International Journal of Multiphase Flow 77 (2015) 244–259

Contents lists available at ScienceDirect

International Journal of Multiphase Flow

journal homepage: www.elsevier.com/locate/ijmultiphaseflow

Numerical analysis of unsteady tip leakage vortex cavitation cloud and unstable suction-side-perpendicular cavitating vortices in an axial flow pump

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ARTICLE INFO

Article history:
Received 18 October 2014
Revised 14 September 2015
Accepted 15 September 2015
Available online 25 September 2015

Keywords:
Cavitation
Tip leakage vortex (TLV)
Suction-side-perpendicular cavitating vortices
Cavitation
Collapse
Instability
Axial flow pump

ABSTRACT

The objective of this work is to simulate and analyze the formations of three-dimensional tip leakage vortex (TLV) cavitation cloud and the periodic collapse of TLV-induced suction-side-perpendicular cavitating vortice (SSPCV). Firstly, the improved SST k–ω turbulence model and the homogeneous cavitation model were validated by comparing the simulation result with the experiment of unsteady cavitation shedding flow around the NACA66-mod hydrofoil, and then the unsteady TLV cloud cavitation and unstable SSPCV in an axial flow pump were predicted using the improved numerical method. The predicted three-dimensional cavitation structures of TLV and SSPCV as well as the collapsing features show a good qualitative agreement with the high speed photography results. Numerical results show that the TLV cavitation cloud in the axial flow pump mainly includes tip clearance cavitation, shear layer cavitation, and TLV cavitation. The unsteady TLV cavitation cloud occurs near the blade trailing edge (TE) where the shapes of sheet cavitation and TLV cavitation fluctuate. The inception of SSPCV is attributed to the tail of the shedding cavitation cloud originally attached on the suction side (SS) surface of blade, and the entrainment affect of the TLV and the influence of the tip leakage flow at the tailing edge contribute to the orientation and development of the SSPCV. The existence of SSPCV was evidently approved to be a universal phenomenon in axial flow pumps. At the part-load flow rate condition, the SSPCV may trigger cavitation instability and suppress the tip cavitation in the neighboring blade. The cavitation cloud on the SS surface of the neighboring blade grows massively, accompanying with a new SSPCV in the neighboring flow passage, and this SSPCV collapses in a relatively short time.

Introduction

Cavitation is usually observed in hydraulic machines such as pumps, marine propellers and water turbines. Usually, Cavitation results in erosion damage, noise, vibration and hydraulic performance deterioration by the periodic inception, growth and collapse of vapor bubbles (Higashi et al., 2002). Cavitation in the axial flow impeller is extremely complex, which can be classified into three types: sheet cavitation on the blade surface, back-flow vortex cavitation and tip leakage vortex (TLV) cavitation (Murayama et al., 2006). The attached sheet cavitation on the surface of hydrofoils (Coutier-Delgosha et al., 2003; Dular et al., 2005; Leroux et al., 2005), propellers (Bensow and Bark, 2010; Ji et al., 2011, 2012), and pumps (Horiguchi et al., 2004; Kikuta et al., 2010; Semenov et al., 2004; Tan et al., 2012; Tsujimoto et al., 1996) have been extensively studied. At present, it is possible to predict the mass flow gain factor, cavity shape, cavitation numbers and pressure fluctuation of the cavitating flow in the hydraulic machines (Horiguchi et al., 2006; Shimiya et al., 2008; Tani et al., 2012; Yamanishi et al., 2007). However, the TLV cavitation is highly unsteady and unstable especially in heavy cavitation conditions and the dynamics of TLV cavitation and the related vortex structure in the axial flow pump are extremely complicated.

Recently, it has been recognized that the cavitation instabilities of vortex cavitation, surge cavitation, rotating cavitation in the rotating machinery are due to its unsteady cavitation characteristics (Shimiya et al., 2008). Since the unstable TLV cavitating flow may cause severe damage of hydraulic machines, the cavitation instability control is expected to improve the performance and reliability of the hydraulic machines. Many cavitation control methods, such as expanding the casing diameter upstream of inducer inlet (Kamijo et al., 1993), J-grooves on the casing wall (Shimiya et al., 2008), drag-reducing polymer solution injection (Fruman and Aflalo, 1989) and active mass injection (Chang et al., 2011), have been investigated for the suppression of the surge and rotating cavitation instabilities in the inducers

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http://dx.doi.org/10.1016/j.ijmultiphaseflow.2015.09.006
0301-9322/

and hydrofoils. Wu et al. (Wu et al., 2010, 2011, 2012; Miorini et al., 2012) measured the TLV's structure and turbulence in an axial waterjet pump based on the two-dimensional and stereoscopic particle image velocimetry (PIV) technology, and analyzed the turbulence associated with the TLV. The extensive PIV results show that the TLV in the tip region is highly three-dimensional and the distributions of velocity and vorticity are varied significantly along the blade tip chord direction. Tan et al. (2012) and Zhang et al. (2013, 2015) also observed the tip attached cavitation, the secondary structures of TLV cavitation, alternate blade cavitation and sheet cavitation at the varying flow and pressure conditions using the high-speed imaging. Tan et al. (2015) analyzed the formation and development of the SSPCVs which are similar to the cavitating backflow vortices developed upstream of highly loaded inducers by the high speed imaging. However, a limited number of results are available for the TLV cavitation instabilities in axial flow pumps, although the unsteady TLV cavitation plays an important role in the reliability of axial flow pump. To avoid or at least to reduce such instabilities, it is essential to improve the understandings of the physical phenomenon associated with the unsteady TLV and SSPCVs as well as the mechanisms.

The objective of this study is to understand the correlation between the unsteady TLV cavitation and the related structure. An improved numerical method was validated by comparing the simulations with the experiment of the unsteady cavitation shedding flow around the NACA66 hydrofoil, and then the numerical results of TLV cavitation cloud and the SSPCV collapse in the axial flow pump were compared with the high-speed imaging results.

Numerical method description and setup

Governing equations and cavitation model

The mixture model for vapor/liquid two-phase flows assumes that the fluid is homogeneous, and various fluid components are assumed to share the same velocity and pressure (Zwart, 2004). The continuity and momentum equations for the mixture flow are expressed as:

$$\frac{\partial \rho_m}{\partial t} + \frac{\partial}{\partial x_j}(\rho_m u_j) = 0 \tag{1}$$

$$\frac{\partial}{\partial t}(\rho_m u_i) + \frac{\partial}{\partial x_j}(\rho_m u_i u_j) = f_i - \frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j}\left[(\mu_m + \mu_T)\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} - \frac{2}{3}\frac{\partial u_k}{\partial x_k}\delta_{ij}\right)\right] \tag{2}$$

where the density of mixture ρ_m is defined by the volume fractions as $\rho_m = \alpha_l \rho_l + \alpha_v \rho_v$, and ρ_l, ρ_v are the densities of the liquid and the vapor, respectively; α_l,α_v are the volume fractions of the liquid and the vapor, u_i is the velocity and f_i is the body force term in the i direction, p is the pressure, μ_m is the mixture laminar viscosity, μ_T is the turbulent eddy viscosity of mixture, which is obtained by the following turbulence model.

The cavitation is modeled by the vapor volume fraction mass transfer equation:

$$\frac{\partial(\alpha_v \rho_v)}{\partial t} + \frac{\partial(\alpha_v \rho_v u_j)}{\partial x_j} = \dot{m}^+ - \dot{m}^- \tag{3}$$

where the source terms, $\dot{m}^+$ and $\dot{m}^-$, represent the mass transfer between evaporation and condensation.

Modified turbulent eddy viscosity

The shear stress transport (SST) turbulence model (Menter, 2009) was used to simulate the two-phase turbulence flow in this paper. The validation studies by Bardina et al. (1997) show SST k–ω turbulence model is suitable for predicting the separation flows with adverse pressure gradients. The turbulent eddy viscosity μ_T in SST k–ω is given as:

$$\mu_T = \frac{\rho_m k}{\omega}\frac{1}{\max\left[\frac{1}{a^*}, \frac{SF_2}{a_1\omega}\right]} \tag{4}$$

where k is the turbulent kinetic energy, ω is the specific dissipation rate, S is the strain rate magnitude, $a*$ is a coefficient which damps the turbulent eddy viscosity representing a low-Reynolds-number correction, $a^* = 1$ in a high-Reynolds-number flow, $a_1 = 0.31$, F_2 is a function that is 1.0 for boundary-layer flows and zero for free shear layers.

A large number of validation studies (Coutier-Delgosha et al., 2003, 2004; Dular et al., 2005; Leroux et al., 2005) show the two-equation turbulence models (such as k–ε *and* k–ω models) may over-predict the turbulent eddy viscosity in the region of cavity closure. The turbulent eddy viscosity Eq. (4) in the SST k–ω turbulence model which blends the k–ε and k–ω also has the same limitation for predicting the unsteady cavitating flow (Li, 2012).

In present study, a mixture density equation $f(\rho_m)$ which was proposed by Reboud et al. (1998) was used to modify the turbulent eddy viscosity in the region of cavity closure. The density corrected method (DCM) was validated by Dular et al. (2005), Leroux et al. (2005) and Coutier-Delgosha et al. (2003, 2004) for the cavitating flow around the hydrafoils.

$$f(\rho_m) = \rho_v + [(\rho_m - \rho_v)/(\rho_l - \rho_v)]^n \cdot (\rho_l - \rho_v) \quad (n = \text{constant and } n \geq 1) \tag{5}$$

Thus, the modified turbulent eddy viscosity in the SST k–ω turbulence model is defined as:

$$\mu_T = \frac{f(\rho_m)k}{\omega}\frac{1}{\max\left[\frac{1}{a^*}, \frac{SF_2}{a_1\omega}\right]} \quad (n = \text{constant and } n \geq 1) \tag{6}$$

Coutier-Delgosha et al. (2003) and Dular et al. (2005) suggested $n = 10$, which was adopted by Liu et al. (2012) in the simulation of a pump–turbine. The SST k–ω based DCM with $n = 10$ (Li, 2012) was also validated by simulating the unsteady dynamic cavitation shedding flow around the NACA0015 hydrofoil. So, $n = 10$ was eventually used in the following simulations.

Homogeneous cavitation model

The Rayleigh–Plesset model was used to model TLV cavitation in an axial flow pump. It provided the basis for the rate equation controlling vapor generation and condensation (Zwart et al., 2004). The homogeneous multiphase model with water and vapor was employed. The size change of a single vapor bubble is driven by the pressure difference between the local static pressure, p, and the saturated vapor pressure p_v. Neglecting the second derivative of the bubble radius, which is important only for rapid bubble acceleration, the Rayleigh–Plesset equation can be written as:

$$\frac{dR}{dt} = \sqrt{\frac{2}{3}\frac{|p_v - p|}{\rho_l}} \tag{7}$$

where R denotes the spherical bubble radius.

The number of bubbles per unit volume, N_b, depends on the direction of the phase change. For the bubble growth, i.e. vaporization, N_b is given by

$$N_b = (1 - \alpha_v)\frac{3\alpha_{nuc}}{4\pi R^3} \tag{8}$$

For condensation, N_b is given by

$$N_b = \frac{3\alpha_v}{4\pi R^3} \tag{9}$$