Numerical studies in a centrifugal pump with the improved blade considering cavitation

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Abstract. In this paper, a centrifugal pump with the improved blade for cavitation is studied numerically. A 3D impeller with logarithmic spiral blade profile was designed by the in-house hydraulic design code using a centrifugal pump geometric parameters, and the blade profile curve of suction side on the designed impeller is replaced by a combination of tangent line and circle arc line. The cavitation flows in the centrifugal pump with designed impeller, modified impeller and centrifugal pump spectrum impeller are respectively calculated by two-phase CFD simulation at three flow rates. The tests of the centrifugal pump have been conducted to verify numerical simulation. The effects of designed impeller and modified impeller on hydraulic efficiency, critical cavitation number, cavitation length, head drop performance and vapor cavity distribution in impeller are investigated. The results show that the modification of blade profile curve of suction side can improve the cavitation performance of an impeller and hydraulic efficiency of centrifugal pump. Compared with designed impeller, the critical cavitation number of centrifugal pump with modified impeller decrease by 26.5% under the same flow rate coefficient, and the cavitation intensity in the modified impeller is weakened effectively. The hydraulic efficiency of modified impeller also increases by 4.9%.

1. Introduction
Cavitation in impeller induced instability, pressure pulsations, erosion and pump vibration have been observed in different kinds of pumps[1-3]. Cooper and Antunes[4] have explored the causes in many researches. The primary effect on the cavitation behavior is the geometry at the impeller eye for a pump. There are some geometric factors at the impeller eye have an effect on cavitation, such as, the blade inlet angles inlet and hub diameters and incidence to upstream flow, blade passage throat area and blade profile, etc.

In recent years, the research about the effect of the above factors on pump cavitation has been studied and reported[5-8]. Eugene and Ravi[6] studies the effect of blade leading edge profiles on the cavitation behavior of an impeller. Palgrave and Cooper[7], have conducted visual studies of cavitation and present a general expression for estimating $NPSH$, based on inlet angle and eye diameter. Schiavello and Visser[8] have studied most of the aspects of cavitation. Herbich[9] have conducted experiment of circular arc blades and it shows that the efficiency of log spiral blade is highest. Cooper[10] and Hackworth[11] proposed that an appropriate blade profile can improve effectively cavitation performance of impeller in the patent of anti-cavitation impeller design. Katz[12] and Franc[13] have studied the relationship between cavitation and boundary layer separation of
stationary airfoil, the result shows that a low pressure zone is formed behind the point of boundary layer separation which induces cavitation concomitantly. Dumitrescu[16] have studied the bubble separation on the leading edge of rotating blades, and proposed that the blade profile and low pressure zone also affect the separation of leading edge.

Above researches mainly discuss the improving cavitation performance of pump are traditional methods such as changing geometry parameters of impeller inlet and inlet angle of blades, etc, however, the effect of blade profile is neglected, especially the suction surface of blade. Actually, the blade profile affects directly on the development of low pressure zone in suction surface and the motion of cavities region.

In this paper, we study the influence of the blade profile modification in suction side on the cavitation performance of a centrifugal pump. It is that the blade profile curve of suction side on designed impeller is replaced by a combination of tangent line and circle arc line. The cavitation flow in centrifugal pump with the designed impeller, the modified impeller and IS the centrifugal pump impeller were calculated by two-phase CFD simulation with the Zwart-Gerber-Belamri cavitation model. A test rig was set up to conduct the cavitation performance tests of the centrifugal pump to verify the numerical simulation. The effect of designed impeller and modified impeller on hydraulic efficiency, critical cavitation number, cavitation length, head drop performance and vapor cavity distribution in impeller are investigated.

2. Mathematical model of numerical simulation

2.1. Continuity equation and momentum equation

Since a basic assumption homogeneous multiphase transport equation model, all phases share the same velocity, the governing mixture equations for continuity and momentum and the vapor phase volume fraction of transport equation respectively are:

$$\frac{\partial \rho_m}{\partial t} + \nabla \cdot (\rho_m \bar{u}_m) = 0$$  \hspace{1cm} (1)

$$\frac{\partial \rho_m \bar{u}_m}{\partial t} + \nabla \cdot (\rho_m \bar{u}_m \bar{u}_m) = -\nabla p + \nabla \cdot \left[ (\mu_m + \mu_t) \nabla \bar{u}_m \right] + \frac{1}{3} \nabla \left[ (\mu_m + \mu_t) \nabla \cdot \bar{u}_m \right]$$  \hspace{1cm} (2)

$$\frac{\partial}{\partial t} (\rho, \alpha_v) + \nabla \cdot (\rho \mu \alpha_v) = R$$  \hspace{1cm} (3)

Where $\rho_m$ and $\rho_v$ are the mixture density and vapor phase density, kg/m$^3$; $t$ is the time, s; $\bar{u}_m$ and $\bar{u}_v$ are the mass average velocity and the vapor phase velocity, m/s; $\mu_m$ and $\mu_t$ are the mixture dynamic viscosity and the turbulence viscosity, kg/(m·s); $P$ is the pressure of flow field, Pa.

$\rho_m$, $\bar{u}_m$ and $\mu_m$ are the weighted average of liquid phase and vapor phase which as follows:

$$\rho_m = \rho_l \alpha_v + \rho_v (1 - \alpha_v)$$  \hspace{1cm} (4)

$$\bar{u}_m = \bar{u}_l \alpha_v + \bar{u}_v (1 - \alpha_v)$$  \hspace{1cm} (5)

$$\mu = \mu_l \alpha_v + \mu_v (1 - \alpha_v)$$  \hspace{1cm} (6)

Where $\bar{u}_l$ and $\bar{u}_v$ are the velocity of liquid phase and vapor phase, m/s; $\mu_l$ and $\mu_v$ are the dynamic viscosity of vapor phase and liquid phase, Pa·s; $\rho_l$ is the liquid phase density, kg/m$^3$; $\alpha_v$ is the vapor phase volume fraction.

The interphase mass transfer rate $R$ is:
\[ R = R_v - R_c \]  

(7)

where the source terms \( R_v \) and \( R_c \) denote vapor generation (evaporation) and condensation rates.

### 2.2. Cavitation bubble dynamic equation

Assuming that all the bubbles in a system have the same size, Zwart-Gerber-Belamri cavitation model[14-15] is that all bubbles in system share the same radius, the interphase mass transfer rate \( R \) is:

\[
R = F \frac{3\alpha_n \rho_v}{R_b} \sqrt{\frac{2}{3} \frac{|P_b - P|}{\rho_i}} \text{sign}(P_b - P)
\]

(8)

where the source terms \( P_b \) and \( P \) denote the inside pressure and outside of cavity, Pa; the source term \( F \) is the empirical calibration coefficient; \( R_b \) is the bubble radius, m.

The final form of this cavitation model is as follows:

\[
\begin{align*}
\text{if } p & \leq p_v, \\
R_v &= F_{vap} \frac{3\alpha_{nu} (1-\alpha_n) \rho_v}{R_b} \sqrt{\frac{2}{3} \frac{p_v - p}{\rho_i}}, \\
\text{if } p & > p_v, \\
R_v &= F_{cond} \frac{3\alpha_n \rho_v}{R_b} \sqrt{\frac{2}{3} \frac{p - p_v}{\rho_i}}
\end{align*}
\]

(9) (10)

where \( \alpha_{nu} \) is the nucleation site volume fraction, the source terms \( p \) and \( p_v \) denote the flow field pressure and vapor pressure, Pa; the source terms \( F_{vap} \) and \( F_{cond} \) denote evaporation coefficient and condensation coefficient.

### 3. Computational domain and mesh

#### 3.1 Impeller model

This paper takes a centrifugal pump as the study object, its design flow \( Q_d = 200 \text{m}^3/\text{h} \), head \( H=20\text{m} \). The parameters of impeller of centrifugal pump are listed in table 1. Based on these basic parameters, the 3D impeller with logarithmic spiral blades was designed by the in-house hydraulic design code which using two-dimension hydraulic design theory[17], as shown in figure 1 and figure 2 respectively. Keeping all other parameters as identical, the blade profile curve (AC) of suction side on designed impeller is replaced by the combination of tangent line(AB) and circle arc line(BC) as shown in figure 3. The tangent line(AB) pass through tangent point A on the blade head, and there is an appropriate transition using circular arc at point B.

![Figure 1. Design blade model.](image1)  
![Figure 2. Design impeller model.](image2)  
![Figure 3. Modified impeller.](image3)
### Table 1. Design parameters of impeller.

| $D_1$ (mm) | $D_2$ (mm) | $B_2$ (mm) | $\beta_1$ (°) | $\beta_2$ (°) | $Z$ (rpm) |
|------------|------------|------------|---------------|---------------|-------------|
| 150        | 270        | 30         | 28            | 21            | 6           | 1450        |

### Figure 4. Computational domain and mesh of centrifugal pump

### Table 2. The head coefficient of centrifugal pump at different grid numbers

| Grid number/million | Head coefficient |
|---------------------|------------------|
| 1                   | 0.9311           |
| 2                   | 0.9292           |
| 3                   | 0.9262           |
| 4                   | 0.9206           |

3.2 Meshing and boundary conditions

The 3D computational domain and mesh of the centrifugal pump with designed impeller and modified impeller are shown in figure 4. In order to reduce the influence of grid number on the simulation, we adopt a consistent approach of meshing for different models, and the grid independence for cavitation numerical simulation at same condition has been conducted as table 2. The calculation results indicates that the maximum head coefficient deviation is 1.12%, which shows that the influence of grid to simulation is ignored. And the number of 12 million grids are more appropriate with comprehensive consideration.

The boundary conditions of simulations are set as follows:

- The inlet of the calculation domain is specified by using the averaged mass flow-rate, and the volume fraction of vapor phase is set to zero.
- At the outlet plane, an average static pressure is set according to the total pressure level at the domain inlet, and the value is reduced gradually to realize the cavitation development in impeller of centrifugal pump.
- All solid walls are set as the non-slip wall condition.
- The coupling between impeller domain and others are set mesh motion.
- The vaporization pressure $p_v = 3540$Pa, environment temperature of experiment $T=20{\degree}C$.

4. Results and Discussion

Three dimensionless parameters are defined to describe the cavitation flow in centrifugal pump:

\[
\sigma = \left( p_c - p_v \right) / 0.5\rho_i \tag{11}
\]

\[
\psi = H / \left( U_2^2 / 2g \right) \tag{12}
\]
Flow coefficient \( \phi = Q / \pi D_b U_2 \) \hspace{1cm} (13)

where \( p_\infty \) is inlet static pressure, Pa; \( \rho \) is fluid density, kg/m\(^3\); \( p_v \) is vaporization pressure of liquid phase, Pa; \( H \) is the head of centrifugal pump, m; \( U_2 \) is the peripheral speed of impeller outlet, m/s.

4.1 The reliability verification of numerical simulation

The hydraulic performance curves and the head drop performance curves of experiment and simulated centrifugal pump with the spectrum impeller are respectively shown in figure 5 and figure 6. It can be observed that there is a good correlation between the simulation and experiment. The simulation predictions follow the trend of the experimental data.

![Figure 5. Hydraulic performance curves of centrifugal pump experiment and simulation](image1)

![Figure 6. Head drop performance curves of centrifugal pump experiment and simulation](image2)

4.2 The analysis of hydraulic efficiency

The hydraulic efficiency curves of centrifugal pump with designed impeller and modified impeller are shown in figure 7. Note from the figure that the flow coefficient at the highest hydraulic efficiency of designed impeller and modified impeller are respectively \( \phi = 0.101 \) and \( \phi = 0.103 \), of which the deviation is 1.9%. It indicates that the modified impeller has little influence on designed condition. However, the hydraulic efficiency of modified impeller is higher than designed impeller at same flow rate, with maximum 4.9%.

![Figure 7. Hydraulic efficiency curves of centrifugal pump](image3)

4.3 The analysis of head drop curves

The head drop performances of designed impeller and modified impeller for different flow rates are shown in Figure 8. The cavitation number \( \sigma_{3\%}, \sigma_{5\%} \) and \( \sigma_{7\%} \) corresponding to the 3\%, 5\% and 7\% drop in head are used to represent the cavitation performance of impeller. For all the flow rates, it can be observed that the \( \sigma_{3\%}, \sigma_{5\%} \) and \( \sigma_{7\%} \) of modified impeller are smaller than designed impeller obviously, and the maximal percentage of decrease are respectively 26.5\%, 27.4\% and 28.4\%. It
because that the sum of incidence loss and fraction loss in the impeller entrance decrease.

![Graphs showing head drop curves](image)

Figure 8. The head drop curves of centrifugal pump in cavitating condition

4.4 The analysis of cavitation flow field and cavity length

The sheet cavitation blocking the impeller channels and cavity length are appropriate representation of the amount of developed cavitation. Figure 9 shows the contours of sheet cavitation blocking the impeller channels of the impeller mid-span location at $\sigma=0.14$ for $0.9Q_d$ flow rate. The cavitation inception formed at the leading edgy of blade, and develop to the tail along blade suction surface. It can be observed that the cavitation area of modified impeller is smaller than designed impeller which will decrease the flow loss caused by vapor-liquid two phase flowing. The analysis of above indicates that the modified impeller can shrink the cavitation intensity in impeller effectively.

Figure 10 shows the extent of vapor cavity development on the impeller mid-span location at $\sigma=0.14$. Note that the cavity lengths are non-dimensionalized by the relative distance along blade chord. From the figure, it can be noticed that the cavity length of designed impeller and modified impeller have little difference. However, the vapor volume fraction of modified impeller at same relative distance along blade chord is smaller than designed impeller.

![Cavity distribution and lengths](image)

Figure 9. The vapor cavity distribution in impeller mid-span: (a) Impeller mid-span location; (b) Design impeller; (c) Modified impeller.

Figure 10. Vapor cavity lengths for $\sigma=0.14$ in impeller mid-span
5. Summary and Conclusion
The hydraulic efficiency, head drop performances and cavitation length at different flow rates are studied using computational fluid dynamic (CFD) analyses with the Zwart-Gerber-Belamri cavitation model. It can be concluded that the impeller of adopting the blade profile curve modification of suction side has higher hydraulic efficiency and better cavitation performance. The hydraulic efficiency of modified impeller increased by 4.9%. The head drop performance curves also have clearly demonstrated the superior $\sigma_{35}$ performance of modified impeller, with decreased percentage 26.5%. The growth of the vapor bubbles in modified impeller is smaller than designed impeller, and will result in lesser cavitation damage and longer impeller life.

References
[1] Brennen C E and Braisted D M. 1980 Stability of hydraulic systems with focus on cavitating pumps
[2] Greitzer E M 1981 J. Fluid Eng. 103(2): 193-242
[3] Cooper P, Sloteman D P , Graf E and Vlaming D J 1991 Elimination of Cavitation Related Instabilities and Damage in High-Energy Pump Impellers Proc. of the 8th Int. Pump Users Symp. (Houston, Tex, 5-7 Mar 1991)
[4] Cooper P and Antunes F F 1983 Cavitation in Boiler Feed Pumps Symposium Proceedings: Power Plant Feed Pumps-State of the Art. EPRI CS-3158 (Cherry Hill, New Jersey, 2-4 June 1982)
[5] Bakir F, Rey R, Gerber A G, Belamri T and Hutchinson B 2004 Int. J. of Rotating Machinery 10(1) 15-25
[6] Balasubramanian R, Bradshaw S and Sabini E. Influence of Impeller Leading Edge Profiles on Cavitation and Suction Performance Proc. of the 27th Int. Pump Users Symp. (Houston, Tex, 12-15 Sep 2011)
[7] Palgrave R and Cooper P 1986 Visual studies of cavitation in pumping machinery Proc. of 3rd Int. Pump Symp. (Houston, Tex, 20-22 May 1986)
[8] Schiavello B and Visser F C Pump Cavitation-Various NPSHR Criteria, NPSHA Margins, and Impeller Life Expectancy Proc. of the 25th Int. Pump Users Symp. (Houston, Tex, 23-26 Feb 2009)
[9] Heribich J B. 1962 Modifications in Design Improve Dredge Pump Efficiency (Bethlehem: Fritz Engineering Lab, Lehigh University)
[10] Cooper P and Sloteman D P 1993 Impeller for centrifugal pumps: U.S. Patent 5,192,193[P]. 1993-3-9.
[11] Hackworth M, Eslinger D and Harrell N R 2009 Impeller for centrifugal pump: U.S. Patent 7,549,837[P]. 2009-6-23.
[12] Katz J 1984 J. Fluid Mech. 140(4): 397-436.
[13] Franc J P and Michel J M 1985 J. Fluid Mech. 154 63-90
[14] Dumitrescu H and Cardos V 2010 J. of Aircraft 47(5) 1815-9
[15] Plesset M S. 1949 J. of Applied Mechanics 16 228-31
[16] Zwart P J, Gerber A G and Belamri T 2004 A two-phase flow model for predicting cavitation dynamics Fifth Int. Conf. on Multiphase Flow (Yokohama, Japan, 30 May-4 June 2004)
[17] Zhang Y, Zhang J, Zhu H and Cai S 2014 Advances in Mech. Eng. 2014 803972