An investigation into Reynolds scaling and solidity for a HATT tidal turbine

Matthew Allmark, Stephanie Ordonez-Sanchez, Sujin Wang, Yo Seop Kang, Chul- Hee Jo, Tim O'Doherty, Cameron Johnstone

Abstract—**Scale model testing has formed a vital part of modelling research activities, allowing modelling researchers to validate code and model set ups against experimental data. Generally, scale testing to-date has proceeded at the 1/30th to 1/20th scale which is in line with the size of facilities available for testing such devices. This paper presents a fundamental study into the effects of Reynolds Number and Rotor Solidity when testing HATTs at 1/5th scale, 1/20th scale and 1/30th scale. The paper utilises a mixture of 1/30th scale (0.5 m diameter) experimental data for two, three and four rotor setups, 1/20th scale (0.9m diameters) experimental data for a three-blade rotor setup.**

Keywords— **Reynolds scaling, solidity, horizontal axis turbines, experiments, flume tank.**

I. Introduction

FOR the last 15-20 years, development and testing of scale model turbines have been utilised in both academic research and by commercial developers. The use of scale model testing has allowed developers to further understand design decisions during early technology readiness levels (TRLs) with relatively small investments needed. In terms of research, the use of scale model turbines has proliferated and allowed researchers to understand the fundamental fluid dynamics, loading mechanisms and efficiencies associated with a variety of turbine rotor set-ups.

Challenges arising from up-scaling devices are not isolated to the marine energy industry. Relating the physical interactions of scaled up turbines with the flow unsteadiness of large scale devices has become almost impossible in wind energy in the latest years. To overcome this constraint, [1] attempted to reach appropriate scaling Reynolds numbers in the order of $5.3x10^6$ to $9.7x10^6$ by using a high pressure wind tunnel. In a similar context, the New European Wind Atlas is a project funded by the European Commission which seeks to develop downscaling methodologies for the wind turbine industry [2]. One of their work packages is dedicated to providing a detailed and accurate description of a wide range of geographical and meteorological conditions. These data will be publicly available to mitigate the problem arising with numerical model validation. It is worth noting that attaining a project of such magnitude within the marine energy sector would be extremely ambitious given the current status of it.

In the marine energy sector, scale model testing has formed a vital part of modelling research activities, allowing researchers to validate analytical and numerical model set-ups against experimental data. Generally, scale testing to-date has proceeded at the 1/30th to 1/20th scale which is in line with the size of facilities available for testing such devices. Some of the most used facilities for tidal energy testing in Europe are those partly described in [3]. This research relates to devices with turbine diameters in the order of 0.5 m to 1.5 m, at least when referring to horizontal axis tidal turbines (HATTs); e.g. [4] and [5].

The establishment of nursery sites, however, has allowed for the development and testing of 1/5th scale devices, which is often a crucial step in moving towards a higher TRL full-scale deployments. However, information related to these devices usually remains confidential and, thus, investigations related to scalability effects are only being partially explored. The lack of information does not only limit the knowledge related to empirical work but also the use of numerical or analytical solutions applied to full scale devices.

Relating the outcomes from numerical models or experiments at small scale to full-scale applications is of vital importance in the marine energy industry. To the authors' knowledge, there is only one publication detailing experimental campaigns aimed at addressing this issue [6]. In Reference [6], a comprehensive evaluation of numerical and experimental studies to demonstrate the effects of scalability in marine renewable energy applications using two scaled turbine rotors was undertaken. It was found that the larger turbine developed

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slightly higher power coefficients than the reduced device. These results were associated to Reynolds dependency and blockage. Although their study demonstrated the importance of understanding scaling effects on HATTs, the investigation was focused on a two-bladed rotor design with the blade profile section based on a NACA aerofoil family. Therefore, the study presented by Reference [6] should be broadened to inform particular scaling effects from other types of devices and also blade designs.

This paper presents a fundamental study into the effects of Reynolds Number and rotor solidity when testing HATTs at a 1/20th scale and 1/30th scale. The paper utilises a mixture of 1/30th scale (0.5 m diameter) experimental data for two, three and four rotor set-ups and a 1/20th scale (0.9 m diameter) experimental data for a three blade rotor set-up.

II. METHODOLOGY

To study the effects of Reynolds scaling and solidity on HATT performance, a series of experiments were conducted with two different turbine prototypes: a small scale device of 0.43 m diameter and a 0.9 m turbine diameter turbine. Both devices are described in the following sections along with the testing campaigns carried out in two facilities.

Fig. 1. Four bladed 1/30th horizontal axis turbine used in the testing campaign.

A. *Solidity investigations*

A scaled horizontal axis turbine of 0.43 m in diameter was utilised in this testing campaign. The rotor was composed of two, three and four blades. The rotor blades were designed using a Wortmann FX 63-37 profile and were a scaled down version of the rotor used in [5]. Figures 1 and 2 show a four bladed turbine and the test rig, respectively.

The tests were carried out at a flume tank located at the Inha University in South Korea. The cross-sectional area of the flume is 1.0 m x 1.0 m and the working section is of 3.0 m. The turbine hub centre was installed at mid water, approximately 0.4 m from the free surface. The blockage ratio was calculated as 20%.

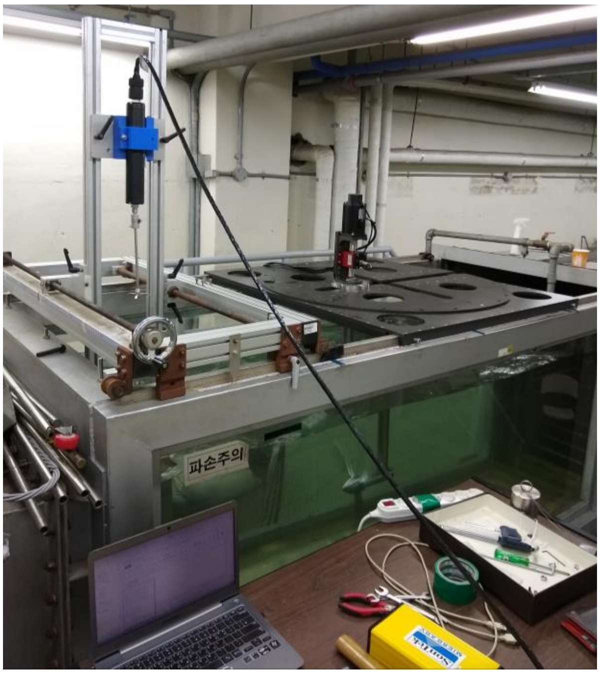

Fig. 2. Flume tank at Inha University.

The turbine assembly consisted of a rotor hub connected to the transmission drive shaft via a keyway mechanism. A bevel gear coupled the main and the secondary shaft which ran inside the mast up to the upper surface. The secondary shaft was attached to an encoder and a motor in the dry, as seen in Figure 2. A torque transducer and the encoder were used to quantify the power generated by the turbine. The load of the rotor was regulated using a torque control mechanism.

The solidity of a turbine can be quantified using Equation 1, where B relates to the blade number, c is the blade chord, considered in these calculations at 75% of the blade length (m) and R is turbine radius (m), in this case 0.22 m. Therefore, the solidity of the turbine can be modified by changing the rotor radius, the chord or the number of blades. For this investigation, only the number of blades were modified adjusting the rotor hub to two, three and four blades as it can be observed in Figure 1. The solidity numbers for each of these settings can be seen in Table 1.

$$\sigma = \frac{Bc}{\pi R^2} \tag{1}$$

The optimum pitch for each configuration was obtained using BEMT theory. Four flow velocities were considered in the experiment to give an indication of Reynolds independence. For most of the tests, 10 rotational speeds were captured. At least one full repetition of the entire dataset was taken. A sampling rate of 100 Hz was employed to record the torque measurements. Each test lasted approximately 120 s. Table 1 depicts the solidity number and the Reynolds numbers quantified for each of the rotor settings and the flow velocities. Here Reynolds number is generated using the blade chord length at 75% of the blade span.

TABLE I

REYNOLDS NUMBER AND SOLIDITY VALUES CALCULATED FOR EACH OF THE TEST CASES

Blades /Flow Velocity	0.8 m/s	0.9 m/s	1.0 m/s	1.1 m/s	Solidity
2 (1/30th scale)	1.15E+05	1.29E+05	1.43E+05	1.56E+05	0.05
3 (1/30th scale)	1.10E+05	1.29E+05	1.43E+05	1.56E+05	0.08
4 (1/30th scale)	1.14E+05	1.29E+05	1.42E+05	1.56E+05	0.10
3 (1/20th scale)			2.16E+05		0.08

A flow characterisation was conducted at the end of the testing campaign. The flow velocity was quantified for each test condition at a number of points across the rotor plane section. The flow measurements were acquired with an Acoustic Doppler Velocimeter, as observed in Figure 2. The data capture was set to 50 Hz, and each point was recorded for three minutes. Figure 3 shows the positions used to capture the flow regime of the flume.

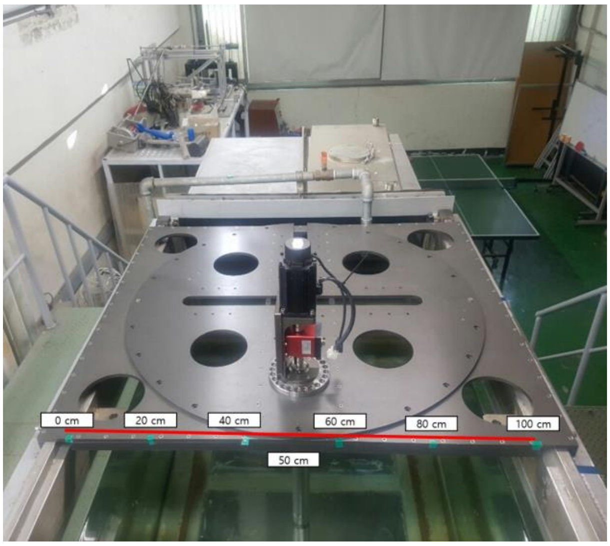

Fig. 3. The green marks at the top panel show the positions where the ADV measurements were taken across the flume. These measurements were obtained at hub height and 0.25 m above hub height (close to the blade tip location).

B. Scaliability

For the second part of this analysis, the data obtained for the three bladed 1/30th scaled turbine is compared with a 0.9 m rotor diameter turbine (1/20th scale prototype). This turbine was developed as part of the Supergen Grand Challenges project Dylotta. The three bladed horizontal axis turbine has already been tested in two test campaigns, in the tow tank located at CNR-INM (former INSEAN) in Rome, Italy and the circulating flume tank located in the installations of Ifremer in Boulogne Sur Mer, France. More information on the equipment and testing campaigns can be found in [7] and [8] and a figure of the scaled up turbine can be seen in Figure 4.

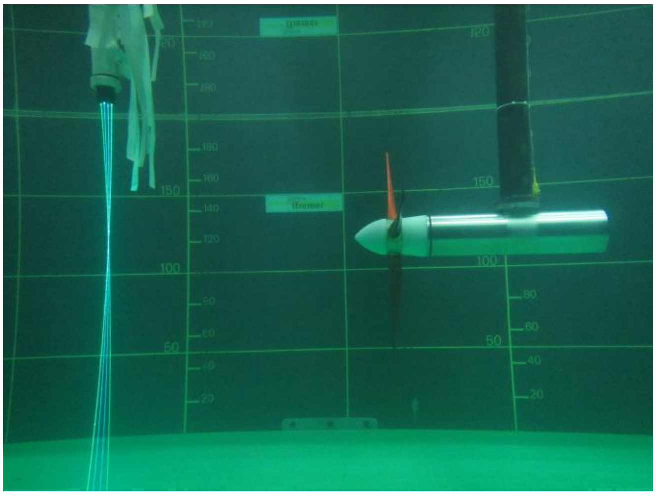

Fig. 4. Flow velocities across the flume cross sectional area at two water depths.

C. Non dimensional parameters

Non dimensional performance parameters for all the data sources are compared. The characteristic curves are considered for, 'Reynolds Independence'. Here Reynolds number is generated using the characteristics length of the blade chord at 75% of the blade length. Power and angular velocity results are normalised based on the flow characteristics using the following equations:

$$C_P = \frac{T\Omega}{2\rho U^3 A} \tag{2}$$

$$TSR = \frac{\Omega R}{U} \tag{3}$$

where T is the measured average torque given in Nm, and Ω is the angular velocity in rad/s resulting in the total hydrodynamic power output generated by the turbine in Watts. A is the swept area of the rotor in meters which is considered here as 0.16 m^2 for the small turbine and 0.64 m^2 for the large prototype. U denotes the flow velocity measured with an ADV and is given in m/s. The density of the water used to calculate C_P was 1000 kg/m^3. The average C_P is related to the tip speed ratio of the turbine (TSR), given by Equation 3.

Blockage corrections were not implemented due to insufficient information related to thrust values. Instead of a blockage correction, the comparative results are occasionally presented against the maximum power coefficient for each data set.

III. Results and Discussion

The results presented in the following sections include the outcomes from the flow characterisation, power measurements for the two, three and four bladed rotors with a comparison of the three-bladed scaled up prototype version.

D. Flow characterisation

Figure 5 presents the variations of the flow velocities across the flume and at two locations on the water column (hub height and 0.25 m above hub height, close to the blade tip location). It can be observed in Figure 5 that the flow developed on the flume is highly variable in both sides of the channel. The flow variability decreases marginally at flow velocities of 0.8 m/s. The flow velocities obtained at hub height seemed to have a larger magnitude than those closer to the free surface in the mid-region of the flume. However, average values were in the regions of 0.8 m/s, 0.9 m/s, 1.0 m/s and 1.1 m/s with mean standard deviation values of 0.04 m/s for all the flow velocities. There were no substantial differences on the standard deviation values at low (0.8 m/s) or high (1.1 m/s) flow velocities.

However, looking at Figure 6, it is visible that the turbulence intensity around that section of the channel is of about 5.7%. Also, the turbulence intensity is slightly higher when setting the flow at a speed of 1.1 m/s. This turbulence intensity is only 8.3, 5.7 and 1.1% higher than those values obtained for the flow settings of 0.8, 0.9 and 1.0 m/s, respectively. The turbulence intensity is significantly high compared to other flow regimes where the scaled-up version of the turbine has been tested. For example, previous tests conducted at the circulating flume tank at Ifremer, Boulogne Sur Mer, France indicated turbulence intensity levels closer to 1.9% [9]. For the tests undertaken at the tow tank CNR-INM (former Insean), the experiments were conducted in still water and thus, no turbulence effects were identified during the testing [5].

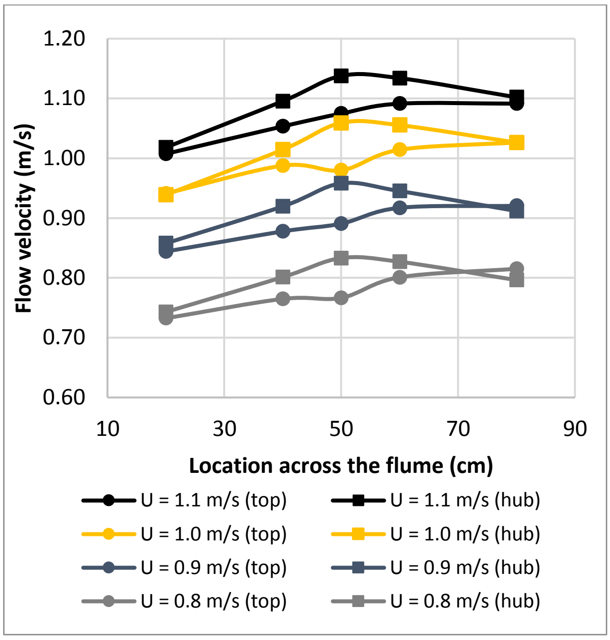

Fig. 5. Flow velocities across the flume cross sectional area at two water depths.

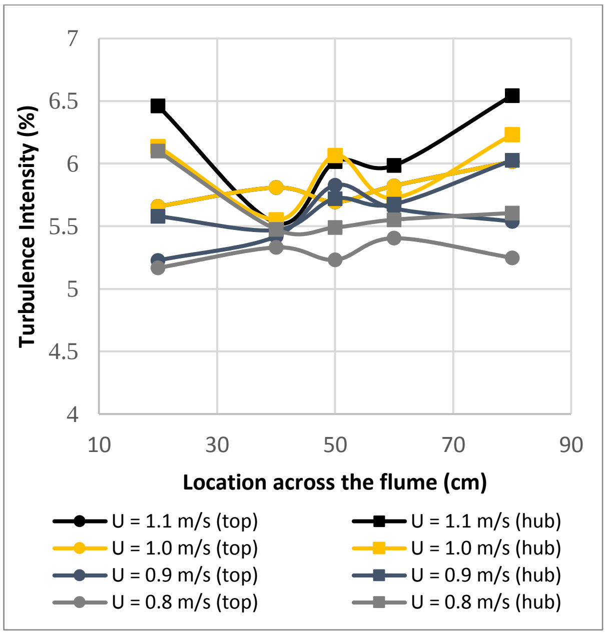

Fig. 6. Turbulence intensity across the flume cross sectional area at two water depths.

E. Solidity – blade number

Figures 7 to 10 show the power coefficients obtained for each of the configurations at four flow velocities. Good repeatability was achieved between the experiments. Power performance fluctuations between repeated tests were in the order of 2% of the mean values for most of the tests. The largest discrepancies were obtained for TSRs > 6 (closer to freewheeling), with exception of one repeated experiment of the "2 bladed, 1.0 m/s r3" test which shows scattered data for most of the operating range. It is likely that this data spread is related to a mechanical fault on the test rig; however, uncertainty on the experiments was reduced by repeating the tests at least twice, which is a rare practice within the field of marine renewables due to cost and time restrictions associated with the use of large facilities.

It is evident that as the solidity increases (4 bladed), a slight increase in the peak power performance is achieved at lower values of TSRs (in a region of 3.5 to nearly 5), compared to the other configurations. The C_P increased by 5% and 19% when compared to the peak power of the 3 bladed and 2 bladed configurations, respectively. An increase of performance between the 3 and 2 bladed configurations was in the order of 14.5 %, this comparison relates data only at peak power conditions which is in the order of 3.5 to 5 depending on the test case. In general, this increment of peak power was achieved from larger to smaller solidity rotor configurations; however, these discrepancies were harder to identify when looking at the 1.0 and 1.1 m/s flow speed cases. This disagreement can be explained by the high fluctuations encountered in the flow. When looking at the mean and fluctuating parts of the flow speeds at 1.0 and 1.1 m/s, an overlap of similar magnitudes is observed in both flow conditions ($1.0\,^{0.96}_{1.05}$ m/s and $1.08\,^{1.04}_{1.12}$ m/s.) This flow speed overlapping was not observed at any other instances.

A steep decrease of power with the increase of TSR in the four bladed configuration is also apparent in all the curves. This characteristic may be undesirable for the operation of turbines in highly fluctuating environments (Figs 7-10). Moreover, the costs associated with the utilisation of turbines with a large number of blades (> 3) may be adverse as devices will become less reliable. For example, it has already been observed that turbines may experience unwanted changes in optimum pitch settings due to the variability of the environment [10]; thus, developing high fluctuations of thrust and torque of adjacent blades which will affect their structural integrity [11]. This problem may exacerbate if the distance between blades is further reduced.

It must be noted that power coefficients presented in Figures 7-10 are on occasions closer to magnitudes of 0.6 which are unrealistic for horizontal axis turbines. These values are related to the ratio between the rotor diameter and the cross section area of the flume which is in the order of 20%. Thus, additional figures are included in this section which are normalised in terms of the maximum C_P, as shown in Figures 11-13.

In Figure 11 the performance curve related to the 1.0 and 1.1 m/s flow velocity cases reaches freewheeling faster than at lower flow speeds of 0.8 and 0.9 m/s. It was also observed that the power performance reached at 1.0 and 1.1 m/s was of a lower magnitude than that observed at lower flow rates. In Figures 12 and 13, it is not clear if Reynolds independence has been achieved for the 3 and 4 bladed configurations as the spread of the data is clustered locally without following a specific sequence. This data spread may be due to mechanical irregularities seen on the test rig; e.g. motor overheating. More information on Reynolds number and its effects on small scale devices is included in the following section.

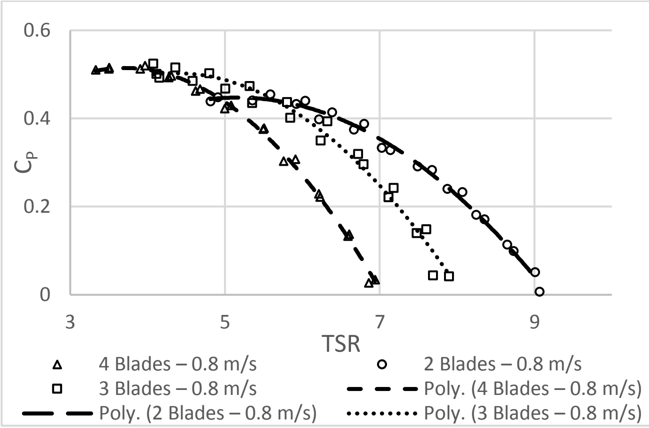

Fig. 7. Comparison of power coefficient curve at a flow velocity of 0.8 m/s.

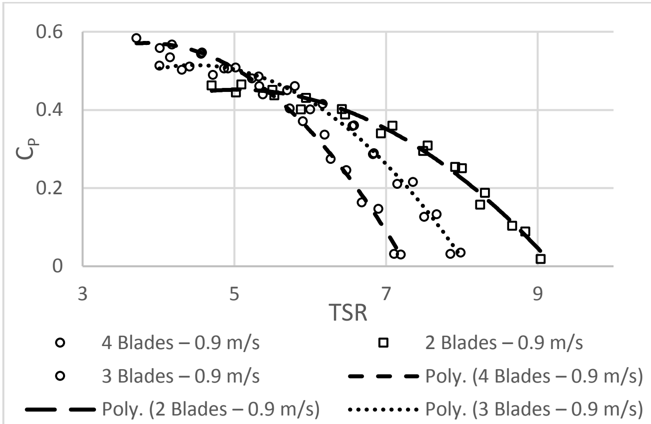

Fig. 8. Comparison of power coefficient curve at a flow velocity of 0.9 m/s.

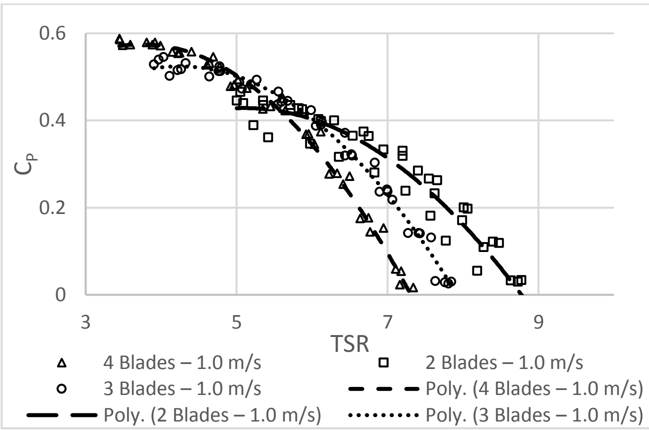

Fig. 9. Comparison of power coefficient curve at a flow velocity of 1.0 m/s.

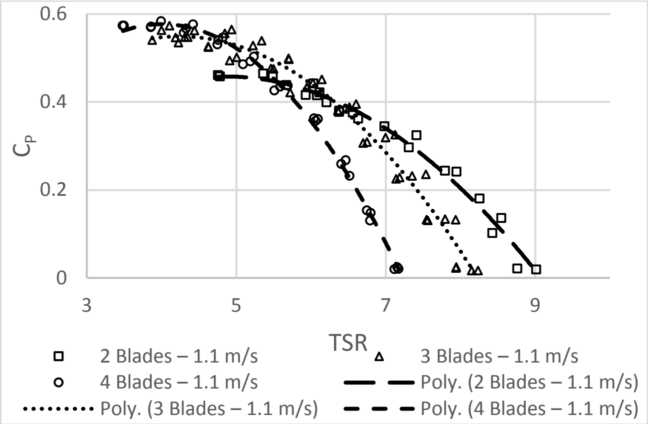

Fig. 10. Comparison of power coefficient curve at a flow velocity of 1.1 m/s.

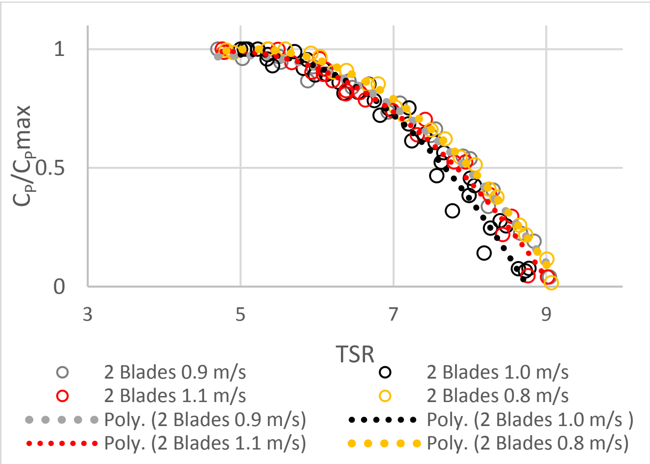

Fig. 11. Comparison of power coefficient curve of the 2 bladed rotor operating at four flow velocities.

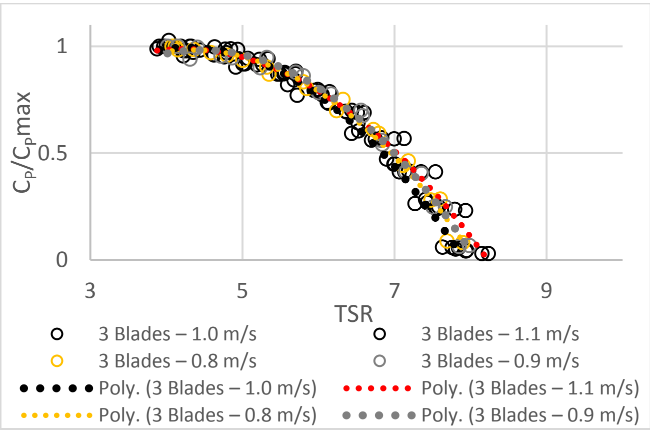

Fig. 12. Comparison of power coefficient curve of the 3 bladed rotor operating at four flow velocities.

To complement the information obtained during the testing campaign, a blade element momentum model was utilised to predict the power of the 2, 3 and 4 rotor configurations. To facilitate the comparison between the physical and the analytical outcomes, the data was normalised by the peak power coefficient. It can be seen in Figure 14 that the power coefficient curves of the 2 and 3 bladed configurations from both the experiment and the analytical model have a similar shape. It is interesting to see that the model does not predict quite well the behaviour of the 4 bladed configuration. One reason may be related to the inability of BEMT to predict the wake circulation in rotors defined by high solidity values which will affect power output predictions.

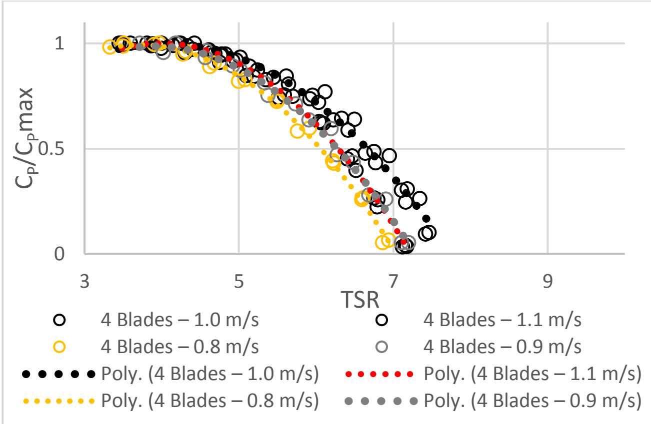

Fig. 13. Comparison of power coefficient curve of the 4 bladed rotor operating at four flow velocities.

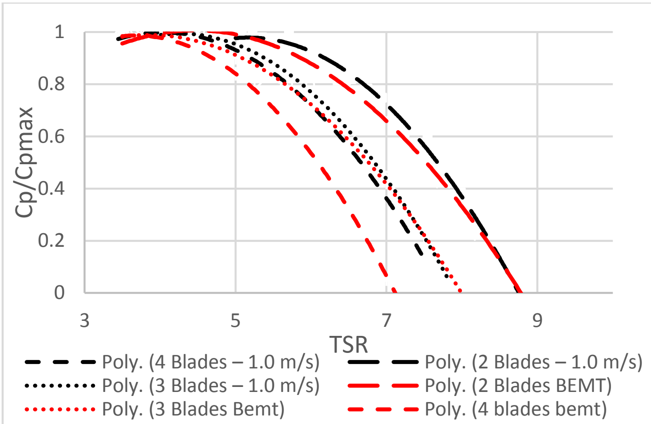

Fig. 14. Comparison of power curve of 2, 3 and 4 bladed turbine.

F. *Scaling investigations*

Figure 15 and 16 includes the data obtained from the physical testing undertaken for the 1/20th and 1/30th scaled prototypes and the prediction values from the analytical model. It was found that at a larger scale the analytical model predicts a peak power coefficient 4% higher than that calculated for the 1/30th scale. The reason for this is related to the aerodynamic coefficients used as an input on the blade element momentum model. It seems that freewheeling operation falls within the same range between the analytical and experimental data of the 1/30th scale prototype. The latter does not hold true for the 1/20th scale, where it seems that the model predicts a sharper decay than that shown by the physical data. When looking at the C_P coefficients of the 1/20th scale it was found that the power coefficient predicted by the model is 6% higher than that obtained from the experiments which include 2 testing campaigns at 2 different facilities.

The differences in power and range of operation may be related to Reynolds numbers achieved at each of the testing campaigns, as well as, blockage conditions. It has been observed by [6] that Reynolds independence for a 2 bladed rotor of 0.5 m in diameter was reached only at flow speeds higher than 1.26 m/s. Therefore, it may be possible that this study is within the Reynolds dependence region thus affecting the operational capabilities of the scaled devices. However, it should be noted that the blade shape used in [6] was substantially more slender than the blade geometries used in this research, i.e. the Wortmann blades are 30% thicker than those used in [6]. Therefore, larger Reynolds numbers were produced in this study compared to those achieved in [6].

Another factor that may influence the characteristic curves of the turbines may be related to the blockage ratio conditions which can influence the performance results of a device even when low blockage conditions exist [3]. Further experiments and validation of the results obtained in this investigation should be undertaken.

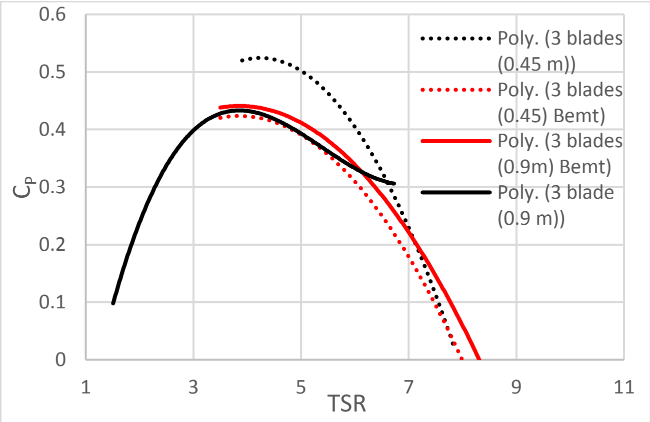

Fig.15. Comparison of the power curve of the 3 bladed turbine, for experimental and BEMT results for the 1/30th and 1/20th scales.

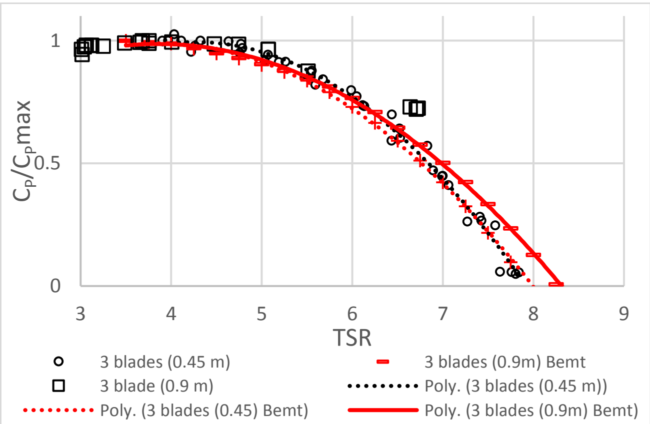

Fig. 16. Comparison of the power curve of the 3 bladed turbine, for experimental and BEMT results for the 1/30th and 1/20th using normalisation values again maximum C_P.

IV. Conclusion

The influence of Reynolds number and solidity of a turbine on the power performance of small scale horizontal axis turbine prototypes was studied in this paper. It was observed that the use of more than 3 blades on a horizontal axis turbine has little improvement on the power achieved by a device, while it may incur in additional capital costs at commercial deployments.

Power performance comparisons between a 1/20th and a 1/30th scaled turbine were undertaken using two turbine prototypes and an analytical model. Due to the facility constraints experienced during the testing of the 1/30th scale turbine, the performance comparisons were done in terms of maximum power coefficient. It was found that at a larger scale the analytical model predicts higher power coefficients. The data from the physical modelling implies that the operating range is reduced when working with smaller scales.

Future work will include CFD simulations to study the issues arising with scaling and solidity of tidal stream turbines.

Acknowledgement

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