Effect of Inlet Flow Pattern on Cavitation Inception of Centrifugal Pumps*

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Flow patterns near the inlet portion of an impeller and cavitation inception were examined experimentally using a centrifugal pump with 3, 7 and 9 blade impellers. At low flow rates, velocities at the inlet of seven and nine blade impellers have non-uniform distributions inbetween hub and shroud, which cause a large drop of pressure on the suction surface of the blades near the shroud. In the impeller with three blades, a large vortex was created near the suction side of its inlet and this prevents severe drop in the pressure as seen in the seven blade impeller. Three kinds of cavitation were observed in the impeller passages according to the pressure distribution on the blade surface.

Key Words: Fluid Machine, Centrifugal Pump, Cavitation, Velocity Distribution, Pressure Distribution

1. Introduction

There are increasing demands for pumps with a higher suction performance, and studies on improvement of the performance have been carried out by many researchers in relation to the flow patterns near the inlet portion of the impeller passage. As the flow entering the impeller passage of centrifugal pumps is deflected from the axial to the radial direction, it is affected by impeller rotation, and thus is much distorted in the impeller inlet. It is necessary to obtain detailed knowledge on the flow pattern there in order to design impellers with high suction performance. Many studies have been carried out concerning the effect of blade inlet geometry on the pump performance and the inception of cavitation. For instance, Yokoyama⁽¹⁾ has experimentally examined the effects of the blade shape at the leading edge on the cavitation inception, and Senoo et al.⁽²⁾ have investigated the relationship between the number of impeller blades and cavitation characteristics of the pump.

The detailed nature of the cavitation inception and the relative flow patterns at the inlet of impeller passages, however, remain unclarified. In this study, velocity distributions in the impeller eye and pressure distributions on the blade

surface were experimentally examined, and the different types of cavitation generated inside the impeller passage were observed.

2. Nomenclature

- a : distance along blade centerline
- A_1 : area of impeller inlet
(blade thickness neglected)
- A_s : cross sectional area of suction pipe
- c : absolute flow velocity
- $\bar{C} = c/u$
- C_p : pressure coefficient
 $= (p - p_s) / (\rho u^2 / 2)$
- p : static pressure
- Q : rate of discharge
- r : radial distance
- r_1 : radius of impeller inlet
- r_2 : radius of impeller exit
- s : thickness of blade (= 10 mm)
- u : peripheral speed at measuring point
- w : relative flow velocity
- $\bar{W} = w/u$
- α : absolute air content
(corrected to standard temperature and pressure)
- α_s : saturation air content
(corrected to standard temperature and pressure)
- Γ, Γ' : circulation per blade
and its dimensionless value⁽³⁾
- σ_d : desinent cavitation parameter
 $= \text{NPSH} / (u^2 / 2g)$
- ϕ_1 : coefficient of flow rate $= Q / A_1 u_1$
- Subscripts
- m : meridional component
- u : peripheral component
- s : suction pipe
- 1 : just before impeller inlet
- 2 : just after impeller exit

3. Test Apparatus and Procedure

General arrangement of the test

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apparatus is shown in Fig. 1. Water from a head tank was led to a pump through a straight suction pipe with its length 18 times of its diameter and was returned to the tank. The discharge rate of the pump was measured with a nozzle installed at the inlet of the suction pipe, and was regulated by a throttling valve at the outlet of the pump. Both the suction and discharge head were measured at the sections of ① and ② in Fig. 1, respectively. Experiments were carried out on a centrifugal pump having an inlet diameter of 150 mm and a specific speed of $n_s = 284$ (min^{-1} , m^3/min , m).

Besides a normal impeller with seven blades, two different impellers with three and nine blades respectively, were also employed in order to examine the effect of the number of blades upon the cavitation inception. These three impellers have blades with the same profile, of which the inlet and outlet angles are $\beta_1 = 22^\circ 20'$ and $\beta_2 = 25^\circ 18'$, respectively, and thickness is 10 mm.

3.1 Velocity and pressure measurement

Velocity distribution just before the impeller passage were measured at a radial distance of $r/r_1 = 0.86$. Velocities on the seven and the nine blade impellers were measured at eight measuring points with equal angular distance along B-B as shown in Fig. 3(a), and velocities on the three blade impeller were measured at ten points with unequal angular intervals

Fig. 1 Schematic outline of experimental equipment

Fig. 2 Dimensions of impeller

between two blades as shown in Fig. 3(b). A cylindrical Pitot tube of 2.5 mm diameter was mounted on the impeller and traversed parallel to the impeller axis at these measuring points. Pressures obtained by the probe were transmitted to a stationary system by a mechanical seal. The pitch angle of the flow against stem of the tube can be neglected because of a large peripheral velocity component. The wall pressures on the blade surface near the leading edge were measured by a static holes drilled along three lines parallel to the shroud or hub surfaces as shown in Fig. 3(a). Measurements were taken at flow rates of $\phi_1 = 0.13$, 0.26 and 0.39, respectively, at a rotational speed of $n = 600$ min^{-1} , at which the similarity relations with respect to the pump performance were ascertained to hold and no cavitation bubbles were observed inside the impeller passages.

3.2 Cavitation test

To examine the cavitation inception, suction head of the pump was controlled by changing the pressure in the suction tank with a vacuum pump. Cavitation limit was decided upon by use of a stroboscopic light through a transparent casing and a stethoscope to listen to the cavity collapsing noise. Cavitation tests were performed at a rotational speed of $n = 800$ to 1000 min^{-1} , at which the cavitation similarity relations were checked to hold. During the cavitation tests relative air content of water was kept unchanged at $a/a_s = 0.55$ under the atmospheric pressure and water temperature was kept between 283 and 293K.

(a) $Z = 7, 9$

(b) $Z = 3$

Fig. 3 Measuring points at impeller inlet

4. Equations Expressing Experimental Results

4.1 Circulation per impeller blade

Provided that there is no pre-rotation before the impeller, the circulation around one blade may be assumed as $\Gamma = C_{u2} \cdot t_2$, where C_{u2} and t_2 denote the peripheral component of the absolute flow velocity and the blade pitch at the impeller exit, respectively. Using the peripheral speed of the impeller u_2 and the chord length of the blade l , Γ can be made dimensionless as

$$\Gamma' = \frac{\Gamma}{u_2 l} = \frac{C_{u2} t_2}{u_2 l} \quad (1)$$

Assuming that the total pump efficiency η is nearly equal to the hydraulic efficiency of the impeller η_h , as is the case in closed type impellers, the total head coefficient ϕ can be written as

$$\phi = \frac{H}{u_2^2/g} = \frac{u_2 \cdot C_{u2} \cdot \eta_h}{u_2^2} = \frac{C_{u2}}{u_2} \cdot \eta \quad (2)$$

Thus, using Eqs. (1) and (2) the dimensionless value of Γ' is given as

$$\Gamma' = \frac{\phi t_2}{\eta l} \quad (3)$$

4.2 Cavitation inception parameter σ_{bi}

Assuming that cavitation occurs on the blade surface when the minimum pressure p_{min} reaches the saturated vapor pressure of water p_v , the pressure coefficient at the minimum pressure point, C_{pmin} , is given by the following equation

$$C_{pmin} = \frac{p_{min} - p_{si}}{\rho u_2^2/2} = \frac{p_v - p_{si}}{\rho u_2^2/2} = \frac{p_v - p_{si}}{\rho u_1^2/2} \left(\frac{r_1}{r_2} \right)^2 \quad (4)$$

where p_{si} denotes the pressure in the suction pipe.

Cavitation inception parameter σ_{bi} can be defined by the equation as

$$\sigma_{bi} = \frac{p_{si} - p_v + \rho C_{pmin}^2/2}{\rho u_1^2/2} = \frac{p_{si} - p_v}{\rho u_1^2/2} + \left(\frac{A_1}{A_s} \right)^2 \phi_i^2 \quad (5)$$

Using Eqs. (4) and (5), the value of σ_{bi} based on the minimum value of C_{pmin} can be given as

$$\sigma_{bi} = \left(\frac{A_1}{A_s} \right)^2 \phi_i^2 - C_{pmin} \left(\frac{r_2}{r_1} \right)^2 \quad (6)$$

5. Experimental Results and Discussions

5.1 Velocity distributions before impeller passage

Figures 4(a), (b) and (c) show the absolute velocity distributions for $\phi_1 = 0.39$, 0.26 and 0.13, in the seven blade impeller measured at the section of $r/r_1 = 0.86$, upstream of the impeller passages, where the flow is changing from the axial to radial direction. At the large flow rate of $\phi_1 = 0.39$ it is seen that the velocity exhibits almost uniform distribution inbetween the hub and shroud, but non-uniform distribution inbetween the blades, the velocity decreasing in region

(a) $Z = 7$, $\phi_1 = 0.39$

(b) $Z = 7$, $\phi_1 = 0.26$

(c) $Z = 7$, $\phi_1 = 0.13$

Fig. 4 Velocity distributions before impeller passage ($Z = 7$)

B near the suction side of the blades and increasing in region M near the pressure side.

At the flow rate of the maximum efficiency point, $\phi_1 = 0.26$, however, the distribution of the meridian velocity components inbetween the hub and shroud tends to become non-uniform due to the effect of the existence of blade behind, the meridian component of velocity being reduced to almost zero in the suction side region B near the hub. The non-uniformity as shown above is lessened as the flow goes on in the blade passage. On the other hand, the peripheral component of the velocity is reversed in the regions near the blade edges, B and L, and in the regions M and N, separated from them. This is probably due to the effect of the circulation generated around the blades and the displacement effect of the blade existing just behind the measuring section.

When the flow rate is reduced to $\phi_1 = 0.13$, a reverse flow in the radial direction occur near the hub in region B. Unlike the case of $\phi_1 = 0.26$ and 0.39, the peripheral component of the velocity has the same direction as the impeller rotation, as is seen clearly near the shroud side, showing occurrence of a pre-rotation in the impeller inlet.

The velocity distributions in the passage of the nine blade impeller exhibit

Fig. 5 Velocity distributions before impeller passage ($Z = 3$, $\phi_1 = 0.13$)

Fig. 6 Vectorial indication of relative velocity distribution ($Z = 3$)

almost the same tendencies as the seven blade impeller, when the flow rates are $\phi_1 = 0.39$, 0.26, and 0.13, respectively. In this case the non-uniformity of the velocity distributions is decreased due to the increased number of blades.

Figure 5 illustrates the velocity profiles in the three blade impeller at a flow rate of $\phi_1 = 0.13$. Compared with the profiles in the seven blade impeller, larger non-uniformity of the meridional component is seen in the peripheral direction, showing the existence of inward flow in the region of B, near the suction side of the impeller passage. This non-uniformity in velocity distributions is considered to be brought about by wider impeller passages due to fewer impeller blades, which results in an increase in the circulation per one blade and brings about a decrease in the relative flow angle in the region near the leading edges of the blades.

On the other hand, the large peripheral component of velocity can be seen in the rotating direction, which indicates occurrence of a large pre-rotation in the impeller eye. This complicated distribution of velocity components before the impeller passages as shown in Fig. 5 coincides well with the results obtained from an oilfilm surface flow pattern (4).

Figure 6 shows a vectorial distribution of the relative velocities measured midway between hub and shroud, for $\phi_1 = 0.13$ and 0.39. Changes in the relative flow angle due to capacity change are found to be small in the regions of ML and MN distant from the leading edges of the blades but large in the regions near the inlet such as N, B and L. Negative flow angles as seen in the regions of L and B for $\phi_1 = 0.13$ show existence of a large vortex as shown by dotted lines in the suction side of the inlet section.

5.2 Pressure distributions on blade surfaces

Pressure distributions on the blade surface of the seven blade impeller at different blade heights are shown in Fig. 7. At a large flow rate of $\phi_1 = 0.39$, the pressure distributions along three sections with different heights show the same profile, which means that the flow is uniform at the impeller inlet in the hub-shroud direction as confirmed in Fig. 4(a). At this flow rate the stagnation point is seen to lie at the leading edge of the blade where the pressure coefficient C_p assumes its maximum.

At the optimum flow rate of $\phi_1 = 0.26$, however, the pressure distributions on the suction side differ with the passage height due to the non-uniformity of flow in the hub-shroud direction. At this flow rate the stagnation point is shifted slightly downstream to the pressure side and the drop of pressure at $a/s = 0.5$ decreases considerably compared with that of $\phi_1 = 0.39$. When the flow rate is decreased to $\phi_1 = 0.13$, such a large pressure drop as seen at $\phi_1 = 0.39$ cannot be realized and pressure distributions mea-

sured at different blade heights show different profiles both on the pressure and suction sides. This is due to the increased non-uniformity of the flow at the inlet region of the impeller. The decrease in blade loading near the shroud is due to existence of a pre-rotation as shown in Fig. 4(c).

Fig. 7 Pressure distributions on blade surface ($Z = 7$)

Fig. 8 Pressure distributions on blade surface ($Z = 3$)

Figure 8 shows pressure distributions on the blade of the three blade impeller when $\phi_1 = 0.39$ and 0.26 . Increased blade loading due to the decreased number of blades causes a larger pressure drop on the suction side than that of the seven or the nine blade impeller. The distribution of C_p on the suction side also differs with the blade height, which is due to a three-dimensional flow occurring at $\phi_1 = 0.39$. When the flow rate is reduced to $\phi_1 = 0.26$, the vortex region formed in the suction side of the blade prevents large pressure drops as compared with those for $\phi_1 = 0.39$.

5.3 Cavitation limit

In Fig. 9, changes in the desinent cavitation parameter σ_d for the three impellers are shown against flow rate. In the seven blade impeller the cavitation parameter σ_d remains almost unaltered in the range $0.03 < \phi_1 < 0.40$, where cavitation occurs on the suction side. Detailed observation of the seven blade impeller reveals that clouds of cavities are scattered near the blade surface at $a/s = 0.5$ and they travel downstream in a thick layer along the blade surface when the rate is below 0.28 . Hereinafter, these cavitating states are termed C2 cavitation. At larger flow rates of $\phi_1 > 0.28$ cavities are seen to be formed at the region of $a/s = 0.5$, fixed to the blade surface in a thin sheet (called C1 cavitation). The difference in the appearance of cavitation inception exactly corresponds to the pressure distributions observed on the impeller blades as shown in Figs. 7 and 8. Namely, the sheet cavities of C1 cavitation were brought about by a steep pressure drop observed at $a/s = 0.5$ on the suction side at an increased flow rate.

In the nine blade impeller the change in the value of σ_d against flow rate is small and the transition of the cavitation state from C2 to C1 cavitation occurs when the flow rate exceeds 0.24 approximately. In the three blade impeller, however, the values of the cavitation parameter are greatly increased because of an increase in pressure loading and a drop in the suction side wall pressure. In

Fig. 9 Changes in cavitation parameter σ_d

this case another type of cavitation is observed when the flow rate is below $\phi_1 = 0.30$, namely, cavitation bubbles were observed in the suction side apart from the blade surface (C3 cavitation). The smaller values of the cavitation parameter for C3 cavitation are caused by the vortex occurring on the suction side of impeller inlet. At a large flow rate, however, clouds of cavities with a certain thickness (that is, C2 cavitation as seen in the seven and the nine blade impellers) are generated.

In the same figure the values of σ_d for the cavitation generated in the pressure side of the seven and the nine blade impeller are also plotted.

5.4 Relation between σ_d and blade loading

As shown in Fig. 9 the desinent cavitation parameter σ_d depends largely on the blade loading. Using the circulation about a blade Γ , the changes in σ_d can be estimated as follows. Based on the wing theory, the pressure difference between the pressure and suction sides, Δp , is proportional to the lift exerted by the blade as $\Delta p \propto \rho U \Gamma$, where U is the velocity of undisturbed flow. Assuming that the pressure drop in the suction side $-\Delta p_s$ is also proportional to Δp , the following relation is obtained; $-\Delta p_s \propto \rho U \Gamma$. If the velocity U is replaced by the mean relative velocity at the inlet of the impeller passages, $-\Delta p_s$ is given as

$$-\Delta p_s \propto \rho U \Gamma = \rho \Gamma u_i \sqrt{1 + \phi_1^2} \dots \dots \dots (7)$$

The above relation is made dimensionless as

$$-C_{ps} \propto \Gamma' \sqrt{1 + \phi_1^2} \dots \dots \dots (8)$$

Using the relation of Eq. (6) and assuming that $\sigma_{bi} \approx \sigma_d$ (σ_{bi} ; incipient cavitation parameter), the following relation is established.

$$\Gamma' \propto \frac{\sigma_d - (A_1/A_s)^2 \phi_1^2}{1 + \phi_1^2} \equiv \sigma'_d \dots \dots \dots (9)$$

In Fig. 10 the relation given by Eq. (9) between σ'_d/Γ' and ϕ_1 is examined using experimental data on the three kinds of impellers. The curves for the seven and the nine blade impellers are seen to

Fig. 10 Relation between σ_d and blade loading

coincide well with each other and the value of σ'_d/Γ' tends to be constant at a large flow rate. At the low flow rate of ϕ_1 less than 0.26, the value of σ'_d/Γ' decreases due to the transition of cavitation from C1 to C2 type. On the other hand the value of σ'_d/Γ' for the three blade impeller increases linearly with the flow rate. A slight change in the slope of the curve can be seen at $\phi_1 = 0.30$, corresponding to the transition of the cavitation nature from C2 to C3. In Fig. 10 the data obtained by Takamatsu et al. (5) are also plotted for comparison. In spite of numerical difference it is found that the cavitation inception can be estimated by the relation between σ'_d/Γ' and ϕ_1 regardless of the number of blades if geometrical configuration is the same.

5.5 Relation between σ_d and σ_{bi}

According to the classical theory of cavitation inception, cavitation generally starts when the local minimum pressure falls to the saturated vapor pressure of the liquid p_v . In Fig. 11 the inception cavitation parameter σ_{bi} calculated by Eq. (6) using the above assumption and the desinent cavitation parameter σ_d are compared for the three different impellers. The agreement between σ_d and σ_{bi} is good at the flow rates of $\phi_1 > 0.28$ in the seven blade impeller and $\phi_1 > 0.24$ in the nine blade one, where C1 cavitation dominates the flow passages. But the discrepancy is remarkable in the three blade impeller due to the effect of a large vortex formed in the suction side of the passage inlet and the displacement of the minimum pressure point away from the blade surface.

5.6 Effect of air content on σ_d

The inception of cavitation is known to depend largely upon the dissolved air content in the liquid. The effect of the dissolved air on the desinent parameter

Fig. 11 Comparison of σ_d and σ_{bi}

σ_d is shown in Fig. 12 against the relative air content α/α_s . At the flow rate of $\phi_1 = 0.39$, where C1 cavitation dominates the impeller passages, the value of σ_d remains unaltered within the region of $\alpha/\alpha_s < 0.7$, but increases gradually when α/α_s exceeds 0.7. Similarly at the flow rate of $\phi_1 = 0.13$, where C2 cavitation is generated, the curves of σ_d exhibit almost the same tendencies as those for C1 cavitation. In the three blade impeller, however, the value of σ_d is almost constant within the experimental range of air contents when $\phi_1 = 0.13$ and as long as C3 cavitation dominates the impeller passages.

6. Conclusions

Flow patterns inside the impeller eye and cavitation inception were studied experimentally using centrifugal pumps with 3, 7 and 9 blade impellers, respectively. The results can be summarized as follows:

(1) At large flow rates, velocities at inlet sections of the seven and the nine blade impellers show almost uniform distributions inbetween hub and shroud. A large drop of pressure on the suction surface of the blade is observed at $\alpha/s =$

0.5. When the flow rate is decreased, the distributions of the inlet velocity and wall pressure on the suction surface of the blades exhibit non-uniform profiles inbetween hub and shroud.

(2) In the passage of the three blade impeller a large vortex is formed near the suction side of the blade inlet and this prevents severe drop in the pressure on the blade suction surface.

(3) Three kinds of cavitation inception in the impeller passage were found according to the pressure distributions on the blade surface. In the seven and the nine blade impellers cavities are fixed to the blade surface forming a thin sheet profile when the flow rate is large, but they disperse into a thick layer on the blade at a small flow rate. In the three blade impeller, cavitation is generated in the vortex core away from the suction surface of the blades.

(4) Prediction of cavitation inception in the impellers with any blade number can be made by considering the circulation per one blade of the impellers.

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Fig. 12 Effect of air content on σ_d