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Numerical and experimental investigation of tip leakage vortex trajectory and dynamics in an axial flow pump ☆

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ABSTRACT

Tip leakage vortex (TLV) in an axial flow pump was simulated by using the shear-stress transport (SST) k–ω turbulence model with a refined high-quality structured grid at different flow rate conditions. The TLV trajectories were obtained by using the swirling strength method corresponding to the cross-sections of streamlines of the TLV. High-speed photography experiments were conducted to observe the TLV trajectory based on cavitation tracing bubbles in an axial flow pump with a transparent casing. The TLV trajectories predicted by the SST k–ω turbulence model agreed well with the visualization results. The numerical and experimental results show that the starting point of the TLV is near the leading edge at part-load flow rate condition ($Q/Q_{BEP} = 0.85$), and it moves towards the trailing edge to approximately 20% blade chord at the design flow rate condition ($Q/Q_{BEP} = 1.0$). At large flow rate conditions ($Q/Q_{BEP} = 1.2$), the starting point of the TLV shifts to about 40% blade chord, and the relative angle between the TLV trajectory and the blade chord is gradually reduced with the increased flow rate. Detailed statistics of the fluid dynamics of the end-wall shear layer and the TLV at design and off-design conditions were discussed based on the numerical results. The shear layer and jetting flow in the tip gap are highly affected by the pressure difference between the pressure side (PS) and suction side (SS). It was also found that the distributions of static pressure, turbulent kinetic energy (TKE) and vorticity inside the TLV core are associated with the TLV structure which is affected by blade loading and operation conditions of the axial flow pump.

1. Introduction

The radial clearance between impeller tip and casing wall in an axial flow pump is indispensable for its operation. However, the tip clearance is the key source of complex tip leakage flow and the tip leakage vortex (TLV). Tip leakage flow is generated by the pressure difference between the pressure side (PS) and the suction side (SS) of the blade, and it rolls up into a TLV due to the interactions of the end-wall boundary with the main channel flow as well as the relative impeller movement [1]. A TLV can cause several adverse effects on the machine performance, including blockage of the main flow of the passage [2], efficiency losses [3], noise generation [4] and the onset of stall [5]. TLV structure control is expected to improve the performance and reliability of an axial flow impeller [6]. To avoid or at least to reduce such adverse effects of a TLV by design and operation, it is essential to improve the understanding of the physical phenomena associated with the TLV structure and trajectory [7] at different operating conditions.

A TLV in an axial flow pump was firstly observed by Rains in 1954 [1], and he used the clearance flow jet model based on the slender body approximation to describe the TLV, in which three-dimensional tip leakage flow was predicted by a two-dimensional cross-flow. Donghyun et al. [8,9] discussed the effects of tip leakage structures, velocity and pressure fields in a linear cascade, and found that larger tip-gap sizes are indicative to tip leakage cavitation judged by the levels of negative mean pressure and pressure fluctuation. However, the fundamental studies above used is using the simple 2D model. Recently, Wu et al. [10–13] measured the TLV's structure and turbulence in an axial water-jet pump based on the two-dimensional and stereoscopic particle image velocimetry (PIV) technology, and analyzed the turbulence associated with the TLV. The extensive PIV experimental results show that the TLV in the tip region is highly three-dimensional and the velocity and vorticity distributions vary significantly along the blade tip chord direction. Tan et al. and Zhang et al. [14,15] also observed tip attached cavitation, TLV cavitation secondary struc-

☆ Fully documented templates are available in the elsarticle package on CTAN.

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Nomenclature

Q	flow rate (m^3/h)
Q_{BEP}	design flow rate (m^3/h)
H	Head (m)
D_2	impeller diameter (m)
D_3	casing diameter (m)
D_h	hub diameter (m)
h	tip clearance size (m)
p	static pressure (Pa)
r	radius (mm)
r^*	nondimensional radius: $r^* = 2r/D_3$
v_z	axial velocity (m/s)
U	peripheral velocity (m/s)
k	turbulence energy (m^2/s^2)
y^+	nondimensional distance from the wall
ρ	density (kg/m^3)
g	gravitational acceleration (m/s^2)
n_s	specific speed $n_s = \frac{3.65n\sqrt{Q}}{H^{3/4}} = \frac{3.65 \cdot r/min \cdot \sqrt{m^3/s}}{m^{3/4}}$
n	rotational speed (r/min)
M	torque (N m)
Ω	angular velocity (rad/s)
ω	vorticity: $\omega = (\nabla \cdot \overline{v})(s^{-2})$
P	shaft power (kW)
η	pump efficiency (%): $\eta = gQH/(3600M\Omega)$
Ψ	head coefficient: $\Psi = (2\pi)^2 gH(\Omega D_2)^{-2}$
Φ	flow coefficient: $\Phi = 2\pi Q \Omega^{-1} D_2^{-3}$
C_p	pressure coefficient: $c_p = \frac{p}{\frac{1}{2}\rho U_{tip}^2}$

Abbreviations

TLV	tip leakage vortex
SS	suction side
PS	pressure side
LE	leading edge
TE	trailing edge
TKE	turbulent kinetic energy
BEP	best efficiency point

tures, alternate blade cavitation and sheet cavitation at varying flow and pressure conditions using high-speed imaging. It is clear that the tip leakage flow is caused by the pressure difference between the PS and the SS of the blade, which forces a backflow through the blade's tip clearance, and then the backflow becomes a wall jet flow at the suction side of the blade. Finally it is entrained to roll up into a three dimensional TLV with the interaction of the forward passage flow [10–13]. The turbulence associated with the TLV was revealed in the axial waterjet pump using the PIV measurement in the past few years. Yet, the tip gap size in an axial flow pump is extremely small, and it is changing during the operation of the pump because of the inevitable shaft runout. Besides, the reflection of the casing is an important factor which results in a PIV measurement error. Thus, it is very difficult to measure the tip jet flow accurately. Considering some limitations and costs on the PIV experiments, the computational fluid dynamics (CFD) simulation is a effective way to study tip flow and vortex flow in an axial flow pump, especially for revealing the TLV dynamics.

The objective of this study is to compare the simulation of the TLV trajectories and dynamics in an axial flow pump at different flow rate conditions with the previous results of TLV measurement[10–13]. The swirling strength method was used to identify the numerical TLV trajectories. The simulated trajectories were also compared with the high-speed imaging results. Systematic and detailed analyses of flow fields at different chord fraction sections and flow rate conditions were made in the investigated axial flow pump. Although an axial flow pump model was used in our experiments, the studys conclusions are universal.

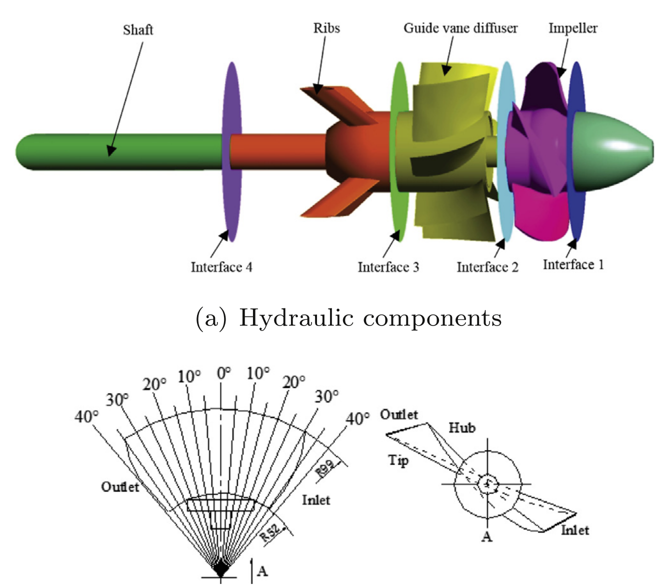
The numerical setup is described in Section 2. The experiment setup is described in Section 3. The results are discussed and compared in Section 4, followed by conclusions in Section 5.

2. Numerical setup, pump model and meshing

2.1. SST k–ω turbulence model, pump geometry and meshing

The SST k–ω model was developed by Menter [16] to effectively blend the robust and accurate formulation of the k–ω model in the near-wall region with the free-stream independence of the k–ϵ model in the far field. The validation studies by Bardina et al. [17] showed the SST k–ω model is suitable for predicting flow separation with adverse pressure gradients. In order to achieve a good tradeoff between accuracy and computational time, the SST k–ω turbulence model in ANSYS CFX was used to simulate the turbulent vortex flow in the present problem. Fig. 1 shows the 3D geometry of the impeller blades of a scale axial flow pump model with the specific speed $n_s = 728$. The geometrical parameters of the pumps are as follows: impeller diameter $D_2 = 0.198$ m, flow rate at best efficiency point $Q_{BEP} = 390\ m^3/h$, head $H = 3.2$ m, rotation speed $n = 1450$ r/min, and a hub ratio $D_h/D_2 = 0.455$. The number of impeller blades is 4, and the number of guide vanes (stator) is 7. The tip clearance size h is 1×10^{-3} m. Tip clearance Reynolds number $Re_h = U_{tip} h v^{-1}$ is 1.38×10^4, blade chord Reynolds number $Re_c = U_{tip} c v^{-1}$ is 1.55×10^6, and inflow Reynolds number $Re_{in} = 4Q(\pi(D_2 - D_{h2}))^{-1} D v^{-1}$ is 7.91×10^5.

As shown in Fig. 1a, the TLV simulation including the flow in the inlet pipe, impeller, guide vane diffuser and discharge elbow was conducted using the sliding mesh technique. The whole computa-



(a) Hydraulic components

(b) Impeller blade geometry

Fig. 1. Pump geometry.