Sensors and Actuators A 171 (2011) 260–265

Contents lists available at ScienceDirect

Sensors and Actuators A: Physical

journal homepage: www.elsevier.com/locate/sna

Modeling characterization and optimization design for PZT transducer used in Near Field Acoustic Levitation

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ARTICLE INFO

Article history:
Received 8 December 2010
Received in revised form 17 June 2011
Accepted 26 June 2011
Available online 19 July 2011

Keywords:
Acoustic levitation
FEA
Coupled field
Size effect
Optimization

ABSTRACT

An acoustic actuator with large radiation surface has been used in a non-contact levitation system. The actuator is composed of a Langevin transducer and a disk-shape radiator. A comprehensive modeling approach has been developed to simulate electrical and mechanical characterization. A coupled-field finite element analysis has been performed considering the vibrator with flexural mode. Identical experimental conditions were used in the simulation to investigate size effect and optimize the transducer design. The stimulated results show good agreement with experimental measurements. This model may be employed to facilitate the analysis and optimization of Near Field Acoustic Levitation transducer assemblies. Moreover, this paper demonstrated the important role that modeling plays in the design, fabrication and optimization of coupled transducers.

1. Introduction

Piezoelectric transducers are widely used to send and detect signals in many areas, such as nondestructive evaluations, acoustics, medical imaging and in many other ultrasonic applications [1]. Among those applications, acoustics has progressed considerably during the last decade, and piezoelectric ceramics are commonly used as the driving source to produce waves [2]. Near Field Acoustic Levitation (NFAL) is used in noncontact handling and transportation of small objects to avoid contamination. The major advantage of NFAL lies in the fact that any material, insulator or conductor, magnetic or non-magnetic, can be manipulated by acoustic levitation and transportation [3].

NFAL is used to handle planar objects slightly above the manipulator surface which is connected to a high-intensity acoustic vibrator. The actuator for noncontact levitation consists of one classical Langevin transducer and a vibration disk with a large diameter. The gas squeeze film, which is created between the acoustic vibrator and the planar object by rapid vibrations, generates a time-averaged levitation force. This type of acoustic transducer has many advantages, including resonance in flexural mode with relatively low frequency, large radiation surface, and improved impedance matching. Since Prof. Langevin developed the first sandwich ultrasonic transducer by embedding piezoelectric rings between two pieces of metal and employed it for high intensity vibration, there have been intensive efforts in modeling and formulating such transducers. For fundamental modeling, Mason had proposed the Equivalent Circuit Method (ECM) [4], while Castillo [5] built a one-dimensional KLM model that takes into account laterally clamping for a piezoelement. When studying thick high power ultrasonic radiators, Mori and Tsuda [6] innovated a method called Apparent Elasticity Method. Recently, Abdullah [7] provided a model with correct prediction, but the model did not consider the working head. Hirase et al. [8] had explained the variation of high-power PZT transducer parameters and mechanical losses. Tressler [9] and Uzgur [10] investigated the effects of material properties and dimensional changes on the fundamental resonance frequency of cymbal transducer using ANSYS. In all published works on conventional Langevin transducers, the working head has not been included in the dynamic modeling. However as used in NFAL, the working surface of the transducer should be large enough to hold a planer object. The lateral dimension of the working head is much larger than the transducer. Therefore the effect of the head to the whole apparatus should not be ignored. It has been observed that the radiation behavior is affected not only by physical and mechanical characteristics of elements and axial dimensions, but also by the lateral dimensions and cross section shape of the transducer. Only when maximum diameter of the transducer is less than a quarter of the sound wavelength, the lateral effect can be ignored

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doi:10.1016/j.sna.2011.06.020

[2]. Additionally, Liu et al. [3] studied the levitation force induced by squeeze film and found that the vibration mode shape of the radiator is crucial to predict the nonlinear levitation behavior. Therefore, it is important to study a coupled system concerning both longitudinal motion of transducer and flexural mode of vibrator. The mechanism and the vibration characteristics of this kind of coupled transducers have not been studied up to date.

In this work, a special type of piezoelectric transducer has been modeled and optimized. General constitutive relations and dynamic equations for a couple-field transducer have been determined. Since transducer characteristics must be predictable, the analysis of these transducers is essential in the design stage [11]. The coupled transducer has been modeled and identical experimental conditions have been simulated by using ANSYS code program. Conventional working head has a dimension no more than the transducer, but in our case, the work surface is much larger than the cross section of the transducer, which generates lower frequency flexural mode. Modal and harmonic analysis have been performed for electrical and mechanical characteristics. Working mode has been chosen based on the maximization of electromechanical conversion efficiency and energy radiation. The relationship between the resonance frequencies and the geometrical dimensions of transducer has been examined, and the effect of the location of fixed point has been discussed which provides an efficient and versatile means for optimization design of the transducer. Very good agreement has been obtained between the modeled and experimentally measured impedance spectra in the frequency range up to 40 kHz, confirming the validity of the model. Compared with conventional transducer modeling, our model and simulation results exhibited a remarkably high predictive potential, which allowed the control of the device geometry and the vibration behavior of the acoustic transducer used in the NFAL.

2. Governing equation

Fig. 1 shows the dimensions of an ultrasonic transducer. It contains of a bolt-clamped Langevin type transducer, a horn and a sound radiation surface. The Langevin transducer is basically composed of a couple of PZT rings sandwiched between two metal

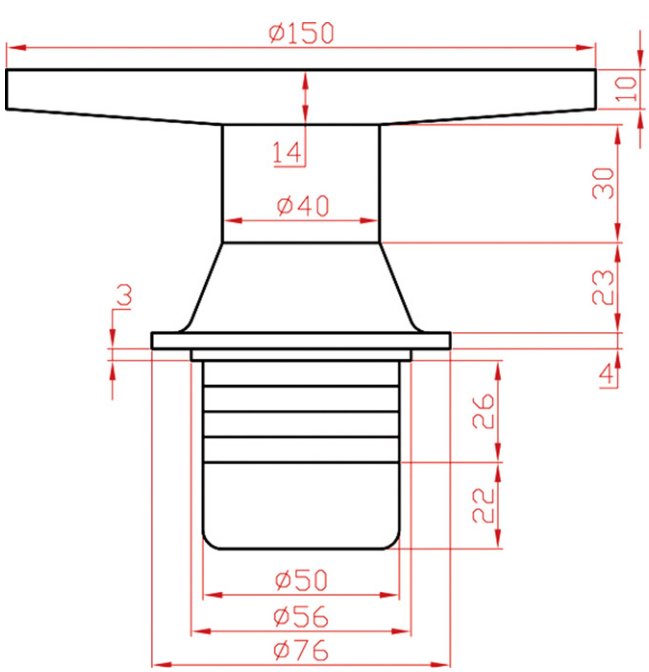

Fig. 1. The dimensions of an transducer used in NFAL.

Table 1
Matrices for piezoelectric-structure coupled system.

Matrix	Description
$M_{UU} = \int_V \rho_S N^T N dV$	The kinematical constant mass matrix
$K_{UU} = \int_V B^T c^E B dV$	The elastic stiffness matrix
$K_{U\Phi} = \int_V B^T e^T B_E dV$	The piezoelectric coupling matrix
$K_{\Phi\Phi} = \int_V B_E^T \varepsilon^S B_E dV$	The dielectric stiffness matrix
$D_{UU} = a\int_V \rho_s N^T N dV + b\int_V B^T c^E B dV$	The damping matrix a, b: Damping coefficients
$F = \int_V N^T N_{FB} f_B dV + \int_\Gamma N^T N_{FS} f_S d\Gamma + N^T f_P$	External force: body force, surface force, point force
$Q = -\int_\Gamma N_E^T N_{QS} q_S d\Gamma - N_E^T q_P$	Electric charge: surface and point electric charge

masses. In our case, there are four PZT rings in the middle of the transducer, with 25 mm radius and a thickness of 6.5 mm each. The PZT elements are excited to resonate in length-extensional mode at low frequency, avoiding the need of high driving voltage. The structure is usually pre-stressed in order to increase the mechanical strength of PZT elements and to endure high electrical power. The piezoelectric transducer can be represented by a combination of elastic material and active material. The vibration plate has a large surface to hold an object.

The constitutive relations given by IEEE Std. [12] for piezoelectric media describe the coupling between the mechanical and the electrical quantities of the system in tensor notation [13]:

$$S = TS^E + Ed$$

$$D = Td + \varepsilon^T E \tag{1}$$

where S is mechanical strain, T is the mechanical stress, E is the applied electrical field (V/m), D is the electrical displacement (C/m^2); S^E, d and ε^T are mechanical compliance at constant electrical field (m^2/N), piezoelectric charge constant (m/V), and dielectric permittivity under constant stress (F/m), respectively.

By using Hamilton's principle for non-conservative systems [14] under quasi-static assumption, the dynamic equations for a piezoelectric structure could be directly extracted from the variational principle. After discretizing the variables and substituting elements matrices into global matrices, the matrix form of the dynamic equations of coupled piezoelectric-structure system can be written as [15],

$$\begin{bmatrix} M_{UU} & 0 \\ 0 & 0 \end{bmatrix} \begin{bmatrix} \ddot{U} \\ \ddot{\Phi}_E \end{bmatrix} + \begin{bmatrix} D_{UU} & 0 \\ 0 & 0 \end{bmatrix} \begin{bmatrix} \dot{U} \\ \dot{\Phi}_E \end{bmatrix} + \begin{bmatrix} K_{UU} & K_{U\Phi} \\ K_{U\Phi}^t & K_{\Phi\Phi} \end{bmatrix} \begin{bmatrix} U \\ \Phi_E \end{bmatrix} = \begin{bmatrix} F \\ Q \end{bmatrix} \tag{2}$$

where U is the nodal displacement vector, Φ_E is the electrical potential. The first term represents inertia, the second term is damping and the third term is stiffness. The definitions of matrices presented in (2) are all described in Table 1.

Base on coupled-field theory, the dynamics of the transducer can be formulated by means of a finite element approximation.

3. FEA modeling

Finite element analysis (FEA) is a popular and powerful method in the design of piezoelectric transducers. It has been successfully applied to study and optimize transducer designs [16,17]. The transducer geometry is treated as a 3D quart symmetry model in this study using a commercially available FEM package (ANSYS12.1, ANSYS Inc., Canonsburg, PA) [18]. The ANSYS multiphysics solver with sequential approach and the Physics environment files are

suitable for the solution of the electrostatic-structural coupled-field analysis. The finite element model is applied to evaluate the electromechanical performance and operational characteristics of the transducer as a function of frequency using the harmonic analysis.

The vibration plate has a large surface to hold an object. Unlike conventional transducers, the maximum diameter of the transducer is more than a quarter of the sound wavelength. When excited by piezoelectric rings, the plate has its own mode shape, not rigid. Moreover, the plate is so big that its effect on the resonant frequency must be considered. In other words, the lateral and radial modes on the plate should not be ignored. Meanwhile the electromechanical coupling strongly acts on the system dynamic characteristics.

The thin electrodes are neglected in the FEA. The coupled equipotential boundary condition was applied to simulate the physical conductive behaviors of the electrodes. This is a good assumption as the piezoelectric pieces actually have a thin silver coating to insure excellent electrical contact. Moreover, the influence of fillets and chamfers at the corners has been ignored. Although transducer performance has been observed to drift slightly during operation as the ceramic pieces warm up due to mechanical losses, the temperature effects have been ignored in this study.

When an object is levitated, the levitation force is a load acting as boundary condition on the coupled system. However as the levitation force is small enough, we can ignore the light loading to make the model simplified. Later experimental measurements reveal that the levitated load barely influence the resonant frequency and the displacement distribution of the sound radiation surface.

Because of the complexity of the structure, the sufficiently dense and the regular mesh are required to guarantee the accuracy of the results. The analyzed transducer is composed of four PZT-8 rings, a steel cylinder- shaped back mass (45#) and an aluminum horn (Al6061). The steel bolt is made in (ST304). The vibration plate is made in aluminum (Al6061). Three-dimensional coupled-field elements (Solid5), which include both mechanical and electrical degree of freedom (DOF), are used to build the active PZT materials.

The relative dielectric permittivity matrix $[\varepsilon_r^s]$ at constant strain is:

$$[\varepsilon_r^s] = \begin{bmatrix} 900 & 0 & 0 \\ 0 & 900 & 0 \\ 0 & 0 & 600 \end{bmatrix}$$

The piezoelectric stress matrix $[e]$ (stress developed/electric field applied at constant strain) is:

$$[e] = \begin{array}{c} \\ X \\ Y \\ Z \\ XY \\ YZ \\ XZ \end{array} \begin{array}{c} \begin{array}{ccc} X & Y & Z \end{array} \\ \begin{bmatrix} 0 & 0 & -4.1 \\ 0 & 0 & -4.1 \\ 0 & 0 & 14.0 \\ 0 & 0 & 0 \\ 0 & 10.3 & 0 \\ 10.3 & 0 & 0 \end{bmatrix} \end{array} \mathrm{C/m^2}$$

The elastic coefficient matrix $[C]$ is:

$$[C] = \begin{array}{c} \\ X \\ Y \\ Z \\ XY \\ YZ \\ XZ \end{array} \begin{array}{c} \begin{array}{cccccc} X & Y & Z & XY & YZ & XZ \end{array} \\ \begin{bmatrix} 14.9 & 8.11 & 8.11 & 0 & 0 & 0 \\ 0 & 14.9 & 8.11 & 0 & 0 & 0 \\ 0 & 0 & 13.2 & 0 & 0 & 0 \\ 0 & 0 & 0 & 3.4 & 0 & 0 \\ 0 & 0 & 0 & 0 & 3.13 & 0 \\ 0 & 0 & 0 & 0 & 0 & 3.13 \end{bmatrix} \end{array} \mathrm{N/m^2}$$

Material properties of the other components are shown in Table 2.

Table 2
The material properties of metal components.

Standard code	AL 6061	Steel 45#	ST304
Modulus of elasticity (N/m^2)	6.83×10^{10}	2.1×10^{11}	2.07×10^{11}
Poisson's ratio	0.33	0.3	0.292
Density (kg/m^3)	2700	7850	7868

3.1. Modal analysis

Modal analysis in FEM is normally used to study the mechanical behavior. In this case, model solution has been employed to inspect the resonant frequencies of the electrical–mechanical coupled system, corresponding mode shapes and the location of the nodal plane. The block Lanczos method is chosen as this method is recommended by ANSYS instructions [19]. No structural constraints were used for the modal analysis, which produces a simulation of an unrestrained transducer assembly. Two electrical conditions were performed separately. In resonance condition, a constant voltage of zero was applied at all electrical contacts of ceramic rings, which simulate "short-circuit" condition. In anti-resonance condition, only the negative poles or the positive poles of the PZT rings were connected to zero voltage as the common ground while the other poles were left free open, which simulate the "open-circuit" condition.

The first three available modes of the radiation plate are shown in Fig. 2. Obviously the displacement distributions of the large radiation surface are quite different. The vibration modal shape serves as the boundary condition of the gas squeeze film between the transducer and levitated object. The flexural boundary is extremely crucial for the nonlinear levitation behavior [20].

3.2. Harmonic analysis

In modal solution, there are several resonant frequencies and corresponding mode shapes. However, most of the resonance modes may not be excited electrically. In view of that the harmonic analysis is performed to validate the correct modeling and correct modal analysis results. The transducer is driven by an ideal voltage generator and the electrical impedance responses are simulated in the frequency domain. The internal structural losses of the transducer have been taken into account by applying an appropriate damping ratio (0.36%), which is defined as the ratio of the loss energy over the kinetic energy when harmonic response analysis is done.

The analysis frequency range is from 1 kHz to 40 kHz and the input peak voltage is 110 V. The calculated relationship between

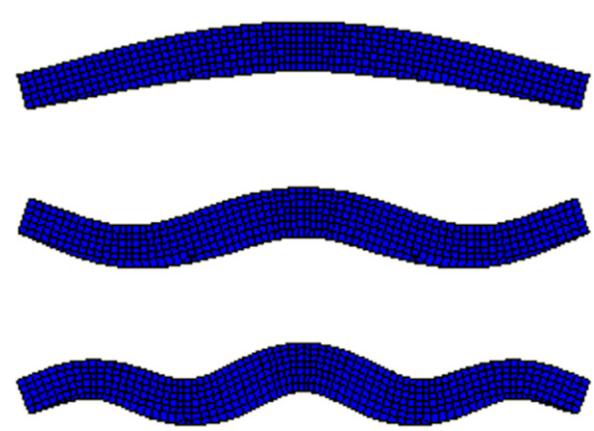

Fig. 2. The vibration displacement of the sound radiator in first three modes.

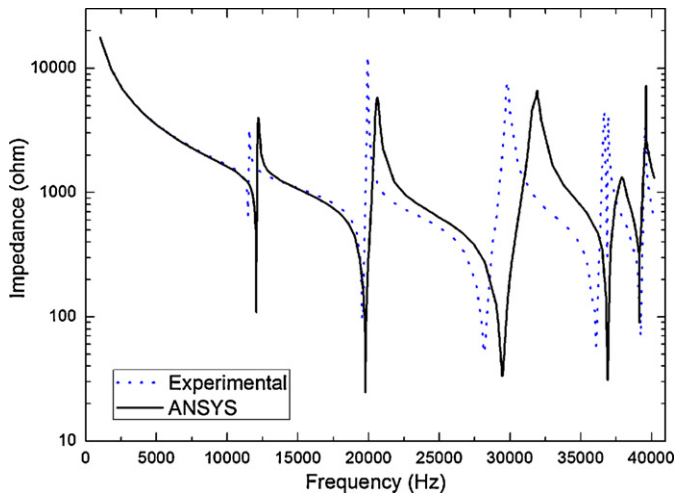

Fig. 3. Impedance by ANSYS simulation and experimental measurement against frequency.

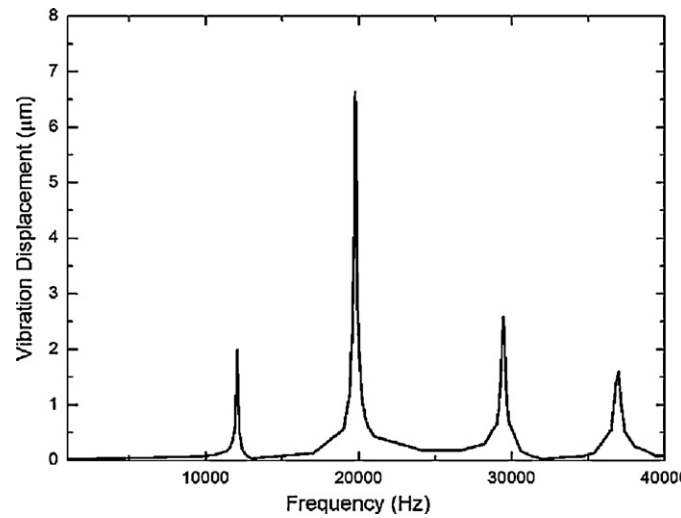

Fig. 4. Vibration displacement at the center of the radiator.

the input electrical impedance and frequency of the transducer is shown in Fig. 3. The "zeros" and "poles" correspond to the resonant frequencies of the device under "short" and "open" circuit conditions, respectively. The short-circuit condition corresponds to zero voltage across an electric element and zero impedance. The open-circuit condition corresponds to zero current and maximum impedance. The short- and open-circuit frequencies can be used to determine the electro-mechanical coupling coefficient.

An impedance analyzer was used to measure the electrical impedance of the device. The lowest impedance appears at the 2nd resonant frequency. Fig. 3 also presents the comparison between simulation and experimental results. Good agreement has been achieved in 1st and 2nd resonances. However, errors occur in higher resonant modes due to the neglecting of frequency dependence of the loss input in the simulation. In practical applications, acoustic devices are usually driven in lower resonant mode to avoid some nonlinear errors.

Fig. 4 illustrates the vibration displacement at the center of the sound radiator against frequency. The working surface produces maximum vibration displacement when driven at the 2nd resonant frequency. Therefore it is more efficient in radiation and generates more levitation force.

Working mode is essential in transducer excitation. Usually transducers are excited at the fundamental resonant frequency. This is because without coupling, only the longitudinal mode of the transducers is considered for which the fundamental mode holds the highest energy efficiency. However for a coupled transducer used in the NFAL applications, the highest energy conversion may not happen at 1st resonance. Harmonic analysis offers a convenient method to compare each resonance and help us chose the working mode. Among all modes in our case, the 2nd resonance has the minimum impedance and maximum vibration. As a result, the 2nd mode is chosen in the levitation experiment for better efficiency in electromechanical energy conversion and sound radiation. In our experimental testing, the transducer was excited at its 2nd resonant frequency by a frequency tracking generator.

During the levitation, the floating object acts as the load on the transducer. In harmonic simulations, we added load up to 10 N on the vibrator surface, but such small load did not produce significant change in the resonant frequency, impedance and vibration displacement. This also has been validated in experimental measurement.

4. Size effect

In some levitation cases, transducers should be operated at a desired frequency according to the dynamic requirements of the levitation. Moreover the excitation amplifiers work in certain bandwidth. Therefore it is important to be able to change the resonant frequency in design optimization. There are many factors affecting the resonant frequency. The geometrical dimension of the vibrator plays the leading role among these factors. In other words, desired resonant frequency can be obtained by changing the dimension of sound radiator. Table 3 shows shifted frequency by changing the thickness of the vibrator only. In the experimental study, we chose the vibrator with 14 mm thickness to better match with the amplifier bandwidth.

The ultrasonic transducers are usually fixed at the node plane to achieve better energy efficiency. We can find the node plane with zero displacement based on the modal solution. To show the importance of the location more explicitly, we have simulated several cases by changing the location of the fixed plane. With the same input power, the vibration displacement of the radia-

Table 3
Frequency with different vibrator thickness.

Vibrator thickness (mm)	First mode frequency (Hz)		Second mode frequency (Hz)	
	Resonant	Anti-resonant	Resonant	Anti-resonant
12	11,408	11,583	18,796	19,602
13	11,761	11,865	19,376	20,207
14	12,048	12,177	19,778	20,596
15	12,292	12,449	20,298	21,106
16	12,633	12,751	20,917	21,786

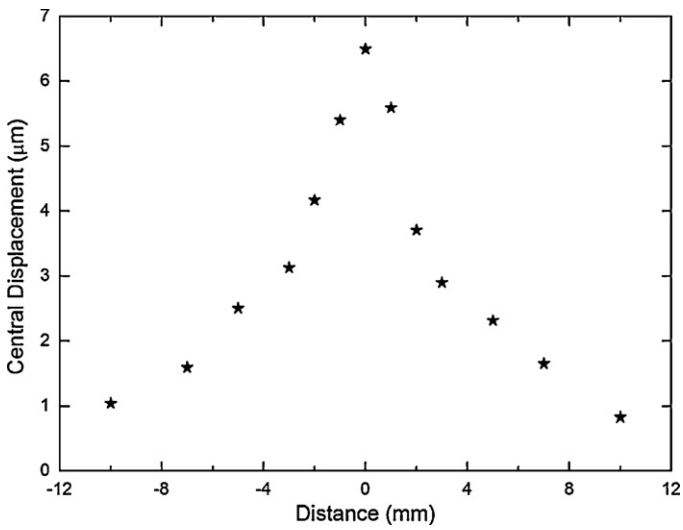

Fig. 5. Displacement at the vibrator center vs. the distance of fixed plane from node plane.

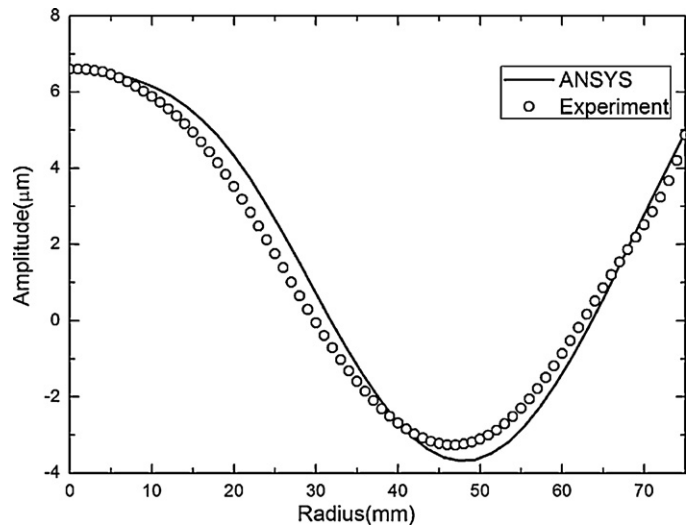

Fig. 7. Vibration displacement distribution of the sound radiation surface.

tion surface decreases as moving the fixed plane away from the node plane. Fig. 5 shows the central displacement of the vibrator fixed at different locations, in which d is the distance from the node plane. All cases are excited at 2nd resonant frequency. The largest displacement happens at $d = 0$. As the fixed plane departs from the nodal plane, the displacement drops steeply. The input energy is consumed mostly by the solid structure so that less energy can be radiated for levitation. The result indicates that the transducer needs to be fixed on node plane to reduce the energy loss.

5. Experimental validation

For the experimental verification, we built a levitation system in a clean room with a constant temperature environment of 20 °C. Fig. 6 illustrates the configuration of the experimental measurement setup. A rigid aluminum plate is placed on the radiation surface as the reflector. This disk has the same radius as the vibrator surface, and is connected to the load cell on a position stage. The levitation distance is measured by a laser displacement sensor and the displacement distribution of the vibrator is measured by a laser scanning vibrometer (Polytec, PSV-300F-B).

Fig. 3 compares the impedances of the simulation with that of the measurement. Good agreement has been achieved due to our coupled modeling. We obtained the resonant frequency of the whole device by an automatic tracing frequency scanner. The measured 1st and 2nd resonant frequencies are 11,520 Hz and 19,580 Hz, respectively. The errors by ANSYS modal analysis are 4.58% and 1.01%, respectively. Although the modal analysis is designed for ideal free boundary cases, it does provide a good guidance to our experimental design. In the experiments, the excitation peak voltage is 110 V and the total input power is 80 W.

Fig. 7 shows the comparison of displacement distributions on sound radiator by ANSYS and experimental measurements. Because the device is simulated at the 2nd mode, the plate has two node circles. The maximum error of the vibration displacement is 6.23%. The result indicates the flexural mode and coupling of the transducer.

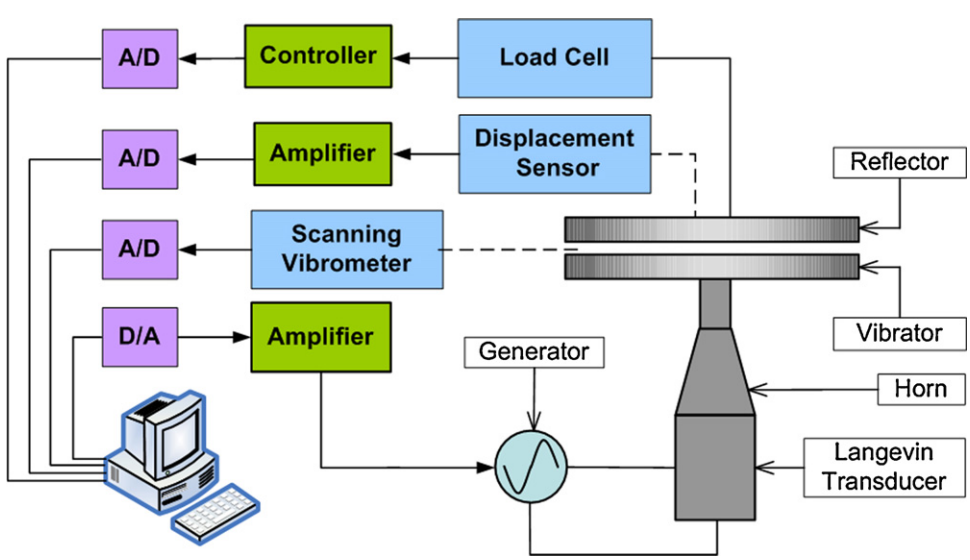

Fig. 6. Schematic of driving and measuring system of NFAL.

6. Conclusions

The PZT transducer used in NFAL has a large radiation surface, which caused the coupling of the longitudinal mode of the Langevin type transducer with the flexural mode of the top plate. In this paper a coupled FEA model has been present to predict the transducer performance in the design stage. Modal and harmonic analyses have been conducted to optimize the model. Identical experimental conditions have been simulated to investigate the mechanical and electrical behaviors. Working mode is of importance to mechanical, electrical and acoustical performance during levitation. The 2nd mode has been chosen as the operating mode based on energy perspective. When the vibrator works at the 2nd mode, it has the largest displacement and lowest electrical impedance. The FEA model also presents a method to predict the vibration distribution of the radiation surface. Further calculations in acoustic field can be performed using this simulated vibration shape as the boundary condition to save time consuming instead of experimental measurements.

The size effect on the dynamic performance of the NFAL transducer has been discussed. Geometric parameters have been optimized to achieve higher efficiency of electromechanical conversion and acoustic radiation. Proper thickness of the plate has been chosen to obtain a desired resonant frequency. Moreover, we show that the fixed plane should locate at the nodal plane so that the largest displacement can be obtained with a given input.

The discrepancy between simulation and experiment come from two kinds of errors. One is the systematic error; the other is the random error. The systematic error originated from ANSYS code. ANSYS uses linear piezoelectric effect in the calculations, without considering nonlinearity under high electric field. The uncertain factors come from manufacturing, measurement and excitation of the transducers. Over all, the errors are well within the designed tolerance.

Our modeling method offers a powerful and reliable tool for predicting the dynamic behavior of NFAL transducers. In particular, further characteristics in gas squeeze film can be calculated using the FEA data instead of experimental measurements. The developed design procedure provides an optimal tool to construct a NFAL transducer, which also can be used in the areas of nondestructive evaluation, underwater acoustics and bioengineering.

Acknowledgements

This research was supported in part by National Natural Science Foundation of China under Grant Nos. 50675132 and 91023035, the Science & Technology Commission of Shanghai Municipality under Grant No. 09JC1408300.

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Biographies

Jin Li was born in 1983 in Sichuan, China. She received the B.S. degree in mechanical engineering from Shanghai Jiao Tong University, Shanghai, China, in 2006. She is currently working toward the Ph.D. degree at Shanghai Jiao Tong University in the Research Institute of Robotics, mechanical engineering school. Her research focuses on piezoelectric actuators and acoustic levitation.

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