Original Paper

Influence of Blade Outlet Angle and Blade Thickness on Performance and Internal Flow Conditions of Mini Centrifugal Pump

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Abstract

Mini centrifugal pumps having a diameter smaller than 100mm are employed in many fields; automobile radiator pump, ventricular assist pump, cooling pump for electric devices and so on. Further, the needs for mini centrifugal pumps would become larger with the increase of the application of it for electrical machines. It is desirable that the mini centrifugal pump design be as simple as possible as precise manufacturing is required. But the design method for the mini centrifugal pump is not established because the internal flow condition for these small-sized fluid machines is not clarified and conventional theory is not suitable for small-sized pumps. Therefore, we started research on the mini centrifugal pump for the purpose of development of high performance mini centrifugal pumps with simple structure. Three types of rotors with different outlet angles are prepared for an experiment. The performance tests are conducted with these rotors in order to investigate the effect of the outlet angle on performance and internal flow condition of mini centrifugal pumps. In addition to that, the blade thickness is changed because blockage effect in the mini centrifugal pump becomes relatively larger than that of conventional pumps. On the other hand, a three dimensional steady numerical flow analysis is conducted with the commercial code (ANSYS-Fluent) to investigate the internal flow condition. It is clarified from the experimental results that head of the mini centrifugal pump increases according to the increase of the blade outlet angle and the decrease of the blade thickness. In the present paper, the performance of the mini centrifugal pump is shown and the internal flow condition is clarified with the results of the experiment and the numerical flow analysis. Furthermore, the effects of the blade outlet angle and the blade thickness on the performance are investigated and the internal flow of each type of rotor is clarified by the numerical analysis results.

Keywords: Mini centrifugal pump, Performance, Internal flow condition, Blade outlet angle, Blade thickness

1. Introduction

Pumps are used widely in the field as the fluid pumping machine and it is recently expected to apply it to the mobile fuel cell and medical instruments.[1-2] Especially, the research related to the ventricular assist pump has been conducted actively all over the world. And some of the research represents significant breakthroughs for troubles like a blood clot which occurs when the mini centrifugal pump are used for the ventricular assist pump by the adoption of the magnet pump. Then the ventricular assist pumps are made practicable in Europe using the magnet pump. In addition to that, experimental investigation of the internal flow condition of the ventricular assist pump has been conducted with numerical investigation.[3-4] The diameter of the ventricular assist pump and pump for the cooling of electrical devices like personal computers is about a dozen millimeters and this kind of pump belongs to the mini pump type. Therefore, there are many cases that the conventional standard and theory cannot be applied due to the small size of the pump and the design method is not established.[5] 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 semi-open impeller. But it is considered that the semi-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 semi-open impeller designed by the original design method in which the blade outlet angle and blade number were increased.[6-8] The two dimensional impeller is often used for the mini centrifugal pump because the advantage of the three

dimensional impeller for the performance can not be fully realized in the case of the mini centrifugal pump. Furthermore, mini centrifugal pump has a tendency to be of low specific speed which means high head and low flow rate and it was clarified that sufficient performance could be accomplished with the two dimensional impeller under the condition of the low specific speed.[9] Then the semi-open impeller for the mini centrifugal pump with 55mm impeller diameter is adopted for this research to take simplicity and maintenance into consideration. In the case of the mini centrifugal pump, blade thickness becomes relatively large against the size of the pump compared to the conventional turbo-pump in order to keep the minimum strength of the rotor. In this research, the influence of blade outlet angle on performance of the mini centrifugal pump with the two dimensional impeller is investigated by using the three types of the impeller with different blade outlet angle. The numerical analysis is conducted and internal flow conditions are shown and the relation between performance and internal flow are discussed. Furthermore, the effect of the blade thickness on the performance and the internal flow condition of the mini centrifugal pump with two dimensional blades is considered by means of experimental and numerical results.

2. Experimental Procedure and Numerical Analysis Conditions

2.1 Experimental Apparatus and Method

A test rotor of a mini centrifugal pump was designed based on the conventional design method under the condition that design head, flow rate and rotational speed were $H_d=2.0$ [m], $Q_d=16.7$ [l/min] and $N_d=2230$ [min⁻¹] on the assumption that the mini centrifugal pump could be used for cooling of electrical devices. Then, specific speed was $N_s=171$ [min⁻¹.m³/min.m]. Figure 1 and Table 1 show the rotor of this test pump and primary dimensions of the rotor respectively. The rotor had two dimensional impeller and the inner diameter of the rotor $D_1=27$ mm, the outer diameter $D_2=55$ mm and blade width $B=4.7$ mm. It was difficult to obtain adequate head for the mini centrifugal pump with the semi-open impeller because of the low Reynolds number effect and a leakage flow from a tip clearance.[5] The blade outlet angle influences head of the pump. Therefore, three types of impellers with different blade outlet angles were prepared in this research. The blade inlet angle of TypeA was $\beta_1=15^\circ$ and the blade outlet angle of TypeA was $\beta_2=22.5^\circ$ which was recommended by the conventional design method. Further the blade outlet angle of TypeB and TypeC were $\beta_2=45^\circ$ and $\beta_2=60^\circ$ respectively under the condition that other design parameters were the same as that of TypeA. The blade thickness of TypeA, TypeB and TypeC was $t=2.0$ mm. Blockage effect of the mini centrifugal pump became relatively larger than that of large size turbo-pumps because the blade thickness became relatively large against the rotor diameter for the case of the mini centrifugal pump to keep the strength of the impeller. In order to keep the flow passage, the blade number could be reduced. But, a slip at the outlet of the rotor became large for the mini centrifugal pump and the influence of the slip would be large by the decrease of the blade number.[5] Therefore, we focused on the blade thickness in this research. Three types of impeller with thin blade thickness were prepared to investigate the effect of the blade thickness on performance and internal flow condition. TypeD, TypeE and TypeF were impellers in which only the blade thickness was changed as $t=1.0$ mm from TypeA, TypeB and TypeC respectively. The suction diameter of the pump was $D_{in}=26$ mm and the discharge diameter of it was $D_{out}=13$ mm. The sectional view of the volute casing was rectangular in shape and the inner diameter at the volute was 62mm. Therefore, the clearance at the volute tongue was 3.5mm because the outer diameter of the rotor was $D_2=55$ mm. The schematic diagram of experimental apparatus is shown in Fig.2. Water was used for the experiment. For the pressure performance evaluation, the static head differences on the wall between $2D_{in}$ upstream and $2D_{out}$ downstream of the rotor were measured. Then, the total pump head were 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 installed far downstream of the pump and torque was measured by a torque meter. Then the shaft power was calculated by the torque and the rotational speed measured by a rotational speed sensor. The shaft power was evaluated by the torque eliminating the torque using a disc without the impeller in this performance test. Then, the hydraulic efficiency of the pump η was calculated as the ratio of the water power to the shaft power.

Fig. 1 Test pump rotor (TypeA)

Table 1 Primary dimensions of rotors

Geometry	A	B	C	D	E	F
Inlet diameter D_1 [mm]	27					
Outlet diameter D_2 [mm]	55					
Inlet width b_1 [mm]	4.7					
Outlet width b_2 [mm]	4.7					
Blade number Z [-]	6					
Blade thickness s [mm]	2			1		
Blade inlet angle β_1 [deg]	15					
Blade outlet angle β_2 [deg]	22.5	45	60	22.5	45	60

2.2 Numerical Analysis Method and Conditions

In the numerical analysis, the commercial software ANSYS-Fluent was used and the numerical analysis was conducted with a numerical model which was the same with the test section of the mini centrifugal pump used in the experiment under the three dimensional steady condition. 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 numerical grids used for the numerical analysis are shown in Fig.3. The numerical domain at the inlet was $5D_{in}$ upstream of the test section and one at the outlet was $5D_{out}$ downstream. The constant velocity and the constant pressure were given as the boundary conditions at the inlet and the outlet respectively. Fine grids were arranged near the tip clearance and the blade. The numerical analyses were performed at five flow rate points of 80%, 90%, 100% 110% and 120% of the design flow rate.

3. Experimental and Numerical Results

3.1 The Effect of Blade Outlet Angle and Blade Thickness on Pump Performance

Figure 4 shows the performance curves of each rotor TypeA, B, C, D, E, F obtained by the experiment. The tip clearance was $c=0.5\text{mm}$ and the flow rates ranged from shut off flow rate to large flow rate in the experiments. Horizontal axis is flow rate Q and vertical axis of each Fig.4(a),(b),(c) are head H , shaft power L and efficiency η . Focused on the head curves in Fig.4(a), the head $H=1.42\text{m}$ was obtained at the design flow rate $Q_d=16.7\text{l/min}$ for TypeA which was lower than the design head $H_d=2.0\text{m}$. On the other hand, the head increased according to the increase of the blade outlet angle and the head $H=2.02\text{m}$ was obtained at the design flow rate $Q_d=16.7\text{l/min}$ for TypeC which had the largest outlet angle. In the case of thin blade thickness $t=1.0\text{mm}$, the head $H=1.52\text{m}$ was obtained at the designed flow rate $Q_d=16.7\text{l/min}$ for TypeD which was larger than TypeA by $\Delta H=0.10\text{m}$ and the head $H=2.17$ was obtained for TypeF which was larger than TypeC by $\Delta H=0.15\text{m}$. It could be found from these results that the head became larger according to the decrease of the blade thickness. It was confirmed from Fig.4(c) that efficiency in the case with thin blade thickness $t=1.0\text{mm}$ at larger flow rate was higher than that with thick blade thickness $t=2.0\text{mm}$. Further, the difference of the efficiency at large flow rate $Q=23.4\text{l/min}$ between TypeA and TypeD of the same blade outlet angle $\beta_2=22.5^\circ$ was $\Delta\eta=4.4\%$. On the other hand, the difference of the efficiency at large flow rate $Q=23.4\text{l/min}$ between TypeC and TypeF of the same blade outlet angle $\beta_2=60^\circ$ was $\Delta\eta=5.9\%$. Therefore, it was clarified that the blade thickness influenced the efficiency especially at large flow rate in mini centrifugal pumps with different blade outlet angles. Figure 5 shows the performance curves of each rotor TypeA, B, C, D, E, F obtained by the numerical analysis. It was found that the head and shaft power of numerical analysis were different from the experimental results. Furthermore, the head of TypeC and F with large blade outlet angle at partial flow rate did not have the same shape as the experimental head. It was considered that the difference between the experimental results and the numerical analysis results would be caused by the separation because of the blade geometry especially for the large blade outlet angle case and the flow conditions at the partial flow rate. Therefore, we need to conduct the numerical analysis in unsteady flow condition in near future to improve the precision of the numerical analysis. But the qualitative tendency that the head increased according to the increase of the blade outlet angle and decrease of the blade thickness was

Fig. 2 Schematic diagram of experimental apparatus

Fig. 3 Numerical grids

Fig. 4 Performance curves by experiment

confirmed. Further, the tendency that the efficiency of the case with thin blade thickness $t=1.0\text{mm}$ became larger than that with thick blade thickness $t=2.0\text{mm}$ was also observed in Fig.5(c). Therefore, qualitative tendency of the experimental results could be captured by the numerical analysis. Then, internal flow conditions were investigated by the numerical analyses results.

3.2 The Effect of Blade thickness on Internal Flow Condition

Figures 6(a),(b) show sectional velocity vectors of TypeA and TypeD at non-dimensional width $b/B=0.25$ respectively. The tip clearance was $c=0.5\text{mm}$ and the flow rate was the design flow rate $Q_d=16.7\text{l/min}$. The velocity vectors are shown as relative velocity in the rotor and absolute velocity in the casing. The non-dimensional width is defined as a ratio of distance from the hub to a casing width B with 0 and 1 corresponding to the hub and the shroud. Enlarged figures of the flow condition in Fig.6 near the outlet of the volute of TypeA and TypeD are shown in Figs.7(a),(b) respectively. It was confirmed from Fig.6(a),(b) that the flow conditions of both TypeA and TypeD were similar in that water flowed along the blade. Although, it could be observed from Fig.7

Fig. 5 Performance curves by numerical analysis

Fig. 7 Enlarged velocity vectors
($b/B=0.25$, $\beta_2=22.5^\circ$, $c=0.5\text{mm}$, $Q=16.7\text{l/min}$)

Fig. 6 Internal flow conditions ($b/B=0.25$, $\beta_2=22.5^\circ$, $c=0.5\text{mm}$, $Q=16.7\text{l/min}$)

that the velocity decreased from blade to blade in the case with thin blade thickness $t=1.0\text{mm}$ due to the increase of the size of the blade passage. Yet, the flow conditions of TypeA and TypeD were almost the same. Therefore, it was considered that the increase of the head in the case of TypeD with thin blade thickness at the design flow rate $Q_d=16.7\text{l/min}$ in Fig.4(a) was mainly caused by the decrease of the frictional loss. Then, theoretical frictional loss evaluated by the following equation on the assumption that one dimensional uniform flow was used to estimate the frictional loss. Table 2 shows the difference of theoretical frictional loss between the rotor with blade thickness $t=2.0\text{mm}$ and that with blade thickness $t=1.0\text{mm}$ at each blade outlet angle. The difference of the head between the rotor with blade thickness $t=2.0\text{mm}$ and that with blade thickness $t=1.0\text{mm}$ at each blade outlet angle obtained by the numerical results are also shown in Table 3.

$$h_f = \frac{c_e \lambda (l/4m) w_m^2}{2g} \quad (1)$$

The ratio of the frictional loss to the head difference at each blade outlet angle was 73% for blade outlet angle $\beta_2=22.5^\circ$, 51% for blade outlet angle $\beta_2=45^\circ$ and 21% for blade outlet angle $\beta_2=60^\circ$ from Tables 2 and 3. Therefore, the frictional loss against the head difference between the rotor with blade thickness $t=2.0\text{mm}$ and that with blade thickness $t=1.0\text{mm}$ was large in the case of blade outlet angle $\beta_2=22.5^\circ$. On the contrary, the frictional loss against the head difference was small in the case of blade outlet angle $\beta_2=60^\circ$. The difference of the head between TypeC and TypeF with blade outlet angle $\beta_2=60^\circ$ could be caused by other reasons related to the flow condition in addition to the change of the frictional loss.(See the discussion below about the mixing loss) Then, flow conditions were focused by numerical results. Figures 8(a),(b) show sectional shear rate of TypeA and TypeD at non-dimensional width $b/B=0.25$ respectively. The tip clearance was $c=0.5\text{mm}$ and the flow rate was the design flow rate $Q_d=16.7\text{l/min}$. The shear rate of TypeA with thick blade thickness was higher than that of TypeD with thin blade thickness in blade to blade and near the blade surface. It could be found that the frictional loss was large in the case of blade outlet angle $\beta_2=22.5^\circ$ from these results. Figures 9(a),(b) show sectional velocity vectors of TypeC and TypeF at non-dimensional width $b/B=0.25$ respectively. The tip clearance was $c=0.5\text{mm}$ and the flow rate was the design flow rate $Q_d=16.7\text{l/min}$. Flow distributions of TypeC and TypeF became non-uniform in circumferential direction and water didn't flow out the flow passage near the tongue. Therefore, the mixing loss of blade outlet angle $\beta_2=60^\circ$ at the outlet of the rotor could be larger than that of blade

Table 2 Difference of theoretical frictional loss

Blade outlet angle β_2 [deg]	Difference of theoretical frictional loss [m]
22.5	0.0338
45	0.0169
60	0.0128

Table 3 Difference of head

Blade outlet angle β_2 [deg]	Difference of head ΔH [m]
22.5	0.0460
45	0.0333
60	0.0617

Fig. 8 Shear rate distributions
($b/B=0.25$, $\beta_2=22.5^\circ$, $c=0.5\text{mm}$, $Q=16.7\text{l/min}$)

outlet angle $\beta_2=22.5^\circ$. The enlarged figure of flow condition in Fig.9 in blade to blade of TypeC and TypeF are shown in Figs.10(a),(b) respectively. The uniform velocity region was wide for TypeF with thin blade thickness and the mixing loss would decrease for TypeF. It was thought that the influence of the flow condition in the case of blade outlet angle $\beta_2=60^\circ$ on the head difference between TypeC and TypeF would be large compared to the influence of the flow conditions on the head difference between TypeA and TypeD of blade outlet angle $\beta_2=22.5^\circ$.

Fig. 9 Internal flow conditions ($b/B=0.25$, $\beta_2=60^\circ$, $c=0.5\text{mm}$, $Q=16.7\text{l/min}$)

Fig. 10 Enlarged velocity vectors ($b/B=0.25$, $\beta_2=60^\circ$, $c=0.5\text{mm}$, $Q=16.7\text{l/min}$)

4. Concluding Remarks

For the purpose to investigate the effect of blade outlet angle and blade thickness on the performance and the internal flow condition of a mini centrifugal pump having a two dimensional blade, experimental and numerical analyses were conducted. As a result, the following conclusions were obtained.

1. The head of the mini centrifugal pump increased according to the decrease of the blade thickness. Further the efficiency near large flow rate was improved by the decrease of the blade thickness.
2. The relative velocity increased according to the increase of the blade thickness in the case of blade outlet angle $\beta_2=22.5^\circ$ and the frictional loss mainly influenced the head difference between TypeA and TypeD. On the other hand, fluid loss related to flow condition influenced on the head difference between TypeC and TypeF in the case of blade outlet angle $\beta_2=60^\circ$.

Nomenclature

b	Axial width from hub [mm]	P	Static pressure [Pa]
B	Casing width [mm]	P_T	Total pressure [Pa]
c	Tip clearance [mm]	Q	Flow rate [l/min]
c_e	Empirical constant	Q_d	Design flow rate [l/min]
D_1	Inner diameter of impeller [mm]	t	Blade thickness [mm]
D_2	Outer diameter of impeller [mm]	w_m	Root mean square relative velocity at inlet and outlet of rotor [m/s]
D_{in}	Diameter of suction pipe [mm]	β_1	Blade inlet angle [°]
D_{out}	Diameter of discharge pipe [mm]	β_2	Blade outlet angle [°]
g	Gravitational acceleration [m/s ²]	η	Efficiency [%]
H_d	Design head [m]	$\Delta\eta$	Efficiency difference [%]
ΔH	Head difference [m]	λ	Coefficient of Pipe friction
l	Blade chord length [m]	Subscripts	
L	Shaft power [W]	1	Impeller inlet
m	Averaged hydraulic radius of blade passage [m/s]	2	Impeller outlet
N	Rotational speed [min ⁻¹]	d	Design point
N_d	Design rotational speed [min ⁻¹]		

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