Numerical simulation of cavitation effects influenced by centrifugal pump inlet parameters

L F Zhao, Y Wang, C Ning, Z C Liu, Z T Zhu and S F Xie
Research Center of Fluid Machinery Engineering and Technology, Jiang Su University, Zhenjiang 212013, China
E-mail: 1020655375@qq.com

Abstract. Cavitation has great influence on performance of the centrifugal pump. However, there is still no effective design to overcome this problem. Blade leading edge of centrifugal pump impeller is the initial position of cavitation. The leading edge geometry shape not only has a great influence on the cavitation inception and its development, but also a great influence on the flow state near the impeller inlet. In this paper, the numerical simulation method is adopted. Cavitation of four different models (including rectangular-shape blade model, circular-arc-shape blade model, elliptical-shape blade model and cusp-shape blade model) are simulated under the same condition by changing the NPSHA value. The influence of different blade models on cavitation performance is analyzed. The results show that the deviation between the simulated data and experimental data is within the deviation range. The head of rectangular-shape blade model and circular-arc-shape blade model are higher than those of elliptical-shape blade model and cusp-shape blade model. However, the head of rectangular-shape blade model and circular-arc-shape blade model is smaller than the latter under the low effective cavitation margin. What’s more, the head of the models with trimmed blade are higher than the head of the models with untrimmed blade under the working condition but are smaller under the low effective cavitation margin.

1. Introduction
Centrifugal pumps are widely used in chemical industry, agriculture, municipal engineering etc. Its internal cavitation has always been the major problem in fluid machinery internal flow research. Because cavitation can result in cavitation erosion, performance degradation, vibration, noise and a series of problems, so it must be considered in fluid machinery design. In recent years, the cavitation researches mainly focus on the theoretical study, experimental study and numerical analysis. Biao Huang, Guoyu Wang and Haitao Yuan proposed a density modify based cavitation model for cavitation flow computations [1]. Eric Goncalvès, Boris Charrière studied the mass transfer term between phases [2]. MØRCH K. A studied some special characters of cavitation nuclei [3]. Frédéric Caupin and Eric Herbert pointed out that metastable state between liquid and vapor plays an important role in establishment of cavitation model [4]. Xu Yu, WU Yulin, TANG Xuelin derived the cavitation flow control equation with the kinetics method [5]. Yuan Danqing, Chen Xiangyang, Bai Bin, Cong Xiaoqing pointed out that cavitation and cavitation erosion is the research emphasis of the future [6]. What’s more, the free streamline approach tells us that smooth shedding and mutation shedding also play an important role in the form of the different cavitation shape [7]. Experimental study mainly focuses on PIV, high-speed digital camera and other advanced imaging technology, which are used to study structure and characteristics of cavitation. Xu Y B has studied the resistivity imaging technology.
between liquid-vapor phases [8]. Shi L L has studied the digital image recognition technology based on gas-liquid two phases flow parameters [9].

Blade leading edge of centrifugal pump suction side is the initial position of the cavitation, here the rapidly rotating leading edge of blade interacts strongly with the low speed liquid, which results in big pressure gradient in the blade suction surface area, and it is easier to cause cavitation under the low inlet pressure. What’s more, the different leading edge of blades have great effect on the interaction method between liquid and blades, and the form of shedding vortex of the blade suction surface are different, which affects the degree of internal flow turbulence, also the performance.

This paper mainly focuses on the liquid flows around different solid obstruction and its complex cavitation flow structure under different conditions, also its effect on the performance of pump. Based on the above idea, numerical simulation is applied to study influence on the cavitation structures of four different models with different blades leading edge.

2. Model hydraulic parameters and the experimental dates
A good hydraulic performance centrifugal pump model is analyzed. The geometric parameters of the model are shown in table 1, and the experimental dates are shown in table 2.

![Table 1. Geometric parameters of the pump](image)

| Impeller outer diameter(D2) | Blade width at impeller outlet(b2) | Suction diameter(Ds) |
|----------------------------|----------------------------------|---------------------|
| 173 mm                     | 20 mm                            | 90 mm               |

![Table 2. Experimental dates](image)

| Flow rate (m³/h) | Head (L/S) | Revolving speed (r/min) | Efficiency (%) | Power (kw) | NPSHC (m) |
|------------------|------------|-------------------------|----------------|------------|-----------|
| 60               | 16.7       | 37.78                   | 70.89          | 8.72       | 2.92      |
| 100              | 27.8       | 33.98                   | 81.51          | 11.36      | 3.37      |
| 120              | 33.3       | 29.02                   | 77.44          | 12.25      | 4.09      |

Figure 1 shows the four kinds of blades .ZF represents the blade with rectangular shape leading edge, YH represents the blade with circular arc shape leading edge, TY represents the blade with elliptical shape leading edge, and JJ represents the blade with cusp shape leading edge.

![Figure 1. Shape of different blade leading edges.](image)

3. Steady-state numerical calculations

3.1 Boundary conditions and mesh generation
Because the flow near inlet of impeller and volute outlet is not constant, the length of inlet pipe and the volute outlet are properly lengthened, about 3 to 4 times the diameter of the corresponding location. ICEM -CFD 14.5 is used to generate the unstructured grid, and the number of grid of the four models is about 700000, and the grid density near every tongue and back of blade suction surface of impeller are increased.
Numerical simulation shows 72 group results of four different model pumps under different inlet pressure, convergence accuracy of simulation is set to $10^{-6}$. The inlet of pipe is set to pressure parameter, and volute outlet is set to velocity parameter.

3.2 Cavitation model

The tendency for a flow to cavitate is characterized by the cavitation number, defined as:

$$C_p = \frac{P - P_v}{\frac{1}{2} \rho U^2}$$  \hspace{1cm} (1)

where $P$ is a reference pressure for the flow, and $P_v$ is the vapor pressure for the liquid, and the denominator represents the dynamic pressure.

The Rayleigh-Plesset equation provides the basis for the rate equation controlling vapor generation and condensation. The Rayleigh-Plesset equation describing the growth of a gas bubble in a liquid is given by:

$$R^2 \frac{d^2 R}{dt^2} + \frac{3}{2} \left( \frac{dR}{dt} \right)^2 + \frac{2\sigma}{\rho_f R} = \frac{P_v - P}{\rho_f}$$ \hspace{1cm} (2)

where $R$ represents the bubble radius, $P_v$ is the pressure in the bubble (assumed to be the vapor pressure at the liquid temperature), $P$ is the pressure in the liquid surrounding the bubble, $\rho_f$ is the liquid density, and $\sigma$ is the surface tension coefficient between the liquid and vapor.

This equation gives the conditions of mechanical equilibrium of bubble in liquid. The first item of the equation represents second order vibration, the second item is for the related item of radius change over time derivative, the third is for surface tension, and the fourth item is for the difference of inside and outside pressure of the bubble.

Neglecting the second order terms (which is appropriate for low oscillation frequencies) and the surface tension, this equation reduces to:

$$\frac{dR}{dt} = \sqrt{\frac{2}{3} \frac{P_v - P}{\rho_f}}$$ \hspace{1cm} (3)

For

$$dV_b = 4\pi R_b^2 dR_b$$ \hspace{1cm} (4)

And then

$$\frac{dV_b}{dt} = 4\pi R_b^2 \frac{dR_b}{dt} = 4\pi R_b^2 \sqrt{\frac{2}{3} \frac{P_v - P}{\rho_f}}$$ \hspace{1cm} (5)

$$\frac{dm_b}{dt} = \rho_b \frac{dV_b}{dt} = 4\pi R_b^2 \rho_b \sqrt{\frac{2}{3} \frac{P_v - P}{\rho_f}}$$ \hspace{1cm} (6)

If there are $N_b$ bubbles per unit volume, the volume fraction can be expressed as:

$$\gamma = \frac{V_b N_b}{3 \pi R_b^3} = \frac{4}{3} \frac{V_b n}{\pi R_b^3}$$ \hspace{1cm} (7)

And then the mass transfer equation can be expressed as:
\[ \dot{m}_b = N_B \frac{dm_B}{dt} = \frac{3 \gamma_s \rho_s}{R_B} \sqrt{\frac{2 P_v - P}{\rho_f}} \]  

(8)

A correction coefficient number \( F \) is considered for the empirical correction, the value of \( F \) may be different in process of condensation and vaporization. If the condensation process is considered, the bubble radius \( R_b \) is replaced by bubble nuclear radius \( R_{nuc} \), the mass transfer is neglected because the mass transfer is very slow during the growing of bubble, the density of interior of bubble will decrease with growing of the bubble, \( \gamma_g \) is replaced by \( \gamma_{nuc} (1 - \gamma_g) \), and then:

\[ \dot{m}_b = F \frac{3 \gamma_{nuc} (1 - \gamma_g) \rho_s}{R_{nuc}} \sqrt{\frac{2 |P_v - P|}{\rho_f}} \text{sgn}(P_v - P) \]  

(10)

\( \gamma_{nuc} \) is the volume fraction of the nucleation sites.

In the simulation of CFX, \( R_{nuc} = 1 \mu m \); \( \gamma_{nuc} = 5 \times 10^{-4} \); \( F_{vap} = 50 \); \( F_{cond} = 0.01 \).

4. Calculation results

4.1 Cavitation performance

Figure 2 shows the pump cavitation performance curve of the four different models. All the cavitation performance curves are d type curves [10], which means that the performances curve have two extreme value points (as shown in figure 2). What’s more, the cavitation curve of YH have four extreme values. The head of circular arc shape blade model is higher than the head of rectangular shape blade model about 0.5 meter, and higher than the head of elliptical shape blade model and cusp shape blade model about 1.5 meter under normal working conditions. However, the head of circular arc shape blade model decreases faster than the others under low NPSHA. In addition, the head of the models with trimmed blade are higher than the head of models with untrimmed blade under the normal working condition. However, it is smaller than the latter under the low effective cavitation margin, which means that the head of models with trimmed blade decrease faster than the head of models with untrimmed blade under the low effective cavitation margin.

The cavitation performance curve of elliptical shape blade model is very similar with the cavitation performance curve of cusp shape blade model, as shown in figure 2.

![Figure 2. Cavitation performance of the four models](image)

Figure 3 shows the changes of relevant physical parameters of the four models near the impeller inlet location about 2 mm, 25 points are selected on the straight line within the dark color area (as
shown in picture a). Picture b is the pressure coefficient distribution picture, and its trend is consistent with the pressure distribution (as shown in figure d) because the value of $C_p$ is proportional to the value of pressure. Pressure value of YH decreases firstly, and increases slowly near the suction surface of blade, while the pressure values near blade suction surface of other three models always decline. ZF model has the smallest value at the same location among the four models. The pressure distribution of TY model is almost the same with the pressure distribution of JJ model, while the pressure value of TY model is slightly bigger than the pressure value of JJ model at the same location.

Figure 3(c) gives the vapor phase volume fraction of the 25 points position which was told above. The change trend of volume fraction of ZF, TY, JJ models are almost exactly the same, but the volume fraction value of ZF is bigger than the volume fraction value of the JJ and TY at the same location. The volume fraction value of YH is bigger than the value of the other three models near blade suction surface, but smaller than the latter in the location where it is far away from the blade suction surface.

4.2 Physical parameter of the position near the impeller inlet analysis (inlet pressure $P = 100000$ Pa)

![Figure 3](image)

**Figure 3.** Values of physical parameters in the Graphical position at $P = 2 \times 10^4 \, Pa$

Figure 4 shows the pressure distribution cloud pictures of the four models under normal condition ($P = 100000 \, pa$) near the impeller inlet about 2 mm. The low pressure area of TY and JJ are relatively larger than the low pressure area of ZF and YH. The blade suction surface near the blade leading edge of ZF is easier to form low pressure area which can be seen from the picture ZF, while the blade suction surface near the blade leading edge of YH is the most difficult to form the low pressure area. However, YH has the biggest pressure value, as the picture shown in the red areas, which means
higher head.

Figure 4. Pressure distribution of section near impeller inlet

4.3 Physical parameters of the section near impeller inlet analysis (inlet pressure $P = 20000$ Pa)

Figure 5 shows the volume fraction distribution cloud picture of the section near impeller inlet about 2 mm when the pressure of impeller inlet is 20000 Pa. As is shown in the picture, YH has the biggest volume fraction value in this section (which can be seen from the dark area). In addition, the volume fraction distribution of TY is almost the same with JJ, and the volume fraction distribution of ZF is similar to YH. In the cavitation area, the volume fraction value of YH and ZF are larger than TY and JJ. The area in the block diagram of YH and ZF has a flow structure which is formed by the interaction flow between vapor and liquid. However, this phenomenon does not occur in TY and JJ.

Figure 5. Vapor phase volume ratio distribution

Figure 6 shows the relative velocity distribution of the vapor phase of the corresponding location in figure 5. The relative velocity of the edge of the cavitation area is smaller, and the relative velocity of the center of the cavitation area is relatively larger. The relative velocity distributions of YH and ZF are relatively homogeneous. In block diagram of YH, the relative velocity of vapor phase is very slow, which is due to the condensation of liquid to vapor, this phenomenon also exists in ZF. However, TY and JJ do not have this phenomenon.

Figure 6. The relative velocity of vapor phase

5. Conclusion and discussion

This paper studies four different hydraulic models cavitation performance, we can reach the following conclusion:
1) The hydraulic performance of the models with trimmed blade is better than the model with untrimmed blade under normal working condition. Perhaps the main reason is that the shedding vortex frequency and its strength of trimmed models are different from the untrimmed models.

2) In the case of the low effective cavitation margin, the head of models with trimmed blade decrease slightly faster than the models with untrimmed blade.

3) Different hydraulic models have different cavitation performance curve shape. Although the cavitation performance curves of four kinds of models are similar to d type curve, its extreme value will change with different blades.

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