on the face of the reed and the hammer excited the reed on its back. Figure 11 displays the results of measurements performed on a moistened natural cane alto saxophone reed (Vandoren, Traditional Cut, Strength 3.5). The measurement system provides consistent and repeatable excitations over nine trials with a low peak force amplitude. Measurements, such as these, can be used to extract material parameters for spatially varying one-dimensional beam model using optimization as shown in the lower plot.

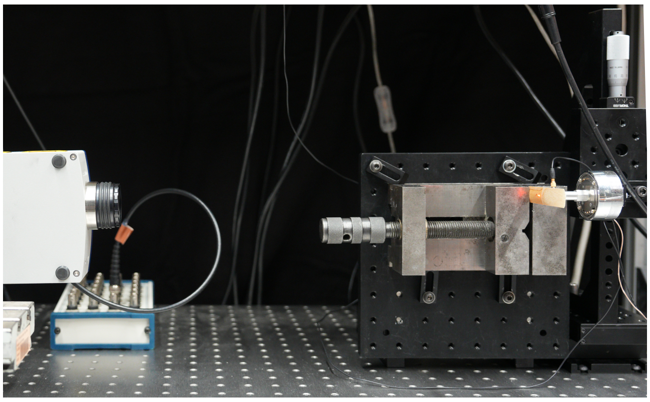

Figure 10: Measurement setup for experimental modal analysis of single reeds. The reed is clamped in a vise and excited from the back using a force hammer. The deflections on the front of the reed are measured using an LDV.

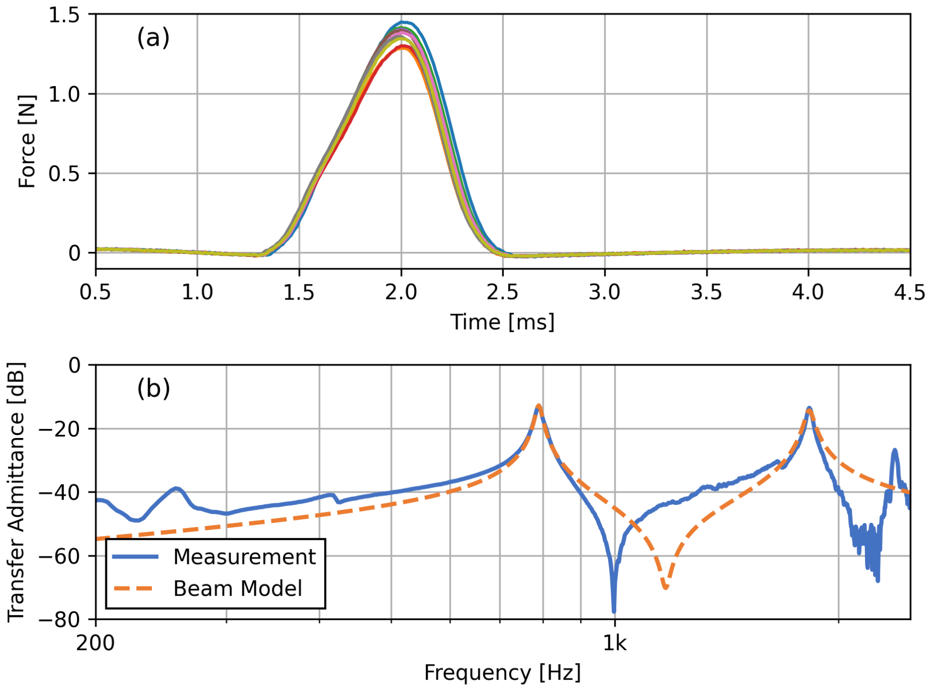

Figure 11: (a) Force pulses used to excite the alto saxophone reed. (b) Resulting averaged admittance transfer measurement and beam model response fit.

5. CONCLUSIONS

This paper presented a technical solution for an automatic excitation system for mechanical admittance measurements. Test measurements using two different vibration speakers demonstrated high reproducibility. A possible method for generating an excitation signal is proposed. Test measurements were taken to evaluate the performance of the driving mechanism under different conditions. Response sensors in the form of an accelerometer and an LDV system were used. System-related limits due to the low stroke of the proposed speakers are pointed out. However, our experience has shown that the strike is reproducible at a wide range of forces, which legitimizes further development. In order to be able to offer affordable tools to interested instrument makers in particular, the sensors could also be replaced by inexpensive components in the future. We hope that this description will serve as an impulse for an open source project.

6. AUTHOR CONTRIBUTIONS

Conceptualization and methodology, A.M., C.D.; algorithm implementation, A.M., C.D.; hardware and software, A.M.; data collection, curation and visualization, A.M., C.D.; writing – original draft, A.M.; writing – review and editing, A.M., C.D.; 3D design, A.M., C.D.; All authors have read and agreed to the published version of the manuscript.

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8. APPENDIX

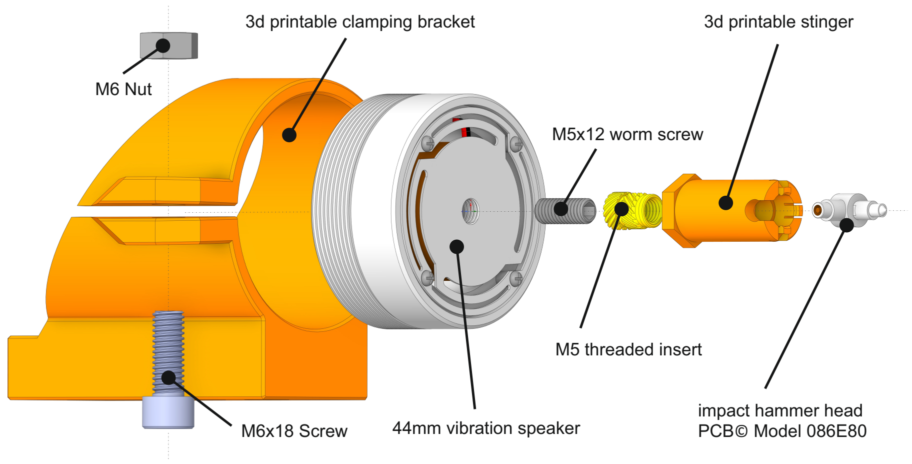

Figure 12: exploded view of the components using the 44 mm vibration speaker [17].

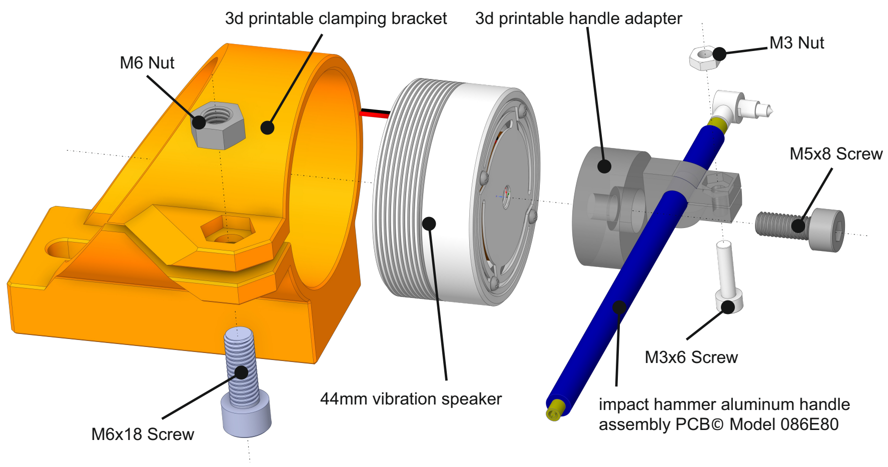

Figure 13: exploded view of alternative components using the 44 mm vibration speaker. In this configuration the hammer is clamped on its aluminum handle assembly [18].

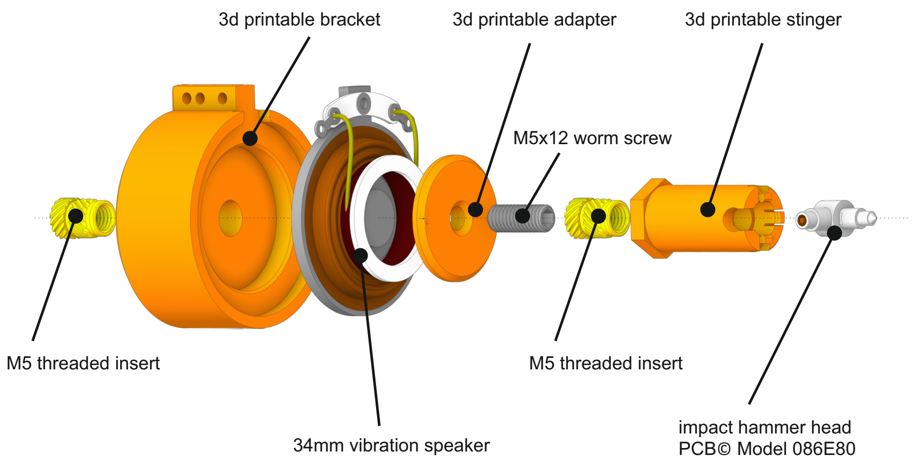

Figure 14: exploded view of the components using the 34 mm vibration speaker [5]. As the vibrational speakers coil is mounted much more flexible than on the 44 mm type we only recommend to use the in center mounted impact hammer setup [19].