Cavitation in a Shaft-less Double Suction Centrifugal Miniature Pump

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Abstract

Based on the consideration that the cavitation would affect the operation stability of miniature pumps, the 3-D turbulent cavitating flow in a test pump was simulated by using a mixed cavitation model and k-ω SST turbulence model. In order to investigate the influence of inlet geometry parameters on the cavitation performance of the miniature pump, two more impellers are designed for comparison. Based on the results, the following conclusions are drawn: 1) Cavitation performance of the double suction shaft-less miniature pump having different impeller is equivalent to the centrifugal pump having ordinary size, though the flow passage at impeller inlet is small; 2) The miniature pump having radial impeller can produce much higher pump head, but lower cavitation performance than that having the impeller based on the conventional design method; 3) It is believed that by applying the double suction design, the miniature pump achieved relatively uniform flow pattern upstream the impeller inlet, which is favorable for improving cavitation performance.

Keywords: double suction, shaft-less, miniature pump, cavitation, numerical simulation

1. Introduction

With the development of lighter and smaller fluid machines, many kinds of miniature pumps are used in the important systems. The technical progress in those social applications shows the direction for improving the operating security and stability of miniature pumps. Blood pumps, as a typical usage in the miniature pump field, the operating stability depends on not only the hydraulic performance, but also the cavitation performance. Generally, the cavitation is harmful for fluid machines. For example, the cavitation in the hydro turbine induces the instability. When cavitation becomes serious, it causes the performance drop, and induces the noise, vibration, and the damage of the material in the flow passage[1]. There is some bubble nucleation in the human being blood. When the nuclei pass the flow field with low pressure, the cavity may form and cavitation may occur in the blood pump. This situation is very harmful for the case of a blood pump. Further, for the application of cooling systems for large integrated circuits and fuel cells, cavitation performance in miniature pumps is also very important.
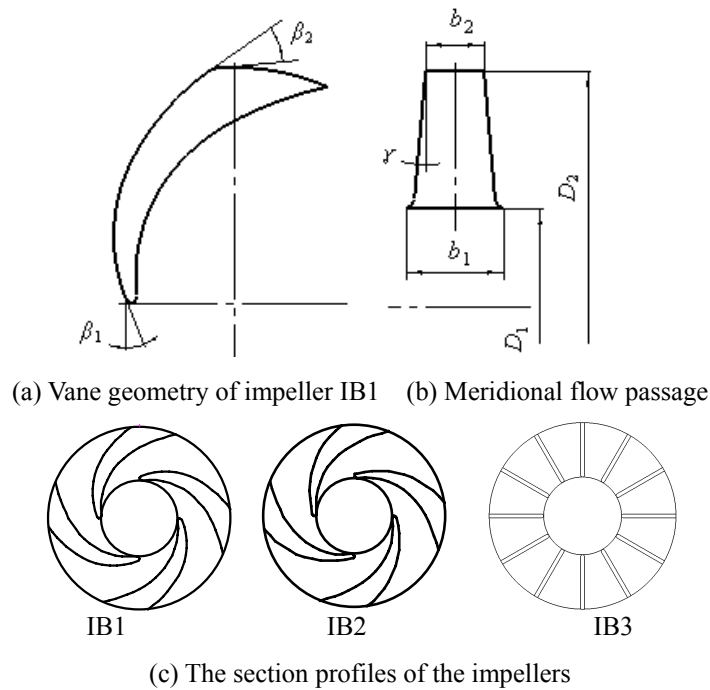
In recent years, numerical simulation technology develops rapidly with the computer technology and computational fluid dynamics (CFD). However, the numerical simulation which focuses few on the cavitation in miniature pumps[2-4]. In this paper, three impellers are simulated to discuss the influence of geometry parameters on cavitation performance for a miniature pump.

2. Miniature Pump Design

Three double suction closed-type impellers i.e. IB1, IB2, and IB3 are designed. The double suction structure is helpful to form the symmetrical flow in a miniature pump, and is favorable for balancing the axial force of the rotor. The geometrical parameters of those impellers are shown in Table.1. The vane geometry of impeller IB1 is shown in Fig.1(a) and (b). Besides, Fig.1(c) shows the meridional flow passage of three impellers.

Table 1 Geometrical parameters of pump impellers

Impeller		IB1	IB2	IB3
Diameter of impeller exit	D_2 (mm)	24	24	24
Diameter of impeller inlet	D_1 (mm)	10	10	10
Vane width at trailing edge	b_2 (mm)	3	3	3
Vane width at leading edge	b_1 (mm)	5	5	5
Vane angle at trailing edge	β_2 (°)	29	29	90
Vane angle at leading edge	β_1 (°)	22	12	90
Wrap angle of a vane	φ (°)	85	82	0
Number of vane	Z	4	4	12
Blade shape		curved	curved	straight

(a) Vane geometry of impeller IB1   (b) Meridional flow passage

IB1   IB2   IB3

(c) The section profiles of the impellers

Fig. 1 Configuration of double suction impeller

For comparison, two more impellers are designed based on the impeller IB1 for the miniature pump. Impeller IB2 has smaller vane angle at leading edge than that of impeller IB1 by 10°. Note that the vane width at leading edge and trailing edge, vane angle at trailing edge, vane wrap angle and vane number are the same for two impellers. Impeller IB3 has 12 radial vanes for the purpose of obtaining higher pumping head with the same impeller size.

3. Simulation Methods

The steady 3-D turbulent cavitating flow in the miniature pump is simulated. Sine the characteristic Reynolds number of the pump is 1.5×10^5, which is lower than the range of critical Reynolds number 5.0×10^5 [5], the SST k-ω turbulence model[6] is used. For the cavitating flow, the mixture cavitation model is adopted. This cavitation model neglects the slip between different phases. The cavitation procedure is simulated by two phases (liquid water and non condensable gases) and three parts (liquid water, stream, and non condensable gases). By solving the volume fraction transport equation, the cavitating flow can be got. It is noted that the growth or collapse of cavity bubble is controlled by the Rayleigh-Plesset equation.

For the boundary condition, the total pressure condition is set at both pump inlets, and the flow discharge is set at the pump outlet. And non slip boundary condition is adopted for the solid walls. The first up-wind scheme is applied and the convergences criterion is 10^{-4}.

Since the cavitating flow would significant influenced by boundary flow near a solid wall, the quality of mesh generation near wall region would dramatically influence the precision of cavitation simulation. Thus, the mesh refinement has been treated carefully. Near the wall 6 grid layers have been inserted, and the grid thickness is set to 30 μm, which satisfies the y+ value less than 9. Fig.2 shows y+ contour in the mid-span surface along a single vane suction surface for cavitation and non-cavitation conditions. Note that r_0 is the inlet radius of the vane, the radial distance from vane leading edge is normalized by the impeller outlet radius R_2. According to Fig.2, at both conditions, the y+ distribution satisfactorily meet the logarithm law. Thus, standard wall function can be used.
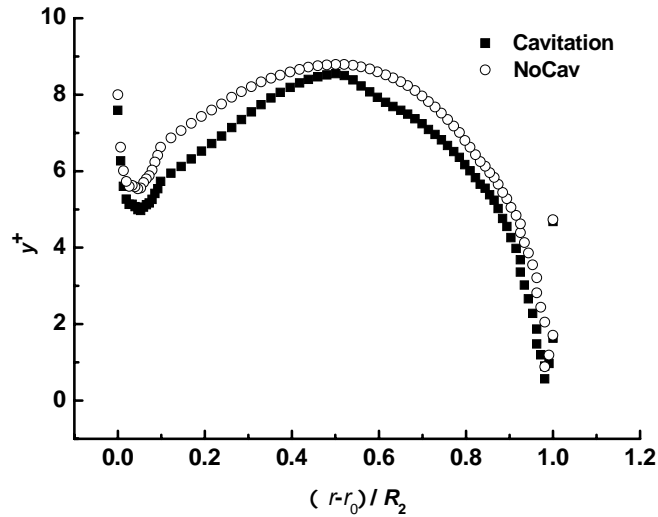

Fig. 2 y+ distribution along a vane suction surface

Fig.3 shows structured mesh of an impeller. The total number of the grid nodes is approximately 990,000. And the full passage flow domain of the pump is divided as rotating domain (including the impeller and both inlet pipes), and stationary domain (including two pump inlet and volute casing). The rotating domain and stationary domain are connected by sliding interfaces.
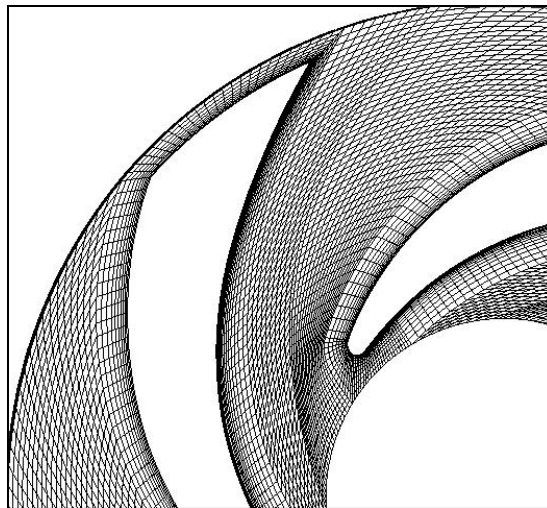

Fig. 3 Mesh for the impeller

4. Results and Analysis

4.1 Cavitation performance

Fig.4 shows the cavitation performance by static-state calculation. Here, IB2 and IB3 are calculated at design operation condition, while IB1 is calculated at both flow-rate of design condition i.e. Q_d and a large flow operation condition i.e. $1.25Q_d$.
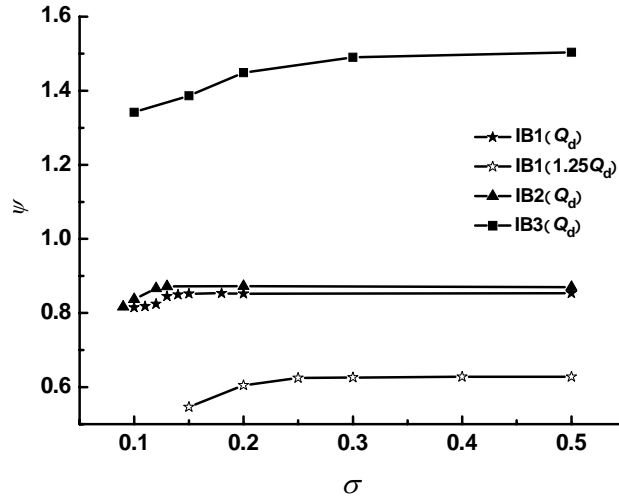
Fig. 4 Relation between ψ and σ for three impellers

Besides, head coefficient and Thoma's cavitation number are defined as follows.

$$\psi = \frac{H}{U_2^2 / 2g} \tag{1}$$

$$\sigma = \frac{\text{NPSH}}{H_d} \tag{2}$$

The critical *NPSH* value i.e. *NPSH*c is calculated at the cavitation coefficient where the pump head decreases by 3% compared with the non-cavitation condition. From Fig.4, it is noted that the critical values of NPSH at design operation condition for IB1, IB2 and IB3 are 0.13, 0.1, and 0.2 respectively. *NPSH*c at $1.25Q_d$ for IB1 is 0.2, which depicts that the value of *NPSH*c would increase with the flow rates near the design operation condition.

Similar with specific speed, cavitation specific speed *C* can fully describe cavitation performance. It is defined as:

$$C = \frac{5.62n\sqrt{Q}}{\sqrt[4]{NPSH_c^3}} \tag{3}$$

where, *n*: pump rotational speed; *Q*: flow discharge; *NPSH*c: critical net positive suction head.

The cavitation specific speed can represent the cavitation performance of a pump. The calculated values of cavitation specific speed for IB1, IB2 and IB3 are 1493, 1818 and 1081 respectively. For the pump having IB1 and IB2, the cavitation performance is acceptable if compared with that of the ordinary pump, whose cavitation specific speed usually have the value of 1200~1400. For the pump with impeller IB3, though the cavitation performance is not so good as that with IB1 and IB2, the impeller can produce much higher head.

4.2 Cavitation flow in the miniature pump

Fig.5 shows pressure contour at the mid-span surface along vane IB1 surfaces. The result is achieved at design flow rate, and the cavitation number is 0.13. Fig.6 shows the cavity volume fraction distribution along impeller passage at design flow i.e. Q_d and large flow condition i.e. $1.25Q_d$, and the cavitation number of both conditions is 0.13.
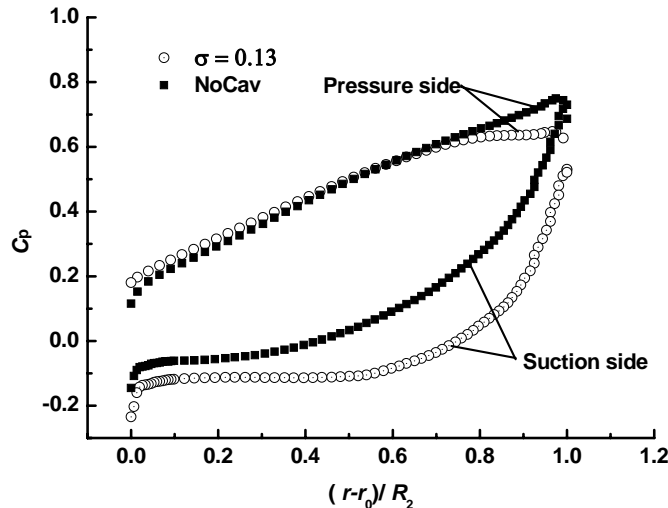

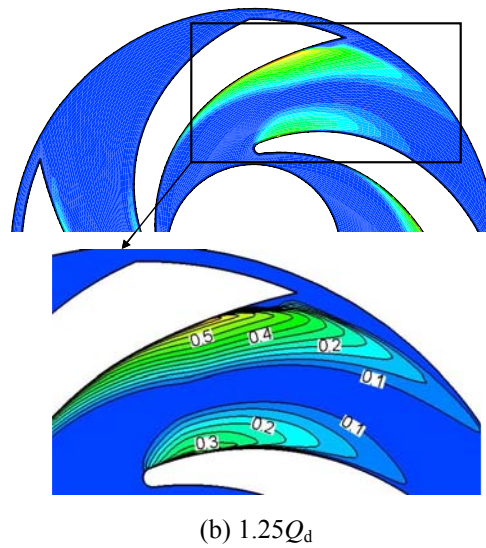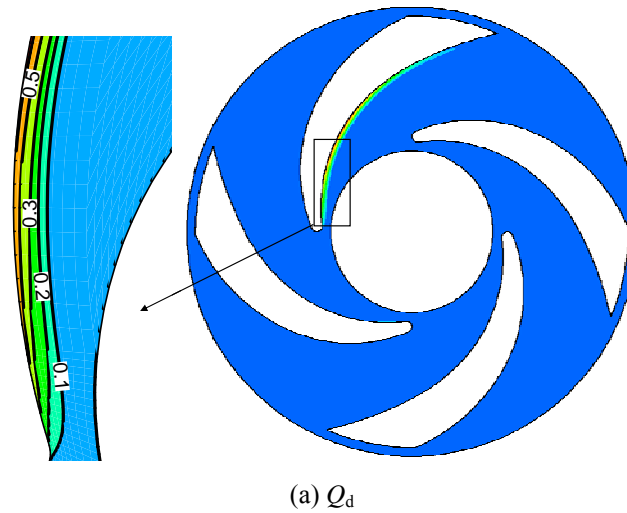**Fig. 5** Pressure along vane surface at mid-span for IB1

194

(a) Q_d

(b) $1.25Q_d$

Fig. 6 Cavity volume fraction for impeller IB1（σ =0.13）

From those results, it is shown that the cavity appears along the suction surface of the vane, and the static pressure at vane suction surface dramatically decreases when the pump is operated at critical cavitation condition i.e. σ_c. Since the static pressure on the vane pressure surface changes only near the trailing edge, the present load acting on the vane becomes much larger than that at non-cavitation condition. Though the cavity does not expand in vane-to-vane passages due to the narrow flow channel, the hydraulic efficiency will obviously drop when cavitation develops to critical cavitation number.

With the increase of the flow rate, the cavity shown in Fig.6(b) grows rapidly. Along the suction surface of vane IB1, the cavity expands remarkably. Further, the cavity attaches not only on the vane suction surface, but also on the leading edge of the vane pressure surface. The results illustrate that there are two cavity regions, which may block the flow passage severely, and further lower the hydraulic performance dramatically.

In order to compare the cavitation performance among those three impellers, the cavitating flow for IB2 and IB3 when cavitation number equals to 0.13 is also shown in Fig.7 and Fig.8. For the case of IB2, though there is little cavity occurring at the leading edge of one single vane suction surface, the cavity volume is very small, and would not cause the performance drop. For the case of IB3, the cavitation has fully developed, and the cavity grows toward middle region of the vane-to-vane passage. This cavity in IB3 would cause rapid decrease of the pump head. Thus, the development of cavitating flow in those three pumps is much different, and cavitation performance for impeller IB2 is the best.

4.2 Further consideration

In generally opinion, the geometry parameters at impeller inlet is crucial for cavitation performance[7]. Since the cavity first appears at the impeller inlet nearby, the flow pattern at the inlet mainly decides the cavitation performance of the pump. Vane angle at the leading edge would directly affect the inlet flow. And it would cause attack loss if the angle is not appropriate. Thus, reasonable vane angle at the leading edge would help to improve the cavitation performance.
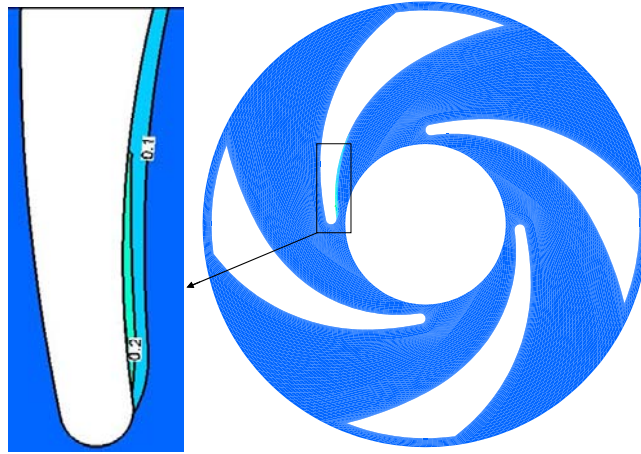
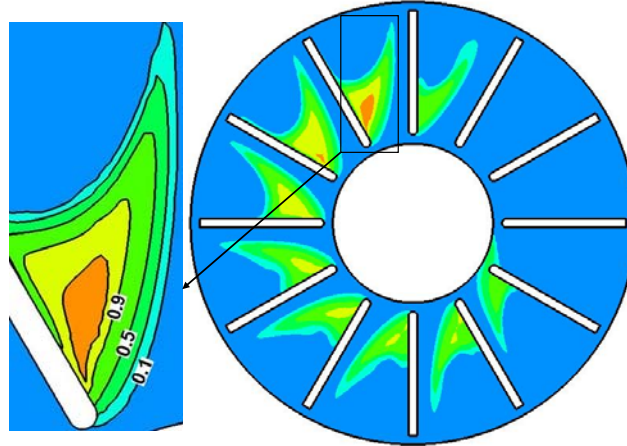**Fig. 7** Cavity volume fraction for impeller IB2 ($\sigma = 0.13$, Q_d)

**Fig. 8** Cavity volume fraction for impeller IB3 ($\sigma = 0.13$, Q_d)

Though the inlet geometry structure decides the pump cavitation performance in theory, it is difficult to relate a geometrical parameter with cavitation performance directly and quantitatively. Based on the literatures [8~9], it is believed that the uniform flow upstream of the impeller inlet is very important to influence cavitation performance of a pump. For convenience, the pressure coefficient i.e. C_p is defined:
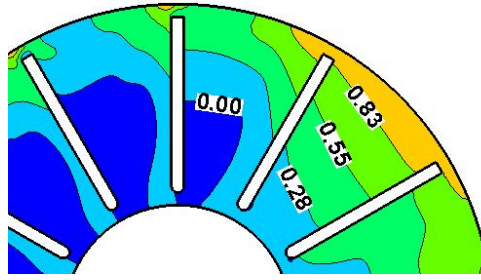
$$C_p = \frac{p - p_0}{0.5\rho U_2{}^2} \tag{1}$$

where p_0: reference static pressure at impeller inlet.
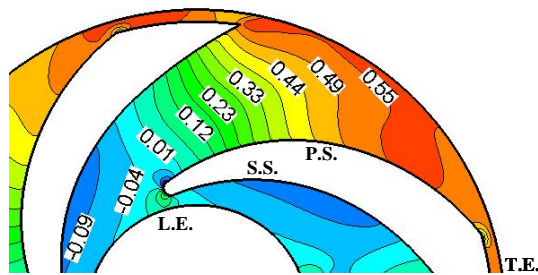
(a) Impeller IB1

(b) Impeller IB2

196
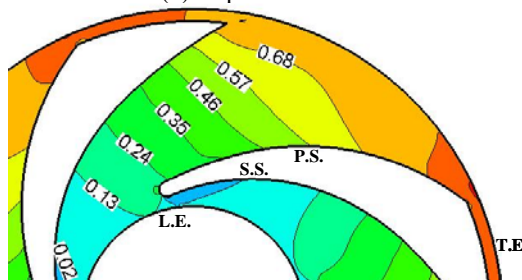
(c) Impeller IB3

Fig. 9 C_p distribution at vane mid-span (σ=0.13, Q_d)

Fig.9 shows the total pressure coefficient contour at the mid-span surface for IB1, IB2 and IB3 at σ=0.13 and Q_d. It is noted that the pressure distribution at vane leading edge for impeller IB2 is more uniform than impeller IB1. Besides, the low pressure region near suction surface for IB2 is also smaller than the case of IB1. On the other side, the total pressure distribution for IB3 is not uniform compared with impeller IB1 and IB2, and there is a large bubble zone with low total pressure. From the flow pattern for those impellers, uniform flow condition near the leading edge of impeller vane will benefit better cavitation performance.

However, it is believed that by applying the double suction design, the cavitation performance for all impellers has been improved compared to single suction structure miniature pump.

5. Conclustion

1) Cavitation performance of the double suction shaft-less miniature pump having different impeller is equivalent to the centrifugal pump having ordinary size, though the flow passage at impeller inlet is small.

2) The miniature pump having radial impeller can produce much higher pump head, but lower cavitation performance than that having the impeller based on the conventional design method.

3) It is believed that by applying the double suction design, the miniature pump achieved relatively uniform flow pattern upstream the impeller inlet, which is favorable for improving cavitation performance.

Acknowledgments

Nomenclature

b_1	Vane width at leading edge [mm]	n	Rotational speed of impeller [min⁻¹]
b_2	Vane width at trailing edge [mm]	$NPSH$	Net positive suction head [m]
C	Cavitation specific speed [min⁻¹·m³min⁻¹.m]	p	Static pressure [Pa]
C_p	Pressure coefficient	ρ	Fluid Density[kg·m⁻³]
D_1	Diameter at impeller inlet [mm]	Q	Flow discharge [l·min⁻¹]
D_2	Diameter at impeller exit [mm]	Q_d	Flow discharge at design point [l·min⁻¹]
D_2	Diameter at impeller exit [mm]	r	Radial distance from the impeller axis [mm]
G	Gravitational acceleration [m·s⁻²]	R_2	Radius at impeller exit, $= D_2/2$
H	Pump head [m]	U_2	peripherical speed at impeller exit [m·s⁻¹]
H_d	Pump head at design point [m]	Z	Number of blades
β_1	Inlet blade angle [°]	φ	Wrap angle of a vane [°]
β_2	Outlet blade angle [°]	σ	Thoma's cavitation number

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