Modeling and Experimental Study on Near-Field Acoustic Levitation by Flexural Mode

Pinkuan Liu, Jin Li, Han Ding, *Member, IEEE*, and Wenwu Cao

***Abstract*—Near-field acoustic levitation (NFAL) has been used in noncontact handling and transportation of small objects to avoid contamination. We have performed a theoretical analysis based on nonuniform vibrating surface to quantify the levitation force produced by the air film and also conducted experimental tests to verify our model. Modal analysis was performed using ANSYS on the flexural plate radiator to obtain its natural frequency of desired mode, which is used to design the measurement system. Then, the levitation force was calculated as a function of levitation distance based on squeeze gas film theory using measured amplitude and phase distributions on the vibrator surface. Compared with previous fluid-structural analyses using a uniform piston motion, our model based on the nonuniform radiating surface of the vibrator is more realistic and fits better with experimentally measured levitation force.**

I. Introduction

Acoustic levitation is one of the most suitable methods for a wide range of applications. The major advantage lies in the fact that any material, insulator, or conductor, magnetic or nonmagnetic, can be manipulated by acoustic levitation and transportation without physical contacts. In micro-assembly, it is difficult to handle and transfer the device or component of micro electromechanical system (MEMS) due to their fragility and surface sensitive characteristics. Classical assembly processes are usually based on mechanical contact, which may result in the destruction of fragile parts or cause some degree of surface damage. Thus, many situations require manipulating such fragile parts without physical contacts [1], [2].

High-intensity acoustic levitation provides an inexpensive noncontact method with magnetic field and electric field immunity, which are useful characteristics for clean room and precision actuators [3], [4]. Although magnetic levitation is also a noncontact method, it suffers from the drawbacks of particle accumulation and low frequency vibrations [5], [6].

There are 2 types of acoustic levitation methods used for manipulating parts [7]: standing-wave acoustic levitation (SWAL) and near-field acoustic levitation (NFAL). In standing-wave levitation, small particles can be levitated in the pressure nodes of an acoustic standing wave between a vibrating plate and a reflector; the levitation is appropriate for containerless processing of materials. In near-field levitation, the reflector is replaced by the levitated object itself, and the near-field levitation is more suitable for guiding or transfer of surface sensitive parts [8].

In NFAL, a planar object atop a vibrating surface is levitated in the near-field range by the acoustic radiation emanating from the vibrating surface. Hashimoto *et al.* [9] illustrated the phenomenon using a planar plate levitated about one-tenth of an acoustic wavelength from an ultrasonically vibrating surface. The principle has been used for developing noncontact ultrasonic motors and noncontact handling and transportation of ultra-clean glass plates for liquid crystal displays. Friend [10], [11] designed a linear bearing using the NFAL principle, where a slider weighing 90 g can be transported linearly at a speed of 138 mm/s.

NFAL is used to handle planar objects slightly above the manipulator surface with a high-intensity acoustic vibrator. The gas squeeze film, which is created between acoustic vibrator and planar object by rapid vibrations, generates a time-averaged levitation force. Both in-phase longitudinal mode, like piston motion, and flexural vibration mode can be employed as radiation sources. The closer the object approaches the radiating source the larger the levitation force it generates. Although practical applications had been developed, theoretical treatment of NFAL is still rather primitive. Chu and Apfel [12] derived a formula based on Eulerian and Langrangian method to express the acoustic radiation pressure by a vibration piston. Minikes and Bucher in [13] investigated the levitation force by flat driving surface and traveling wave vibrations. The key assumption in previous studies [9], [10], [12]–[14] is that the radiator had been assumed to be a rigid surface with a uniform displacement and no deformation. However, in practical situations, the radiator obviously has nonuniform elastic deformation when the radiator is excited by a time–varying source. Meanwhile, the shape of the nonuniform radiator surface should be used as the boundary in solving the nonlinear differential equation of gas squeeze film. Those previous theoretical models cannot accurately describe the pressure distribution of the squeeze film in NFAL because the deformation of the radiator surface had been ignored. It was found experimentally that the suspension force is highly dependent on the vibration distribution on the radiation surface [15], [16].

In the present paper, the measured flexural mode shape of the radiator along the radius is used as exact boundary conditions in solving the gas squeeze film problem. Be-

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P. Liu, J. Li, and H. Ding are with the School of Mechanical Engineering, Shanghai Jiao Tong University, Shanghai, China.

P. Liu and H. Ding are also with the State Key Laboratory of Mechanical System and Vibration, Shanghai Jiao Tong University, Shanghai, China (e-mail: pkliu@stju.edu.cn).

W. Cao is with the Materials Research Institute, The Pennsylvania State University, University Park, PA.

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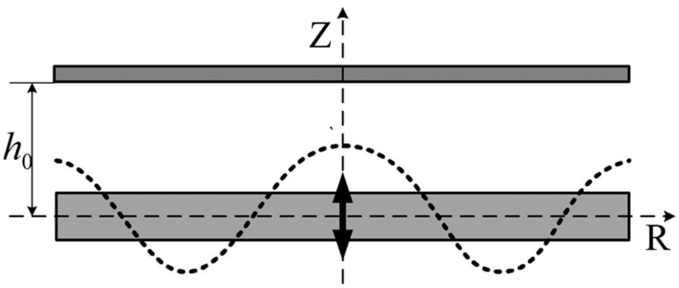


Fig. 1. Model structure for near-field acoustic levitation.

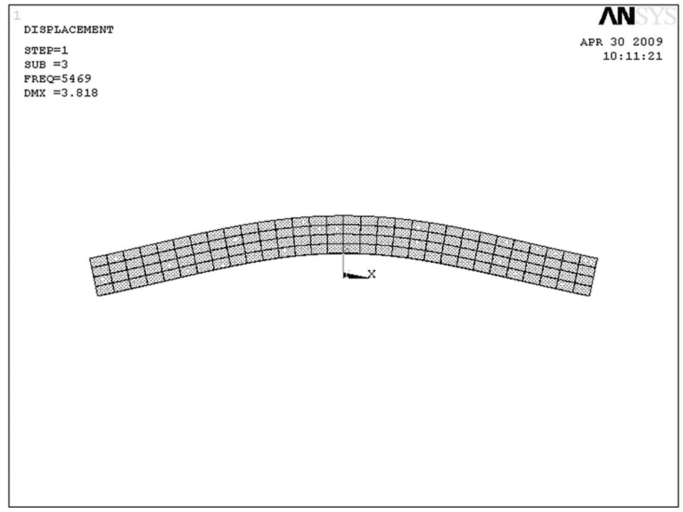


Fig. 2. First-mode shape of the vibrating plate.

cause the deformation of the radiator changes the boundary condition of gas squeeze film, the complexity of the convection-diffusion differential equation is increased [17]. A higher order finite-difference scheme is introduced here to perform the numerical solution.

Our theoretical procedure contains 2 steps. First, based on the classical thick plate theory, mode shape and corresponding frequencies of the vibrator were calculated by finite element method using ANSYS (ANSYS Inc., Canonsburg, PA). The resonance frequencies were used to guide the design of the experimental setup, and then the vibration amplitudes and phases of 500 points on the vibrator surface were measured using an optical interferometer. Second, a higher order numerical method is employed to solve the equation of gas squeeze film problem using the exact boundary conditions obtained from the experimentally measured data. The numerical levitation force is then compared with the experimentally measured force as a function of the levitation height.

II. Theoretical Model

As illustrated in Fig. 1, we consider a vibrating surface of radius R with displacement along the vertical z axis at frequency f. A reflector of the same dimension is placed initially at a distance h_0 from the vibrating surface and is assumed to be rigid (cannot move and deform) [14]. Between the vibrator and reflector, there is a thin layer of air whose thickness is much smaller than the radius, which is conventionally called gas squeeze film. Gas squeeze film generates NFAL phenomenon to levitate the reflector.

An aluminum disk connected to the top of an ultrasonic horn is served as the vibrating surface. Minikes and Bucher [13] studied a first-order perturbation solution for the radiation pressure generated by the normal vibration of a flat piston. According to their assumption, the disk vibrates uniformly (independent of R) with certain amplitude along the z-direction, for which the displacement on the surface can be expressed as a function of time only. In reality, because the diameter of the disk is much lager than the end of the driving horn, the vibrating disk cannot oscillate in a piston motion at resonance. In other words, the displacement on the surface changes with both time and radius.

To model this complex system with nonuniform surface vibration, we assume that the levitation force in the gas film does not affect the vibration of the sound source. Therefore, the coupling between air and the solid plate can be ignored so that we can solve the problem of the vibrating disk independently. This point had been verified experimentally. We have measured the resonance frequencies of the vibrator under 30 N and 60 N loads and found no difference from the case without load.

Based on classical thick plate theory [18], [19], the disk has several resonance modes. The one we can employ to generate an axisymmetrical levitation force should be an axisymmetrical vibration mode, in which points at the same radius have exactly the same displacement at a given time. Figs. 2 and 3 show the 1st- and 2nd-mode shape of the vibrating plate and the corresponding natural frequencies are 5469 Hz and 21 024 Hz, respectively. The natural frequency obtained from modal analysis of ANSYS was later used to guide our experimental design, including the driving Langevin transducer, horn, and vibrating plate [20], [21].

The finite element method (FEA) results indicate that the largest deflection magnitude occurs in the center of the plate and the displacement of the radiation surface is a function of time and radius. As long as the applied voltage is in the linear range of the piezoelectric material, the vibration is harmonic. In our experimental setup, the applied voltage is sinusoidal with a constant amplitude, which results in a steady-state harmonic vibration with a constant amplitude. At the same vibration amplitude, excitation of the 2nd mode produced a larger levitation force than with the 1st mode. Therefore, we chose the 2nd mode when performing real experiments.

The flexural vibrating surface squeezes the compressible air film, generating a time-averaged pressure. Because the Reynolds number is low in the present case, we assume that the fluid inertia is negligible compared with the viscous forces [22]. In addition, we assume that the thin gas film is isothermal because of its low heat capacity. Also,

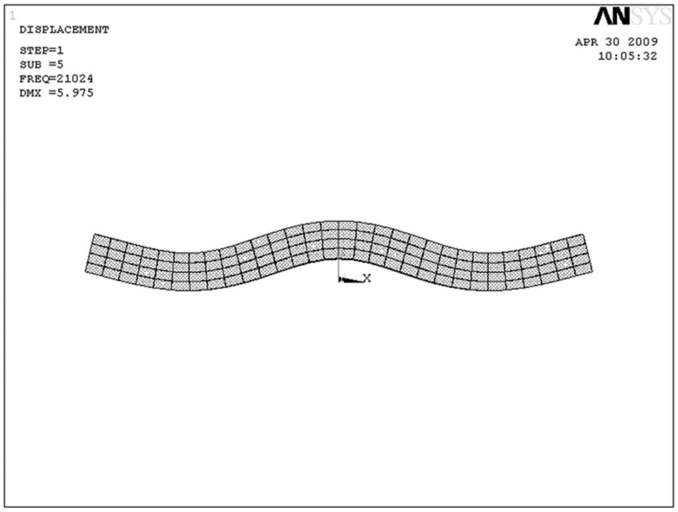


Fig. 3. Second-mode shape of the vibrating plate.

the high squeeze number reveals that the pressure gradient in the normal direction may be ignored. Consequently, the problem is reduced to 1 dimension.

The Reynolds equation is employed as the governing equation for pressure distribution in the squeeze film, which is suitable for laminar, isothermal, and compressible thin fluid [23]. The Reynolds equation is derived from the classical Navier-Stokes equation with continuity restriction

$$\frac{\partial}{\partial X}\left(H^{3}P\frac{\partial P}{\partial X}\right)=\sigma\frac{\partial(PH)}{\partial T}, \tag{1}$$

where p_a is the atmosphere pressure, $P,\ H,\ X,\ T$ are the dimensionless pressure, mean clearance, horizontal coordinate, and time, respectively: $P=p/p_a$, $H=h/h_0$, $X=x/R_0$, $T=\omega\cdot t$. The squeeze number σ is defined by

$$\sigma=\frac{12\omega\mu\cdot {R_0}^{2}}{p_{a}h_{0}^{2}}, \tag{2}$$

where R_0 is the radius of the squeeze film.

The equation in polar coordinates is given by

$$\frac{1}{R}\frac{\partial}{\partial R}\left(PRH^{3}\frac{\partial P}{\partial R}\right)=\sigma\frac{\partial(PH)}{\partial T}. \tag{3}$$

The boundary condition and initial condition are as follows:

1) The pressure at the disk edge is equal to the atmosphere pressure, whereas the pressure gradient in the center of the disk is zero, i.e.,

$$P(R=1,T)=1,\quad \frac{\partial P}{\partial R}(R=0,T)=0. \tag{4}$$

2) The dimensionless thickness of the gas film is

$$H(R,T)=1+Y(R)\cdot\sin(T), \tag{5}$$

where $Y(R)$ is the dimensionless vibration amplitude on the vibrator surface at point R. And the real displacement of the vibrator surface is given by

$$d(R,T)=Y(R)\cdot\sin(T)\cdot h_{0}. \tag{6}$$

3) At $t=i$, the pressure in the film is equal to the atmosphere pressure,

$$P(R,T=0)=1. \tag{7}$$

The time averaged pressure at each point is given by

$$P_{\text{average}}(R)=\frac{1}{2\pi}\int_{0}^{2\pi}P(R)\cdot dT. \tag{8}$$

The total levitation force on the surface is therefore given by

$$F=2\pi\int_{0}^{1}R(P_{\text{average}}-1)dR. \tag{9}$$

III. Numerical Solutions

In the present treatment, the gas film thickness H changes with R unlike in the pistonlike models so that it is impossible to obtain analytical solutions. To treat such problems, a higher accuracy finite-difference scheme is needed in numerical calculations. Here we used the high-resolution central schemes for convection-diffusion equations presented by Kurganov *et al.* [24]. This second-order semidiscrete central scheme has a specific simplicity. It does not require any information about the eigen-structure of certain problem beyond the CFL[1] (Courant-Friedrichs-Lewy Condition)-related speeds, $a_{j+1/2}(t)$. The general 1-D non-homogenous convection-diffusion equation is expressed as follows:

$$\begin{aligned}\frac{\partial}{\partial t}u(x,t)+A\cdot\frac{\partial}{\partial x}f\left(u(x,t)\right)=\\ B\cdot\frac{\partial}{\partial x}Q\left[u(x,t),u_{x}(x,t)\right]+S\left(u,x,t\right),\end{aligned} \tag{10}$$

where $u(x,t)$ is a conserved quantity, $f(u)$ is the nonlinear convection flux, $Q[u(x,t),u_x(x,t)]$ is the dissipation flux satisfying the (weak) parabolicity condition, and the source term $S(u,x,t)$ is a function of $u(x,t)$, x, and t.

Treating the hyperbolic and the parabolic parts of (10) simultaneously results in the following conservative scheme:

[1] CFL: Courant-Friedrichs-Lewy Condition. In mathematics, the Courant-Friedrichs-Lewy Condition is a condition for convergence while solving certain partial differential equations numerically.

$$\begin{aligned}\dot{u}_j(t) = -A \cdot \frac{G_{j+1/2}(t) - G_{j-1/2}(t)}{\Delta x} \\ + B \cdot \frac{K_{j+1/2}(t) - K_{j-1/2}(t)}{\Delta x} + S_j(t),\end{aligned} \tag{11}$$

where

$$\begin{aligned}
G_{j+1/2}(t) &= \frac{f(u^+_{j+1/2}(t)) + f(u^-_{j+1/2}(t))}{2} \\
&\quad - \frac{a_{j+1/2}(t)}{2}[u^+_{j+1/2}(t) - u^-_{j+1/2}(t)], \\
K_{j+1/2}(t) &= \frac{1}{2}\Bigg[Q\left(u_j(t), \frac{u_{j+1}(t) - u_j(t)}{\Delta x}\right) \\
&\quad + Q\left(u_{j+1}(t), \frac{u_{j+1}(t) - u_j(t)}{\Delta x}\right)\Bigg], \\
a^n_{j+1/2} &= \max\left\{\rho\left(\frac{\partial f}{\partial u}(u^-_{j+1/2})\right), \rho\left(\frac{\partial f}{\partial u}(u^+_{j+1/2})\right)\right\}, \\
u^+_{j+1/2} &= u_{j+1}(t) - \frac{\Delta x}{2}(u_x)_{j+1}(t), \\
u^-_{j+1/2} &= u_j(t) + \frac{\Delta x}{2}(u_x)_j(t).
\end{aligned}$$

Here, $G_{j+1/2}(t)$ is the numerical convection flux, $K_{j+1/2}(t)$ is a reasonable approximation to the diffusion flux, and $(u_x)^n_j$ is defined as

$$(u_x)^n_j \coloneqq \min\bmod\left(\frac{u^n_j - u^n_{j-1}}{\Delta x}, \frac{u^n_{j+1} - u^n_j}{\Delta x}\right), \tag{12}$$

where min mod $(a, b) := 1/2[\mathrm{sgn}(a) + \mathrm{sgn}(b)]$.

Substituting the semidiscrete central scheme of the 1-D nonhomogenous convection-diffusion equation into (3) we obtain

$$u(x,t) = P(R,T) \tag{13}$$

$$f(u(x,t)) = P^2(R,T) \tag{14}$$

$$A = -\frac{1}{2\sigma R}\left[3RH(R,T)\cdot(-\sin T)\cdot\dot{y}(R) + H^2(R,T)\right] \tag{15}$$

$$Q(u(x,t), u_x(x,t)) = P(R,T)\cdot\frac{\partial P(R,T)}{\partial R} \tag{16}$$

$$B = \frac{H^2(R,T)}{\sigma} \tag{17}$$

$$S(x,t) = \frac{P(R,T)}{H(R,T)}\cos T \cdot y(R). \tag{18}$$

The numerical solution of the pressure distribution in the air film is a function of time and radius. The total levitation force can be integrated from the pressure distribution by (8) and (9). The pressure distribution along the radius reveals that near the edge of the disk ($R = 60$ mm), the mean pressure is equal to the atmosphere, whereas the largest time averaged pressure occurs at the center of the vibrator ($R = 0$ mm). Furthermore, the air near the edge ($R = 60$ mm) barely experiences any compression or decompression, so no squeeze effect take place near the edge. The squeeze phenomenon takes place close to the center of the film disk, where the mean pressure is above the atmospheric pressure.

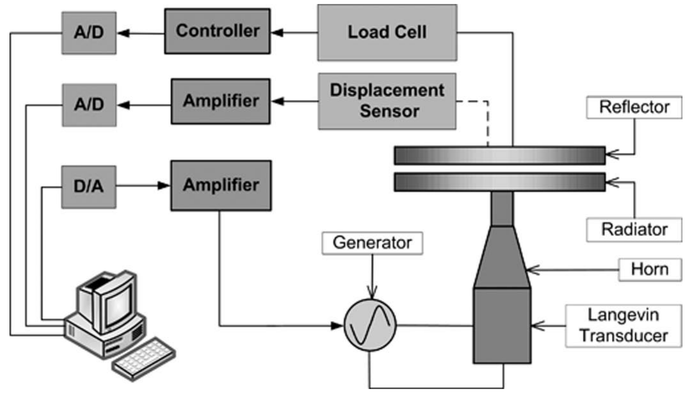


Fig. 4. Schematic of driving and measuring system of near-field acoustic levitation.

IV. Experimental Study

Fig. 4 illustrates the configuration of the experimental setup. The radiator is excited at the 2nd resonant frequency by a conical horn that is driven by a pre-stressed sandwich transducer also referred to as Langevin-Bolt transducer (LBT) [7]. There are 4 piezoelectric ring elements in the middle of the transducer, with a 19-mm radius and a thickness of 5 mm. The piezoelectric elements are driven in its thickness mode to generate acoustic vibrations. The conical horn is designed to magnify the vibration amplitude of the Langevin transducer. An aluminum plate with a 60-mm radius and 9-mm thick is used as the vibrator, which is screwed onto the horn of the LBT. As shown in Fig. 5, 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 3-degree-of-freedom position stage is used to position the reflector precisely over the radiator and move the reflector along the z-direction to change the levitation distance. The levitation distance was measured with laser displacement sensor, and the corresponding levitation force was measured by the load cell. The measurements were carried out while keeping the vibration amplitude at 0.05 mm at the center point of radiation surface.

The displacement pattern of the vibrator surface was measured by a Polytec scanning vibrometer (PSV-300F-B, Polytec GmbH, Waldbronn, Germany). As shown in Fig. 6, we have probed more than 500 points on the radiation surface of the flexural plate. The vibration amplitude and phase of each point was recorded. These measured

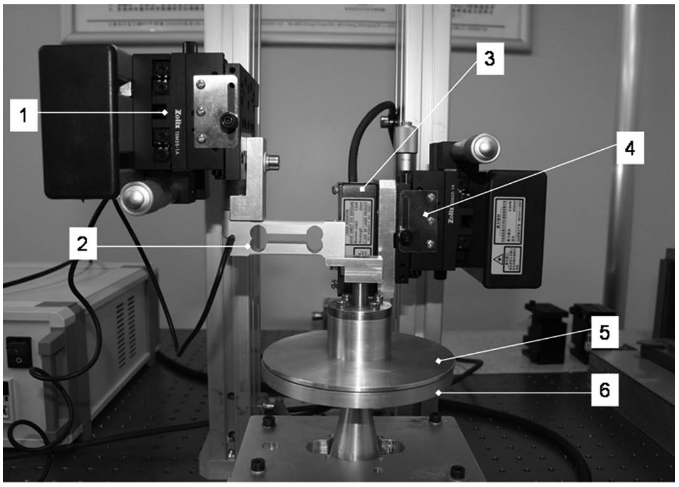


Fig. 5. Experimental setup of near-field acoustic levitation: (1) positioning stage, (2) load cell, (3) displacement sensor, (4) positioning stage, (5) reflector, and (6) vibrator.

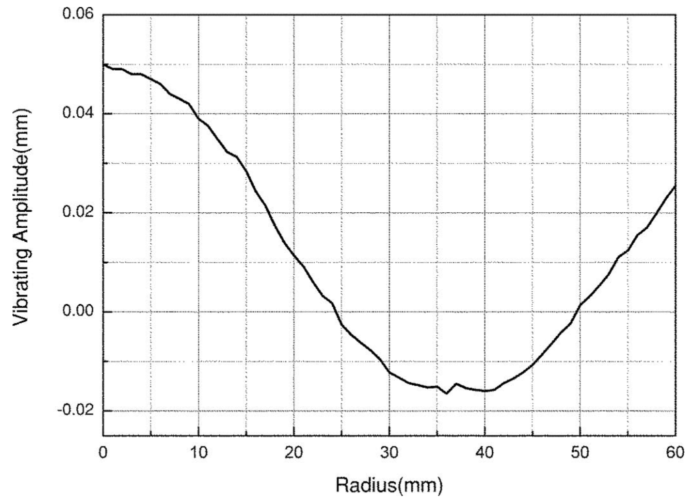


Fig. 7. Radiator surface vibration amplitude as a function of radius measured by a Polytec scanning vibrometer (PSV-300F-B).

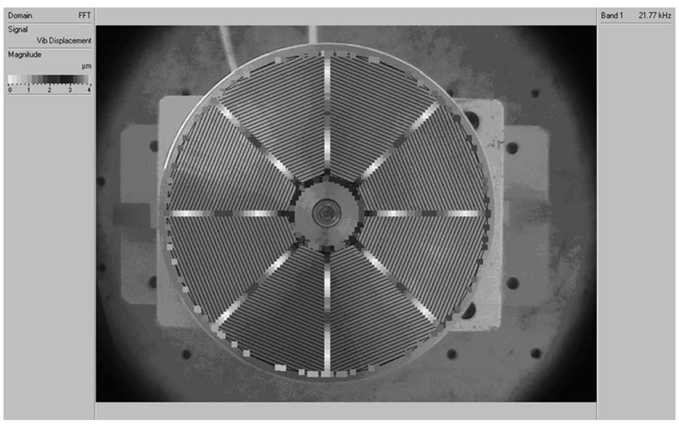


Fig. 6. Points measured on the vibrator surface.

TABLE I. Comparison of Frequencies in Modal Analysis and Experimental Measurements.

	Frequency, kHz
Modal Analysis	21.024
Measurement (Load = 0 N)	21.765
Measurement (Load = 30 N)	21.765
Measurement (Load = 60 N)	21.765

amplitudes and phases were used as boundary conditions in numerical solution of the gas squeeze film, see (5) and (6). The measured vibration amplitude as a function of radius is shown in Fig. 7.

We have obtained the resonant frequency of the whole system by an automatic tracing frequency scanner. Table I shows the comparison between the frequencies obtained by ANSYS modal analysis and experimental measurements. The 2nd natural frequency obtained from ANSYS modal analysis is only about 3.5% off the measured resonant frequency. Although the modal analysis was designed for ideal free boundary cases, it does provide good guidance to our experimental design.

Because the levitation force changes with levitation distance and the structure is coupled with acoustic field, when the levitation distance is small, the load is higher, which may affect the resonant frequency of the system. To verify this point, we have measured the resonant frequencies of the system at different loads as shown in Table I. The measured results indicate that when the load is within 60 N, there is no measurable difference of resonant frequency from the load free case. Essentially, in our setup, the load is too small to have any significant effect in the vibrating system when the system is in a strong forced vibration. As a result, the coupling of the structure response from the acoustic field can be ignored, which confirms the decoupling assumption in our theoretical treatment.

The levitation force is plotted in Fig. 8 as a function of levitation distance. Clearly visible in Fig. 8 is the steep rising levitation force close to the vibrating surface. The force decreases quickly with increasing levitation height from the radiator. Although the pistonlike models can provide the general trend correctly (dashed line), the numerical values are far off from that of experiments. Our model with nonuniform surface displacement showed great improvement (solid line). Although our theoretical results are much better in comparison, it is still not satisfactory near the vibrator surface. One reason is that the relative nonuniformity becomes very large, and the pressure gradient term cannot be ignored. The pressure gradient can cause higher order effect in the squeeze film with increasing levitation force, which requires a multidimensional nonhomogenous convection-diffusion equation instead of the 1-D equation solved in this paper. Nevertheless, predictions from our model are much closer to the experimental values than previous pistonlike models. As the levitation distance increases, the agreement between our model and the experimental values becomes better and better. Excellent agreement between experimental data and our theoretical prediction was found when the levitation dis-

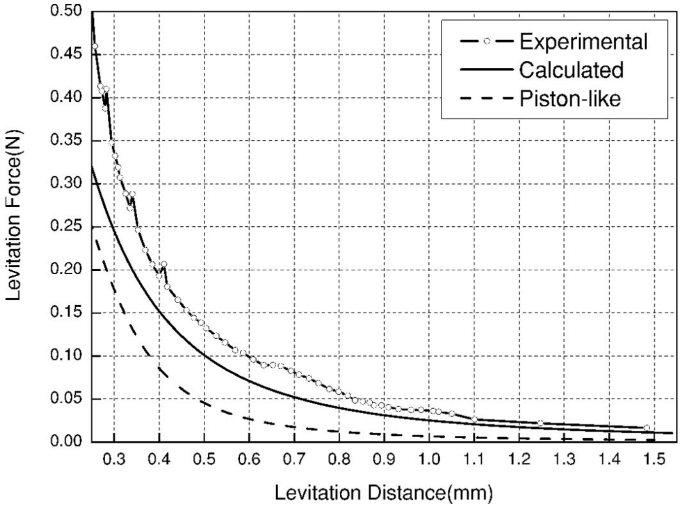


Fig. 8. Comparison of levitation forces obtained by measurement (circle), our calculation (solid line), and calculation using pistonlike approximation (dash line).

tance is beyond 1.1 mm. On the other hand, the pistonlike model never gives satisfactory predictions.

V. Summary and Conclusions

Near-field acoustic levitation technology offers an excellent approach for noncontact handling of micro-parts, surface-sensitive wafers, and substrates. An important advantage of using NFAL to transfer planar parts is that the method is not subject to any restrictions in terms of size or shape of the feed paths.

Theoretical treatment of acoustic levitation problem is still rather limited up to date. The well-used pistonlike models for analyzing the squeeze gas film problem can reduce the problem into 1-D so that analytic solutions can be obtained. However, it does not provide an accurate description of the experimental measurements. Considering the vibrator surface should be nonuniform, we have derived a theoretical procedure by using real measured surface displacement distributions as boundary conditions to solve the squeeze gas film problem. Because of the nonuniform surface displacement, one cannot obtain analytic solutions anymore, so we have used higher order numerical methods to derive the solutions. Compared with experimentally measured levitation force values, the new model results showed significant improvement over the pistonlike models. In fact, excellent agreement between theory and experiment occurs when the levitation distance is beyond 20 times of the vibration amplitude.

There are still discrepancies between theory and experiments when the levitation height is less than 10 times of the vibration amplitude even with our improved model. One important reason is that the gas clearance decreases so that the pressure gradient in the normal direction may not be ignored. To include such effects, one must solve multidimensional nonhomogenous convection-diffusion differential equations.

References

[1] V. Vandaele, P. Lambert, and A. Delchambre, "Non-contact handling in microassembly: Acoustical levitation," *Precis. Eng.*, vol. 29, no. 4, pp. 491–505, 2005.

[2] G. Reinhart and J. Hoeppner, "Non-contact handling using high-intensity ultrasonics," *CIRP Ann.*, vol. 49, no. 1, pp. 5–8, 2000.

[3] N. Taniguchi, *Nanotechnology: Integrated Processing Systems for Ultra-Precision and Ultra-Fine Products,* Oxford: Oxford University Press, 1996.

[4] K. H. Park, S. K. Lee, J. H. Yi, S. H. Kim, Y. K. Kwak, and I. A. Wang, "Contactless magnetically levitated silicon wafer transport system," *J. Mechatron.*, vol. 6, no. 5, pp. 591–610, 1996.

[5] J. Hesselbach, S. Büttgenbach, J. Wrege, S. Bütefisch, and C. Graf, "Centering electrostatic microgripper and magazines for microassembly tasks," presented at Microrobotics and Microassembly III, Newton, MA, 2001.

[6] M. F. Zäh, A. Zitzmann, and M. Schilp, "Non-contact handling in microfabrication," presented at the European Society for Precision Engineering and Nanotechnology, Aachen, Germany, 2003.

[7] G. Reinhart and J. Hoeppner, "Non-contact handling using high-intensity ultrasonics," *CIRP Ann. Manuf. Technol.*, vol. 49, no. 1, pp. 5–8, 2000.

[8] S. Ueha, "Recent development of ultrasonic actuators," in *Proc. IEEE Ultrasonics Symp.*, vol. 1, 2001, pp. 513–520.

[9] Y. Hashimoto, Y. Koike, and S. Ueha, "Acoustic levitation of planar objects using a longitudinal vibration mode," *Jpn. J. Appl. Phys.*, vol. 16, no. 3, pp. 189–192, 1995.

[10] T. Ide, J. Friend, K. Nakamura, and S. Ueha, "A non-contact linear bearing and actuator via ultrasonic levitation," *Sens. Actuators A*, vol. 135, pp. 740–747, Aug. 2006.

[11] D. Koyama, T. Ide, J. R. Friend, K. Nakamura, and S. Ueha, "An ultrasonically levitated noncontact stage using traveling vibrations on precision ceramic guide rails," *IEEE Trans. Ultrason. Ferroelectr. Freq. Control*, vol. 54, no. 3, pp. 597–604, Mar. 2007.

[12] B. Chu and E. Apfel, "Acoustic radiation pressure produced by a beam of sound," *J. Acoust. Soc. Am.*, vol. 72, no. 6, pp. 1673–1687, Dec. 1982.

[13] A. Minikes and I. Bucher, "Levitation force induced by pressure radiation in gas squeeze films," *J. Acoust. Soc. Am.*, vol. 116, no. 1, pp. 217–226, Apr. 2004.

[14] H. Nomura and T. Kamakura, "Theoretical and experimental examination of near-field acoustic levitation," *J. Acoust. Soc. Am.*, vol. 111, pp. 1578–1583, Apr. 2002.

[15] E. Matsuo, Y. Koike, K. Nakamura, S. Ueha, and Y. Hashimoto, "Holding characteristics of planar objects suspended by near-field acoustic levitation," *Ultrasonics*, vol. 38, pp. 60–63, Mar. 2000.

[16] J. Li and P. Liu, "Analytical and numerical study on near-field acoustic levitation force in gas squeeze films," presented at the 5th Int. Workshop on Piezoelectric Materials and Applications in Actuators, The Pennsylvania State University, University Park, PA, 2008.

[17] M. Remesikova, "Solution of convection-diffusion problems with non-equilibrium adsorption," *J. Comput. Appl. Math.*, vol. 169, no. 1, pp. 101–116, 2004.

[18] A. W. Leissa, *Vibration of Plates.* Washington, DC: U.S. Government Printing Office, 1970, NASA SP-160.

[19] S. Timoshenko and J. Goodier, *Theory of Elasticity*, 3rd ed. New York: McGraw-Hill, 1970, pp. 213–256.

[20] A. Iula, R. Carotenuto, and M. Pappalardo, "An approximated 3-D model of the Langevin transducer and its experimental validation," *J. Acoust. Soc. Am.*, vol. 111, pp. 2675–2680, Jun. 2002.

[21] E. Moreno, P. Acevedo, M. Fuentes, A. Sotomayor, L. Borroto, M. E. Villafuerte, and L. Leija, "Design and construction of a bolt-clamped Langevin transducer," in *Proc. 2nd Int. Conf. Electrical and Electronics Engineering*, 2005, pp. 393–395.

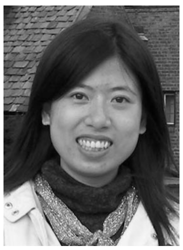
[22] M. Biet, F. Giraud, and B. Lemaire-Semail, "Squeeze film effect for the design of an ultrasonic tactile plate," *IEEE Trans. Ultrason. Ferroelectr. Freq. Control*, vol. 54, no. 12, pp. 2678–2688, Dec. 2007.

[23] A. Minikes and I. Bucher, "Coupled dynamics of a squeeze-film levitated mass and a vibrating piezoelectric disc: Numerical analysis and experimental study," *J. Sound Vibrat.*, vol. 263, no. 2, pp. 241–268, 2003.

[24] A. Kurganov and T. Eitan, "New high-resolution central schemes for nonlinear conservation laws and convection-diffusion equations," *J. Comput. Phys.*, vol. 160, no. 1, pp. 214–282, 2000.

Pinkuan Liu was born in 1969 in Hubei, China. He received the B.S. degree in precision machine and instrument engineering and M.S. and Ph.D. degrees in mechatronic engineering from Harbin Institute of Technology (HIT), Harbin, China, in 1991, 1998, and 2003, respectively.

From 1998 to 2003, he worked at the Robotics Institute of the HIT and was appointed an associate professor in August 2002. From September 2003 to August 2005, he worked as a postdoctoral fellow at the Institute of Machine and Production Engineering of Technical University of Braunschweig, Germany. He joined Shanghai Jiao Tong University (SJTU) in September 2005 and is currently working at the Research Institute of Robotics of the SJTU. His research interests include nanopositioning, micromanipulation, smart actuators and sensors, and precision parallel robots.

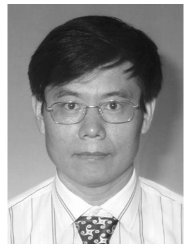


Jin Li was born in 1983 in Sichuan, China. She received the B.S. degree in mechanical engineering from Shanghai Jiao Tong University, Shanghai, China, in 2006.

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.

Han Ding was born in 1963 in Anhui, China. He received the Ph.D. degree from the Huazhong University of Science and Technology (HUST) in 1989. Supported by the Alexander von Humboldt Foundation, he worked at the University of Stuttgart, Germany, from 1993 to 1994.

From 1994 to 1996, he worked at the School of Electrical and Electronic Engineering, Nanyang Technological University, Singapore. From 1997 to 2001, he was a professor at HUST. He joined Shanghai Jiao Tong University in September 2001, where he is now a Cheung Kong Chair Professor (Special Appointment of the Yangtze Scholars Award Plan) and the director of the Robotics Institute. Dr. Ding was the recipient of the National Distinguished Youth Scientific Fund of China (former Premier Fund) in 1997. His research interests include robotics, manufacturing automation, smart sensors, and intelligent maintenance.



Wenwu Cao was born in 1957 in Jilin, China. He received his B.S. degree in theoretical physics from the Jilin University, Changchun, China, in 1982, and the Ph.D. degree in condensed matter physics from The Pennsylvania State University, University Park, PA, in 1987.

He is currently holding a joint appointment with the Department of Mathematics and the Materials Research Institute of The Pennsylvania State University as a professor of mathematics and materials science. He is also a faculty member in the Bioengineering Department and the Materials Science and Engineering at Penn State. His research interests are in both theoretical and experimental work in the areas of condensed matter physics and materials science, including theories on proper and improper ferroelastic phase transitions, static and dynamic properties of domains and domain walls in ferroelectric and ferroelastic materials, as well as performing measurements on second- and third-order elastic constants, linear and nonlinear dielectric constants, and piezoelectric constants in single crystals and ceramics. His current research includes nonlead piezoelectric materials, domain engineering process, nonlinear effects of solids, simulation design of piezoelectric sensors, transducers and actuators for underwater acoustics and medical ultrasonic imaging, ultrasonic NDE, and signal processing.

Dr. Cao is a member of the American Physical Society and the Materials Research Society.