

Performance Prediction of a Free Stream Tidal Turbine with Composite Bend-Twist Coupled Blades

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Abstract

Free stream tidal turbines are a source of growing interest in the marine renewable energy field. Some designs that use an actuated mechanism to control individual blade pitch, as the tidal current varies and even during a revolutionary cycle, face the challenge of ensuring high reliability under the loads imposed on the device in the harsh sub sea environment. Adaptive materials have been used in the wind industry to create bend-twist coupled blades in an effort to increase turbine efficiency, thus increasing annual energy capture and reducing blade fatigue loads. A computationally efficient, yet realistic, empirical model has been developed in order to estimate the amount of induced twist present on a bend-twist coupled blade in a tidal stream. The method has been incorporated into a Blade Element Momentum code, modified to predict the performance of free stream tidal devices. Results show that the use of bend-twist coupled blades could improve the energy capture of a 20m free stream horizontal axis tidal turbine by around 12.5% without a significant increase in production costs.

1. INTRODUCTION

Horizontal Axis Tidal Turbine (HATT) design has to confront problems that do not occur when operating such a system in air, and as a result the blade section, plan form, and twist will differ subtly from those used on a Horizontal Axis Wind Turbine (HAWT). Due to differences in fluid density, for instance, the thrust on a HATT is more than three times greater than that experienced by a HAWT of a given rated power, despite the tidal device having a significantly smaller swept area. Other alterations to the blade load regime that differ from those present on a HATT include cavitation, free surface boundary layer interference, wave loading, and variation in static pressure and velocity across the vertical water column.

There is an interest in the use of composite materials for potential improvements in hydrodynamic and structural performance of HATTs. In addition to the obvious advantages of high strength-to-mass and strength-to-stiffness ratios, anisotropy of the laminated fibre composites can be designed to allow three dimensional tailoring of the blade deformation. Passive control of a turbine blade can be achieved by integrating advanced composite materials into the blade structure and taking

advantage of the directionality of anisotropic composite material. Anisotropic structures show different levels of elastic coupling, depending on the ply angle in the layers that comprise such a material.

A structure that undergoes both bending and twisting due to a pure bending load is said to exhibit bend-twist coupling. This type of behaviour has been identified as a potential method for load reduction and enhanced stall control of wind turbines [1-6]. Although many advances have been made in the use of composites to improve the performance of aerospace structures, only limited work has been carried out in the marine industry [7, 8]. Preliminary studies have shown that the use of bend-twist coupled blades may also improve the performance of HATTs [9].

In the course of one revolution, the HATT blade will encounter a non-uniform inflow velocity, with implications for the design of the blade. Using a relatively simple bend-twist coupled blade could enable the turbine to deal with the severe complex loads induced upon it whilst also maximizing the energy capture of the device

The aim of this work is to develop a computationally efficient model in order to estimate the amount of induced twist present on a bend-twist coupled blade in a tidal stream. This model takes into account the effect on the induced twist of fibre orientation, blade loading and cross section, material mechanical properties, and shell thickness. The method has been incorporated into a Blade Element Momentum code, modified to predict the performance of free stream tidal devices; such that the performance of a HATT with composite bend-twist coupled blades could be estimated.

There is an apparent increase in annual energy extraction, similar to that found with a variable pitch blade, and a decrease in the fatigue loads experienced by the turbine blade. The modification to the structure should not significantly increase blade cost but would remove the need for a complex and expensive pitch varying mechanism.

It has been shown that, when compared to a free stream tidal turbine with fixed blades of a similar configuration, a HATT that utilises composite bend-twist couple blades can reduced fatigue loading, optimise performance and increase the annual energy capture.

2. BEND-TWIST COUPLING

An adaptive textile composite, sometimes referred to as a smart material or intelligent material, is a structure tailored to exhibit desirable elastic deformation behaviour not necessarily proportional to the imposed load. An example of such a structure would be a box beam so tailored that an imposed cantilever load results in twisting as well as bending, although no torsional load was imposed. Reversible behaviour would be exhibited if, in addition, an imposed torsional loads results in bending as well as twisting, although no cantilever load was imposed. This structure has bend-twist coupling.

In the process of improving HAWT performance, new methods are continually being sought for capturing additional amounts of energy, alleviating loads and controlling the rotor. Composites offer several advantages over metals such as superior fatigue characteristics, high stiffness to weight ratio, construction of complex geometries, and a potential for aeroelastic tailoring. Composite blade elastic coupling can not only be tightly controlled but varied over the span by appropriate selection of ply angles, thicknesses and spanwise layup. Resulting anisotropy provides direct bending-axial torsion elastic coupling not possible with metallic blades. There are added benefits of such as a dramatic reduction in manufacturing costs when compared to steel. A 60-70% reduction in production costs was seen in the marine propeller industry upon embarkation of research into coupled composite propeller blades, along with smoother power take up, reduced blade vibration, reduced noise, and better fatigue performance [8].

Wind turbines carry loads primarily by twisting and bending, much like tidal turbines. Coupling between bending and twisting can be used to reduce loads and improve fatigue performance. Incremental loads are reduced because when the blade deforms in bending the bend-twist coupling produces a decrease in the blade twist and angle of attack. The level of load reduction depends on the twist distributed along the blade length, which is controlled by the level of bend-twist coupling. The level of bend-twist coupling depends on the blade cross sectional geometry, the level of anisotropy in the structural material, and the material distribution [10].

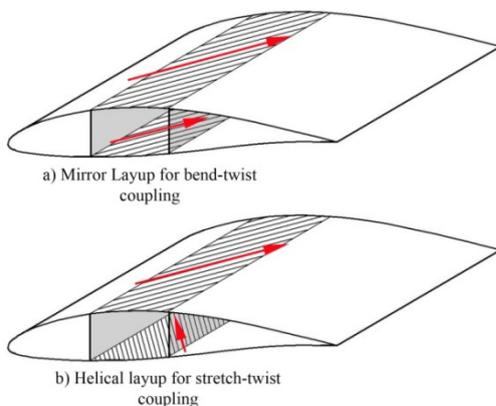


Figure 2.1: Composite lay-up for a) a bend-twist and b) a stretch-twist coupled blade.

Figure 2.1 illustrates the lay-up required for both bend-twist coupling and extension-twist coupling in a turbine blade. In the case of coupled spars, it is considered beneficial to use braided planforms rather than a more conventional lamination method. A key advantage in using braided planforms is the relative ease of manufacturing compared to conventional laminated composites. Moreover, the structural box is continuous, and fatigue/damage tolerance performance is better than conventional laminated composites [10]. Figure 2.2 illustrates three common lay-up methods for box beam manufacture. It is clear that the single box lay-up, Figure 2.2a), will not result in any elastic coupling in the beam; whereas both the double box and the patch lay-up would produce a bend-twist coupled beam.

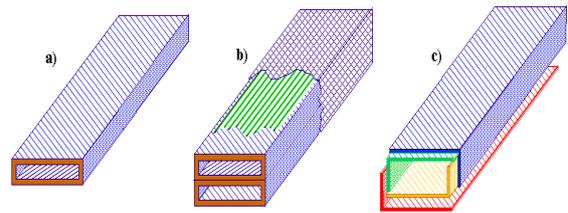


Figure 2.2: a) Single box lay-up, b) double box lay-up and c) patch lay-up

The primary advantage of the use of flexible composites in the production of turbine blades is the ability to tailor the structure to deflect in response to load changes. This allows the shape of the blade to adjust automatically to a changing inflow. This can be exploited to reduce the changes in thrust and cavitation that occur when the inflow changes locally/globally. In the case of a circumferentially varying inflow, changes in velocity cause changes in thrust which are transmitted to the shaft. The change in loading may also result in blade surface and tip vortex cavitation. A flexible composite turbine blade can be designed such that the change in loading causes a change in blade shape that reduces the magnitude of the change in thrust and delays the inception of cavitation.

In turbines with adaptive blades, the induced twist due to elastic coupling is a variable parameter which depends on the turbine run condition as well as material and structural characteristics of the blade. An adaptive blade has a dynamic topology that varies with the flow speed and rotor angular velocity.

Using a relatively simple bend-twist coupled blade enables the turbine to deal with the severe complex loads induced upon it whilst also maintaining a more consistent power output of the turbine, thus reducing the cost of the generator but maximising the energy capture of the device. It also negates the need for a variable pitch blade therefore improving maintenance feasibility and costs. Furthermore it decreases the chance of overpowering the generator or gearbox, or rather enables a more predictable peak level of power to be reached thus reducing the need for an overrated, and much more costly, generator. . By delaying the onset of stall and cavitation inception, and reducing the loads experienced

by the structure, the turbine becomes more efficient over a greater range of tip speed ratios. The turbine can capture more energy from an increased range of flow velocities and ultimately the annual energy capture of the device is increased.

3. **BLADE ELEMENT MOMENTUM CODE**

A tidal turbine generates electricity from the kinetic energy of the flow. Analysis of a tidal turbine assumes that the power generated is proportional to the cube of the flow velocity or

$$P \propto V^3 \quad (3.1)$$

From this a relationship for the power coefficient can be derived as

$$C_{pow} = \frac{P}{\frac{1}{2} \rho A V^3} \quad (3.2)$$

In Equation (3.2), $A = \pi D^2 / 4$ and is the swept area of the turbine, and ρ is the density of the fluid.

The basis Blade Element Momentum (BEM) code used in this analysis was Cwind [11] and subsequently modified for tidal turbine performance [12]. BEM balances the changes in fluid axial and angular momentum for flow through a discrete number, typically 10 – 20, of annuli and the work done on the corresponding blade sections at the effective local sectional angle of attack. The analysis requires the spanwise distribution of blade chord and pitch and the two-dimensional sectional performance, lift and drag, of the chosen blade sections. A modified Goldstein tip loss correction method accounts for the use of a finite number of blades.

The code output provides overall power, thrust and torque data for a specified turbine diameter at a fixed flow velocity and tip speed ratio (TSR). For the spanwise sections it is possible to examine the effective angle of attack, contributions to axial and tangential forces, centripetal load, and local cavitation index at top dead centre for a given HATT immersion.

In this study, the turbine RPM is held constant and the flow velocity allowed to vary over a specified range, the results of interest being the non-dimensional (C_p - λ) power curve for the turbine where,

$$C_{pow} = \frac{16nQ}{\rho D^2 V^3} \quad (3.3)$$

And

$$\lambda = \frac{\pi n D}{V} \quad (3.4)$$

Under normal circumstances, the blade geometry is fixed for such calculations; however, in this case a separate algorithm based in MATLAB was run alongside the BEM code in order to determine the performance of an adaptive device. The aim is to simulate blade motion that may be achieved with an adaptive blade without requiring a full composite blade design process and associated computation of the coupled fluid-structural response of the device to the hydrodynamic loads.

3.1. **Modifications to Code**

During the course of the research the BEM code has been modified in several different ways in order to achieve more optimal performance analysis of tidal energy extraction devices. One of the more important modifications is that of the fluid density on which the calculations are carried out, salt water being 1025 times denser than air. Other modifications include optimisation parameters which can be useful at the initial design stage of a tidal turbine in that a few optimal designs may be achieved in a short period of time giving the engineer a sound base point from which to further optimise the device.

For the evaluation of the performance of a turbine with bend-twist coupled blades, the main body of the code is run as a program based in Fortran 77; with a separate piece of code written in MATLAB R2008a which incorporates the algorithm to determine the induced twist, and hence the overall final twist, of the blade. The MATLAB code calls the main BEM program at various points in order to gain data required to complete calculations. This is detailed further in Section 5.

There is then a separate algorithm based in MATLAB which calculates the annual energy capture of each tidal device under examination. This is detailed further in Section 5.

4. **THE INDUCED TWIST MODEL**

A blade constructed of unbalanced composite laminates under axial or bending moments experiences a torsional deformation known as induced twist. The magnitude and direction of the aerodynamic force applied on the blade, and consequently the blade and turbine performance, strongly depends on the angle of attack of the blade. Angle of attack depends upon the inflow angle and twist angle of the blade. Blade twist angle may be expressed as a combination of three elements: pretwist, pitch angle and induced twist due to elastic coupling [12].

The induced twist due to elastic coupling of a tidal turbine blade changes the effective angle of attack of the flow, Equation (4.1). This therefore affects the hydrodynamic characteristics of the turbine blade and hence the performance of the device as a whole.

$$\alpha = \varphi + \beta - \beta_0 - \gamma \quad (4.1)$$

Where α is the angle of attack, ϕ is the inflow angle, β is the induced twist, β_o is the blade pre-twist and γ is the pitch angle of the blade – all in degrees.

The algorithm used in this study is one that has been developed for the performance prediction of wind turbines [13]. The design algorithm is based on analysis rather than synthesis and requires the use of a performance prediction program for the tidal turbine. With any device that uses adaptive blades – specifically bend-twist coupled blades – the induced twist is the parameter that affects the device performance. The induced twist depends both on the hydrodynamic loading experienced by the turbine blade and the structural and material properties of the blade. Therefore it is apparent that some form of structural analysis must be undertaken in order to predict the induced twist with any degree of accuracy.

The model estimates the induced twist as a function of the blade topology and the turbine run condition. Induced twist, $\beta(r)$, can be expressed as:

$$\beta(r) = \beta^*(r)\beta_T \quad (4.2)$$

Where $\beta^*(r)$ is the normalised induced twist and β_T is the maximum value of induced twist at the blade tip.

It is assumed that the material of the blade is uniform, such that the mechanical properties of the blade, fibre orientation and shell thickness do not vary in the spanwise direction. Maheri et al [13] have shown that the effect of fibre orientation on the normalised induced twist is minimal and therefore it is not considered in the algorithm.

Considering the blade as a box beam with bend-twist coupling, then β^* can be derived as:

$$\beta^* = \frac{\int_0^{y^*} (1 - y^*) dy^*}{\int_0^1 (1 - y^*) dy^*} = (2 - y^*)y^* \quad (4.3)$$

Where y^* is the normalised radius of the turbine and is defined by Equation (4.4).

$$y^* = \frac{r - R_{hub}}{R - R_{hub}} \quad (4.4)$$

Where R is the radius of the turbine, r is the instantaneous radius under analysis and R_{hub} is the hub radius, all in metres.

It should be noted that Equation (4.3) estimates β^* without any information regarding blade material except the fact that the material properties are constant spanwise along the blade. In a bend-twist coupled beam, the bending moment is mostly due to lateral loading and partly due to the torsional loading of the beam acting through the coupling effect. Each blade only experiences

a small pitching moment when compared to the lateral loading and therefore the induced bending moment due to the elastic coupling can be neglected, thus enabling the flapwise bending moment to be considered independent of the material properties.

As such β_T can be split into two parts; one part relating to the hydrodynamics of the blade, M_{hub} , and one part relating to the material properties (shell thickness, fibre orientation, and mechanical properties) of the blade, f .

$$\beta_T = fM_{hub} \quad (4.5)$$

M_{hub} is calculated directly for various tidal stream velocities and turbine radii using the BEM program. B_T needs to be estimated at a reference point in order for further calculation to be undertaken. This reference point can be taken as the design flow velocity, V_d . The reference induced twist at the blade tip at this flow velocity is an independent variable – or rather a design variable – and as such must be chosen to suit the turbine. Using M_{hub} at V_d and the chosen induced tip twist the material factor, f , can be calculated. This material factor does not vary with the tidal current velocity and therefore can be used to calculate β_T at other flow velocities.

More detail on the methodology and validation for the induced twist model against experimental data is detailed by Maheri et al [13].

5. MATLAB ALGORITHM

The MATLAB code runs the modified version of Cwind remotely in order to determine the necessary turbine performance parameters. Initially the normalised induced twist and the material factor at the design flow velocity are calculated and then these values are used to determine the induced twist at other flow velocities. The pre-twist of the turbine blade will be modified in order to cancel the effect of the induced twist at V_d , such that the ordinary blade and the adaptive blade are identical in exterior geometry at this flow velocity.

The program obtains initial data assisting the designer in the most optimum choice for the initial induced twist angle at the tip. Once this is settled, the material factor, f , is calculated and the performance of the turbine assessed over a range of tidal flow velocities.

The outputs of the code include turbine performance data such as mechanical power, thrust loading, torque and blade twist data.

The algorithm is run as illustrated in Figure 5.1. Once the results of the BEM simulation are complete the predicted performance can be used to obtain an estimate of the annual energy capture of the device.

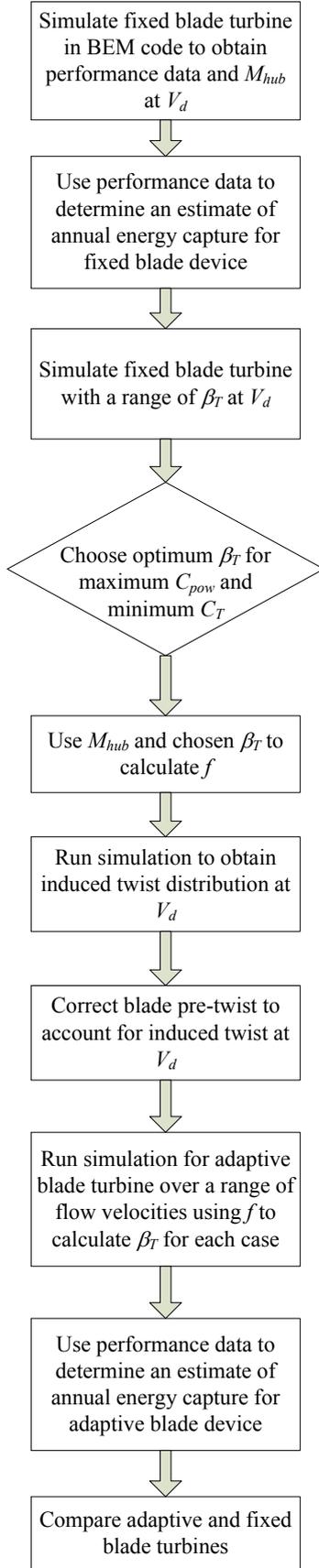


Figure 5.1: Flow chart of the bend-twist calculation procedure

6. ANNUAL ENERGY CAPTURE

To calculate the expected annual energy capture for a tidal turbine, the frequency of occurrence of each flow velocity, and the power generated at each flow velocity, need to be known.

A program has been developed in MATLAB in order to obtain an annual energy estimate for the device. The tidal cycle has been approximated by a double sinusoid; one with a period of 12.4 h representing the diurnal cycle, and the other a period of 353 h representing the fortnightly spring-neap period. The following equation, as shown by Fraenkel [14], provides a model for the prediction of the velocity, V , of a tidal current:

$$V = \left[K_0 + K_1 \cos\left(\frac{2\pi t}{T_0}\right) \right] \cos\left(\frac{2\pi t}{T_1}\right) \quad (6.1)$$

K_0 and K_1 were derived from the variation of the flow velocity over a tidal cycle.

In Equation 4, $T_0 = 353$ hours (1 lunar month) and $T_1 = 12.4$ hours. The variation due to the daily tidal cycle is taken to be $\cos\left(\frac{2\pi t}{T_1}\right)$ and due to the lunar month tidal cycle taken to be $\cos\left(\frac{2\pi t}{T_0}\right)$.

The maximum flow velocity for the spring tide was 2.5m/s and the ratio of maximum spring to maximum neap tide was 2.0. Since the maximum spring tide occurred at $t = 0$, an equation in terms of K_0 and K_1 could be derived:

$$V_{spring} = 2.5 = K_0 + K_1 \quad (6.2)$$

From plotting the sum of the two tidal variations it could be seen that the maximum neap tide occurred at approximately $\cos\left(\frac{2\pi t}{T_1}\right) = 1$ and $\cos\left(\frac{2\pi t}{T_0}\right) = -1$.

A second relationship was derived as:

$$V_{neap} = 1.25 = K_0 - K_1 \quad (6.3)$$

Or

$$\frac{V_{spring}}{V_{neap}} = 2.0 = \frac{K_0 + K_1}{K_0 - K_1} \quad (6.4)$$

Figure 6.1 illustrates the monthly tidal velocity variation. It is clear that the peak velocities happen infrequently and thus a device that is able to draw more power from the flow at lower velocities is more desirable.

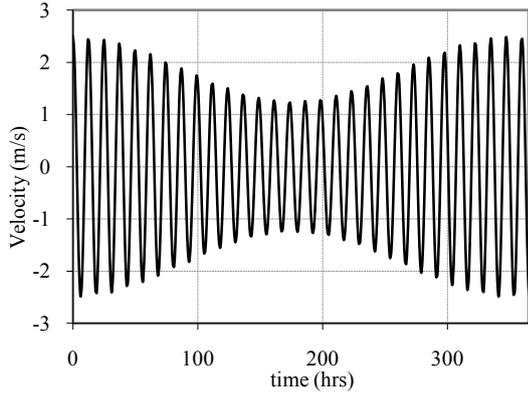


Figure 6.1: Monthly tidal velocity variation

The two simultaneous equations could then be solved to give $K_o = 1.875$ and $K_i = 0.625$, and a final relationship for flow velocity variation with time of:

$$V = \left[1.875 + 0.625 \cos\left(\frac{2\pi t}{353}\right) \right] \cos\left(\frac{2\pi t}{12.4}\right) \quad (6.5)$$

For a maximum peak spring flow velocity of 2.5m/s, and assuming that the maximum mean spring current velocity is twice that of the maximum mean neap velocity, the two constants can be found to be $K_o = 1.875$ and $K_i = 0.625$ m/s. The frequency of occurrence of each flow velocity annually was obtained using a developed MATLAB code and this, coupled with the power data obtained using the BEM code and associated bend-twist simulation algorithm was used to compute a value for the annual energy capture of both the fixed blade and the adaptive blade turbine.

7. RESULTS

The principle particulars for the free stream, horizontal axis turbine under analysis are shown in Table 7.1.

Table 7.1: Turbine principle particulars

Rotor radius	20 m
Hub radius	2 m
Number of blades	3
Pitch angle	4°
Rated power	1.3 MW
RPM	14 rpm
Design flow velocity	2.5 m/s

The power curve for the fixed blade, base rotor is illustrated in Figure 7.1.

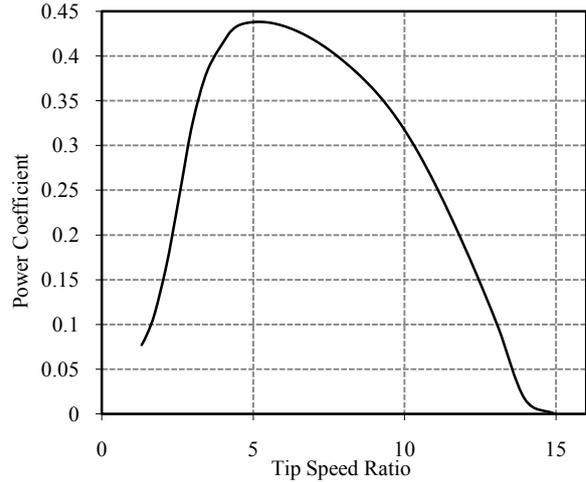


Figure 7.1: The reference power curve for the fixed blade turbine

Thus M_{hub} at the design flow velocity was calculated and β^* could be found. The next stage was to choose an initial value for the tip twist at the design flow velocity. The BEM code and associated algorithm was run for a series of β_T and a power and thrust curve, this time plotted against initial induced tip twist, was developed and is shown as Figure 7.1.

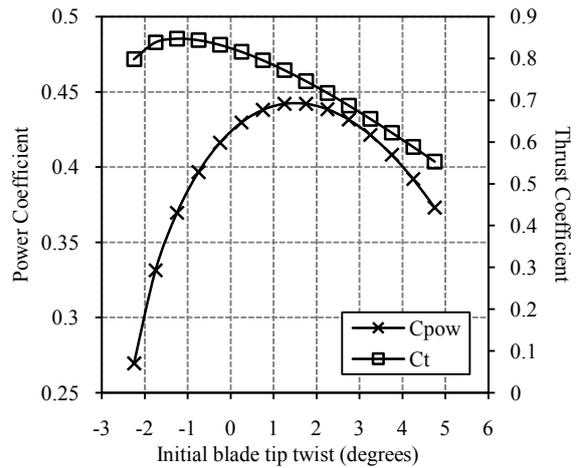


Figure 7.2: Power and thrust coefficient plotted as a function of initial blade tip twist

In order to achieve the optimum performance from the turbine, a high value of C_{pow} and a low value of C_T are required. C_T tends to decrease as β_T increases, whilst C_{pow} is maximum for $\beta_T = 1.5^\circ$. The value of $\beta_T = 1.5^\circ$ was considered to be optimum for this turbine as C_T was approximately half of the maximum value and C_{pow} was maximum with a value of 0.446. For this value of β_T the material factor, f , was calculated to be 0.000462.

Using this value of f , the twist distribution of the turbine was adjusted such that the device maintained the optimal shape at the design flow velocity. The performance of the turbine with bend-twist coupled blades, but no other alterations, was then calculated.

It is interesting to observe the relationship between overall blade tip twist and TSR. This is illustrated in Figure 7.3. With increasing flow velocity, decreasing TSR, the twist at the tip of the blade increases more rapidly with the blade twisting towards stall. Overall, the blade tip twist seems to decrease exponentially with an increase in TSR.

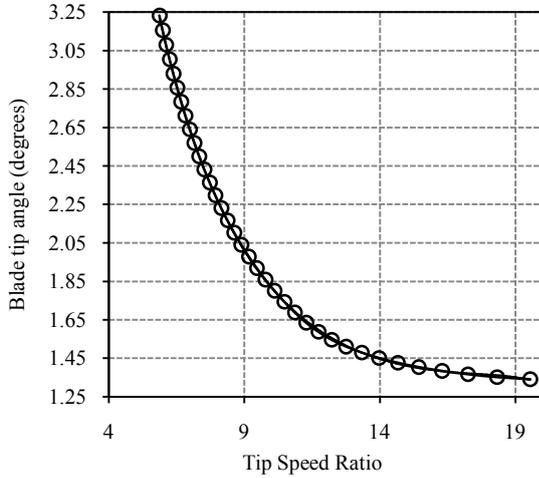


Figure 7.3: The relationship between overall blade tip twist and TSR

Figure 7.4 compares the power curves for the fixed and adaptive blade turbines over the operating range of tip speed ratios (TSR), and hence flow velocities.

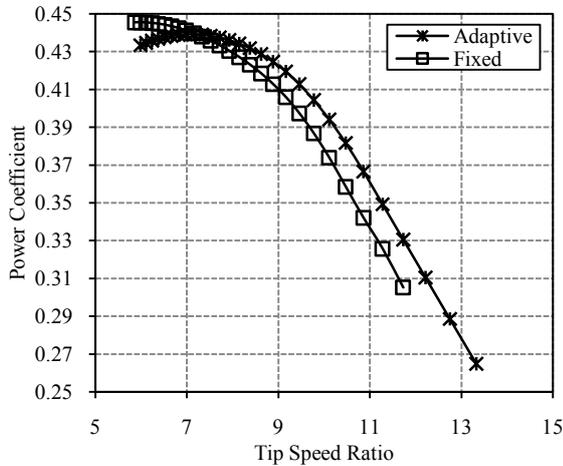


Figure 7.4: Comparison of the fixed blade turbine and adaptive blade turbine power curves

It can be observed that whilst the fixed blade has a higher maximum value of C_{pow} the adaptive blade exhibits increased values of C_{pow} over the range of TSR from 7.5 – 13.3, and also starts operating at higher TSR which equates to a lower flow velocity. Since the peak flow velocity in a tidal stream occurs infrequently, it is likely that the turbine with better performance over the lower flow velocities (higher TSR) will have an improved annual energy capture.

The thrust force on the rotor is directly applied to the blades and hub, but also the support structure of the device, and thus considerably influences the design.

Figure 7.5 illustrates the change in thrust coefficient, C_T , as a function of tip speed ratio for the fixed blade and adaptive blade turbines.

$$C_T = \frac{8T}{\rho\pi V^2 D^2} \quad (5)$$

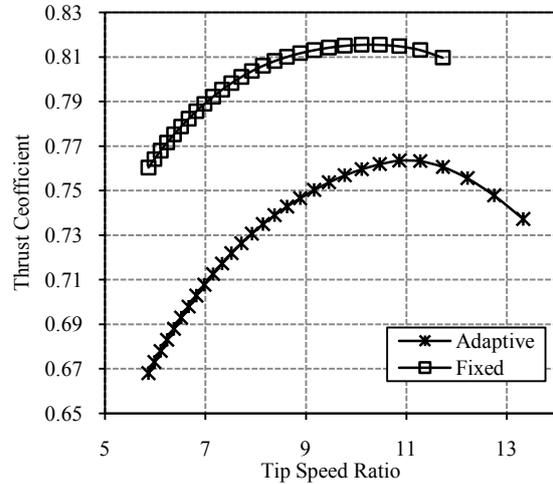


Figure 7.5: Comparison of the fixed blade turbine and adaptive blade turbine thrust curves

It can be seen that the use of adaptive blades significantly reduces the thrust coefficient over the full range of TSR, thus reducing the harmful loads on the turbine structure. This has positive connotations for the design and maintenance of the device, in that the amount of material used and the level of maintenance required may both be reduced, thereby reducing the overall cost of the HATT.

The performance data was then used in the annual energy capture program and the energy capture of both the fixed blade and adaptive blade turbine was calculated over the period of one year. The adaptive blades improved the energy capture of the device by 12.5% over that of the fixed bladed device. It is thought that this is due to the more optimal performance over the higher TSR (lower wind velocities) which occur more frequently over both the spring and neap tidal cycles. This increase in energy capture, coupled with the reduction in thrust force on the device, makes the concept of adaptive blades an attractive proposition for future turbine designs.

8. CONCLUSIONS

The use of composite materials, and the design of their lay-up to create bend-twist coupling in a free stream tidal turbine blade, could improve annual energy capture and reduce thrust loading on a tidal device.

An initial design program has been developed to estimate the coupling in a composite bend-twist adaptive blade and calculate the performance of a device using such blades.

The performance of such a device was compared to a fixed bladed turbine, with the only alteration in design between the two being that of the adaptive blades.

For this specific case it was observed that the fixed bladed turbine had a higher maximum value of C_{pow} , but the adaptive bladed turbine preserved a higher value of C_{pow} for a wider range of TSR. This is an equivalent effect to the use of a controllable pitch blade. The reduction in C_{pow} would allow the use of a lower rated generator.

The turbine with adaptive blades was also observed to have a significant reduction in thrust force over the operating range of TSR. It is thought that the use of adaptive blades on a HATT could provide significant gains in the costs associated with construction and maintenance of the device due to the decreased thrust loading.

Overall the adaptive bladed turbine showed a 12.5% increase in annual energy capture when compared to the fixed blade device.

It is thought that the potential improvements in using adaptive blades for a free stream, horizontal axis, tidal turbine are such that further work as to the development and testing of such blades should be carried out.

9. FUTURE WORK

In order to take the concept of an adaptive free stream tidal turbine blade further, much work will be required involving both numerical computation and practical experimentation.

Initially a simple Finite Element Analysis (FEA) of a bend-twist coupled box beam will be set up. Several different carbon fibre bend-twist box beams will then be manufactured and tested with the intention of developing a relationship between the lay-up angle and number of plies to the level of coupling apparent in a beam. This will then be used as a design tool alongside the BEM code to optimise the blade topography for a tidal turbine.

Following on from that, it is intended that a model in which the fluid flow over the blades of the turbine and the structural response to the resulting pressures can be computed simultaneously and interact with one another – a Fluid Structure Interaction package.

Ultimately it is intended that optimised model bend-twist coupled blades will be manufactured for testing in both a wind tunnel, and a cavitation tunnel.

10. ACKNOWLEDGEMENTS

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11. NOMENCLATURE

Variable	Units	Definition
A	m ²	Swept area of turbine
C_{pow}	-	Power coefficient
C_T	-	Thrust coefficient
D	m	Turbine diameter
K_0	-	First tidal cycle constant
K_1	-	Second tidal cycle constant
M_{hub}	Nm	Bending moment at hub
P	W	Power
Q	Nm	Torque
R	m	Turbine radius
R_{hub}	m	Hub radius
T	N	Thrust
T_0	Hours	Variation due to daily tidal cycle
T_1	Hours	Variation due to monthly tidal cycle
V	m/s	Flow velocity
V_d	m/s	Design flow velocity
V_{neap}	m/s	Maximum neap tidal current
V_{spring}	m/s	Maximum spring tidal current
f	-	Material factor
n	rps	Revs per second
r	m	Instantaneous radius
t	Hours	time
y^*	-	Normalised radius
α	(°)	Angle of attack
β	(°)	Induced twist angle
$\beta^*(r)$	(°)	Normalised induced twist
β_0	(°)	Blade pre-twist angle
β_T	(°)	Induced twist at the blade tip
γ	(°)	Pitch angle
λ	-	Tip Speed Ratio (TSR)
ρ	kg/m ³	Fluid density
φ	(°)	Inflow angle

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