Experimental investigation on suction performance and cavitation instabilities of turbopumps with two different inducers

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Abstract. In the present study, the suction performance and the cavitation instabilities in turbopumps with two different inducers designed with different design incidence angle are experimentally investigated in the wide range of operating flow rate. Compared to the Inducer A, Inducer B is designed with larger incidence angle, therefore it has slightly larger inlet and outlet angles. The identical centrifugal impeller is used as a main impeller. As a result, the total head of pump with inducer B is confirmed to be larger than that with inducer B especially at large flow rates, while the shaft power is almost the same, resulting in the better efficiency with the inducer B. The suction performance is better with inducer B at large flow rates. Two kinds of instabilities, the cavitating whirling vortex and the surges are mainly observed for both inducers, but they are limited at low flow rates. The occurrence ranges of these cavitation instabilities are wider with inducer B.

1. Introduction

In turbopumps, miniaturization and high power density can be achieved by increasing the rotational speed. However, with high rotational speed, cavitation occurs, resulting in various problems such as deteriorations of head and efficiency, machine vibration, noise generation, erosion, etc. [1]. Installing an inducer upstream of main impeller is an effective way to improve the suction performance of turbopumps. However, it is known that various kinds of cavitation instabilities such as cavitation surge and rotating cavitation occur under operating conditions of low suction pressure [2]-[4]. For long time operation with inducer, instability-free operation is expected as well as improved suction performance in the wide operating range from shut-off to over flow rates. In our previous studies we have tried to suppress aforementioned instability phenomena, especially the cavitation surge, by installing the obstruction plate [5] or connecting with reduced-diameter suction pipe [6] upstream of helical inducers. It has been found that suppression of inlet back flow is effective in suppressing the instability phenomenon and, as a result, improving the suction performance. However, in either method, the effect is insufficient in an extremely low flow rate region. For further improvement at such extreme conditions, it seems to be necessary to combine the method with inducer blade shape suitable for suppressing the inlet backflow and the flow instabilities.
In the present study, from the viewpoint of the design and shape of inducer, the effects of design incidence angle of inducer on the suction performance and the cavitation instabilities in the wide flow rate range are experimentally investigated by employing two different inducers in combination with an identical main impeller.

2. Experimental method

Figure 1 shows the schematic view of test section of turbopump. Table 1 shows the specifications of two test inducers. The two inducers have been designed with the different incidence angles at the designed flow rate, therefore the different inlet blade angle, while the tip diameter, the blade number (three), meridional shape and so on are the same. Inducer A has moderate inlet and outlet blade angles, while Inducer B has larger inlet and outlet blade angles. The same centrifugal impeller with backward blades is employed as a main impeller, whose blade number is 15.

![Test section](image)

**Figure 1. Test section.**

|                   | Inducer A | Inducer B |
|-------------------|-----------|-----------|
| Number of blades  | 3         |           |
| Inlet hub-to tip ratio, $D_h/D_t$ | 0.38      | 0.38      |
| Inlet tip blade angle ratio | 1         | 1.14      |
| Outlet tip blade angle ratio | 1         | 1.23      |
| Tip clearance ratio, $c/D_t$ | 0.003     | 0.003     |

All Experiments are conducted in a closed loop cavitation tunnel. To obtain the hydraulic performance of turbopump, the flow rate $Q$, the head $H$, the torque $T$ are measured respectively by the electromagnetic flow meter installed downstream of test pump, static pressure transducers which are connected to pressure taps $\bar{\delta}$ as shown in Figure 1, and by the torque meter installed between the driving shaft and the motor. The rotational speed of test pump $N$ is kept constant by frequency control inverter. Temperature of working fluid, water, is also kept constant during the experiments with the aid of water cooling system equipped in the tunnel. To evaluate the time averaged values, we obtain the instantaneous data for 20 seconds with the sampling frequency of 1,500 Hz, and the flow coefficient $\phi$, head coefficient $\psi$, the shaft power coefficient $\lambda$, the efficiency $\eta$ are calculated by the following equations.

$$\phi = \frac{Q}{AU_2}, \quad \psi = \frac{H}{U_2^2/2g}, \quad \lambda = \frac{T\omega}{\rho AU_2^3}, \quad \eta = \frac{\rho gQH}{T\omega} = \frac{\phi \psi}{2\lambda}$$

(1)

where $A$ is the passage area of the main impeller exit, $U_2$ the peripheral velocity of impeller exit, $\omega$ angular shaft speed, $g$ gravitational acceleration.

To evaluate the suction performance of test pump, the measurements are conducted with reducing the tunnel pressure by a vacuum pump connected to a reservoir tank, while keeping the flow rate constant. The net positive suction head (NPSH) is calculated from the static pressure measured at $\bar{\delta}$ upstream of
inducer and the dynamic pressure based on the area-averaged velocity. The non-dimensional NPSH $\tau$ is defined by using the inducer tip speed $U_t$ as

$$\tau = \frac{\text{NPSH}}{U_t^2 / 2g}$$

(2)

In addition to the suction performance, we obtain the unsteady pressures at 4 locations $\mathbb{B}$, two of which locate just upstream of inducer with different azimuth locations and another two on the scroll casing ($\mathbb{C}$) wall as shown in Figure 1, to detect the occurrence of flow instabilities. Sampling frequency of the measurements is 5,000 Hz. The observation of cavitation is also conducted through the acrylic casing $\mathbb{3}$ by using a high-speed camera with the frame rate of 1,500 frame/s.

All experimental results presented below have been obtained under the constant shaft rotational speed of $N=3,000 \text{ min}^{-1}$. We have confirmed that the similitude against shaft speed well holds through the comparisons with $N=4,000 \text{ min}^{-1}$.

3. Results and discussion

3.1. Non-Cavitating Performance

Figure 2 shows the measured hydraulic performance of test pumps with the two inducers under non-cavitating state. The head coefficient $\psi$, the shaft power coefficient $\lambda$ and the efficiency $\eta$ normalized by those with Inducer A at the design flow rate (subscript of $dA$) are plotted against the flow coefficient normalized by the design flow coefficient, $\phi/\phi_{d}$. It can be seen that the head coefficient is larger with Inducer B than with Inducer A especially at larger flow rates, while the shaft power coefficient is almost the same, resulting in the better efficiency with Inducer B. The similar shaft power indicates the similar theoretical head, therefore the difference of head seems to come from the difference in the head loss between Inducers A and B. However, since the incidence angle to the inducer should be larger with Inducer B, the shock loss would be larger with it, which is in contradiction with the observed result. Therefore we think that the flow matching between inducer exit flow and the main impeller inlet is better with Inducer B, resulting in the better performance with it.

![Figure 2. Non-cavitating performance curves with Inducers A and B.](image)

3.2. Cavitating performance

Figure 3 shows the comparisons of suction performance curves between Inducers A and B at flow coefficients of (a) $\phi/\phi_{d}=0.14$, (b) 0.57, (c) 1.0 and (d) 1.42. The head and shaft power coefficients $\psi$ and $\lambda$ normalized by those at non-cavitation conditions (subscript of non-cavi) are plotted against the non-dimensional NPSH $\tau$.
At the flow rates below the design flow rate $\phi/\phi_d < 1$, the difference of suction performance between with Inducers A and B can be hardly seen. At the lowest flow rate $\phi/\phi_d = 0.14$ (Figure 3 (a)), both the head and shaft power coefficients $\psi$ and $\lambda$ gradually decrease with the decrease of NPSH $\tau$ in the region below $\tau < 0.15$. As shown later, the cavitation surge occurs in this region with gradual decrease of $\psi$ and $\lambda$, therefore the performance breakdown here is associated with the occurrence of cavitation surge. On the other hand at the moderate flow rates of $\phi/\phi_d = 0.57$ and 1.0 (Figure 3 (b) and (c)), the head and shaft power coefficients $\psi$ and $\lambda$ suddenly decrease. At those flow rates, the cavitation mainly develops in the tip region, and the tip leakage vortex cavitation extends not into the blade passage but toward upstream with the decrease of NPSH. As a result, the blade passage is kept unblocked and the performance is thought to be well maintained until the extreme development of cavitation.

However, at the large flow rate of $\phi/\phi_d = 1.42$, a clear difference of suction performance curves between with Inducers A and B can be seen. With Inducer B, the tip cavitation in the inducer still dominates and the performance curves are similar to those observed at the lower flow rates of $\phi/\phi_d = 0.57$ and 1.0. On the other hand, with Inducer A, the head coefficient $\psi$ gradually decreases while the shaft power coefficient $\lambda$ is kept constant before the final breakdown. Since the latter indicates that the theoretical head is also kept constant, the main impeller is well working perhaps without significant development of cavitation in it. From the observation, it is confirmed that the cavitation starts at the tip of the blade and spreads along the blade surface as the NPSH $\tau$ decreases. With the further decrease of $\tau$, the flow passage of inducer is blocked, which seems to result in the increasing of the head loss leading to the head breakdown.

Figure 3. Suction performance curves with Inducers A and B.
3.3. Cavitation Instabilities

The measurements of pressure fluctuations as well as the high-speed camera observation have been carried out to investigate the occurrence of cavitation instabilities. In the test pump with Inducers A and B, distinct flow instabilities are only observed at the low flow rates. Figures 4 and 5 show the typical results of FFT analysis of pressure fluctuations measured (a) upstream of inducer and (b) on the scroll casing wall for Inducers A and B respectively. The flow rate is \( \phi/\phi_d = 0.14 \). We also check the cross-correlation of pressure signals at the same streamwise location with different azimuth angles, from which we identify if the fluctuation is caused by rotating instabilities or axial (surge-like) instabilities (zero phase angle). In these waterfall plots, the horizontal axis is the normalized frequency \( F = f/f_N \) (\( f_N \): shaft speed frequency), the vertical axis is the non-dimensional NPSH \( \tau \) and the height is the normalized amplitude of pressure fluctuation defined by \( |\Delta \psi_s| = |p'_s|/(\rho U_t^2/2) \).

By comparing the results with Inducers A and B, very similar spectra can be seen, meaning that the similar instabilities occur. With the decrease of the non-dimensional NPSH \( \tau \), the component with normalized frequency of \( F \approx 0.25 \) is firstly observed almost only upstream of inducer (indicated by blue circles in Figures 4 and 5). The frequency \( F \) is almost kept constant with the decrease of \( \tau \), but before diminishing it becomes gradually lower. From the phase analysis of the pressure fluctuations and visual observation, this phenomenon is found to be caused by the cavitating whirling vortex in the inlet pipe around the rotating axis (Figure 6). Because of very low flow rate operation, the backflow from inducer to the inlet pipe is significant, which has a strong swirling velocity component, resulting in a formation of whirling longitudinal vortex around the rotating axis. Besides the component of almost constant frequency with \( F \approx 0.25 \), another component can be found at the similar frequency of \( F \approx 0.25 \) for the both upstream and downstream pressure fluctuations as indicated by green arrows. From the phase analysis, this fluctuation is found to be caused by the surge-like oscillation. Because the frequency becomes lower with the decrease of \( \tau \), this phenomenon seems to be related with cavitation.

With the further decrease of the non-dimensional NPSH \( \tau \), the BPF component with \( F = 3 \) becomes larger only in the upstream pressure fluctuations. After that, this component suddenly disappears with the appearance of very large components with the low normalized frequency of \( F \approx 0.02 \) in the upstream and downstream pressure fluctuations as indicated by red circles in Figures 4 and 5. From the phase analysis, this component is found to be due to the occurrence of surge-like mode, and in the visual observation, significant cavity volume fluctuation is observed with the same frequency, from which this mode is identified to be the cavitation surge phenomenon.

![Figure 4. FFT analysis of wall static pressure fluctuations at \( \phi/\phi_d = 0.14 \) (Inducer A).](image-url)
Figure 5. FFT analysis of wall static pressure fluctuations at $\phi/\phi_d=0.14$ (Inducer B).

Figure 6. Observation of cavitating whirling vortex (Inducer A).

Figure 7 summarizes the occurrence map of the cavitation surge phenomenon and cavitating whirling vortex in the normalized NPSH $\tau$ and the normalized flow coefficient $\phi/\phi_d$ plane. In these Figures, 3% and 5% head drop NPSH are plotted by the solid and broken lines, respectively. The red and blue symbols respectively indicate the occurrences of cavitation surge and the cavitating whirling swirling vortex, the black closed symbols indicate the maximum amplitude point of BPF component, and the black open symbols indicate no significant pressure fluctuation is observed.

By comparing two maps for (a) Inducer A and (b) Inducer B, it is easily noticed that the cavitation surge region is wider with Inducer B in the both flow rate and NPSH ranges. Also, the whirling vortex region is wider with Inducer B. However, the suction performance at larger flow rates is again confirmed to be better with Inducer B. All these are thought to be due to the larger inlet blade angle of Inducer B; the incidence angle is larger for Inducer B at the same flow rate condition. Actually if we plot those maps by the flow rate divided by shockless flow rate $m = Q/A_i U_i \tan \beta_{1b,t}$ ($A_i$: inlet flow area of inducer, $U_i$: inducer tip speed, $\beta_{1b,t}$: tip inlet blade angle of inducer) instead of $\phi/\phi_d$, we can obtain the similar maps for Inducers A and B as shown in Figure 8. This seems to indicate that the suction performance of turbopump and the occurrence of cavitation instabilities are greatly involved to the inlet flow condition of inducer if the identical main impeller is used.
4. Conclusions
In the present study, the effects of design incidence angle of inducer on the suction performance and instability characteristics of turbopump are experimentally investigated. Two inducers A with the moderate design incidence angle and B with the larger incidence angle are used in combination with identical main impeller. The results are summarized as follows.

(1) In terms of hydraulic performance, the turbopump with Inducer B is better especially at larger flow rates. But this seems not due to the difference of incidence angle but to that of exit blade angle between the two inducers.

(2) As for the suction performance, the head and the shaft power gradually decreases with the decrease of NPSH at the low flow rates with the both Inducers A and B. This gradual deterioration of performance is associated with the occurrence of cavitation surge. At the medium flow rates below the design flow rate, the performance breakdown suddenly occurs in the both head and shaft power. At the large flow rate, the suction performance with Inducer A is much worse. It seems that the cavitation extends along the blade surface due to small incidence angle, which easily blocks the blade passage, resulting in the earlier head breakdown.

(3) The flow instabilities are only observed at the lower flow rates for the both Inducers A and B. Two types of instabilities are mainly observed; cavitating whirling vortex along the rotating axis in the inlet pipe, and the cavitation surge phenomenon with low frequency.

(4) The occurrence range of the above two instabilities in terms of the flow rate and NPSH is larger with Inducer B. This is thought to be due to larger incidence angle. Actually if we plot the occurrence range of instabilities in terms of the flow rate normalized by the shockless flow rate, the similar occurrence map of instabilities are obtained with the both Inducers A and B. This seems to indicate that the suction performance of turbopump and the occurrence of cavitation instabilities are greatly involved to the inlet flow condition of inducer if the identical main impeller is used.
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