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VORTEX LEVITATION

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ABSTRACT

In this paper, a new pneumatic levitation method, called vortex levitation, is introduced. Vortex levitation can achieve non-contact handling by blowing air into a vortex cup through a tangential nozzle to generate a swirling air flow. First, its basic characteristics are investigated by measuring the pressure distribution and lifting force. We confirmed that negative pressure is caused by the centrifugal force of the swirling air flow to apply a lifting force. It was also revealed that the pressure distribution varies as the gap thickness between the vortex cup and the work piece changes, and thereby the lifting force is dependent on the gap thickness. Thereafter, its dynamical characteristics are investigated and we proposed a spring–damper model by modelling the lifting force caused by swirling air flow and the resistive force related to squeeze film air damping separately. The proposed dynamical model is verified by means of an experiment of loading process. The calculation result can reach a good agreement with the experimental result.

Keywords: Non-contact handling, Swirling air flow, Squeeze damping, Dynamical model

INTRODUCTION

Usually a work piece is brought into contact with a handling device in order to be picked up and moved. Such contact methods are often accompanied by surface scratching and static electricity. Especially in the semiconductor production process, the inherent disadvantages of contact handling often lead to defective products. Therefore, many non-contact handling approaches had been proposed and have proven effective. Among them pneumatic levitation approaches, which use air flow to apply a lifting force to a work piece, are put into various practical applications because air flow is magnetic free, generates little heat, requires no control loop to obtain a stable state and nearly maintenance free for their simple structures. One typical pneumatic approach based upon Bernoulli principle that is most often used in practical applications is called Bernoulli levitation. However, its large air consumption often leads to a great air power loss through supply pipes. This paper introduces a new pneumatic non-contacting handling approach named vortex levitation that uses air swirling flow. As reported in many previous studies, negative pressure can be generated by swirling air flow and its technical applications include separation of particles and improvement of combustion, among others. Consequently, vortex levitation applies negative pressure caused by swirling flow to achieve non-contact handling. Vortex levitation is characterized by its low air consumption.

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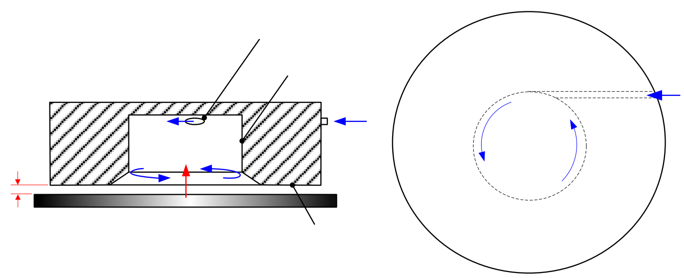
STEADY-STATE CHARACTERISTICS

Mechanism of Vortex Levitation

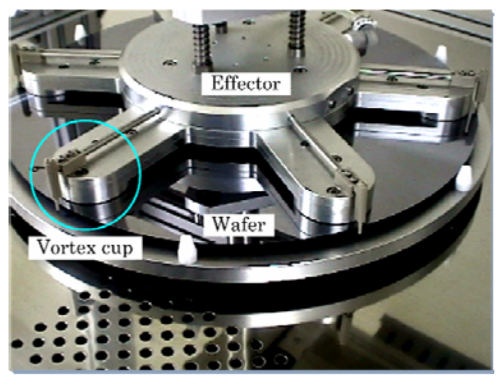
Similar to cyclones where low pressure is caused by air swirling, a simple structure called the vortex cup (hereafter referred to as cup) is used to generate an air swirling flow. As can be seen in Fig.1(a), the cup is made up of a circular cylinder and a tangential nozzle inserted above. A fillet is cut at the bottom to direct air out of the cup. Compressed air is blown through the nozzle into the cup, and then spins along the circular wall to create a negative pressure in the central area by centrifugal force. This negative pressure will be applied as a lifting force to a work piece placed under the cup, which will then pick it up hold it at an equilibrium position where the weight is balanced by the lifting force. Because air is supplied continuously, the work piece will keep levitating with a gap of hundred micrometers from the cup, through which air can be discharged into the atmosphere. For this reason, the work piece never contacts the cup. Fig.1(b) is a noncontact handling system in which a manipulator is equipped with a set of vortex cups to achieve better stability and a larger lifting force. In this paper, only one typical-sized vortex cup is used for experiments and analysis. A sketch illustrating its design and a list of its dimensions are given in Fig.2 and Table 1, respectively.

Pressure Distribution

Fig.3 plots the radial pressure distribution at different gap thicknesses for the case in which the supply flow rate is set to be constant. It is observed from this figure that the pressure inside the cup is distributed



a) Sketch of vortex cup



b) Non-contact handling system

Fig. 1. Vortex levitation

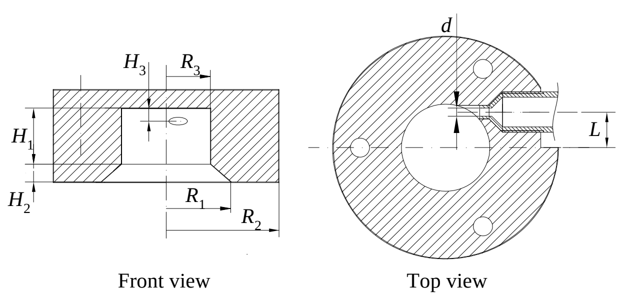


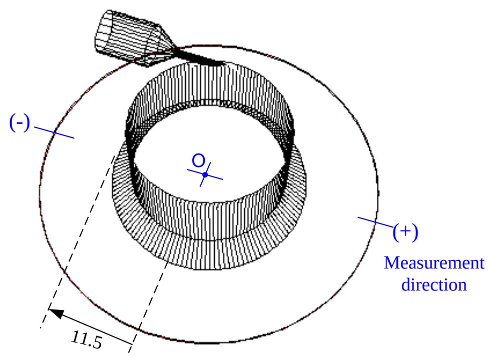
Fig. 2. Size of vortex cup

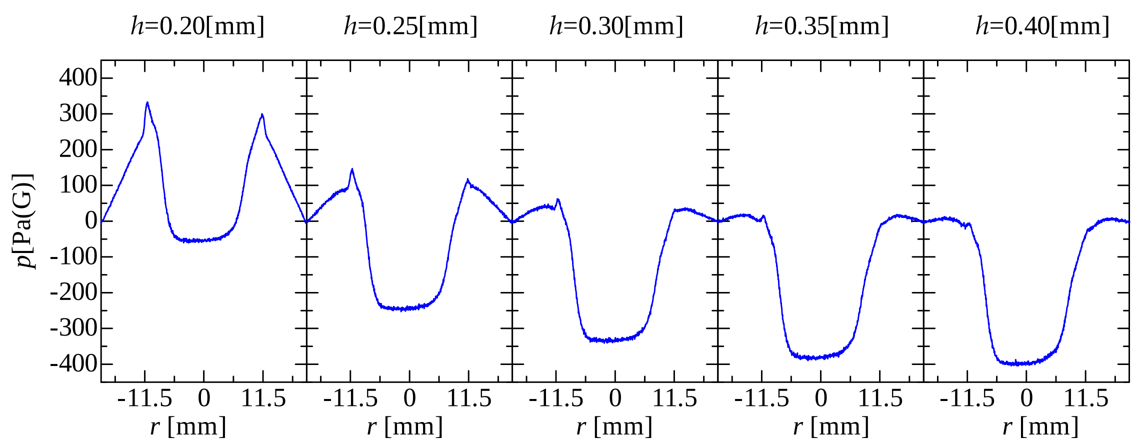
Table 1. Size (unit: mm)

H_1	8.5
H_2	2.0
H_3	2.0
R_1	11.5
R_2	20.0
R_3	9.0
d	1.5
L	7.8

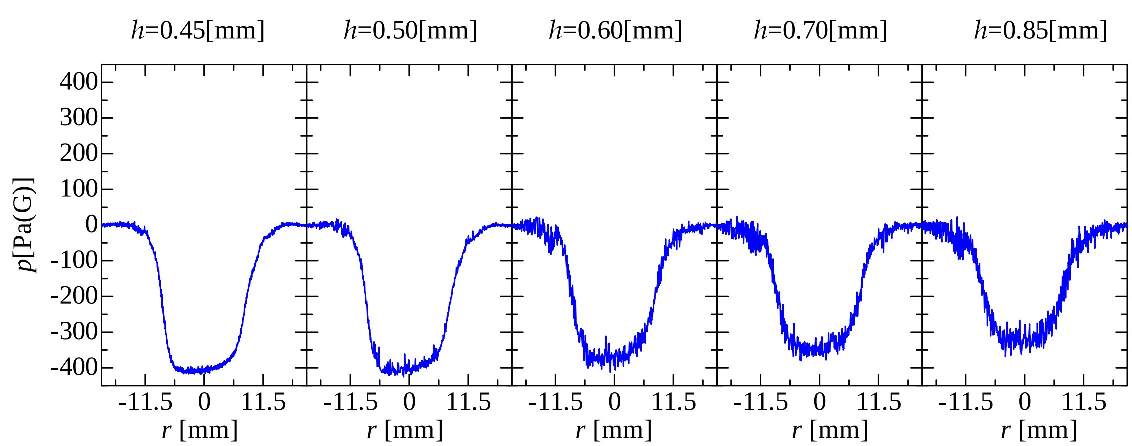
Tangential nozzle

along the radial direction and pressure at the center is lower than at the periphery. When air flows into the thin gap between the work piece and the skirt of the cup, a pressure distribution over atmosphere is formed along the gap due to viscous effect. Another important observation is that the pressure distribution varies according to the gap thickness. As the gap is enlarged, the pressure distribution shifts toward the vacuum with a uniform distribution. However, once the gap thickness becomes big enough, the negative pressure inside the cup slowly recovers toward atmospheric pressure.





a) Pressure distribution (*h*=0.20 ~ 0.40[mm])



b) Pressure distribution (*h*=0.45 ~ 0.85[mm])

Fig. 3. Pressure Distribution (*Q*=15.7[$10^{-5}m^3/s$])

Lifting Force

As is stated above, the negative pressure inside the cup varies depending on the gap thickness. Consequently, Lifting force caused by the negative pressure is dependent on the gap thickness. Fig.4 is the experimental result in case of Q=27.1[$10^{-5}m^3/s$]. It is shown that the lifting force increases as the gap between the cup and the work piece is enlarged. However, the lifting force will decrease slowly after it reaches a maximum as the gap becomes bigger and bigger. As an example, assume a 30 [g] work piece is handled by the cup, which is plotted by a broken line in Fig.4. This line intersects the lifting force line at two points A and B where the weight is balanced by the lifting force. If the work piece gets closer to the cup than A, it will obviously fall back to A because the lifting force is smaller than its weight. If it comes into the region between A and B, the lifting force becomes bigger than its weight to be able to pull it back to A. However, the work piece will fall down once it gets further away than B due to the insufficient lifting force. Therefore, A is defined as a stable levitation position and B is defined as the levitation boundary position. The region from the bottom of the cup to B is called the stable levitation region.

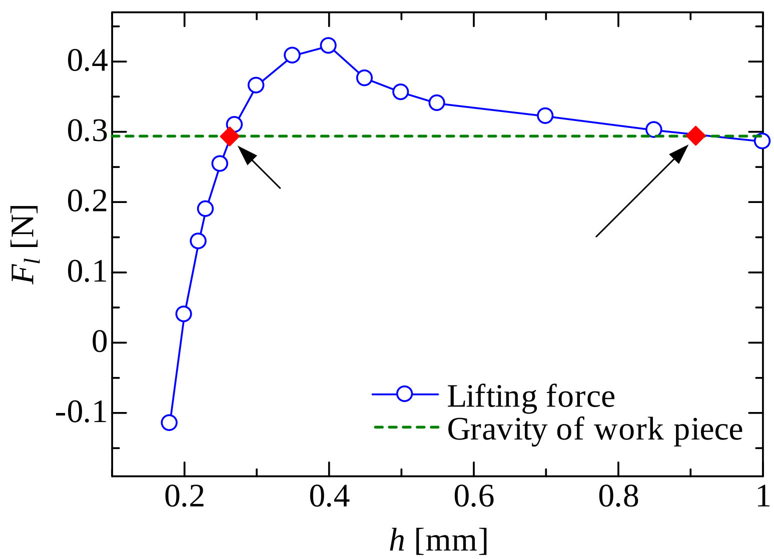


Fig. 4. Pressure Distribution (Q=27.1[$10^{-5}m^3/s$])

DYNAMICAL CHARACTERISTICS

Considering some practical uses, for example, in a semiconductor production process where each wafer is repeated loading and unloading during transportation and in the case where the work piece is displaced from equilibrium, efforts are obviously necessary to make clear how the lifting force responses and thereby influences work piece behavior while the relative distance changing with time. In this section, we proposed a spring-damper model by considering and modeling the lifting force caused by swirling air flow and the damping force related to squeeze film air damping separately. A sketch of the model is displayed in Fig.5. F_l and F_d denote lifting force and damping force, respectively.

A

Lifting Force and Spring Model

Suppose that lifting force caused by swirling flow has an enough rapid response relatively to the movement of the work piece. Therefore, a nonlinear spring model is used to indicate that lifting force is only dependent on the gap thickness. Fig.4 is a typical $F_l - h$ curve that can be obtained experimentally.

Stable

Levitatio

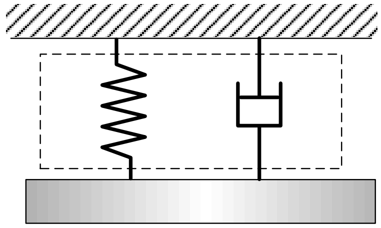


Fig. 5. Spring-damper model

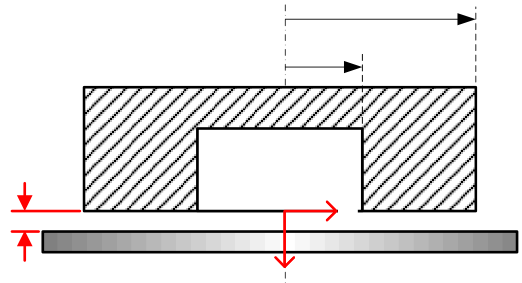


Fig. 6. Geometrical model of squeeze film

Damping Force and Damper Model

In pressure dynamic response experiment, squeeze film air damping caused primarily by the annular skirt gives rise to a pressure hysteresis while the work piece repeating sinusoidal movement relatively to the cup. Thereby, an extra damping force is caused to act upon the work piece. As shown in Fig.6, we model the squeeze film air damping geometrically with a volume at the center and an annular gap on the surrounding. Assume the cup is fixed, and set a cylinder coordinate (r, θ, z) at its bottom. Ambient pressure denoted by p_a is equal to atmosphere pressure. Theoretical analysis is given as follows.

First, consider pressure distribution and variation through the thin gap ($R_1 < r < R_2$) for the case where it is subject to varying gap thickness $h(t)$. The governing equations are derived from Navier-Stokes equations by assuming:

I. Air flow in the gap is laminar primarily because of low Reynolds numbers of this case.
II. Viscous effect is dominant in the very thin gap so that the inertia effect is often negligible for the very small geometries.
III. Distributions at r and θ directions of velocities u_r and u_z are small enough to be negligible in comparison with the intense distributions at z direction.
IV. Pressure distributions at z and θ directions are too small to be regarded.
V. Air state change is isothermal.

According to the assumption I, II and III, the motion equation of r direction is given by

$$\frac{\partial p}{\partial r} = \mu \frac{\partial^2 u_r}{\partial z^2} \tag{1}$$

From the assumption I, II, III and IV, the motion equation of z direction can be written as

$$\frac{\partial^2 u_z}{\partial z^2} = 0 \tag{2}$$

Boundary conditions are known as u_r=0 [m/s] at z=0[m], u_r=0 [m/s] at z=h[m], u_z=0 [m/s] at z=0[m] and u_z=dh/dt at z=h[m]. Thus, we can integrate Equ. (1) and (2) with the boundary conditions to obtain velocity distributions as

$$u_r = \frac{z^2}{2\mu}\frac{\partial p}{\partial r} - \frac{hz}{2\mu}\frac{\partial p}{\partial r} \tag{3}$$

$$u_z = \frac{z}{h}\frac{dh}{dt} \tag{4}$$

Considering the assumption V, conservation equation becomes

$$\frac{1}{r}\frac{\partial (pru_r)}{\partial r} + \frac{\partial (pu_z)}{\partial z} + \frac{\partial p}{\partial t} = 0 \tag{5}$$

The following equation is derived by substituting Equ. (3) and (4) into Equ. (5)

$$\frac{\partial p}{\partial t} = \frac{h^2}{12\mu}\left[p\frac{\partial^2 p}{\partial r^2} + \frac{p}{r}\frac{\partial p}{\partial r} + \left(\frac{\partial p}{\partial r}\right)^2 \right] - \frac{p}{h}\frac{dh}{dt} \tag{6}$$

This equation is well-known Reynolds Equation. Experimental results of last section shows that pressure and its change is very close to atmosphere pressure, that is, $p/p_a \approx 1$. Consequently, Equ. (6) is simplified as

$$\frac{\partial p}{\partial t} = \frac{h^2}{12\mu}\left[p_a\frac{\partial^2 p}{\partial r^2} + \frac{p_a}{r}\frac{\partial p}{\partial r} + \left(\frac{\partial p}{\partial r}\right)^2 \right] - \frac{p_a}{h}\frac{dh}{dt} \tag{7}$$

Pressure distribution through the gap and variation versus time can be determined by using finite difference method and central difference to solve Equ. (7) with boundary conditions of $p=p_0$ at $r=R_1$ and $p=p_a$ at $r=R_2$. Here, p_0 is dependent on air state change in the central volume and its governing equation is

$$p_0(V_0 + h\pi R_1^2) = mR\theta \tag{8}$$

The differential equation of Equ. (8) is given by

$$\frac{dp_0}{dt} = \frac{R\theta\dfrac{dm}{dt} - p_0\pi R_1^2\dfrac{dh}{dt}}{V_0 + h\pi R_1^2} \tag{9}$$

where dm/dt is mass flow passing through the gap inlet ($r=R_1$). Since velocity distribution of u_r had been known in Equ. (3), the mass flow at the gap inlet ($r=R_1$) is expressed as

$$\frac{dm}{dt} = -2\pi R_1\rho_a\int_0^h u_r\,dz = \frac{\pi R_1\rho_a h^3}{6\mu}\left.\frac{\partial p}{\partial r}\right|_{r=R_1} \tag{10}$$

Thus, we can rewrite Equ. (9) and simplify it with $p \approx p_a$ as follows.

$$\frac{dp_0}{dt} = \frac{R\theta\dfrac{\pi R_1\rho_a h^3}{6\mu}\left.\dfrac{\partial p}{\partial r}\right|_{r=R_1} - p_a\pi R_1^2\dfrac{dh}{dt}}{V_0 + h\pi R_1^2} \tag{11}$$

According to the derivations above, Equ. (7) and (11) suggest that pressure and thereby resistive force due to squeeze film air damping are dependent on not only the work piece's moving velocity dh/dt but also its position h. Given that the gap between the skirt and the work piece is split into n equal parts at r direction, the damping force F_d is given by

$$F_d = p_0\pi R_1^2 + \sum_{i=1}^{n} 2\pi\Delta r r_i p_i \tag{12}$$

where Δr is grip interval and equal to (R_2 -R_1)/n.

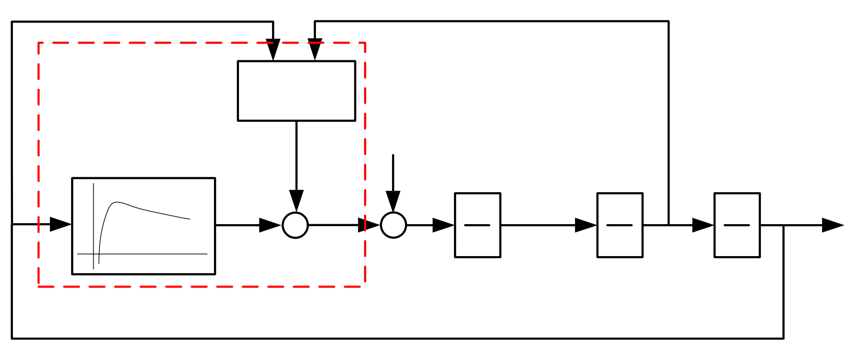


Fig. 7. Block diagram of Motion Equation

Next, using a force balance on the work piece at z direction, the equation of motion for $h(t)$ becomes

$$M\frac{d^2h}{dt^2} = Mg - F_l - F_d \tag{13}$$

Fig. 7 displays a block diagram of Equ. (13).

Loading Process Experiment

In many practical cases, for example, in a semiconductor manufacture, the handling device equipped with vortex cups usually approaches a work piece very closely (h=0.5~1.0[mm]), and then is blown compressed air to apply a lifting force to the work piece and pick it up. The resulting behavior of the work piece is a typical transient response that concerns the dynamic characteristics of the cup. Hence, next, a loading process experiment is conducted to verify our theoretical analysis.

In our present analysis, an interest is only taken in the instantaneous vertical movement of the work piece when the cup picks it up. However, lifting force is always accompanied by a deviating torque resulting from the deviation of lifting force and gravity. This certainly has a considerable affect on vertical movement and makes the measurement difficult. In order to make a correct measurement on the vertical movement, a work piece with a special design is used. As displayed in Fig.8, a rigid and light stick is installed to a flat disk. Most of the mass is attached to the lower end of the stick. Such a design can result in a very big mass moment of inertia about the upper center. Consequently, the work piece's roll caused by the deviating torque will get very slow to be negligible during the measurement period that is less than 0.2[s]. By this means, we can measure vertical movement correctly with a laser displacement meter in a relatively short period. In addition, arrays of thru holes are drilled on the table to disregard squeeze film effect that is caused between the table and the work piece.

The experimental procedure is as follows: First, adjust the supply flow rate to a given value by using the pressure regulator. Next, place a work piece under the cup and set an initial gap thickness between them. And then blow air into the cup, at the same time, record the vertical movement of the work piece with the laser displacement meter.

First, experimental condition is set as Q=25.2[10^{-5} m^3/s (ANR)] and M=20[g]. The resulting transient movement is shown in Fig.9 (a). Here, time zero is set when the work piece starts to move. The work piece approaches to the cup and oscillates due to its inertia. Thereafter, the oscillation is damped to zero in several periods. The work piece is like to be attached to a spring, at the same time, subject to a

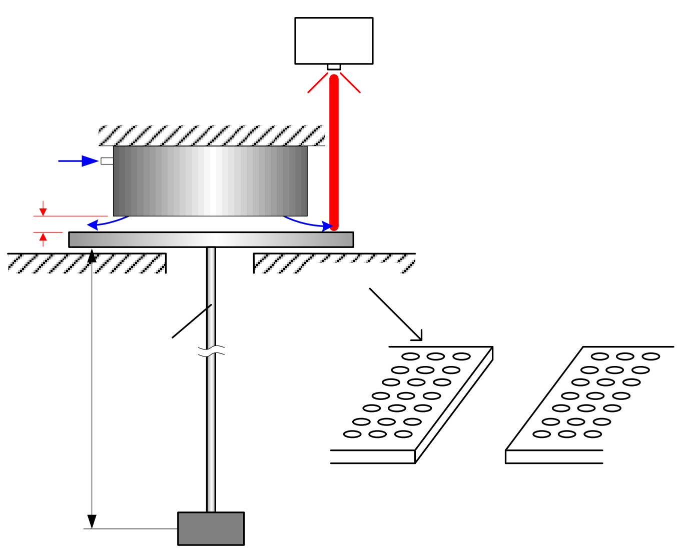
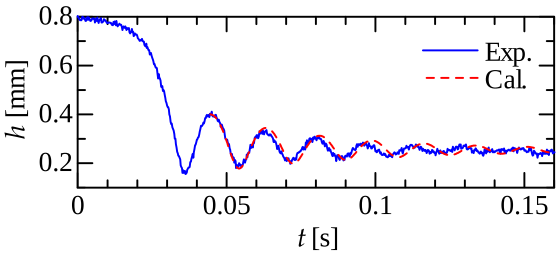
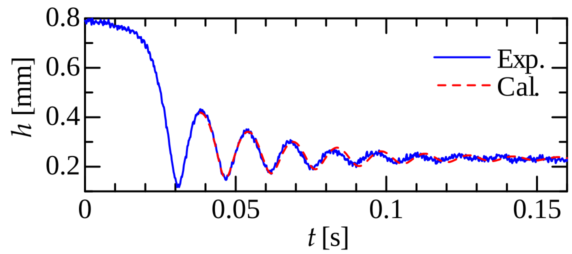


Fig. 8. Experimental Apparatus of Loading Process

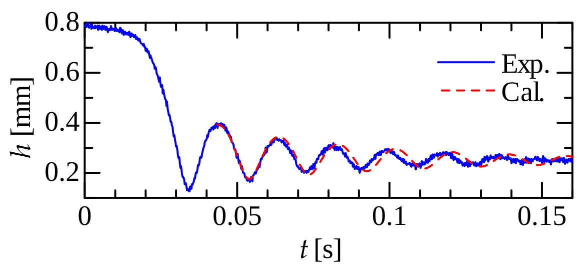
damping force. In Fig.9 (b) and (c), we change supply flow rate and mass to confirm that the oscillatory frequency is dependent on the supply flow rate and the mass. However, despite those various experimental settings, oscillation can be damped to zero in a short time. From a viewpoint of energy conversation, a more straightforward explanation is that kinetic energy of the work piece is dissipated due to air viscosity. Fig.9 also give a detailed comparison between experimental results and analytical results calculated with proposed dynamic model of Fig.6 in Matlab®. Here, analytical calculation is only conducted after the first period because some complex factors like developing time of swirling flow is concerned from beginning (t=0[s]) to the first period. Note that analytical results show a good agreement with experimental results. Our proposed dynamic model is proved to be valid.



(a) Q=25.2[×10^{-5} m^3/s (ANR)], M=20[g]



(b) Q=27.1[×10^{-5} m^3/s (ANR)], M=20[g]



(c) Q=27.1[×10^{-5} m^3/s (ANR)], M=26[g]

Fig. 9. Experimental Result of Loading Process

CONCLUSION

Vortex levitation can achieve non-contact handling by blowing air into a vortex cup through a tangential nozzle to generate a swirling air flow. In this paper, its steady-state and dynamical characteristics are investigated experimentally and theoretically. In steady-state analysis, it is revealed that the pressure distribution varies as the gap thickness between the vortex cup and the work piece changes, and thereby the lifting force is dependent on the gap thickness. Thereafter, in dynamical investigation, we proposed a spring–damper model by modelling the lifting force caused by swirling air flow and the resistive force related to squeeze film air damping separately. The proposed dynamical model is verified to be valid by means of an experiment of loading process.

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APPENDIX I. NOTATION

The following symbols are used .

Symbol	Quantity	Unit
F_l	Lifting force	[N]
F_d	Resistive force	[N]
G	Acceleration of gravity	[m/s^2]
h	Gap thickness	[m]
m	Mass of air	[kg]
M	Mass of work piece	[kg]
p	Pressure	[Pa]
p_a	Atmosphere pressure	[Pa]
Q	Supply flow rate	[m^3/s(ANR)]
R	Gas constant	[J/(kg·K)]
t	Time	[s]
u	Velocity	[m/s]
V_0	Central volume	[m^3]
r, □, z	Cylinder coordinates	
□	Temperature	[K]
□	Viscosity	[Pa·s]
□$_a$	Atmosphere density	[kg/m^3]