The prediction of the hydrodynamic performance of marine current turbines

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Abstract

The development of a blade element momentum (BEM) model for the hydrodynamic design of marine current turbines is presented. The model includes routines for interpolation of 2D section data and extrapolation for stall delay. The numerical model is compared with experimental data obtained from tests of an 800 mm diameter model rotor carried out in a cavitation tunnel. The theoretical predictions are in good agreement with the experiments. Using this validated model, a typical 3D rotor is used to demonstrate parametric variations of the design parameters. The effect of tip immersion on possible cavitation is assessed for this rotor. The model is then used to solve the dynamic effects of a tidal profile. The effect of an increase in blade roughness is presented, indicating a relatively small reduction in power. This work demonstrates that the numerical model developed can provide a useful tool for the investigation of the hydrodynamic design and operation of marine current turbines.

Keywords: Tidal stream; Marine currents; Marine current turbines; Tidal energy; Ocean energy; Turbine design

1. Introduction

Electrical energy extraction from marine currents offers the promise of regular and predictable energy [1]. The location and viability of such devices to extract energy from marine currents has been a focus of several investigations [2–4] and a detailed review article [5]. These highlight several advantages and possible commercial viability for several locations throughout the world, particularly where the mean peak tidal currents are over 2 m/s (~4 knots). One example is the race of Alderney in the Channel Islands, which has mean peak tidal currents around 2.5 m/s and could theoretically supply 7.4 TWh per year of electrical energy [6,7]. The majority of the designs for energy conversion are based upon the design of a horizontal axis wind turbine, which is the focus of this paper [8].

The success of using marine current turbines to tap the ocean currents is dependent on predicting their hydrodynamic performance. Methodologies need to be established that will describe the physical and operational performance of the turbines, allowing their design to be investigated and performance evaluated. Much can be transferred from the design and operation of wind turbines [9], and ship propellers [10]. There are however a number of fundamental differences in the design and operation of the marine current turbine, which will require further investigation, research, and development. Particular differences entail changes in Reynolds number, different stall characteristics, and the possible occurrence of cavitation [11].

This paper describes the development of a numerical model, based on the blade element momentum (BEM) theory [9] for predicting the performance of marine current turbines and the spanwise distributions of blade loadings. The BEM theory is well established for modelling rotor dynamics including marine propellers and wind turbines. While BEM theory can successfully predict the spanwise loading on narrow blades, such as those used for wind or marine turbines, it does not provide information on chordwise loading. Other methods, such as the 2D panel methods, can be used to predict the chordwise pressure...
distributions and loadings, which are of particular relevance to the study of stall characteristics and the occurrence of cavitation [12]. The development of the theory and numerical model are described, together with comparisons with experimental data and examples to illustrate applications of the methodology.

The work described here forms a part of a wider research programme being conducted by the Sustainable Energy Research Group at the University of Southampton. Relevant to this work is an EPSRC funded programme, which included detailed experimental studies of an 800 mm diameter turbine reported in Refs. [13,14]. These experiments used for validation were carried out in the cavitation tunnel at QinetiQ, Haslar. The viscous solution that interacts with the incompressible potential flow via a surface transpiration model. Recent cavitation tunnel tests of possible sections toward the tip of a marine current turbine have been compared with XFoil [12]. The results showed good agreement with pressure distributions and lift curves, but underestimate drag coefficients at incidences approaching stall angle. To extrapolate up to the stall delay angle the method proposed by Snel et al. was used [21]. Further extrapolations beyond the stall delay angle were achieved using Viterna and Corrigan’s methodology for post stall predictions [22].

The overall numerical model demonstrating how the numerical program can be used to provide both spanwise and chordwise loadings and cavitation inception checks is shown in the flow chart in Fig. 1.

### 3. Comparison of theory with experiments

The developed theory has been validated using the cavitation tunnel and test tank experimental results for an 800 mm diameter model marine current turbine reported in Refs. [13,14]. The numerical codes for this research are based on BEM theory and were adapted from a program for wind turbine design developed by Barnsley and Wellicome [15]. Details of the basic equations used are given in Appendix A. Derivations of these equations and program flow charts are presented in Ref. [16].

The 2D panel code XFoil [17] was used to derive the lift and drag coefficient data for the blade elements. XFoil is a linear vorticity stream function panel method with a viscous solution that interacts with the incompressible potential flow via a surface transpiration model. Recent cavitation tunnel tests of possible sections toward the tip of a marine current turbine have been compared with XFoil [12]. The results showed good agreement with pressure distributions and lift curves, but underestimate drag coefficients at incidences approaching stall angle. To extrapolate up to the stall delay angle the method proposed by Snel et al. was used [21]. Further extrapolations beyond the stall delay angle were achieved using Viterna and Corrigan’s methodology for post stall predictions [22].

The developed theory has been validated using the cavitation tunnel and test tank experimental results for an 800 mm diameter model marine current turbine reported in Refs. [13,14]. These experiments used for validation were carried out in the cavitation tunnel at QinetiQ, Haslar. The same chord, pitch, and thickness distributions of the rotor were used in both the experiments and theory are presented in Table 1.
For the experiments, the rotor is attached to a main shaft, which drives a DC generator from a pulley through a belt carried up through the vertical support tube. An in-line strain gauge dynamometer mounted next to the turbine was used to measure the thrust and torque. This dynamometer was designed to run wet, so measurements could be made before any bearing or seal losses. The strain gauge bridge circuit is connected via a slip-ring assembly to conditioners, and output signals were acquired on a computer using a data acquisition system. The electrical power is absorbed with rheostats, which also allowed regulation of the rotor speed.

Fig. 2 shows the basic section lift and drag data for variation in thickness ratio derived using Xfoil [17] and variations in thickness ratio derived using XFoil [17] and with the extrapolated stall angles. Fig. 4 shows examples of the predicted spanwise outputs for $x$, $C_T$, $C_L$, and $C_D$ for fixed $20^\circ$ hub pitch angle and a range of TSR values.

On comparing Figs. 3a and 4a, the blade is stalling for TSR = 4 around $r/R = 0.4$. Fig. 4b shows a peak in the power gradient close to $r/R = 0.84$ and a reduction for $r/R > 0.84$ due to the tip loss factor. Fig. 4c shows constant axial flow factors around 0.3 for the majority of the blade, close to the optimum of 1/3 derived from the Betz limit [9]. Fig. 4d shows low tangential flow factors indicating the appropriateness of using the BEM theory. Total $C_P$ and $C_T$ values derived from integration of the spanwise distributions are shown in Figs. 5 and 6. Also included in Figs. 5 and 6 are the experimental results. It is seen that the theoretical $C_P$ results for hub pitch angles of $20^\circ$, $25^\circ$, and $27^\circ$ are close to the experiments up to a TSR of about 7 after which they are a little higher than the experiments. The theoretical values for the $15^\circ$ hub pitch angle are correct up to a TSR of about 5 after which they continue to increase, rather than decrease. The reasons for this are not clear. The theoretical $C_T$ values for hub pitch angles of $20^\circ$, $25^\circ$, and $27^\circ$ are a little lower than the experimental results. Again, the predictions for $15^\circ$ hub pitch angle are erroneous above a TSR of about 5.5.

Based on the reasonable correlation between theory and experiment, the theoretical model is considered to provide a satisfactory representation of the performance characteristics. This is particularly the case when using the theory for preliminary design exercises, investigations, and parametric studies. It should be noted that refinements to the theory have not yet been carried out to bring the theoretical predictions more in-line with the experimental data.

4. Example applications and implications

4.1. General

A number of cases have been investigated to illustrate typical applications of the theory. These entail an investigation of the rotor design for a $20 \text{ m}$ diameter as suggested for some of the devices for the Alderney Race [5,6], calculations of cavitation inception, the influence of the tidal velocity profile, and the effects of an increase in blade drag due to an increase in roughness and fouling.

4.2. Example predictions for a $20 \text{ m}$ diameter rotor

For sample calculations of a three-bladed $20 \text{ m}$ rotor, the turbine blades were assumed to be scaled versions of the model used for the validation exercise. This example turbine is rated for a tidal speed of $2 \text{ m/s}$. At higher tidal speeds, the turbine is assumed to be pitched to maintain a constant rated power and constant rpm. At the design speed of $2 \text{ m/s}$, a TSR of 6 was assumed, with a hub pitch of $20^\circ$, operating at 11.5 rpm. This TSR is close to the peak
$C_p$ of 0.45 shown in Fig. 5 and results in a rated shaft power of 580 kW for this device. In reality, after subtracting drive chain and generating losses the output may be closer to 500 kW. The thrust loading for the design case is 490 kN ($C_T = 0.76$), as shown in Fig. 6.

Predictions for power and thrust for marine current speeds up to 3 m/s are presented in Fig. 7 for four pitch angles from 20° to 34.6° at the design RPM of 11.5 rpm. Point A in Fig. 7 and the data given in Table 2 show the design case (2 m/s). Point B gives the device performance at the increased flow speed of 3 m/s. In order to maintain a constant power operation the device would have a pitch angle of 34.6°. For Point B, the predicted performance is $TSR = 4$, $C_p = 0.14$, and $C_T = 0.17$. The thick solid line in Fig. 7 represents design operation using pitch regulation. Consequently, at the same power, the thrust loading has
Fig. 4. Spanwise characteristics of inflow angle, power coefficient and axial and tangential inflow factors for 20° hub pitch angle and a range of TSR.
(a) Alpha distribution, (b) power gradient distribution, (c) axial flow factor and (d) tangential flow factor.

Fig. 5. Theoretical power coefficient simulations for various hub pitch angles compared with experimental data points [13,14].

Fig. 6. Theoretical thrust coefficient simulations for various hub pitch angles compared with experimental data points [13,14].
almost halved (Fig. 7b). A more detailed methodology for matching the design loads of a turbine to resource has been developed and discussed in [18].

4.3. Predicting cavitation inception

Cavitation will occur on the turbine blade when the local pressure falls to, or below, the vapour pressure of the sea water. It can be predicted by comparing the local pressure distribution with the cavitation number [10]. A cavitation number, \( \sigma \), is defined as

\[
\sigma = \frac{P_0 - P_V}{0.5 \rho W^2} = \frac{P_{AT} + \rho gh - P_V}{0.5 \rho W^2}.
\]

The local head of water (\( h \)) on the blade is calculated at the blade at the uppermost position (\( h = h_t + R - r \)), where \( h_t \) is the tip immersion. The local chordwise cavitation number and lift coefficient are presented in Fig. 8 for Points A and B for different tip immersion depths. The results show as expected a decrease in \( \sigma \) as the tip is approached due the faster speeds and smallest immersion.

A summary at \( r/R = 0.95 \) is shown in Table 3 for Points A and B, and plots of the chordwise pressure distribution are presented in Fig. 9. These pressure distributions were calculated using XFOIL assuming \((t/c = 13\%)\). The results indicate that for Point A, even with 2 m tip immersion, cavitation is unlikely as the minimum \( C_{\text{press}} \) is still only half of \( \sigma \). However, for Point B, with the blades heavily pitched, cavitation is possible for the small immersion cases. It should be noted that the region of minimum pressure is now on the face of the foil close to the leading edge. This type of cavitation is known as face sheet cavitation.

The choice of blade section can be important when considering cavitation. In this study, NACA 63-8xx sections have been used. Cavitation tests for this and alternative 2D sections with 15% thickness chord ratio are reported in [12]. The shape of the cavitation inception envelope defining the region of cavitation free operation is dependent upon the section profile. For example, Fig. 10 shows the operation region of cavitation free for both NACA 63-815 and NACA 63-215, which has less camber. The envelope of cavitation free operation is centred on a lower \( C_L \) for the NACA 63-215 section than for the NACA 63-815 section. It should also be noted that the XFOIL theory had a tendency to underpredict the region of cavitation free operation. This is further discussed in [12] and data are presented for a further two-section profiles. Consequently for this design, if NACA 63-2xx sections were used instead, then cavitation would be more likely at Point A, but not at Point B.

In summary, the cavitation is possible both for the design case or when the blade is heavily pitched for power regulation. Consequently, cavitation could occur on both sides of the blade and therefore all areas of operation would need to be considered when designing a rotor. The choice of blade section could result in reducing the possibility of cavitation at one end of operation.

Predictions of the various forces on the blade are required in order to carry out a structural analysis, including checks on the blade stiffness and hub bending moments. Fig. 11 shows the spanwise normal and axial force distributions for Points A and B. For further detailed
structural calculations, the chordwise loadings are also required, for example at $r/R = 0.95$, as shown in Fig. 9.

4.4. Effect of tidal velocity profile

For coastal waters, the tidal boundary layer is generally assumed to extend to the full tidal depth, $d$. An approximate tidal velocity profile can be estimated assuming a power law of the form:

$$U_0(z) = \bar{U}_0 \left( \frac{z}{d} \right)^{\frac{1}{7}},$$

where $U_0$ is the mean tidal velocity at the surface, and $d$ is the tidal depth.

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Table 3
Comparisons of blade conditions at different tip immersions at $r/R = 0.95$

<table>
<thead>
<tr>
<th>$h_t$ (m)</th>
<th>Case</th>
<th>$\sigma$</th>
<th>$\alpha$</th>
<th>$C_L$</th>
<th>$\text{Max } (-C_{\text{press}})$</th>
</tr>
</thead>
<tbody>
<tr>
<td>2</td>
<td>A</td>
<td>1.81</td>
<td>0.84</td>
<td>0.72</td>
<td>0.9</td>
</tr>
<tr>
<td>5</td>
<td>A</td>
<td>2.25</td>
<td>0.84</td>
<td>0.72</td>
<td>0.9</td>
</tr>
<tr>
<td>10</td>
<td>A</td>
<td>2.99</td>
<td>0.84</td>
<td>0.72</td>
<td>0.9</td>
</tr>
<tr>
<td>2</td>
<td>B</td>
<td>1.73</td>
<td>-5.18</td>
<td>0.04</td>
<td>2.4</td>
</tr>
<tr>
<td>5</td>
<td>B</td>
<td>2.16</td>
<td>-5.18</td>
<td>0.04</td>
<td>2.4</td>
</tr>
<tr>
<td>10</td>
<td>B</td>
<td>2.86</td>
<td>-5.18</td>
<td>0.04</td>
<td>2.4</td>
</tr>
</tbody>
</table>

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Fig. 8. Local spanwise cavitation number and lift coefficient for Point A (2 m/s, 11.5 rpm, 20° hub pitch) and Point B (3 m/s, 11.5 rpm, 34.6° hub pitch) calculated with the turbine blade at the uppermost position for tip immersion depths of 2 and 5 m. (a) Cavitation number and (b) lift distribution.

Fig. 9. Chordwise distributions of pressure at $r/R = 0.95$ for Point A (2 m/s, 11.5 rpm, 20° hub pitch) and Point B (3 m/s, 11.5 rpm, 34.6° hub pitch).

Fig. 10. Cavitation inception envelope from the experiments and predictions presented in Fig. 9 of Ref. [12].
where $\bar{U}_0$ is the mean velocity, and $z$ is the vertical position \[19\]. Preliminary measurements of tidal velocities over the Arklow Bank suggest that Eq. (2) is suitable for these initial calculations \[20\]. For more accurate site-specific predictions, the use of logarithmic equations that take into account seabed roughness scales may be more appropriate \[19\]. For detailed design of a farm of current turbines, measurements would need to be taken for a particular site over a period of a month in order to derive accurate velocity profile data. Then powers and oscillations in forces on current turbines could be predicted with more confidence.

For these preliminary studies, Eq. (2) has been used to demonstrate the scale of the possible dynamic forces. Fig. 12 demonstrates a typical velocity profile across a marine current turbine, based on a 10 m radius turbine in 30 m deep sea and a tidal speed of 2 m/s. Based on the power law and this design, there will be a 20% decrease in the tidal velocity across the blade. This clearly indicates that for optimum performance the pitch of the blade should be continually altering, as close to the sea surface cavitation will be the design criterion while toward the sea bed stall will be the design criterion.

The local solved spanwise distributions with a turbine blade at 0°, 60°, 120°, 180° are shown in Fig. 13. These show that at a design TSR = 6 with a 2 m/s velocity flow, the local TSR at the tip varies from around 6.2 to 7.7 (Fig. 13b). This results in a change in loadings as it rotates (Fig. 13(c–d)). Assuming a three-bladed turbine with blades A, B, and C as indicated in Fig. 12, the predicted power and thrust for each blade over one rotation are shown in Fig. 14(a and b). Fig. 14(c and d) demonstrates the summed contribution for the three blades. The resulting rotor force oscillations are up to about ±1% while individual blades are seen to suffer force oscillations of up to about ±3%. This should be taken into account in studies of fatigue loadings on the blades (Figs. 13 and 14).

4.5. Blade fouling

As a first approximation, the influence of an increase in roughness and fouling has been simulated by assuming an increasing in section drag coefficient by up to 50%. The effect of surface roughness does not tend to have much effect on the lift slope but can alter the angle at which stall can occur. This has been demonstrated in experiments for thicker wind turbine sections discussed in \[23,24\]. The results from this approximation are shown in Fig. 15. There is little change in spanwise $C_L$ distribution (Fig. 15a) and significant increase in spanwise distribution of $C_D$ (Fig. 15b) for TSR of 6. Fig. 15c shows the corresponding spanwise $C_P$ distribution and Fig. 15d shows $C_P$ to a base of TSR for different values of $C_D$. For example, an increase in $C_D$ of 50% leads to a very small change in $C_P$ up to a TSR of about 4 and a decrease in $C_P$ of about 6–8% at higher TSR. This approximates to a loss in power of around 40 kW for the design case of 580 kW (Point A: TSR = 6, speed = 2 m/s) (Fig. 15).
Fig. 13. Spanwise characteristics for $z$, $dC_p/d(r/R)$, $C_L$, $C_D$ for four blade angles at a design TSR = 6 (2 m/s). (a) Power components, (b) thrust components, (c) power sum and (d) thrust sum.

Fig. 14. Oscillations in predicted power and thrust over the blade passage (20 m diameter turbine at TSR = 6). (a) Power components, (b) thrust components, (c) power sum and (d) thrust sum.
5. Conclusions

(1) A numerical model based on the BEM has been developed and presented. This model has been validated successfully using cavitation tunnel tests on an 800 mm diameter rotor. The model includes routines for interpolation of 2D section data and extrapolation for stall delay to allow predictions off design.

(2) The work has been extended to the consideration of a large-scale device operating in the sea. An example of a case study on a 20 m diameter rotor turbine demonstrates the ability of the model to predict the expected performance characteristics, in the form of power and thrust curves.

(3) The prediction of cavitation was carried out for certain cases with relatively shallow tip immersion. It was found that cavitation could be avoided with the use of suitable designs and choice of 2D sections. Care should therefore be taken to predict cavitation inception for all situations if marine current turbines are operated, as back sheet cavitation may occur close to the design case, but with pitch regulation designs, face sheet cavitation may occur.

(4) The effect of tidal velocity profile on blade loadings has been illustrated. It was found that the fluctuations in blade loadings were not insignificant and will need to be taken into account in structural fatigue studies. A similar methodology could be applied to other sources of non-uniform flow, such as that due to waves or oblique flow.

(5) The effect of blade fouling was investigated and it was found that a significant decrease in power can arise at higher TSR. Care will therefore be required in this aspect of turbine maintenance, depending on factors such as areas of operation and sea temperature.

(6) The use of the validated model as a design tool for marine current turbines has been demonstrated by a series of example applications. Overall, these example applications have provided an analysis of some of the operational factors that need to be considered in the design process. Large-scale validations of the model will be possible when real data on a device in operation are made available to the researchers. The work is continuing and will also investigate aspects of device operations in arrays.

Appendix A. The blade element momentum model

The performance of a part of the rotor between radius \( r \) and radius \( (r+dr) \) is analysed by matching the blade forces generated by the blade elements (as 2D lifting foils) to the momentum changes occurring in the fluid flowing through the rotor disc between the radii. The direction of the blade forces and angles is shown in Fig. A.1. An outline of the equations used is set out below and derivations are presented in Ref. [16].
A.1. Momentum considerations

Equating the thrust on an element of the blade to the axial momentum change, the torque on an element to the angular momentum change and introducing a Goldstein factor $k$ to take into account of a finite number of blades leads to the following equations for the thrust ($T$) and torque ($Q$) gradients:

$$\frac{dT}{dr} = 4\pi p r [U_0^2 a(1-a)k + (a'\Omega)k^2], \quad (A.1)$$

$$\frac{dQ}{dr} = 4\pi^3 \rho U_0 \Omega a'(1-a)k. \quad (A.2)$$

A.2. Blade element considerations

The local lift and drag gradients are defined by

$$\frac{dL}{dr} = \frac{1}{2} \rho c B W^2 C_L \quad (A.3)$$

and

$$\frac{dD}{dr} = \frac{1}{2} \rho c B W^2 C_D, \quad (A.4)$$

where $c$ is the local blade chord, and $B$ is the number of blades. The rotor thrust and torque gradients are then defined by

$$\frac{dT}{dr} = \frac{dL}{dr} \cos \phi + \frac{dD}{dr} \sin \phi, \quad (A.5)$$

$$\frac{dQ}{dr} = r \left[ \frac{dL}{dr} \sin \phi + \frac{dD}{dr} \cos \phi \right]. \quad (A.6)$$

Combining Eqs. (A.1), (A.2) (A.5), and (A.6) yields equations for axial ($a$) and tangential ($a'$) inflow factors.

These equations are solved by iteration of $\phi$.

$$\frac{a}{1-a} = \frac{\sigma_k}{4\pi \kappa \sin^2 \phi} \left[ C_X - \frac{\sigma_k C_y^2}{4\pi \kappa \sin^2 \phi} \right], \quad (A.7)$$

$$\frac{a'}{1+a'} = \frac{\sigma_k C_y}{4\pi \kappa \sin \phi \cos \phi}, \quad (A.8)$$

where $x = r/R$, $C_X = C_L \cos \phi + C_D \sin \phi$, $C_Y = C_L \sin \phi - C_D \cos \phi$, and solidity ratio $\sigma_k = cB/2R$.

A.3. Power and thrust

The application of Eqs. (A.1)–(A.8) yields solutions for the power and thrust gradients:

$$\frac{dC_p}{dx} = \frac{2TSR(1-a)^2 \sigma_k x C_y}{\pi \sin^2 \phi}, \quad (A.9)$$

$$\frac{dC_T}{dx} = \frac{2(1-a)^2 \sigma_k C_x}{\pi \sin^2 \phi}. \quad (A.10)$$

The integration of these gradients yields solutions for power and thrust.

A.4. Tip loss factor

The Goldstein momentum averaging factor $\kappa$ has been used to evaluate tip losses. The following equations provide a good fit to the Goldstein charts used in the marine propeller field:

$$\kappa = \frac{2}{\pi} \cos^{-1} \left( \frac{\cosh(xf)}{\cosh(f)} \right), \quad (A.11)$$

$$f = \frac{B}{2x \tan \phi} - \frac{1}{2}. \quad (A.12)$$

A.5. Stall delay

Predictions of marine current turbine stall performance are required if stall is used to control the power. Stall regulation may be a cost-effective approach, as pitch regulation under water is further complicated due to sealing problems and complex access for maintenance. The basic 2D section data were derived using Xfoil [7]. The data extrapolated to up to the stall delay angle using the method proposed by Snel et al. [21]. Post stall extrapolations were achieved using Viterna and Corrigan methodology [22]. A fuller discussion of stall delay is given in [16].

References


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