HYDRODYNAMIC DESIGN OF TIDAL CURRENT TURBINE AND THE EFFECT OF SOLIDITY ON PERFORMANCE

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ABSTRACT

Blade element momentum (BEM) flow model in conjunction with pattern search optimization algorithm embedded in the National Renewable Energy Laboratory (NREL) tool HARP_Opt was used to design a horizontal axis tidal current turbine (HATCT). The numerical method was validated with the experimental data and a good agreement was achieved. The designed turbine has a 3 bladed rotor of 4 meters diameter and rated mechanical power of 20 kW. Performance metrics of the rotor for steady and uniform flow was simulated at flow speeds from 0.5-3.5 m/s. The turbine achieved its rated power at a tip speed ratio (TSR) 5.7 with a peak C_P value of 0.47 and peak thrust of 16.7 kN-m. Additionally, a series of simulations were performed at TSR from 1-10 to obtain performance curve for the turbine at a design current velocity of 1.5 m/s. Effect of solidity on performance was quantified by varying the number of turbine blades. The value of C_P increased by 1.5% with increasing the number of blades from 2 to 3. The value of C_P further increased by only 0.2% with increase in number of blades from 3 to 4. The value of turbine thrust was minimally effected by increase in the number of blades. However, the value of thrust per blade increased with a reduction in number of blades. The increase in flap moment and thrust per blade with reduction in number of blades could have serious consequences for the structural integrity of the turbine.

KEYWORDS: Tidal Current Turbine, Marine Hydrokinetic Turbine, Hydrodynamic Design, Blade Element Momentum (BEM), Solidity, Performance.

INTRODUCTION

Tidal energy has the potential to contribute electricity to the grids in numerous coastal countries. Estimates of global tidal energy potential greatly vary with figures ranging from100-17500 TWh/yr (Borthwick, 2016; Laws & Epps, 2016; Pelc & Fujita, 2002). A significant portion of this potential is located in shallow basins which provide an attractive opportunity for the future energy supplies (Borthwick, 2016). UK has one of the largest resources estimate with more than 50.2 TWh/yr (Pelc & Fujita, 2002), whereas whole of the Western Europe have a resource potential of up to 105.4 TWh/yr (Pelc & Fujita, 2002). The tidal energy resources are globally spread and every coastal country has some part in the global resource potential. Several locations with significant in stream tidal energy potential are also available in Pakistan. An estimated power of 1.1 GW can be produced from the tidal resources of the creek system in the Indus delta (Khan, 2010). Tidal currents in these creeks have flow velocities from 2-2.5 m/s with some locations containing flow speeds as high as 4 m/s.

The current status of tidal energy development is such that some of the prototype systems are being tested in the intended environment (Magagna & Uihlein, 2015). Whereas, some of the demonstration systems are operating in the operational environment at pre-commercial (Magagna & Uihlein, 2015) and commercial scales (Neill *et al.*, 2017). The technology readiness level (TRL) of tidal energy is as in Fig. 1. Apart from the industrial breakthroughs and R&D efforts, extensive academic research is also going on to develop the tidal energy technology. The academic research conducted during the past decade can be easily divided into research related to resource characterization and technology development. Research related to the resource characterization are utilizing site measurements and numerical modeling for the identification and characterization of potential sites. In the area of numerical modeling for resource characterization, Bryden and Melville (2004) and Couch and Bryden (2004) developed a 1D numerical model. This model was intended to estimate the maximum energy extraction from a tide in open channel. Garrett and Cummins (2005) developed equation for the estimation of maximum energy extraction from a channel. Resource characterization is still a very active research area and

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the researchers are trying to map the global tidal energy resource and characterize the potential sites.

TRL 9	Full commercial application, technology available for consumer
TRL 8	First of kind commercial system. Manufacturing issues solved
TRL 7	Demonstration system operating in operational environment at pre-commercial scale.
TRL 6	Prototype system tested in intended environment
TRL 5	Large scale prototype tested in intended environment
TRL 4	Small scale prototype built and tested in a laboratory environment
TRL 3	Applied research. First laboratory test completed, proof of concept
TRL 2	Technology formulation. Concept and application have been formulated
TRL 1	Basic research. Principals postulated and observed but no experimental proof available

Figure 1: TRL of ocean energy technologies (Magagna & Uihlein, 2015)

Research related to technology development started with an understanding of the hydrodynamics (Batten *et al.*, 2006) that later translated into the design and development of turbines (Bir *et al.*, 2011). The design and development was followed by the quest to predict the device performance before its development and after installation. The numerical methods mostly used by the researchers for the performance prediction of TCTs include Blade Element Momentum (BEM) and Computational Fluid Dynamic (CFD) simulations of the Reynolds Navier Stokes Equation (RANS). More recently, both these model have been combined in a new model called the coupled BEM-CFD model to take advantage of the strength of each model (Malki *et al.*, 2014; Turnock *et al.*, 2011). The BEM method is computationally very efficient and has been widely used for design studies of the TCTs. Batten *et al.* (2007) developed a BEM model for the design of TCT and compared its performance with experimental data (Batten *et al.*, 2008). The model results agreed well with the experimental data. Several design and performance parameters like cavitation inception and the effect of velocity profile, blade fouling and blade pitch angle on turbine performance were quantified. More recently, O'Rourke *et al.* (2015) used the BEM model in unsteady formulation to quantify the effect of velocity shear and yaw misalignment on the performance of TCT. This model also showed a good match with experimental data. In addition to its computational efficiency and ease of use, the BEM model has the advantage that it can be easily combined with an optimization algorithm and economic model. RANS (CFD) based numerical models are computationally very expensive. But the recent tremendous increase in computational power has resulted in the increased use of this model for the performance studies like (Hee Jo *et al.*, 2012; Lawson *et al.*, 2011; Li *et al.*, 2015; Ma *et al.*, 2016; Tian *et al.*, 2016; Zhang *et al.*, 2014) with acceptable accuracy. Experimental studies have also been used by (Batten *et al.*, 2007; Hill *et al.*, 2014; O'Doherty *et al.*, 2009) to evaluate the performance metrics of their devices. Some real sea test have also been reported in the recent literature by (Jing *et al.*, 2017; Liu *et al.*, 2016). Research related to performance and design studies has explored the basic design parameters affecting the performance, hydrodynamics and novel designs.

This paper outlines the design and optimization of Horizontal axis tidal current turbine (HATCT) and the effect of solidity on turbine performance. The turbine blades are designed using a Blade element momentum (BEM) flow model and optimized for efficiency with pattern search optimization algorithm embedded in the National Renewable Energy Laboratory (NREL) tool HARP_Opt. The turbine design was carried out at uniform and steady flow for a velocity range of 0.5-3.5 m/s. Additionally, the performance curve for the turbine for TSR 1-10 was also simulated. Performance data computed from the numerical model was validated with experimental data and a good match was observed. The effect of variation in number of blades on the turbine performance was also quantified using BEM flow model. Because, the number of blades is an important economic consideration and limited research has been conducted to investigate its effect on the performance and design of turbine. This study adds new knowledge on this important topic through a computationally efficient numerical method.

Hydrodynamic design procedure Power Output

The hydrokinetic energy available in any stream is determined as (Schubel & Crossley, 2012):

$$P_{expect} = \left(\frac{\rho \pi D^2 V^3}{8} \right) \tag{1}$$

According to Bitz limit only 16/27 of this available power can be extracted by a hydrokinetic turbine(Vennell, 2013) and this is the maximum power coeffecient (C_p) that a stand alond TCT can achieve. In addition, the power train efficiency (η) sets another theoretical upper limit of 0.9 for the maximum power can that can be achieved.

Blade Air Foil

Design of the turbine blade involve specific foil shapes located along the blade span having suitable twist angles and chord lengths. The most important aerodynamic considerations for selecting an airfoil is the glide ratio or turbine effeciency (C_L/C_D). But there are other aspects like the sensitivity to surface roughness which must also be duly considered. The tidal turbine operates in water and their blades face more loads as compared to wind turbine due to the higher density of water. For tidal application airfoils with relatively larger percentage thicnkess are recommended.

Thicker airfoils also have the advantage to easily accomodate other necessary arrangements to make the blade more strong. Sudden increase in current velocity will create sudden variation in the local angle of attack (AOA). This sudden variation may force the turbine into a pre-stalled or stalled zone. Therefore, airfoils having low probability of stall should be selected for tidal turbine application. Some special purpose airfoils are being developed for tidal turbines (Goundar & Ahmed, 2013; Singh & Choi, 2014). However, this study uses a 24% thick NACA 63-series airfoil that has a comparatively large minimum pressure coeffecient to avoid cavitation. This airfoil is less sensitive to leading edge roughness and known for a delayed stall. This airfoil has previosly been used in the famous National Renewable Energy Laboratory (NREL) 550 Kw hydrokinetic turbine(Lawson *et al.*, 2011).

Tip Losses

Rotating turbine blades form vertices at the blade tip. These vertices are a source of loss and causes a decrease in the lif force of the blade. The blade tip losses are determined using eqution (2) presented by Ludwing Prandtl (Hee Jo *et al.*, 2012):

$$f_{tip} = \frac{2}{\pi} \cos^{-1} \left[e^{\left(\left(N/2 \right)(1-\mu)/\mu \right) \sqrt{1+\left(\lambda\mu^2/\left(1-a^2\right) \right)}} \right] \tag{2}$$

Where, "μ" is "r/R" with "R" representing the total radius of the turbine and "r" the radius of the respective blade element. The axial component of velocity is represented by U_{disk} determined as.

$$U_{disk} = U_\infty - aU_\infty \tag{3}$$

Where 'a' is the axial induction factor, U∞ is the free stream velocity and 'N' represents the number of blades.

Blade Chord

Mathematical expression for the determination of blade chord at a blade element is provided by the Schmitz formula (Hee Jo *et al.*, 2012):

$$C = \frac{16\pi r}{C_L B} sin^2 \left(\frac{1}{3} tan^{-1} \left(\frac{R}{\lambda r} \right) \right) \tag{4}$$

Where C is the blade chord at a blade element at radius r, number of blades in the turbine is B, λ represents the tip speed ratio and R is the radius of turbine. "C_L" is the design co-efficient of lift and its value is taken as 80% of the value corresponding to the design of angle of attack (AOA) from the drag polar data of the airfoil.

Blade Twist

Fig.2 provides a description of the blade angles. In this Figure α is the design AOA taken as the AOA for maximum glide ratio. The incidence angle is 'ϕ', (V(1-a)) is the component of velocity acting in the axial direction. Ω_r (1+a') is the tangential velocity component and 'β' is blade twist. Twist at a blade element is determined from equation (5) (McCosker, 2012).

$$\beta = \frac{2}{3} tan^{-1} \left(\frac{R}{\lambda r} \right) - \alpha \tag{5}$$

The relation of twist angle β, inlet angle ϕ, and AOA **α** is as in equation (6) and (7) (Jo *et al.*, 2013).

$$\phi = \alpha + \beta \tag{6}$$

$$\phi = tan^{-1} \left(\frac{1-a}{\lambda\mu\left(1+a'\right)} \right) \tag{7}$$

Where, *a* is the axial flow induction factor, a' is the tangential flow induction factor and μ is the ratio of radial direction from hub to tip and is defined as r/R.

Tip Speed Ratio (TSR) And Angular Velocity

Tip speed ratio (TSR) is a non-dimensional parameter representing the ratio of blade tip velocity to relative flow velocity and mathematically expressed as (Vermaak *et al.*, 2014):

$$\lambda = \frac{\Omega R}{V} \tag{8}$$

Where, "Ω" is the angular vecoity or speed of the turbine in rads/s, mathematically expressed as:

$$\Omega = \frac{2\pi N}{60} \tag{9}$$

Turbines with higher optimum tip speeds are more effecient but they are subjected to higher hydrodynamic and centrifugal stresses (Nasir Mehmood & Khan, 2012). Blades designed for higher TSR have comparatively smaller chords that will require less material, will have low weight and cost.

Hydrodynamic Performance Parameters

The important hydrodynamic parameters include lift, drag, thrust and torque. Equation for calculating local/ sectional lift, drag, thrust and torque are as in equation (10) to (13):

$$dL = 1/2\rho U^2 C C_L dr \tag{10}$$

$$dD = 1/2\rho U^2 C C_D dr \tag{11}$$

$$dT = 1/2\rho U^2 \left(C_L Cos\varnothing + C_D Sin\varnothing\right) Cdr \tag{12}$$

$$dQ = 1/2\rho U^2 \left(C_L Sin\varnothing - C_D Cos\varnothing\right) Crdr \tag{13}$$

In the blade element theory model, sectional lift, drag, thrust and torque are calculated at every blade element/ station. Sum of all the sectional parameters gives the total values for a single blade and is multiplied by the number of blades to get the values of hydrodynamic parameters for the turbine. The relations for calculating the thrust coefficient (C_T) and torque coefficient (C_q) are as in equation (14) and (15):

$$C_T = C_L Cos\varnothing + C_D Sin\varnothing \tag{14}$$

$$C_Q = C_L Sin\varnothing - C_D Cos\varnothing \tag{15}$$

Geometric Design

Three dimensional geometric model of the turbine was created in Autodesk Inventor Professional shown in Fig. 3.

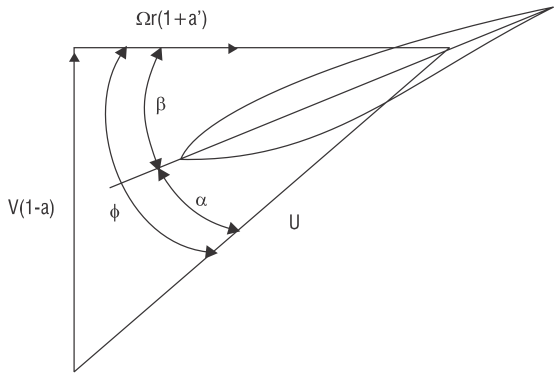

Figure 2: Angle definition of an airfoil

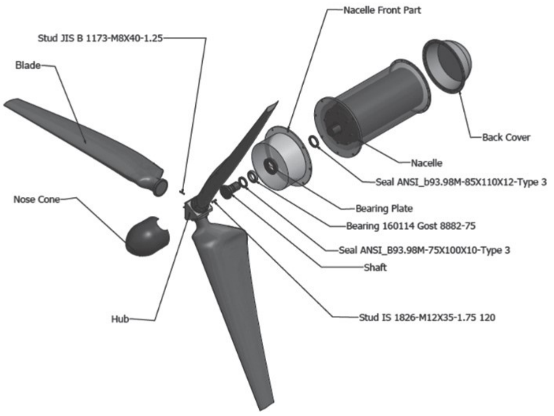

Figure 3: 3D exploded view of the designed

The blade has been modeled on the basis of design sequence provided in table 1. Station 1-4 is the blade transition region. The transition region does not contribute to the power output but adds strength to the turbine blade. The blade to hub joint is a bolted connection. Hub in a turbine holds the blades and prevents unwanted movements of the blade. The design criterion for the hub is the "principle of direct and short force transmission path" by Phal and Bietz (Matousek, (2012)). The shaft was designed using ASME design code for ductile material. Purpose of the Nacelle is to enclose the alternator and protect it from the exposure to water.

Validation

A 2 bladed experimental model turbine of 0.5m diameter was designed to compare the numerical results of this study with its experimental data. This turbine model is referenced as RM1 and is a non-proprietary reference model developed as a study object for open source research by the reference model project sponsored by U.S department of energy. Details of the turbine design and experimental data are available in (Hill *et al.*, 2014). Fig. 4 provides a comparison of coefficient of power numerically computed using a BEM model in this study and the experimental data (Hill *et al.*, 2014).

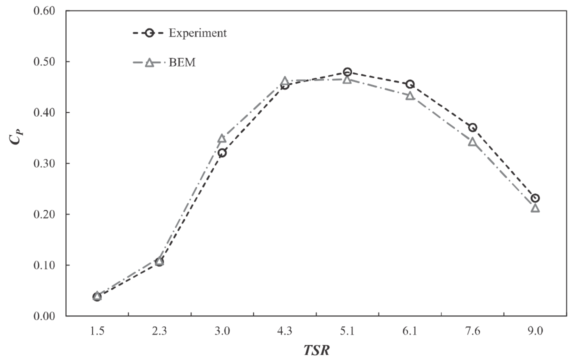

Figure 4: Comparison of experimental and numerically simulated performance curve.

The average relative error C_P between BEM computations in this study and experiment is 5.3% with a maximum error of 9.5% at TSR 2.27. Other, BEM performance studies like (Liu *et al.*, 2017) have also provided an error of up to 8% that has been widely accepted by the research community. Hence, the BEM predictions for C_P agrees well with the experimental data.

RESULTS AND DISCUSSION

Hydrodynamic Design Results

Horizontal Axis Rotor Performance Optimization (HARP_Opt) code was used to design a 4m diameter, 3 bladed turbine with a rated mechanical power of 20 Kw. Blade element momentum theory (BEM) flow model and pattern search optimization algorithm was utilized to optimize the turbine design for a steady and uniform flow condition. The optimization objective for this case was the turbine efficiency to be optimized for a range of flow speed from 0.5-3.5 m/s. The shape of the blade was defined through curve fits. The pattern search algorithm quantified the optimum twist and chord distribution for the blade elements. Results of the optimized chord and twist distribution are as in table 1. The control configuration for the turbine designed in this study is variable speed variable pitch (pitch to feather). This configuration controls the rotor to operate at the optimal TSR (max efficiency) till the rotor achieve its rated power or meet the minimum or maximum rotor speed. In the current design the rotor achieved its rated power of 20 Kw at flow velocity of 1.9 m/s. After achieving rated power, it is kept constant through the pitching mechanism by decreasing the AOA.

The hydrodynamic performance of TCT is represented either by the turbine power and thrust or their coefficients. The performance of the turbine for flow speeds of 0.5-3.5 m/s is as in Fig. 5. The optimum TSR for the designed turbine is 5.7. It is also a common practice to describe the turbine performance in terms of non-dimensional parameter TSR. A representative design velocity of 1.5 m/s was selected because a velocity of 1.5-2 m/s is generally considered adequate for economic power production (Kempener & Neumann, 2014). The perfromance curve of the designed turbine at flow speed of 1.5 m/s for TSR range of 1-10 is as in Fig. 6.

Number of Turbine Blades

The number of blades have an effect on the stability, efficiency and cost considerations. A rotating disc is the most stable mechanical component (Hee Jo *et al.*, 2012). Three blades in a turbine can satisfy the minimum stability requirements and enables stable operation of the turbine like a rotating disc Solidity is the ratio of the area covered by the blade to the total swept area of the turbine and is mathematically expressed as:

Where, B is the turbine number of blades, C is the mean chord and R is the radius of the turbine. Solidity for a turbine can be varied by the variation in number of blades or chord length. The solidity in tidal turbine has not been much investigated for its effects on tidal turbine performance and most of the knowledge is transferred from the wind turbine research. To the best of the author's knowledge, the only other study has been performed by

Table 1: Characteristics of the designed turbine blade

Station	r/R	Radius	Pre-Twist	Chord	% Thick	Thickness	Airfoil
(-)	(-)	(m)	(deg)	(m)	(100*t/c)	(m)	(-)
1	0.12	0.230	13.0	0.100	100	0.10	NACA6_1000
2	0.15	0.290	13.0	0.102	98.2	0.10	NACA6_0982
3	0.18	0.350	13.0	0.162	55.3	0.09	NACA6_0553
4	0.21	0.410	13.0	0.233	30.8	0.07	NACA6_0308
5	0.24	0.470	13.0	0.279	24	0.07	NACA6_0240
6	0.27	0.530	13.0	0.306	24	0.07	NACA6_0240
7	0.30	0.590	13.0	0.322	24	0.08	NACA6_0240
8	0.33	0.650	13.0	0.330	24	0.08	NACA6_0240
9	0.36	0.710	13.0	0.334	24	0.08	NACA6_0240
10	0.39	0.770	12.3	0.334	24	0.08	NACA6_0240
11	0.42	0.830	11.6	0.331	24	0.08	NACA6_0240
12	0.45	0.890	11.0	0.326	24	0.08	NACA6_0240
13	0.48	0.950	10.4	0.320	24	0.08	NACA6_0240
14	0.51	1.010	9.9	0.313	24	0.08	NACA6_0240
15	0.54	1.070	9.4	0.305	24	0.07	NACA6_0240
16	0.57	1.130	8.9	0.295	24	0.07	NACA6_0240
17	0.60	1.190	8.4	0.285	24	0.07	NACA6_0240
18	0.63	1.250	8.0	0.275	24	0.07	NACA6_0240
19	0.66	1.310	7.6	0.264	24	0.06	NACA6_0240
20	0.69	1.370	7.2	0.252	24	0.06	NACA6_0240
21	0.72	1.430	6.8	0.239	24	0.06	NACA6_0240
22	0.75	1.490	6.3	0.227	24	0.05	NACA6_0240
23	0.78	1.550	5.9	0.213	24	0.05	NACA6_0240
24	0.81	1.610	5.5	0.199	24	0.05	NACA6_0240
25	0.84	1.670	5.1	0.184	24	0.04	NACA6_0240
26	0.87	1.730	4.6	0.169	24	0.04	NACA6_0240
27	0.90	1.790	4.1	0.153	24	0.04	NACA6_0240
28	0.93	1.850	3.6	0.137	24	0.03	NACA6_0240
29	0.96	1.910	3.1	0.120	24	0.03	NACA6_0240
30	0.99	1.970	2.5	0.102	24	0.02	NACA6_0240

Morris (2014). This study used the Reynolds Averaged Navier Stokes (RANS) based computational fluid dynamic (CFD) model which is computationally very expensive and laborious. Since, the decision on turbine number of blades is an initial design stage decision therefore the use of computationally expensive RANS (CFD) model is not advisable for such studies. Further investigations are required to add new knowledge on this important topic and utilize efficient numerical method to contribute towards the convergence of tidal energy technology. The main focal point of the current study is to fill this gap and therefore used a BEM flow model. Three tidal turbines with similar blade design as in table 1 were simulated by varying the number of blades as 2, 3 and 4 to study the effect of solidity on turbine performance. Results of the study in the form of performance curves are as in Fig. 7. The optimum TSR for the 2 bladed device was near TSR 8, whereas for the 3 and 4 bladed device it was near TSR 5. It is induced from this observation that devices with less number of blades have a higher optimal TSR and therefore require smaller size gear box. Smaller gear box size and number of blades reduces the cost of the

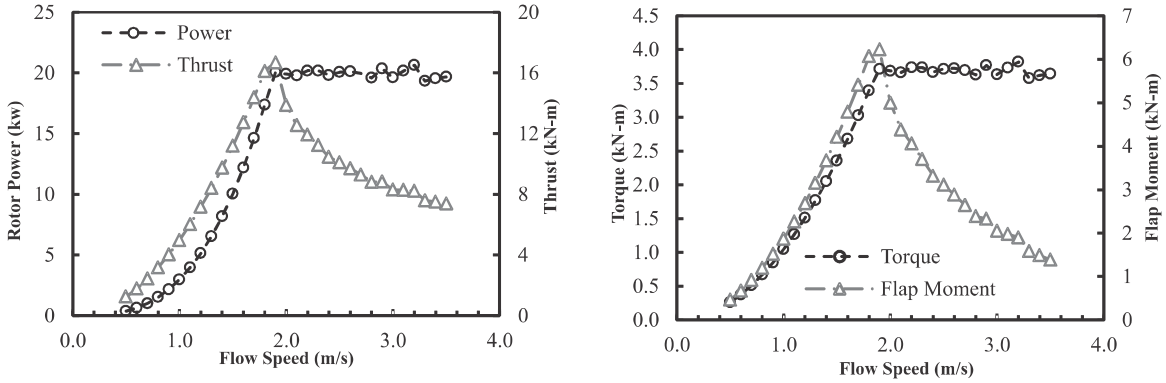

Figure 5: Performance of the designed turbine for flow speed of 0.5-3.5 m/s.

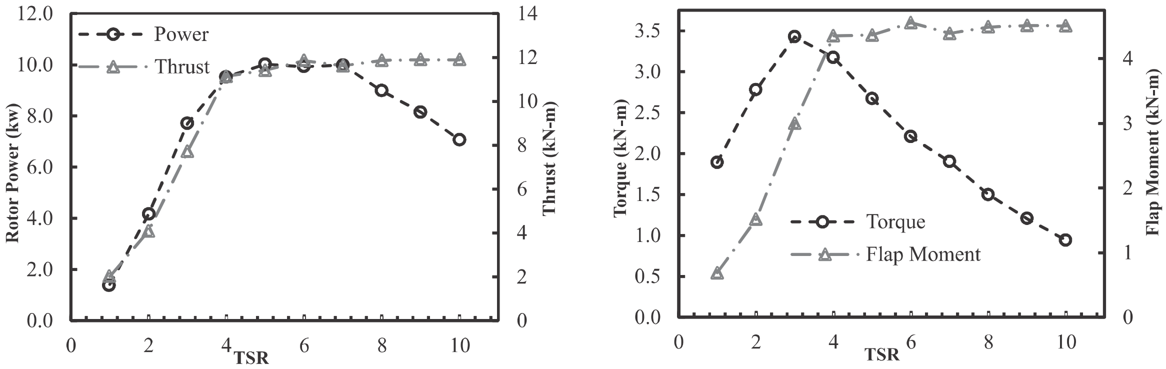

Figure 6: Performance of the designed turbine at flow velocity of 1.5 m/s for TSR range 1-10.

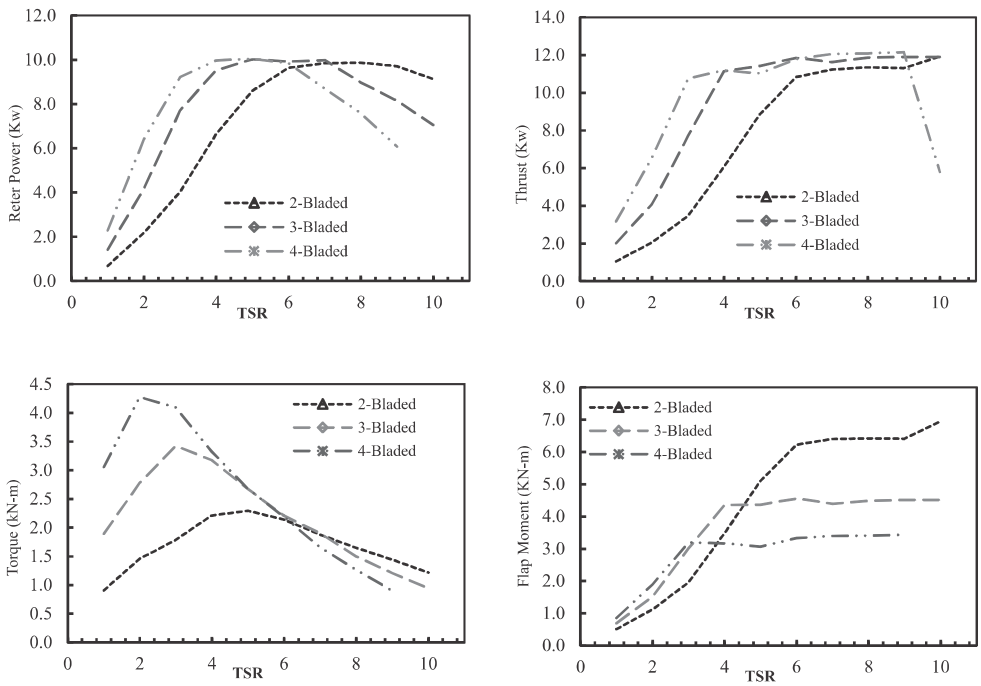

Figure 7: Effect of number of blades on performance at flow speed of 1.5 m/s

turbine. On the other hand, devices operating at higher TSR creates a more significant wake that could make it complicated for operation in an array. The chances of cavitation are also higher at higher TSR. Further, cyclic loadings will be induced on

Figure 7: Effect of number of blades on performance at flow speed of 1.5 m/s at TSR range 1- the drive train and the tower effects will also be more prominent for large two bladed devices operating in a current shear. The value of C_P in this study was observed to increase by 1.5% with increasing the number of blades from 2 to 3. The value of C_P further increased by only 0.2% with an increase in the blades from 3 to 4. Hence, it is clearly established that increasing the number of blades from 3 to 4 does not cause any appreciable increase to the turbine efficiency. The value of C_q increased by 49% and 86% with increasing the number of blade from 2-3 and 2-4 respectively. The value of thrust is minimally effected by the increase in number of blade. However, the value of thrust per blade reduced with increasing the number of blades. The value of thrust continuously increased for 2 bladed turbine after passing the optimum TSR, stayed nearly constant for the 3 bladed device and decreased for the 4 bladed turbine. The implication of this finding is that in case of failure, turbines with more solid rotors would have to withstand lower loads.

CONCLUSIONS

Blade Element Momentum (BEM) flow model in conjunction with pattern search optimization algorithm was used to design a 3 bladed tidal current turbine. The numerical method was 1st validated with experimental data of DOE RM1 model scale turbine. This validated method was then used to evaluate the performance matric of the designed turbine for uniform and steady flow condition for a tidal current velocity range of 0.5-3.5 m/s. The turbine achieved its rated power at a TSR 5.7. The peak C_P value of the turbine was 0.47 with a peak thrust of 16.7 kN-m. Additionally, the performance curve for the designed turbine for TSR range of 1-10 was also evaluated at a design velocity of 1.5 m/s. The effect of variation in number of blades on the turbine performance was investigated by simulating the performance of a 2, 3 and 4 bladed device of the same design that was designed in this work. An increase to the number of blades beyond 3 did not cause any increase to the efficiency of the turbine. However, an appreciable increase in torque was achieved by increasing the number of blades. Thrust of the turbine was minimally effected by the increase in number of blade. However, the value of thrust per blade considerably increased with a reduction in solidity. The implication of this finding is that 2 bladed turbines would be subjected to continuously increasing loads with increasing current speed. For a 3 bladed device the load remain constant after a certain TSR and for a 4 bladed device the load decreased with increasing TSR. It can be induced from this finding that in case of failure, turbines with more solid rotors would have to withstand lower load. The increase in flap moment and thrust per blade with reduction in number of blades could have serious consequences for the structural integrity of the turbine. Moreover, the increased deflection due to increased flap moment will affect the designed output power of the blade. The addition of the fourth blade caused a negligible increases to the efficiency. Therefore the additional manufacturing cost of the fourth blade cannot be justified. The optimal number of blades for a turbine may vary from one site to another and from location to location in an array. However, the effect of variation in number of blades on the structural integrity of the turbine needs further investigation by using a fluid structure interaction model.

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