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Technical paper

Methodology for shape optimization of ultrasonic amplifier using genetic algorithms and simplex method

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ABSTRACT

Designing devices for ultrasonic vibration applications is mostly done by intuitively adjusting the geometry to obtain the desired mode of vibration at a specific operating frequency. Recent studies have shown that with optimization methods, new devices with improved performance can be easily found. In this investigation, a new methodology for designing an ultrasonic amplifier through shape optimization using genetic algorithms and simplex method with specific fitness functions is presented. Displacements at specific functional areas, main functionality, and mode frequency are considered to determine the properties of an individual shape to meet the stated criteria. Length, diameter, position of mountings, and further specific geometric parameters are set up for the algorithm search for an optimized shape. Beginning with genetic algorithms, the basic shape fitting the stated requirements is found. After that the simplex method further improves the found shape to most appropriately minimize the fitness function. At the end, the fittest individual is selected as the final solution. Finally, resulting shapes are experimentally tested to show the effectiveness of the methodology.

1. Introduction

Recent developments in ultrasonic vibration assisted machining call for new designs of tools, horns, boosters, and other components. While a wider application range of the hybrid technology of ultrasonic vibration assistance for many manufacturing technologies is of great interest, one key challenge is the design of the system in order to achieve the desired resonance vibration. The most common mode used in ultrasonic-vibration-assisted machining is the longitudinal mode of an axisymmetrical horn, which can be easily obtained when solving the Webster Horn Equation [1].

$$\frac{\partial^2 u}{\partial z^2} + \frac{1}{A(z)}\frac{dA(z)}{dz}\frac{\partial u}{\partial z} = \frac{1}{c^2}\frac{\partial^2 u}{\partial t^2} \tag{1}$$

where t is time, u is displacement, $A(z)$ is the cross sectional area as function of position z, and c is the acoustic velocity. The acoustic velocity c can be obtained with

$$c = \sqrt{\frac{E_Y}{\rho}} \tag{2}$$

where E_Y is the material's Young's modulus, and ρ the material density. In case of a harmonic motion, Eq. (1) can be rewritten as

$$\frac{\partial^2 u}{\partial z^2} + \frac{1}{A(z)}\frac{dA(z)}{dz}\frac{\partial u}{\partial z} = \frac{\omega^2}{c^2}u \tag{3}$$

where ω is the angular frequency. Using Eq. (3) the length of axisymmetric horns can easily be calculated given a specific resonance frequency. The design of tool holders and horns can be obtained by solving the above equations [2–4] for various $A(z)$. Based on these findings, a great variety of axisymmetric horns has been found and are used in the industry today. In medical engineering, a novel ultrasonic vibration tool for surgery has been designed and tuned to the appropriate frequency for the optimal configuration [5]. A percussive drill system was designed for rock coring on planetary robotic missions using ultrasonic vibration assistance to reduce power and torque requirements [6]. More challenging are new designs for ultrasonic vibration assisted machining by combining two modes for operational purposes [7]. The longitudinal-torsional composite mode allows advanced applications for machining like drilling. Tsujino et al. designed a one-dimensional longitudinal-torsional vibration converter using diagonal slits within the resonating structure [8]. Designing transducers for ultrasonic assisted wire bonding with finite element method has been discussed in [9] with the goal of matching simulation results with experimental results. Enhancing vibration performance and matching simulation with experimental results has also been discussed in [10] for ultrasonic block horns. Properly

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designing a rotary ultrasonic milling tool with finite element method is introduced in [11]. For many of the mentioned designs of horns and ultrasonic vibration components, intuitive design strategies were used by evaluating the nodal displacements of modes simulated with an FEM software. A common non-automated design procedure can be found in [12], which outlines the step-by-step procedure to manually design an ultrasonic device. Automating the intuitive/manual design process can be done by shape optimization.

Many optimization methods have been applied for finding shapes and structures that provide good results for ultrasonic vibration applications. Combining multi-objective decision making such as the NIMBUS method with the finite element method can provide very good designs as shown in [13]. Another very good optimization was introduced in [14] to find advanced transducer designs while satisfying conflicting optimal values in the design space. In [15], design of experiments is used to find the correct parameters for an ultrasonic linear motor and perform a sensitivity analysis for each parameter and their interactions with each other. Based on Eq. (1), a ultrasonic horn optimization method is done in [16]. Porto et al. developed a genetic algorithm to optimize the amplitude of a surgical ultrasonic transducer by changing the length of specific geometric parts at a given frequency [17].

Designing the components for ultrasonic vibration assisted machining is generally challenging, because the high frequency vibrations need to precisely occur at the tool edge or a preferred location. While the maximum amplitude is desired at the tool, minimal vibration should occur at the clamping or mounting of the ultrasonic vibration device. In this investigation, a new methodology for a shape optimization [18] of ultrasonic vibration amplifiers and reducers using generic algorithm (GA) and simplex method [19,20] is introduced. Since the GA is capable of searching for the optimum of the entire design space for non convex problems, it serves as a global search method. The subsequent simplex method, as a local search method, is used to further refine the found optimum. The optimized shapes are experimentally tested by conducting an FFT Analysis and measuring displacements using a laser vibrometer.

2. Optimization methodology and parameterization

2.1. Methodology

To optimize the shape of a structure via shape optimization, specific parameters need to be altered while the optimization algorithm tries to satisfy the defined criteria. Such criteria can be stress levels, compliance, volume and others that are to be maximized or minimized. As mentioned earlier, most of the ultrasonic vibration assisted machining use the longitudinal resonance mode to impose oscillations to the tool. For this optimization, the longitudinal resonance mode in z-direction is described by the displacements of specific structural parts when exciting the structure with the operating frequency of $\omega = 35$ kHz with an amplitude of $x_0 = 10\,\mu$m. This frequency is set by the generator (see Section 3) used later in the experiments. The GA uses a fitness function that is to be minimized in order to find the optimal solution to the defined problem. For designing the structure, the finite element method (FEM) results of a modal analysis [21] and harmonic response analysis [22] are evaluated based on the displacements of the mesh nodes at the specific structure parts like cutting edge and mountings. The nodal displacement for describing the modal shape is used, because vibration modes are usually graphically evaluated (e.g. [12]) through the nodal displacements. Therefore, the global fitness function is made up from the following set of dimensionless functions. At the functional area or tool edge, the sum of displacements in z-direction is to be maximized while the sum of displacements in y-direction and the sum of displacements in x-direction are to be minimized, as stated by

$$f_1 = \frac{\sum_{i=0}^{n}|u_{xi}| + \sum_{i=0}^{n}|u_{yi}|}{\sum_{i=0}^{n}|u_{zi}|} \overset{!}{=} min \tag{4}$$

where u_x is the nodal displacement in x-direction, u_y is the nodal displacement in y-direction, and u_z is the nodal displacement in z-direction as shown in Fig. 1. With this function, bending or torsional modes will have a higher fitness value than the longitudinal mode. To ensure an even displacement across all mesh nodes at the functional area, the sum of difference between the displacement of a single node and the average displacement in z-direction needs to be minimized, which is stated by

$$f_2 = \frac{\sum_{i=0}^{n}(|u_{zi}| - |\bar{u}_z|)}{|\bar{u}_z|} \overset{!}{=} min \tag{5}$$

with

$$\bar{u}_z = \frac{1}{n}\sum_{i=0}^{n} u_{zi} \tag{6}$$

being the average displacement in z-direction. Considering the overall longitudinal mode and the placement of the mountings needed for fixing the device, it is necessary to position the mountings at the vibration nodes of the device, because minimal vibration should occur at the mountings. This is done with

$$f_3 = \frac{|x_{m1} - x_{n1}|}{|x_{n1}|} + \frac{|x_{m2} - x_{n2}|}{|x_{n2}|} \overset{!}{=} min \tag{7}$$

where x_{m1}, x_{m2} are the positions of the mountings 1 and 2, and x_{n1}, x_{n2} are the locations of the vibration nodes 1 and 2. The values for the location of the vibration nodes are selected by finding two minima of the displacement values in z-direction along the axis of rotation. Since the input amplitude is 10 μm and a booster or amplifier aims to increase the amplitude while an amplitude reducer aims to decrease the amplitude of an ultrasonic vibration system, the fitness function

$$f_4 = \left| A_R - \frac{|\bar{u}_{zo}|}{|\bar{u}_{ze}|} \right| \overset{!}{=} min \tag{8}$$

is formulated with A_R being the amplitude ratio between input and output, $\bar{u}_{zo}$ is the average amplitude in z-direction at the output, and $\bar{u}_{ze}$ is the average amplitude in z-direction at the input. While a harmonic response analysis is used to obtain the results for the so far described fitness functions, a modal analysis is additionally performed to obtain the resonance frequency. Ideally, the ultrasonic device resonant frequency or mode should be equal to the operating frequency of the transducer (Section 3). This condition is stated by

$$f_5 = \frac{|\omega_s - \omega_t|}{\omega_t} \overset{!}{=} min \tag{9}$$

where ω_s is the frequency of the vibration mode and ω_t is the targeted frequency, which is 35 kHz for this ultrasonic device. Here, Eqs. (5), (7) and (9), are normalized to remove any dimensions.

The global fitness function for the GA optimization results in

$$f = \sum_{i=1}^{6} R_i f_i \overset{!}{=} min \tag{10}$$

where R is a penalty factor which can be specified for each individual fitness functions. For the following optimization, the penalty factors are set to $R_1 = 10$, $R_2 = 10$, $R_3 = 10$, $R_4 = 100$, and $R_5 = 10$. These factors are chosen based on relevance of each function. For the present optimization, ensuring the correct amplification (f_4) is