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Wave Energy Harvesting Turbine: Performance Enhancement

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Abstract

The performance of an oscillating water column (OWC) wave energy system depends upon several factors including its turbine performance. OWC turbines have inherently low efficiency and low operating range. Through this work, the shape of a Wells turbine was tried to modify to enhance the efficiency and enlarge the operating range. A numerical investigation with steady state flow condition has been reported to change blade thickness from hub to tip. A new concept to combine NACA0015 and NACA0024 at the hub, the midspan and the tip section were developed and the blade was produced through smooth polynomial so that the blade will have different thickness along the span. A commercial code ANSYS-CFX® v14.5 was used for the simulations. The turbulence k-ω SST model was adopted and the reference turbine performances were compared with existing results reported in the literatures. It was found that a blade thinner at midspan gives lesser flow separation, which helps increasing the turbine performance.

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Keywords: Wells Turbine, Axial flow turbine, Wave energy conversion

Introduction

Wells turbine, which is an axial flow self-rectifying low-pressure air turbine, rotates continuously in one direction by the reversing flow or bi-directional flow. The turbine is used in an oscillating water column (OWC) wave

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energy system. The system performance greatly depends upon the turbine's power production capability under different wave conditions. The wave condition varies in each wave cycle as well as varies with weather condition. The turbine blades are constructed using symmetric airfoil and the angle of attack on the airfoil is 90°. Curran and Gato (1997) compared single-plane and multi-plane wells turbines. Torresi et al. (2008) reported an accurate description of the steady three-dimensional flow-field in a high solidity Wells turbine. Taha and Sawada (2010) compared computational and experimental results with tip clearance to chord length ratios of 0.0056 and 0.0111. It was shown clearly that the tip clearance (TC) significantly influences the turbine performance. Halder and Samad (2014, 2015) studied the effect of uniform tip gap and casing treatment to delay flow separation and to increase turbine performance.

Several authors reported effect of blade-profile shape modification to improve overall efficiency of the Wells turbine numerically or experimentally (Table 1). Takao and Okuhara (2013) reported that increasing blade thickness hub to tip enhanced the turbine performance and improving the stall characteristics. However, no information was found in the literatures on different profiles to delay stall.

In the present work, a numerical study for a new concept to vary blade profile thickness from hub to tip of a Wells turbine has been reported. The results were compared with experimental and numerical results. Different profile thickness for wide flow coefficients were simulated and details analysis has been reported.

Table 1: Blade-profile modifications to enhance Wells turbines performance.

Design modification	Advantage	Description	Profile
Mohamed et al., (2011)	Increased power output (average relative gain: +11.3% Improved efficiency: 1%	Incident angle varied: 5 to 14°	NACA0021
Raghunathan and Tan, (1985)	The NACA0021 produced the peak efficiency. Efficiency drop: ~10% with blade roughened blade.	Thicker and modified aerofoil blades improved performance of the turbine.	NACA0024,NACA0021, NACA0015H,NACA0015, NACA0012
Suzuki and Arakawa, (2008)	Efficiency improved at an angle of attack <7°. Stall angle= 10° was smaller.	fan-shaped blades with different sweep angles	NACA0021, NACA0012
Takao et al., (2006)	Peak efficiency higher	Optimum blade profile: NACA0015	NACA0015,NACA0020CA9,H SIM 15-262133-1576
Thakker and Abdulhadi, (2007)	Higher power output	Preferable rotor blade profile CA9	NACA0015,NACA0020,CA9, HSIM 15-262133-1576
Takao and Okuhara, (2013)	Improved efficiency and stall characteristics	The blade thickness increases gradually from hub to tip	NACA0015, NACA0020,NACA0025

Nomenclature

h	Hub-to-tip ratio	TC	Tip clearance
LE	Leading edge	T	Shaft torque
N	Speed of rotor, rpm	t	Blade thickness
PS	Pressure surface	U_{tip}	Rotor velocity
Q	Volume flow rate	V	Axial velocity
RANSE	Reynolds averaged Navier-Stoke equations	z	Number of rotor blade
RB	Rotor blade	ω	Rotational speed
R_{mid}	Mid Span radius	ρ	Density
R_{tip}	Tip radius	η	Efficiency
s	Turbine solidity	*	Non-dimensional parameter
ΔP*	Pressure drop coefficient		
ΔP*	Stagnation pressure drop		
T*	Torque Coefficient		

Computational Methodology

As stated earlier, the Wells turbine profile was varied from hub-to-tip. Fig. 1 shows the variation of profile thickness using NACA0015 and NACA0024. The left most cases is the original blade (NACA0015) and it has same profile all along the span. Similarly, case A has NACA0024 profile. The other profiles (cases B, C, D and E) are mixed profiles from hub-to-tip, which makes the blade thinker or thicker at different sections. The profiles in between the hub and the midspan, and midspan and tip were produced through a third order polynomial curve.

The turbine performance was numerically investigated by solving the 3D incompressible Reynolds Averaged Navier-Stokes equations (RANSE) and k-ω SST turbulence model. The RANSE were discretised by a finite volume method. The governing equations used for the simulations were continuity, momentum and energy. A finite volume based solver ANSYS CFX® v14.5 was used to solve the equations. High resolution and first order numerical schemes were implemented. The solutions were considered converged once the maximum residual values were reached to a value of 10E-5.

Table 2 shows the specification of the turbine, which rotates at a constant velocity (=2000rpm). A single blade passage with periodic boundary conditions was used for the computations. Table 3 shows meshing and boundary conditions. The computational domain was divided into four and six times of the blade chord length along the upstream and downstream of the rotor, respectively (Fig. 2).

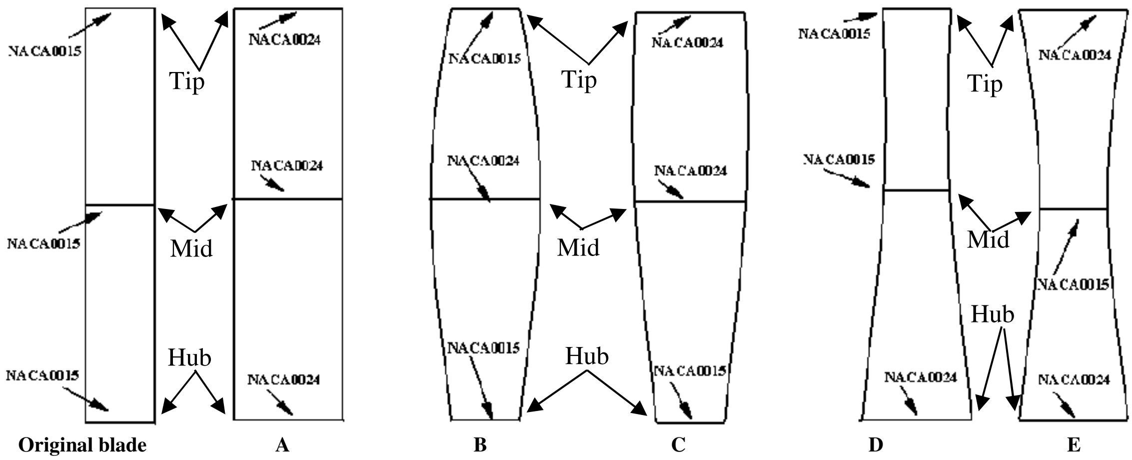

Fig. 1. Variation of blade profile thickness from hub-to-tip

Table 2. Specifications of rotor.

Parameter	Dimension
Blade profile	NACA0015,NACA0024
Blade chord length, C	0.125m
Blade number, z	8
Blade thickness, t	15% of C
Solidity	0.644m
Tip radius	0.3m
Hub radius	0.2m
Hub/Tip ratio	0.67m
Mean radius	0.25m
Tip clerance of the chord length	1mm

Result and Discussion

Present numerical results were validated with an existing results available in the literatures (Curran and Gato, 1997; Torresi et al., 2008) (Fig. 3) and match well with the results of original blade (Fig. 3). Fig. 3(a) shows the torque coefficient (T*) variation with different blade profiles from hub-to-tip. The torque coefficient is highest for a blade having midspan-thickness lower (NACA0015) while the tip and hub sections were wider (NACA0024). Fig. 3(b) shows that the peak efficiency is almost constant for all the cases.

Table 3. Meshing and boundary conditions.

Parameter	Description
Flow domain	Single blade
Interface	Periodic
Mesh/Nature	Unstructured
Fluid	Air
Turbulence model	k-Ω SST
Inlet	Velocity
Outlet	Pressure
Hub	Wall
Casing	Wall
Blade	Wall
Residual RMS criteria	1x10-5
Mass imbalance	0.001

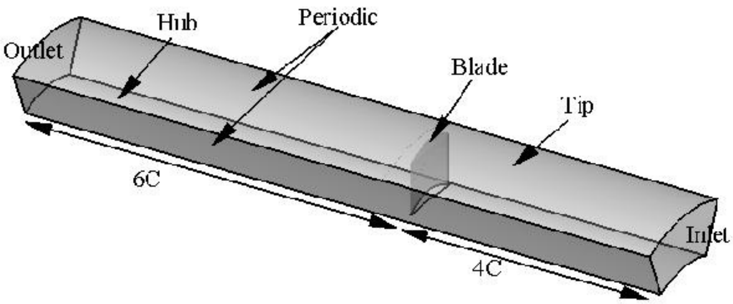

Fig. 2. Computational domain

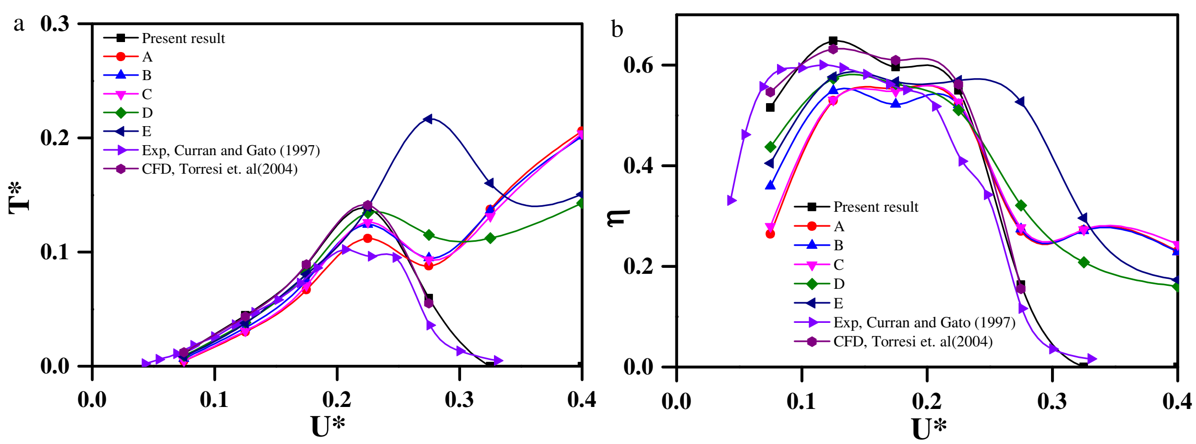

Fig. 3. (a) Torque coefficient; (b) Efficiency

Fig. 4 shows the static pressure distribution and blade-loading curve at the midspan of the blade passage. At lower flow coefficients, the flow is attached and there is not much variation in pressure on the blade surfaces. The airfoil profiles are taken at Midspan. At U*=0.275, the blade E is showing higher pressure on suction surface and hence this gives a delayed separation. The separated flow can have lower power transfer to the blade and hence has lower torque (Fig 3(a)). It can also be found that the torque is higher at higher flow coefficient and hence it gives a wider operating range. This observation is consistent with an existing study (Mohamed et al., 2011).

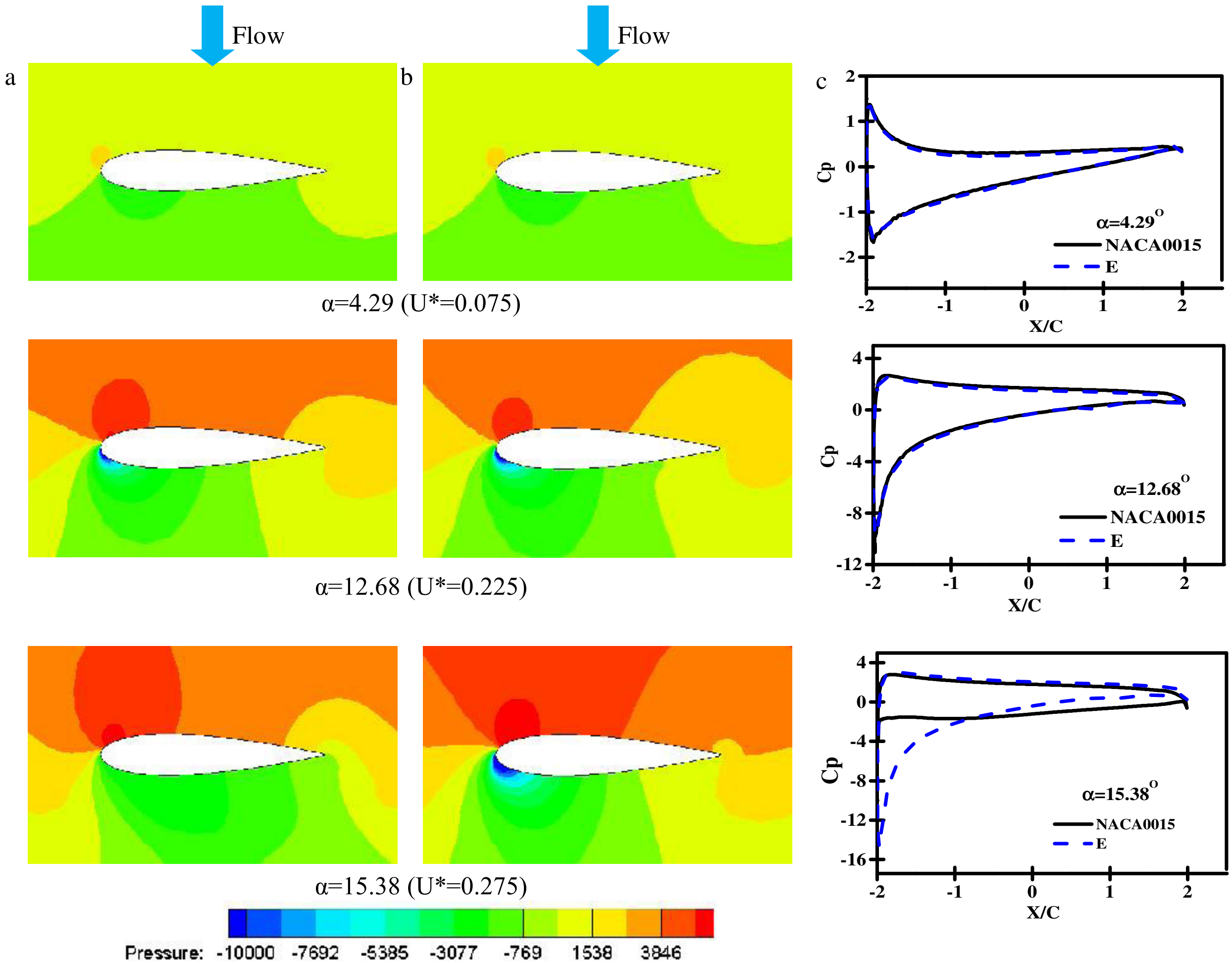

Fig. 4. Static pressure distribution: a) Original blade (NACA0015), b) Modified blade (E) and (c) blade loading with original and Modified blade (E)

Fig. 5 shows the streamline pattern at the midspan of the blade passage. At lower flow coefficient (U*=0.075), there is no circulation at the trailing edge and the streamlines are attached to the blade surfaces. As the flow coefficient increases, flow separation occurs at the trailing edge. With higher flow coefficient (U*=0.275), the flow circulation is found at the trailing edge. However, the size of the circulation is much larger in case of standard aerofoil (NACA0015). As a result, the torque is lower in the original aerofoil (Fig. 3(b)). It was already reported that increasing blade thickness enhances turbine performance and improves the stall characteristics (Takao and Okuhara, 2013).

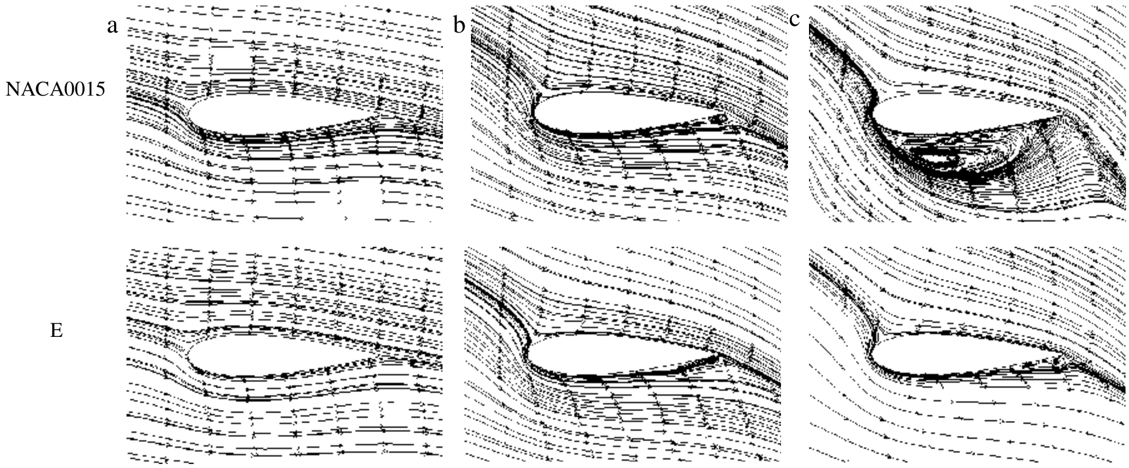

Fig. 5. Streamline line on the midspan of the flow passage: (a) U*=0.075; (b) U*=0.225; (c) U*=0.275;

Conclusion

A novel concept to vary blade thickness from hub-to-tip of a Wells turbine blade was reported and the blade performance has been evaluated. Reynolds averaged Navier-Stoke equations were solved with *k-w* SST model. The numerical results were validated with the existing experimental results.

It was found that a lower thickness at the midsection produces larger torque and hence larger power. The reason behind the performance enhancement was the delayed flow separation and increased blade loading.

It was found that the peak efficiency was not changed much because of blade shape change. An increase operating range of the turbine was also found through this investigation.

By further shape modification with larger number of parameters, it can produce further power enhancement.

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