FILM AIR BEARINGS GENERATED BY ULTRASONIC VIBRATIONS

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Abstract

Film air bearings generated by a piezoelectric bimorph present many advantages in comparison with conventional air bearing systems, because they are autonomous and they take a reduced place. This paper first presents the working principle, then theoretical formulation for the fluid domain. Several prototypes are designed using the ATILA finite element code and are built. The experimental procedure is detailed with a view to measuring the air bearing thickness and its relative variations.

Introduction

With a view to obtaining specific displacement conditions, such as friction cancellation, higher speed or non contact transportation of planar objects, film air bearings using piezoelectric bending elements [1] have many advantages in comparison with conventional air bearing systems, because they are autonomous and they take a reduced place.

To generate an air bearing, a high-pressure difference has to be created between the external pressure and the pressure in the fluid layer. Static air bearings require an external source of pressurized air, and aerodynamical bearings are subject to wear during startup/shutdown phases. Film air bearings using a piezoelectric bimorph allow to avoid these disadvantages. Air bearing is performed by a vibration of either the wearing surface [2-4], or the object to be lifted.

In this paper, after the explanation of the working principle, two theoretical models describing the pressure in the fluid are presented : the first one uses the polytropic law, the second one the acoustic radiation pressure [5]. The two approaches give exactly the same results. Several prototypes are then designed using the ATILA finite element code and are built [6]. Finally, the experimental procedure is detailed with a view to measuring the air bearing thickness and its relative variations. It relies upon the measurement of the air bearing capacitance, using amplitude modulation. Adding several masses on the bimorph, the variations of the air bearing thickness and its relative variations are studied. Comparing experimental results with numerical results allows to understand physical behavior of the system and to show the limitations of some hypothesis.

Working principle

In the considered system, the film air bearing is obtained by imposing vibrations on the object to be lifted (Fig. 1). Therefore, a piezoelectric bimorph (Fig. 2), made of one passive disk stuck on one piezoelectric disk is used. The system is excited at the frequency of its first bending mode. The nodal vibration line is used for the connection system, thus all the vibration energy is confined in the element. For a sufficiently high vibration frequency, an air bearing appears between the bimorph and the ground.

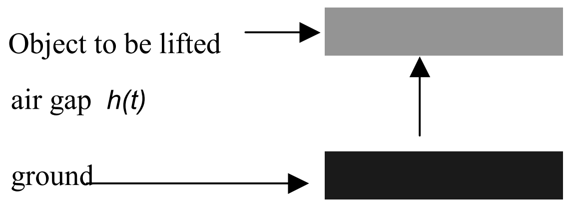

Figure 1: Vibrating surface creating a film air bearing.

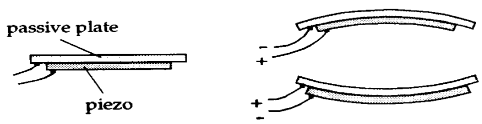

Figure 2: Bending motion of the piezoelectric bimorph

Theoretical formulation : polytropic law

Piston motion of the surface.

Considering the vibrating surface as a piston, and if h_0 is the averaged thickness of the air gap, then the thickness h is written as:

$$h = h_0(1-\xi\cos(\Omega t)) \qquad (1)$$

where ξ is the relative variation of the thickness of the air gap. When the vibration frequency is high enough (around 20 kHz) and when the air thickness is small (around 10µm), the lateral air flux is reduced to zero. Gas remains stationary and is undergoing periodic cycles of compression and decompression. If gas is considered as ideal, then the polytropic approximation is:

$$p\,V^k = constant \qquad (2)$$

Where p is the pressure of the gas, V film volume, and k = 1 or 1.4 if isotherm or adiabatic conditions are considered. Then, air pressure is:

$$p = p_0(V_0/V)^k = p_0(h_0/h)^k = \frac{p_0}{(1-\xi\cos(\Omega t))^k} \qquad (3)$$

where V_0 corresponds to the gas volume when the thickness is h_0, p_0 is the surrounding air pressure. To know the averaged pressure p_m, p of equation (3) is integrated on one cycle. It is always greater than p_0. The pressure difference $p_{diff} = p_m - p_0$ is the pressure that allows to compensate the gravitational forces and then the object is in levitation.

A first simple method consists in doing limited development of equation (3) :

$$p=p_0(1+k\xi\cos(\Omega t)+k(k+1)\xi^2\cos^2(\Omega t)/2...) \quad (4)$$

Then, the average on one cycle gives:

$$p_{diff} \approx p_0\, k(k+1)\xi^2/4 \quad (5)$$

The lifted mass is, with the displacement $U = h_0\, \xi$:

$$M \approx p_0\, k(k+1)U^2\, \pi R^2 / 4gh_0^2 \quad (6)$$

Relation (6) gives a first estimation of the weight that can be in levitation.

Bending motion of the surface

In the actual system, bimorph has not a piston motion but a bending motion. Therefore, the previous equations are modified to take into account the displacement distribution as a function of the radius $\xi(r) = U(r)/h_0$. Then, the lifting force is:

$$F = \int_0^R \left(\frac{1}{T} \int_0^T \frac{p_0\, dt}{\left(1-\xi(r)\cos(\Omega t)\right)^k} - p_0 \right) 2\pi r\, dr \quad (7)$$

Considering small $\xi(r)$, then the lifted mass is:

$$M \approx p_0\, k(k+1)U_{eq}^2\, \pi R^2 / 4gh_0^2 \quad (8)$$

with

$$U_{eq}^2 = \frac{2}{R^2} \int_0^R U^2(r)\, r\, dr \quad (9)$$

Relation (8) is very similar to the lifted mass in the piston case, but here the displacement distribution is taken into account.

Theoretical formulation : acoustic approach

The study of the acoustic radiation pressure, which is an area of non linear acoustics, appears to be necessary for the understanding of the problem [5]. It is the mean excess pressure experienced by a material surface in a sound wave. It is defined on the material surface only. In the problem, considering a piston motion of the plate, let us write the displacement of a fluid particle $u(x)$ in the fluid layer as:

$$u(x) = x \frac{U}{h_0} \cos(\Omega t) \quad (10)$$

Where x is the direction perpendicular to the plate plane. For $x = 0$, the displacement of the fluid particle is equal to 0 and for $x = h_0$, the displacement is equal to U. Eq. (36) of [5] gives the excess pressure $\langle p^L - p_0 \rangle$ in this 1D case, in the Lagrangian system :

$$\left\langle p^L - p_0 \right\rangle = \rho_0 c_0^2 \left[-\left\langle \frac{\partial u(x)}{\partial x} \right\rangle + \frac{k+1}{2} \left\langle \left(\frac{\partial u(x)}{\partial x} \right)^2 \right\rangle \right] \quad (11)$$

Where ρ_0 is the ambient density and c_0 is the ambient sound speed. Eq. (29) of [5] gives the excess pressure $\langle p^E - p_0 \rangle$ in this 1D case, in the Eulerian system :

$$\left\langle p^E - p_0 \right\rangle = \left\langle p^L - p_0 \right\rangle + \rho_0 c_0^2 \left[\left\langle u(x) \frac{\partial^2 u(x)}{\partial x^2} \right\rangle \right] \quad (12)$$

These expressions are related to the conservation of mass in a closed system. In our case, taking the time average, using Eq. (10), both Eq. (11) and (12) give exactly Eq. (5). Thus, the acoustic radiation approach is an alternative approach to get the value of the pressure difference and of the lifted mass. Moreover, only the linear wave equation is solved to determine the wave.

In the same way, if a bending motion is considered instead of a piston motion of the plate, Eq (8) and (9) are obtained using an acoustic approach.

In the 3-D case, considering the conservation of the mass in a closed system, a numerical approach, using the finite element method [6], is necessary. The work is under progress.

Prototypes

Many prototypes have been built. First, a finite element analysis with the help of the ATILA code [6] allows the resonance frequencies and the position of the nodal point, used to fix the system, to be determined. To get a volumic displacement non-equal to zero on the vibrating face of the prototype, the bimorph transducer is partially clamped by a greater mass at the circumference, which total thickness is 4.5 mm (Fig. 3). The thickness at the center of the brass disk is 1.5 mm; the thickness of the PZT is 1 mm. The external radius of the PZT disk is 15 mm, whereas the external radius of the brass is 18.5 mm. The resonance frequency is 13.5 kHz. At the position of the nodal point, a rubber ring allows to add variable masses on the system (Fig. 4).

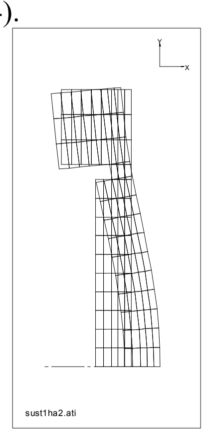

Figure 3: Prototype. Displacement field at the frequency resonance is superposed to the rest position.

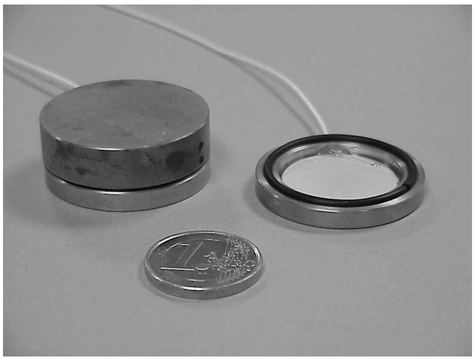

Figure 4. Picture of the prototype.
A rubber ring (right picture) allows variable masses on the system (left picture) to be added

Experimental set-up

Electrical measurement of the air gap thickness.

First, resonance and antiresonance frequencies as well as damping are measured with the help of the impedance.

The basic circuit used to measure the thickness of the air gap is a capacitive divider (Fig. 5)

C1 is a given capacitance, *Ca(t)* is the capacitance of the air-gap, the electrodes are one on the vibrating face of the prototype, the other on the static plane that has to be conductor. A HF generator, which pulsation is ω, supplies the capacitive divider. Because *C1* is smaller than *Ca(t),* then, *V2(t)* can be written as :

$$V2(t) \approx V1\,\sin(\omega t)\,\frac{C1}{\varepsilon_0 S}\,h_0\,(1-\xi\cos(\Omega t)) \qquad (13)$$

It corresponds to an amplitude-modulated signal and its modulation rate is directly related to ξ, relative variation of the air gap thickness. Because *C1*, ε_0 and *S* are known, h_0 and ξ are easily determined with the help of *V2 (t)*.

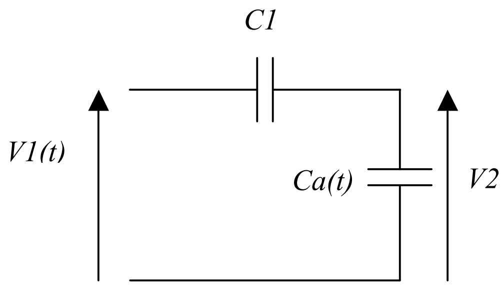

Figure 5: Scheme used for the measurement of capacitance of the air gap thickness

Because of the bending motion of the bimorph, instead of a piston motion, then, the capacitance *Ca(t)* is written as:

$$Ca(t) = \iint_S \frac{\varepsilon_0\, dS}{h_0\left(1-\xi(r)\cos(\Omega t)\right)} = \frac{2\pi\,\varepsilon_0}{h_0}\int_0^R \frac{r\,dr}{\left(1-\xi(r)\cos(\Omega t)\right)} \qquad (14)$$

Results

Experimental results

Several given masses are added on the prototype. Table 1 presents the results for a constant excitation (V= 27 Veff, i = 21 mAeff) and different loadings.

With the averaged capacitance, the averaged thickness h_0 is deduced: $h_0=\varepsilon_0 S/Ca$, with ε_0 the permittivity and *S* the surface of the disk. Then, with the help of the amplitude modulation rate *(ΔV/V)* and of the numerical displacement distribution on the bimorph (Fig. 6), the displacement at the center of the bimorph *U(r=0)* is deduced. Even if the numerical displacement distribution is used to deduce the displacement at the center of the bimorph, knowing the averaged displacement, this result is referred as the *experimental displacement*.

More important is the loading, smaller is the averaged thickness of the air. The value of the displacement at the center of the bimorph *U(r=0)* is identical (around 12µm), because the excitation is identical in all the cases, and because there is a mechanical decoupling between the bimorph and the masses. The difference observed for the smaller mass has to be explained.

Total mass (g)	Averaged Capacitance *Ca(t)*	*ΔV/V* (%)	h_0 (µm)	*U(r=0)* (µm)
26.5	244 pF	2.65	39	9.36
115	338 pF	5.5	28.3	12.73
174	411 pF	6.85	23.4	12.2
188	420 pF	8	22.9	13.2

Table 1. For a constant electrical excitation and for different masses, averaged capacitance *Ca(t)*, amplitude modulation rate *(ΔV/V)*, averaged thickness of the air gap displacement h_0, displacement at the center of the bimorph *U(r=0).*

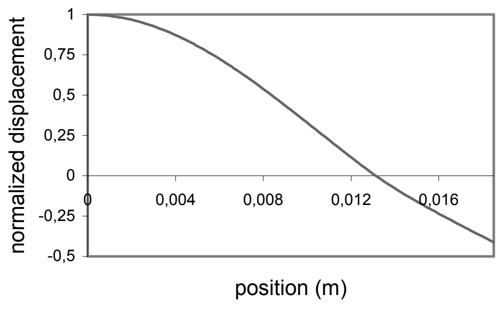

Figure 6 : Displacement of the transducer as a function of the position.

Theoretical and numerical results

The value of *k* in Eq. (2) (polytropic approach) and in Eq. (11) (acoustic approach) raises the question of the acoustic in the air layer, which is analyzed using

adiabatic [3] or isotherm [4] conditions. In air, the thermal characteristic length is 6E-8 m. At 13.5 kHz, the thermal boundary-layer thickness is around 20 µm, which is the same order of magnitude than the thickness of the air gap (Table 1). The value of k is neither 1 nor 1.4 and an intermediate condition has to be used [7]. Nevertheless, Eq. (8) can be used to determine the value of k, using the following procedure.
A linear finite element analysis has been carried out with the help of the ATILA code [6]. An harmonic analysis is performed, for the second case of table 1. The transducer is meshed, with a 28-µm air gap (Fig. 7) and is excited at 13.5 kHz, with a current equal to 21 mA. The variations of the displacement of the bimorph as a function of the radius give the $U(r)$ function (Fig. 6). By changing the sound speed in the fluid layer, two extreme cases are considered : first the fluid in the layer is adiabatic, second the fluid in the layer is isotherm. One can notice that the displacement of the structure is roughly independent of the condition in the fluid. Using this displacement $U(r)$, U_{eq} is calculated (Eq. 9). Because M is known (115 g), Eq. (8) gives a second degree equation in k. The value of k deduced by this method is not correct, because the U_{eq} value from the finite element modeling is too high. Many hypothesis have been studied to explain the mistake. First, the disk has been polished by hand and thus the surface is not perfectly plane: the external border is more eroded than the center. A finite element calculation has shown that U_{eq} from Eq (9) decreases as the active surface of the disk (i.e. R) decreases. The small differences have an important impact on the $\Delta V/V$ values but the value of h_0 is correct. Then, one can also ask the question of the lateral flux. Therefore, the previous mesh is used (Fig. 7). Fig. 8 presents the variations of the pressure in the air layer. Using the pressure derivative in the radial direction, the relative variation of the fluid volume on one cycle can be calculated. Calculations have been performed for $k = 1$ and $k = 1.4$ in the fluid layer. The results are different in both cases and the relative variation of the fluid volume on one cycle is around several percents. Thus the pumping effect is not negligible.

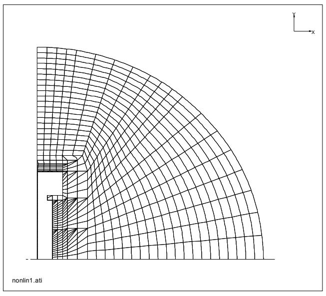

Figure 7: Finite element mesh of the fluid domain for the linear finite element analysis.
Transducer is not meshed.

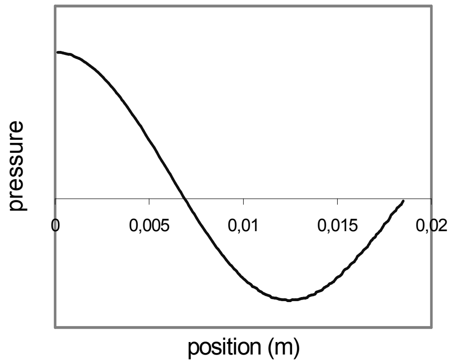

Figure 8: Variations of pressure in the air gap versus the position.

Conclusion

A film air bearing system has been designed and built. When the structure is idealized, the pressure in the fluid layer has been determined using the polytropic law and the acoustic radiation pressure. Both approaches have given the same results. Then, using the electrical measurement of the capacitance, the averaged thickness and its relative variations have been evaluated. Differences have been observed between experimental results and theoretical results. Thus, a numerical approach [6] is required to calculate the conservation of the mass in a closed system. Experimental analysis will now be carried out using a laser vibrometer, to measure with precision the averaged thickness, its relative variations and the displacement distribution on the bimorph.

References

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