CAVITATION PERFORMANCE OF A CENTRIFUGAL IMPELLER SUITABLE FOR A MINI TURBO-PUMP

Xianwu Luo
Kyushu Institute of Technology, Kitakyushu, Japan
a584105k@tobata.isc.kyutech.ac.jp

Michihiro Nishi
Kyushu Institute of Technology, Kitakyushu, Japan
nishimech.kyutech.ac.jp

Kouichi Yoshida
Kyushu Institute of Technology
yoshida@mech.kyutech.ac.jp

Hitoshi Dohzono
Kyushu Institute of Technology
b104303h@tobata.isc.kyutech.ac.jp

Ken Miura,
Kyushu Institute of Technology
Toshiba Carrier Co., Ltd, Japan
ken1.miura@glb.toshiba.co.jp

ABSTRACT

A mini turbo-pump, which is defined as the size having its impeller diameter between 5 mm and 50 mm by the authors, is needed for cooling small electronic devices and fuel cells, etc. Since the mini pump will act as a key part in these systems, the hydraulic performance including cavitation performance is of great importance. The authors’ group has demonstrated that an impeller having more vanes than the conventional is favorable for a mini pump, using a 34 mm dia. semi-open impeller. In the present study, the proposed design concept is examined whether it is appropriate from the viewpoint of cavitation performance. The tests are conducted under the rotational speed of 11,000 rpm using two impellers called as Imp. B and Imp. C. The former has the conventional shape of 36 mm dia. with five vanes and the latter has the proposed shape of 34 mm dia. with 12 vanes. As we see experimental results are reasonably predicted by the turbulent flow analysis with VOF model, we investigate numerically the cavitation performance of a modified impeller of Imp. C, which consists of six original long vanes and six splitter vanes to reduce the inlet blockage due to vane thickness.

Keywords: cavitation performance, load ratio, turbulent flow, inlet geometry, centrifugal impeller, mini turbo-pump.

1 INTRODUCTION

Recently, new needs for mini turbo-pumps, which are defined as the size having impeller diameter between 5 mm and 50 mm by the authors, are recognized in various social areas. For example, a mini turbo-pump is expected to use as heat controlling systems for small electric motors and other electric devices, fuel cells, etc. Under these conditions, since a mini turbo-pump will act as a key part in those systems, both high performance and stable operation are necessary. If the liquid temperature rise is high, cavitation is much easier to occur, and unfavorable problems such as deterioration of hydraulic performance, noise and vibration, erosion, etc. are used to arise.

However, a review of available literatures shows that rather a few researches have treated a mini turbo-pump [1], and few study was made to clarify its cavitation performance [2]. In this paper, we have investigated cavitation performances of two mini-pump impellers experimentally, where one is designed by the proposed concepts suitable for a mini turbo-pump [1], and the other is designed by the conventional method. To discuss the results, the cavitation performances of both impellers are predicted by the turbulent flow analysis. A further study on the cavitation performance for the proposed impeller is made to see the effect of the inlet geometry numerically.

NOMENCLATURE

- $b$: blade width (m)
- $BF$: blockage factor, see Eq.(5)
- $c$: axial tip clearance between impeller tip and casing (m)
- $D$: impeller diameter (m)
- $g$: gravitational acceleration (m/s$^2$)
- $H$: head rise (m)
- $L$: curvature length of vane profile (m)
- $n$: rotational speed (1/min)
- $NPSH$: net positive suction head (m)
- $P$: absolute static pressure (pa)
- $Q$: flow rate ($m^3$/min)
- $r$: radius (m)
- $S$: suction specific speed ($m, m^3/min, rpm$)
- $Z$: number of blade
- $\delta$: blade thickness (m)
- $\rho$: density of water (kg/m$^3$)
- $\lambda$: clearance ratio
- $\sigma$: Thoma's cavitation number
- $\phi$: flow coefficient
- $\psi$: head coefficient
- $\tau$: input power coefficient

Subscripts

- $0$: reference section
- $1$: impeller inlet
- $2$: impeller exit
- $cal$: calculated value
- $cr$: critical value
- $exp$: experimental value
- $imp$: impeller
- $in$: internal (efficiency) or input (power)
- $r$: radial component
- $u$: peripheral component
\( \gamma \): vapor

\( z \): axial component

2 TEST RIG AND EXPERIMENTAL METHOD

2.1 Test impellers

Fig. 1 shows two impellers prepared for the cavitation tests. Imp. C, which is shown in Fig. 1(a), is designed with a proposed concept suitable for a mini pump \(^1\), while Imp. B, which is shown in Fig. 1(b), is designed by the conventional method. The former has the outer diameter of 34 mm and the latter has that of 36 mm. As the both are semi-open impellers, there is an axial clearance between the blade tip of an impeller and the pump front cover. The clearance at the impeller exit of 0.2 mm is used in this paper. Their geometrical parameters are shown in Table 1.

![Fig. 1 Test impellers](a) Imp. C (b) Imp. B)

Table 1 Specification of Test Impellers

<table>
<thead>
<tr>
<th>Impeller type</th>
<th>C</th>
<th>B</th>
</tr>
</thead>
<tbody>
<tr>
<td>Outlet diameter ( D_2 ) (mm)</td>
<td>34.0</td>
<td>36.0</td>
</tr>
<tr>
<td>Inlet diameter at hub ( D_{in} ) (mm)</td>
<td>9.1</td>
<td>18.0</td>
</tr>
<tr>
<td>Inlet diameter at tip ( D_{it} ) (mm)</td>
<td>22.0</td>
<td>25.0</td>
</tr>
<tr>
<td>Outlet width ( b_2 ) (mm)</td>
<td>2.9</td>
<td>4.4</td>
</tr>
<tr>
<td>Blade inlet angle at hub ( \beta_{in} ) (°)</td>
<td>63.8</td>
<td>21.5</td>
</tr>
<tr>
<td>Blade inlet angle at tip ( \beta_{it} ) (°)</td>
<td>34.0</td>
<td>31.1</td>
</tr>
<tr>
<td>Blade outlet angle ( \beta_2 ) (°)</td>
<td>60</td>
<td>22.5</td>
</tr>
<tr>
<td>Blade number ( z )</td>
<td>12</td>
<td>5</td>
</tr>
<tr>
<td>Front shroud type</td>
<td>Semi-open</td>
<td></td>
</tr>
</tbody>
</table>

2.2 Test rig and test condition

Fig. 2 shows the test system for a mini turbo-pump schematically. In order to secure the constant flow temperature, a cooling unit has been installed. An inverter motor is used so that the rotational speed of the mini pump is selectable between 1,000 rpm and 12,000 rpm, though only high rotational speed was used in this study. The static pressure at the pump inlet is adjusted by a vacuum pump, which is connected to the settling tank. The flow discharge, which is controlled by a gate valve, is measured by a magnetic flow meter. The static pressures at the inlet pipe, outlet pipe, and the impeller exit (not possible for Imp. C due to narrow space) are measured by pressure sensors. A compact torque meter is used to measure the torque input to the pump and the rotational speed.

By choosing 11,000 rpm as the test rotational speed, we investigate the hydraulic performances including cavitation performance of two mini turbo-pumps.

![Fig. 2 Test rig for a mini turbo-pump]

2.3 Parameters for pump performance

For a semi-open impeller, tip clearance is an important parameter. The following dimensionless axial tip clearance is specified at the impeller outlet.

\[
\lambda_2 = \frac{c_2}{b_2} 
\]

Net positive suction head, i.e. \( NPSH \), is one of the most important parameters to measure cavitation performance for a turbo-pump. It is defined by equation (2).

\[
NPSH = \left( P_0 - P_v \right) / \rho g + \frac{V^2}{2g} 
\]

Thoma’s cavitation number is also a notable parameter for cavitation performance. It is derived as the ratio of \( NPSH \) to pump (or impeller) head.

\[
\sigma = \frac{NPSH}{H} 
\]

Like the specific speed, which is a parameter to show the type of a rotor, suction specific speed \( S \) can represent the cavitation performance of a pump. Its definition is given by equation (4).

\[
S = \frac{n \sqrt{Q}}{NPSH_{cr}} 
\]

where \( NPSH_{cr} \) is decided as the value of \( NPSH \) when 3% drop of the pump (or impeller) head reaches.

![Fig. 3 Blockage factor for two impellers]

\[
\text{BF} = \frac{1}{1.5} 
\]

1 Motor 2 Torque meter 3 Test pump 4 Ball valve 5 Flow meter 6 Vacuum meter 7 Water level 8 Tank 9 Gate valve 10 Cooling unit 11 Thermometer

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\( n \): rotational speed

\( Q \): flow rate
As the thickness of vane is considered to affect the cavitation performance, we use the following inlet blockage factor:

\[
BF = \frac{2\pi \frac{r_i}{r_1} - Z\delta_v}{2\pi \frac{r_i}{r_1}}
\]  

(5)

These factors for both impellers in the radial direction from hub to tip are shown in Fig. 3. We see that the blockage factor is almost the same near the hub but there is difference between them near the tip region, where \( BF \) of Imp. C is a little bit larger than that of Imp. B.

3 NUMERICAL METHODS

3.1 Three-dimensional flow

We have also tried to predict the cavitation performance of the mini pump by the numerical analysis of three-dimensional turbulent flow. The calculation has been carried out based on RANS equations with k-ε model. In this study, since our concern is steady performance of the mini impellers, one impeller channel is treated assuming the inflow is axisymmetric. Fig. 4 shows the meridional section of the test pump with Imp. C. A-A1 section, which is far from the impeller inlet, is adopted as the inflow plane and F-F1 section, which is the just downstream of the vaneless diffuser exit, is assigned for the outflow plane of the computational domain. When cavitating flow was treated, the inflow plane was set at C-C1 section. For both impellers, the mesh nodes for the area from C-C1 section to F-F1 section are over 170,000, while mesh nodes for the extended suction area from A-A1 to C-C1 are over 100,000. Fig. 5 shows the mesh surface for the case of Imp. C. It is noted that this selection is acceptable from the preliminary calculation.

3.2 Cavitation model

If we investigate the cavitation performance of a pump numerically, we need to use some cavitation model at present, though the cavitation is essentially unsteady phenomena. Since every available cavitation model, which has been developed just to approximate some features of typical cavitating flows \([3]\), is regarded as imperfect, we had better confirm it by the experiment.

In this study, a VOF (Volume of Fluid) model developed on the basis of multi-component fluid is adopted. The governing equations describe the cavitation process involving two-phase (liquid and gas) three-component (liquid, vapor, and non-condensable gas in the form of micro-bubbles) system, where no-slip between two phases, and thermal equilibrium between all phases are assumed. Further, the Rayleigh-Plesset equation, which occurs as a source term at the volume-fraction scalar equation, is used to control the vapor generation or construction.

3.3 Boundary conditions and calculation conditions

The boundary conditions are as follows: (1) Total pressure is set at the inflow plane (A-A1 section, or C-C1 section in the case of cavitation prediction), (2) Mass flow-rate is given at the outflow plane (F-F1 section), (3) For those solid surfaces, the wall law is applied. Further, periodic flow condition is used at two sides of the computation domain.

A commercial code named as CFX-TASCflow has been used for the numerical analysis. The calculation is conducted under the rotational speed of 11,000 rpm, and at several flow rates around the best efficiency point for each of those impellers.
4 RESULTS AND DISCUSSION

4.1 Characteristic curves for two mini pumps

Fig. 6 shows the characteristic curves for two mini pumps. As the axial tip clearance at the outlet of both impellers is 0.2 mm, the clearance ratio \( \lambda_2 \) is 0.069 for Imp. C and 0.045 for Imp. B. The experimental results such as pump head coefficient \( \psi \), input power coefficient \( \tau_{\text{in}} \) (the loss due to mechanical seal and bearing is excluded) are plotted against flow coefficient \( \phi \). For the case of Imp. B, impeller head coefficient \( \psi_{\text{imp}} \) estimated from the measured static pressure at impeller outlet is also shown. For comparison, the numerical results such as input power and impeller head (Imp. B only) predicted near the best efficiency points are plotted in the same figure. The following features are observed in the results:

1) Characteristic curves are quite different between Imp. B and Imp. C, though they have the similar specific speed and diameter. Much larger power transmission is achieved by using the latter impeller, size of which is a little bit smaller than that of the former [1].

2) The time-average performances for both impellers are predicted satisfactorily.

3) Internal efficiency of Imp. C larger than 0.6 is observed between \( \phi =0.11 \) and \( \phi =0.25 \). This feature is qualitatively explainable from numerical results as follows: Major part of relative flow entering Imp. C doesn’t vary too much with flow coefficient in the above range due to the change of reverse flow thickness upstream of impeller inlet, which is observed in the tip region.

4.2 Cavitation performance of mini pumps

Cavitation performances of two mini pumps are shown in Fig. 7, where pump head coefficient \( \psi \) (impeller head coefficient \( \psi_{\text{imp}} \) in the case of Imp. B) and input power \( \tau_{\text{in}} \) are plotted against Thomas’s cavitation number \( \sigma \). The results of Imp. C at three operation conditions near the best efficiency point are shown in Fig. 7(a), while the result of Imp. B at the best efficiency point is shown in Fig. 7(b). For comparison, numerical results of input-power are shown in these figures. In the case of Imp. B, impeller head coefficient is also plotted. From them, we can see the following features:
1) If we use the usual criterion of the critical cavitation condition that corresponds to 3% pump head drop, the critical value $\sigma_{cr}$ at the best efficiency point is 0.185 for Imp. C, which is less than that for Imp. B ($\sigma_{cr}$ is 0.213 in the case of Imp. B). On the other hand, the suction specific speed $S$ for Imp. C is 1,178 (m, m³/min, m), and this value for Imp. B is 1,291 (m, m³/min, m). As it is said that $S$ for the usual size centrifugal pump is around 1,200~1,600 (rpm, m³/min, m), the present mini pumps have nearly the same cavitation performance as usual-size pump.

2) Thus, to select the greater number of vanes is not bad from the viewpoint of cavitation performance.

3) The critical cavitation number increases with the pump discharge similar to a usual size pump.

4) Cavitation performance of Imp. B was satisfactorily predicted by the numerical analysis. And the prediction for Imp. C was regarded as fair, though it underestimated the effect of cavitation on the hydraulic performance, as shown in Fig. 7(a).

To see the cavitating flow in each pump impeller operated at the best efficiency point near $\sigma_{cr}$, distribution of void fraction in the mid-span surface between two adjacent vanes is correlated and shown in Fig. 8 (a) and (b) using color plot. The former corresponds to Imp. C and the latter is for Imp. B. We can suspect that the cavity in the flow passages of both impellers appears on the suction surface just downstream of the leading edge. But, its development is different from each other. In the case of Imp. C, the cavity looks to grow in the downstream direction along the suction surface, while that in Imp. B tends to grow toward the center of the vane-to-vane passage from the suction surface

Each plot of velocity vector in the mid-span surface for both impellers is shown in Fig. 9 (a) and (b) respectively. we can observe the smooth through-flow in the case of Imp. C, and a large-scale separation behind the bubble cavitation in the case of Imp. B. These results imply that the performance drop will be caused by the passage blockage due to the cavity at the impeller inlet for Imp. C, and both the blockage and the separation will cause performance drop in Imp. B.
The numerical results shown in Fig. 10 are non-dimensional components of absolute velocity at H-H1 section upstream of the impeller inlet. $Vu$, $Vr$ and $Vz$ denote peripheral, radial and axial components respectively. Pre-swirl and backflow are clearly observed near tip region for both impellers operated even near the best efficiency point. This will be caused by the tip clearance at the semi-open impeller inlet, which is not regarded as small in the case of mini pump. Thus, it indicates us that the selection of inlet vane-angle should be made considering the incoming flow. In the present case, we know that the velocity distribution for Imp. C is more uniform than that for Imp. B from Fig. 10. This flow condition will be favorable for the cavitation performance of Imp. C, as local pressure drop causes the cavitation, even though many vanes are used.

4 CONCLUSION

From the present investigation, the following can be concluded:

1) The test mini turbo-pumps have nearly equal cavitation performance to a usual size centrifugal pump.

2) The impeller having many vanes like twelve, which is suitable for a mini pump has comparable cavitation performance to a conventional impeller having a few vanes.

3) The cavitation performance for the conventional impeller near the best efficiency point is predicted satisfactory by the numerical analysis, which is based on RANS equations with k-ε model and VOF model. But, the prediction for the proposed impeller is fair. Further investigation on the numerical method is still necessary for the general application.

REFERENCE


2 Luo, X., Nishi, M., and et al, 2002, Cavitation Performance of a High-Speed Mini Turbo-Pump, 4th International Conference on Pumps and Fans, E1, Tsinghua University, Beijing


Table 2 Mean unit loads for two test impellers

<table>
<thead>
<tr>
<th>Impeller type</th>
<th>C</th>
<th>B</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length of vane on mid-span $L/D_2$</td>
<td>0.479</td>
<td>0.512</td>
</tr>
<tr>
<td>Number of vane $Z$</td>
<td>12</td>
<td>5</td>
</tr>
<tr>
<td>Load of each vane $τ_{in}/Z$</td>
<td>0.0258</td>
<td>0.0173</td>
</tr>
<tr>
<td>Mean unit loads $(τ_{in}/Z)/(L/D_2)$</td>
<td>0.0538</td>
<td>0.0338</td>
</tr>
</tbody>
</table>

Fig. 10 Absolute flow distribution upstream of the impeller inlet (at H-H1 section) (Imp. C: $λ_2 =0.069$, $φ =0.202$, imp. B: $λ_2 =0.045$, $φ =0.089$)