AUTOMATED DESIGN AND OPTIMIZATION OF RECTANGULAR PLATE SONOTRODES FOR SQUEEZE FILM LEVITATION

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Keywords: Ultrasonic, mode-tracking, modal assurance criterion, finite element method

Abstract. *The careful handling of sensitive workpieces is constantly gaining importance in production engineering. Ultrasonic non-contact handling through squeeze film levitation (SFL) is thereby a rather new and promising approach. The required ultrasonic systems consist of well known parts, like an ultrasonic transducer, an ultrasonic horn and a manipulator. As manipulators, either tonpilz sonotrodes are used for small workpieces or rectangular plate sonotrodes for larger workpieces. So far, only very little research has been conducted on the design of the latter. In the paper at hand, an automated design and optimization process for rectangular plate sonotrodes considering the specific demands of the SFL on the mode shape is presented and validated. After summarizing the basics of the squeeze film levitation as well as outlining the existing models and design approaches for the required components, the automation procedure with a simple optimization algorithm is introduced. Following a description of the employed finite element model, the modal assurance criterion (MAC) for the mode-tracking is introduced. Since the objective function of the optimization problem also uses the MAC for the evaluation of the mode shape, the synthesis of an idealized mode vector is described. Finally, the automated design and optimization process is validated through the experimental analysis of an exemplary plate sonotrode.*

1 INTRODUCTION

With the steady miniaturization of chips in everyday electronic devices, the need for gentle handling during their manufacturing and assembly constantly rises. In this context Vandaele [1] derived that for microassembly non-contact handling should be employed. Furthermore, [1] compared various different levitation techniques and deduced that non-contact ultrasonic handling through "acoustic levitation is the most suitable method for a wide range of applications".

1.1 Ultrasonic non-contact levitation

When talking about ultrasonic levitation, the two different principles standing wave levitation (SWL) and squeeze film levitation (SFL) need to be regarded [2]. The first effect (SWL) was originally developed by the space agencies NASA and ESA for the containerless processing in agravic environments. An acoustic standing wave is therefore generated between an ultrasound source and a reflector. In the pressure nodes of the resulting standing wave, small solid or liquid specimens can be placed and contactless restraint during experiments.

The second ultrasonic phenomena (SFL), which can be used for non-contact handling, is capable of handling larger and heavier objects than the SWL [3] and was introduced to production engineering by Hashimoto et. al. [4] in order to handle flat workpieces. When such a flat workpiece is brought close to an ultrasound emitting manipulator, a very small gap filled with a fluid (e.g. air) is formed. In the scientific literature different opposing explanation models concerning the governing physical effects can be found. While some models [4; 5] are based on acoustic effects, most of the publications agree on explanations from the field of fluid dynamics [6–8]. As the focus of this paper is not on the working principles of the squeeze film levitation, only the most dominant effect will be explained.

Since the ultrasound source performs an oscillating motion, perpendicular to its surface, the volume of the small gap between the manipulator and the workpiece periodically increases and decreases. Due to the high frequency, only the fluid at the boundaries is able to leave and re-enter the gap. Instead the vast majority of the fluid is "trapped" inside the gap and is being compressed and decompressed. When the small boundaries, which exchange fluid with the surrounding environment, are neglected, adiabatic conditions can be assumed for the remaining gap. Considering the non-linear relationship between the pressure and volume for adiabatic conditions depicted in Figure 1, the mean pressure p_m resulting from a harmonic variation of the volume surmounts the ambient pressure p_0. [6]
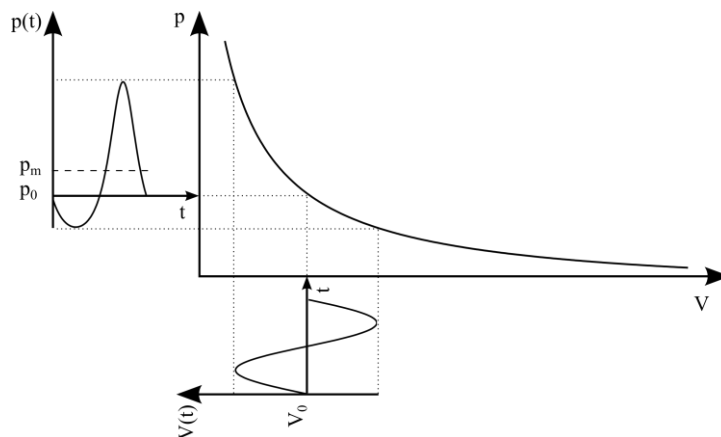

Figure 1: Contribution of the adiabatic compression to the SFL [6]

As depicted for the example in Figure 2, the handling devices used for the SFL most often consist of the same three components: a langevin bolt transducer (LBT) as ultrasound source, an ultrasonic horn to boost the amplitude and a plate sonotrode as manipulator.
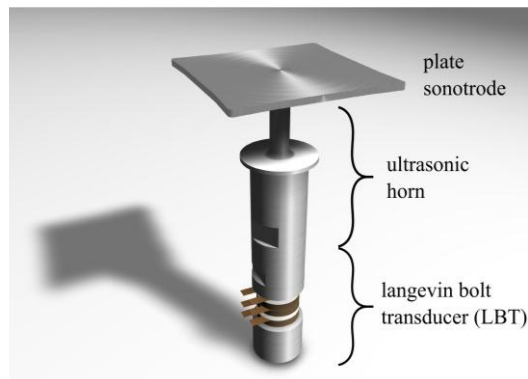

Figure 2: Ultrasonic handling system for SFL

Depending on the size of the workpiece and the driving frequency, the sonotrode either performs a piston-like rigid body movement or some sort of bending mode. The second possibility is the common operating mode when handling larger objects, in which case the ultrasonic horn is connected to an antinode of the plate sonotrode. If necessary, the plate sonotrode can be supported in additional nodes. For both connections a rigid and punctual contact is desirable. According to [9], the plate sonotrode should be dimensioned in such a fashion, that it completely covers the workpiece and therefore applies the levitation force uniformly over the entire surface of the workpiece. If the handling system is moved together with the workpiece, the dimensions should exceed those of the workpiece as little as possible to reduce the unused radiated ultrasound to a minimum.

1.2 Design of ultrasonic devices

In order to reduce the mechanical damping and maximize the vibration amplitude, the entire system should operate at its eigenfrequency. By assuming appropriate boundary conditions at the interface (e. g. floating bearings), the three components can be modelled and considered individually [10].

With various detailed models at hand, current research is committed to the optimization of ultrasonic transducers and horns. Simulating the ultrasonic transducer, horn and including a fluid, Heikkola et. al. [11] perform a multiobjective optimization of the horn geometry. Andrade et. al. [12] use the pressure field of a SWL as objective function to find the optimal geometry of a horn and reflector. Schorderet et. al. [13] couple a structural and electromechanical FEM simulation to optimize a parameterized ultrasonic system for a maximum displacement at a given voltage.

The third component, the plate sonotrodes, can be differentiated into circular and rectangular plates. The circular plates, also referred to as tonpilz sonotrodes, are often used for research of the SFL [6; 14; 15] and are one of the most popular types of sonar transducers [16]. Because their mode shape is axi-symmetric, one-dimensional analytical models can be used for their design [17]. In numerous other publications, for example [16; 18], the FEM is used for the design of tonpilz transducers.

In contrast to the widely spread tonpilz sonotrodes, only little research deals with the design of rectangular plates. For SFL the rectangular plates can either be designed as monolithic long transfer paths or short tiles. In the first case, usually a one-dimensional bending mode

shape, resembling that of a beam, is used [19; 20]. Examples for the design of such plates by the means of FEM can be found in [21; 22]. For the second case, the dimensions of the length and width are approximately equal which results in two-dimensional complex mode shapes. For the best support of the workpiece, an equal distribution of vibration amplitude over the entire plane is desirable [9]. The deformation of an exemplary plate sonotrode is shown in Figure 3 in a false-color plot, where white symbolizes maximal displacement and black no vibration. So far, no design method for the second kind of rectangular plate sonotrode has been proposed.
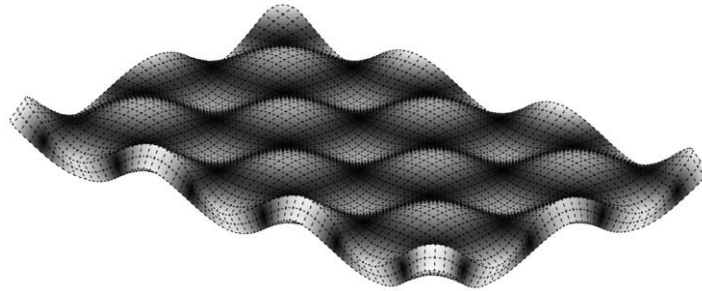
Figure 3: Exemplary target mode shape of a rectangular plate sonotrode

2 AUTOMATED DESIGN

As the desired vibration is of a rather complex shape and can be influenced by virtually all available parameters, a manual design is not feasible. Furthermore, the variable dimensions of all possible plates render a general analytical solution impossible as the required maximal ratio of thickness to length and width of 1/20 for the thin plate theory [23] cannot be guaranteed. For these two reasons, an automated numerical design and optimization of the plate sonotrodes is proposed by utilizing the FEM.

2.1 Automation procedure

To be able to focus on the overall design process, only a simple optimization algorithm, namely the random walk algorithm, is implemented in this paper as visualized in Figure 4.
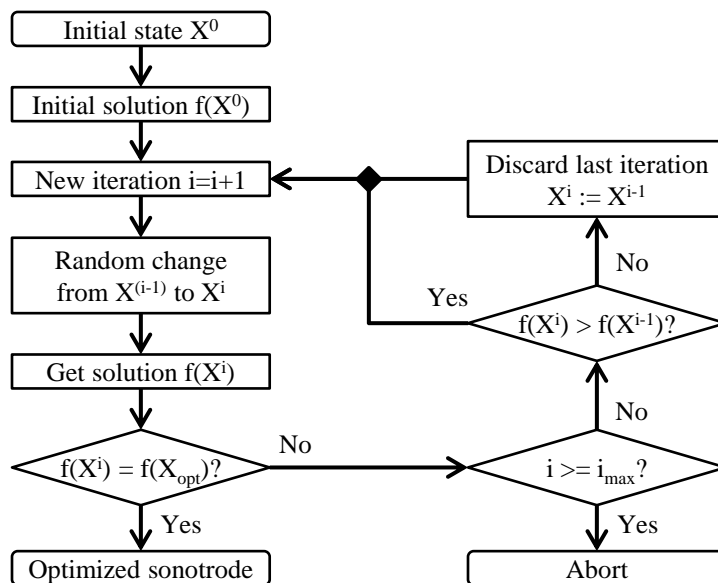
Figure 4: Flow chart of the automated optimization process

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Starting with a parameterized model, calculations for the initial solution are performed. In the following loop an iterative optimization is performed with random changes to the design vector. If a new state offers better results than the previous, the optimization is continued from there, otherwise the new state is rejected and the optimization is continued with new random changes from the old state. These iterations are repeated until either a solution is found, that satisfies the objective functions, or a maximum number of iterations is reached. For the initial solution and during each iteration, the following steps must be carried out. First the parameters for the current iteration are determined and the simulation model is generated. Then, the finite element model is solved and the required information is retrieved from the results. The last step is evaluating the objective functions and checking the fulfillment of the abort criterions.

While the optimization algorithm and all directly related functions are implemented in MATLAB, some tasks are performed in a commercial finite element program. For generating the simulation model, solving the problem and retrieving the required results, the commercial finite element program ADINA is used. As ADINA has no interface to MATLAB, all communication between these two programs has to be performed with text files. While ADINA can be execute from a command window with batch-files, MATLAB acts as a master and controls the overall program flow. The later is done by evaluating various log files and requesting ADINA to write certain information, like specific results, to text-files.

The powerful programming capabilities of MATLAB enable the creation of complex programs with various functions. In essence, the optimization program is compartmentalized into four sections: the control of the overall program flow, the model creation, the interface to ADINA and the evaluation of the results. As the interfaces between these sections are generalized, various functions from the different sections can easily be combined and new features can quickly be implemented in the optimization procedure.

2.2 Finite Element Model

First, a two dimensional sheet with the outline of the plate is created from the given parameters. The sheet is then automeshed with 2D-Solid 9-node rectangular elements of approximately the same size, which are assigned a simple linear isotropic material model. By extruding the two-dimensional mesh by the thickness of the plate, 27-node 3D-Solid elements are created. Through this method, the total number of degrees of freedom (DOF) can effectively be reduced compared to a common three-dimensional automeshing with tetrahedral elements, which in turn significantly boosts the simulation speed.

Besides some simulation specific parameters like mesh density and the material properties, the finite element model is created using only five geometric parameters, which are shown in Figure 5.
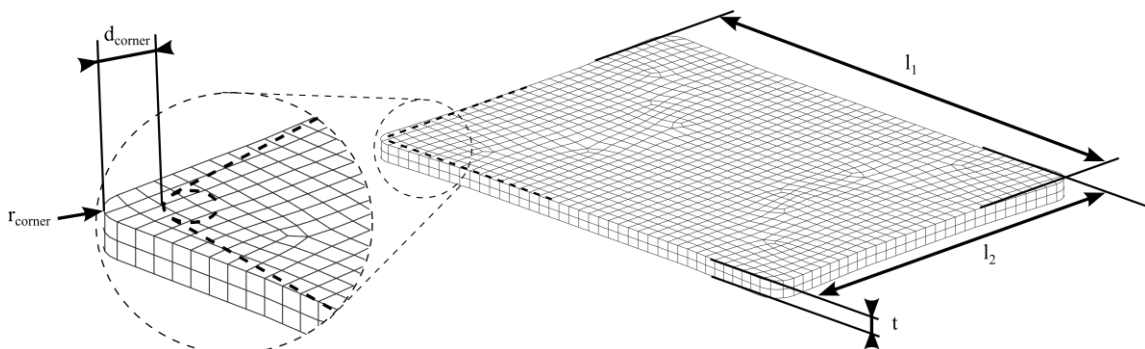

Figure 5: Geometric parameters of the plate sonotrode

The simulations are performed in modal space with free vibrations, without any boundary conditions. This approach is valid, as the ultrasonic horn will later be attached to an antinode at a small area, where almost no curvature exists.

2.3 Mode-tracking

When using modal analysis for the design and optimization of structures, specific eigenmodes need to be evaluated in multiple subsequent simulations. As the changes made to the structure between these simulations can cause crossing of eigenmodes in respect to their frequency, a so called mode tracking is necessary. One possibility for mode-tracking offers the modal assurance criterion (MAC). The MAC was first introduced by Allemang and Brown in 1982 "to provide a measure of consistency between estimates of a modal vector" and compare "estimates of modal vectors originating from different sources" [24]. When comparing modal vectors, they must be of the same length and their entries need to correspond to the same degrees of freedom (DOF). If the vectors originate from two different sources (a and b), one vector needs to be converted to the other sources modal space. This is usually done by rearranging and omitting some entries of the modal vectors. With the two complex valued modal vectors Φ_a and Φ_b, the MAC value is defined in Eq. (1).

$$MAC_{a,b} = \frac{|\Phi_a \Phi_b^*|^2}{\Phi_a \Phi_a^* \Phi_b \Phi_b^*} \tag{1}$$

Even so the MAC value is explicitly not defined as an orthogonality check, as the modal vectors are not weighted by the mass matrix, it can be interpreted accordingly. When its value is near zero and all prerequisites are met, the compared mode vectors belong to linearly independent eigenmodes. When the MAC value approaches unity instead, it offers a strong indication that the mode vectors originate from the same eigenmode. [24]

2.4 Optimization problem

The most common scenario is to design a plate sonotrode which works with a given ultrasonic LBT to transport a specific workpiece. As all components should operate in resonance, the LBT defines the target frequency of the plate sonotrode. The first objective function in Eq. (2) consequently is the minimization of the difference between the resonance frequencies of the LBT and the plate.

$$min\,|f_{LBT} - f_{plate}| \tag{2}$$

Closely related to the resonance frequency is the desired mode shape. While the objective is a regular distribution of nodes and antinodes with equal amplitude, as shown in Figure 3, the reality is often a rather distorted distribution with varying amplitudes, especially on the borders of the plate. An easy way to evaluate the mode shape, is the utilization of the MAC value already provided by the mode-tracking algorithm. For this purpose, the reference mode vector for mode-tracking Φ_{ref} is not taken from the first simulation run, but an idealized synthetic mode vector is used. The second objective function is formulated in Eq. (3).

$$max(MAC_{ref,target}) \tag{3}$$

In order to influence the resonance frequency of the plate, the material properties and all five geometric parameters of the model can be adjusted. The material and plate thickness are usually selected by process and production criteria and therefore not available as a design variable. The other geometric variables however, can be changed within certain boundaries and are well suited as design variables. The design vector v is defined in Eq. (4) as

$$v = \begin{Bmatrix} l_1 \\ l_2 \\ d_{corner} \\ r_{corner} \end{Bmatrix}, \tag{4}$$

with design constraints derived from the earlier mentioned requirements of the plate size.

While all four design variables are capable of influencing both objective functions, they possess a different impact. The main plate dimensions l_1 and l_2, have a significant effect on the resonance frequency and mostly scale the mode shape, whereas the design of the corners mostly effects the distribution of the antinodes and only slightly changes the frequency. These different behaviors enable a separation of the two objective functions and significantly reduce the effort required for the optimization. The two objectives, resonance frequency and mode shape, are pursued in an iterative scheme with two optimization cycles as depicted in Figure 4 and only half the original design vector each. While an improvement of one objective function might result in a deterioration of the other, the different impact of the variables guarantees a convergence.

2.5 Synthetic mode vector

Because of the variable dimensions of the plate sonotrode, the number of antinodes in each direction are unknown and every reference mode vector must be created individually. Furthermore, the exact locations of the antinodes must be determined as accurately as possible. Otherwise the mode vector of the simulation might not include the maximum value of each antinode when the optimal mode shape is reached and the optimization algorithm pursues a distorted mode shape.

The first step is selecting the DOF to be included in the mode vectors. Since the target mode shape of the plate sonotrode is characterized through the even distribution of antinodes, all antinodes on the plate can be connected with straight lines, parallel to the plate edges. Additional lines can then be inserted in between every two lines, marking the nodes of the plate. While nodes cannot positively contribute to the nominator of the MAC, the lack of a node in only one mode vector increases the denominator and therefore degrades the overall MAC value. Consequently all DOF at all grid positions shall be included in the mode vector.

So far, however, the number of antinodes in each dimension of the plate and their locations are still unknown. Again, the key lies in the regular distribution of the antinodes. A good approximation of the antinode spacing at the center of the plate can be obtained from an infinite plate, by considering an area with the antinodes of the two-dimensional bending vibration at the corners, and nodes at the center of the edges and the center of the area. By definition this edge length equals the half wavelength of the bending vibration. Along the edges of this area, no in-plane displacement or curvature can be observed, as the neighboring areas of the same size must have equal and compatible boundaries. When placing boundary conditions at the edges of the area, restricting the translational and rotational DOF accordingly, the area can be isolated and simulated separately. With this model, the appropriate wavelength for a target frequency can easily be determined by varying the edge length.

Before each simulation run, starting at the very center of the plate, the grid for the mode vectors is created in steps of a quarter wavelength. However, when approaching the edge of the plate, the spacing between the antinodes decreases due to the free edge and the dimensions of the plate are not necessary a multiple of the quarter wavelength. Therefore the creation of grid-lines is stopped just outside of the plate, and the outer most grid-lines are then moved to coincide with the edges of the plate. For a smoother transition, the last gridlines on the plate are moved midway between the edges and the last but one gridlines on the plate.

Starting at any corner the gridlines are numbered subsequently in both directions. While grid positions belonging to two uneven numbered gridlines are antinodes, alternating in direction respective sign, all grid positions belonging to at least one even numbered gridline are nodes. The main steps of the synthesis of the idealized mode vector, are depicted in Figure 6.
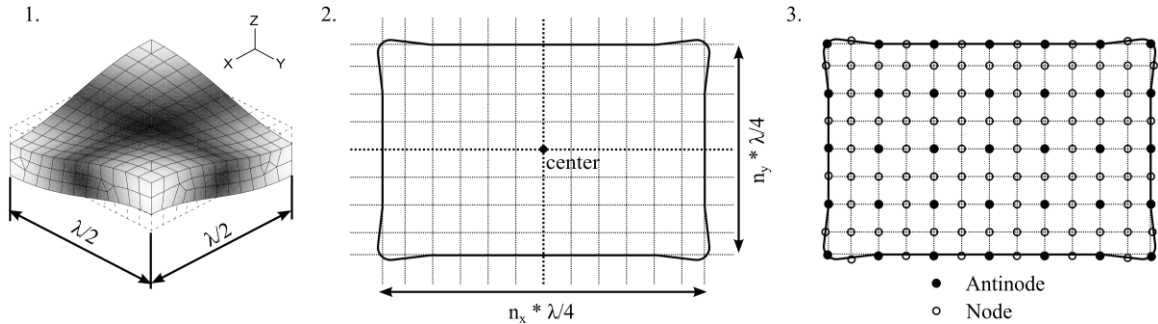

Figure 6: Synthesis of the idealized mode vector

3 EXPERIMENTAL VALIDATION

By means of the presented automated optimization process, an exemplary rectangular plate sonotrode was designed. Starting with a LBT and ultrasonic horn at a resonance frequency of 35,021 Hz, a circular workpiece of 140 mm in diameter was assumed. Due to its excellent ultrasonic properties and its good processability, the aluminum alloy EN AW 5083 was chosen as plate material. The automated optimization took about 80 minutes on an up-to-date workstation and delivered a plate of roughly 148 mm edge length.

In order to validate the separate design process of the plate sonotrode, the plate was then manufactured and assembled with the LBT and horn. During the experimental validation the excitation was performed by applying sweep signals of 10 Volts peak-peak to the LBT. The system response was measured using a 3D scanning vibrometer of the type PSV-400-3D from Polytec utilizing the laser Doppler effect, capable of synchronizing the measured vibrations from various locations and visualizing the vibrations of the plate in the frequency domain. Table 1 offers a comparison of the simulation and experiment with the amplitudes of the vibrations represented by false-color plots.
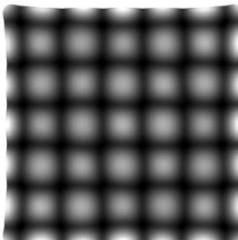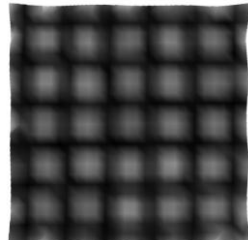
	Simulation	Experiment
Resonance frequency	35010 Hz	35113 Hz
Deviation to LBT	-11 Hz (0,031 %)	92 Hz (0,263 %)
Mode shape		

Table 1: Experimental validation of an optimized plate sonotrode

Considering the large influence of small variations of material properties and manufacturing tolerances on the resonance, the relatively small deviation of the first objective function can be explained by multiple reasons. Furthermore, even the heating of the LBT during its

application usually results in resonance shifts of more than 100 Hz. The second objective function, the regular form of the mode shape, is also very well fulfilled. As can be seen, the antinodes are distributed regularly and are of approximately the same amplitude. All together, a good agreement between the simulation and the experiment can be stated.

4 CONCLUSION

Rectangular plate sonotrodes are of great importance to the squeeze film levitation. Unfortunately their design goals cannot be appropriately included in simplified models and very little research on their design has been conducted so far. Therefore, in this paper, an automated design and optimization process for rectangular plates has been presented. One of the main obstacles, the evaluation of the mode shape, was successfully conquered by using the MAC on the simulation results and an idealized synthetic mode vector. Finally, the automated process was validated at the example of a small plate sonotrode.

Further work will be needed to predict the required amplitude for the squeeze film levitation of a workpiece and include this as an additional objective function in the design process.

5 ACKNOWLEDGEMENTS

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