Research on axial thrust of the waterjet pump based on CFD under cavitation conditions

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Abstract. Based on RANS equations, performance of a contra-rotating axial-flow waterjet pump without hydrodynamic cavitation state had been obtained combined with shear stress transport turbulence model. Its cavitation hydrodynamic performance was calculated and analysed with mixture homogeneous flow cavitation model based on Rayleigh-Plesset equations. The results shows that the cavitation causes axial thrust of waterjet pump to drop. Furthermore, axial thrust and head cavitation characteristic curve is similar. However, the drop point of the axial thrust is postponed by 5.1% comparing with one of head, and the critical point of the axial thrust is postponed by 2.6%.

1. Introduction

Waterjet propulsion is also known as waterjet pump. It is a device that uses reaction spraying flow to push vessel forward by a pump. Due to the advantages of waterjet pump hydraulic performance, such as superior resistance to cavitation characteristics, and structural characteristics of the device make it more and more suitable and rapid development momentum [1].

Waterjet pump power stability is of foundation to the ship stability so the research of waterjet thrust characteristics is essential. Liu Chengjiang, Wang Yongsheng et. , introducing how to obtain waterjet propulsion thrust theory, experiment and numerical calculation method, validated CFD method is feasible and credible [2]. Jules W. Lindau, Christopher Pena et. analyzed of cavitation performance AxWJ-2 type waterjet pump, discovered the relationship between waterjet thrust broken and bubbles blocked rotor runner [3]. And usually, waterjets during acceleration and cornering produces cavitation[4]. But there is rarely studies on relationship between waterjet cavitation and thrust in terms of the internal flow at home and abroad [5-7].

In this paper, the use of CFD technology, a new type of rotary axial waterjet pump cavitation was calculated at design conditions, and the predicted axial thrust and internal flow of cavitation under different conditions, and preliminary analyzed of the relationship between the two.

2. Research model and computation method

Contra-rotating axial flow pump’s design parameters used in this paper are: design flow \( Q_d \) taken to 0.3 \( \text{m}^3/\text{s} \), first stage impeller head taken to 2 m, secondary impeller head taken to 8m. Contra-rotating axial flow pump specific speed 600. First stage impeller blade number 5, secondary impeller blades number 4, vanes leaf number is 7. Flow components joint model shown in figure 1 (figure 1 using Cartesian coordinate system),from the perspective of imports to exports, first stage impeller speed of -1450 r/min, direction is clockwise, secondary impeller speed of 1450 r/min, direction is
counterclockwise. Propeller model was three-dimensional modeling using Pro-ENGINEER, calculating area from entrance to exit and no exit tube in ordinary waterjet simulation, this study's propeller was added an export tube with special use, as shown in figure 1.

The grids of the three-dimensional model was generated by the ICEM CFD (integrated computer engineering and manufacturing code for computational fluid dynamics). The structured grids in computational regions are shown figure 2. The total number of grid elements is 1845627 for all the domains.

The boundary conditions of steady flow simulation were almost the same under both cavitation and non-cavitation conditions. The total pressure at the inlet of the suction pipe was applied, and then the outlet boundary condition was mass flow rate. The impeller surfaces were set up as a rotating wall. All the other walls were stationary.

3. Governing equation and cavitation model

In the mixture model of the vapor/liquid two-phase flow, the fluid is assumed homogeneous so that the multiphase fluid components were assumed to share the same velocity and pressure [8]. A cavitation process is expressed by the mass transfer equation. Equation (1) has demonstrated the conservation equation of the vapor volume fraction. Note that the source terms and represent evaporation and condensation.

\[
\frac{\partial}{\partial t} (\rho_v \alpha_v) + \frac{\partial}{\partial x_i} (\rho_v u_i \alpha_v) = R_v - R_c
\]  

(1)

\[ \rho = \rho_v \alpha_v + \rho_i (1 - \alpha_v) \]  

(2)

Where \( \alpha_v \) is volume fraction of one component.
\[ R_e = F_e N_b \rho \sqrt{\frac{2}{3}} \frac{|p_v - p|}{\rho_i} \text{sign}(p_v - p) \]  
\[ R_c = F_c N_c \rho \sqrt{\frac{2}{3}} \frac{|p_v - p|}{\rho_i} \text{sign}(p_v - p) \]

For the vaporization, \( N_b \) is given by
\[ N_b = \frac{3\partial \rho}{4\pi R_b^3} (1 - \partial) \]  
For the condensation, \( N_b \) is given by
\[ N_b = \frac{3\partial}{4\pi R_b^3} \]

In equation (3) and equation (4), \( P_v \) is vapor pressure, \( F_e \) is vaporization coefficient, \( F_c \) is condensation coefficient. Empirical coefficients used to represent the vapor increasing or reducing time step. For the vaporization, \( F_e = 50 \). For the condensation, \( F_c = 0.01 \). In the present study, SST turbulence model was selected.

4. Results and discussion
Figure 3 and 4 shows pump head and pump axial thrust of the cavitation performance of the waterjet pump. With the decrease of the net positive suction head (NPSHA), the pump head decreases. NPSHA = 3.76 is approximated the net positive suction head critical (NPSHC) for the pump while head down 2.6%. Head decreases caused by increasing bubbles in the impeller. With the appearance of cavitation, the flow inside the pump state changes and the change accumulate to a certain extent on the performance of a drop head.

We find it similar to figure 4 and figure 3, i.e. the axial thrust of the pump also decreases with the decrease of the NPSHA, reaching a critical point decline intensified.

The decline-starting point of the axial thrust delays 5.1% compared with the head decline-starting point, critical point delays 2.6%. When cavitation occurs, the axial thrust decline is earlier than head fall; it is possible vacuoles might hinder the interaction between the blade and the working medium. But this does not mean cavitation is the direct cause of head drop.

![Figure 3. Cavitation characteristic curve of the pump at \( Q_a \).](image-url)
4.1. Vapor bubbles distribution

This section the design point of axial thrust decline curve was selected to investigate the relationship between the development of the cavitation distribution in impeller channel and the decline of the axial thrust in conjunction with the impeller cavitation shown in Figure 5. It is analyzed the influence of cavitation based on 5 condition points (a–e) on the decreased curve of axial thrust in figure 4.

Figure 5 shows the distribution of bubbles of five operating points, while the cavitation region is composites by the volume fraction of 10% iso-surface.

When NPSHA was not reduced, slight cavitation occur in the suction side of the first stage impeller tip as shown in figure 5(a). When NPSHA was 4.78 m, the cavity area of the suction around the secondary impeller tip increased, while covered import to export side. Meanwhile, bubbles were found at pressure side near the hub of the first stage impeller, so was the suction side near the tip of the secondary impeller. When NPSHA dropped to 3.76 m, bubbles came out at the center of the suction side of the secondary impeller, which may be reasons and signs for the axial thrust decline beginning; When NPSHA was 2.99 m, bubble had covered 2/3 of the suction side of the first stage impeller, while the surface area of the bubble at the hub also further expanded, and at the same time the axial thrust had dropped 12.57%.

(a) NPSHA=10.02 m  (b) NPSHA=4.78 m  (c) NPSHA=3.76 m

(d) NPSHA=3.50 m  (e) NPSHA=2.99 m

Figure 5. Distribution of the vapor bubbles at Q_d
4.2. Impeller surface axial thrust

Figure 6, X-axis represents NPSHA, where Y-axis represents the axial thrust of every part of the impeller. With the NPSHA decreasing, the axial thrust of the impellers each surface reduced. The difference is that the axial thrust of the first impeller rose, when NPSHA reached the maximum was 4.78 m, an increase of 15.26%. Then it began to decrease, when NPSHA was 3.76 m, the axial thrust and starting point (NPSHA = 10.02 m) flat, slightly decreased 0.04%. Secondary impeller axial thrust curve by a slow decline to a sharp decline in the critical point of 3.76 m, showing a similar trend in figure 4. This will also be consistent with the analysis of figure 5(c).

![Figure 6. Axial thrust of impellers.](image)

Figure 7, the X-axis represents the position of the blade, 0 said blade inlet, 1 blade outlet; Y-axis represents the pressure coefficient of the blade. Pressure coefficient $C_p$ is defined as

$$C_p = \frac{p - p_{in}}{\frac{1}{2} \rho v_{in}^2}$$

Where $C_p$ is pressure coefficient, $p$ is blade surface static pressure, $p_{in}$ is pump inlet static pressure, $v_{in}$ is the tip peripheral speed of first stage impeller inlet side.

In figure 7(a), under conditions of no cavitation, a "low-voltage spikes "later in the work surface of impeller inlet side came out. In figure 8, the low pressure region near the inlet of the blade face appeared extending from the hub. From the above results that cavitation had occurred here, cavitation flow lines tail twisted, bubble extruded flow channel, changing the flow structure at the entrance. The suction side of the impeller blade pressure is the lowest part, high flow rate, and prone to cavitation.

With NPSHA = 4.78, 3.76 m, blade surface pressure decreased, but the basic trend remained unchanged. When NPSHA = 2.99 m, the blade face pressure was further reduced, area of "low spike" increased and figure 5(e) showed cavitation area near the hub of the first stage impeller further expansion. In this case the pressure at the suction side of the blade inlet increased, and smoothly extended to the rear of the blade.

In figure 7(b), when NPSHA reduced from 10.02 m to 4.78 m, pressure coefficient curve did not change significantly. NPSHA down to 3.50 m, the pressure coefficient decreased overall, and NPSHA down to 2.99 m, the pressure coefficient sharply declined.

5. Conclusion and recommendations

1) Waterjet pump axial thrust declines caused by cavitation. With NPSHA decreasing, the axial thrust and head of the pump appear a slight decline after the rise at beginning and decline intensified at the critical point. The decrease starting point of the axial thrust delays 5.1% compared with the head starting point, critical point delays 2.6%.
2) Axial thrust of the Impellers present cavitation critical point, while the initial stage of cavitation performance is of different nature. Cavitation initial stage, axial thrust of the first stage impeller rose. When NPSHA = 4.78 m it reached a maximum and then began to decline. The difference is that the initial stage of the axial thrust of the secondary impeller was gradually decreased. With NPSHA down to 3.76 m, the axial thrust appeared steep drop phenomenon.

3) We find that the curve of axial thrust presents similar trend to head cavitation characteristics, but both the starting drop point and the critical drop point is different. The next step is to explore the possible relationship between the two. In addition, this paper only preliminary reveals changes in axial thrust under cavitation conditions, but also analyzes the relationship between cavitation and axial thrust from the effects of cavitation on the flow structure.

![Figure 7. Distribution of $C_p$ at 0.5 span.](image)

![Figure 8. Distribution of the pressure and velocity of first stage impeller’s work side at NPSHA=3.25 m](image)

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