Granular dampers for the reduction of vibrations of an oscillatory saw

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A B S T R A C T

Instruments for surgical and dental application based on oscillatory mechanics submit unwanted vibrations to the operator's hand. Frequently the weight of the instrument's body is increased to dampen its vibration. Based on recent research regarding the optimization of granular damping we developed a prototype granular damper that attenuates the vibrations of an oscillatory saw twice as efficiently as a comparable solid mass.

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1. Introduction

Surgical and dental instruments for mechanical material ablation by design often suffer from strong vibrations. These vibrations disturb the haptic sensing, leading to uncomfortable operation and may damage the joints of the operator's hand. The vibrations may lead to decreased accuracy of the instrument's application and can trigger uncomfortable or painful sensations in the patient. One method to reduce these unwanted vibrations are granular dampers, that is, cavities filled with grains. Under vibration, the particles collide dissipatively leading to an attenuation of their total kinetic energy and, thus, to damping of the vibration.

There is a long history of research on granular damping, e.g. Refs. [1–5]. In comparison to other types of dampers, granular dampers have several advantages: (a) They are extremely simple and, therefore, cheap—just cavities with particles inside (e.g. Refs. [6–8]); (b) they require little or no maintenance; (c) they hardly undergo aging unlike viscous liquids or components of active dampers; (d) they do not require a resting part against which the vibration is damped; (e) unlike viscous dampers their operation is almost independent of temperature [9]. Also they are suitable for sterilization as required for medical instrumentation (e.g. Refs. [10,11]).

Other applications of granular dampers include bank note processing machines [12], tennis rackets [13], mechanical tools (like the dead blow hammer [14]), turbine blades [15] and even the space shuttle main engine [3].

As a *proof of principle*, in this work we show that granular dampers, also called *particle dampers*, are suitable for reducing unwanted oscillations of saws used for example in medical applications [16].

2. Experimental setup

We concentrate our studies on saws using an electro-mechanically driven oscillating blade. As a reference, a medical saw of type Aesculap Acculan 3Ti (GA673) was examined and its mode of operation was analyzed. Since a model of this saw was not available for modifications, a less expensive saw, not certified for medical purposes, but using the same working principle was used to conduct the experiments (BOSCH PMF 10, 8 LI).
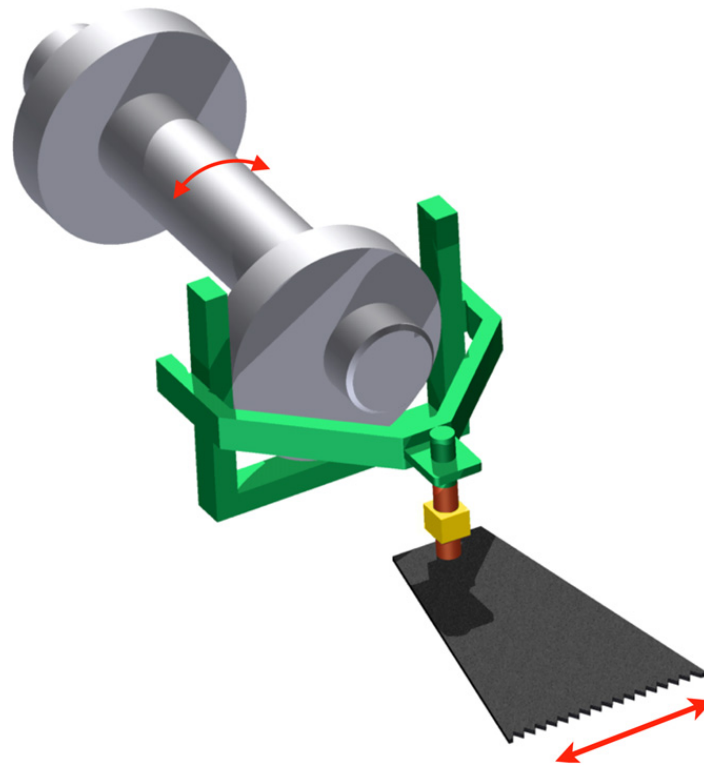

Fig. 1. Sketch of the oscillatory saw's driving mechanism. An eccentric disk connected to a rotating axis (gray) drives the oscillatory motion of the saw blade (black) through a fork (green). The surrounding body, including the saw's handle (not shown), is connected to the motor (not shown) and to the rotation bearing (yellow). For better visualization, the eccentricity of the disk was exaggerated. (For interpretation of the references to colour in this figure legend, the reader is referred to the web version of this article.)
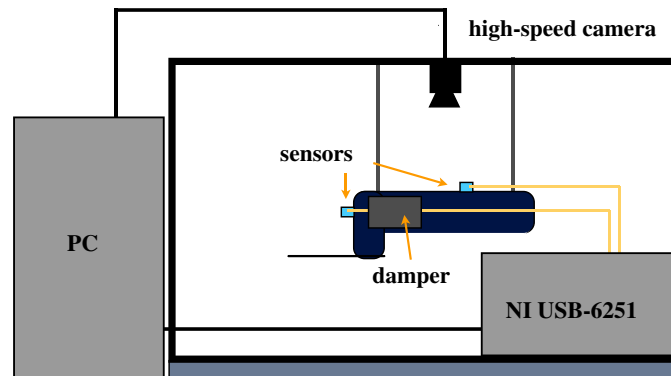
Fig. 2. Experimental setup.

Fig. 1 illustrates the generation of the oscillatory motion. An electric motor drives an axis carrying an eccentric disk. The eccentric disk is framed by a fork that is connected to the saw blade. When the eccentric disk is rotated the fork is pushed left and right, thus, generating the oscillatory motion of the blade. The vibrating blade causes a counter-acting motion of the saw's body including the handle due to conservation of angular momentum.

The aim of our research is to apply granular dampers to either reduce the total mass of the device while keeping the vibrational properties invariant or to reduce the vibrations of the handle leaving the total mass of the saw invariant.

The experiments were conducted using a setup as shown in Fig. 2. The saw was suspended by strings in a solid metal frame and was equipped with accelerometers (Analog Devices AD 22285) next to the blade (front part of the saw) and on top of the handle to measure the strength of the vibrations. The momentary acceleration data provided by the sensors were fed into a data acquisition device (NI USB-6251) connected to a PC. The sample rate was 10 kHz and the measurement time was 21 s, so we had 210,000 data points per measurement for our analysis. The front part of the saw where the largest vibration amplitudes are expected was modified to carry granular dampers attached laterally to the saw (see Fig. 3). A high speed camera (Redlake MotionScope M3) was employed to record the saw's motion. The amplitude of the oscillation of the saw was measured by attaching a marker onto the saw that shows a distinct grayscale value in the high-speed recording and then determining its centroid for each frame.
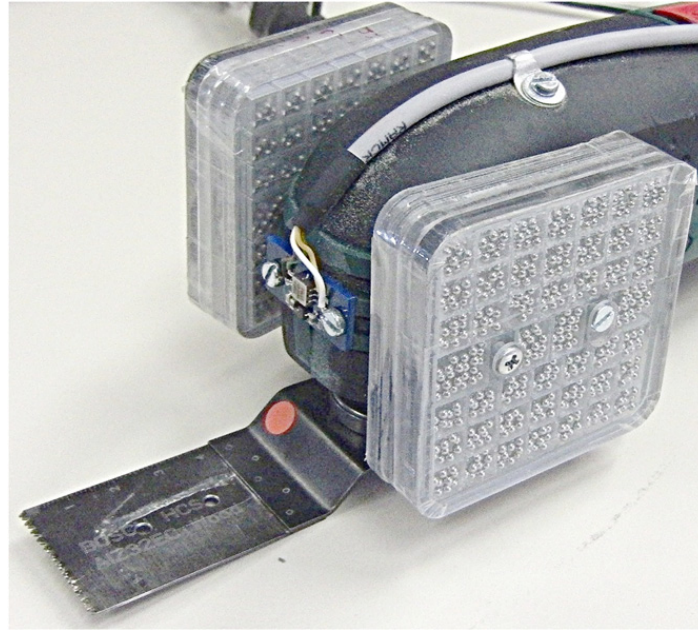
Fig. 3. Saw with prototypical dampers attached. One of the acceleration sensors is fixed at the front side of the case. The red label attached to the blade is one of the optical markers used for high-speed video recording of the vibration. (For interpretation of the references to colour in this figure legend, the reader is referred to the web version of this article.)

3. Damper design

As shown recently [5], the optimal length, L_{opt}, of a granular damper along the direction of the oscillation is given by

$$L_{opt} = \pi \frac{A_0}{2} \sqrt{\frac{M}{M + Nm}} + \sigma_{layer}, \tag{1}$$

where A_0 is the peak to peak amplitude of the oscillation, M is the net mass of the oscillating object (here the saw without the granular damper), N is the number of beads of mass m and σ_{layer} is the filling height of the damper. Eq. (1) indicates that the size of an optimal damper depends only on the amplitude of the oscillation but is independent of the frequency.

Eq. (1) was derived for granular dampers attenuating the vibration in linear direction. Nevertheless, although the mechanics of the saw is different from a linear spring, Eq. (1) can be applied with some modifications. The handle of the saw is approximately a cylindrical torsion pendulum whose rotation axis is perpendicular to the axis of the cylinder. Driven by the inertia of the oscillating blade, the pendulum rotates with a tiny amplitude, $\varphi_0 \approx 2 \cdot 10^{-3}$ rad, around its center of mass. The motion obeys $T = J\ddot{\varphi}$, where T is the torque exerted by the blade on the saw. The dampers are mounted close to the end of the handle, at distance $H/2$, where H is the length of the saw (length of the axis of the cylinder). There, the dampers feel the periodic force $F = 2T/H$. The moment of inertia of a rod of length H and mass M_R, rotating around its stable axis is $J = M_R H^2/12$. Finally, for small elongation, φ, we write $\varphi = 2x/H$, where x is the displacement of the damper, thus, x and F are the coordinate of the damper oscillating linearly around $x = 0$ and the force acting on the damper, respectively. Combining these equations, we obtain the equation of motion of the damper in x-direction: $F = 1/3 M_R \ddot{x}$. Consequently, Eq. (1) is applicable to our system if we replace M by $M_R/3$.

There are some side conditions for the design of the damper: First, the damper should not add too much total mass to the saw for convenient usage. Second, the particles should be free to move in the direction of the saw's oscillation but not form a sediment of resting particles, due to gravity, which do not contribute to the dissipation of energy, see Ref. [17] for a detailed discussion. The formation of larger sediments can be avoided by subdividing the damper into separate compartments, each containing only a small number of particles. Third, the size and outer shape of the damper should allow for a normal operation and handling of the saw. These conditions together with the modified Eq. (1) gave rise to the following design (see Fig. 4): The damper combines 46 small cavities in a polycarbonate box forming a rectangular arrangement of size 60×60 mm^2 and height 4 mm. Each of the cavities contains 9–23 spherical steel beads of diameter 2 mm. The dimensions of the cavities were chosen accordingly to Eq. (1) for the optimal damper size, taking into account the amplitude of the oscillation, $A_0 \approx 300$ μm, as determined by means of the high-speed camera, $M = 1$ kg and $Nm = 180$ g resulting in an optimal damper length of $L_{opt} = 2.4$ mm. To maximize the damping, three layers of dampers were stacked to a damping unit. Fig. 4 shows a sketch of the designed dampers, and Fig. 3 shows a photo of a pair of damping units mounted laterally to the saw.
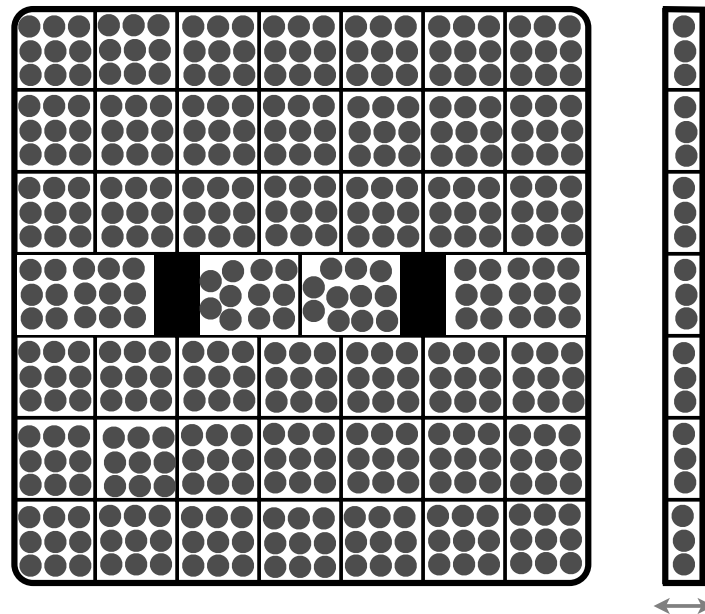
Fig. 4. Schematic drawing of the dampers in front (left) and side (right) view. The black rectangular areas in the middle are the attachment points and the gray double arrow in the side view illustrates the direction of the blade motion.
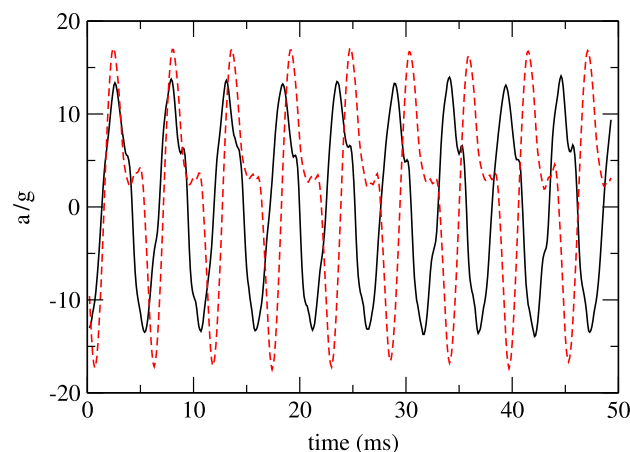

Fig. 5. Characteristic vibrations under load (red dashed line) and without load (black full line) for low driving power. The acceleration measured synchronously to the blade motion is given in units of gravity g (color online).

4. Experimental results

We analyzed the acceleration data and the high-speed video recordings for the unloaded saw where the saw was freely suspended as sketched in Fig. 2 and for the loaded case, where the device was applied to saw a piece of wood. Fig. 5 shows characteristic vibrations of the saw with and without load. By showing an approximately sinusoidal acceleration profile it justifies the use of Eq. (1) as a starting point for the optimization.

We noticed, however, that the data for the saw under load were not sufficiently (quantitatively) reproducible. This may be attributed to the mechanism which drives the oscillatory motion of the saw: In the unloaded case, the torque leading to the oscillation is caused by the moment of inertia of the oscillating blade only, which is a well defined constant. Under load, there is an extra moment resulting from the interaction of the saw with the material the saw is applied to. Naturally, the material is gradually degraded by the saw leading to a non-constant torque. A possible way to imitate the load such that the system stays stationary would be to add an extra mass to the blade, leading effectively to an enhanced torque due to increased moment of inertia and, thus, to more intense vibration of the saw.

In this study, we restrict ourselves to the experimental investigation of the vibration damping for the unloaded case. To this end, we analyze the damping efficiency for two different speed settings of the unloaded, that is freely suspended, saw.

Table 1 lists the vibration characteristics of the undamped saw.

To assess the efficiency of granular dampers in an unbiased way we equipped the saw alternatively with granular dampers and with passive solid masses of the same total mass. We performed experiments using five different setups and two different settings of the saw's vibrational agitation, summarized in Table 2.

Table 1
Vibration characteristics of the saw.

Frequency (low power)	(181 ± 1) Hz
Frequency (high power)	(220 ± 3) Hz
Amplitude (peak–peak)	$\approx 300\,\mu$m
Max. acceleration (low power)	± 13 g
Max. acceleration (low power, with load)	± 18 g
Max. acceleration (high power)	± 20 g

Table 2
Description of the experimental configuration.

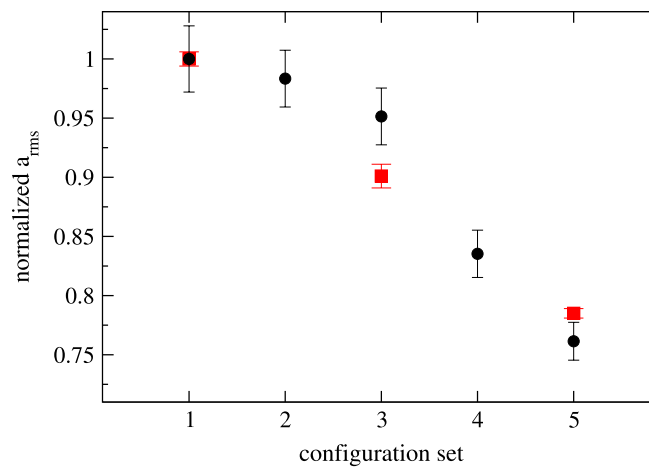| Set	Type of damper	Mass (g)	Driving power
1	None		Low/high
2	Solid mass	90	High
3	Solid mass	180	Low/high
4	Granular (steel beads)	90	High
5	Granular (steel beads)	180	Low/high

Fig. 6. Vibrational acceleration in various damper configurations (see Table 2) normalized to the root mean square of the acceleration without damper for low driving power (red squares) and high driving power (black circles) with error bars from up to 21 unique measurements. (For interpretation of the references to colour in this figure legend, the reader is referred to the web version of this article.)

Fig. 6 shows the average acceleration at low and high driving power (see Table 1) of the saw with and without dampers. The plotted value is the average of the root mean squares (a_{rms}) of the acceleration a_i measured on the saw's body:

$$a_{\mathrm{rms}} = \sqrt{\frac{1}{n}\sum_{i=0}^{n-1}\left(a_i - \frac{1}{n}\sum_{i=0}^{n-1}a_i\right)^2}, \qquad (2)$$

normalized by the corresponding value for the undamped case. The number of points in the data set was $n = 2.1 \times 10^5$, recorded at sampling rate 10 kHz and a_i is the instantaneous acceleration. The error bars result from 2σ standard deviation of a_{rms} determined from up to 21 independent measurements per set (the standard deviation inside the single sets was much smaller than comparing the independently measured sets and is therefore neglected).

Comparing the data for the undamped saw (set 1) with the damping by a solid mass (sets 2 and 3) with the data for the granular damper (sets 4 and 5) we notice that the granular damper attenuates the vibration about twice as efficiently as the solid mass. As expected, two damping units operate more efficiently than one (configurations 2 and 4 in comparison to 3 and 5 in Fig. 6). Nevertheless, these data do not allow to provide a reliable scaling law.

To see which frequencies were affected by the dampers, a Fourier transform of the signal was computed. Fig. 7 compares the power spectrum of the acceleration with and without the dampers attached. All peaks of the spectrum were clearly attenuated up to about 10 dB by the action of the dampers. The achieved attenuation of the vibration concerns the lowest modes which are the most important ones for practical applications.

5. Discussion

We investigated the attenuation of the vibrations of an oscillatory saw's handle by means of two different passive damping mechanisms: (a) A solid mass was attached to the handle to increase the inertia of the saw's body. Since the
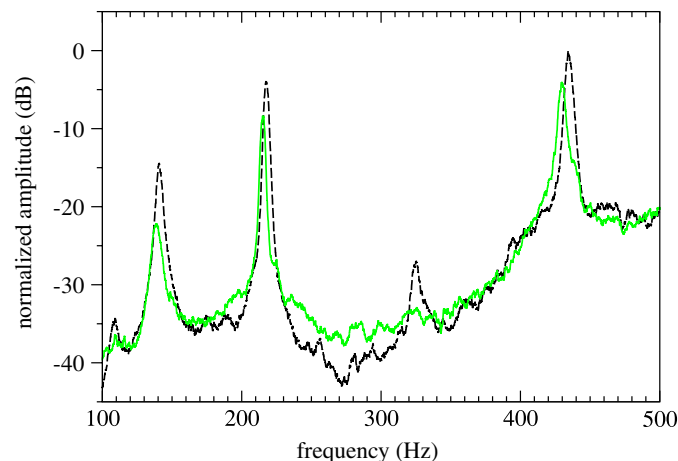
Fig. 7. Power spectrum of the acceleration data recorded at the high power setting. One can see the power reduction with the damper configuration set 5 (green solid line) against the undamped case (black dashed line) from configuration set 1 (color online).

saw is suspended by strings which transmit almost no torque, the amplitude of the body's vibrations is reduced due to the conservation of angular momentum, albeit the mass does not dissipate any mechanical energy. (b) Granular dampers of the same total weight as in case (a) were attached to the handle, here part of the kinetic energy is dissipated due to inelastic collisions. The size of the granular damper was chosen such that the optimal dissipation rate can be expected [5].

We find that granular dampers operate more efficiently than solid mass dampers if the geometry of the dampers is optimized with respect to the specific amplitude of the vibration. Consequently, by means of granular dampers one can achieve either more efficient damping while keeping the total weight of the saw invariant or alternatively the weight of the saw can be reduced by keeping the residual vibration of the handle constant.

It was our aim to provide a proof of principle for a practical application of granular dampers rather than to optimize the damper such that the most efficient attenuation is achieved. Such an optimization is possible, but always depends on the specific application.

Acknowledgments

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