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Feasibility Study Using Computational Fluid Dynamics for the Use of a Turbine for Extracting Energy from the Tide

D.A. Egarr[1], T. O'Doherty[1], S. Morris[1] and R.G. Ayre[2]

[1]Cardiff University, Queen's Buildings, The Parade,
PO Box 925, Cardiff, CF24 0YF, Wales, UK

[2]Tidal Hydraulic Generators Ltd, 7 Dolan Court, Broad Haven,
Haverford West, Pembrokeshire, SA62 3JN, Wales, UK

Abstract

A 4 blade tidal turbine has been modelled using computational fluid dynamics and validated against experimental data prior to modelling a 3 blade turbine. This is the first stage in the process of optimising a turbine to extract energy from the tide. The 3 blade turbine has been modelled in a number of tidal flows between 0.51 and 3.09m/s (1 and 6 knots) and the power predictions studied as well as the magnitude of the force on the turbine. The redevelopment of the flow downstream from the turbine has also been studied.

Introduction

The Kyoto agreement of 1997 was that each country that acceded the agreement will reduce or limit its greenhouse gas emissions by an amount agreed, such that the overall emissions of such gases is reduced by at least 5% compared to 1990 levels in the commitment period 2008 to 20012 [6]. It should be noted that the protocol acknowledges developing countries and in some cases, allows particular countries, such as Australia and Iceland to increase their emissions. For those that have to reduce emissions, alternative sources to fossil fuels are required for producing electricity. The British government currently has no plans for development of further nuclear power plants [4]. This leaves renewable sources of energy as a viable option to reduce greenhouse gases. The British government has 'confirmed a timetable up to 2015 when 15.4% of electricity generated in the UK is from renewable sources' [5]. One of the most popular renewable sources is wind energy, whereby a single turbine with 66m diameter rotor blades is capable of producing 1.5MW.

Another source of renewable energy is that harnessed from the sea. Recently, interest has focussed on capturing the energy in the tides. The gravitational force of attraction of the moon and sun on the earth and the rotation of the earth results in a rhythmic rise and fall of ocean levels relative to the coastline [1]. Thus, unlike wind and solar energy, tidal currents are predictable. In addition to this advantage the available power flux from the tide is around 500W/m^2 compared to 40W/m^2 for air. This is based on the density of water being some 815 times that of air and the tidal velocity of the order of 25% that of air. There is a large quantity of literature on the methods that can be utilised for extracting energy from the tides but the use of turbines in tidal streams, due to the advances in technology, is only now becoming reality with the world's first full scale offshore tidal turbine being constructed in 2003 [2].

This paper looks to assess the feasibility of energy generation from the tide, using the Computational Fluid Dynamic (CFD) package FLUENT. Initial models are validated against full scale experimental trials of a 4 blade turbine. Modelling of a 3 blade turbine has also been undertaken.

Experimental trials

Tidal Hydraulic Generators Ltd. (THGL) undertook the experimental trials in 2002 on the Cleddau River near Milford Haven in Pembrokeshire, UK, using a four blade, 5.5m diameter turbine. These trials were reported to be the first large scale tests to be undertaken on a river in the UK. The blades were constructed from Glass Reinforced Plastic (GRP) and held on a welded steel tube assembly. The turbine was mounted on a swivel system so that it could be easily lowered from the back of a barge during slack tide, shown in figure 1. When lowered, the turbine was held 3.35m, to the hub centre, below the surface. An RPM sensor disc measured the angular velocity of the turbine through a drive system. A load was applied, via a lever arm, to the drive shaft and thus the power extracted by the turbine could be determined using equation (4). Table 1 contains data measured in an average tidal flow of 0.9m/s (1.75 knots). Hence, with applied load, geometry and gear rates known the power generated was determined using equation (4).

Figure 1 (Provided by THGL © 2002). The 4 blade, 5.5m turbine on the Cleddau River during experimental trials.

The tidal flow varies due to the tide change, hence a constant tidal flow is therefore difficult to capture at which a set of data can be taken.

Series A		Series B	
Turbine angular velocity, / rad/s	Power, / W	Turbine angular velocity, / rad/s	Power, / W
1.49	1677	1.48	0
1.21	2724	1.26	1650
0.79	2820	1.05	1965
0.66	2464	0.87	2456
		0.64	2161

Table 1. Experimental data, measured in a 0.9m/s (1.75 knot) tidal flow (Data provided by THGL Ltd.)

The blade used in the experimental trials was surveyed to produce a set of vertex data. The vertices were connected to form a blade comprising of over 900 faces. Further 3D CAD manipulation resulted in the blade comprising of 6 faces. This was used in the CFD modelling.

Computational Fluid Dynamics

Computational Fluid Dynamics is a tool used in a variety of engineering fields, such as automotive, process and micro-processor. The CFD software solves the 'Reynolds Averaged Navier-Stokes' (RANS) equations and, depending on the viscous model chosen, relates the Reynolds stresses to the mean velocity gradients, or closes the RANS equation by solving transport equations for the Reynolds Stresses [3]. This yields a flow field for which quantities such as fluid velocity, pressure and turbulence can be obtained. The advantage of CFD is that it is relatively inexpensive compared to experimental trials; providing a validation is made between CFD and experimental data. This validation and further modelling of a 3 blade turbine is presented. This is the first stage of the optimisation of a turbine for extracting energy from the tide.

Turbine Power

The maximum theoretical efficiency of a frictionless turbine, found to be 59.3%, was derived by A. Betz in 1920. The derivation of which is documented in many fluid mechanics texts such as White [7]. A real turbine is not frictionless, as is assumed by the derivation of the Betz limit, thus the force acting on the turbine is made up of the shear force on the turbine blades and the force due to the static pressure, hence

$$F = F_S + F_P \tag{1}$$

Where: F = Total force, N F_S = Shear force, N
F_P = Static pressure force, N

To calculate the force on a turbine blade that has been divided into finite faces, the forces in the x, y and z components must be considered. The force due to the static pressure is given by the product of the static pressure and the area vector of the face, hence, in the x-component

$$F_{Px} = p.A_x \tag{2}$$

Where: F_{Px} = Force due to static pressure in x-component, N
p = Static pressure acting on the element, N/m^2
A_x = Area vector of the face in x-component, m^2

Similar expressions apply for the y and z component.

The torque is required only for the components of the forces in which the plane of the turbine lies. Hence, if the turbine axis is in the z-component, then only the forces in the x and y components are required and for simplicity, consider the case for the origin of the axis of rotation of the turbine at (0,0,0). Then the total torque acting on the turbine is the summation of the torque acting on each face. The torque on each face is given by the cross-product of the distance and force vectors, hence:

$$T = \sum_{n=1}^{n=N} \left[r_x F_y - r_y F_x \right] \tag{3}$$

Where: T = Torque, Nm
r_x = Distance in x-component, m
r_y = Distance in y-component, m
F_x = Force vector in x-component (sum of shear force and static pressure force), N
F_y = Force vector in y-component (sum of shear force and static pressure force), N

The power being extracted by the turbine is then given by:

$$P = T\omega \tag{4}$$

Where: ω = Angular velocity of turbine, rad/s
P = Extracted power, / W

The maximum power available to a turbine is the kinetic energy of the fluid in a stream tube whose diameter is equal to the diameter of the turbine. Thus:

$$P_{\max} = \frac{1}{2}\rho A v^3 \tag{5}$$

Where: $P_{\max}$ = Power available, W
ρ = Density of fluid, kg/m^3
A = Cross-sectional area of stream tube, m^2
v = Free stream velocity of fluid, m/s

This allows the efficiency of the turbine, η, to be defined as

$$\eta = 100.\frac{P}{P_{\max}} \tag{6}$$

Meshing and grid dependency

A check for grid dependency was made by comparing two meshes created from tetrahedral elements; one comprising of 300 000 cells and two cell zones, with a non-conformal interface and the other 800 000 cells and one cell zone. The significance of a grid with two cell zones, with a non-conformal interface, is that averaging errors occur across the interface. This is an inherent problem when modelling rotational and/or translational parts. The difference in power prediction between the two grids was 0.32%, hence a grid of 300 000 cells was considered grid independent. The actual grid used comprised of 500 000 cells, shown in figure 2, to achieve a slightly higher resolution grid downstream of the turbine for studying the flow field. Custom sizing functions were used to attain a slightly higher quality grid in parts of the model.

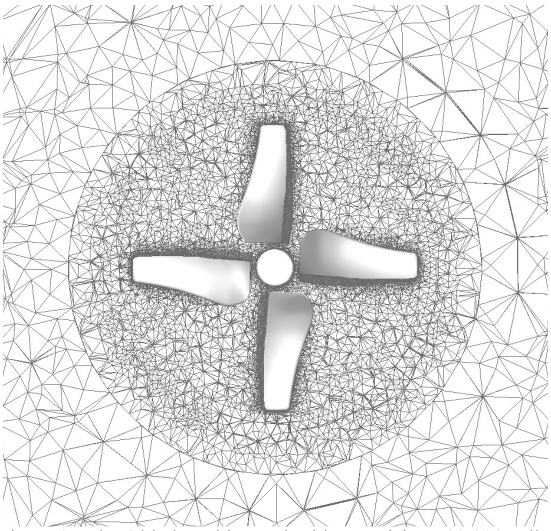

Figure 2. The 4 blade turbine and grid comprising 500 000 cells.

The topology of the region in which a tidal turbine can be located varies from site to site. It was felt that for assessing the power of the turbine and studying the flow field downstream from the turbine a domain the shape of a rectangular channel with a flat bed would be sufficient and a free surface represented by a frictionless wall. A frictionless wall was chosen over the use of the 'Volume of Fluid' (VOF) multiphase model, because it was felt the extra computation time in using the VOF model was not justified, as the flow field around the free surface was not of interest. Initially, models were created with different widths, to choose a width of channel that minimised boundary effects, but which was sufficiently small to keep the grid size as low as possible.

The width of the channel chosen was approximately 6 times the turbine's diameter and the sides of the channel were specified as frictionless walls, to reduce boundary layer effects. In all the models, the turbine axis of rotation was parallel to the flow and a steady state solution was attained. This was justified by the fact that the turbine arrangement would allow the 'head' to swivel on a stanchion to face the tidal flow at all times. The inlet to the channel was specified as a velocity inlet, where a constant velocity was specified across the entire surface of the inlet. This ensured that the free stream velocity of the fluid approaching the turbine was that specified at the inlet. The velocity profile due to boundary layer effects adjacent to the bed of the model therefore developed within the model. In this region the grid was fairly coarse and had negligible, if any effect on the flow around the turbine. A range of turbulence intensities at the inlet were investigated and it was found that for the domain in which the turbine was being modelled, a turbulence intensity of 2% was appropriate.

Validation

The CFD model of the 4 blade, 5.5m diameter turbine was simulated in a 0.9m/s (1.75 knot) tidal flow. The experimental data sets are small, since their collection was difficult and expensive. Hence, the experimental data in table 1 was averaged; a polynomial was fit to each set and at a number of angular velocities, the power for each polynomial calculated, for which an average was taken. Figure 3 shows good agreement between the CFD and an average of the experimental data. There is a 4% difference in peak power and approximately 15% difference in the angular velocity at which the peak power occurs.

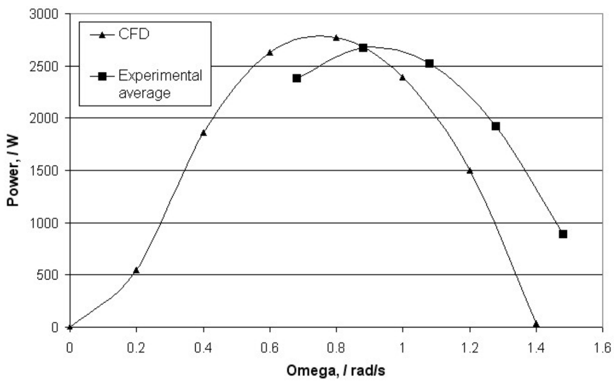

Figure 3. Comparison of CFD and experimental power for a 5.5m diameter turbine in a 0.9m/s (1.75 knot) tidal flow.

3 Blade Turbine

In order to reduce the cost of manufacture, a 3 blade turbine was preferable and to match the power of a 4 blade 5.5m diameter turbine, the diameter of the 3 blade turbine was increased to six metres. Thus, having acquired good agreement between CFD and experimental data, modelling of a 3 blade, 6m diameter turbine was undertaken, shown in figure 4, using a scaled version of the blade used for the four blade turbine.

To ensure that the optimum power from the 3 blade turbine was achieved, the blade pitch was investigated, to ensure that the blade setting was appropriate for the new geometry. Having established the maximum efficiency of the turbine at a minium of 3 blade pitches, the optimum blade pitch was identified by plotting efficiency vs. blade pitch and fitting a third order polynomial to the data. The pitch at which the turning point of the polynomial occurred was determined by means of differentiation of the polynomial. The turbine was then re-modelled at the estimated optimum blade pitch and the procedure repeated until the optimum blade pitch of 2.6° was attained. As this was the method utilised, the blade pitches for which efficiency data have been acquired are not evenly distributed in figure 5.

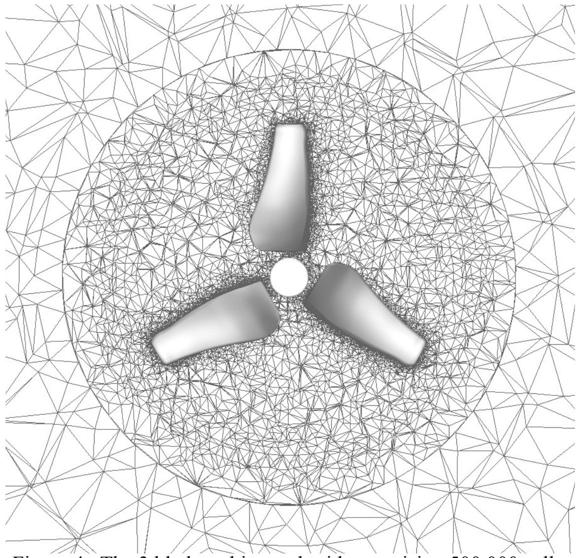

Figure 4. The 3 blade turbine and grid comprising 500 000 cells.

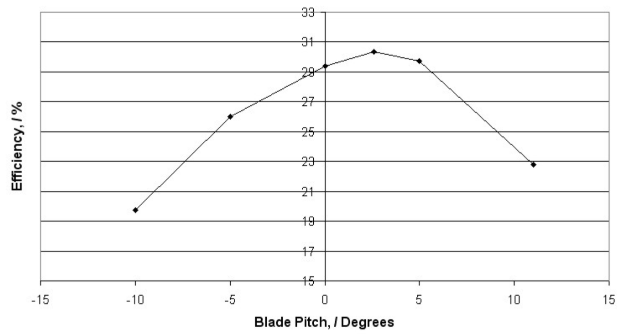

Figure 5. Variation in efficiency with blade pitch of a 3 blade, 6m diameter turbine in a 3.09m/s (6 knot flow).

With the optimum blade pitch established, the turbine was modelled in a range of tidal flowrates. Figure 6 shows how the power varies with tidal flow.

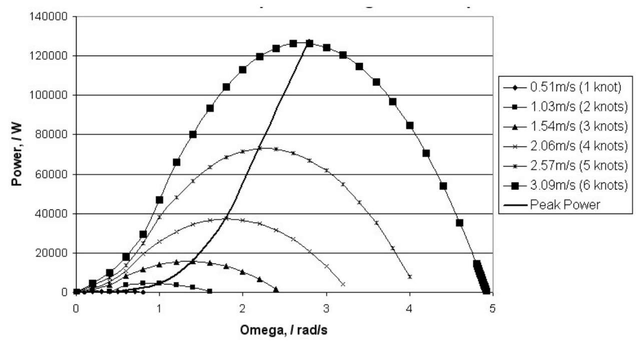

Figure 6. Three blade turbine power predictions for a range of current velocities (Blade pitch of 2.6 degrees at the tip).

From figure 6 it can be seen that there is a clear function between the maximum power attainable for the given turbine and the tidal flow, given by:

$$P = 4289v^3 = 5755\omega^3 \tag{7}$$

Where: v = Free stream velocity of fluid, m/s
ω = Angular velocity of turbine, rad/s

However, when considering the optimal power extraction by the turbine, the increasing force that is exerted on the turbine via the blades must be considered as the turbine would have to be supported by a structure anchored to or piled into the sea bed. The magnitude of the force on the turbine has therefore been considered at each tidal flow and angular velocity at which a data point is presented in figure 7.

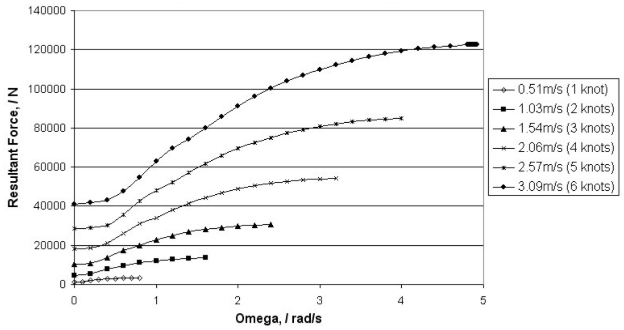

Figure 7. Force magnitude on the 3 blade turbine for a range of current velocities (Blade pitch of 2.6 degrees at the tip).

From figure 7, as the turbine approaches the free wheeling speed, the force on the turbine starts to plateau. This was investigated in more detail (indicated by the greater number of points in figures 6 and 7) as the turbine in a 3.09m/s (6 knot) tidal flow approaches the free wheeling velocity. The conclusion is that the peak force on the turbine appears to occur when the turbine is free wheeling.

As with wind turbines potential sites will exist where tidal turbine farms would be desirable. Hence where turbines could be in line with each other, then the development of the flow downstream from a turbine is of interest. This information will then aid the optimum positioning of additional turbines. Figure 8 shows the velocity magnitude of the fluid along the turbine axis at peak power extraction in a 3.09m/s (6 knot) tidal flow. The turbine is positioned at 0m. It can be seen that as the fluid approaches the turbine, the fluid in effect 'sees' the turbine and the velocity drops. The fluid velocity must clearly drop to zero as the axis passes through the turbine hub. Immediately behind the hub of the turbine there is a recirculation zone, which forces the velocity magnitude to peak and then drop. Once past this the fluid gradually recovers and reaches 90% of the free stream at approximately 70m downstream of the turbine i.e. approximately 12 turbine diameters. An 80% recovery is achieved however in the first 40m i.e. approximately 7 turbine diameters. This indicating that the closest position for a downstream turbine based on the velocity magnitude would be approximately 7 turbine diameters. Vortex shedding has also been examined and was found to decay considerably faster than the distance required for redevelopment of the flow.

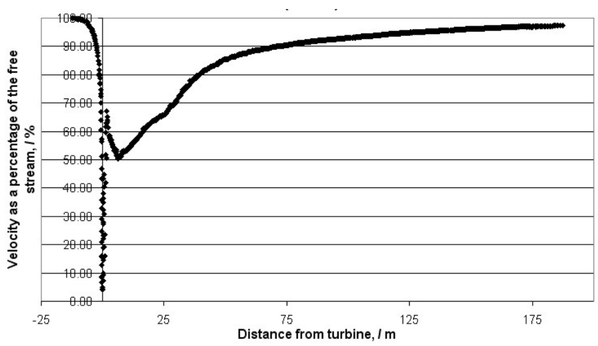

Figure 8. Development of the flow downstream from the turbine at peak power extraction in a 3.09m/s (6 knot) tidal flow.

This recovery in the downstream velocity is best illustrated in figure 9, which shows an iso-surface of velocity magnitude of 2.27m/s (74% of the free stream velocity) around the turbine, with all contours coloured by velocity magnitude. Here the flow is passing from right to left, with the turbine situated at the extreme right of the iso-surface. As the flow develops downstream, the envelope effectively collapses in on itself as the free stream velocity recovers.

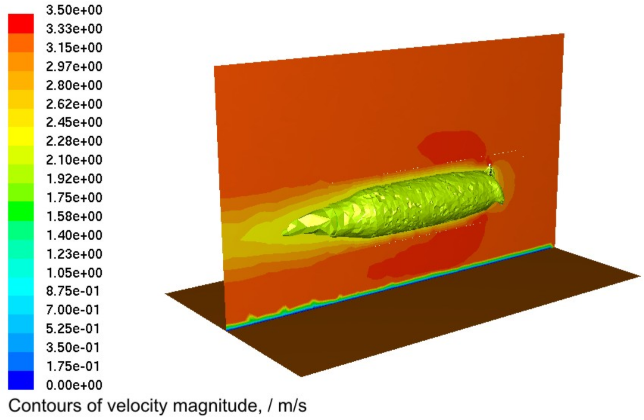

Figure 9. Contours of velocity magnitude in a 3.09m/s (6 knot) tidal flow.

Conclusion

It has been found that in using CFD, a good prediction of power extraction by a turbine can be made. Work has concentrated on a 3 blade turbine, for which power predictions have been made at a number of flowrates. The force on the turbine has been investigated and found to peak at the free wheeling velocity. The redevelopment of the flow along the turbine axis has been investigated, as well as the velocity field around the turbine. Although the efficiency of this particular blade is 31%, the work presented here is the first stage in the process of detailed modelling of tidal turbines. More recent work has found that a tidal turbine can be designed to have an efficiency of 40 to 45%. The results presented here will differ slightly for an optimised turbine blade, but it is expected that the trend of the data will remain the same.

This work clearly indicates the feasibility for well designed tidal turbines, positioned in suitable tidal streams, to extract energy from the tide. Further work is to be undertaken to model an improved blade and other parameters such as scaling and improved boundary conditions.

Acknowledgements

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