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Original Paper

Performance and Internal Flow Condition of Mini Centrifugal Pump with Splitter Blades

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Abstract

Mini centrifugal pumps having a diameter smaller than 100mm are employed in many fields. 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, mini centrifugal pumps with simple structure were investigated by this research. Splitter blades were adopted in this research to improve the performance and the internal flow condition of mini centrifugal pump which had large blade outlet angle. The original impeller without the splitter blades and the impeller with the splitter blades were prepared for an experiment. The performance tests are conducted with these rotors in order to investigate the effect of the splitter blades on performance and internal flow condition of mini centrifugal pump. On the other hand, a three dimensional steady numerical flow analysis is conducted with the commercial code (ANSYS-CFX) to investigate the internal flow condition in detail. It is clarified from experimental results that the performance of the mini centrifugal pump is improved by the effect of the splitter blades. Blade-to-blade low velocity regions are suppressed in the case with the splitter blades and total pressure loss regions are decreased. The effects of the splitter blades on the performance and the internal flow condition are discussed in this paper.

Keywords: Mini centrifugal pump, Performance, Internal flow, Splitter blade

1. Introduction

Pumps are used widely in industrial field as fluid pumping machine and it is recently expected to apply it to mobile fuel cells and medical instruments.[1-2] Especially, research related to ventricular assist pumps have been conducted actively all over the world. And some of the researches represent 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 magnet pump. Then, the ventricular assist pumps are made practicable in USA and Europe using the magnet pump. In addition to that, experimental investigations of internal flow condition of the ventricular assist pump have been conducted with numerical investigation.[3-4] The diameters of the ventricular assist pump and pump for the cooling of electrical devices like personal computers are about a dozen millimetres and these kind of pumps belong to the mini pump. Therefore, there are many cases that the conventional standard and theory can’t 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 performance of the mini centrifugal pump based on the conventional design method using a closed impeller was better than that using a 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 maintenance of the pump and manufacture of an 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 blade outlet angle and blade number were increased.[6-8] The two dimensional impeller is often used for the mini centrifugal pump because advantages of the three dimensional impeller for the performance can’t be fully realized in the case of the mini centrifugal pump. Furthermore, the mini centrifugal pump has a tendency of low specific speed which means high head and low flow rate. 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 was adopted for this research to take simplicity and maintenance into consideration. It was clarified that the head of the mini centrifugal pump decreased due to the viscous effect from the previous research and the effect of the increase of the blade outlet angle on the increase of the head was confirmed under the condition that the rotational speed and pump size

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were kept constant. On the contrary, the efficiency decreased according to the increase of the blade outlet angle because of the complex internal flow condition.[10] Therefore, splitter blades were adopted in this research to improve the performance and the internal flow condition of the mini centrifugal pump which had large blade outlet angle. 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 splitter blades on the performance and the internal flow condition of the mini centrifugal pump with two dimensional blades were investigated and suitable splitter blade length was discussed by the experimental and numerical results.

2. Experimental Procedure and Numerical Analysis Conditions

2.1 Experimental Apparatus and Method

A test rotor of the 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=2.0m, Q_d=16.7l/min and N_d=2230min^{-1} on the assumption that the mini centrifugal pump could be used for cooling of electrical devices. Then, specific speed was Ns=171min^{-1},m^3/min,m. The blade inlet angle was β_1=15° and the blade outlet angle was set as β_2=22.5° which was recommended by the conventional design method for the first impeller named TypeA. However, 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] Therefore, the large blade outlet angle β_2=60° 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 that of the first model. This base model rotor without splitter blades was named as TypeC. On the other hand, three types of rotor having splitter blades with different blade chord were prepared. The splitter blade chord lengths of three types of the rotor TypeG-1, TypeG-2 and TypeG-3 were 1/2, 3/8 and 1/4 of the main blade chord respectively. The splitter blades were installed at the mid blade-to-blade region in circumferential direction and radial position of the trailing edge of the splitter blades was the same with the trailing edge of the main blade. The blade thickness of main blade and splitter blade of all types were t=2.0mm. Figure 1 and Table 1 show the rotor having splitter blade of this test pump and primary dimensions of the rotors respectively. The rotor had two dimensional impeller and the inner diameter of the rotor D_1=27mm, the outer diameter D_2=55mm and blade width B=4.2mm. The suction diameter of the pump was D_{in}=26mm and the discharge diameter of it was

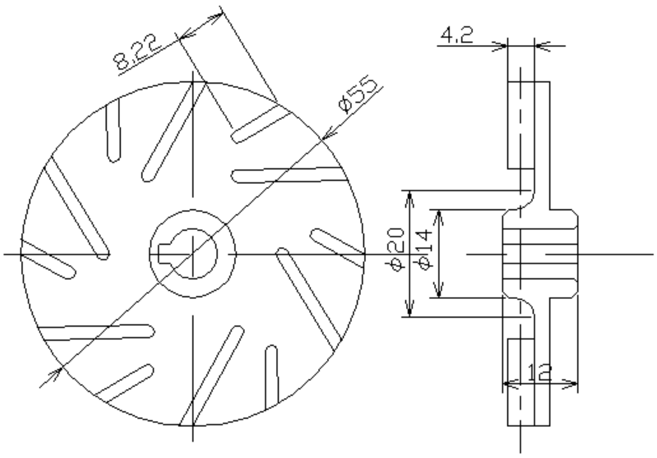


Fig. 1 Test pump rotor (TypeG-1)

Table 1 Primary dimensions of rotors

Geometry	TypeC	TypeG-1	TypeG-2	TypeG-3
Inlet diameter at hub D_{1h} [mm]	27	27	27	27
Inlet diameter at tip D_{1t} [mm]	27	27	27	27
Outlet diameter D_2 [mm]	55	55	55	55
Inlet width b_1 [mm]	4.2	4.2	4.2	4.2
Outlet width b_2 [mm]	4.2	4.2	4.2	4.2
Main Blade number Z [-]	6	6	6	6
Splitter Blade number Z_s [-]	0	6	6	6
Blade thickness s [mm]	2	2	2	2
Blade inlet angle β_1 [deg]	15	15	15	15
Blade outlet angle β_2 [deg]	60	60	60	60
Splitter blade length l_{sb} [mm]	-	8.2	6.2	4.1

D_{out}=13mm. Sectional view of a volute casing was rectangular in shape and an inner diameter at the volute was 62mm. Therefore, a clearance at a volute tongue was 3.5mm because the outer diameter of the rotor was D_2=55mm. 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 mechanical loss and disc friction loss 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.

2.2 Numerical Analysis Method and Conditions

A numerical flow analysis was conducted to investigate the internal flow condition in detail. In the numerical analysis, the commercial software ANSYS-CFX 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-ω 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 that 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 six flow rate points of 80%, 90%, 100% 110% 120% and 140% of the design flow rate. Grid point numbers of inlet pipe, rotor region, casing region and outlet pipe region were 50,000 points, 1,200,000 points, 700,000 points and 50,000 points respectively.

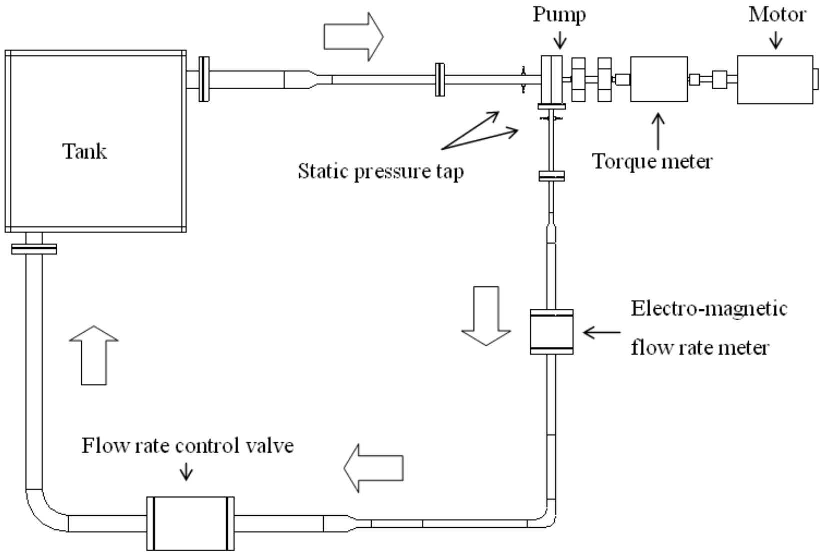


Fig. 2 Schematic diagram of experimental apparatus

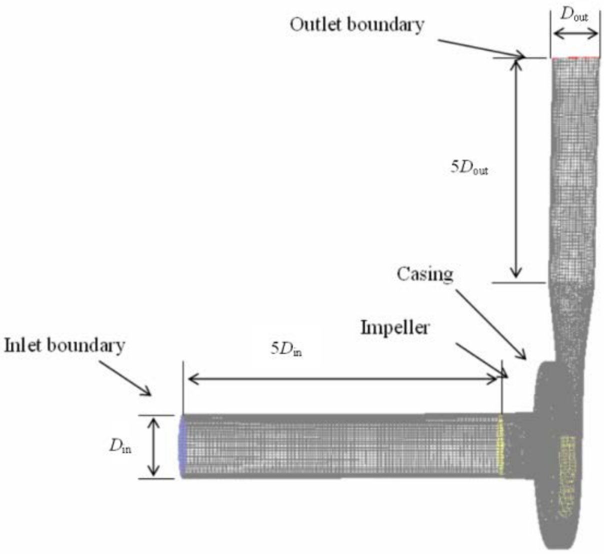


Fig. 3 Numerical grids

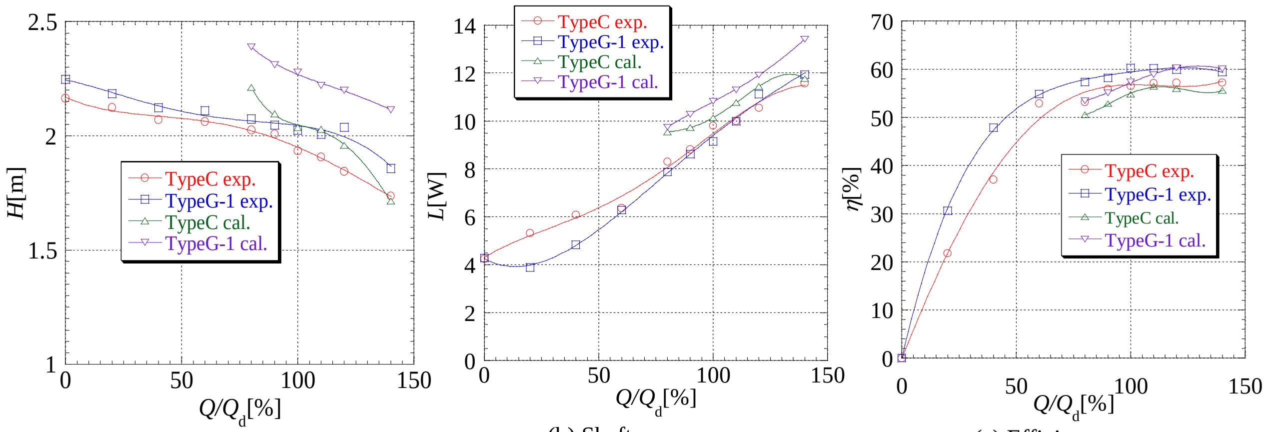
1. Experimental and Numerical Results

3.1 The Effect of The Splitter Blade on Pump Performance and Internal Flow

Figure 4 shows the performance curves of each rotor TypeC and TypeG-1 obtained by the experiment. The tip clearance was c=0.5mm and performance test was conducted in wide flow rates range from shut off flow rate to large flow rate in the experiments. Horizontal axis is flow rate Q and vertical axes of each Fig.4(a),(b),(c) are head H, shaft power L and efficiency η. The numerical analysis results were also shown in Fig.4 to compare the numerical results with the experimental results. Focused on the head curves of the experiments in Fig.4(a), the head (H=1.93m) was obtained at the design flow rate Q_d=16.7l/min for TypeC. On the other hand, in the case of TypeG-1, the head (H=2.07m) over the designed head (H=2.0m) was obtained at the design flow rate. Further, the head of TypeG-1 was larger than that of TypeC in all flow rates, where experiment was conducted. The efficiency of TypeC was η=56.5% at the design flow rate and efficiency of TypeG-1 was larger than that by about 3.7% at the design flow rate. Furthermore, the efficiency of TypeG-1 was also larger than that of TypeC in all flow rates, where experiment was conducted. It was clarified from the experimental results that the performance of the mini centrifugal pump improved by the effect of the splitter blades.

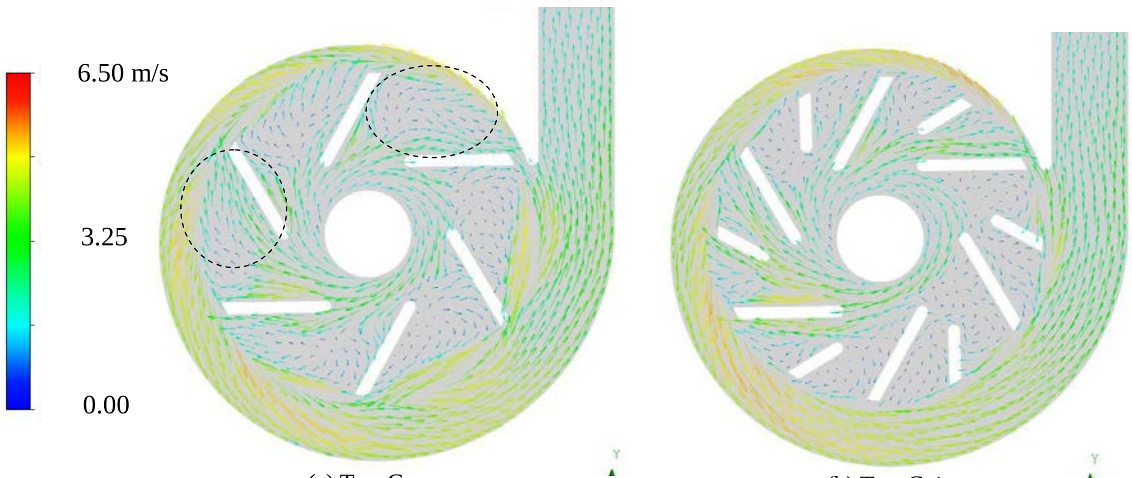
It was found that the quantity of the head, the shaft power and the efficiency of numerical analysis were different from the experimental results in Fig.4. It was considered that the difference between the experimental results and the numerical analysis results would be caused by the numerical conditions; grid number, turbulence model and steady condition, although, the flow condition was complex with the separation because large blade outlet angle was adopted in these impellers. Therefore, we need to check the grid dependence and so on in near future to improve the precision of the numerical analysis. But the qualitative tendency of the performance curves of the experiments that the head decreased and the shaft power increased according to the increase of the flow rates was also observed by the numerical results. Further, the qualitative tendency that the head and the efficiency of TypeG-1 were higher than that of TypeC was also confirmed by the numerical analysis results in Fig.4. Therefore, qualitative tendency of the experimental results could be captured by the numerical analysis. Then, internal flow conditions were investigated by the numerical analysis results.

Figures 5(a),(b) show axial cross-sectional velocity vectors of TypeC and TypeG-1 at non-dimensional width b/B=0.5 respectively. The tip clearance was c=0.5mm and the flow rate was the design flow rate Q_d=16.7l/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; b/B=0 and 1 correspond to the hub and the shroud. The blade-to-blade flow distribution weren't uniform; high velocity and low velocity regions existed on the pressure surface and near the suction surface respectively. It was also found that there were blade-to-blade secondary flows for TypeC as shown in broken circles. Furthermore, there was a blade-to-blade high velocity region at the outlet of the rotor near the casing tongue. Low velocity regions were found in the casing near the trailing edge of the impeller for TypeC and the mixing loss could be caused in the casing. On the other hand, the blade-to-blade high velocity region near the casing tongue and the blade-to-blade secondary flows were suppressed for the case of TypeG-1



(a) Head curves (b) Shaft power curves (c) Efficiency curves

Fig. 4 Performance curves



(a) TypeC (b) TypeG-1

Fig. 5 Velocity vectors (b/B=0.5, c=0.5mm, Q=16.7l/min) due to the splitter blades. Further, low velocity regions in the casing were suppressed for TypeG-1.

Figures 6(a),(b) show axial cross-sectional total pressure distributions of TypeC and TypeG-1 at non-dimensional width b/B=0.5 respectively. The tip clearance was c=0.5mm and the flow rate was the design flow rate Q_d=16.7l/min. The total pressure distributions near the outlet of the rotor became uniform in circumferential direction for TypeG-1 due to the splitter blades, although there was relatively large difference of total pressure in circumferential direction for TypeC. The low velocity regions near the trailing edge of the impeller were suppressed in the case with the splitter blades in Fig.5(b) and the total pressure loss regions near the trailing edge of the impeller was decreased for TypeG-1 in Fig.6(b). Therefore, the effects of the splitter blades with the half main blade chord length on the internal flow conditions were confirmed from these results.

3.2 The Effect of The Splitter Blade length on Pump Performance and Internal flow

The effects of the splitter blade with the half main blade chord length on the performance and internal flow were confirmed. On the other hand, it is suitable for the mini centrifugal pump to minimize the length of the splitter blade because the precise manufacture is difficult for the size limit of the mini centrifugal pump. Then, the performance tests were conducted using the rotors with the short blade chord length. Figure 7 shows the performance curves of each rotor TypeG-1, TypeG-2 and TypeG-3 obtained by the experiment. The tip clearance was c=0.5mm. Horizontal axis and vertical axis of each Fig.7(a),(b),(c) are the same in Fig.4(a),(b),(c). The numerical analysis results were also shown in Fig.7 to compare the numerical results with the experimental results. The head decreased in the case of the shortest blade chord length(TypeG-3) and the efficiency of the TypeG-1 was the highest in 3 types rotors. It was found from the experimental results that the half of the main blade length was the best blade chord length of the splitter blade in 3 types of rotors having splitter blades lengths from 1/2 to 1/4 of the main blade chord length. On the other hand, focused on the numerical analysis results, the qualitative tendency that head decreased and the shaft power increased according to the increase of the flow rates was confirmed in Fig.7. Further, the head and the efficiency of the numerical analysis results for TypeG-1 were higher than that for TypeG-2 and TypeG-3 at the design flow rate. Then, internal flow conditions were investigated to clarify the causes of the performance difference of TypeG-1, TypeG-2 and TypeG-3 by the numerical analysis results at the design flow rate.

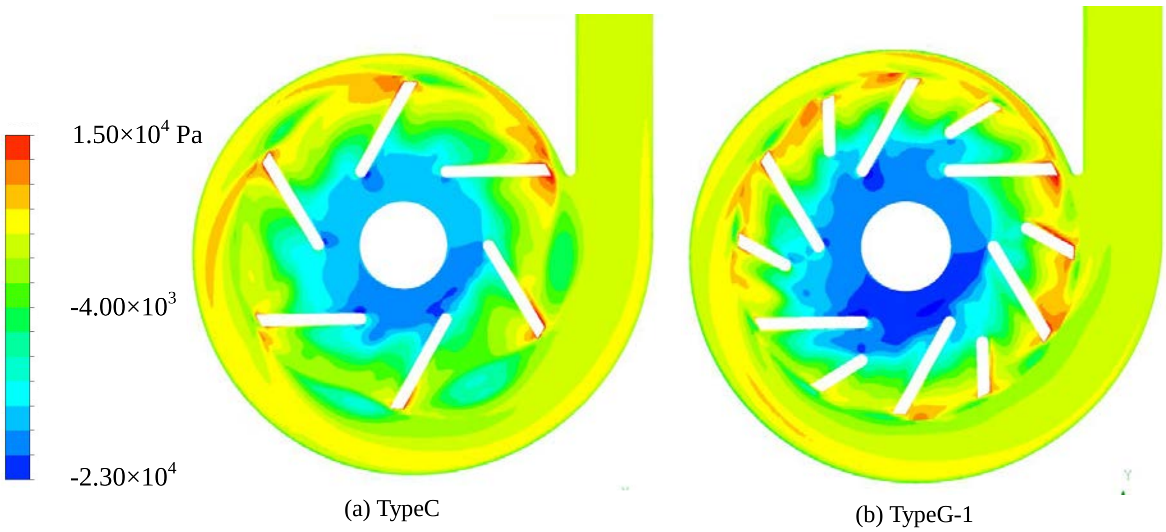


Fig. 6 Total pressure distributions (b/B=0.5, c=0.5mm, Q=16.7l/min)

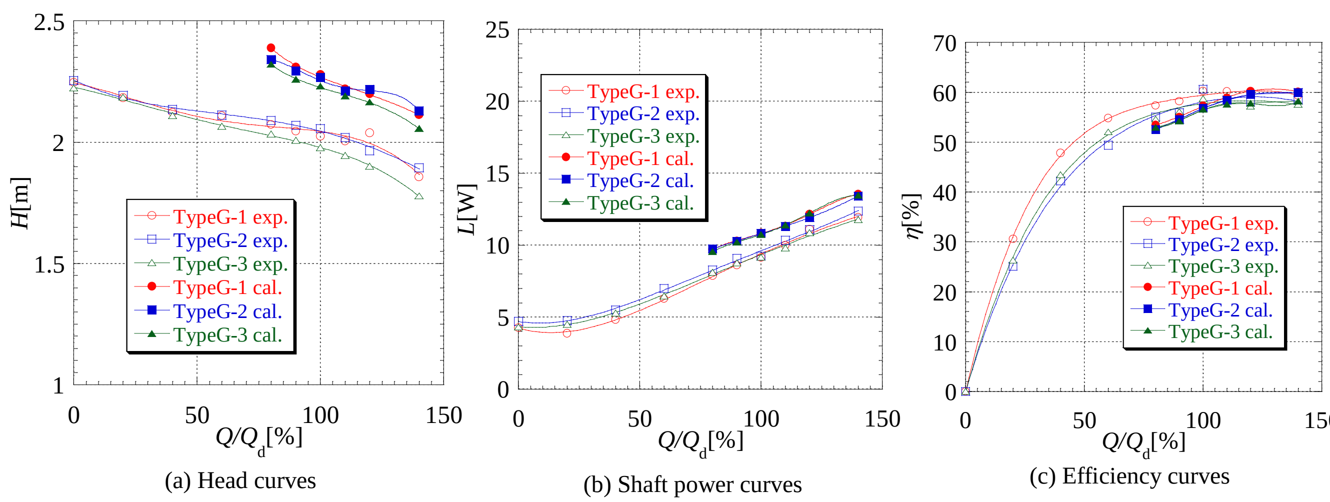


Fig. 7 Performance curves

Figures 8(a),(b),(c) show axial cross-sectional velocity vectors of TypeG-1, TypeG-2 and TypeG-3 at non-dimensional width b/B=0.5 respectively. The tip clearance was c=0.5mm and the flow rate was the design flow rate Q_d=16.7l/min. The velocity vectors are shown as relative velocity in the rotor and absolute velocity in the casing. Focused on the flow condition near the leading edge of the splitter blade, inclined flows to the pressure surface observed in Fig.8(b),(c) for TypeG-2 and TypeG-3 were suppressed for TypeG-1. Figures 9(a),(b),(c) show axial cross-sectional total pressure distributions of TypeG-1, TypeG-2 and TypeG-3 at non-dimensional width b/B=0.5 respectively. The tip clearance was c=0.5mm and the flow rate was the design flow rate Q_d=16.7l/min. The total pressure distribution at the outlet of the impeller became uniform for TypeG-1, although, the low total pressure regions were observed near the trailing edge of the impeller for TypeG-3. Therefore, the lengths of the splitter blade for TypeG-2 and TypeG-3 were too short to improve the performance and the internal flow condition. TypeG-1 was the best for 3 types of rotors on the point of view of the performance and internal flow condition. The optimum length and position of the splitter blade need to be investigated as future research works.

4. Concluding Remarks

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

1. The head and the efficiency of TypeG-1 were larger than that of TypeC in all flow rates, where experiment was conducted.
2. The blade-to-blade high velocity region near the casing tongue and the blade-to-blade secondary flows were suppressed for the case of TypeG-1 due to the splitter blades.
3. TypeG-1 was the best for 3 types of rotors having the blade chord length from 1/2 to 1/8 of the main blade chord length on the point of view of the performance and the internal flow condition.

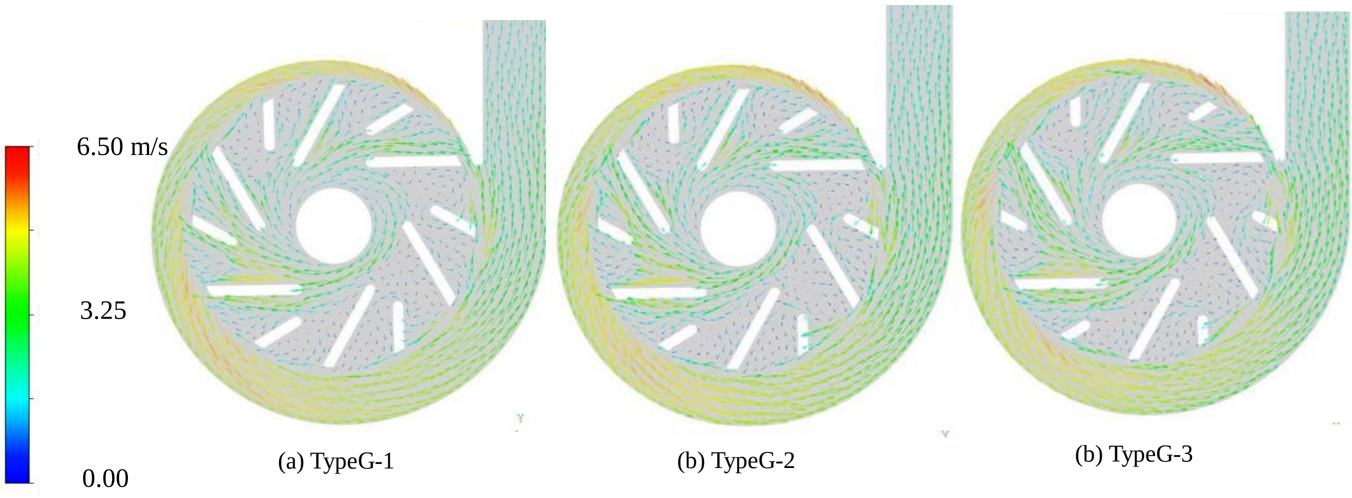


Fig. 8 Velocity vectors (b/B=0.5, c=0.5mm, Q=16.7l/min)

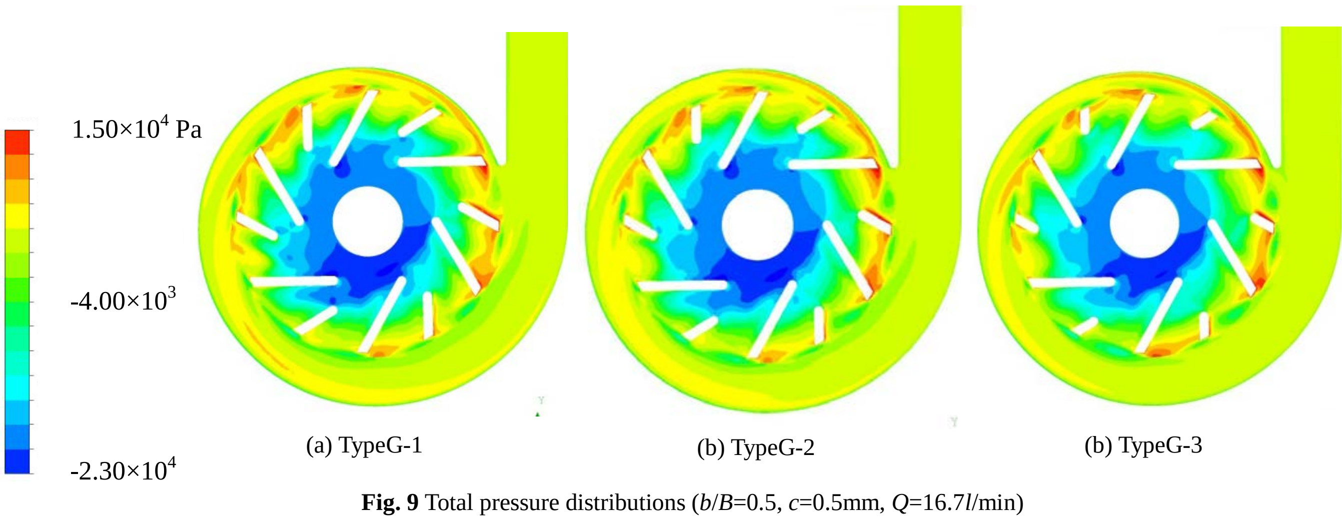


Fig. 9 Total pressure distributions (b/B=0.5, c=0.5mm, Q=16.7l/min)

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]
D_1	Inner diameter of impeller [mm]	Q_d	Design flow rate [l/min]
D_2	Outer diameter of impeller [mm]	t	Blade thickness [mm]
D_in	Diameter of suction pipe [mm]	β_1	Blade inlet angle [°]
D_out	Diameter of discharge pipe [mm]	β_2	Blade outlet angle [°]
H_d	Design head [m]	η	Efficiency [%]
l	Blade chord length [mm]		
l_sb	Splitter blade chord length [mm]	**Subscripts**	
L	Shaft power [W]	1	Impeller inlet
N	Rotational speed [min^{-1}]	2	Impeller outlet
N_d	Design rotational speed [min^{-1}]	d	Design point

References

[1] Akamatsu, T., and Tsukiya, T., 1998, “Development of a centrifugal blood pump with magnetically suspended impeller and the related fluid mechanical problems,” Sadhana, Vol. 23, No. 5-6, pp. 597-603.

[2] Horiguchi, H., Matsumoto, S., Tsujimoto, Y., Sakagami, M., and Tanaka, S., 2009, “Effect of Internal Flow in Symmetric and Asymmetric Micro Regenerative Pump Impellers on Their Pressure Performance,” International Journal of Fluid Machinery and Systems, Vol. 2, No. 1, pp. 72-79.

[3] Shao, J., Liu, S., Yuan, H., and Wu, Y., 2008, “Numerical Simulation and PIV Measurement on the Internal Flow in a Centrifugal Pump,” ASME Fluids Engineering Division Summer Meeting, Jacksonville, USA, Proc. FEDSM2008-55025.

[4] Wu, Y., Yuan, H., Shao, J., and Liu, S., 2009, “Experimental Study on Internal Flow of a Mini Centrifugal Pump by PIV Measurement,” International Journal of Fluid Machinery and Systems, Vol. 2, No. 2, pp. 121-126.

[5] Shigemitsu, T., Fukutomi, J., and Nasada, R., 2009, “Study on Performance and Internal Flow Condition of Mini Turbo-Pump,” Proceedings of the 10th Asian International Conference on Fluid Machinery, Kuala Lumpur, Malaysia, CD-ROM, AICFM112.

[6] Nishi, M., Liu, S., Yoshida, K., Tsukamoto, H., and Kouya, S., 1998, “Tip-Clearance Effect on the Performance of a High Speed Mini Turbopump,” Proceedings of The Third International Conference on Pumps and Fans, Beijing, China, pp. 223-231.

[7] Liu, S., Nishi, M., Yoshida, K., and Tsukamoto, H., 2000, “A Study on a High Speed Mini Turbopump(2nd Rep: Effect of Bypass Flow on The Pump Performance,” Proceedings of 2nd Symp. on Fluid Machinery and Fluid Engineering 1, pp. 633-636.

[8] Liu, S., Nishi, M., Yoshida, K., 2001, “Impeller Geometry Suitable for Mini Turbo-Pump,” ASME J.Fluids Engineering, Vol. 123, No. 3, pp. 500-506.

[9] Choi, Y., Kurokawa, J., and Matsui, J., 2006, “Performance and Internal Flow Characteristics of a Very Low Specific Speed Centrifugal Pump,” ASME J.Fluids Engineering, Vol. 128, No. 2, pp. 341-349.

[10] Shigemitsu, T., Fukutomi, J., and Kaji, K., 2011, “Influence of Blade Outlet Angle and Blade Thickness on Performance and Internal Flow Conditions of Mini Centrifugal Pump,” International Journal of Fluid Machinery and Systems, Vol. 4, No. 3, pp. 317-323.