Numerical analysis of head degrade law under cavitation condition of contra-rotating axial flow waterjet pump

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Abstract. In order to study the flow-head characteristic curve, the SST turbulence model, homogeneous multiphase model and Rayleigh-Plesset equation were applied to simulate the cavitation characteristics in contra-rotating axial flow waterjet pump under different conditions based on ANSYS CFX software. The distribution of cavity, pressure coefficient of the blade at the design point under different cavitation conditions were obtained. The analysis results of flow field show that the vapour volume distribution on the impeller indicates that the vapour first appears at the leading edge of blade and then extends to the outlet of impeller with the reduction of Net Positive Suction Head Allowance (NPSHA). The present study illustrates that the main reason for the decline of the pump performance is the development of cavitation, and the simulation can truly reflect the cavitation performance of the contra-rotating axial flow waterjet pump.

1. Introduction
Cavitation flow are complicated because it is a rapid phase change phenomenon, which often occurs in the high-speed or rotating fluid machineries. Cavitation adversely affects the performance of hydraulic machinery, cause noise, vibrations, erosion and mechanical structure damage[1-3]. So it is necessary to study cavitation performance of hydraulic machinery.

Experimental methods and CFD are the main approaches to research the cavitation performance. Significant progress has been made on researching cavitation performance using experimental methods[4-6]. In recent years, because of the evolution of CFD technologies and the time-consuming of experimental investigation, numerical simulation has been widely used to reach the cavitation performance and the internal flow fields in hydraulic machinery.

With the development of waterjet propulsion technology, the contra-rotating axial flow waterjet pump is considered to be an ideal form of propulsion for its compact structure, good cavitation performance and wide stable operation zone[7]. When it runs under the off-design conditions, the torque and thrust alter significantly due to cavitation[8]. Liu chengjiang found that Inlet duct cavitation lags behind the blade cavitation, and the nozzle cavitation in form of spatial cavitation occurs ahead of the blade cavitation, but there is no cavitation on nozzle wall[9]. Jules W. Lindau, Christopher Pena[10-11] explored the cavitation performance of waterjet pump, AXWJ -2,designed by Michael, then the results show the thrust breakdown coincide with cavitation choking in rotator passage. In this study, the cavitation performance of contra-rotating axial flow waterjet pump will be revealed under different conditions.
2. Physical model
The computational domain established by the Pro/E software consists of inlet flow channel, front impeller, axial clearance, rear impeller, guide vanes, outlet flow channel. Pump heads $H_f=2\text{m}$, $H_r=8\text{m}$ and flow $Q=0.3\text{m}^3/\text{s}$ were specified in pump design. The rotational speed were determined as $N_r=N_i=1450\text{r/min}$. The numbers of blades were chosen as 5, 4 and 7 for the front impellers, rear impellers and guide vanes respectively. The physical model of contra-rotating axial flow waterjet pump is shown in Figure 1.

3. Numerical simulation
3.1 Mesh generation
The grid is generated for each part by the software ANSYS ICEM CFD and the grid number and distribution are guaranteed consistently at the interface transition. In order to effectively control the mesh distribution in the contra-rotating axial flow waterjet pump, the whole hexahedral strategy is adopted. It can be observed that the results on successive grids become closer to each other and with the further refinement the results become grid independent. The final computational grid number is 1840000. Figure 2 show the grid for the computational domain: (a) Grid for front impeller, rear impeller and guide vanes (b) Grid for contra-rotating axial flow waterjet pump (c) Grid for the pump just with the front impeller.

3.2 Numerical methods
For the boundary conditions, in the cavitation simulation process, the mass flow rate was imposed at the outlet and the total pressure was specified at the inlet, where gas volume fraction of 0 and liquid volume fraction of 1. The results of noncavitating flow was used as the initial flow field of the cavitating case to improve the convergence speed and accuracy. The saturated vapour pressure is 3169pa, when the liquid temperature is 25$^\circ\text{C}$. The frozen rotor interface model was employed for the interface between the rotational domains and stationary ones. The wall of impellers was set as rotational boundary conditions, while others are defined as no slip boundary conditions.

3.3 Cavitation model and turbulence model
Cavitation terms, Rayleigh-Plesset model, are used in this study. The evaporation and condensation rates are given as follows:

$$R_e = F_e N_b \rho_v 4\pi R_b^2 \sqrt{\frac{2 \left| p_v - p_l \right|}{\rho_l}} \text{sign}(p_v - p)$$

(1)

$$R_c = F_c N_b \rho_l 4\pi R_b^2 \sqrt{\frac{2 \left| p_v - p_l \right|}{\rho_v}} \text{sign}(p_v - p)$$

(2)

where $p_v$, vapour pressure; $F_e$, evaporation coefficient; $F_c$, condensation coefficient. $F_e=50$, $F_c=0.01$. The turbulence model, Shear Stress Transport model, is employed to make the control equation complete.
4. Results and discussion

4.1 The cavitation performance curves

Net Positive Suction Head (NPSH) is usually used to describe the problem of cavitation in pumps, which could be expressed as

\[
\text{NPSH} = \frac{P_{\text{in}}}{\rho g} + \frac{v_{\text{in}}^2}{2g} - \frac{P_r}{\rho g}
\]  

(3)

where \( P_{\text{in}} \) and \( v_{\text{in}} \) are the static pressure and velocity in the fluid close to the front impeller inlet, respectively.

The comparison of cavitation performance for the pump just with the front pump and the contra-rotating axial flow waterjet pump, called case 1 and case 2 respectively in the following study, is shown in Figure 3. The critical NPSH, \( \text{NPSH}_c \), is defined as the NPSH value when the head of the pump drops by 3%. The results show that \( \text{NPSH}_c \) in case 2 (NPSH=3.76m) is smaller than that in case 1 (NPSH=4.58m), which means the cavitation phenomenon in case 2 appears later than case 1. The comparison of cavitation performance for case 1 and case 2 will be revealed in detail from three different cavitation conditions.

Figure 2. Grid for the computational domain: (a) front impeller, rear impeller and guid vanes. (b) axial flow waterjet pump (case 1). (c) pump just with the front impeller (case 2).
4.2 The pressure coefficient distribution under different conditions
Cavitation may alter the pressure on the pressure surface and suction surface, according to the results showed in Figure 3, some detailed investigation will be proceed. Fig.4 shows CFD-Calculated surface pressure coefficient $C_p$ distribution around front impeller blades under different NPSHA at the design flow rate of 0.3m$^3$/s. The pressure coefficient $C_p$ is defined as

$$C_p = \frac{P - P_m}{0.5 \rho u^2}$$  \hspace{1cm} (4)

where $P$ is the static pressure, $P_m$ is the inlet total pressure, $u$ is circumferential velocity of the impeller.

From Figure 4, it shows that the $C_p$ distribution is similar at 0.3-0.7 streamwise under different NPSH before the deep cavitation occurs. The difference between case1 and case2 increases with the decrease of NPSH value at 0.8-1.0 streamwise(close to the tip of blade). Before the deep cavitation occurs, the trends of $C_p$ distribution under different NPSH are the same and the $C_p$ on the suction surface in case2 is always little higher than that in case1. The results show the good cavitation performance of the contra-rotating axial flow waterjet pump (case2).

4.3 The vapour volume fraction distribution
The vapour volume fraction distributions on blades of contra-rotating axial flow waterjet pump are shown in Figure 5. The cavitation phenomenon first occurs at the tip of the blades' suction surface, with the NPSH decrease, the surface covered by vapor bubbles increases. When NPSH=2.74m, the vapour region covered 75% of the suction surface of the blades, the cavitation region extended to the outlet of the impeller, at this time, the deep cavitation occurs.

Figure 6 shows the vapour volume fraction distribution in the flow passage contra-rotating axial flow waterjet pump under three conditions: the incipient cavitation condition (NPSH=4.58m), the critical cavitation condition (NPSH=3.76m), the deep cavitation condition (NPSH=1.74m). When NPSH=4.58m, the cavitation just occurs at the leading edge on the suction side of the front impeller, there is no cavitation phenomenon in the rear impeller. When NPSH=3.76m, the vapour covered on the pressure side of the front impeller and leading edge of suction side of the rear impeller. When NPSH=1.74m, the deep cavitation occurs, the flow passage is filled with bubbles, which choked the channel. Cavitation choking significantly alter the torque and the thrust. From Figure 6, it shows the cavitation first occurs in the front impeller and then extend to the rear impeller and guide vines due to the pre-pressure on the rear impeller.
Figure 4. Pressure coefficient $C_p$ distribution around front impeller blades

Figure 5. Vapour volume fraction distribution on blades

5. Conclusions
1) The NPSHc (NPSH=3.76m) of contra-rotating axial flow waterjet pump is smaller than that (NPSH=4.58) of the pump just with the front impeller. The contra-rotating axial flow waterjet pump has good cavitation performance.

2) With the NPSH decrease, the surface covered by vapor bubbles increases. At the deep cavitation condition, the vapour region covered 75% of the suction surface of the blades, the cavitation region extended to the outlet of the impeller.

3) Cavitation first occurs in the front impeller and then extend to the rear impeller and guide vines due to the pre-pressure on the rear impeller.
Figure 6. Vapour volume fraction distribution in the flow passage

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