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Hydroelastic Vibration of a Circular Diaphragm in the Fluid Chamber of a Reciprocating Micro Pump

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ABSTRACT

Reciprocating diaphragm micro-pumps are the most common type among indirectly–driven micro-pumps. This paper addresses the hydroelastic vibration of a circular elastic diaphragm interacting with the incompressible and inviscid liquid inside the cylindrical chamber with a central discharge opening. Taking into account axisymmetric vibration of the diaphragm, the fluid pressure exerted upon the plate is formulated using linear Bernoulli's equation. The kinematic and compatibility conditions are incorporated into the elastic vibration of the circular plate to derive the governing eigen-matrix equation. Numerical results are presented for different materials for diaphragm (silicon and glass) and pumped liquid (water and methanol). Normal frequencies of the coupled system, wet mode shapes of diaphragm and fluid oscillation modes are presented in numerical simulations. It is seen that the hydroelastic interaction lowers the natural frequencies considerably. However, the wet mode shapes for diaphragm vibration are very similar to the dry mode shapes. Finally, the effects of chamber height and operating fluid density on the normal frequencies are illustrated for the lowest four modes.

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1. INTRODUCTION

Spanning from DNA samplers [1] to cooling systems of microelectronic devices [2], micro scale fluid transport plays a significant role in an increasing number of engineering structures. Micro pumps as the essential component of micro fluidic delivery systems, based on their constitution and actuation mechanism, are categorized into several groups. The reciprocating diaphragm micro pumps are the most common type [3]. These pumps are composed of three main parts; elastic diaphragm, fluid chamber, and inlet/outlet valves. The actuation mechanism forces the diaphragm to deflect. Thus, it alters the volume of the fluid chamber, drives the fluid out during the discharge stroke and also draws it into the chamber during the suction stroke.

Design and optimal performance considerations require the dynamic characteristics of the micro pump be fully determined. Other than the properties of the diaphragm and working fluid separately, the interaction between them has been observed to be largely responsible for constituting the overall vibratory characteristics of micro pumps. Elastic diaphragms are most commonly circular or rectangular plates fabricated by etching silicon wafers and clamped over the cylindrical or cubic chambers [4, 5]. Therefore, the diaphragm-fluid interaction is basically attributed to the coupling between a clamped plate and its adjacent liquid. This subject has been investigated by numerous researchers so far. Kwak and Kim [6] used Henkel transform and Rayleigh quotients to introduce dimensionless added masses for axisymmetric vibration of circular plates on water. Later, Amabili and Kwak [7] expanded the same procedure to evaluate wet mode shapes of vibration for circular plates with any kind of uniform boundary conditions. Bauer [8] evaluated the asymmetric oscillation frequencies of an inviscid incompressible liquid contained inside a cylindrical cavity with an elastic plate cover. Same problem was also analyzed by Tariverdilo et al. [9] using two separate approaches; Fourier Bessel series and Hamiltonian Variational Principle. Jeong [10] and Jeong et al. [11], theoretically determined the natural frequencies of two identical circular/rectangular plates

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coupled by an incompressible, inviscid fluid employing Fourier series expansion and Rayleigh-Ritz method. Compressibility of fluid was also investigated in some articles. Jeong and Kim [12] added the compressibility assumption to the formulations of Ref. [10] to analyze the coupling of clamped plate submerged in a compressible liquid. Gorman and Horacek [13] applied Galerkin method to the combined Helmholtz and Euler-Bernoulli equations to analyze the coupled plate-fluid system. They used convergence of subsystem energies to reveal the sensitivity of the coupled structure to the assumed dry modes. Liu [14] investigated the electrostatic pull-in of circular diaphragm using a reduced order model and compared the outcome with the results of FEM. Francais and Dufour [15] also studied pull-in instability in electrostatically actuated micro pump accounting for hysteresis phenomenon. Jiankang and Lijun [16] employed shallow water assumption to the problem of thin micro pump to obtain the eigen-matrix equation governing the vibration of piezoelectrically diaphragm. They showed that while the natural frequencies were discernibly lowered by the liquid-plate coupling, the dry mode shapes of the diaphragm vibration were an acceptable template of wet diaphragm vibration. Using experimental verification, Ayela and Nicu [17] investigated frequency shift of circular diaphragms working with biomedical mixtures. Their work revealed that for liquids having viscosity less than 10cP, the viscous effects do not affect the dynamic characteristics of the micropump. Machauf et al. [18] proposed a novel structure for the electrostatic micro pump. Capitalizing on the electric permittivity of the working fluid, they showed that decreasing the fluid gap favors the flow rate in a nonlinear manner. In micro and nano applications, the thickness of elements is typically on the order of microns and sub-microns. At this scale, size effect could affect the analyses results. On the other hand, Sadeghian et al. [19] showed that for silicon thickness greater than $1\mu m$, the effective Young's modulus reaches an asymptotic equal to its macro-scale value (see also experimental results reported by [20-23]). Accounting for the thickness of plates considered in this study, analysis of obtained results will not be affected by size. This paper investigates the fluid structure coupling between the circular clamped plate and the partially bounded cylindrical liquid chamber adjacent to it. The liquid is assumed inviscid and incompressible. The plate is assumed to deflect axisymmetrically. Considering compatibility of deflections and using Fourier-Bessel series, the governing relation is obtained in terms of an eigen-matrix equation where the effect of the liquid emerges as an added mass matrix. Vibratory mode shapes are evaluated for diaphragm and liquid and the structure is investigated for the change in its natural frequencies under various geometrical and material conditions.

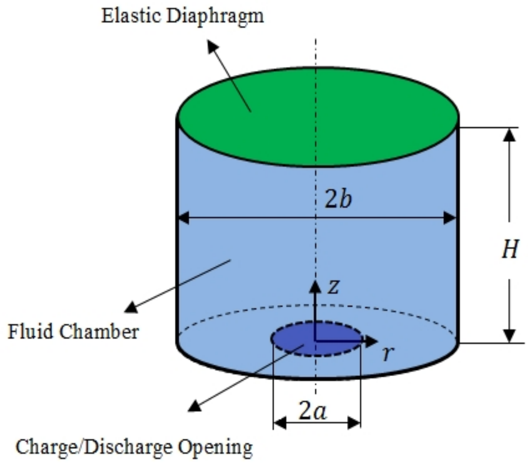

Figure 1. Cylindrical reciprocating micro pump with central opening

2. MATHEMATICAL MODEL

The considered micro pump structure is shown in Figure 1. It consists of a cylindrical chamber of radius *b* and height *H*. Top surface of the pump is assumed to be an elastic circular micro plate (diaphragm) which is deflected when acted upon by the force of the actuation mechanism. Amongst the actuation methods used in diaphragm micro pumps, piezoelectric and electrostatic actuations are the most common type where each one has its own merits. Piezoelectric actuation is easy to control and does not face dynamic instabilities. On the other hand, electrostatic actuation is easy to mount and integrate. In order to study the free vibration of the diaphragm, the actuation mechanism is not included in the figure.

Bottom surface of the chamber is composed of an annular rigid section and a central hole through the contained liquid of density ρ which is pumped out during discharge course and sucked in during suction course. The bottom opening is connected to the inlet and outlet valves (not shown here) which rectify the fluid flow in/out of the pump. A cylindrical coordinate system is defined (as shown in the figure) with the center at bottom and z axis directed upward. The analysis is based on the assumption that liquid sloshing is inviscid, incompressible and irrotational. Liquid movement inside the chamber may be defined by the velocity potential function *Φ* for the boundary value problem which is stated as follow:

$$\nabla^2\phi = 0, \quad 0 \le r \le R, \; 0 \le z \le H \qquad \text{(1a)}$$

$$\left.\frac{\partial \phi}{\partial r}\right|_{r=b} = 0 \qquad \text{(1b)}$$

$$\left.\frac{\partial\phi}{\partial z}\right|_{z=0} = \begin{cases} 0 & a<r<b \\ \frac{1}{\pi a^2}\int_0^b \frac{\partial w}{\partial t}(2\pi r)\,dA & 0<r<a \end{cases} \quad (1c)$$

$$\left.\frac{\partial\phi}{\partial z}\right|_{z=H} = \frac{\partial w}{\partial t} \quad (1d)$$

where, w denotes the vertical deflection of the diaphragm. The Laplace's equation (Equation (1a)) is a statement of incompressibility inside the chamber domain and boundary condition (Equation (1b)) denotes the impermeability condition on the side wall. It is assumed that liquid velocity along the bottom opening is radius-independent. Thus, it is directly proportional to the rate volume change which is caused by the diaphragm deflection (Equation (1c)). The kinematic compatibility of velocities between the moving diaphragm and its adjacent liquid is stated by Equation (1d). In order to obtain preliminary formulation for the velocity potential, the Laplace's equation is solved using method of separation of variables and accounting for the boundary condition (1b) which leads to a linear combination of permissible modes:

$$\phi(r,z,t)=\sum_{m=1}^{\infty} J_0(\alpha_m r)[X_m(t)\cosh(\alpha_m z)+Y_m(t)\sinh(\alpha_m z)] \quad (2)$$

where, $m=1, 2, \ldots$ and the radial eigenvalues α_m are obtained by satisfying B. C. at the side wall:

$$J_0'(\alpha_m b)=0 \quad (3)$$

Unknown time-dependent factors X and Y are to be determined later. The coupled equation of motion of the diaphragm having uniform density ρ_p and thickness h can be written in the following form:

$$\frac{Eh^3}{12(1-v^2)}\nabla^4 w + \rho_p h\frac{\partial^2 w}{\partial t^2} = P \quad (4)$$

where, E and v denote the Young's modulus and Poisson's ratio, respectively. The hydrodynamic liquid pressure, P, can be obtained from the linearized Bernoulli's equation:

$$P = -\rho_f \left.\frac{\partial\phi}{\partial t}\right|_{z=H} \quad (5)$$

where, ρ_f is the fluid density. The lateral deflection of the diaphragm may be formulated as the summation of natural vibration mode shapes of a circular plate in air (ψ_n):

$$w(r,t)=\sum_{n=1}^{\infty} q_n(t)\psi_n(r), \quad n=1,2,3\ldots \quad (6a)$$

$$\psi_n(r)=J_0(\beta_n r)I_0(\beta_n b)-J_0(\beta_n b)I_0(\beta_n r) \quad (6b)$$

where, q_n are the unknown generalized coordinates of plate vibration. The values of eigenvalues β_n which satisfy the clamped boundary conditions of the plate are shown in Table 1.

TABLE 1. Radial eigenvalues for plate vibration and fluid oscillation

m, n	**1**	**2**	**3**	**4**	**5**	**6**
$\beta_n b$	3.196	6.306	9.434	12.587	15.740	18.843
$\alpha_m b$	3.831	7.015	10.185	13.338	16.491	19.594

In order to calculate the unknown factors, Equations (4) and (6) are inserted into Equations (1c), (1d), so we have:

$$\sum_1^{\infty}\alpha_m Y_m J_0(\alpha_m r)=\begin{cases} 0 \\ \frac{1}{\pi a^2}\int_0^b \sum_1^{\infty}\psi_n(r)\dot{q}_n(t)(2\pi r)\,dr \end{cases} \quad (7)$$

$$\sum_{m=1}^{\infty}\alpha_m J_0(\alpha_m r)(X_m\sinh(\alpha_m H)+Y_m\cosh(\alpha_m H))=\sum_{n=1}^{\infty}\psi_n(r)\dot{q}_n(t) \quad (8)$$

$$\frac{Eh^3}{12(1-v^2)}\sum_{n=1}^{\infty}\nabla^4\psi_n(r)q_n(t)+\rho_f h\sum_{n=1}^{\infty}\psi_n(r)\ddot{q}_n(t)= \lrcorner$$
$$-\rho_f\sum_{m=1}^{\infty}J_0(\alpha_m r)(\dot{X}_m\cosh(\alpha_m H)+\dot{Y}_m\sinh((\alpha_m H)) \quad (9)$$

Upon using the orthogonality of Bessel functions (for both velocity potential and plate radial modes), and defining the following integrands:

$$\delta_n=\int_0^b r\psi_n(r)dr \qquad \lambda_{mn}=\int_0^b rJ_0(\alpha_m r)\psi_n(r)dr$$

$$\gamma_m=\int_0^b rJ_0^2(\alpha_m r)dr \qquad k_n=\frac{Eh^3}{12(1-v^2)}\int_0^b \psi_n(r)\nabla^4\psi_n(r)dr \quad (10)$$

$$\kappa_m=\int_0^a rJ_0(\alpha_m r)dr \qquad \xi_n=\rho_p h\int_0^b r\psi_n^2(r)dr$$

the governing equations may be restated by the following matrix equations:

$$[S_1]_{M\times M}\{Y\}_{M\times 1}=[S_2]_{M\times N}\{\dot{q}\}_{N\times 1} \quad (11)$$

$$[S_4]_{M\times M}\{X\}_{M\times 1}+[S_5]_{M\times M}\{Y\}_{M\times 1}=[S_6]_{M\times N}\{\dot{q}\}_{N\times 1} \quad (12)$$

$$[M]_{N\times N}\times\{\ddot{q}\}_{N\times 1}+[K]_{N\times N}\times\{q\}_{N\times 1}=-\rho_f\left([S_8]_{N\times M}\times\{\dot{X}\}_{M\times 1}+[S_9]_{N\times M}\times\{\dot{Y}\}_{N\times 1}\right) \quad (13)$$

Note that the assumed modal summations are truncated to N modes for diaphragm vibration and M modes for liquid oscillation. The coefficient matrices are specified as:

$$S_1(m,m)=\alpha_m\gamma_m \qquad S_2(m,n)=\frac{2}{a^2}\kappa_m\delta_n$$
$$S_3=S_1^{-1}\times S_2 \qquad S_4(m,m)=\alpha_m\gamma_m\cosh(\alpha_m H) \quad (14)$$

$S_6(m,n) = \lambda_{mn}$ $\quad S_5(m,m) = \alpha_m \gamma_m \sinh(\alpha_m H)$

$S_8(n,m) = \lambda_{mn} \cosh(\alpha_m H)$ $\quad S_9(n,m) = \lambda_{mn} \sinh(\alpha_m H)$

$M(n,n) = \xi_n$ $\quad K(n,n) = k_n$

Finally, by evaluating the unknown factors X and Y from Equations (11) and (12) and substituting into Equation (13), the dynamic equation for the lateral movement of the diaphragm with its surrounding fluid is obtained as:

$$[K]\{q\} + [M + M']\{\ddot{q}\} = 0 \tag{15a}$$

$$M' = \rho_f \left(S_8 \left[S_5^{-1} S_6 - S_5^{-1} S_4 S_3 \right] + S_9 S_3 \right) \tag{15b}$$

It is observed that effect of the contained liquid on the vibration of the elastic diaphragm emerges as an equivalent added mass. Thus, the coupled system will possess lower values of natural frequency than the dry diaphragm.

3. NUMERICAL RESULTS

In this section, the natural frequencies and mode shapes of the couple structure is calculated. The geometrical and material properties used in the analysis are shown in Table 2. Two materials are considered for the diaphragm; silicon and Pyrex glass which are used most in micro scale pumps (Ref. [3]). Moreover, the pumping fluid properties are given according to two commonly used liquids in micro fluidic devices with electrostatic actuation mechanism; water and methanol.

Figure 2 shows five natural frequencies of the coupled system (refered to as wet frequencies) and the frequencies of diaphragm without fluid coupling (refered to as dry frequencies). It is observed that, as anticipated, the wet frequencies increase as exponential order. They are also lower than the dry ones and the difference is larger for dense fluids. The value of the frequencies is considerably lower in the case of Pyrex glass diaphragm which is less stiffer compared to silicon.

Diaphragm deflection mode shapes with and without fluid coupilng are illustrated in Figure 3. The change in the mode shapes is not the same for different modes. It shows the radial distribution of added mass on the diaphragm for each vibration mode. Corresponding to these, are the fluid velocity patterns inside the pump chamber as shown in Figure 4 by the use of velocity vectors.

The normal frequencies of the system defined as the ratio of wet frequency to dry frequency are depicted in Figure 5 for various H/b values. Normal frequencies possess an ascending trend with tendency to reach convergence at higher modes. The higher value of normal frequencies indicates that the effect of fluid added mass tends to fadeat higher modes. It means that less amount of fluid succeeds to co-vibrate with the moving diaphragm. Moreover, the frequencies increase as the fluid height increases. This effect is more obvious for small chamber heights.

TABLE 2. The Data used in the calculations

Parameter	Value
Pump radius, b	100 μm
Pump chamber height, H	50 μm
Discharge radius, a	50μm
Diaphragm thickness, h	2μm
Young's modulus, *E*	
Silicon	162GPa
Pyrex glass	67GPa
Diaphragm density, ρ_d	
Silicon	2300 Kg/m^3
Pyrex glass	2230 Kg/m^3
Fluid density, ρ_f	
Water	1000 Kg/m^3
Methanol	750 Kg/m^3
Poisson's ratio, *v*	
Silicon	0.22
Pyrex glass	0.23

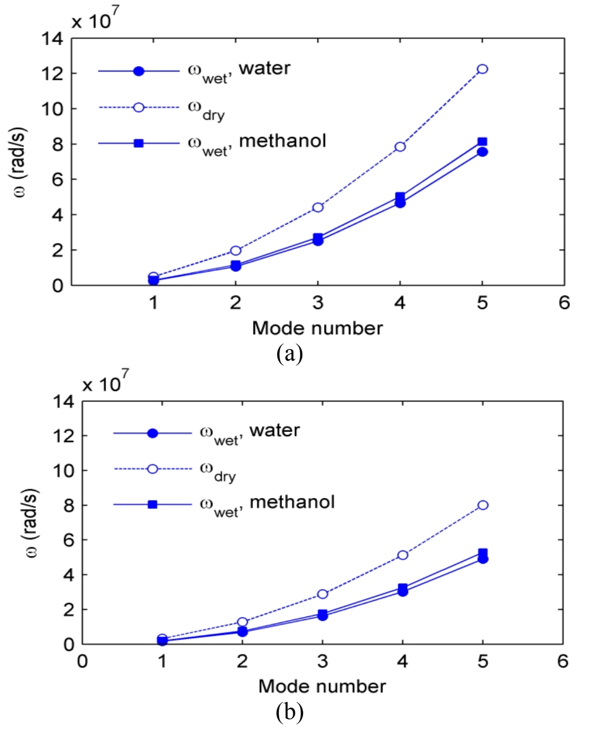

Figure 2. Natural frequencies of the diaphragm with and without fluid coupling, (a) Silicon diaphragm, (b) Pyrex glass diaphragm

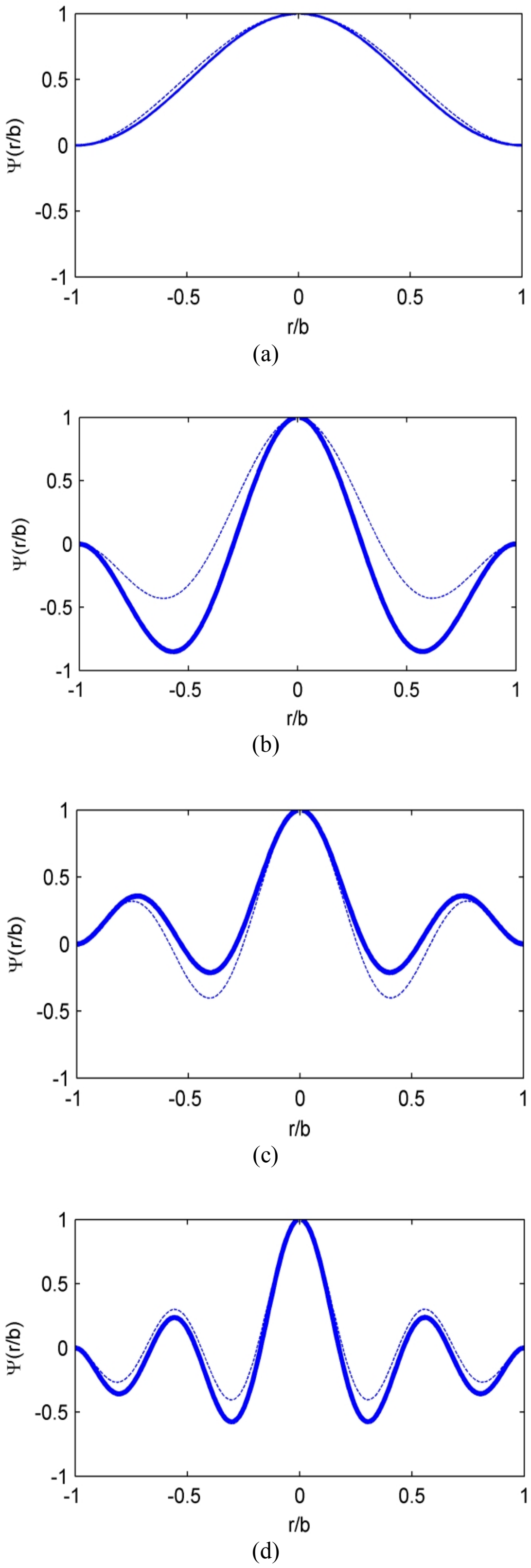

Figure 3. Mode shapes for diaphragm vibration (dashed line: dry mode, solid line: wet mode), (a) First mode, (b) Second mode, (c) Third mode, (d) Fourth mode

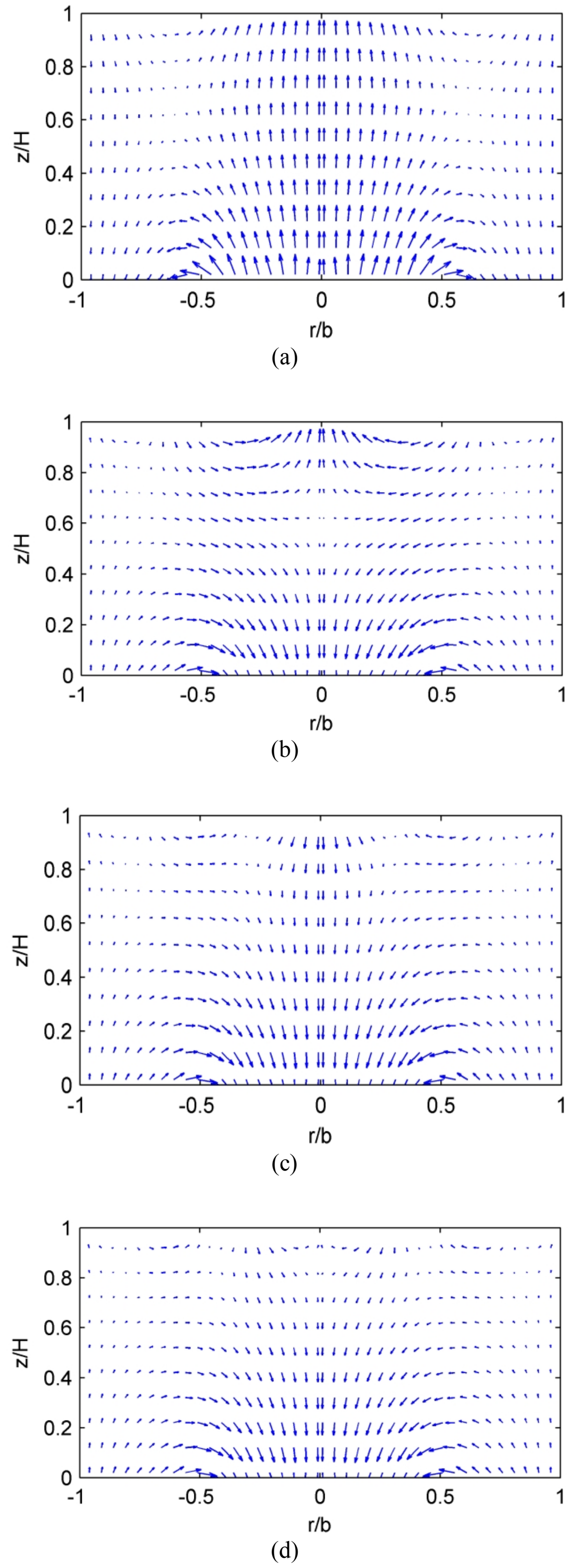

Figure 4. Oscillation modes of the liquid inside chamber (a) First mode, (b) Second mode, (c) Third mode, (d) Fourth mode

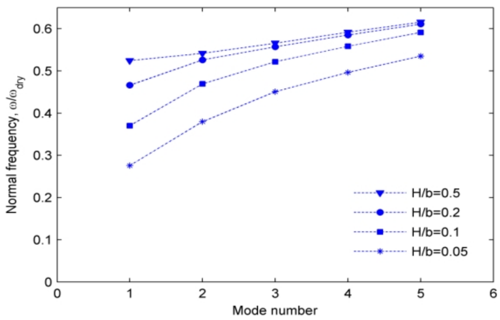

Figure 5. Normal Frequencies for different height to radius ratios

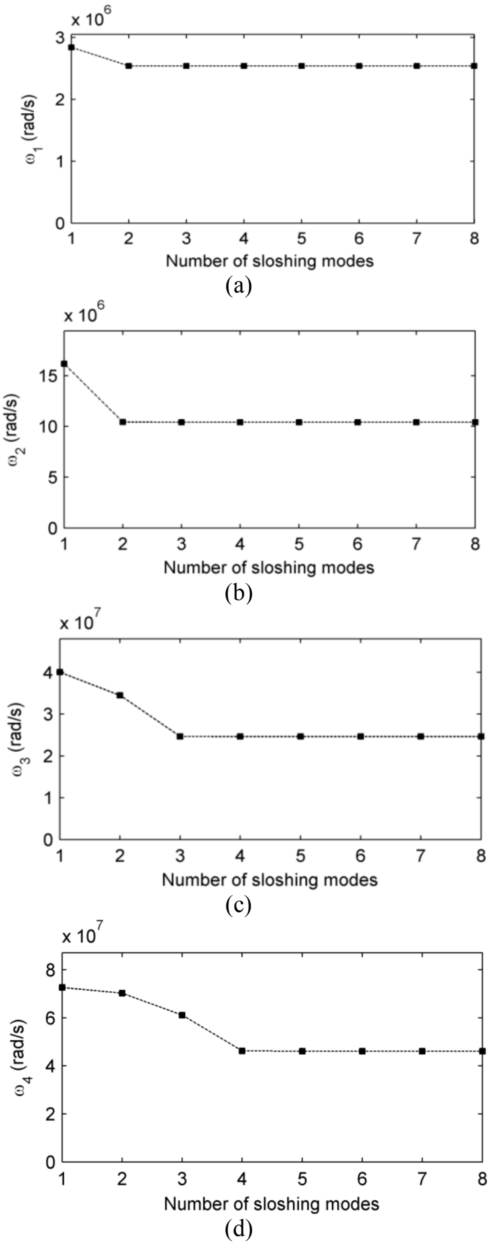

Figure 6. Convergence of natural frequencies based on sloshing modes (a) First mode, (b) Second mode, (c) Third mode, (d) Fourth mode

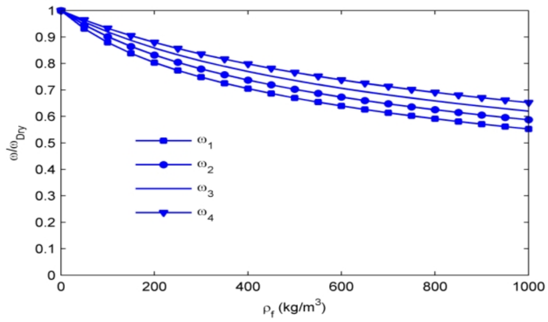

Figure 7. Effect of fluid density on natural frequencies

Convergence diagrams in terms of number of fluid oscillation modes are shown in Figure 6. It is observed that higher modes frequencies need more modes considered in the fluid oscillation.

Finally, the influence of fluid density on the normal frequencies of the coupled diaphragm is shown in Figure 7. As observed, the value of normal frequency for all modes becomes one when the fluid density is set equal to zero (no fluid) indicating the dry diaphragm condition. The normal frequencies tend to decrease as the density increases. This trend has a pronounced effect at lower densities. It is once again noticed that higher modes frequencies are less affected by density alterations compared to lower modes.

The smaller change of wet frequencies in higher modes could be explained by referring to Figure 4, which indicates the fluid field velocity associated with each mode. Comparing the fluid field of the first mode with those of third or fourth modes reveals that the fluid velocity near elastic diaphragm in the first mode is substantially higher than those for the third and fourth modes. This means that the associated added mass for the higher modes will be smaller compared to the first mode. Therefore, it is anticipated that the change in the natural frequency of the higher modes becomes smaller.

4. CONCLUSION

The problem of fluid-structure interaction was investigated in the case of reciprocating micro pump. The elastic diaphragm was assumed in form of a clamped circular plate covering the cylindrical liquid chamber. The chamber laterally is bounded to rigid side wall while vertically partially bounded at the bottom surface. The method of Fourier-Bessel series was used using modal expansions for plate deflection and fluid velocity potential. Assuming incompressibility, fluid pressure was simulated by the use of Bernoulli's equation leading to a coupledeigen-matrix equation for the vibration of diaphragm. The absolute and normalized values of natural frequencies were depicted for different material and geometrical specifications.

The compatibility of the coupled system was illustrated by plotting operating fluid mode shapes. It was shown that the wet frequencies were lower than the dry frequencies and converged as the fluid oscillation modes increased. Fluid chamber height proved to have a favorable effect on increasing the frequencies especially at lower modes. It was observed that lower modes frequencies were more affected by a change in density compared to higher ones.

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Circular Plate

چکیده

میکرو پمپهای نوع دیافراگمی نوع متداول از میکروپمپهای عمل کننده غیر مستقیم می باشند. عملکرد آنها بر اساس حرکت رفت و برگشتی دیا فراگم به همراه شیرهای یک طرفه مربوطه می باشد. در این مقاله ارتعاشات مربوط به دیافراگم مدور یک میکرو پمپ در مجاورت سیال تراکم ناپذیر و غیر لزج که در داخل محفظه استوانه ای قرار دارد مورد بررسی قرار می گیرد. در مرحله مدلسازی فرض می گردد که محفظه میکرو پمپ دارای یک مجرای خرجی مرکزی در قسمت تحتانی می باشد. با در نظر گرفتن ارتعاشات متقارن برای غشاء، فشار ناشی از سیال مجاور بر مبنای معادله خطی برنولی مدلسازی شده است. در استخراج فرم ماتریسی معادلات حاکم، شرایط سینماتیکی و سازگار سیال–سازه بکار گرفته شده است. در مرحله شبیه سازی، مواد مختلفی برای دیافراگم و سیالات متفاوتی بعنوان سیال عمل کننده در نظر گرفته شده است. فرکانسهای طبیعی سیستم کوپل شده، شکل مودهای دیافراگم خیس و شکل مودهای سیال عمل کننده در نتایج شبیه سازی ارائه شده اند. نتایج شبیه سازی نشان می دهند که علی رغم تفاوت جزئی در شکل مودهای غشاء خشک و خیس، فرکانسهای سیستم کوپل شده به شدت تحت تاثیر سیال عمل کننده می باشند. در نهایت اثرات ارتفاع محفظه میکرو پمپ و تغییرات دانسیته سیال عمل کننده بر روی چهار مود اول سیستم مورد بررسی قرار گرفته است.

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