

Considerations of Improved Tidal Stream Turbine Performance Using Double Rows of Contra-Rotating Blades

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Abstract

This paper describes work involved with modelling, using the CFD package FLUENT, a contra-rotating double row set of 3 bladed tidal turbines. The design of these turbines is to assess the potential increase in the power, torque and axial thrust generated over a conventional single row propeller.

A single row 3 bladed horizontal axis tidal turbine (HATT) has been created and validated with data from a 0.5 m diameter laboratory scale turbine. This data has then been scaled up to 10 m diameter turbines. A series of models have been produced at the 10 m diameter scale which incorporates 2 rows of blades. The spacing between each row of blades has been increased to establish the wake characteristics and the turbine characteristics, in particular for power, torque and axial thrust for each design scenario. Each model is compared against the single row turbine. The row spacing has been non-dimensionalised to the turbine hub diameter to provide a more pragmatic approach to the spacing selection.

The results from the CFD models show that there is a negligible increase in the power generated but an increase in the axial load on the turbine. The nett torque acting on the device is, however, considerably reduced, and potentially negated, so potentially helping the turbine to align to the tidal flow.

Keywords: CFD, Contra-rotating blades, power, torque

Nomenclature

a	= swept area (m^2)
b_T	= blade tip chord length (m)
C_p	= power coefficient
C_t	= axial thrust coefficient
D	= hub diameter (m)
F_a	= axial thrust (N)
L	= axial spacing (m)
L_c	= characteristic length (m)
P	= power (W)
P_a	= available power (W)
r	= turbine radius = (5 m)
T	= torque (Nm)
v	= velocity (m/s)

ρ	= density (kg/m^3)
ω	= angular velocity (rad/s)
θ	= blade pitch angle ($^\circ$)

Acronyms

BEM	= beam element method
CFD	= computational fluid dynamics
HATT	= horizontal axis tidal turbine
MRF	= moving reference frame
RANS	= Reynolds Averaged Navier-Stokes
RSM	= Reynolds Stress model
TSR	= tip speed ratio
UDF	= user defined function
Vmsp	= Mean-Spring-Peak velocity

1 Introduction

Tidal stream resources around the UK have the potential to generate 15.6 TWh, which is approximately 4% of the UK electricity demand [1]. However, this figure is slightly reduced by the 'significant impact factor' (SIF) as presented by Black and Veatch [2]. The resulting figure of 12 TWh/y represents the UK tidal stream resource that could be economically exploited if the technology were to be developed and deployed.

To date UK tidal stream technology has resulted in a number of installed full scale devices. Marine Current Turbines (MCT) introduced the world's first offshore tidal stream turbine, The Seaflow was built into the seabed 1.5 km off shore from Lynmouth, Devon. It comprises an 11 m diameter twin bladed turbine and is capable of producing 300 kW of electricity at a tidal flow of about 2.8 m/s (5.5 knots) [3]. MCT has also developed the more recent 1.2 MW SeaGen project at Strangford Lough off the coast of Northern Ireland which will supply up to 1000 homes with electricity [4,5]. MCT have also commenced studies for an array of turbines producing 10.5 MW situated in the fast flowing waters within The Skerries off the coast of Anglesey, North Wales [6].

The clear advantage of tidal stream turbines is that they can be sized to suit the requirements of the local environment, i.e. coastal restrictions, tidal flow, tidal range, seabed topography, etc., and can be placed on either an individual or 'farm' configuration. As such, no large civil works, as compared to tidal impoundment schemes, are required and this method would therefore be less disruptive to wildlife, marine activity (and possibly the coastline) and would not

present a significant barrier to water transport. It has been stated that the ideal site for a tidal stream turbine is to be within 1 km of the shoreline and at a depth of 20 to 30 m [7]. The ideal tidal speed is 2 to 3 m/s (between approx. 4 and 6 knots) as higher speeds can lead to blade loading problems [8].

A recent survey on the extractable tidal resource distribution by depth suggests that 63% of the total resource is at depths greater than 40 m with a Mean-Spring-Peak velocity (V_{msp}) range between 2.5 m/s and 5.5 m/s and above. Although more challenging to deploy and maintain, there is considerable resource at depths greater than 40 m where the resource is estimated to be 28% with a V_{msp} of 5.5 m/s and above. Between 30 m and 40 m depth the V_{msp} ranges between 2.5 m/s and 3.5 m/s with an extractable resource of 18%, [2]. It is within this latter velocity and depth range that tidal turbines are initially being developed, such as those previously discussed. It is unlikely that attention will be given to depths less than 25 m as the peak resources is around 3.4%.

The Carbon Trust report on resource availability splits the UK into 5 resource regions - The Channel Islands, Northern Islands, North West, Pentland and South West [9]. The total energy yield from each of these sites is 3.017 TWh/y, 1.045 TWh/y, 2.033 TWh/y, 8.12 TWh/y and 1.229 TWh/y, respectively [9]. For the South West, which includes four locations within the Bristol Channel, namely Barry, Foreland Point and South and North Lundy the total energy yield from these four locations is 712 GWh/y, representing around 58% of the total energy yield from the South West region and around 5% of the total UK resource, making the Bristol Channel a viable energy source.

This paper considers the characteristics of a 10 m diameter, three bladed tidal stream turbine positioned in deep water, i.e. 50 m. The fixing or anchoring of a tidal stream turbine is briefly considered including free floating turbines. The torque generated, however, will have a propensity to turn the whole device. An option of counteracting this problem, and investigate the possible increase in power generated, via a design of 10 m diameter, double rows of contra-rotating blades is discussed and compared to the single turbine. The concept of using rows of contra-rotating blades is not a new concept, being used as early as the 19th century, and has been successfully used for propulsion systems for both aircraft and shipping and is described in depth by Carlton [10]. This paper will also evaluate the potential for use in HATT designs regarding only the characteristics. What is not considered is the likely high increase in both capital and maintenance expenditure for additional complexity, gearing, blades, etc.

1.1 Stanchion and turbine interaction

To allow the operation of a HATT there must be a means of fixing the turbine at some depth through the water column. The means by which the turbine is attached will greatly depend on the depth of the water. It is however inevitable that the extraction efficiency of

the turbine will be affected by any fixing structure. At depths greater than 40 m the application of a stanchion as a seabed fixing method becomes more difficult [11]. Another fixing method is via the use of tethering. This system has the advantage of positioning the device higher in the water column without introducing excessive turning moments such as those imposed on a pile or stanchion fixed into the seabed. The tethered device could be attached to the seabed using sinks and floatation devices at the water surface. Alternatively ballast could be applied to the HATT assembly with neutral buoyancy allowing the assembly to float at a predefined depth. It has been suggested that using this type of methodology would have a number of potential advantages over the more conventional stanchion/pile design. One such advantage is the elimination of the torque moment transmitted to the support structure thereby reducing the height from the seabed at which the device could potentially operate [12]. However, the method used to tether a floating HATT will need to limit the amount of yaw induced from the reactive torque generated by the turbine blades during power extraction. In an attempt to overcome this problem, Clarke et al [13] proposed the use of a contra-rotating turbine design where the reactive torque of the rear turbine corrects the yaw, returning alignment and rectilinear flow. It also allows a simple and economic mooring system for deep water applications. A further advantage of the system included the reduction of stable wake vertical elements in the wake of the HATT. This latter feature may have implications when considering the spacing of an array of tethered HATTs. When considering an array of HATTs the comparative costs between tethered and piled were summarised in a 2001 study undertaken to investigate the installation of 3 to 5 m diameter HATTs in 20 m water depth at fixed navigation marks. Capital costs of £400k and £600k for a tethered and fixed installation were given, respectively, [14]. An estimated breakdown of the cost for a farm of pile-mounted tidal turbines was contained within a report by Previsic in 2006 [15] and summarised by Clarke, et al, [16]. The costs consisted of 33% for the structural steel elements, 16% for the turbine installation, 35% for the power conversion system. This meant that the stanchion mounting system made up a considerable proportion of the overall cost. This then makes a valuable case for free floating systems that can be tethered to the seabed.

2 Reference CFD models

In accordance with the deep water conditions outlined by Black and Veatch [2], a 50 m water depth was selected for the CFD model. The model consists of a single 3 bladed 10 m diameter HATT located in a cylindrical Moving Reference Frame (MRF) with its axis of rotation 25 m below the surface boundary, Figure 1. The blades used the Wortmann FX 63-137 profile as the basis of the design with the key geometric values being: length 3.8 m, tip and root chord lengths 0.6 and 1.5 m respectively and a twist of

34°. The design was originated using a mixture of an in-house BEM programme and CFD modelling. The turbine model is a scaled version of a 0.5 m diameter turbine which has been validated against data from a physical model in a water flume with a uniform velocity profile with a measured turbulence intensity of 5%. A maximum uncertainty in the velocity profile, measured by a pitot tube, was estimated to be better than 5% over the water depth. This was limited by the resolution of the instrumentation. Details of the validation are presented elsewhere by Mason-Jones [17]. It should be noted that for the purpose of considering the effect of the contra-rotating turbine arrangement the model is compared only to a single turbine, i.e. without the presence of a stanchion. As an approximation the presence of a stanchion, with a diameter approximately equal to that of the hub diameter, the reduction on the extracted power could be ~11% [17]. The turbine volume was modelled using a cylindrical MRF volume with its axis of rotation through the centre of the hub to allow the angular velocity of the turbine to be varied. Given the complex shape of the blades and geometry between the blades and the hub, the MRF cylindrical volume was meshed with a tetrahedral hybrid scheme. The MRF volume was subtracted from the rectangular channel representing the remainder model. The turbulence intensity was based on the Hydraulic Diameter method, where the characteristic length is defined by $L_c = 0.5b_T$. This resulted in 5% being applied at both the velocity-inlet and pressure-outlet. The velocity-inlet and pressure-outlet were positioned 2 diameters upstream and ~38 diameters downstream of the turbine, respectively.

To simulate an open water scenario zero friction was applied to the sides and surface boundaries of the channel, however for the seabed the no-slip formulation was assumed.

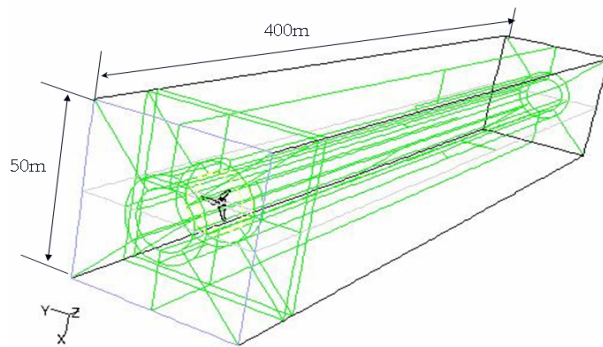


Figure 1 Outline of CFD model

No interaction between surface waves and tidal current was considered, particularly since the distance from the blade tip to the surface was deemed large enough to ensure no significant effect on the turbine characteristics. For uniform velocity conditions, the velocity profile was allowed to develop upstream of the turbine with a peak value of 3.1 m/s (6 knots). The latter velocity was chosen to define peak conditions for the turbine as 3.1m/s is approaching the higher end of

most of the potential UK sites [2]. The density of seawater was taken as 1025 kg/m³.

The meshed model included ~1 million cells around the turbine and a further 90 000 cells for the remaining flow field. The cell count was controlled by the number of cells initially generated while meshing the faces of each turbine blade and hub. A finer mesh density was placed towards the tip of the blade within the last 1/3 of the blade length. The upstream and downstream faces were meshed with increasing mesh densities. The model used the Reynolds Stress Model (RSM) to close the ‘Reynolds Averaged Navier-Stokes’ (RANS) equations that relate the Reynolds Stresses to the mean velocity gradients. This viscous model was chosen since it was expected that the rotation of the turbine, would introduce anisotropic turbulence and the RSM is recommended for such situations, [18].

At each converged steady-state solution, a User Defined Function (UDF) was used to extract the torque (T) and axial thrust force (F_T). The peak torque (T_{max}) was calculated at every converged solution by integrating and resolving forces at each cell face via the UDF. The product of $T\omega$ (W) was then used to calculate the peak power P_{max} (W). Finally, the power coefficient for the 10 m swept area was calculated using Equation 1.

$$C_p = \frac{T\omega}{0.5\rho av^3} \quad (\text{Equation 1})$$

From the angular velocity (ω) sweeps, run over a range of blade pitch angles, a series of power curves were developed. The pitch angle (θ) is defined as the angle between the chord of the blade and the normal to the rotational axis of the turbine hub. To determine the optimum pitch angle (θ_p), 5 models were created with the pitch angle varying from 0° to 12°. Figure 2 shows the peak power obtained at each pitch angle over the angular velocity sweep. The results show that peak power extraction occurred with a blade pitch of 6°. From the power curves it is also evident that the angular velocity at which peak power occurs, shifts with changes in θ . With the blade pitch at 6°, the peak power occurred at an angular speed of 2.25 rad/s whilst the lowest peak power occurs with a blade pitch of 12° where the angular velocity at peak power is approximately 1.8 rad/s.

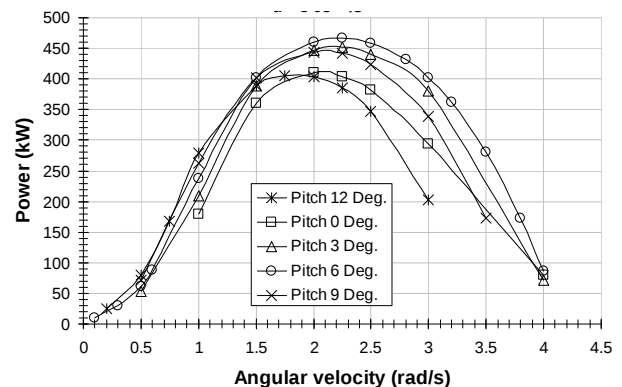


Figure 2 Power extracted with blade pitch variation.

In addition, the Tip Speed Ratio (TSR), which is defined by equation 2, shifts from an optimal value of 3.6 at 6° to 2.92 at 12° indicating that at larger pitch angles the rotational velocity of the turbine must decrease to obtain optimum power.

$$TSR = \omega r / v \quad (\text{Equation 2})$$

Two further inlet velocities corresponding to tidal velocities of 1.54 m/s and 2 m/s were selected to identify any variation in the pitch angle required to maintain optimum power extraction. It was found that the optimum pitch angle for the blade proved to be insensitive to tidal velocity, at least within the specified range.

The power coefficient was calculated for each blade pitch and tidal velocity and normalised to the maximum value as shown in Figure 3. This emphasises the fact that peak power extraction occurs at an optimum pitch of 6°.

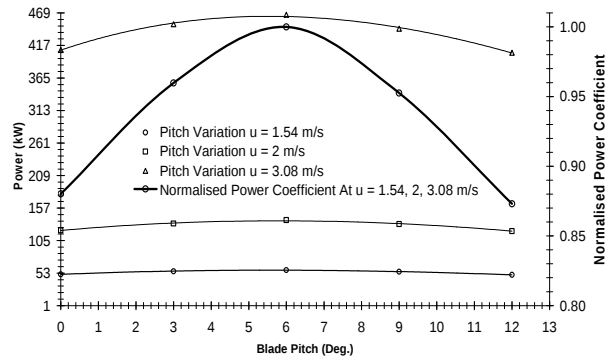


Figure 3 Turbine Power characteristics

Having identified the optimal pitch angle as 6°, further data were generated by using smaller incremental values of ω to add additional detail to the power curve and to include torque and axial thrust profiles under the same peak flow conditions. Figure 4 gives the full performance characteristics. The torque generated for a stationary turbine (Start-up) is approximately 96 kNm which increases to a maximum of 275 kNm at 1.3 rad/s. After approximately 1.5 rad/s the torque decreases relatively linearly to zero. In reality however the torque can only approach zero at a high rotational velocity since zero torque would imply no lift forces and hence zero angular velocity through stall. The turbine will remain just below the complete stall angular velocity at a freewheeling state.

The general shape of the power curve is cubic except for a slight 'tail' towards the lower rotational velocities, in this case between 0 and 0.6 rad/s. Following the relatively linear decrease in torque after its peak at 1.3 rad/s the power maintains a steady increase along the curve. At the inflection of the curve a maximum power of approximately 466 kW was obtained at a rotational velocity of 2.25 rad/s around 0.95 rad/s above the peak torque. The torque then reduces to zero along the remaining curve. The shape of the power curve is due to the product $T\omega$. At a peak

torque of 275 kNm, the angular velocity is 1.3 rad/s giving a power of 356 kW which is 110 kW below the peak. A 10 m diameter swept area gives 1.2 MW of available resource when using Equation 1, resulting in a C_p of ~0.4.

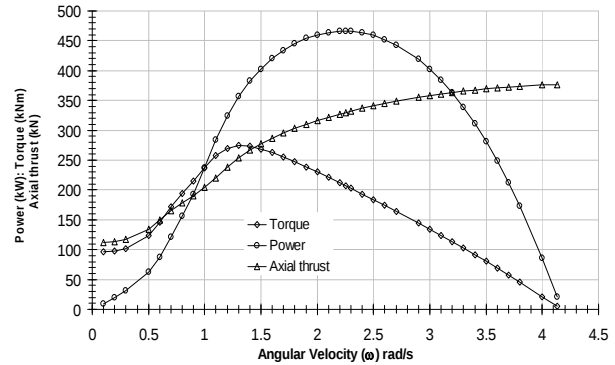


Figure 4 Variation of power, torque and axial load with angular velocity

The axial thrust coefficient, which is defined by Equation 3, was calculated for each blade pitch angle.

$$C_T = \frac{F_T}{0.5 \rho a v^2} \quad (\text{Equation 3})$$

Figure 5 shows how the axial thrust and power coefficients, C_T and C_p respectively, are affected as θ is varied with an inlet velocity of 3.1 m/s.

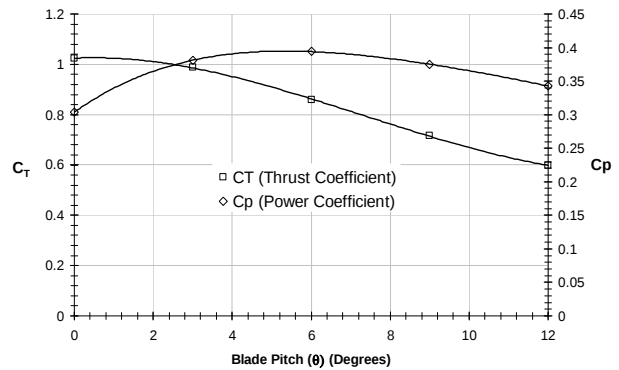


Figure 5 Variation in power and axial thrust coefficients with blade pitch angle.

Again peak C_p occurs at a blade pitch angle of 6°. A greater rate of decay in power extraction is indicated as θ approaches 0° with a 23% reduction in peak C_p , however as θ approaches 12°, C_p is only reduced by 13%. As for C_T , it varies between 0.58 and 1 for angles of θ between 0° and 12°. Therefore, as θ approaches 12° a 33% reduction in C_T is realised while maintaining a C_p of approximately 35%.

In addition, the modelling described above was repeated for a shallower domain of 20 m depth. All other conditions were maintained. The findings were that there were minor differences in the turbine characteristics. These differences produced slight

increases in the peak values of power (~ 30 kW), torque (~ 10 kNm) and axial thrust (~ 10 kN). For the purposes of this study, the data was assumed to be the same. This model will be referred to as the reference model in the rest of the paper.

3 Contra-rotating turbine performance

The operating conditions of the turbine presented in the reference studies (deep and shallow waters) are idealistic in the sense that there is no direct influence on the efficiency of power extraction from the seabed fixture, such as a stanchion. However, for applications where a relatively even velocity profile through the water column can be taken advantage of, the use of a stanchion maybe impractical for various reasons. An alternative to a stanchion mounted turbine is a free floating tethered system using contra-rotating turbines [13]. In such a system the torque generated by the rear turbine could be used to counteract that of the front turbine, thereby limiting the inefficiency introduced through yawing and hence misalignment with the upstream velocity field [12].

To give a first order approximation on the operation of a contra-rotating turbine assembly, using the same design HATT discussed earlier in this paper and by Mason-Jones [19], a series of contra-rotating steady-state CFD models were developed with varying axial spacing L , where L is the distance between the faces of rear and front hubs, Figure 6. The blades of the front and rear turbines were set at the optimum pitch angle of 6° with the rear blades rotated through 180° to account for the reversed rotation. The diameter and fundamental blade geometry of the rear turbine matched that of the front turbine. The velocity-inlet boundary used for this model was the same as that defined for the reference models, that is, a uniform velocity profile with a peak (developed) velocity of 3.1 m/s. As in the other cases a series of angular velocity sweeps were performed to identify the peak performance of each turbine under the given conditions and as a result of any near face interaction between the devices. The reference domain channel dimensions of $5D$ were used to limit flow concentration from the boundary effects. The turbulence specification method was once again based on the hydraulic diameter method giving a turbulence intensity of 5% .

With the introduction of a second turbine the maximum memory required to run the model on a single processor was exceeded. The volume was divided into three zones, the first contained the cells defining the sea, the second and third zones defined the front and rear turbine volumes respectively. The total number of tetrahedral cells in the turbine zones was 1.94 million, equally divided between them. The sea zone consisted of $202\,527$ tetrahedral cells. This gave a total cell count of 2.14 million cells for the whole domain. To overcome the computational limitations, the model used parallel processing with the domain split into 2 partitions using the principal axis method [18], Figure 7. This gave a total of $1\,070\,080$ cells for partition 1 and $1\,072\,566$ cells for partition 2, each

containing one of the turbines and approximately half the sea cells.

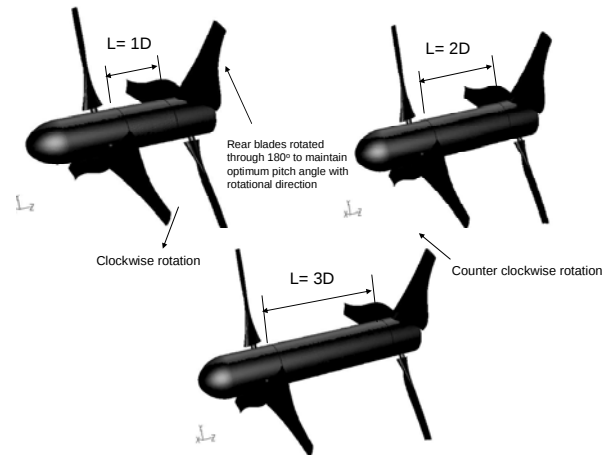


Figure 6 Contra-rotating turbines with varying axial spacing.

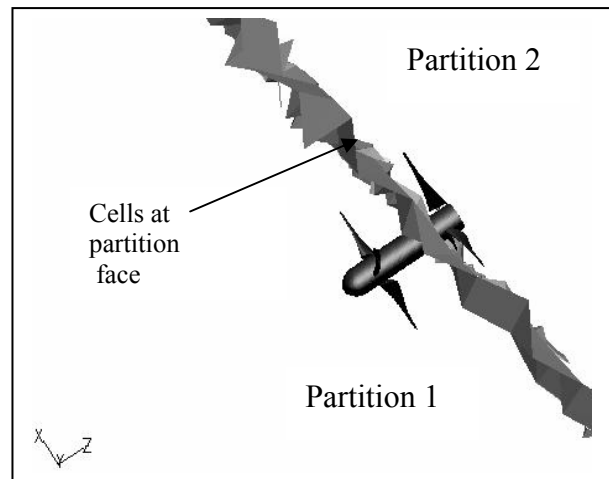


Figure 7 Mesh partition for $L=3D$ contra-rotating mesh

3.1 Power Characteristics

The modelling discussed for the contra-rotating turbine takes no account of the practicalities of designing and manufacturing such turbines. As discussed earlier the front row of blades have been optimised for their power generation and set to 6° . The rear row of blades uses the same design as the front row and has, for simplicity, been set at -6° . This has allowed the water swirling out of the front row of blades to be intercepted by the rear row. Figure 8 shows only the curves for $L = 1D$ and $3D$ for clarity. However, values of L between these follow the curves shown and can be interpolated from the data shown. It should also be reiterated that that data shown for the single turbine does not include a stanchion and that the extracted power could be around reduced to ~ 400 kW ($C_p \approx 0.34$).

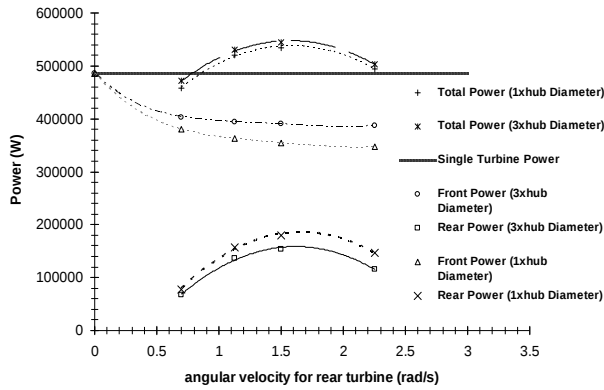


Figure 8 Power curves for 10m diameter contra-rotating turbines

The values shown for power generated from the front row of turbines are for $\omega = 2.25$ rad/s in all cases, which equates to the peak power for the turbine in the reference model. Figure 8 shows the effects of introducing the rear row of contra-rotating blades with regard to the power generation, where only the angular velocity of the rear row is varied. It is clearly noticeable, irrespective of the separation distance, (L) that the power curve of the rear row of blades follows the basic profile expected from a single turbine; although at much lower values. The power generated by the front row, however, shows a serious decrease with the curve dropping to an asymptotic level after the point in which the rear row's curve peaks, i.e. $\omega > 2$ rad/s. What can also be deduced from the figure is that, the greater the power generated by the rear row the lower the power generated by the front row. It is possible that as the rear row of blades rotates at a higher rate it causes an increased blockage, which in turn is 'seen' by the front row. Also since the velocity is lower after the front row of blades (and also more turbulent) due to the power removed, then the working range of the rear row of blades is also reduced. Hence after the rear row's power has peaked it rapidly decays until it is freewheeling, whilst, as said, the power generated by front row has become asymptotic. The result is the total power curve shows that, under these simplistic conditions, it is possible to generate around 13% more power, giving a $C_p \approx 0.46$ (or upto 33% if a stanchion is introduced). This also shows that with some optimisation of the angular velocities between the two rows of blades, and the blade pitch angles of both rows, there could be the potential to further increase this value.

3.2 Torque characteristics

Figure 9 shows the percentage of the torque generated by rear row of turbine blades as a function of the front row of blades. Since this row is contra-rotating the torque generated by the front row is obviously acting in the opposite direction to that generated by the rear row. As with the reference model the torque decays with an increasing angular velocity and will peak ahead of the peak power. Certainly from the point of view of countering the yaw that would be

created by tethering the turbine in the reference model, this design would only reduce the nett torque to 65% of that generated by the reference model. This would then only result in an increase in Power of $\sim 0.22\%$ ($C_p \approx 0.43$). However, the indications are that with the rear row of blades correctly sized and angled, the nett torque could be reduced significantly and the C_p increased. That is, the rear row of blades would have to be increased in length over that of the front row of blades, thus resulting in an increase in the force acting on the blades at an increased radial position. Alternatively, the number of blades would need to be increased which will change all the characteristics since the rotational speed of the peak power will change.

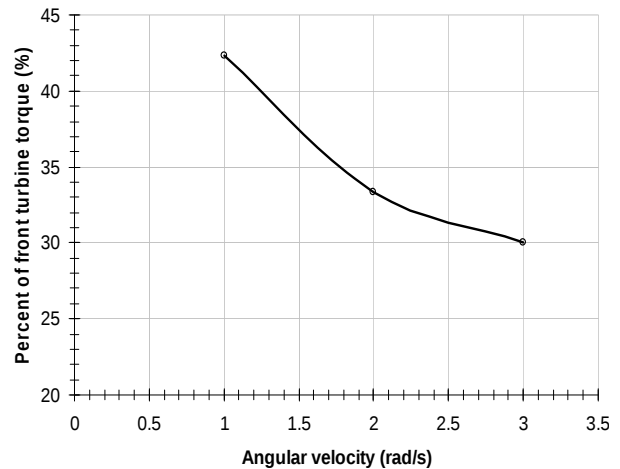


Figure 9 Percentage of rear turbine torque to front turbine

3.3 Axial Thrust characteristics

Interestingly when examining the axial thrust data it can be seen that there is only a small change in the level of thrust as a result of the variation in row separation, L (Figure 10). As L increases there is a small decrease in the axial thrust on the rear row of blades and a similarly small increase in the axial thrust on the front row of blades, over the angular velocity range. This results in little total change for any given angular velocity.

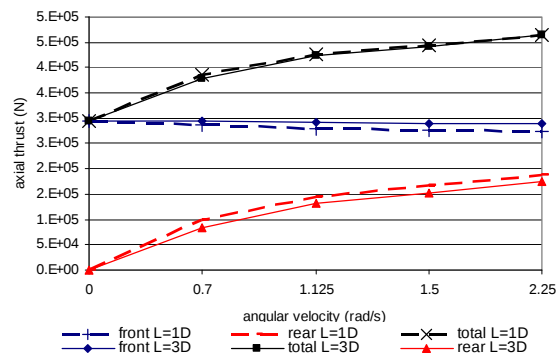


Figure 10 Predicted axial thrust for the contra-rotating turbine

However, comparing the point at which the total power peaks, i.e. $\omega = 2.25$ rad/s (constant) for the reference turbine and $\omega = 1.7$ rad/s for the rear blades

of the contra-rotating turbine then the C_T shows significant differences, Figure 11. For the reference turbine $C_T = 0.84$ whereas for the contra-rotating turbine $C_T = 1.18$, i.e. an increase in the axial thrust of $\sim 40\%$. This gives an axial thrust of ~ 460 kN compared to ~ 350 kN on the reference turbine. It should be noted that if the axial thrust on the reference turbine included a stanchion then the peak axial thrust would increase to the order of 650 kN [19]. Obviously if the aim was to reduce the nett torque to zero there would be a considerable increase in the axial thrust. So whilst the turbine wouldn't yaw due to the torque, the cables tethering the turbine to the seabed would need to have a large increase in their rating.

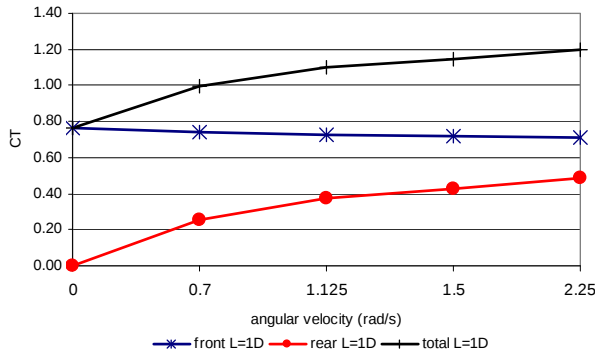


Figure 11 Thrust coefficient for the contra-rotating turbine.

3.4 Wake characteristics

When considering Figure 12 and 13 of the reference turbine, it can be seen that the flow is as expected around a partial blockage. Initially the fluid slows upstream as it approaches the turbine. Some of the fluid then travels through the blade spacings, having been deflected off the blades, whilst some is deflected around the outside of the swept area. The fluid is accelerated off the tips causing the wake to expand such that the maximum wake diameter can be as much as three times the turbine diameter. With momentum being conserved, the velocity immediately downstream of the turbine collapses from the far field velocity of 3.1 m/s to around a half this value (~ 1.6 m/s), for a blade pitch angle of 6° in deep water. As the wake collapses in on itself the centreline velocity makes a steady recovery. However, the wake takes nearly 20 turbine diameters to reach 71% recovery and 40 diameters for a 90% recovery Figure 12. This figure also shows the effect of blade pitch angle on the wake characteristics. It is clear that as the blade angle increases so the drop in the velocity immediately downstream of the blades reduces. This is as expected since at 0° the blades present a larger blockage than at higher angles. Further downstream, however, the wake recovery does not 'recognise' this difference and a 71% and 90% recovery occurs at approximately the same distance.

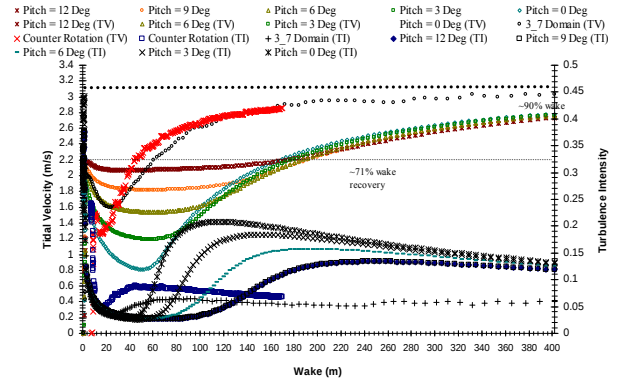


Figure 12 Comparison of wake recovery for the reference turbine with the contra-rotating turbine at both 50 m and 20 m water depths. Inlet velocity = 3.1 m/s TI = 5%.

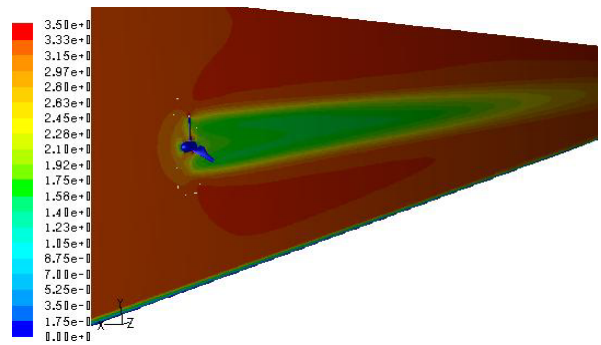


Figure 13 Fluent contour plot of reference turbine centreline velocity m/s.

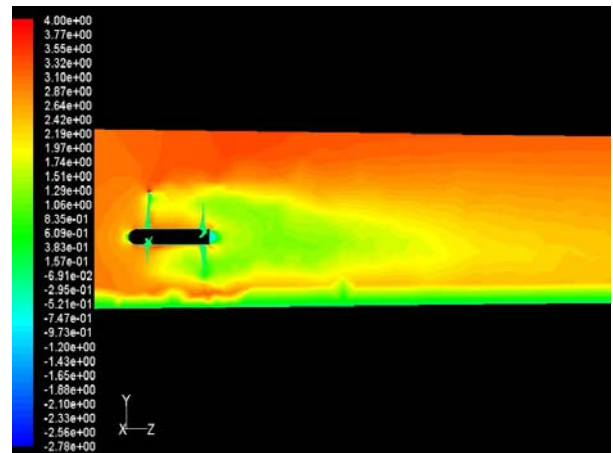


Figure 14 Fluent contour plot of contra-rotating turbine centreline velocity m/s.

In the shallower water the recovery occurs earlier, but the reduction in the velocity magnitude immediately downstream of the blades is the same at 1.6 m/s. However the recovery occurs faster than for deeper water with 71% at ~ 5 turbine diameters and 90% at ~ 15 turbine diameters. The contra-rotating turbine has a greater drop in the velocity immediately downstream of the blades, dropping to $\sim 37\%$ (i.e. 1.15 m/s) of the upstream velocity. The recovery is initially faster than the reference turbine with 71% at

~4 turbine diameters as can be seen in Figure 12 and Figure 14. This is then followed by a recovery that is similar to that seen with the reference turbine, i.e. 90% at ~15 turbine diameters.

The turbulence intensity of the contra-rotating turbine downstream of the blades, peaking at ~8% is ~33% higher than that of the reference turbine, at ~6%. When compared to the deep water data for the reference turbine which peaks at 20%, it can be estimated that the turbulence intensity for a contra-rotating turbine would be ~27%. It can also be estimated that the wake of the contra-rotating turbine would have a 71% recovery by ~12 turbine diameters.

The explanation for the initial recovery occurring sooner for the contra-rotating turbine is that the turbulence off the back of the front blades is higher, therefore mixing occurs at a faster rate with some of the surrounding, higher velocity water. As the level of turbulence reduces so the rate of recovery slows and matches that of the reference turbine. In the deeper water the flow is not confined therefore as the wake can expand to its full diameter the expansion causes a greater increase in the level of turbulence. The axial velocity surrounding the turbine is able to stretch and carry the mixing area further downstream than the shallower, confined flow.

4 Conclusions

An increase in power generation can be achieved with the use of a contra-rotating turbine over a single turbine.

With correctly designed blade rows, i.e. blade length, number of blades and row spacing, it should be possible to provide a zero nett torque to prevent yawing of tethered turbines.

There is a large increase in the axial thrust acting on the contra-rotating turbine when compared to a single turbine. This does not take into account the axial thrust present if a stanchion was used. It does mean that the tethering ropes would have to be suitably rated to withstand the increased axial thrust.

The initial wake recovery for the contra-rotating turbine is faster than that of a single turbine, although full recovery is the same.

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