PIV Measurement of Internal Flow in Mini Centrifugal Pump

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Abstract

The diameter of the ventricular assist pump and cooling pump for electrical devices like servers is about a dozen millimetres and these kinds of pumps belong to the mini pump. The internal flow condition of the mini centrifugal pump has to be clarified because there are so many types of mini pumps according to special specifications and shapes. Therefore, in this research, an open impeller was adopted as a mini centrifugal pump with 55mm impeller diameter in consideration of reduction in impeller manufacturing cost and its internal flow was investigated. Since the outlet angle of the test impeller is large, the maximum efficiency flow rate moves to a large flow rate and the relatively high head and efficiency were obtained in large flow rate region. In the present paper, the velocity and vorticity distribution near the volute tongue are shown and the internal flow in the mini centrifugal pump is clarified based on the PIV measurement results.

Keywords: Centrifugal pump, PIV, Internal flow, Performance

1. Introduction

Pumps are widely used in the industrial field and are expected to be applied to the fuel cells and medical devices in recent years [1,2]. Especially, the research related to the ventricular assist pump has been conducted actively all over the world. Some of the researchers represent significant breakthrough for the trouble like the blood clot, which occurs when the mini centrifugal pump is used for the ventricular assist pump, by the adoption of the magnetically suspended pump. Then, the ventricular assist devices are in practical use in USA, Europe, Japan and et al. using the magnetically suspended pump. In addition to that, experimental investigations of the internal flow condition of the ventricular assist pump have been conducted with the numerical analysis [3]. The cavitation flow of a shaft-less double suction miniature pump having 24mm impeller diameter was investigated using CFD [4]. Horie et al. clarified the effect of leakage flow through the gap and the volute shape for the magnetically suspended blood pump with 50mm diameter by CFD [5,6]. Two-stage centrifugal blood pump with the impeller diameter of 40mm for cardiopulmonary support system was developed by Horiguchi et al. and its performance was examined by experiment and CFD [7,8]. On the other hand, Wu et al. conducted the experiment related to the internal flow of the mini centrifugal pump with 100mm diameter using PIV [9]. Keller et al. investigated the unsteady flow condition inside the volute of the centrifugal pump by PIV measurement and clarified the vortex structure near the volute tongue with the rotation of the impeller [10]. The blade-to-blade flow condition was measured using PIV by Zhang et al. and showed the internal flow of the low specific speed centrifugal pump [11]. Diameter of the ventricular assist pump and cooling pump for servers is less than 100mm and these kinds of pumps belong to the mini pump. Therefore, there are many cases that the conventional design standard and theory can’t be applied due to the small size of the pump and the design method is not established [12]. It was clarified by Nishi et al. that the performance of the mini centrifugal pump based on the conventional design method using the closed impeller was better than that using the open impeller. However, it is considered that the open impeller is suitable for the mini centrifugal pump on the point of view of the maintenance of the pump and the manufacture of the impeller. On the other hand, it was also verified by Nishi et al. that adequate performance could be obtained even in the open impeller designed by the original design method, in which the blade outlet angle and blade number were increased [13-15]. The two dimensional impeller is often used for the mini centrifugal pump because the mini centrifugal pump can’t fully demonstrate the advantages concerning the performance of the three dimensional impeller. Furthermore, the mini centrifugal pump has a tendency getting lower specific speed with the high head and low flow rate. It was clarified that the sufficient performance could be obtained with the two dimensional impeller under the condition of the low specific speed [16]. Therefore, the two dimensional open impeller was adopted as a mini centrifugal pump having 55mm
impeller diameter in consideration of reduction in impeller manufacturing cost. Then, the internal flow of this test pump was investigated by PIV measurement.

In the present paper, the internal flow near the casing tongue is clarified using the mini centrifugal pump having 55mm impeller diameter. The internal flow near the casing tongue of the PIV measurement is compared with the numerical analysis results to evaluate the validity of the PIV measurement results. The vortex structure near the casing tongue was visualized by the PIV measurement results when the trailing edge of the impeller approaching and leaving the casing tongue.

2. Experimental Apparatus

2.1 Design specification and experimental apparatus for performance test

A test rotor of the mini centrifugal pump was designed based on the conventional design method under the condition of the design head $H_0=2.0\text{m}$, flow rate $Q_d=16.7\text{l/min}$ and rotational speed $N_d=2230\text{min}^{-1}$ on the assumption that the mini centrifugal pump can be used for the cooling of electrical devices. Therefore, the specific speed is $N_s=171\text{min}^{-1}, \text{m}^3/\text{min}, \text{m}$. The blade inlet angle of the original model was $\beta_1=15^\circ$ and the blade outlet angle was set as $\beta_2=22.5^\circ$ which was recommended by the conventional design method. However, it was difficult to obtain adequate head for the mini centrifugal pump with the open impeller because of the low Reynolds number effect and the leakage flow from a tip clearance [12]. Therefore, the large blade outlet angle $\beta_2=60^\circ$ was set as the base model to keep the head of the pump in this research under the condition that other design parameters were the same as those of the original model. Figure 1 shows schematic diagram and picture of the base model. Table 1 is the primary dimensions of the original and base rotor. The rotor has two-dimensional impeller and is made of the aluminum. The inner diameter of the rotor $D_1=27\text{mm}$, outer diameter $D_2=55\text{mm}$ and blade width $B=4.2\text{mm}$. The suction diameter of the pump is $D_{in}=26\text{mm}$ and its discharge diameter is $D_{out}=13\text{mm}$. The sectional view of the volute casing is rectangular and the inner diameter of the volute is 62mm. Therefore, the clearance at the volute tongue is 3.5mm because the outer diameter of the rotor is $D_2=55\text{mm}$.

The schematic diagram of the experimental apparatus for the performance test is shown in Fig. 2. Water was used for the experiment. For the pressure performance evaluation, the static head difference on the wall between $2D_{in}$ upstream and $2D_{out}$ downstream of the rotor was measured by the pressure sensor Kyowa Electronic Instruments, PGM-1KG. Then, the pump total head was evaluated by adding the dynamic head difference of the sectional averaged axial velocity to the corresponding measured static head difference. The rotor was driven by the motor. The flow rate $Q$ was obtained by a magnetic flow meter KEYENCE FD-UH25G installed far downstream of the pump and the torque was measured by a torque meter ONO SOKKI SS-010. The shaft power was calculated by the torque and rotational speed measured by a rotational speed sensor ONO SOKKI MP-981. The shaft power was evaluated by the torque eliminating the mechanical loss and disc friction loss using a disc without the impeller in this performance test. Then, the hydraulic efficiency of the pump $\eta$ was calculated as the ratio of the water power to the shaft power.

2.2 PIV measurement instrument

The schematic diagram and picture of PIV measurement instrument are shown in Fig. 3 and Fig. 4 respectively. FlowMaster Stereo-PIV by LaVision was used for PIV measurement. The test section was designed as same as the experimental apparatus for the performance test, the whole test section for PIV measurement was made of the acrylic resin as shown in Fig. 5 including the test impeller in Fig. 6. The experimental apparatus for PIV measurement was a circulating water channel and the flow rate was controlled by a valve. Static pressure taps were set on the wall between $2D_{in}$ upstream and $2D_{out}$ downstream of the rotor and the static pressure was measured by a small digital pressure switch SMC ZSE80F-02L-T-MA. The pump total head was measured by the same method of the performance test. Then, the flow rate of the PIV measurement test was evaluated by the pump total head obtained by the PIV measurement and the head curve obtained by the performance test in section 2.1. The Oriental motor BLM5400HP-AS and inverter BMUD400-S were used to keep the constant design rotational speed of the pump. The double pulse YAG laser LaVision DPIV-L125, with 135mJ power was used as a light source and double pulse timing widths(repetition rate) was set as 5$\mu$s (0.2MHz) considering the range of the measurement velocity. A trigger pulse from the rotor was detected by the photo interrupter Panasonic CNZ1023. The phase locked averaged data were measured by inputting a trigger pulse from the rotor to PTU(Programmable Timing Unit). The measurement plane was axial sectional plane at the mid of the blade width $b/B=0.5$ near the volute tongue and its area was about $58\text{mm}^2\times58\text{mm}$ because the calibration plate size was $58\text{mm}^2\times58\text{mm}$. Fluorescent particles EBM FLUOSTAR with particle diameter 15$\mu$m were
Table 1 Primary dimension of rotors

<table>
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<tr>
<th>Geometry</th>
<th>Original model</th>
<th>Base model</th>
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<tr>
<td>Inlet diameter at hub $D_{1h}$ [mm]</td>
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<td>27</td>
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<tr>
<td>Inlet diameter at tip $D_{1t}$ [mm]</td>
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<tr>
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<td>55</td>
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<tr>
<td>Inlet width $b_1$ [mm]</td>
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<td>4.2</td>
</tr>
<tr>
<td>Outlet width $b_2$ [mm]</td>
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<td>4.2</td>
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<tr>
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<td>6</td>
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<tr>
<td>Blade thickness $s$ [mm]</td>
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<tr>
<td>Blade outlet angle $\beta_2$ [deg]</td>
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<td>60</td>
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<tr>
<td>Arc geometry</td>
<td>1Arc</td>
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Fig. 2 Schematic diagram of experimental apparatus for performance test

Fig. 3 Schematic diagram of PIV measurement instrument
used as tracer particles because of its overwhelming high-luminance and followability to water. FLUOSTAR is a fluorescent particle, which absorbs laser from the light source efficiently and scatters the light with wave length 580nm, different from the laser wave length 532nm of the light source. Therefore, we can only get the scattering light of fluorescent particles using high pass filter OG570 in Fig. 7 attached on the lens of the camera and suppress the error vector in the post process. The captured image using FLUOSTAR is compared to the image obtained in the previous research using particle Nylon12 in Fig. 8. The scattered light was suppressed using the FLUOSTAR with high pass filter and clear particles image was obtained as shown in Fig. 8. Two CCD camera with its resolution 2048×2048 pixels were used and velocity vector and vorticity distribution were calculated by the PIV measurement software DaVis(Lavision).

Fig. 4 PIV measurement instrument

Fig. 5 Test section for PIV measurement

Fig. 6 Test impeller for PIV measurement

Fig. 7 High pass filter

Fig. 8 Captured image
3. Numerical analysis condition

Numerical flow analysis was conducted to compare the internal flow obtained by the PIV measurement. In the numerical analysis, the commercial software ANSYS CFX16.2 was used and the numerical analysis was conducted with the three-dimensional model, which is the same with the test section of the mini centrifugal pump used in the performance test. Water was assumed to be incompressible and isothermal water and the equation of the mass flow conservation and Reynolds Averaged Navier-Stokes equations were solved by the finite volume method. The standard wall function was utilized near the wall and the standard $k-\omega$ model was used as the turbulence model. The computational grids used for the numerical analysis is shown in Fig. 9. The inlet of the numerical domain is $5D_{in}$ upstream of the test section and the outlet is $5D_{out}$ downstream of it. The constant velocity and constant pressure are given as the boundary conditions at the inlet and outlet respectively. The tip clearance is also made in the numerical analysis, which is the same with the experimental apparatus for the performance test and the number of elements within the tip clearance was 8. The numerical analysis was performed at 6 different flow rate points from $0.8-1.4Q_d$. Grid point numbers of the inlet pipe, rotor region, casing region and outlet pipe region are 240,191 points, 4,376,568 points, 1,130,000 points and 266,179 points respectively, so the total grid points are 6,012,938 points. The unsteady numerical analysis was conducted and the coupling between the rotor and casing was accomplished by the sliding mesh (Transient Rotor Stator Model). The time step number per one rotor rotation is 180 and the time step is $t = 1.495 \times 10^{-4}$ s. The data of one rotor rotation were obtained after 6 rotor rotations in unsteady numerical analysis.

4. Result and Discussion

4.1 Performance curve of the test mini centrifugal pump

Figure 10 shows the performance curves of the test mini centrifugal pump obtained by the experiment. The tip clearance is $c=0.5\text{mm}$ and the performance test was conducted in wide flow rates range from the shut off flow rate to large flow rate in the experiment. Horizontal axis is the flow rate $Q$ and first and second vertical axis of Fig. 10(a) is the head and shaft power respectively. The vertical axis of Fig. 10(b) is the efficiency $\eta$. The unsteady numerical analysis result is also shown in Fig. 10 to compare the numerical result with the experimental result. Focused on the head and efficiency curves of the experiment, we obtained the head $H=1.93\text{m}$ at the design flow rate $Q_d=16.7\text{l/min}$ and the efficiency was $\eta=56.5\%$ at the same flow rate $1.0Q_d$. The flow rate of the maximum efficiency shifts larger flow rate $1.2Q_d$ because the test rotor having the large blade outlet angle $\beta_2=60^\circ$. The maximum efficiency is not so high, however, the relatively high head and efficiency are kept in large flow rate region. On the other hand, when we compared the numerical analysis result and experimental result, the numerical analysis results of head, shaft power and efficiency represented the qualitative tendency of the experimental results and numerical analysis data of the performance accorded with the experimental data qualitatively for some data. Therefore, we concluded the unsteady RANS could capture the trend of the performance of the test mini centrifugal pump.

4.2 Comparison between PIV measurement result and CFD result

In this research, we tried to focus on the flow condition near the casing tongue. It is difficult to validate the PIV measurement result using the other measurement technique because the test pump was small with 55mm rotor diameter. The performance of the test mini centrifugal pump could be captured well in Fig. 10, so we considered that the flow condition of the numerical analysis would show the adequate data of the internal flow for the test mini centrifugal pump. Then, we compared the PIV measurement result and numerical analysis result to investigate the validity of the PIV measurement results. The rotational position of the impeller is defined by the rotational angle of the trailing edge(TE) on the suction surface of the impeller from the casing tongue in Fig. 11. The anti-crock wise direction of the impeller is a positive for the rotational angle $\theta$ and its angle is zero when the TE on the suction surface of the impeller accords with the casing tongue. In order to compare the flow condition near the casing tongue, we defined the tongue flow angle $\alpha$ represented by Fig. 12. The sampling points for the data are 3mm away from the casing tongue and set at 0.5mm interval in the horizontal line in Fig. 12. The center of the casing tongue is the origin $x=0$ mm and $x$ is a positive in right direction in Fig. 12. The flow angle is the angle of the velocity vector from the vertical line in Fig. 12 at each sampling point and it is a positive in crock wise
Direction. The tongue flow angle at the design flow rate $1.0Q_d$ is shown in Fig. 13. The rotational angle $\theta_r = -24^\circ$ in Fig. 13. The trend of the tongue flow angle distribution between the PIV measurement and numerical analysis is almost the same in the range $-1.0 \text{mm} < x < 1.0 \text{mm}$, however, there is some difference between them $x < -1.0 \text{mm}$, $1.0 \text{mm} < x$. This difference may have occurred in RANS because it is difficult to reproduce the vortex and separation in the experiment. The value of the tongue flow angle is not so different each other and the tongue flow angle is minus in the region $-2.0 \text{mm} < x < 1.5 \text{mm}$ for both case, which corresponds recirculation flow from the casing to the impeller at the design flow rate $1.0Q_d$ in the test pump. Therefore, we determined the PIV measurement results show the proper flow condition of the test mini centrifugal pump. The tongue flow angle at the large flow rate $1.4Q_d$ is shown in Fig. 14. The rotational angle $\theta_l = -24^\circ$ in Fig. 14. There is a discrepancy at the large flow rate between the PIV measurement and numerical analysis. It shows the difficulty to capture the tongue flow at large flow rate by numerical analysis based on RANS.

![Fig. 10 Performance curves of test mini centrifugal pump](image1)

![Fig. 11 Rotational position of impeller](image2)

![Fig. 12 Definition of tongue flow angle $\alpha_t$](image3)

![Fig. 13 Tongue flow angle at design flow rate $1.0Q_d (\theta_r=-24^\circ)$](image4)

![Fig. 14 Tongue flow angle at large flow rate $1.4Q_d (\theta_l=-24^\circ)$](image5)
4.3 PIV measurement result

The absolute velocity vector overlapped with the velocity contours and the vorticity distribution at the design flow rate $1.0Q_d$ are shown in Fig. 15 and 16 respectively. The velocity vector shows the direction of the velocity and contour represents its magnitude. Anti-crock wise vorticity is a plus and crock wise vorticity is a minus in Fig. 16. The absolute velocity increases with the increase of the radial position and becomes high near the trailing edge on the suction surface of the impeller. The strong anti-crock wise vorticity occurs on the suction surface of the impeller because the test impeller has the large blade outlet angle $\beta_2=60^\circ$ and causes the separation on the suction surface of the impeller. In addition to that, the crock wise vortex shedding from the trailing edge of the impeller is induced and large slip is caused at the outlet of the impeller. When the trailing edge of the impeller approaches the casing tongue $\theta_r=0^\circ$, this vortex region from the trailing edge of the impeller is divided into two regions, volute inlet and discharge throat regions, and the strong interaction between the flow from the trailing edge and the casing tongue is generated.

The velocity vector and its contours at the partial flow rate $0.8Q_d$ is shown in Fig. 17. The absolute velocity at the radial outer region becomes larger than that at the design flow rate $1.0Q_d$, which corresponds the increase of head at the partial flow rate in Fig. 10(a). The distribution of the velocity contours at $0.8Q_d$ is similar to that at $1.0Q_d$, however, there is the difference of the magnitude of the velocity. Figure 18 shows the vorticity distribution at the same flow rate $0.8Q_d$. Anti-crock wise vorticity is a plus and vice versa. The characteristics of the vorticity distribution at the partial flow rate $0.8Q_d$, the anti-crock wise vorticity on the suction surface of the
impeller, crock wise vortex shedding from the trailing edge of the impeller and interaction between the flow from the trailing edge and the casing tongue, are the same with that at design flow rate $1.0Q_d$. The vorticity magnitude at $0.8Q_d$ is smaller than that at $1.0Q_d$, so the noise and vibration related to the vortex will be small at $0.8Q_d$. The velocity vector and its contours at the large flow rate $1.4Q_d$ is shown in Fig. 19. The velocity at the radial outer region becomes small with the increase of the flow rate in Fig. 15, 17 and 19. The stagnation point on the casing tongue shifts the casing inlet side and causes the radially outer flow near the casing tongue, when the trailing edge of the impeller approaches the casing tongue $\theta_r = 0^\circ$.

The vorticity distribution at the large flow rate $1.4Q_d$ is shown in Fig. 20. Anti-crock wise vorticity is a plus in Fig. 20. The magnitude of the vorticity on the suction surface of the impeller and vorticity shedding from the trailing edge of the impeller becomes larger with the increase of the flow rate as shown in Fig. 16, 18 and 20. The high vortex region on the suction surface of the impeller attaches on the suction surface at $1.4Q_d$ because the radial velocity increases with the increase of the flow rate and flow angle becomes large. The vorticity near the casing tongue shows the maximum value at $\theta_r = 0^\circ$ and vorticity near throat wall becomes large with the interaction between the flow from the trailing edge and the casing tongue.

![Fig. 18 Vorticity distribution at partial flow rate $0.8Q_d$](image1)

![Fig. 19 Absolute velocity vector overlapped with velocity contours at large flow rate $1.4Q_d$](image2)

![Fig. 20 Vorticity distribution at large flow rate $1.4Q_d$](image3)
5. Concluding Remarks

The internal flow condition near the casing tongue of the mini centrifugal pump having large blade outlet angle $\beta_2=60^\circ$ was investigated by PIV measurement. The flow rate of the maximum efficiency shifts larger flow rate $1.2Q_d$ and the relatively high efficiency is kept in large flow rate region. The performance of the test mini centrifugal pump could be captured well by the numerical analysis. The trend of the tongue flow angle distribution between the PIV measurement and numerical analysis is almost the same in the range $-1.0\text{mm} < x < 1.0\text{mm}$ at the design flow rate. Furthermore, the value of the tongue flow angle is not so different each other and the tongue flow angle is minus in the region $-2.0\text{mm} < x < 1.5\text{mm}$ for both PIV measurement and numerical analysis at the design flow rate. On the other hand, there is a discrepancy in the tongue flow angle at the large flow rate between the PIV measurement and numerical analysis.

The strong anti-croch wise vorticity occurs on the suction surface of the impeller because the test impeller has the large blade outlet angle $\beta_2=60^\circ$. In addition to that, the crock wise vortex shedding from the trailing edge of the impeller is induced and large slip is caused at the outlet of the impeller. When the trailing edge of the impeller approaches the casing tongue $\theta_t=0^\circ$, the vorticity near the casing tongue shows the maximum value and the strong interaction between the flow from the trailing edge and the casing tongue is generated. The stagnation point on the casing tongue shifts the casing inlet side and causes the radially outer flow near the casing tongue at the large flow rate $1.4Q_d$ when the trailing edge of the impeller approaches the casing tongue $\theta_t=0^\circ$.

Nomenclature

- $b_1$: Inlet width of impeller [mm]
- $b_2$: Outlet width of impeller [mm]
- $b$: Axial width from hub [mm]
- $B$: Blade width [mm]
- $D_1$: Inner diameter of impeller [mm]
- $D_2$: Outer diameter of impeller [mm]
- $D_{in}$: Diameter of suction pipe [mm]
- $D_{out}$: Diameter of discharge pipe [mm]
- $H_d$: Design head [m]
- $L$: Shaft power [W]
- $N$: Rotational speed [min$^{-1}$]
- $N_d$: Design rotational speed [min$^{-1}$]
- $Q$: Flow rate [l/min]
- $Q_d$: Design flow rate [l/min]
- $s$: Blade thickness [mm]
- $x$: Horizontal position from casing tongue [mm]
- $z$: Blade number
- $\alpha$: Tongue flow angle [°]
- $\beta_i$: Blade inlet angle [°]
- $\beta_o$: Blade outlet angle [°]
- $\eta$: Efficiency
- $\theta_t$: Rotational angle [°]

References

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