Suction Performance and Cavitation Instabilities of Turbopumps with Three Different Inducer Design

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Abstract

In the present study, the suction performance and the cavitation instabilities in turbo-pumps with three different inducers designed with different design incidence angle are experimentally investigated in the wide range of operating flow rate. Three inducers L with the lowest design incidence angle, M with the moderate one and H with the largest one are used in combination with identical main impeller. As a result, the total head of pump with inducer H is confirmed to be the largest especially at large flow rates, while the shaft power is almost the same, resulting in the best efficiency with the inducer H. The suction performance is the best with inducer H at large flow rates and is the best with inducer L at low flow rates. Two kinds of instabilities, the cavitating whirling vortex and the surges are mainly observed for all three inducers, but they are limited at low flow rates. The occurrence ranges of these instabilities in terms of the operating flow rate is the widest with inducer H. However, those in terms of the shockless flow rate ratio is similar for all three inducers: This fact can contribute to establish some guideline to the pump operation avoiding serious flow instabilities.

Keywords: Cavitation, Turbopump, Inducer, Cavitation performance, Design incidence angle

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]. To suppress these cavitation instabilities, there have been many studies on cavitating inducers especially for rocket engine turbopumps which operate in the limited flow rate range around design flow rate [5]-[8].

For long time operation of general-use turbopumps 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. However, the inlet backflow and the cavitation surge often occur at low flow rate range [9], [10] and they are hardly avoided because of their nature without suppression devices. In our previous studies, we have tried to suppress the cavitation surge, by installing the obstruction plate [11] or connecting with reduced-diameter suction pipe [12] upstream of helical inducers. It has been found that the suppression of inlet back flow is effective in narrowing the onset range of cavitation surge and in weakening the amplitude even if it occurs. As a result, the suction performance is also improved. 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 these method with the inducer having blade shape suitable for reducing the inlet backflow.

From the viewpoint of the inducer blade shape, the leading edge sweep back and reducing the inlet blade angle are effective to suppress cavitation instabilities [13]-[15]. They seem to be very successful ways but are still to be limited to near the design flow rate condition. In the design criteria provided by NASA [16], the range of incidence angle is recommended as 30-50% of the inlet blade angle. However, this criterion is not necessary to be appropriate for the general-use turbopump inducer. In the present study, 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. In our previous report [17], two different inducers designed with high and middle design incidence angles have been employed in combination with an identical main impeller to investigate the effects of incidence angle on the suction performance and cavitation instabilities. In this paper, another inducer with the low design incidence angle is also employed and experimented to obtain the concrete knowledge and findings on the effects of inducer design incidence angle.

2. Experimental method

Figure 1 shows the schematic view of test section of turbopump. Table 1 shows the specifications of three test inducers. The three 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 L, M and H has the smallest, moderate and largest inlet and outlet blade angles respectively. The same centrifugal impeller with backward blades is employed as a main impeller, whose blade number is 15.

![Fig. 1 Test section.](image)

<table>
<thead>
<tr>
<th>Inducer</th>
<th>Inducer L</th>
<th>Inducer M</th>
<th>Inducer H</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of blades</td>
<td>3</td>
<td>3</td>
<td>3</td>
</tr>
<tr>
<td>Inlet hub-to tip ratio, (D_h/D_t)</td>
<td>0.38</td>
<td>0.38</td>
<td>0.38</td>
</tr>
<tr>
<td>Inlet tip blade angle ratio</td>
<td>0.86</td>
<td>1</td>
<td>1.14</td>
</tr>
<tr>
<td>Outlet tip blade angle ratio</td>
<td>0.89</td>
<td>1</td>
<td>1.23</td>
</tr>
<tr>
<td>Tip clearance ratio, (c/D_t)</td>
<td>0.003</td>
<td>0.003</td>
<td>0.003</td>
</tr>
</tbody>
</table>

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 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 1500 Hz, and the flow coefficient \(\phi\), head coefficient \(\psi\), the shaft power coefficient \(\lambda\), the efficiency \(\eta\) are calculated by the following equations.

\[
\phi = \frac{Q}{AU^2}, \quad \psi = \frac{H}{U^2/2g}, \quad \lambda = \frac{T\omega}{\rho AU^2}, \quad \eta = \frac{\rho g Q H}{T \omega} = \frac{\phi \psi}{2 \lambda}
\]

where \(A\) is the passage area of the main impeller exit, \(U\) 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 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

\[
\tau = \frac{NPSH}{U_t^2/2g}
\]

In addition to the suction performance, we obtain the unsteady pressures at 4 locations, two of which locate just upstream of inducer with different azimuth locations and another two on the scroll casing wall as shown in Figure 1, to detect the occurrence of flow instabilities. Sampling frequency of the measurements is 5000 Hz. The observation of cavitation is also conducted through
the acrylic casing by using a high-speed camera with the frame rate of 1500 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 three inducers under non-cavitating state. The head coefficient \( \psi \), the shaft power coefficient \( \lambda \) and the efficiency \( \eta \) normalized by those with Inducer M at the design flow rate (subscript of \( dM \)) are plotted against the flow coefficient normalized by the design flow coefficient \( \phi / \phi_d \). It can be seen that the head coefficient is almost same at lower flow rates. However, especially at larger flow rates, the head coefficient is larger with Inducer H, while the shaft power coefficient is almost the same, resulting in the best efficiency with the Inducer H. 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 L, M and H. However, since the incidence angle to the inducer should be the largest with Inducer H, the shock loss would be the largest with it, which is in contradiction with the obtained result. Therefore we think that the flow matching between inducer exit flow and the main impeller inlet is the best with Inducer H, resulting in the best performance with it.

Fig. 2 Hydraulic performance curves of test pump with three different inducers

#### 3.2 Cavitating performance

Figure 3 shows the comparisons of suction performance curves of (a) with Inducer L, (b) with Inducer M and (c) with Inducer H. The head coefficient \( \psi \) normalized by those with Inducer M at the design flow rate is plotted against the non-dimensional NPSH \( \tau \).

At larger flow rates \( \phi / \phi_d = 1.28 \text{ and } 1.42 \), the difference of suction performance between three inducers can be apparently seen. The head coefficient \( \psi \) suddenly drops at low NPSH \( \tau \) with Inducer H. However, it gradually decreases with the decrease of NPSH \( \tau \) with Inducer L (at \( \phi / \phi_d = 1.28 \text{ and } 1.42 \)) and with Inducer M (at \( \phi / \phi_d = 1.42 \)). It has been confirmed that the shaft power coefficient \( \lambda \) keeps almost constant, indicating that the theoretical head is also kept constant and the main impeller is well working perhaps without significant development of cavitation in it. From the observation with Inducer L and M, 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 cause the gradual increase of the head loss leading to the head breakdown.

At moderate flow rates \( \phi / \phi_d = 0.421 \text{ – } 1.14 \), the head coefficient \( \psi \) suddenly decreases with all inducers. It has been confirmed that the shaft power coefficient \( \lambda \) also suddenly decreases at those flow rates. Since the shaft power coefficient relates with theoretical head of main impeller, it can be said that the head breakdown is caused by the decrease of the theoretical head of impeller. At those cases, 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.

At the low flow rates \( \phi / \phi_d = 0.145 \text{ and } 0.289 \), the head coefficient \( \psi \) begins to decrease at high NPSH \( \tau \) and the decrease is more gradual than those at the moderate flow rates. At extremely low flow rates \( \phi / \phi_d = 0.145 \), it can be seen that the suction performance is the best with Inducer L.

Figure 4 summarizes the suction performance of three inducers. In these figures, 3% and 50% head drop NPSH \( \tau \) are plotted by the solid and broken lines, respectively. The horizontal axes are (a) normalized flow coefficient \( \phi / \phi_d \) and (b) shockless flow rate ratio \( m = (Q / A_1) / U_{1\delta \beta} \tan \beta_{1\delta} \) (\( A_1 \): inlet flow area of inducer, \( \beta_{1\delta} \): tip inlet blade angle of inducer). From Fig. 4(a), it can be seen that 3% head drop NPSH at lower flow rates is the lowest with Inducer L while that at higher flow rates is the highest with Inducer H. On the other hand, from Fig. 4(b), 3% head drop NPSH is almost the same between three inducers. This seems to indicate that the suction performance of turbopump is greatly involved to the inlet flow condition of inducer if the identical main impeller is used. For 50% head drop, it can be seen that the NPSH is the highest with the Inducer L and is slightly lower with the Inducer H than with M.
3.3 Cavitating 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 L, M and H, distinct flow instabilities are only observed at the low flow rates. Figures 5 show the typical results of FFT analysis of pressure fluctuations measured upstream (left figure) of inducer and on the scroll casing wall (right figure) for Inducers (a) L, (b) M and (c) H, respectively. The flow rate is $\phi/\phi_d=0.145$. We have also checked 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 $|\Delta\psi/| = |p_1|/(\rho U_c^2/2)$.
By comparing the results with Inducers L, M and H-type, 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 Figure 4). 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 Fig. 4. 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 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 three maps for (a) Inducer L, (b) Inducer M and (c) Inducer H, it is easily noticed that the cavitation surge region is the widest with Inducer H in the both flow rate and NPSH ranges. Also, the whirling vortex region is widest with Inducer H. However, the suction performance at larger flow rates is again confirmed to be the best with Inducer H. All these are thought to be due to the largest inlet blade angle of Inducer H; the incidence angle is the largest for Inducer H at the same flow rate condition. Actually if we plot those maps by the shockless flow rate ratio $m$, that is the flow rate normalized by the shockless flow rate, instead of $\phi/\phi_d$, we can obtain very similar maps for Inducers L, M and H 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.
Conclusion

In the present study, the effects of design incidence angle of inducer on the suction performance and instability characteristics of turbopump are experimentally investigated. Three inducers L with the lowest design incidence angle, M with the moderate design incidence angle and H with the largest 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 H is the best 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 three 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 Inducers L, M and H. This is more apparent with the Inducer H. 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 both head and shaft power. At the large flow rate, the suction performance with Inducer L 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 all Inducers. 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 the largest with Inducer H. This is thought to be due to the largest 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 Inducers L, M and H. 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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Nomenclature

\[ A \] Cross-section area of impeller outlet [m²] \hspace{1cm} \[ U_2 \] Tip velocity of impeller [m/s]

\[ c \] Tip clearance [m] \hspace{1cm} \[ U_t \] Tip velocity of inducer [m/s]

\[ D_t \] Tip blade diameter of inducer [m] \hspace{1cm} \[ \beta_{tb,t} \] Inlet tip blade angle of inducer [deg.]

\[ D_h \] Hub diameter of inducer [m] \hspace{1cm} \[ \eta \] Efficiency \( = \rho q H / (T \omega) = \phi \psi / 2A \)

\[ F \] Normalized frequency \hspace{1cm} \[ \lambda \] Shaft power coefficient \( = T \omega / (AU_z^2) \)

\[ g \] Gravity acceleration [m²/s] \hspace{1cm} \[ \rho \] Density of water

\[ H \] Inducer head [m] \hspace{1cm} \[ \tau \] Normalized NPSH \( = NPSH / (U_z^2 / 2g) \)

\[ N \] Rotational speed of inducer [min⁻¹] \hspace{1cm} \[ \phi \] Flow coefficient \( = Q / (AU_z^2) \)

\[ NPSH \] Net positive suction head [m] \hspace{1cm} \[ \psi \] Head coefficient \( = H / (U_z^2 / 2g) \)

\[ Q \] Flow rate [m³/s] \hspace{1cm} \[ \omega \] Angular shaft speed

\[ T \] Torque [Nm]

References


