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Impeller inlet geometry effect on performance improvement for centrifugal pumps †

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Abstract

This research treats the effect of impeller inlet geometry on performance improvement for a boiler feed pump, who is a centrifugal pump having specific speed of 183 $m \cdot m^3 min^{-1} \cdot min^{-1}$ and close type impeller with exit diameter of 450 mm. The hydraulic performance and cavitation performance of the pump have been tested experimentally. In order to improve the pump, five impellers have been considered by extending the blade leading edge or applying much larger blade angle at impeller inlet compared with the original impeller. The 3-D turbulent flow inside those pumps has been analyzed basing on RNG k-ε turbulence model and VOF cavitation model. It is noted that the numerical results are fairly good compared with the experiments. Based on the experimental test and numerical simulation, the following conclusions can be drawn: (1) Impeller inlet geometry has important influence on performance improvement in the case of centrifugal pump. Favorite effects on performance improvement have been achieved by both extending the blade leading edge and applying much larger blade angle at impeller inlet; (2) It is suspected that the extended leading edge have favorite effect for improving hydraulic performance, and the much larger blade angle at impeller inlet have favorite effect for improving cavitation performance for the test pump; (3) Uniform flow upstream of impeller inlet is helpful for improving cavitation performance of the pump.

Keywords: Inlet geometry; Cavitation; Hydraulic performance; Centrifugal impeller

1. Introduction

A boiler feed pump has been designed to satisfy high temperature (such as 280°C) and high pressure (up to 6.4 MPa) operation condition, where hydraulic performance and cavitation performance are both vital for the stability of the pump. In this study, the boiler feed pump has been tested experimentally, and five other impellers with revised inlet geometries basing on the original impeller i.e. Imp A0 are treated by numerical method so as to improve the pump performance. The impellers have the same impeller exit diameter and blade number. Imp A0 has been designed conventionally, namely by one dimension method. Other impellers are designed basing on Imp A0, but their inlet geometries are designed with different concepts: Imp A1, Imp A2 and Imp A3 have extended blade leading edge, while Imp A4 and Imp A5 have much larger blade angle at impeller inlet. Three dimensional steady turbulent flow in total flow passage of those pumps is analyzed basing on RANS equations, RNG k-ε turbulence model and VOF cavitation model. Based on these results, the effect of impeller inlet geometry on performance improvement for the pumps is discussed.

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2. Test pump and its impellers

The specification of test pump with Imp A0 is shown at Table 1. The test pump has the specific

Table 1. Specification of Imp A0 for the test pump.

Inlet diameter [mm]	at hub D_{1h}	115
	at tip D_{1t}	184
Blade angle at leading edge [°]	at hub β_{1h}	37.5
	at tip β_{1t}	20.5
Impeller diameter D_2 [mm]		450
Blade width at exit b_2 [mm]		28
Blade angle at exit β_2 [°]		40
Blade number Z		6
Impeller type		closed

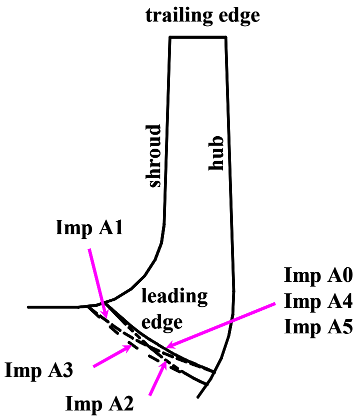


Fig. 1. Meridional configuration for test impellers.

speed of 183 m·m^{3}min^{-1}· min^{-1} ($\frac{n\sqrt{Q}}{H^{3/4}}$), and its impeller i.e. Imp A0 is designed conventionally, namely by one dimensional method. It is noted that the blade angle at impeller exit i.e. β_2 as large as 40° is chosen in order to reduce disc friction.

For the test pump, whose specific speed is low, the impeller inlet geometry is believed to be important for hydraulic performance and cavitation performance based on Yokoyama's studies [1, 2]. In this research, five other impellers have been considered. Those five impellers have almost the same geometries except that the inlet geometry is different as shown at Fig. 1 and Fig. 2.

As shown at Fig. 1, Imp A1 has leading edge extended toward pump inlet along the shroud, and Imp A2 has leading edge extended forward along the hub, while Imp A3 has leading edge extended forward along both shroud and hub based on Imp A0. Though Imp A4 and Imp A5 have the same leading edge as Imp A0, their blade angles are larger than that of Imp A0 as shown at Fig. 2.

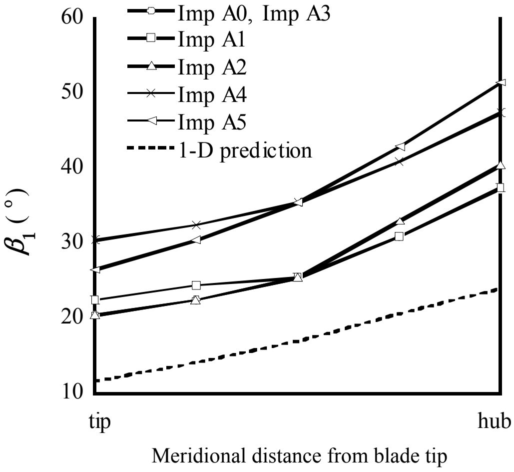


Fig. 2.Blade angle distribution along leading edge.

Fig. 2 shows that all impellers have larger blade angle than the flow angle predicted by one dimensional method [3] at the design point of test pump with Imp A0. Compared with the case of Imp A0, Imp A1, Imp A2, and Imp A3, the blade angles for Imp A4 and Imp A5 are much larger.

3. Methods and explanations

3.1 Experimental test

The hydraulic performance and cavitation performance for the test pump with Imp A0 have been tested experimentally. The rotational speed of the pump is 1480 min^{-1}. The experimental results indicate that the maximum efficiency point is close to the design point where the parameters are: Q = 0.125 m^3·s^{-1}, and H= 62 m.

3.2 Numerical calculation

In order to compare with the experimental data and investigate the flow condition inside the pump, the three dimensional turbulent flow in the total flow passage of the test pump whose calculation domain is shown at Fig. 3 has been simulated based on RANS equations and RNG k-ε turbulence model. For cavitation modeling, the VOF model where Rayleigh−Plesset equation is applied to treat the vapor generation and condensation in the phase change has been used.

In this paper, the cavitating flow in the pump for each impeller has been calculated near the design point of Imp A0 by applying a commercial CFD code i.e. Ansys-CFX.

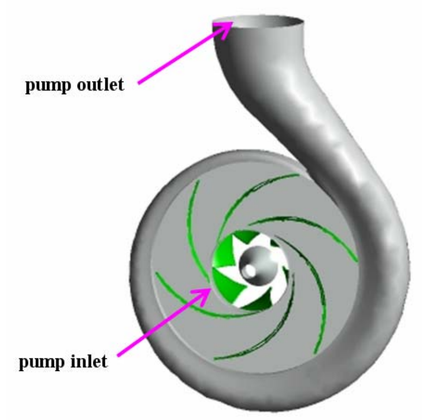


Fig. 3. Calculation domain.

The boundary conditions are as follows: the inlet of calculation domain is specified with averaged total pressure, which is reduced gradually when cavitation development is analyzed; at the outlet of the calculation domain, averaged mass flow-rate is given basing on mass equilibrium; for all solid walls, the non-slip wall condition is specified. Since there are a rotating frame for the impeller area and a stationary frame for pump stationary components, the sliding interface between two frames is treated as GGI method.

3.3 Parameter definition

For the convenience of treating the data from experiments and calculations, several necessary parameters are defined:

(1) Flow coefficient ϕ

$$\phi = \frac{Q}{A_2 u_2} \tag{1}$$

where A_2 : area at impeller exit [m^2], = $\pi D_2 b_2$; u_2: peripheral speed at impeller exit [m·s^{-1}], $=\omega D_2/2$. ω : angular speed [s^{-1}], = $2\pi n/60$.

(2) Head coefficient ψ

$$\psi = H \Big/ \frac{u_2^{\,2}}{2g} \tag{2}$$

(3) Power coefficient τ

$$\tau = \frac{P}{\rho A_2 u_2^3 / 2} \tag{3}$$

Note that the disc friction power [4] has been included to calculate the power input to the pump because the specific speed of the pumps in this study is low.

(4) Thoma's cavitation number σ

$$\sigma = NPSH \big/ H \tag{4}$$

where $NPSH$: net positive suction head [m], $= \frac{(p_0 - p_V)}{\rho g} + \frac{v_0^{\,2}}{2g}$. In the definition of $NPSH$, p_0, v_0: static pressure (Pa) and absolute velocity [m·s^{-1}] at pump inlet; p_v : vapor pressure [Pa], = 3574 Pa).

(5) Total pressure coefficient C_{pt}

$$C_{pt} = \frac{p_t - p_{t0}}{\rho u_2^2 / 2} \tag{5}$$

where p_t , p_{t0}: total pressure for the pump [Pa], and total pressure at pump inlet [Pa].

4. Results and considerations

Fig. 4 shows the characteristic curve for the test pump with Imp A0 at the rotational speed of 1480 min^{-1}. For comparison, the calculation results at the operation point of Q = 0.125 m^3·s^{-1} (ϕ=0.09) are plotted at the same figure. It is noted that near the maximum efficiency point, the numerical prediction is satisfactory except that power coefficient is a little under predicted.

The cavitation performance near the maximum efficiency point for the test pump with different impeller is shown at Fig. 5. Note that the data marked with exp are experimental result, and the others are numerical data in Fig. 5. For the case of Imp A0, the cavitation development predicted by the calculation is similar to the experimental result before the breakdown of pump performance. The numerical results are fairly good though the effect of cavitation on pump performance is a bit under-estimated as many cases for cavitation simulation by applying the present cavitation models [5, 6].

According to Fig. 5, performance breakdown is a little different if head coefficient or power coefficient is considered respectively. This phenomenon is understandable because the flow rate changes when cavitation develops. Usually, 3% performance breakdown point is defined as the critical cavitation point

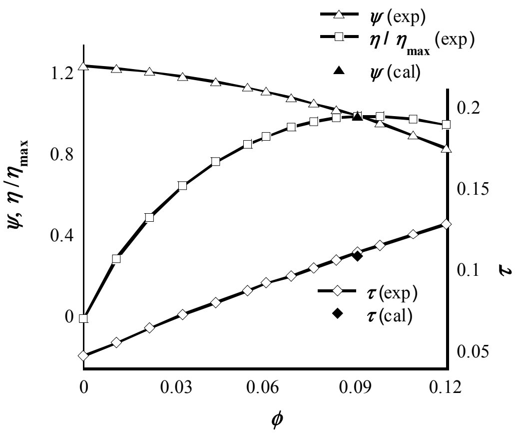
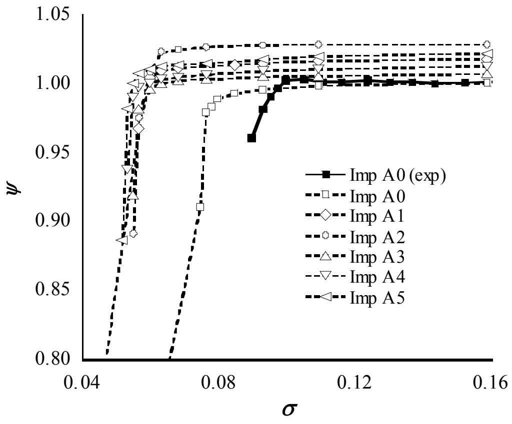
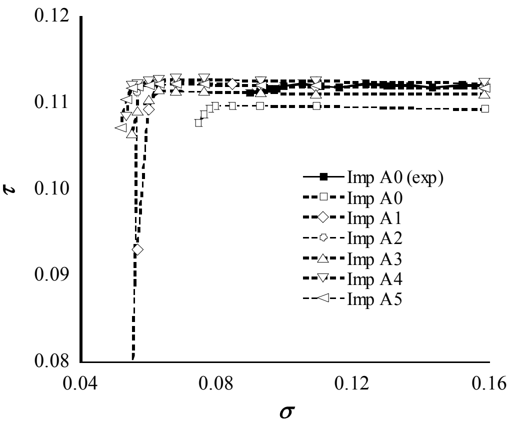


Fig. 4. Characteristic curve for the test pump with Imp A0 (n=1480 min^{-1}).



(a)



(b)

Fig. 5. Cavitation performance for the test pump (n=1480 min^{-1}, ϕ=0.09): (a) ψ vs. σ, and (b) τ vs. σ.

[3]. Compared with the case of Imp A0, both 3% performance breakdown points based on head coefficient drop and power coefficient drop for other impellers have much lower value of cavitation number.

Table 2 shows the comparison of hydraulic performance and cavitation performance for the test pump with different impellers. The critical cavitation number σ_c is calculated basing on 3% head coefficient drop.

From Table 2, it is noted that

(1) Compared with Imp A0, the other impellers with revised impeller inlet geometry basing on the conventional design have better hydraulic performance and cavitation performance;

(2) The extended leading edge such as Imp A1, Imp A2 and Imp A3 may improve hydraulic performance for the test pump. The favorite effect is resulted from the head rise by applying the extended part of blade at impeller inlet;

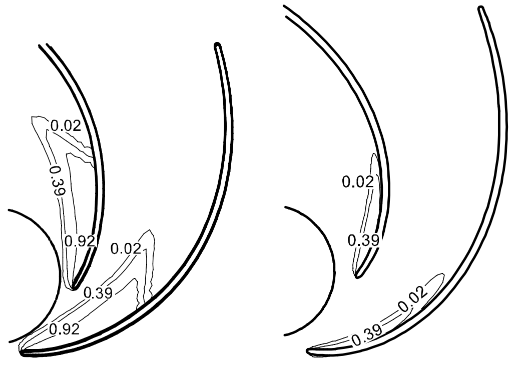
(3) Applying much larger blade angle at impeller inlet may have better effect on cavitation performance compared with extending the blade leading edge. According to Fig. 6 where the digital figure means the void fraction of cavity, the cavity in Imp A5 is controlled near the leading edge along the suction side, while that in Imp A0 expands toward the central blade-to-blade flow passage;

At the downstream of cavitation bubble, there is backflow zone which results in the drop of pump performance. As shown at Fig. 7, Imp A0 has larger backflow zone close to blade suction side compared with Imp A5. Further, there is another backflow zone near the pressure side in Imp A0;

Table 2. Performance for test pump with different impellers (n=1480 min^{-1}, ϕ=0.09).

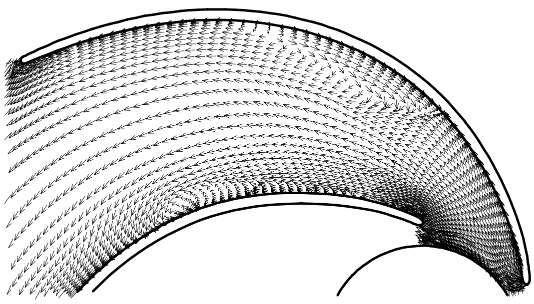
Impeller No.	$\psi / (\psi)^*_{A0}$	$\eta_{max} / (\eta_{max})^*_{A0}$	σ_c
Imp A0 (exp)	1.00	1.00	0.094
Imp A0 (cal)	1.00	1.05	0.079
Imp A1 (cal)	1.02	1.15	0.060
Imp A2 (cal)	1.03	1.18	0.062
Imp A3 (cal)	1.01	1.15	0.059
Imp A4 (cal)	1.01	1.14	0.055
Imp A5 (cal)	1.02	1.15	0.054

$(\psi)^*_{A0}$, $(\eta_{max})^*_{A0}$: experimental data i.e. head coefficient and best efficiency for the test pump with Imp A0.

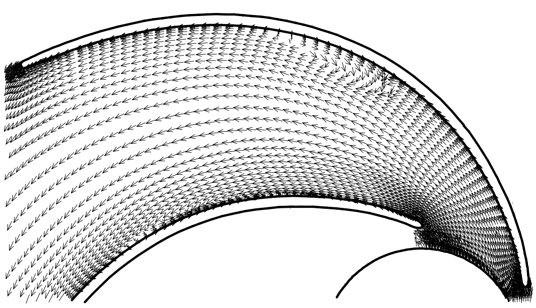


(a) (b)

Fig. 6. Cavity void fraction distribution at mid-span of impeller (n=1480 min^{-1}, ϕ=0.09, σ=0.076): (a) Imp A0, and (b) Imp A5.



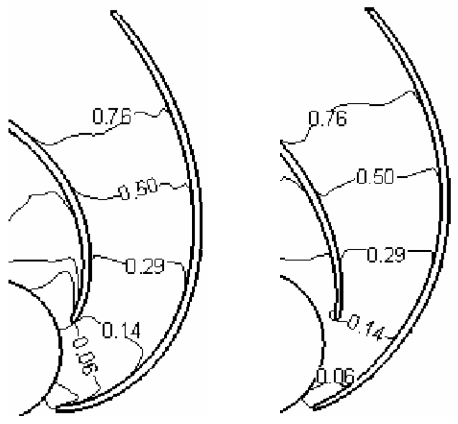
(a)



(b)

Fig. 7. Relative velocity at mid-span of impeller (n=1480 min^{-1}, ϕ=0.09, σ=0.076): (a) Imp A0, and (b) Imp A5.

(4) Fig. 8 indicates Imp A5 has more uniform total pressure distribution at impeller inlet than Imp A0. Thus, uniform flow upstream of impeller inlet is helpful for improving cavitation performance. Our previous research on mini pump also showed the same tendency [7, 8].



(a) (b)

Fig. 8. C_{pt} distribution at mid-span of impeller (n=1480 min^{-1}, ϕ=0.09, σ=0.076): (a) Imp A0, and (b) Imp A5.

5. Concluding remarks

Based on the experimental test and numerical simulation, the following conclusions can be drawn:

(1) Impeller inlet geometry has important influence on performance improvement in the case of centrifugal pump.

Favorite effects on performance improvement have been achieved by both extending the blade leading edge and applying much larger blade angle at impeller inlet;

(2) It is suspected that the extended leading edge toward pump inlet have favorite effect for improving hydraulic performance, and the much larger blade angle at impeller inlet have favorite effect for improving cavitation performance for the test pump;

(3) Uniform flow upstream of impeller inlet is helpful for improving cavitation performance.

Acknowledgments

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Nomenclature

C_{pt} : Total pressure coefficient
D : Impeller diameter, [m]
g : Gravity acceleration, [m·s^{-2}]
H : Pump head, (m)

n : Rotational speed, [min^{-1}]
p : Static pressure, [Pa]
P : Power, (W)
Q : Flow discharge, [$m^3 \cdot s^{-1}$] or [$m^3 \cdot min^{-1}$]
u : Peripheral speed, [$m \cdot s^{-1}$]
β : Blade or flow angle, [°]
ϕ : Flow coefficient
η : Efficiency
ρ : Fluid density, [$kg \cdot m^{-3}$]
σ : Thoma's cavitation number
ψ : Head coefficient
τ : Power coefficient

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