Design of a mechanical resonant station to free jammed micro-mechanisms

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Introduction

Some types of mechanical devices, such as molds, but not only, include a large number (many hundreds) of micro- mechanisms, valves, channels, vents, or other devices subject to deterioration or malfunctioning, frequently caused by unwelcome phenomena such as incrustations, fillings, or reciprocal bonding and adhesion of parts following prolonged periods of intense use [1,2].

Standard cleaning techniques (wet washing with detergents, sandblasting, cleaning with high power lasers, ultrasonic cleaning with chemical solvents) can be highly automated and optimized in terms of time and costs, but they can't solve the problem of blocked micro-mechanisms [1,2]. It is therefore understandable why it is so important the invention of a resonant station that allows the easy unlocking of all micromechanisms at the same time. The station consists of an electromechanical transducer and a closing clamp, which operate in such a way that at least one mechanical resonance mode of the system is excited. For example, a frequency range of considerable practical interest for the purposes of the device is that between 20 kHz and 24 kHz, hence in the ultrasound spectrum.

COMSOL Multiphysics® FEM was essential for the project, in order to obtain the following results:

- design of the electromechanical (piezoelectric) transducer, with high power and efficiency
- design of the closing clamp
- optimization of the mechanical resonance of the assembly.

Resonance

As reported before, the invention relies on mechanical resonance in a mold or other structure comprising micromechanisms to be unlocked. Resonance is a phenomenon [3] in which a vibrating system or external force drives another system to oscillate with greater amplitude at specific frequencies. Frequencies at which the response amplitude is a relative maximum are known as the system's resonant frequencies. At resonant frequencies, small periodic forces have the ability to produce large amplitude oscillations, due to the storage of vibrational energy. Resonance occurs when a system is able to store and easily transfer energy between two or more different storage modes (such as kinetic energy and potential energy in the case of a simple pendulum). However, there are some losses from cycle to cycle, called damping. When damping is small, the resonant frequency is approximately equal to the natural frequency of the system, which is a frequency of unforced vibrations. Some systems have multiple, distinct, resonant frequencies.

For a lightly damped linear oscillator with a resonance frequency Ω, the intensity of oscillations I when the system is driven with a driving frequency ω is typically approximated by a formula that is symmetric about the resonance frequency:

Eq.1
$$I(\omega) \equiv |\chi|^2 \propto \frac{1}{(\omega-\Omega)^2 + \left(\frac{\Gamma}{2}\right)^2}.$$

The intensity is defined as the square of the amplitude of the oscillations. This is a Lorentzian function, or Cauchy distribution, and this response is found in many physical situations involving resonant systems. Γ is a parameter dependent on the damping of the oscillator, and is known as the *linewidth* of the resonance. Heavily damped oscillators tend to have broad linewidths, and respond to a wider range of driving frequencies around the resonant frequency. The linewidth is inversely proportional to the Q factor, which is a measure of the sharpness of the resonance (see later).

A physical system can have as many resonant frequencies as it has degrees of freedom; each degree of freedom can vibrate as a harmonic oscillator. Systems with one degree of freedom, such as a mass on a spring, pendulums, balance wheels, LC tuned circuits have one resonant frequency. Systems with two degrees of freedom, such as coupled pendulums and resonant transformers can have two resonant frequencies. As the number of coupled harmonic oscillators grows, the time it takes to transfer energy from one to the next becomes significant. The vibrations in them begin to travel through the coupled harmonic oscillators in waves, from one oscillator to the next.

Extended objects that can experience resonance due to vibrations inside them are called resonators, such as organ pipes, vibrating strings, quartz crystals, microwave and laser cavities. Since these can be viewed as being made of millions of coupled moving parts, they can have millions of resonant frequencies. The vibrations inside them travel as waves, at an approximately constant velocity, bouncing back and forth between the sides of the resonator. If the distance between the sides is d, the length of a roundtrip is $2d$. To cause resonance, the phase of a sinusoidal wave after a roundtrip must be equal to the initial phase, so the waves reinforce the oscillation. So the condition for resonance in a resonator is that the roundtrip distance, $2d$, be equal to an integer number of wavelengths λ of the wave :

Eq.2

$$2d = N\lambda, \qquad N \in \{1, 2, 3, \ldots\}$$

$$f = \frac{Nv}{2d} \qquad N \in \{1, 2, 3, \ldots\}$$

So the resonant frequencies of resonators, called normal modes, are equally spaced multiples of a lowest frequency called the fundamental frequency. The multiples are often called overtones. There may be several such series of resonant frequencies, corresponding to different modes of oscillation.

The Q factor or *quality factor* is a dimensionless parameter that describes how under-damped an oscilaltor or resonator is and characterizes a resonator's bandwidth relative to its center frequency. Higher Q indicates a lower rate of energy loss relative to the stored energy of the oscillator, i.e., the oscillations die out more slowly. To sustain a system in resonance in constant amplitude by providing power externally, the energy provided in each cycle must be less than the energy stored in the system by a factor of $Q/2\pi$. Sinusoidally driven resonators having higher Q factors resonate with greater amplitudes (and are more stable) but have a smaller bandwidth. The 'best' quality factor varies substantially from system to system : Systems for which damping is important (such as dampers keeping a door from slamming shut) have $Q = 1/2$. Clocks, lasers, and other systems that need either strong resonance or high frequency stability need high-quality factors. Tuning forks have quality factors around $Q = 1000$.

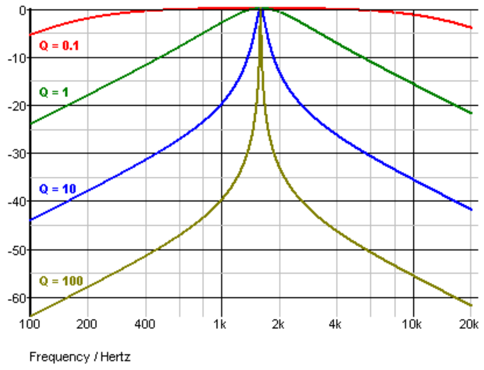

Figure 1: resonance and Q factors

Concerning the use of resonance in a system, it must be said that mechanical resonance may cause violent swaying motions and even catastrophic failure in improperly constructed structures including bridges, buildings, trains, and aircraft. Generally, when designing objects, engineers must ensure the mechanical resonance frequencies of the component parts do not match driving vibrational frequencies of motors or other oscillating parts, a phenomenon known as resonance disaster. Avoiding resonance disasters is a major concern in every building, tower, and bridge construction project. As a countermeasure, shock mounts can be installed to absorb resonant frequencies and thus dissipate the absorbed energy. In addition, engineers who design systems having engines must ensure that the mechanical resonant frequencies of the component parts do not match driving vibrational frequencies of the motors or other strongly oscillating parts.

Instead, the present invention aims at obtaining exactly the opposite result, that is, the maximum resonance vibration with the minimum absorption of forcing energy. Indeed micro-vibration can be the tool to unlock micromechanisms inside a structure subject to wear, filling, crusting, such as a rubber vulcanization mold. If the proper resonance frequency is excited using an electromechanical transducer, vibration close to each micromechanism makes it possibly rotate and vibrate, so that unlocking is achieved without any solvent, liquid, sanding or laser ablation procedure. Moreover, mold typically contains hundreds of micromechanisms, so that a local cleaning/unlocking procedure can be time-consuming while the present invention can achieve unlocking of at least 90% of the total number of micro mechanisms in 5 minutes of pulsed vibration.

The amplitude and frequency of the generated vibrations are optimized in relation to the type and characteristics of the mechanical structure to be treated. In an embodiment of the invention of particular practical interest, the aforementioned generations are in the ultrasound spectrum.

For example, a frequency range of considerable practical interest for the purposes of the present invention is that between 21 kHz and 24 kHz (with reference to the fundamental resonance frequency of the system), hence in the ultrasound spectrum.

Ultrasound Vibration System

The most important devices in the mechanical resonant station are :

- Voltage Generator
- 1 or 2 High Power Ultrasound Transducers
- Clamping Device

As regard the voltage generator that drives the electro-mechanical transducer, it is a high technology device (Titako generator, Unitech Srl [4]), equipped with a smart electrical impedance analyzer that ensures in real time the best operating conditions when the load for the transducer varies. Indeed such generator is autonomously able to perform a search for the optimal operating point, by varying the control frequency of the transducer and detecting the corresponding minimum electrical impedance and maximum $\cos(\varphi)$.

One or two opposite transducers are pressed on the mold by a clamping device that keep the assembly vibrating efficiently (see figure 2).

The electromechanical transducer is based on a standard Langevin structure, but with many design details and innovations to get the maximum efficiency. It is based on a stack of six Hard-PZT-8 piezoelectric rings, pre-stressed through a central bolt and operating electrically parallel, mechanically series. The stack is stressed between a ‘Frontmass’ and a ‘Backmass’ : the design and material choice for the two metallic masses is essential for the tuning and efficiency of the transducer.

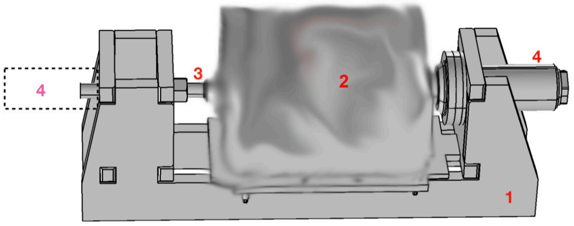

Figure 2: ultrasound resonance station. 1: frame, 2: mold, 3: clamping device, 4: transducer

Modal analysis

For the purpose of the present invention, the first study step to perform is a structural analysis of the mold, to get the frequencies of the strongest resonances.

Such analysis is easily performed with Comsol Solid Mechanics module, where both Eigenfrequency and Frequency Domain analysis are available.

For the most basic problem involving a linear elastic material which obeys Hooke’s Law, the matrix equations take the form of a dynamic 3D spring mass system.The generalized equation of motion is :

$$[M][U''] + [C][U'] + [K][U] = [F]$$

Where : $[M]$ is the mass matrix, $[U'']$ is acceleration, $[U']$ is velocity and $[U]$ is the displacement vector, $[C]$ is the damping matrix, $[K]$ is the stiffness matrix and $[F]$ is the force vector.

For vibrational modal analysis, the damping is generally ignored, leaving only :

$$[M][U''] + [K][U] = 0$$

This is the general form of the eigensystem encountered in structural engineering using the FEM. To represent the free-vibration solutions of the structure, harmonic motion is assumed, so that :

$$[U''] = \lambda[U]$$

And the equation above reduces to the linear transformation:

$$[M][U]\lambda + [K][U] = 0 \qquad \text{Eq.3}$$

where λ is an eigenvalue.

The physical interpretation of the eigenvalues and eigenvectors which come from solving the system are that they represent the resonant frequencies and corresponding mode shapes. Sometimes, the only desired modes are the lowest frequencies because they can be the most prominent modes at which the object will vibrate, dominating all the higher frequency modes.

To perform an Eigenfrequency analysis of the mold, a Comsol model was developed directly from the original mechanical drawing, with no constraints.

It was found that a frequency range of considerable practical interest for the purposes of the present invention is that between 21 kHz and 24 kHz (with reference to the fundamental resonance frequency of the system), hence in the ultrasound spectrum.

Next the transducer and frame-clamp assembly were designed, and the most important part of the system is clearly the electro-mechanical transducer, that excite the resonance of the complete system. As reported before, the transducer is based on a prestressed stack of six Hard-PZT-8 piezoelectric rings, that work electrically parallel, mechanically series. A sketch of such transducer is reported in figure 3. The most important parts to design in such device are the two metallic masses that enclose the piezoelectric stack, since they are responsible for : tuning, efficiency and symmetry (or desired asymmetry) of operation of the transducer. They are generally called backmass and frontmass and, in the present design, are made of Aluminum and Titanium respectively.

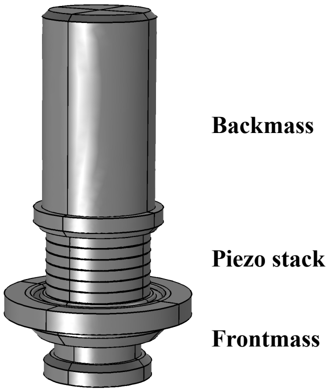

Figure 3: Ultrasound Transducer

A detailed FEM was developed for the transducer, making use of Comsol Solid Mechanics-Piezoelectric coupled modules and a frequency analysis was performed to achieve the best design in terms of high efficiency and clear resonance.

As regard piezoelectricity, the constitutive equations for the material are [5], in *stress-charge* form :

$$\begin{cases} \mathbf{T} = \left[\mathbf{c}^{\mathrm{E}}\right] \mathbf{S} - \left[\mathbf{e}^{t}\right] \mathbf{E} \\ \mathbf{D} = \left[\mathbf{e}\right] \mathbf{S} + \left[\boldsymbol{\varepsilon}^{\mathrm{S}}\right] \mathbf{E} \end{cases} \qquad \text{Eq.4}$$

where **T** is the stress vector, **c** is the elasticity matrix, **S** is the strain vector, **e** is the piezoelectric matrix, **E** is the electric field vector, **D** is the electric displacement vector, **ε** is the dielectric permittivity matrix. The superscripts indicates a zero or constant corresponding field. Eq.(4) take into account both piezoelectricity, both mechanical and electrical anisotropy of the material. Once these matrices have been specified, COMSOL recognizes which equations domains are to be used inside the FEM elements. Hard PZT-8 type ceramic is chosen as active medium, due to its exceptionally high efficiency and *Q* factor.

Model description

A FEM for the resonant station was built using COMSOL Multiphysics® 5.3a, making use of Solid Mechanics and piezoelectric modules, in 3D space dimension. Mesh on all domains was chosen as free tetrahedral and element size was always less than a fifth of signal wavelength. This allows a good compromise between accuracy and computational time. In order to further reduce computational time, symmetry was exploited and FEM was limited to half of the structure, as reported in figure 4.

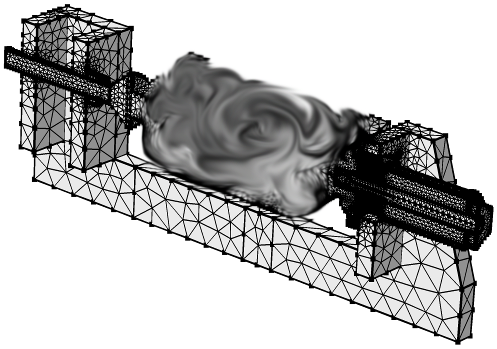

Figure 4: Mesh and symmetry

Boundary conditions are : symmetry on vertical cut plane, fixed constraint on lower part of the clamp. Piezoelectric Material and Electrostatics were imposed on the six piezoceramic rings inside the transducer, were ground and electric potential were set alternately to get electrical parallel connection.

Simulation Results

The most important results of simulations are reported here.

First the Eigenfrequency analysis of the selected mold alone was performed, with the following result for the average vibration on the top surface of the mold :

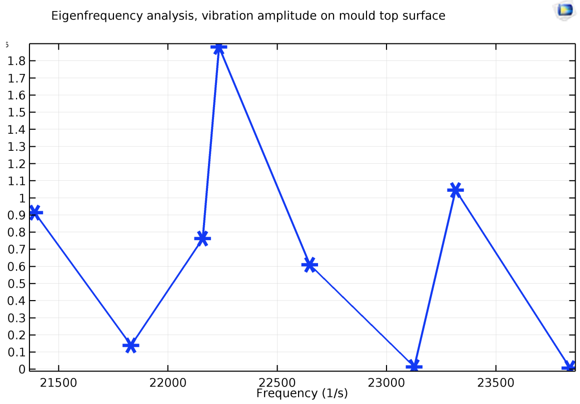

Figure 5: Eigenfrequency analysis

Clearly a strong resonance can be excited at approx. 22.2kHz and the ultrasound transducer was tuned accordingly. Indeed these molds exist in various sizes and the resonance frequencies vary accordingly to their dimensions and mass, in a range from 21 to

24kHz. The results reported here refer only to an aluminum mold with medium size (width=274mm, arc lenght=253mm, height=71mm), for briefness.

Then a frequency analysis was performed for the complete assembly, with a frequency sweep from 21 to 24kHz for the ultrasound transducer driving voltage, in order to record the most important resonances.

The average vibration calculated on the top surface of the mould is reported in figure 6, vs. frequency.

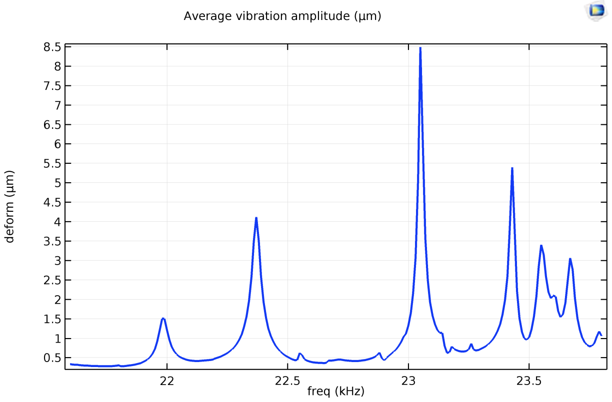

Figure 6: Average of vibration amplitude (µm) on mould top surface

And a deformation map of the structure at resonance (approx. 23kHz) follows :

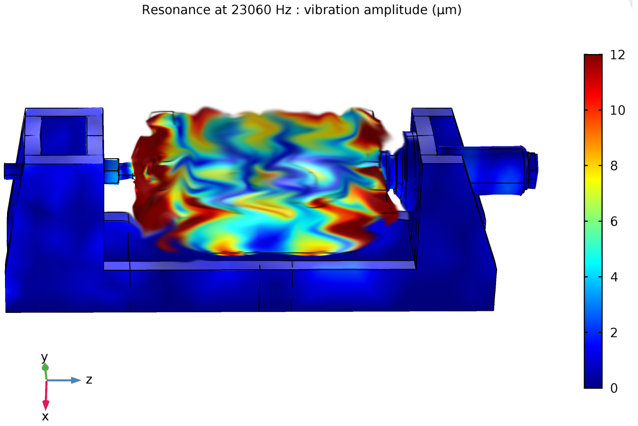

Figure 7: structure deformation (µm) at resonance

From figure 6 it's clear that a strong resonance is present at approx. 23kHz , with an average vibration amplitude of the mold close to 9 microns.

Figure 7 shows that the regions of maximum vibration, at resonance, are inside the mold and not on the external structure, as desirable.

The FEM simulation was essential to optimize the vibration of the system, but it's not easy to establish whether the simulated vibration is sufficient for the release of the micromechanisms in the mold : an experimental test is mandatory.

Prototype and experimental results

A prototype was manufactured according to the FEM design, in order to validate both the model and the invention. The prototype can be seen in figure 8, where the items can be recognized referring to fig.2.

The experimental procedure is the following :

First the dirty mold (with 20-30% of jammed micromechanism) is clamped between the transducer and the opposite support stop (or second transducer) with the appropriate clamping force. Then the generator is switched on, a frequency sweep and a scan of the possible resonances are performed and the generator chooses the best operating point, that makes the system resonate with high amplitude.

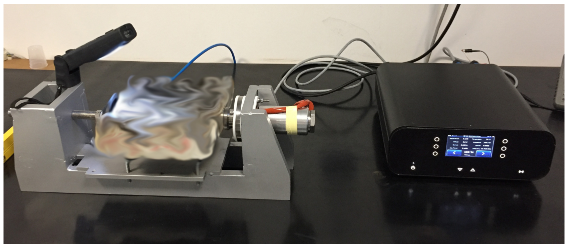

Figure 8: Prototype : resonant station and generator

The optimum resonance frequency found from such procedure was always close to the simulated one, with a discrepancy of less than 5%.

Moreover, some optimization tests were performed in terms of :

- driving power
- continuous vs. pulsed
- clamping force
- vibration time

The best operating condition was found to be pulsed (burst of vibration with short breaks), with very high clamping force (3000-5000N, depending on mold dimensions and weight) and a total time of vibration of 5min. The average driving power was approx. 300W.

Thanks to the accurate FEM design, at least 90% of all micro-mechanisms are always unlocked and clearly vibrate (the remaining ones are often broken or excessively filled with dirt), allowing to say that the invention leads to a qualitative leap in the unlocking process with respect to current manual techniques, which are more laborious and time-consuming.

Conclusions

A 3D FEM was developed for a mechanical resonant station that allows the easy unlocking of jammed micromechanisms inside a mold. The station consists of an electromechanical transducer and a closing clamp, which operate in such a way that at least one mechanical resonance mode of the system is excited. A frequency range of considerable practical interest for the purposes of the device was found between 21 and 24 kHz, hence in the ultrasound spectrum.

COMSOL Multiphysics® was essential for the design of the invention, in order to :
- design and tune the electromechanical transducer, with high power and efficiency
- optimize the mechanical resonance of the assembly

In the end it was possible to manufacture a prototype where at least 90% of jammed micromechanism were unlocked, after 5min. of pulsed vibration.

References

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