Effect of Intake Parameters on Lifting Force of Swirl Gripper

Mitsuhiro NAKAO**, Yuta YAMANOUCHI***, Minoru FUKUHARA**

A swirl gripper, which uses a vane rotated by an electric motor to form a swirling flow, is known to be more energy-efficient than the conventional pneumatic vortex gripper. However, the effect of its design parameters on the lifting force has not been fully investigated. In this study, the design parameters of the intake of the swirl gripper such as the ratio of the inlet radius to the swirl chamber radius and the ratio of the distance from the center of the cup to the center of the intake, are investigated to clarify how they affect the lifting force. The working fluid was air, and the design parameters were varied to experimentally investigate the lifting force and pressure distribution for two different vane speeds. The experimental results suggested that the inlet flow rate increased as the diameter of the intake increased and its position became closer to the center of the swirl chamber, and the gap thickness for the same lifting force increased accordingly. In addition, it was found that there was an optimum intake diameter to obtain the maximum lifting force for the intake position.

Keywords: Swirl gripper, Non-contact handling, Swirling flow, Pressure distribution, Vane

1. Introduction

A work piece is usually in contact with the handling device during lifting and moving operations. Particularly in the semiconductor manufacturing process, such contact can scratch the surface of the work piece and generate static electricity. Contact is also undesirable in some food and pharmaceutical manufacturing processes, as it can cause damage and contamination. Because of these disadvantages of contact handling, much research has been done on non-contact handling[1]. A common method in non-contact conveying devices is to use air as a medium to generate a lifting force on a work piece. Bernoulli gripper [2-4] based on Bernoulli’s theorem is a typical method of non-contact conveyance using air. Another method that uses air is a vortex gripper[5], in which compressed air forms a swirling flow in a cylindrical chamber to generate lifting force. Due to the nature of using compressed air, large energy losses are inevitable in these methods. Li et al. proposed a swirl gripper[6] using an electric motor to form the swirling flow to improve energy efficiency, and investigated its basic characteristics. However, the effect of its shape on the characteristics such as lifting force has not been sufficiently investigated.

The objective of this study is to clarify how the design parameters of the air intakes of swirl gripper affect the lifting force and pressure distribution. The working fluid was air, and the design parameters of the air intakes at the top of the swirl gripper was varied to experimentally investigate the effects on the lifting force and pressure distribution. The ratio of the intake radius to a swirl chamber radius and the ratio of the distance from the chamber center to the air intake to the radius of the chamber were set as design parameters. Then, the pressure distribution above a work piece at the gap thickness where the maximum lifting force was obtained was measured for each design parameter to investigate the effect on the pressure distribution.

2. Nomenclature

\[
\begin{align*}
b & : \text{thicknees of air intake} \\
H & : \text{height of circular cylinder} \\
h & : \text{heigh of vane} \\
h_1 & : \text{height of curved part of vane} \\
h_2 & : \text{height of vertical part of vane} \\
l & : \text{horizontal position of air intake} \\
n & : \text{number of vanes} \\
r & : \text{radius of air intake} \\
R & : \text{radius of circular cylinder} \\
R' & : \text{radius of vane} \\
\theta & : \text{angle of curvature}
\end{align*}
\]

3. Mechanism of the Swirl Gripper

A schematic of the swirl gripper is shown in Fig. 1. Mainly it consists of a set of swirl vanes, motor, set of air intakes and
circular swirl chamber. The axis of the swirl vane is tangent to the inner center of the cup, and the eight swirl vanes are arranged vertically and symmetrically on the axis. When the blades are rotated along the curved direction of the tip, the air in the swirl chamber swirls. The centrifugal force generated by this rotation causes the air to move from the center of the swirl chamber to the wall of the swirl chamber, creating a negative pressure in the central region. When a work piece is placed at the bottom of the cup, this negative pressure acts as a lifting force to lift the work piece. And by balancing the lifting force with the gravity applied to the work piece, this work piece can be held and transported without contact. At the same time, the curved shape of the swirl vanes gives a downward velocity component to the swirling flow, and air flows in from outside the cup through four air intakes installed at 90° intervals at the top of the cup, and is discharged through a small gap between the bottom of the cup and the work piece. The discharged air acts as a pushing force against the work piece, thus realizing non-contact conveyance.

4. Experiment

4.1 Trial Swirl Gripper Used in this Study

Fig. 2 shows the design dimensions of the swirl gripper used for discussion in this paper. As shown in Fig.2, swirl gripper used in this study can be divided into a swirl chamber consisting of a top plate with four air intakes and a cylinder, and a set of swirl vanes. $l$ is the distance of the inlet from the center of the cup, $r$ is the radius of the inlet, $b$ is the thickness of the intake, $R$ is the radius of the swirl chamber, $H$ is the height of the swirl chamber, $R'$ is the radius of the swirl vane, $h$ is the height of the swirl vane, $h_1$ is the height of the curved section, $h_2$ is the height of the vertical section, $\theta$ is the angle of curvature, and $n$ is the number of swirl vanes. In this study, the dimensions of the swirl chamber and swirl blades were set as $R'/R = 0.96$, $h/H = 0.86$, $h_1/h_2 = 0.33$, $\theta = 10^\circ$, and $n = 8$, as shown in Table 1. These dimensions were determined by trial and error. The actual dimensions are $R = 25$ mm, $H = 14$ mm, $h_2 = 9$ mm. The swirl blades were molded from the dedicated model material AR-M2 using a 3D printer (KEYENCE AGILISTA-3100). The dimensions of all the air intakes used in this study are summarized in Table 2. A total of four air intakes were installed at 90° intervals on the upper part of the cup. For the air intakes $\text{①}$ through $\text{⑫}$, the ratio of the air intake radius to the radius of the swirl chamber, $r/R$, was varied in four ways: 0.06, 0.09, 0.12, and 0.18. And the ratio of the distance between the air inlet from the center of the cup to the radius of the swirl chamber, $l/R$, was varied in three ways (Table 2).

| $R'/R$ | $h/H$ | $h_1/h_2$ | $\theta$ | $n$ |
|-------|-------|-----------|--------|-----|
| 0.96  | 0.86  | 0.33      | 10°    | 8   |

| $r/R$ | $l/R$ | $b/H$ |
|-------|-------|-------|
| ① 0.06 | 0.3   | 0.36  |
| ② 0.09 | 0.3   | 0.36  |
| ③ 0.12 | 0.3   | 0.36  |
| ④ 0.18 | 0.3   | 0.36  |
| ⑤ 0.06 | 0.5   | 0.36  |
| ⑥ 0.09 | 0.5   | 0.36  |
| ⑦ 0.12 | 0.5   | 0.36  |
| ⑧ 0.18 | 0.5   | 0.36  |
| ⑨ 0.06 | 0.7   | 0.36  |
| ⑩ 0.09 | 0.7   | 0.36  |
| ⑪ 0.12 | 0.7   | 0.36  |
| ⑫ 0.18 | 0.7   | 0.36  |
ways: 0.3, 0.5 and 0.7. The ratio of the thickness of the air intake to the height of the swirl chamber, \( b/H \), was set as 0.36.

### 4.2 Experimental Methods

Fig. 3 shows the configuration of the experimental apparatus used to measure the lifting force of swirl gripper. The experimental apparatus consists of a Z-stage, support for the Z-stage, transmission mechanism which conveys the rotation of the servomotor to the axis of the vane, swirl gripper, plate (work piece) that assumes the object to be grasped, and electronic balance. The work piece was a rectangular piece of aluminum measuring 110 mm x 110 mm x 16 mm (length x width x height) with a mass of \( 5.0 \times 10^2 \) g.

The motor (Mitsubishi Electric, HG-KR13) is driven by a servo amplifier (Mitsubishi Electric, MR-J4-10A), which has a closed loop to control the rotation speed. An encoder built into the motor detects the rotational speed and feeds back to the amplifier, which sets the input voltage to maintain a constant rotational speed. Therefore, the rotation speed of the motor is used instead of the drive voltage or current as the experimental condition for measuring the lifting force. The rotation of the servomotor fixed to the support is transmitted via a pulley belt to the shaft supported by two bearings, which rotates the vanes inside the cup attached to the end of the shaft.

The lifting force was measured by operating the motor at two different rotational speeds of 2800 rpm and 3600 rpm, with the speed shifted to 2.5 times the rotational speed through a pulley and transmitted to the swivel vane shaft. The rotational speed of the swivel vane shaft was 7000 rpm and 9000 rpm. The gap thickness between swirl gripper fixed to the Z-stage and the work piece could be adjusted by manually raising and lowering the Z-stage attached to the support. The measured gap thickness ranged from 0.20 mm to 3.0 mm, and the measurement interval was 0.1 mm. The work piece is placed on the electronic balance, and the pull force is measured according to the value displayed on the electronic balance.

Fig.4 shows the configuration of the experimental apparatus for measuring the pressure distribution on the work piece when the swirl gripper is in operation, and Fig.5 shows the measurement points. The configuration of the experimental setup differs from that of Fig. 3 in that instead of a plate on the electronic balance, a plate of \( 3.0 \times 10^2 \) g mass with a 0.5 mm diameter hole for pressure measurement is fixed on the X-axis stage. A pressure sensor (ALL SENSORS, 5INCH-D-4V) was connected to the pressure hole to detect the pressure signal. By manually sliding the X-stage, the position of the hydrostatic hole on the work piece could be adjusted, and the measurement point was varied from 0 mm to 70 mm. Pressure measurements were taken end-to-end through the center of the cup, with a measurement interval of 2.0 mm.

### 5. Result and Discussion

We measured the lifting force when the gap thickness was changed in the swirl gripper using the cups ① through ⑫. The results at 7000 rpm and 9000 rpm are shown in Fig. 6 and Fig. 7, respectively. Regardless of the rotation speed and intake shape, the lifting force increased as the gap thickness increased until it reached the maximum lifting force. However, the detailed trend was different, and for \( r/R = 0.06 \), after
reaching the maximum lifting force, the lifting force decreased slightly but remained almost unchanged. In addition, the lifting force was hardly affected by the intake position. On the other hand, in the case of \( r/R = 0.09 \) and 0.12, it was clearly observed that the lifting force decreased as the gap thickness further increased after the maximum lifting force was obtained. The gap thickness showing the maximum lifting force was larger for \( r/R = 0.12 \). For both \( r/R \), the lifting force was affected by the intake position when the gap thickness was small. In the case of \( r/R = 0.18 \), the lifting force increased slowly with the increase of the gap thickness, and the lifting force seemed to be maximum at the gap thickness around 3 mm, but the decrease of the lifting force could not be observed clearly within the experimental range. This difference in the amount of increase in lifting force is thought to be due to the fact that the larger the intake diameter, the larger the flow rate from the outside air into the swirl chamber. As the flow rate increases, the viscous friction in the gap flow increases because the velocity of the flow out through the gap increases. As a result, the positive pressure generated in the gap increases, and the force pushing the workpiece back is considered to increase.

From the viewpoint of stable non-contact conveyance, it is desirable that a repulsive force is generated when the workpiece gets too close to the cup. In other words, it is desirable that the lifting force near the gap thickness 0 be negative. Comparing from such a viewpoint, the repulsion force is larger when the intake position is closer to the center within the scope of the present study, and \( r/R = 0.3 \), which is the closest to the center, is recommended.

The lifting force characteristics are evaluated from the viewpoint of the maximum lifting force. Fig. 8 shows the results of comparing the maximum lifting force of the cups ① through ⑫ at each rotation speed. The trend is different for each intake diameter. For \( r/R = 0.06 \), the maximum lifting force decreases as \( l/R \) increases, while for \( r/R = 0.09 \), the distribution is convex upward with the maximum value at \( l/R = 0.5 \), and for \( r/R = 0.12 \) and \( r/R = 0.18 \), it monotonically
increases as \( l/R \) increases. Fig. 8 also shows that the maximum lifting force is obtained when the intake diameter \( r/R = 0.09 \) for intake positions \( l/R = 0.3 \) and 0.5, and \( r/R = 0.12 \) for \( l/R = 0.7 \). This result indicates that there is an optimum intake diameter to obtain the maximum lifting force for each intake position. This is thought to be the fact that a change in the intake position causes the incoming air in the path to the changed, which is swirled by the vane and released through the gap between the cup and the work piece to atmospheric pressure, thus changing the pressure in the swirl chamber.

Fig. 7 Lifting force against gap thickness with a rotation speed of 9000 rpm

Fig. 8 Comparison of maximum lifting force against the intake position
The pressure distribution on the work piece at the gap thickness where the maximum lifting force was recorded was measured in each cup. The results at 7000 rpm and 9000 rpm are shown in Fig.9 and Fig.10, respectively. In all cups, the pressure distribution in the cup was generally parabolic, with the maximum negative pressure near the center of the cup. The air exiting the chamber is released through the skirt, the gap between the cup and the work piece, to atmospheric pressure, and the pressure increases slowly in the process. The larger \( r/R \) is, the more the pressure drops immediately after leaving the chamber, and this tendency is more pronounced when \( l/R \) is smaller. In general, when a fluid flows from a larger space into a narrow channel, a large inflow velocity causes a contraction and a sudden pressure drop. The larger the \( r/R \), that is, the larger the inflow velocity into the gap, the larger the pressure drops after the inflow, which is thought to be due to the contraction occurring in this gap. Therefore, the smaller \( l/R \) is, the larger the inflow flow is. This is supported by the fact that the negative pressure near the center is small when \( l/R \) is small. Since the negative pressure generated by the vane is greatest near the center, the closer the intake is to the center, the greater the difference from atmospheric pressure, and the greater the flow rate. However, as \( r/R \) increases, the vertical downward momentum of the air passing through the intake increases and acts as an impinging jet against the top surface of the workpiece. This impinging jet inhibits the formation of negative pressure. This is supported by the fact that at \( r/R = 0.18 \) and \( l/R = 0.3 \), the negative pressure at \( x = 27.5 \) mm and 42.5 mm, corresponding to the intake position, is particularly reduced. On the other hand, when \( r/R = 0.06 \), the pressure distribution is almost independent of the intake position. This can be attributed to the fact that the effect of the impinging jet was sufficiently small due to the small intake diameter and the small inflow air flow rate.

Reviewing the results in Fig. 8, we can see a negative correlation between \( l/R \) and maximum lifting force at \( r/R = 0.06 \), and the positive correlation becomes stronger as \( r/R \) increases. This result can be explained by the above discussion. For \( r/R = 0.06 \), the effect of the impinging jet hardly
appeared because of the small inflow air volume, and the smaller the $l/R$ where the inflow flow rate was higher, the larger the maximum lifting force. As $r/R$ increases, the impinging jet gradually affects the formation of negative pressure on the workpiece, and an increase in the inflow of air does not necessarily lead to an increase in the lifting force. Therefore, the maximum lifting force will be higher when the $l/R$ was large and the inflow air volume was suppressed to some extent to reduce the effect of the impinging jet.

6. Conclusions
In this study, we investigated the effects of varying the ratio of the intake radius to the swirl chamber radius and the ratio between the distance from the center of the cup and the center of the intake to the radius of the swirl chamber on the lifting force and pressure distribution in a swirl gripper with air as the working fluid. The conclusions of this paper are as follows:

(1) As the intake diameter increases and the intake position moves closer to the center of the cup, the gap thickness increases for the same lifting force. This corresponds to an increase in the inlet flow rate.

(2) There is an optimum intake diameter to obtain the maximum lifting force for the position of the intake port. In the scope of the present study, the maximum lifting force was obtained at $r/R = 0.09$ when $l/R = 0.3$ and $0.5$, and at $r/R = 0.12$ when $l/R = 0.7$.

References
1) Brandt, E. H.: Levitation in Physics, Science, Vol.243, No.4889, p.349-355 (1989)
2) Benjamin, J. M., Brick Town, N., J.: Pneumatic Probe for Handling Flat Objects, U. S. Patent, 3,425,736 (1969)
3) Paivanas, J. A., Hassan, J. K.: Air Film System for Handling Semiconductor Wafers, IBM Journal of Research and Development, Vol.23, No.4, p.361-375 (1979)
4) Brun, X. F., Melkote, S. N.: Modeling and Prediction of the Flow, Pressure, and Holding Force Generated by a
5) Li, X., Kawashima, K., Kagawa, T.: Analysis of Vortex Levitation, Experimental Thermal and Fluid Science, Vol.32, No.8, p.1448-1454 (2008)

6) Li, X., Kagawa, T.: Development of a New Noncontact Gripper Using Swirl Vanes, Robotics and Computer-integrated Manufacturing, Vol.29, No.1, p.63-70 (2013)