Proceedings of the 13th European Wave and Tidal Energy Conference 1-6 Sept 2019, Naples, Italy

Unsteady loading prediction on tidal turbine blades using computational fluid dynamics

William Finnegan and Jamie Goggins

Abstract— **After wind energy, tidal energy presents the most prominent opportunity for generating energy from renewable sources. However, due to the harsh environment that tidal turbines are deployed in, they are exposed to extreme conditions and loadings. As a consequence of the harsh environment, the loadings on the turbine blades are much greater than that on wind turbine blades and need to be accurately evaluated within their design stage. Numerical modelling offers a relatively inexpensive method of estimating these loadings, compared to physical testing. However, in recent years, computational fluid mechanics has gained in popularity due to its ability to compute non-linear solutions for complex geometries accurately. In this paper, a computational fluid mechanics model for a concept tidal turbine is developed in order to estimate the fatigue loading on the turbine blades during operation. A steady state solution for a single blade is first analysed to estimate static loading on a turbine blade. This is then advanced to a transient model of the full turbine, which is used to estimate the fatigue loading for three flow velocity conditions. A combination of constant flow, tidal flow and the velocity profile due to the presence of surface waves are analysed and discussed within the paper. Finally, a brief discussion on the implications of the study is presented, along with the next stages of the study.**

Keywords—**Blades, Fatigue loading, Tidal energy**

I. Introduction

As the global tidal stream energy moves closer to commercial viability, additional challenges are presented as developers strive to lower the levelised cost of energy in order to challenge the low cost associated with generating energy from fossil fuels. As a result, a greater insight into the environment in which tidal turbine blades are deployed in is sought in order to advance the design of key components to achieve these economic and sustainability targets. One of the most significant operational challenges is the survivability of tidal energy devices when exposed to fatigue loading over their design life. Therefore, it is essential that developers and engineers clearly and accurately understand these fatigue loadings and can allow for them in the design stage of the development of a tidal energy device. A key component of many tidal energy devices is the blades, which converts the kinetic energy of the current into useful mechanical energy. As a result, the blades experience very high thrust forces and need to be designed to deal with these loadings. In recent years, the horizontal axis tidal turbine has become the most popular type of tidal energy device as, based on an EU report by Corsatea and Magagna [1], 76 % of research and development efforts in the tidal energy sector are related to HATT technologies.

Within the design stage of a device, numerical models are used to account for the operational loads on device components. Traditionally, the hydrodynamic modelling tool used for estimating the loadings on the blades of HATTs is the blade element momentum theory [2]. However, in recent years, computational fluid dynamics (CFD) is gaining ground due to increased computational capabilities and higher accuracy under a range of operating conditions. A summary of previously published studies exploring the use of CFD to examine the operation of horizontal axis tidal turbines has been included and discussed in Finnegan et al. [3]. Ahmed et al. [4] explored the fluctuating load on tidal turbine due to velocity shear and turbulence using CFD, which was validated against field data. Additionally, a number of studies have combined these fluid dynamics models with structural analysis algorithms to explore the structural loads and responses of the device components. Grogan et al. [5] developed a hydrodynamic-structural design methodology for a tidal turbine blades in order to compare glass fibre and carbon fibre as blade construction materials for a full scale (1.5MW) tidal turbine. Fagan [6] used advanced hydrodynamic and finite element method (FEM) models, along with a genetic algorithm, to optimise

Paper ID number: 1653. Conference track: Tidal hydrodynamic modelling (THM). This study was funded in part by the Sustainable Energy Authority of Ireland (SEAI) through the Ocean ERA-NET Ocean Energy Prototype Research and Development Programme 2018 as part of the SEABLADE project (Award no.: 18/OCN/102) and Science Foundation Ireland (SFI) through the Marine and Renewable Energy Ireland (MaREI) research centre (Grant no. 12/RC/2302).

W. Finnegan is a Research Fellow in the School of Engineering, the SFI MaREI Research Centre and the Ryan Institute at the National University of Ireland Galway, Galway, Ireland H91 HX31 (e-mail: william.finnegan@nuigalway.ie).

J. Goggins is a Senior Lecturer in the School of Engineering and a Co-PI in the SFI MaREI Research Centre and the Ryan Institute at the National University of Ireland Galway, Galway, Ireland H91 HX31 (e-mail: jamie.goggins@nuigalway.ie).

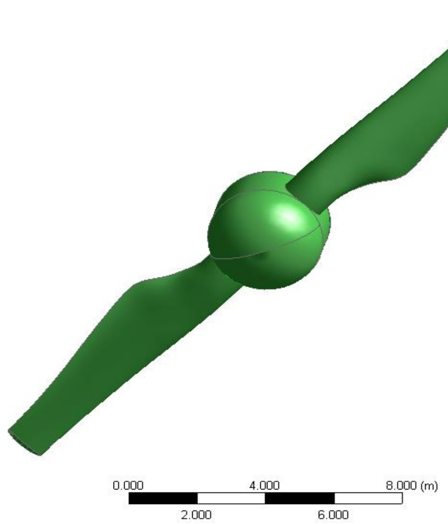


Fig. 1. Schematic of the geometry of the concept 1 MW horizontal axis tidal turbine used in this study.

the mass of the blade and evaluate a number of near-optimum tidal turbine blade designs.

The tidal turbine blades convert the hydrodynamic energy of the tidal current to mechanical energy. Therefore, the fatigue effects from the tidal current on the blades is detrimental to the design life of a tidal turbine, if not properly accounted for. With a number of tidal turbines being recently shown to be reaching commercial viability, there has been an increased understanding of the various fatigue loading on tidal turbine blades. There are five main aspects that have been identified by tidal turbine developers that contribute to fatigue loadings on tidal turbine blades:

- Variation in vertical velocity profile of the tidal flow;
- Shadow effects from the support structure;
- Array effects due to positioning in the wake of other turbines;
- Forces generated from surface waves; and
- Turbulence in incoming tidal flow.

Numerical modelling offers a relatively inexpensive method of testing and validation of early designs, compared to physical models.

In this paper, the operational fatigue loading on the blades of a concept tidal turbine device due to waves and tidal current are explored. The concept tidal turbine device used in this analysis is a 1 MW horizontal axis tidal turbine with two blades, as shown in Fig. 1. In order to achieve this, a CFD model of the blades and hub has been developed using the commercial software ANSYS CFX. The development of the model has been discussed, along with a discussion on the loadings due to water impacting on the blades due to the forces of the wave and tidal current.

II. Methodology

A. *Aim and objectives*

The overarching aim of this study is to develop a numerical model that can accurately predict the operational fatigue loading on tidal turbine blades. This model will allow designers, engineers and developers to allow for these fatigue loads from the early design stage and ensure that components of tidal energy devices are sufficiently designed to perform effectively throughout their full design life span. However, in order to achieve this aim, the following objectives were necessary:

- To develop a CFD model of flow on tidal turbine blades
- To accurately model the vertical profile of the flow velocity that accounts for both waves and tidal current
- To determine the resulting fatigue loadings on the blades due to the flow

B. *Tidal turbine geometry*

The 2-blade tidal turbine geometry for a concept horizontal axis tidal turbine that is used in this analysis is shown in Fig. 1. The diameter of the tidal turbine is approximately 21 m, where the hub is 3 m in diameter and, in turn, is suitable for use in tidal turbine with a capacity of approximately 1MW. The blades are pitched so that the tip of the blade is near 0° so to design for a worst-case scenario as the highest thrust loadings will occur in this position.

C. *Computational fluid dynamics model*

The tidal turbine geometry, which is discussed in Section B and shown in Fig. 1, is inputted into the CFD model, which is developed using the commercial software ANSYS CFX, and is based on the models developed by Finnegan et al. [3,7]. The turbine is modelled as a void within a rotating fluid domain that is located within a larger stationary fluid domain. Within the model, the flow and resulting forces on the turbine are only modelled, however the resulting pressures on the tidal turbine can be inputted into a structural mechanics software to further investigate the resulting structural impacts. The total length of the computational domain is 120m and the total diameter of the model is 80m, where the flow velocity is specified at an inflow boundary that is located 25m upstream of the tidal turbine, which can be seen in Fig. 4. The inflow flow velocity for the study is defined in Section E.

The meshed area for the solver calculation is the two fluid domains – a stationary domain and a rotating domain. A hex dominant method has been defined in the stationary domain with a maximum element size of 1.2m and the edges are divided in 160 divisions, to ensure a uniformly dispersed mesh. The maximum element size defined in the rotating domain is 0.3m. Mesh inflation has been specified at the surface of the tidal turbine, with a

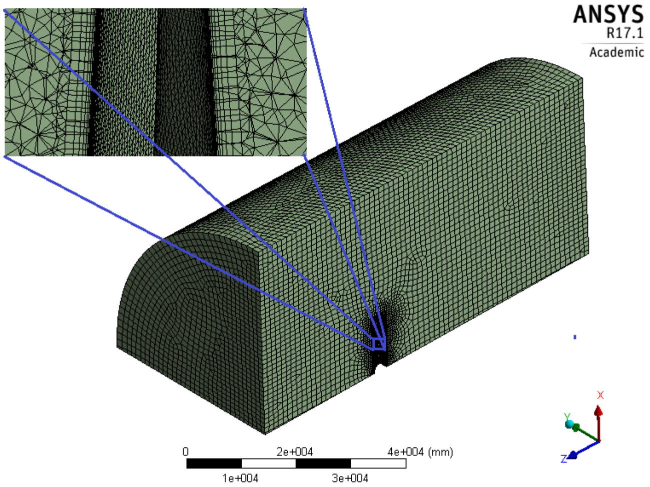


Fig. 2. A section through the domain mesh used in the analysis, including detail of the mesh refinement around the turbine blades.

maximum of 15 layers and a growth rate of 1.5, and a maximum element size on the surface of 0.05m. These refinements to the mesh result in an overall mesh size of approximately 2.8 million elements (or 1.3 million nodes).

The turbulence model used in this analysis is the shear stress transport (SST) turbulence model. Since a transient simulation is necessary, the "transient rotor-stator" method, which is also known as the “sliding mesh approach”, is specified within the rotating domain. In other words, the stationary flow is solved in a fixed frame of reference and the rotating flow field is solved in a rotating frame of reference, where the Coriolis force is included as part of the source terms. A general connection between the stationary and rotating fluid domains is specified. Additionally, the rotational speed of the region is specified along with a General Grid Interface (GGI) mesh connection.

D. *Computational fluid dynamics governing equations*

In this study, the CFD model is developed using the commercial software, ANSYS CFX, where its solver is based on the finite volume technique [8]. This technique divides the region of interest into sub-regions and discretises the governing equations in order to solve them iteratively over each sub-regions. Therefore, an approximation of the value of each variable at points throughout the domain is achieved.

The governing equations that need to be solved by the ANSYS CFX solver are the mass continuity equation for compressible flow, which is given as:

$$\frac{\partial \rho}{\partial t}+\frac{\partial(\rho u_1)}{\partial x}+\frac{\partial(\rho u_2)}{\partial y}+\frac{\partial(\rho u_3)}{\partial z}=0 \tag{1}$$

and the 3-dimensional Navier-Stokes equations, which are given as:

$$\begin{aligned}\rho\left(\frac{\partial u_1}{\partial t}+u_1\frac{\partial u_1}{\partial x}+u_2\frac{\partial u_1}{\partial y}+u_3\frac{\partial u_1}{\partial z}\right) &= -\frac{\partial p}{\partial x}+2\mu\frac{\partial^2 u_1}{\partial x^2}\\ &+\frac{\partial}{\partial y}\left(\mu\left(\frac{\partial u_1}{\partial y}+\frac{\partial u_2}{\partial x}\right)\right)\\ &+\frac{\partial}{\partial z}\left(\mu\left(\frac{\partial u_1}{\partial z}+\frac{\partial u_3}{\partial x}\right)\right)+F_1\end{aligned} \tag{2}$$

$$\begin{aligned}\rho\left(\frac{\partial u_2}{\partial t}+u_1\frac{\partial u_2}{\partial x}+u_2\frac{\partial u_2}{\partial y}+u_3\frac{\partial u_2}{\partial z}\right) &= -\frac{\partial p}{\partial y}+2\mu\frac{\partial^2 u_2}{\partial y^2}\\ &+\frac{\partial}{\partial x}\left(\mu\left(\frac{\partial u_1}{\partial y}+\frac{\partial u_2}{\partial x}\right)\right)\\ &+\frac{\partial}{\partial z}\left(\mu\left(\frac{\partial u_2}{\partial z}+\frac{\partial u_3}{\partial y}\right)\right)+F_2\\ &-\rho g\end{aligned} \tag{3}$$

$$\begin{aligned}\rho\left(\frac{\partial u_3}{\partial t}+u_1\frac{\partial u_3}{\partial x}+u_2\frac{\partial u_3}{\partial y}+u_3\frac{\partial u_3}{\partial z}\right) &= -\frac{\partial p}{\partial z}+2\mu\frac{\partial^2 u_3}{\partial z^2}\\ &+\frac{\partial}{\partial x}\left(\mu\left(\frac{\partial u_1}{\partial z}+\frac{\partial u_3}{\partial x}\right)\right)\\ &+\frac{\partial}{\partial y}\left(\mu\left(\frac{\partial u_3}{\partial y}+\frac{\partial u_2}{\partial z}\right)\right)+F_3\end{aligned} \tag{4}$$

where t is time, ρ is the fluid density, x, y, z are Cartesian coordinates (as shown in Fig. 1), u_1 is the flow velocity in the x-direction, u_2 is the flow velocity in the y-direction, u_3 is the flow velocity in the z-direction, F_1 is the body force on the fluid in the x-direction, $F_2-\rho g$ is the body force on the fluid in the y-direction (vertical), F_3 is the body force on the fluid in the z-direction, p is pressure and μ is viscosity.

E. *Flow velocity definition*

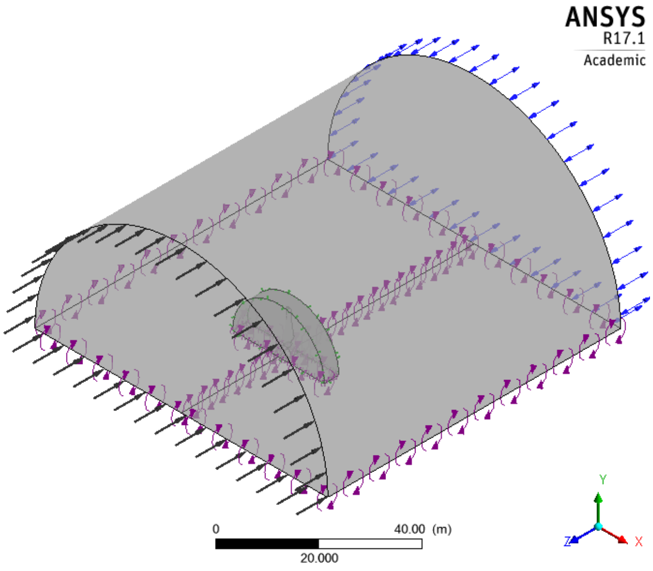
The flow velocity profile defined in this study is a combination of the wave velocity profile and the tidal current velocity profile. In this section, the two velocity profiles are discussed individually and the two are summed to form the flow velocity profile. Within the study, a number of combinations are used to describe the flow in order to investigate the influence of each component on the fatigue loading on the blades.

The velocity profile of the tidal current (U_x) reduces near the seabed due to friction. Therefore, the velocity profile can be described using the following power law [9]:

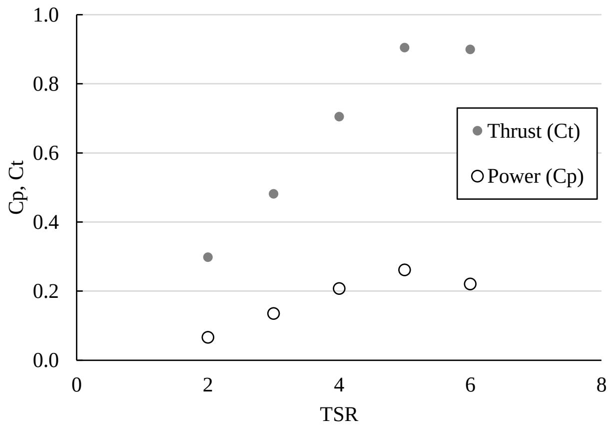
$$U_x=\left(\frac{y}{\beta d}\right)^{1/\alpha}\mathrm{U} \tag{5}$$

where, y is the height above the seabed (m), α is the power law, β is the bed-roughness coefficient, d is the water depth (m) and U is the depth averaged velocity (m/s), which is 3 m/s for the purpose of this study. Based on the findings of Lewis et al. [10], on average the 1/7th power-law (α = 7) with a bed-roughness coefficient (β) of 0.4 was found to accurately represent the velocity profile.

The velocity profile due to the presence of surface waves (U_{wave}) is defined using linear (Airy) wave theory, which originates from Navier Stokes and Euler equations. In this



(a)



(b)

Fig. 3. Details of a CFD model of a single tidal turbine blade showing (a) a schematic of the model domain and (b) the results of a steady state analysis for varying TSRs.

study, only the horizontal velocity component of the ocean wave is accounted for as this will have the most significant effect on the turbine. Therefore, the following equation is used to define U_{wave}:

$$U_{wave} = A\omega \frac{\cosh(k_0 y)}{\sinh(k_0 d)} cos(-\omega t - \varepsilon) \tag{6}$$

where, A is the amplitude of the wave (m), ω is the angular velocity of the wave (s^{-1}) and k_0 is the wavenumber (m^{-1}).

Therefore, the flow velocity, which includes the effects of shear (or velocity profile) in the tidal current (Equation 5) and the presence of surface waves (Equation 6), (U_{in}) is defined as:

$$U_{in} = \left(\frac{y}{\beta d}\right)^{1/\alpha} U + A\omega \frac{cosh(k_0 y)}{sinh(k_0 d)} cos(-\omega t - \varepsilon) \tag{7}$$

III. Results and Discussions

F. Single blade model

Initially, a CFD model of a single blade is developed in order to investigate further the parameters required for the model. This reduced model requires less computational costs due to the reduced size of the model, and in turn the mesh, and the use of a steady state analysis rather than a transient analysis. In order to model the rotation of the tidal turbine in the rotating domain, a 'frozen rotor' method is used within the steady state analysis. In order to reduce the overall size of the computational domain, periodic symmetry boundary conditions have been specified in both domains, as seen in the schematic of the domain given in Fig. 3 (a). The meshing parameters for refining the mesh in both the stationary and rotating domain were derived using this model and are given in Section C.

Within this analysis, a constant velocity of 3m/s is used but the tip speed ratio (TSR) is varied in order to derive the power (C_P) and thrust (C_T) coefficients for the concept horizontal axis tidal turbine blade, which is described in Section B. These non-dimensionalised coefficients (TSR, C_P and C_T) are defined using the following equations [2]:

$$TSR = \frac{\Omega R}{U_{in}} \tag{8}$$

$$C_P = \frac{Q\Omega}{\frac{1}{2}\rho U_{in}^3 A_T} \tag{9}$$

$$C_T = \frac{T}{\frac{1}{2}\rho U_{in}^2 A_T} \tag{10}$$

where Ω is the rotational speed of the turbine (rad/s), R is the radius of the blade (m), Q is the rotor torque (Nm), T is the thrust force (N) and A_T is the area of the turbine.

The results of this steady state analysis are shown graphically in Fig. 3 (b). The thrust coefficient for the turbine blade increases linearly and then levels off as it approaches the theoretical limit, based on Equation 10. The power coefficient for the turbine blade also increases until it peaks and then decreases as the TSR increases past this maximum value. Based on the results from Fig. 3 (b), the peak power production occurs at TSR of 5, which is equivalent to the turbine rotating at 13.39rpm when there is a constant flow of 3m/s. These results represent the static load imposed on the turbine blade and hub as the result of constant flow and may be used as the force applied when performing mechanical static testing of the tidal turbine blade in the flap-wise direction.

G. Fatigue loading

The fatigue loading on the tidal turbine blades is calculated using the transient CFD model of the full turbine geometry (from Fig. 1). The complete domain, including boundaries, is shown graphically in Fig. 4.

A transient analysis of the flow velocity, which is defined in Section E, has been performed in order to investigate the thrust force on the turbine. The thrust force has been selected as it is the most significant loading on the turbine blades when in operation and, therefore, the fatigue loading imparted due to this force will have the most damaging effect on the turbine blades.

Within this analysis, a case study for the turbine operating at 13.39rpm, where the average tidal flow

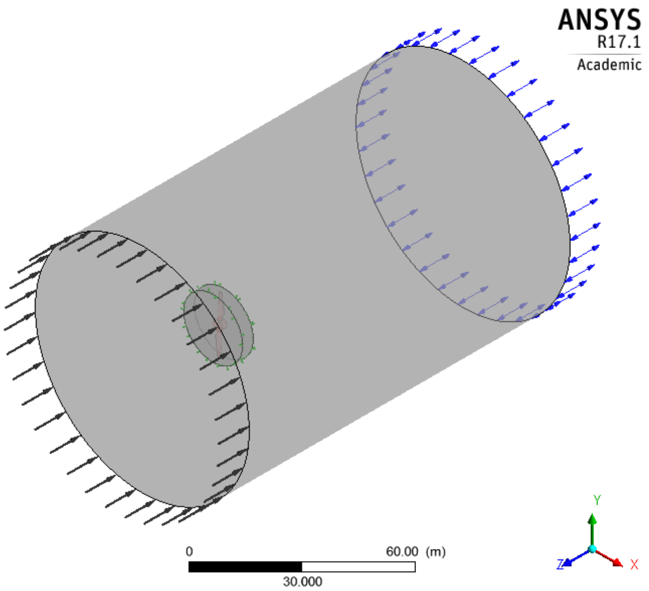


Fig. 4. Graphical representation of the CFD model domain used for the transient analysis in this study.

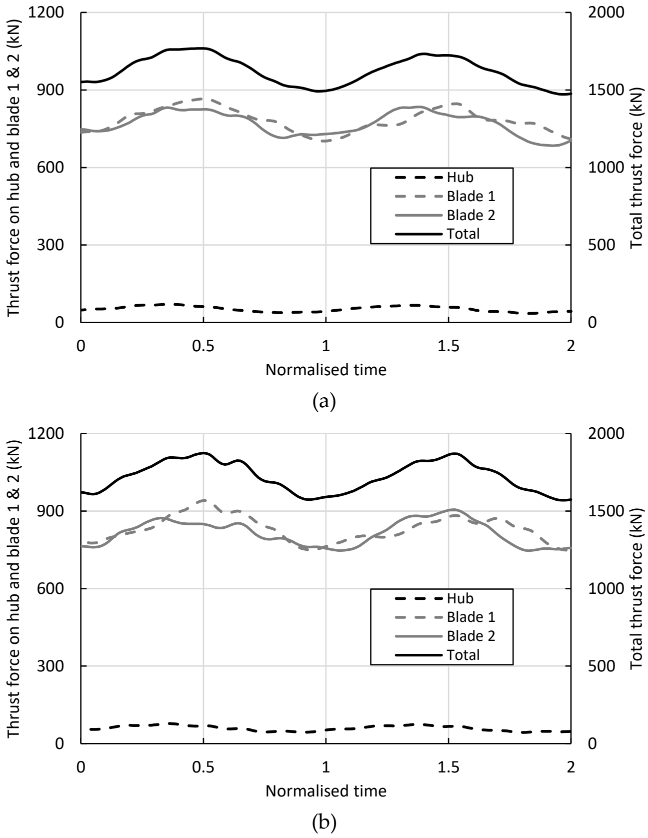


Fig. 5. Results from the CFD analysis of the concept tidal turbine, showing the thrust force on the turbine for (a) a constant flow of 3 m/s in the presence of surface waves and (b) a tidal flow in the presence of surface waves (using Equation (7)).

velocity is 3 m/s, has been assessed. This operating speed for the turbine has been selected based on the results of the steady state analysis, which is discussed in Section F, where 13.39rpm is found to produce the peak power when the average tidal flow velocity is 3 m/s. Additionally, a single linear wave has been selected in order to explore the fatigue loading on the tidal turbine in the presence of surface waves. This wave has a significant wave height of 3m and a wave period of 10s.

In order to explore the effect of the velocity profile on the concept tidal turbine, four velocity profile conditions have been specified:

- A constant flow
- A constant flow in the presence of surface waves
- A tidal flow
- A tidal flow, derived using the power law, in the presence of surface waves

As the thrust force is the most significant force on the tidal turbine blades, this was examined for the concept tidal turbine, where the total thrust force on the turbine and the thrust force on the hub and individual blades were extracted from the results.

Fig. 5 presents the thrust force for the three scenarios detailed in the previous paragraph. The horizontal axis of the graph has been normalised with respect to the period of the waves present. It is evident from both Fig. 5 (a) and (b) that the total thrust force on the turbine varies significantly over the period of the surface wave, which is approximately 16%. This variance between maximum and minimum thrust force is consistent when each blade, where it is approximately 14-17%. However, when the blades are examined individually, it can be seen that further variances per turbine rotation is evident. Although this is smaller in comparison to the overall thrust, it will be significant over the design life of the turbine blade. Additionally, the thrust force on the hub is displayed in Fig. 5 but it is relatively small compared to the other components being analysed and this is as a result of its relatively small surface area.

The thrust force on a single blade has been compared for each of the four velocity profile conditions described in this study, which is shown in Fig. 6. It is evident that the force due to the combination of tidal flow in the presence of surface waves is greater than the constant flow scenarios, when estimating the fatigue loading imparted on the blades. The variance in loading per turbine rotation, which will cause fatigue damage in operation, is even more evident in Fig. 6 and this should be further investigated in order to accurately recreate it in a controlled laboratory setting. Additionally, it is evident from this study, when fatigue testing tidal turbine blades in a laboratory, for certification or design life prediction tests, it is satisfactory to carry out tension-tension fatigue loading.

When the unsteady loadings discussed in this paper are coupled with the other sources of fatigue on turbine blades – presence of a supporting structure and turbulence within the flow – the overall loading on the turbine and, in particular, the tidal turbine blades becomes relatively significant, where it may surpass 30% of the overall thrust force. Therefore, it is essential to accurately predict these unsteady loadings to account for fatigue and should be a central component when certifying tidal turbines, which is currently defined in DNVGL-ST-0164 [11].

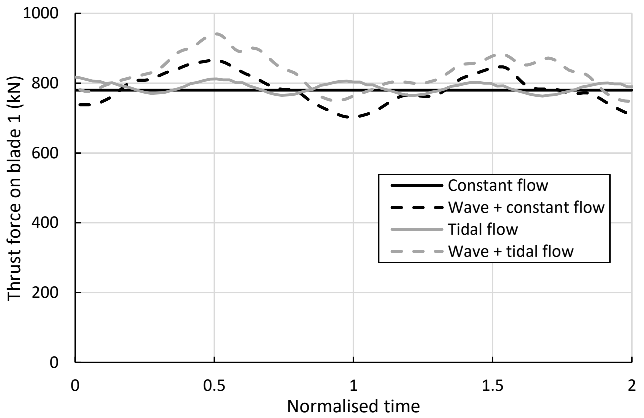


Fig. 6. Results from the CFD analysis showing the thrust force on a single blade for each of the three velocity profile conditions described in this study.

IV. Future Work

Now that tidal energy is reaching commercial viability, there is a larger data set for operational trials available and, therefore, a greater understanding into the operational loadings that are experienced by the turbines. Therefore, the next stages in this study will be to incorporate other sources of fatigue damage into the models. These include shadow effects from the presence of a supporting structure, turbulence in the flow and any other aspects identified by developers.

In the short term, this current model will be advanced by exploring a range of linear wave inputs and expanding the model to include irregular wave, which represent ocean waves. Additional data on the wave and current climate at proposed sites may be incorporated into the model, which would allow for the model to be used to assess the suitability of a specific turbine at a range of tidal energy sites.

V. Conclusion

In this paper, the operational fatigue loading on the blades of a concept tidal turbine device due to waves and tidal current was explored using a concept tidal turbine device. A CFD model of the fluid flow on the tidal turbine was developed using the commercial software ANSYS CFX. The fatigue loading on the turbine was explored for three flow velocity conditions and discussed in Section 3. A significant fatigue loading due to a variance between maximum and minimum thrust force of up to 17% was observed and its implications in design, testing and operation were discussed.

Fatigue testing is well established in the wind energy sector, which is detailed in the structural testing standards DNV-DS-J102 [12] and IEC61400-23 [13]. For the tidal energy sector, a discussion of the requirements for fatigue testing blades and components is presented in the tidal turbine design standard, DNVGL-ST-0164 [11]. The research presented in this study can help to better define the potential impacts of fatigue loading on tidal turbine components and potentially feed into the fatigue testing and design of components, as companies move into the final TRLs and begin full commercialisation.

The key to success for a new energy source, such as tidal energy, is to ensure the levelised cost of tidal energy is kept to a minimum and, preferably, lower than alternative energy sources, which will aid its commercial competitiveness, while making it more attractive for investors. Advanced and comprehensive fatigue testing will de-risk the technology as it will ensure that tidal turbine blades to withstand fatigue effects over their functional lifespan.

Acknowledgement

This study was funded in part by the Sustainable Energy Authority of Ireland (SEAI) through the Ocean ERA-NET Ocean Energy Prototype Research and Development Programme 2018 as part of the SEABLADE project (Award no.: 18/OCN/102) and Science Foundation Ireland (SFI) through the Marine and Renewable Energy Ireland (MaREI) research centre (Grant no. 12/RC/2302).

The first author would like to acknowledge the financial support from the Executive Agency for Small and Medium sized Enterprises (EASME) as part of the LEAPWind project (Agreement no.: EASME/EMFF/2017/1.2.1.12/S1/06/SI2.789307). The last author would also like to acknowledge the support of Science Foundation Ireland through the Career Development Award programme (Grant No. 13/CDA/2200).

References

[1] T.D. Corsatea, D. Magagna, Overview of European innovation activities in marine energy technology, European Commission, Brussels, Belgium, 2013

[2] A.H. Day, A. Babarit, A. Fontaine, Y.P. He, M. Kraskowski, M. Murai, I. Penesis, F. Salvatore, H.K. Shin, Hydrodynamic modelling of marine renewable energy devices: A state of the art review, Ocean Engineering 108 (2015) 46-69

[3] W. Finnegan, E. Fagan, T. Flanagan, A. Doyle, J. Goggins, Operational fatigue loadings on tidal turbine blades using computational fluid dynamics, Renewable Energy (2018).

[4] U. Ahmed, D.D. Apsley, I. Afgan, T. Stallard, P.K. Stansby, Fluctuating loads on a tidal turbine due to velocity shear and turbulence: Comparison of CFD with field data, Renewable Energy 112 (2017) 235-246

[5] D.M. Grogan, S.B. Leen, C.R. Kennedy, C.M. Ó Brádaigh, Design of composite tidal turbine blades, Renewable Energy 57 (2013) 151-162

[6] E. Fagan, Design of fibre-reinforced polymer composite blades for wind and tidal turbines, Civil Engineering, National University of Ireland Galway, Galway, Ireland, 2017

[7] W. Finnegan, J. Goggins, Linear irregular wave generation in a numerical wave tank, Applied Ocean Research 52 (2015) 188-200

[8] ANSYS, ANSYS® CFX (Release 17.1): CFX-Solver Theory Guide, , ANSYS Inc., 2017

[9] R. Soulsby, 1997. Dynamics of marine sands: a manual for practical applications. Thomas Telford

[10] M. Lewis, S.P. Neill, P. Robins, M.R. Hashemi, S. Ward, Characteristics of the velocity profile at tidal-stream energy sites, Renewable Energy 114 (2017) 258-272

[11] DNV GL, Tidal turbines, DNV Standard, DNVGL-ST-0164, Oslo, Norway, 2015
[12] DNV GL, Design and manufacture of wind turbine blades, offshore and onshore wind turbines, DNV Standard, DNV-DS-J102, 2010, pp. 2010-11
[13] International Electrotechnical Commission (IEC), Part 23: Full-scale structural testing of rotor blades, IEC 61400-23:2014 Wind turbines 2014