Numerical Investigation of Suction Performance of Inducer with Splitter Blade

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Abstract

For general-use turbopump inducers, high suction performance is required in a wide operating range including the cut-off point. At the low flow rates, low frequency cavitation surge is known to occur with the strong inlet backflow from the inducer. The reduced inlet blade angle would be favoured for the suppression of this inlet backflow, whereas the reduced inlet blade angle causes the deterioration of suction performance through the reduced inlet throat area. In this study, a splitter blade was adopted for the helical inducer to overcome or relieve these two conflicting problems, and the effectiveness was investigated by CFD considering cavitation. First, the favourable length and the circumferential position of short blade were investigated by 2-D cascade model. Then, the obtained suitable cascade design was applied for the 3-D helical inducer. As a result, the inlet backflow was found to be weakened at the low flow rates as expected, while keeping the good suction performance of inducer in the whole flow rate range.

Keywords: Inducer, Splitter blade, Inlet backflow, Cavitation, CFD

1. Introduction

In turbopumps, miniaturization and high power density are attempted by increasing the rotational speed. However, with high rotational speed, cavitation often occurs, resulting in various problems such as deteriorations of head and efficiency, machine vibration, erosion, etc. [1]. Installing an inducer upstream of main impeller is effective way to improve the suction performance of turbopumps. However, various kinds of cavitation instabilities such as rotating cavitation and cavitation surge are known to occur in cavitating inducers [2]-[4]. To clarify the occurrence mechanism and 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 cavitation surge with strong inlet backflow often occurs in the low flow rate range [9], [10], and it is hardly avoided because of its nature without suppression devices. In our previous studies, we have tried to suppress the cavitation surge by installing the obstruction plate [11] or the reduced-diameter suction conduit [12] upstream of helical inducers. It has been found that the suppression of inlet backflow is effective in weakening/suppressing cavitation surge. The suction performance is also improved in some cases. 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 methods with the inducer having blade shape suitable for reducing the inlet backflow.

From the view point of the inducer blade design, there exist several studies investigating the geometry effect of inducer on the cavitation instabilities [13], [14] as well as on the suction performance [15]. The reduced inlet blade angle seems favourable for the suppression of inlet backflow as well as of cavitation surge at low flow rates. However, the inlet throat area would be reduced by the reduced inlet blade angle, which may lead to the deterioration of suction performance. Actually, in our previous study where the effects of the design incidence angle of inducer on the suction performance and the cavitation instabilities were experimentally investigated [15], the operation range with the occurrence of cavitation surge was narrow in the inducer with small inlet blade angle, whereas the suction performance was not good at high flow rates. Therefore, it is important to find some methodology to overcome or relieve these two conflicting problems to improve the overall performance of inducer in the wide flow rate range.

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In the present study, a splitter blade is adopted for the helical inducer, and its effectiveness is investigated by CFD (Computational Fluid Dynamics) simulation considering cavitation. First, the favourable length and circumferential position of the short blade are investigated by 2-D cascade model, focusing on the suction performance at the higher flow rates. Then, the obtained suitable cascade design is applied for the 3-D helical inducer, and the effectiveness of the splitter blade design is examined by CFD in terms of suction performance as well as of inlet backflow at the low flow rates.

2. 2-D Cascade Model

2.1 Splitter Blade Design

As shown later, we are going to improve the suction performance of an existing flat-plate helical inducer with the tip and hub diameters of 64 mm and 30 mm respectively by applying the splitter cascade design. By referring to the geometrical parameters at the tip of this inducer, we have chosen the flat-plate cascade with the solidity of 2.0, the blade angle of 11.25 deg. and the number of blade of 4 for the baseline cascade. From this cascade, the splitter cascade examined here employs the smaller inlet blade angle of 7.65 deg., aiming at reducing the inlet backflow. Since the low blade loading near the leading edge is known to be effective for the better suction performance, the camber line has a flat shape for the former half part of the blade chord and a circular arc for the aft half part. The outlet blade angle is determined through the parametric study using numerical analysis so that it has the same head as that of the baseline flat-plate cascade at the best efficiency point. In the splitter design, one of every two blades is alternatively shortened from its leading edge. The chord length of the short blade \( c_{\text{short}} \) against that of the original long blade \( c_{\text{long}} \) is set to be \( c_{\text{short}}/c_{\text{long}}=0.50 \) and 0.75 to examine the effect of the length of short blade on the cascade performance. We also examine the effect of the circumferential position of short blades. To do so, the translated distance of short blade from the equal pitch position toward the suction side of long blade \( \theta_s \) is introduced. Assuming that equalizing the ratio of overlap length to the blade pitch for two blade passages gives the equal friction loss, we set the maximum translational distance \( \theta_{s,\text{max}} \) as 0.41 and 0.30 for \( c_{\text{short}}/c_{\text{long}}=0.50 \) and 0.75 respectively. The effect of short blade position is finally examined by setting \( \theta_s/\theta_{s,\text{max}}=0 \) (equal pitch), 1/3, 2/3 and 1.0 (the maximum translation).

2.2 Computational Method

A commercial CFD software ANSYS-CFX ver. 18.0 is used for the numerical simulation. Steady Reynolds-Averaged Navier-Stokes (RANS) equations are solved with employing \( k-\omega \) SST (Shear Stress Transport) model for the turbulence closure. Working fluid is water. Figure 1 shows an example of (a) computational domain and (b) computational grid near the leading edge. The inlet and outlet boundaries are located at 40 times pitch upstream and 24 times pitch downstream of the cascade. The computational domain includes 4 blades, and periodic boundary conditions are defined. The height of computational domain is 1 mm, and 3 grid points are located in the height direction to realize two dimensional flow condition. The hexahedral mesh is used and the number of computational nodes is about 470 thousands for each model. Averaged non-dimensional wall distance of the first node \( y^+ \) is about 0.6. As the boundary conditions, the velocity components in the relative frame are specified at the inlet boundary, while the static pressure is kept constant at the outlet boundary. On the blade walls, non-slip wall conditions are given.

For the evaluation of the suction performance, the homogeneous cavitation model, Zwart-Gerber-Belamri model [16], considering the dynamics of cavitation bubbles with a simplified Rayleigh-Plesset equation is employed. The default values of the model constants given in ANSYS-CFX are used. The static pressure at the outlet boundary is decreased step-by-step to reduce the suction pressure, and at each NPSH (Net Positive Suction Head) condition, the performance is evaluated.

Since the inlet backflow which appears at low flow rates cannot be simulated by this two-dimensional model, we herein focus on the suction performance at the best efficiency point (BEP) of the baseline flat plate cascade. We attempt to find out the appropriate splitter cascade design with employing the low inlet blade angle.

2.3 Results and Discussion

Table 1 summarizes the simulated head and efficiency for the splitter cascades at BEP of the baseline flat-plate cascade (\( \phi=0.121 \) \( \phi=V_o/U_t \), \( V_o \): axial velocity, \( U_t \): cascade moving velocity) in non-cavitating conditions. The head and the efficiency are normalized by those of the baseline flat-plate cascade for comparison. The head \( H \) is evaluated by the total pressure difference of inlet and outlet of the computational domain. The efficiency is evaluated by \( \eta=\rho g V_o^2 A h H / U_t \sum_{i=1}^{N} F_i \) (\( h \): pitch of cascade, \( F_i \): ith blade’s force in the cascade moving direction). By simply having a small inlet blade angle \( (c_{\text{short}}=c_{\text{long}}) \), the efficiency is slightly decreased form that of the baseline cascade. However, the head and efficiency of all splitter cascades are higher than those of the baseline cascade. For the

<table>
<thead>
<tr>
<th>( \theta_s/\theta_{s,\text{max}} )</th>
<th>( C_{\text{short}}=C_{\text{long}} )</th>
<th>( C_{\text{short}}=0.75C_{\text{long}} )</th>
<th>( C_{\text{short}}=0.50C_{\text{long}} )</th>
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<td>1.07/1.02</td>
<td>1.06/1.03</td>
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<td>0.121</td>
<td>1.05/1.02</td>
<td>1.09/1.06</td>
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<tr>
<td>0.25</td>
<td>1.07/1.03</td>
<td>1.10/1.05</td>
<td></td>
</tr>
<tr>
<td>0.50</td>
<td>1.07/1.03</td>
<td>1.10/1.04</td>
<td></td>
</tr>
<tr>
<td>0.75</td>
<td>1.07/1.03</td>
<td>1.10/1.04</td>
<td></td>
</tr>
<tr>
<td>1.0</td>
<td>1.07/1.03</td>
<td>1.10/1.04</td>
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</tr>
</tbody>
</table>

Fig. 1 Computational domain for 2-D cascade model
splitter cascades, the head and the efficiency of $C_{short} = 0.50C_{Long}$ is higher than of $C_{short} = 0.75C_{Long}$. This seems to be due to the reduction of wall friction loss by the decrement of the overlap length for $C_{short} = 0.50C_{Long}$. For $C_{short} = 0.75C_{Long}$, the circumferential location of the short blade does not affect the hydraulic performance. On the other hand, the efficiency becomes higher by translating the short blade for $C_{short} = 0.50C_{Long}$.

Before evaluating the suction performance, the static pressure in the blade passage is evaluated since the cavitation is expected to occur in local low pressure regions. Figures 2 and 3 respectively show the pressure distribution and the relative velocity distribution at $\phi = 0.121$. The pressure and velocity are plotted in the form of pressure coefficient $C_p = (p - \bar{p}_{in})/0.5\rho U_t^2$, $\bar{p}_{in}$: area averaged pressure of inlet, $\rho$: density of water) and normalized relative velocity $w/U_t$ in the frame moving with the cascade. In the baseline flat-plate cascade, the low pressure region is limited on the suction side near the leading edge. On the other hand, for the low inlet blade angle $(C_{short} = C_{Long}, \text{Fig. 2 (b)})$, the remarkable low pressure region appears inside the blade passage. Since the blade thickness is the same for all examined cascades, the effective blockage is larger and the throat is narrower with the small inlet blade angle. As a result, the relative velocity is high in the blade passage as shown in Fig. 3(b), which is the reason for the low pressure there. By applying the splitter blade to this cascade with small inlet blade angle, the low pressure region on the pressure side of short blade is relaxed, while the low pressure and high relative velocity region still exists on the suction side passage of short blade as shown in Fig. 2 (c) and Fig. 3 (c) ($C_{short} = 0.50C_{Long}$, equal pitch). However, by translating the short blade toward the suction side of long blade, this low pressure region with high velocity is further relaxed as can be seen in Figs. 2 and 3 (d)-(f). Finally, the smooth pressure distributions are obtained with $\theta_s = \theta_{s, max}$ for $C_{short} = 0.50C_{Long}$ (Fig. 2 (f)) and with $\theta_s = 2\theta_{s, max}/3$ for $C_{short} = 0.75C_{Long}$ (Fig. 2 (g)). The relaxation of high velocity region seems also to be responsible for the increase of efficiency as has been shown in Table 1, because of the reduction of friction loss there.

![Fig. 2 Static pressure distribution at $\phi = 0.121$](image1)

![Fig. 3 Relative velocity distribution at $\phi = 0.121$](image2)

From the above analysis, it is expected that the splitter cascades with $C_{short} = 0.50C_{Long}$ and $\theta_s = \theta_{s, max}$, and $C_{short} = 0.75C_{Long}$, $\theta_s = 2\theta_{s, max}/3$ can achieve the better suction performance since the low pressure region in the blade passage is well relaxed. Therefore, in the followings, we conduct the numerical simulation considering cavitation for these two splitter cascades as well as for the baseline flat-plate cascade and the non-splitter cascade $(C_{short} = C_{Long})$ with the small inlet blade angle.

Figure 4 shows the suction performance curves at $\phi = 0.121$. Horizontal axis is the non-dimensional NPSH $\tau = (\bar{p}_{in} - p_c)/(\rho U_t^2)$, $\bar{p}_{in}$: mass flow averaged total pressure at inlet. Vertical axis is the normalized head coefficient $\psi/\psi_{noncav}$. Figure 5 shows...
the cavity distribution at NPSH with $\psi/\psi_{noncav}=0.90$. In the case of the baseline flat-plate cascade, the head gradually drops with the decrease of NPSH. The cavitation is not well developed at 10% head drop as can be seen in Fig. 5(a), which indicates that even short partial cavitation deteriorates the head performance in this cascade. On the other hand, the head suddenly drops at high NPSH ($\tau=0.14$) for the non-splitter cascade ($C_{short}=C_{Long}$) with the small inlet blade angle. The cavitation forms along the pressure side of blades as shown in Fig. 5(b), which well agrees with the observed low pressure region in Fig. 2(b). The negative incidence with the increased blockage due to the small inlet blade angle is the main reason for this. By applying the splitter blade to this cascade with small inlet blade angle, the suction performance is significantly improved, and the suction performance of $C_{short}=0.50C_{Long}$ is better than that of $C_{short}=0.75C_{Long}$. From Figs. 5(c) and (d), it can be found that the cavitation forms mainly on the suction sides of both short and long blades in the case with $C_{short}=0.50C_{Long}$, while on the pressure sides in the case with $C_{short}=0.75C_{Long}$. Although the cavitation forms also on the pressure side of long blade in the case with $C_{short}=0.50C_{Long}$, the blockage effect seems not very significant, since the blade passage is wide because of its design with the large translation of short blade toward the pressure side of long blade. As a result, the splitter cascade with more shortened blade $C_{short}=0.50C_{Long}$ shows the better suction performance than the others examined here. Therefore, we decide to apply this cascade shape to the 3-D helical inducer. Since, in the two-dimensional cascade model, any three-dimensional effects such as the tip clearance flow, the inlet backflow and any other secondary flows, are not considered, it is important to re-examine the effect of splitter cascade for the three-dimensional inducer. We conduct the three-dimensional simulation and the results are discussed in the next section.

Fig. 4 Suction performance curves at $\phi=0.121$

3. 3-D Helical Inducer

3.1 Splitter Inducer

In order to examine the effectiveness of the splitter blade against the suction performance and the inlet backflow in inducer, the obtained geometry with $C_{short}=0.50C_{Long}$ and $\theta_s=\theta_s,max$ is adopted for the tip in a helical inducer. The number of blades is four. The radial distribution of blade angle is specified as the helical condition ($r \tan \beta(r) = r_{1} \tan \beta_{1}$). The hub-to-tip ratio is constant from the inlet to the outlet. To clarify the effect of splitter blades on the suction performance and the inlet backflow, the baseline flat-plate inducer and the inducer with the small inlet blade angle (called “non-splitter inducer”) are also designed. Figure 6 and Table 2 show the schematic view and the specifications of inducers.

![Schematic view of inducers](image)

**Table 2 Specifications of inducers**

<table>
<thead>
<tr>
<th>Inducer type</th>
<th>Flat-plate</th>
<th>Non-splitter</th>
<th>Splitter</th>
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<tr>
<td>Tip diameter, $D_t$[mm]</td>
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<td>64</td>
<td>64</td>
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<tr>
<td>Hub diameter, $D_h$[mm]</td>
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<td>30</td>
<td>30</td>
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<tr>
<td>Tip clearance, $c$[mm]</td>
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<td>0.5</td>
<td>0.5</td>
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<tr>
<td>Number of blades, $Z$</td>
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<td>4</td>
<td>Long 2</td>
</tr>
<tr>
<td>Inlet tip blade angle, $\beta_{t1}$[deg.]</td>
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<td>7.65</td>
<td>7.65</td>
</tr>
<tr>
<td>Outlet tip blade angle, $\beta_{t2}$[deg.]</td>
<td>11.25</td>
<td>15.00</td>
<td>15.00</td>
</tr>
<tr>
<td>Tip blade thickness, $t_t$[mm]</td>
<td>1.00</td>
<td>1.00</td>
<td>1.00</td>
</tr>
</tbody>
</table>
3.2 Computational Method

Figure 7 shows a computational model of the splitter inducer. The inlet and outlet boundaries are located at 10\(D_t\) upstream and 6\(D_t\) downstream of the leading edge of inducer, where \(D_t\) is the tip diameter. The computational domain includes two passages between short and long blades in the case of splitter inducer, while only one passage in the other inducer cases. The rotational periodic boundary condition is applied. The hub diameter is enlarged near the outlet boundary to suppress the backflow at the outlet boundary. Hexahedral mesh is used and the number of computational nodes is about 8 millions in the case of splitter inducer, while 4 millions in the other inducer cases. Sixteen grid points are located radially in the tip clearance to well capture the tip leakage flow, which is important for the formation of inlet backflow. Averaged \(y^+\) on the blade surface is about 0.7. The grid dependency has been examined for the performance and the amount of inlet backflow in the non-splitter inducer as described later. The flow in the suction pipe is simulated in the stationary frame while the flow in the inducer and the downstream pipe is in the rotating frame. The rotational speed of inducer is set to be \(N=5000\) min\(^{-1}\). As the boundary conditions, the mass flow rate is specified at the inlet boundary while the constant static pressure at the outlet boundary. On the blade walls, the non-slip wall condition is given.

The suction performance is evaluated in the same way as in the cascade simulation. The cavitation model implemented in ANSYS CFX is used. The static pressure at the outlet boundary is decreased step-by-step to reduce the suction pressure, and at each NPSH (Net Positive Suction Head) condition, the performance is evaluated.

3.3 Grid dependency

The grid dependency check has been carried out using four types of grids, fine, medium, coarse and very coarse grids, for the non-cavitating flow in the non-splitter inducer, as shown in Table 3. The head coefficient \(\psi\), the power coefficient \(\lambda\), the efficiency \(\eta\) and the inlet backflow rate \(Q_{\text{back}}\) at 1\(D_t\) upstream of the inducer at the low flow rates of \(\phi=0.013\) and 0.067 (best efficiency point) are summarized after normalized by those obtained using fine grid. The definitions of the coefficients are explained later. It is found that the hydraulic performance can be well calculated even with the very coarse grid, whereas the quantitative prediction of the inlet backflow requires the finer grid. Since the evaluation of inlet backflow is important in the present study, we have finally decided to use the fine grid in the following simulations.

![Fig. 7 Computational domain and grids for the splitter inducer](image)

<table>
<thead>
<tr>
<th>Grid Type</th>
<th>Number of nodes</th>
<th>(\psi)</th>
<th>(\lambda)</th>
<th>(\eta)</th>
<th>(Q_{\text{back}})</th>
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<tr>
<td>Fine</td>
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<tr>
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<td>569,632</td>
<td>1.01</td>
<td>1.01</td>
<td>1.00</td>
<td>0.988</td>
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</tbody>
</table>

3.4 Results and discussion

Figure 8 shows the hydraulic performance curves. Horizontal axis, left vertical axis and right vertical axis in Fig. 8 (a) is the flow coefficient \(\phi=(Q/\Delta U_t)\), \(Q\): flow rate, \(A=\pi(D_{t1}^2 - D_{t2}^2)/4\), \(D_{t1}\): inner diameter of the casing, \(D_{t2}\): hub diameter), the head coefficient \(\psi=(\bar{p}_{\text{out}} - \bar{p}_{\text{in}})/\rho U_t^2\), \(\bar{p}_{\text{out}}\) and \(\bar{p}_{\text{in}}\): mass flow averaged total pressure at the outlet and inlet boundaries) and the efficiency \(\eta=(\bar{p}_{\text{out}} - \bar{p}_{\text{in}})Q/(2\pi NT/60)\), \(T\): torque of the blades respectively. The vertical axis in Fig. 8 (b) is the power coefficient \(\lambda=(\Delta Tw)/\rho U_t^2\). By comparing the non-splitter inducer with the flat-plate inducer, it is found that the efficiency of the non-splitter inducer is lower in \(\phi>0.080\). This seems to be due to the increment of friction loss by the increased blockage with the small inlet blade angle;
increased flow velocity due to the increased blockage is responsible for the friction loss generation. The power coefficient of the non-splitter inducer is lower than that of the flat-plate inducer in the whole flow rate range; this indicates the theoretical head of the non-splitter inducer is lower. As a result, the head of the non-splitter inducer is lower than that of the flat-plate inducer in $\phi>0.067$. On the other hand, the efficiency of the non-splitter inducer is higher than that of the flat-plate inducer in $\phi<0.080$. This seems to be due to the decrement of the shock loss by the decreased inlet blade angle. By comparing the splitter inducer with the non-splitter inducer, it is found that the efficiency of the splitter inducer is higher in $\phi>0.067$. This seems to be due to the decrement of friction loss by the decrement of the overlap length. The power coefficient of the splitter inducer is almost the same as that of the non-splitter inducer in the whole flow rate range; this indicates the theoretical head of the splitter inducer is almost the same. As a result, the head of the splitter inducer is higher than that of the non-splitter inducer in $\phi>0.067$.

Figure 9 shows the relationship between the operating flow rate and the inlet backflow rate. The backflow rate is evaluated at $1D_t$ upstream from the leading edge of the inducer. By comparing the non-splitter inducer with the flat-plate inducer, it is found that the inlet backflow is weakened in the non-splitter inducer, which indicates that the small inlet blade angle is effective for the suppression of inlet backflow through the reduction of blade loading. In the splitter inducer, the effect of small inlet blade angle is still available; the inlet backflow rate of the splitter inducer is smaller than that of the flat-plate inducer.

Figure 10 shows the suction performance curves of the non-splitter and splitter inducers at $\phi=0.094$, the best efficiency point (BEP) of the splitter inducer. It is clearly found that the suction performance of the splitter inducer is much better than that of the non-splitter inducer. Figure 11 depicts the cavitation region by the isosurface of vapor void fraction $\alpha_v=0.1$ at the normalized NPSH of $\tau=gNPSH/U_t^2=0.067$ and 0.068, where the 20% head drop occurs in the non-splitter inducer. It is clearly seen that the cavity develops significantly into the blade passage and even on the pressure side in the non-splitter inducer, while cavitation in the splitter inducer is limited. The mechanism seems to be similar to the case in the two-dimensional cascade. Actually, Figs. 12 and 13 show the pressure and relative velocity distributions in the tip region ($r=31.5$ mm) at the non-cavitating condition, where the pressure coefficient $C_p=(p-p_{in})/0.5\rho U_t^2$ and the normalized relative velocity $w/U_t$ in the rotating frame are plotted. The low pressure region with the high relative velocity is clearly seen inside the blade passages of the non-splitter inducer. This is the reason why the cavity extends rapidly in the non-splitter inducer.
Figure 14 shows the suction performance curves of the non-splitter and splitter inducers at the medium flow rate of $\phi = 0.067$, that is the BEP of the non-splitter inducer. At this flow rate, the suction performance of the non-splitter inducer becomes better. It is interesting to see that the head decreases more sharply in the non-splitter inducer while gradually in the splitter inducer. Figure 15 depicts the cavitation region at the normalized NPSH of $\tau = 0.020-22$, where roughly 10-20 % head drop occurs in the inducers. It is seen that the cavitation develops into the all blade passages in the non-splitter inducer. On the contrary, in the splitter inducer, the cavitation develops mostly on the suction side of long blade, and the suction side passage of short blade is almost free from cavitation. It seems that this flow passage still works well during the head drop process, resulting in the gradual head decrease in the splitter inducer.
Figure 16 shows the suction performance curves of the non-splitter and splitter inducers at the low flows rate of $\phi = 0.013$. It is seen that the suction performance at this flow rate is similar in the splitter and non-splitter inducers; the head gradually decreases against the decrease of NPSH. Figure 17 depicts the cavitation region at the normalized NPSH of $\tau = 0.028$ and 0.022, where roughly 10% head drop occurs. In the splitter inducer, the cavitation significantly develops on the suction side of long blade, and it extends into the blade passage along the hub wall. On the other hand, the suction side passage of short blade is almost free from cavitation as similar to the middle flow rate case shown in Fig. 15(a). Small cavity appearing at the inlet hub is caused by the reduced pressure there due to the centrifugal force of swirling back flow, while the impact of this cavity at the hub wall is considered to be small on the head performance. Therefore, it seems that the head gradually decreases in the similar mechanism to that at the middle flow rate. On the other hand, in the non-splitter inducer, the cavitation does not yet extend into the blade passage at this NPSH condition. The mechanism of gradual head drop seems to be different from that in the splitter inducer.

4. Conclusion

In the present study, the splitter blade is adopted to the helical inducer and its effect on the suction performance and the inlet backflow were investigated by the numerical simulation. At first, the favourable length and circumferential position of the short blade were investigated by 2-D cascade model, focusing on the suction performance at higher flow rates. The results are summarized as follows.

1. The splitter cascade was effective for the better hydraulic performance. By translating the short blade circumferentially toward the suction side of long blade, the head and the efficiency were slightly but further improved. This seemed to be because the high relative velocity inside the suction side blade passage of short blade was relaxed, reducing the passage friction loss.

2. The suction performance of the splitter cascade was higher than that of the non-splitter cascade. The splitter cascade with the short blade with the chord length half of long blade and with the large circumferential translation showed the better suction performance. It seemed that the relaxed low pressure region with the reduced relative velocity in the blade passage was responsible for the improvement of suction performance.

3. Then, the obtained suitable cascade design was applied for the 3-D helical inducer, and the effectiveness of the splitter blade design was examined by numerical simulation in the wide flow rate range. The results are summarized as follows.

4. The head and the efficiency of the splitter inducer were higher than those of the non-splitter inducer. It seemed to be due to the decrement of friction loss by the decrement of the overlap length.

5. The inlet backflow was weakened in the non-splitter and the splitter inducers. This was achieved simply by reducing the inlet blade angle.

6. The suction performance was well improved at the high flow rates by relaxing the low pressure region with high relative velocity in the blade passage of the splitter inducer.

7. In this study, the proposed splitter inducer design was applied only for a helical inducer with straight hub which is known to be low-head type inducer. The results obtained by the present numerical simulations clearly indicated the effectiveness of proposed splitter design, while the experimental validation should be made for more quantitative evaluation. In addition, for modern design of inducer, the tapered hub is often applied. The swept-back blade with wedge-shaped leading edge is often employed. Those encourage us for further studies to obtain the design guideline of inducer for general-use turbopumps.

Acknowledgments

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Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$A$</td>
<td>Cross-sectional area [$m^2$] ($=\pi(D_c^2-D_h^2)/4$)</td>
</tr>
<tr>
<td>$c$</td>
<td>Tip clearance [$m$]</td>
</tr>
<tr>
<td>$C_{\text{Long}}$</td>
<td>Chord length of long blade [$m$]</td>
</tr>
<tr>
<td>$C_{\text{Short}}$</td>
<td>Chord length of short blade [$m$]</td>
</tr>
<tr>
<td>$r_t$</td>
<td>Tip radius [$m$]</td>
</tr>
<tr>
<td>$t_t$</td>
<td>Tip blade thickness [$m$]</td>
</tr>
<tr>
<td>$T$</td>
<td>Torque of inducer [$Nm$]</td>
</tr>
<tr>
<td>$U_t$</td>
<td>Tip speed/cascade moving speed [$m/s$]</td>
</tr>
</tbody>
</table>

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References


