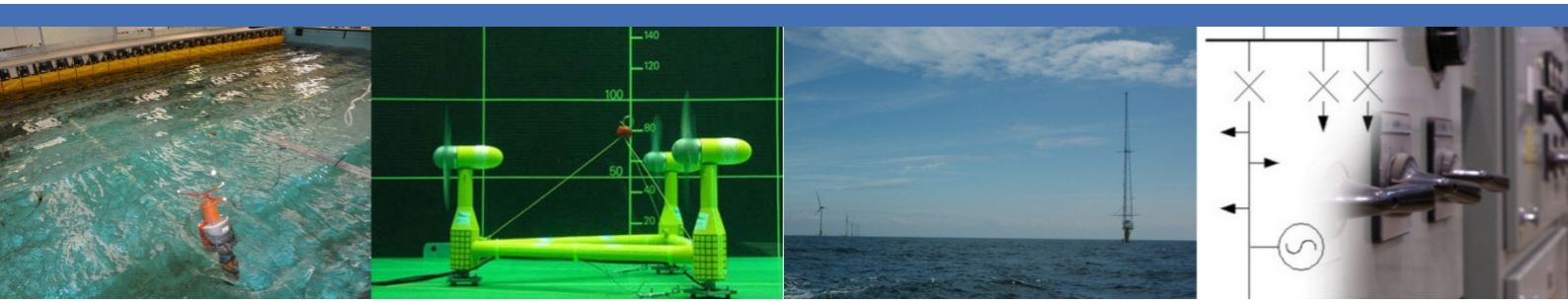
MARINET

Marine Renewables Infrastructure Network



WP2: Marine Energy System Testing - Standardisation and Best Practice

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Report on Comparative Testing of Tidal Devices

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ABOUT MARINET

MARINET (Marine Renewables Infrastructure Network for emerging Energy Technologies) is an EC-funded network of research centres and organisations that are working together to accelerate the development of marine renewable energy - wave, tidal & offshore-wind. The initiative is funded through the EC's Seventh Framework Programme (FP7) and runs for four years until 2015. The network of 29 partners with 42 specialist marine research facilities is spread across 11 EU countries and 1 International Cooperation Partner Country (Brazil).

MARINET offers periods of free-of-charge access to test facilities at a range of world-class research centres. Companies and research groups can avail of this Transnational Access (TA) to test devices at any scale in areas such as wave energy, tidal energy, offshore-wind energy and environmental data or to conduct tests on cross-cutting areas such as power take-off systems, grid integration, materials or moorings. In total, over 700 weeks of access is available to an estimated 300 projects and 800 external users, with at least four calls for access applications over the 4-year initiative.

MARINET partners are also working to implement common standards for testing in order to streamline the development process, conducting research to improve testing capabilities across the network, providing training at various facilities in the network in order to enhance personnel expertise and organising industry networking events in order to facilitate partnerships and knowledge exchange.

The initiative consists of five main Work Package focus areas: Management & Administration, Standardisation & Best Practice, Transnational Access & Networking, Research, Training & Dissemination. The aim is to streamline the capabilities of test infrastructures in order to enhance their impact and accelerate the commercialisation of marine renewable energy. See www.fp7-marinet.eu for more details.

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LEGAL DISCLAIMER

EXECUTIVE SUMMARY

This report intends to provide guidelines to compare different types of tidal energy converters. The first section of the report shows the different groups and sub-groups of converters that can be found currently in the literature. Parameters such as scale factors and blockage corrections that need to be considered when testing tidal energy converters are also discussed briefly in the report. Methods currently adopted to measure structural loads and performance of tidal devices can be found in the report with recommendations on topics such as: data collection, instrumentation and post-processing of data.

The report then gives suggestions on the parameters that can be employed to carry out the comparisons between types of tidal converters (e.g. performance vs. solidity). Other parameters such as blade deflection or wake structure are also suggested as parameters that can be utilised for comparative studies; however, the collection of such data is always dependent on the testing objectives of the experimental session.

A case study is presented at the end of the report. A three bladed horizontal axis turbine was tested in four different facilities as part of the Round Robin testing programme of MARINET. In the results of the study, it can be observed that the performance characteristics of the turbine were slightly influenced by the testing facilities used, especially when the turbine was installed in facilities with turbulent flows. It was concluded that turbulence intensity did not influence the performance of the devices severely. However, the examination of other flow characteristics (e.g. turbulence length scales or wave-current interactions) should be studied in detail in the future.

CONTENTS

1 INTRODUCTION

This report encompasses the procedures and common practices used when comparing different types of tidal energy converters (TEC). It contains a brief description of the range of architecture to be found in existing turbines and the rotor configurations that distinguish each of them. The report also includes a list of parameters which should be considered when undertaking a comparison between different rotor and device configurations. Suggestions of measurement techniques to be adopted when assessing device performance are also included in the report. Finally, recommendations are made on conducting a comparative analysis on a range of different device architectures. Adoption of a more standard rotor blade design provides an initial approach to establishing optimum efficiency but other aspects will also be considered when selecting a specific rotor configuration/type (e.g. mechanical design, economics and structural design). If these are not correctly adhered to, the performance of the designated TEC might not comply with initial expectations and therefore may not prove to be successful at a commercial scale.

2 HORIZONTAL AXIS TIDAL TURBINES

The rotor arrangement of a horizontal axis tidal turbine can vary according to the number of blades and their disposition within the rotor configuration. For example, a turbine can extract energy with the use of single axial, dual co-planar and dual co-axial rotor designs but their configurations result in a different set of operational characteristics. These characteristics are identified for each rotor configuration in turn.

2.1 SINGLE AXIAL ROTOR

To date, single axial rotor configurations have been the most popular for development and testing. Figure 2.1 shows an example of a simple three bladed single axial rotor (SR) design. Most of the TEC designs deployed and tested use this type of rotor configuration. The main advantage of its use is its simplicity compared to other types of rotor configurations. Tidal developers have proposed the use of SRs with distinct numbers of blades, typically ranging from 2 to 6 blades in some instances. An example of the latter can be seen on the Sabella turbine (Dhomé, 2013). The selection of the number of blades on a rotor results in a trade off between the advantages and disadvantages in respect of the torque produced and the rotor speed. Work undertaken by Morris (2014) has shown that three bladed turbines produce more power than two bladed turbines, with a high disparity; however, any increase in power extraction by a 4 bladed turbine compared to a three bladed one is almost negligible. Meanwhile, shadowing effects and transient torque drops due to the presence of rigid support structures are usually more prevalent when using a two bladed rotor systems when compared to rotors equipped with at least 3 blades.

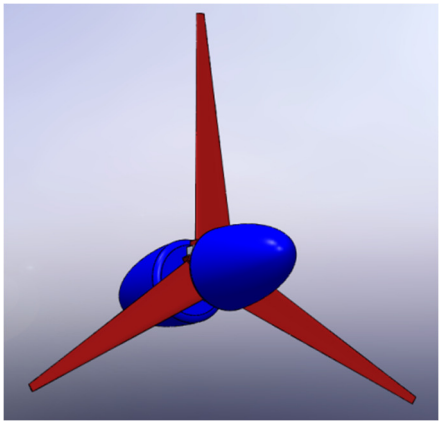

Figure 2.1 Single Axial Rotor

2.2 Dual Co-Planar Axial Turbines

As an evolution from the single axis rotor, dual co-planar, tandem rotors (TRs) typically use two single axis rotors supported from the same structure. Within system operations, these co-planar rotors should, ideally, counter rotate in order to reduce extreme loading on the supporting structure and improve stability of device operation through each rotor counteracting the torque developed by their opposite angular motions, as shown in Figure 2.2. TRs are extensively used in both the aviation and marine propulsion industries. In the marine renewable context, the use of TRs appears to improve the cost efficiency associated with the expensive fixed costs of the supporting structure since the energy extracted is doubled compared to single axis rotor devices. However, since this configuration comprises two rotors side-by-side, the loading imposed on the structural support is doubled and the costs associated with doubling the hold down capacity and increasing the stiffness and strength of the support system to accommodate this will increase. Additional influences on operational performance may be encountered depending on the number of blades used on the rotor configuration, where shadow effects might be introduced, resulting in a reduction in the energy capture capabilities of the system. Ordonez-Sanchez (2013) demonstrated that depending on the structural connection between rotors, vortex shedding can lead to vibration problems with impact on device stability, structural fatigue and operational performance.

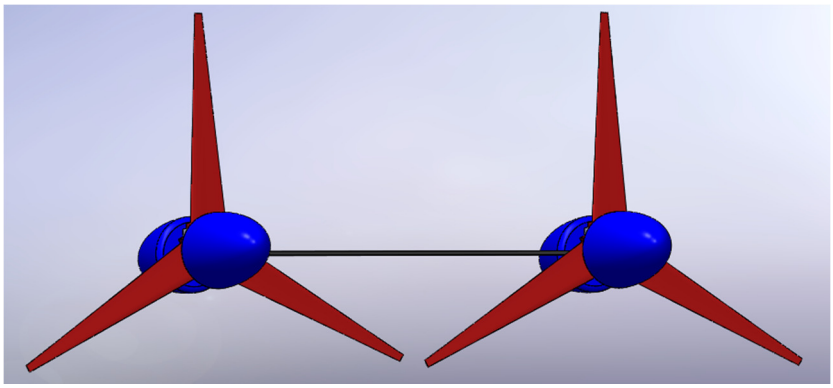

Figure 2.2 Tandem Rotor Tidal Energy Converter

2.3 Coaxial Contra Rotating Rotor

Coaxial, contra rotating rotors (CCRR) consist of axially aligned front (upstream) and rear (downstream) sets of rotors, as demonstrated in Figure 2.3. The front and rear rotors rotate in opposite directions (contra-rotate), to induce reactive torque compensation and inherently balance operational forces. This force compensation ensures the device remains stable during operation. CCRRs have been employed for propulsion systems in the marine and aviation industries (Johnson, 1994). When applied to marine propulsion systems (Ghassemi, 2009), this has demonstrated that contra rotating rotors reduce the reactive torque in the shaft and increase the efficiency of the thrust delivered by the propulsion system. Moreover, CCRRs can provide the required power with smaller rotor diameters, which can be important in shallow water applications. Other improvements have been associated with reductions in noise and vibration compared to what is observed in traditional SRs (Jukola & Ronkainen, 2006). This enhancement in performance has resulted in a reduction to a ship's fuel consumption during operation.

The aviation industry has used CCRRs to increase the manoeuvrability of helicopters due to torque and moment of inertia reduction which simplifies control mechanisms (Chen & McKerrow, 2007). It has also contributed to an increase in stability of hover being obtained when using this type of rotor configuration. The most significant inconvenience found with the use of CCRR based machines is that some aspects of the mechanical design of the drive system can be complex when driven by a single drive shaft. In these instances, the machine's design and production costs can increase considerably when compared to single rotor systems used within the aviation industry. Wider performance benefits of using CCRR configurations can extend beyond the torque balance properties. Investigations by O'Doherty et al. (2009), into the applications of CCRRs within the tidal sector have found wake propagation behind the rotors to be less intense and diffuse quicker from that of a single rotor device. In such

instances, this enables CCRRs to achieve an increased packing density of turbines in a confined site, allowing for more turbines to be deployed and greater power capacity to be extracted from a constrained area of sea bed. Work undertaken by Ordonez-Sanchez (2013) has demonstrated that the power generation from a similar diameter CCRR increased by up to 13% when compared with a conventional single rotor. In contrast when compared with CCRRs, coaxial rotors can produce up to 30% more thrust (Ordonez-Sanchez., 2013). Such comparative performance details were found to be in agreement with the studies undertaken by Kumar et al. (2012), who reported an increase in thrust of up to 35%.

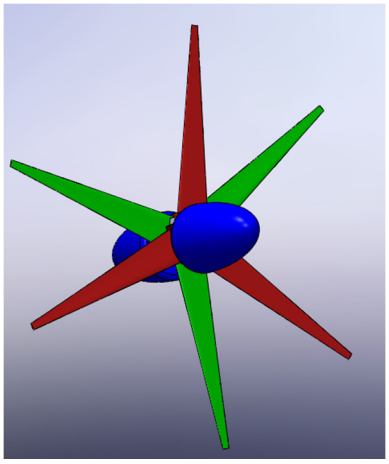

Figure 2.3 Coaxial Contra Rotating Rotor

2.4 Parametric Comparison for Horizontal Axis Turbines

There are two main parameters that can be used as the comparative metrics when investigating different types of tidal turbines: the swept area of the rotor; and the solidity of the rotor. The influence of each of these is described as follows.

2.4.1 Swept Area

The swept or capture area of a horizontal axis turbine is the area occupied by the rotor. For horizontal axis turbines this can be calculated simply as the area of a circle encompassed by the rotor as it rotates one revolution, as represented by

$$A = \pi R^2 \qquad [1]$$

where R is the radius of the turbine rotor, to include the area occupied by the nacelle.

The swept area (m^2) is a parameter which is not only used to compare turbine configurations and inform the potential power capture, but also enables quantification of the blockage ratio, the ratio of the rotor cross sectional area divided by the cross sectional area of the testing environment being used to evaluate the rotor/device performance. High blockage ratios (typically greater than 6%) will influence and affect the accuracy of any performance metrics produced from the testing, as recognised in the Equimar protocols (Stallard, 2011). Empirical correction factors have been derived for moderate blockage factors, i.e. sub 20%, and can be applied to test data to provide correction to test metrics produced. In most laboratory based testing programmes correction for blockage needs to be applied.

As identified in the EquiMar protocols, which quantifies the effect of blockage on the turbine power capture performance metric C_p, in instances when there is high blockage ratios this can result in computed C_p values exceeding 1.

2.4.2 Blockage Correction

When testing tidal turbines in tow or flume tanks, higher power and thrust coefficients may be estimated even if blockage ratios are less than 10%, as mentioned before. The flow stream is modified due to boundary restrictions created by the tank walls and free surface of the flow. This results in an acceleration of the flow through the turbine which will yield a higher power performance compared to that when the turbine operates in an unbounded environment. One suggestion to correct this is presented by Bahaj et al. (2007), who based his study on an actuator disc model where the constrained downstream wake was taken into account in the model. The equations to correct blockage ratios are presented in equations 2-4. A full description of the procedure can be found in Bahaj et al. (2007).

$$C_{PF} = C_{PF}(\frac{U_T}{U_F})^3 \quad [2]$$

$$C_{TF} = C_{TF}(\frac{U_T}{U_F})^3 \quad [3]$$

$$\lambda_F = \lambda_F(\frac{U_T}{U_F}) \quad [4]$$

where C_{PF}, C_{TF} and λ_F are the power coefficient, thrust coefficient and tip speed ratio in free stream condition respectively. U_T and U_F, are the flow speeds recorded in the tank with and without rotor respectively.

In the context of performance evaluation being undertaken in 'real-sea' testing, any discrepancies introduced into these evaluation metrics from the physical environment being used for quantification will primarily limit the scope and applications of correction factors developed for use in indoor laboratory facilities. Other influential factors associated with both laboratory and 'real-sea' testing environments include the test model's proximity to the seabed or the 'free' surface. More importantly, if tidal turbines are being compared, any testing and evaluation should be undertaken in facilities which provide similar blockage ratio conditions and rotor clearance from the bottom and free surface.

2.4.3 Solidity

As the swept area only considers the diameter of the rotor and is independent of the number and geometrical characteristics of the blades, the solidity of the rotor should also be included in any comparative analysis since this will influence rotor speed and the thrust loads experienced.

The solidity of the rotors can be defined as the annular area covered by the blades of a turbine rotor without considering the void space, as defined by Hansen (2008).

It can be calculated as the following equation:

$$\sigma = \frac{Nc(R)}{2\pi R} \quad [5]$$

where σ is defined as the rotor solidity, c is the average chord and N is number of blades on the rotor.

When using tapered blades, an average of the summation of blade chord sections should be performed. However, a good approximation would be to establish an equivalent solidity on varying chords, as depicted by Johnson (1994). Therefore, the following equation can be used:

$$\sigma = \frac{Nc(r{=}0.75)}{2\pi R} \quad [6]$$

It has been demonstrated that even minor changes in rotor solidity can change the performance characteristics of a tidal turbine. Through analytical work undertaken by Morris (2014), it was observed that the turbines structural performance can be improved when a larger number of blades are used in a rotor configuration. Also, as the

distribution of thrust loads is spread over a larger number of blades, the loading experienced on the blade root is lower than that supported by turbine rotors with fewer blades. When considering next generation rotors like CCRTs, the solidity should be quantified by including the number of blades used in both rotating sections.

3 VERTICAL AXIS TURBINES

Vertical Axis Turbines (VAT) have undergone wide scale development in the marine energy sector. VATs can primarily be divided into two main types, Darrieus or Savonious rotor. A brief description of the operational performance of these devices is given below. Studies undertaken by McAdam (2011) identified a number of advantages associated with the use of VATs. These include improved power coefficients associated with rotor power capture when operating in high blockage ratios in the flow. However, due to the rotor blades on VATs continuously varying their angle of attack during operation, this results in large fluctuations in drive torque. To mitigate this, the rotor needs to be designed correctly, with a sufficient numbers of blades each located at specific arc around the circumference of the rotor. While a smaller number of blades making up such a rotor can provide faster rotational speeds, they are prone to stalling when large fluctuations in torque are experienced in the drive train, as found by Falconer (2012), which could become a severe drawback for the selection of these types of systems.

3.1 DARRIEUS TURBINE

The majority of the VATs are evolutions of the design initially proposed by the French engineer Georges Darrieus in 1931. Variants of his design include the cross flow turbines and giro-mills, as shown in Figures 3.1 and 3.2. The use of cross flow turbine rotors within tidal energy applications usually employ blade sections which consist of a constant span and a fixed pitch. In some cases, the turbine rotors are designed with twisted (Gorlov) blades and are installed horizontally so they can also be used in shallow water applications, see Gebreslassie et al. (2013), and Figure 3.1. Most of the turbines of this type are designed with at least 3 blades making up the rotor set, but some designs suggest the use of 6 or more blades, as shown by McAdam (2011).

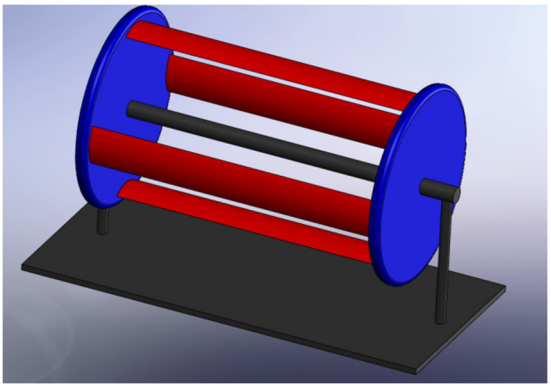

Figure 3.1 Darrieus tidal turbine types: Cross Flow

Giro-mills are another variant of the Darrieus turbine. These turbines have been designed to use blades configured into a fully packed device, as shown in Figure 3.2 and presented by Gretton (2009). A further variant/evolution of the giro-mill turbine is the cyclo-turbine. As its name suggests, the turbine is equipped with a pitch control system which allows a change in pitch of the rotor blades as it rotates around its radius of circumference. Varying the blade pitch as the rotor rotates around its circumference improves the power capture performance of a VAT. Similar to HATs, the swept area and solidity of the rotor have significant impacts on the performance characteristics of this type of device. Both parameters should be taken into account when comparing the performance parameters of these systems or with a different rotor type (e.g. HAT).

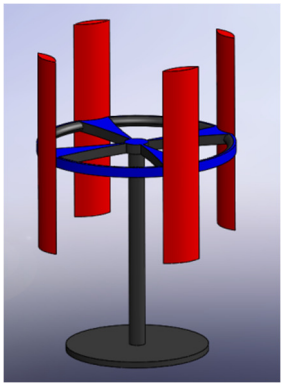

Figure 3.2 Darrieus tidal turbine types: Giro-mills

3.2 SAVONIUS TURBINE

The Savonious turbine is a design first proposed by the Finnish engineer Sigurd Johannes Savonius in 1922. The turbine primarily works as a drag type device and it usually comprises two blades as shown in Figure 3.3. In the marine energy industry, its use has been proposed by various developers, one example is the Carbine turbine (Falconer, 2012). The swept area and solidity are modified by vertically stacking rotor blocks with an equal offset pitch and also, a change of blade length and number of blades. Other modifications include the blade overlap ratio.

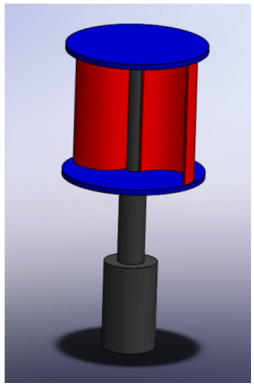

Figure 3.3 Savonious tidal turbine type

3.3 ROTOR PARAMETRIC COMPARISON

As discussed in section 2.4, the main parameters which can be used to compare the performance of different configurations of VATs are:

3.3.1 Swept area

The swept area of a VAT is related to the geometry of the device. This can be calculated by multiplying the rotor height (H) by the rotor diameter (D). This can be used to capture the area swept by most types of vertical axis turbines; however, the developer should be aware that sometimes the rotor diameter can be arbitrary and it will depend entirely on the rotor design. It has been shown by Yaakob et al. (2013) and Falconer (2012) that a stacked Savonious turbine performs better than a single turbine configuration of the same type.

3.3.2 Solidity

Similarly as described in Section 2.4.3, the solidity of VATs can be calculated using equation 5. An analytical study undertaken by Gretton (2009) showed that a 3 bladed rotor performed somewhat better than a four bladed turbine; however, the difference in performance was low.

This in any case shows how influential the solidity of a rotor is on the performance of any type of tidal turbine and thus, is a required parameter for comparison purposes.

4 SCALING AND ROTOR CAPACITY

4.1 SCALE FACTORS

There are two main dimensionless parameters relevant when doing testing of marine energy converters. These are the Froude number (Fr) and the Reynolds number (Re). In undertaking any scaling evaluation, only one parameter can be considered; thus, the devices will only be realistically scaled in terms of the surface effects or blade effects respectively (Massey & Ward-Smith, 1998). These parameters can be quantified as follows:

$$Re = {c(R)V_{flow}}/{v} \quad [7]$$

$$Fr = {V_{flow}}/{\sqrt{gL}} \quad [8]$$

Where:

- c corresponds to local chord at the radial position r
- v is the kinematic viscosity of the fluid
- V_{flow} is the flow velocity
- g is the acceleration due to gravity and
- L is a characteristic length scale (i.e.water depth).

For complete dynamic similarity, any model tests should be carried out at identical values of Fr and Re, compared to the full-size prototype. However, similarity of ambient flow field states such as turbulence intensity should also be considered during the tests.

Since Fr is related to water surface elevation, this scaling parameter must be used if wave motions are included in the experimental programme. While Fr scaling is relatively easy to achieve, Re scaling is not and complete dynamic similarity is practically impossible. The Reynolds number is a key parameter in determining the performance of a turbine rotor. Tests on small models will almost inevitably be conducted at values of Re which are too low for dynamic similarity with full scale. The likely effect on aerofoil sections are well documented (Molland, et al., 2004), consisting of a reduction in peak drag coefficients and a small increase in the lift coefficient as the Reynolds number increases. For a turbine rotor, this creates an increase in the power coefficient throughout the range of operation, which if not taken into account will result in an over-prediction of turbine performance at the field scale.Fortunately the overall effects of reducing the Reynolds number are moderate and progressive in nature. Turbulence is not governed by any dimensionless number. However, the ratio of the turbulent eddies relative to the size of the prototype could be compared to the full size characteristics. Since this is an extremely difficult process, turbulence intensities between laboratory and real case scenarios should be compared.

In this work programme, it has been found that flume tanks are capable of generating turbulence intensities similar to tidal test sites. For example, Rose et al. (2010) reported a turbulence intensity of 7% in a flume tank; meanwhile, Gooch et al. (2009) measured an average value of 10% at the Puget Sound test site in Washington State, USA.

4.2 PERFORMANCE CHARACTERISTICS

When doing parametric comparisons of different tidal turbines or testing one type of turbine in different test facilities, as in the case of the MARINET round robin tidal testing programme (see section 7), performance characteristics of the turbines should be considered in a comparative analysis. As established by Betz's law (Hansen, 2008), a turbine cannot extract completely all the kinetic energy from the flowing fluid. Therefore, the energy yield is limited by the power coefficient (C_p) which is expressed as the ratio of the available power to the power extracted by the turbine. The C_p of each turbine, or the turbine tested at the different test facilities is thus based on several

factors; that is, the blade profile characteristics and the environmental conditions of each specific test site. Hence, the calculation of the power output is determined by:

$$P = 0.5\rho A C_p {V_{flow}}^3 \quad [9]$$

where:

- P is the calculated power generated (predicted) by the specified turbine,
- ρ represents the water density
- A corresponds to the swept area of the rotor.

As the turbine extracts energy from the fluid, a pressure drop is generated in the axial direction of the turbine rotor, which is known as the thrust force. Similarly to the power extracted, the thrust force is related to a thrust coefficient (C_t). The thrust coefficient varies in accordance with the flow velocity and the rotor's angular velocity; and it is measured with respect to the rotor plane (wake centre) (Frohboese & Schmuck, 2010). In an equation which considers an ideal turbine the thrust is calculated as (Hansen, 2008):

$$T = 0.5\rho A C_T {V_{flow}}^2 \quad [10]$$

where:

- T is the calculated thrust load of the turbine

The coefficients C_t and C_p are usually related to the local rotational velocity of the rotor through a non-dimensional parameter called Tip Speed Ratio (λ). In order to obtain the maximum power output, the turbine must operate at its ideal/optimum λ. λ is calculated as follows:

$$\lambda = {}^{\omega\, R}/_{V_{flow}} \quad [11]$$

where λ corresponds to the ratio between the blade tip speed and the flow velocity (V_{flow}). ω symbolises the angular velocity of the rotor in radians per second. Finally, the torque (Q) generated by the turbine can be calculated by:

$$Q = {}^{P}/_{\omega} \quad [12]$$

Specifically, the power, torque and thrust calculated for comparison purposes should be obtained with the use of appropriate sensors and corresponding calibration procedures. These topics are briefly described in Section 5.

5 INSTRUMENTATION AND DATA COLLECTION

5.1 PERFORMANCE

The torque and thrust of a TEC can be measured in a number of ways. Specially designed torque and thrust sensors may be obtained from an off-the-shelf transducer; however, many manufacturers can provide custom made solutions that involve detailed waterproofing design requirements. A common practice is also to design the required measurement device using strain gauge elements to produce an electrical signal (e.g. dynamometers). In small models, bearing friction will be an unduly dominant feature and needs to be minimised. Ideally any torque measuring system should be configured to include friction from rotating bearings. Angular velocity of the rotors can be measured with Hall Effect or proximity sensors. These are used extensively within the tidal energy industry, but their installation might be difficult in small models and water ingress is always a problem. Magnetic reed switches are an alternative and their installation may be less complicated.

With calibrated instrumentation, the performance of the device and the computation of non-dimensional coefficients such as C_p and C_t can be accurately established and 'quality' data sets can be produced for the

calibration and verification of predictive models used for a comparative analysis. If other parameters, such as output voltage and current are measuredthen the electrical power output can also be included in the comparative analysis.

5.2 STRUCTURAL LOADS

Forces on rotor blades are of great importance for tidal turbines. It is therefore desirable to monitor these forces, especially because large deflections on a blade can give erroneous data of the turbine performance. It has been studied analytically and experimentally by Morris (2014) and Ordonez-Sanchez (2013) respectively, that deflection on a turbine blade will increase significantly as the number of blades is reduced in a TEC. Load measurement is usually carried out using strain gauges or purpose-built load cells incorporating resistors or piezoelectric elements. Strain gauges are compact, robust and provide a linear response over a wide range of frequencies. They operate by measuring changes in the electrical resistance of a small element, so can be highly sensitive to temperature fluctuations. In marine applications the presence of large volumes of water tends to ensure temperature stability, but can present its own problems in terms of sealing. The size of the strain gauge has to be considered during the design stages; very small gauges are available for specialised applications but are very fragile and can suffer from instability and fatigue.

5.3 VELOCITY AND TURBULENCE MEASUREMENT

The measurement of the flow velocity, although not compulsory, should be considered in a comparative analysis to provide information of the wake structure of different turbine types. The method used to measure the flow velocity depends on the nature of the test facility. In a towing tank the carriage speed will be controlled via the drive train, but an independent monitoring system might be advisable, especially, if it is intended to measure at least a point in the wake of the device. In this case, flow meters including pitot tube probes can give accurate measurements. In a flume tank, questions of velocity distribution and turbulence arise: conditions within the working section should be investigated prior to the test programme using acoustic or laser Doppler velocimetry probes (ADV, LDV) or other 'point' measuring devices. During testing, continuous monitoring of velocity at a single reference point may suffice. It should be mentioned that the calibration of sensors should be done as specified in various protocols(McCombes, et al., 2010).

5.4 DATA LOGGING AND SAMPLING RATE

One of the most important rules of signal processing techniques is the 'Nyquist-Shannon Sampling Theorem', stating that the sampling rate of the acquired data must be at least twice the expected peak frequencies (Chitode, 2009). For tests on TECs, it is expected that the signal will be mainly related to the rotational speed of the turbine and blade passing frequencies. Compared to rotating machines in general, this might be regarded as the low frequency part of the spectrum, and sampling rates in the region of 30 to 500 Hz should be adequate. This is well within the capabilities of modern data logging systems. As always, there will be some compromise between logging frequencies, storage capacity and sample sizes. The sample size and duration of tests should be determined with reference to established protocols and standards, such as the EquiMar protocols (EquiMar, 2012). For example, (McCombes, et al., 2010) states that the sample size should be calculated using the following relationship:

$$n = \left({}^{s}/_{Uc}\right)^2 \qquad [13]$$

where:

- n is the number of samples
- s is the sample standard deviation
- U_c is the interval in which there is a confidence interval of 95%.

This means that the variability of the data should not be greater than 5%. This is easily achievable in circulating flume tanks where the data can be analysed with auto-regression techniques. This means that a running mean (k) can be instantly compared to the k-1 test. In the case of single tests, such as the ones undertaken in tow tanks, care should be taken as even with a high sampling frequency each run is considered as a single experiment (McCombes, et al.,

2010). Therefore, data can be collected and compared betweena number of sets until the threshold has been satisfied.

If periodicity of the signal is being monitored, three analyses can be undertaken. First, the average of the cycles can be determined. Secondly, spectral moments should be considered from zero to order 2 with a 5% convergence. The third analysis is similar to the former analysis, but in this case, window functions should be applied before spectral moments are calculated. Time and frequency domain analysis should be shown in a comparative analysis while including the uncertainty analysis. More information about this can be found in (McCombes, et al., 2010).

6 PARAMETRIC COMPARISON

The characteristics of different marine turbines or a turbine tested at a number of testing facilities can be related in terms of their performance, loading or wake structure. Each of these forms is detailed in the following sections.

6.1 PERFORMANCE CHARACTERISTICS

Turbine configurations with different rotor characteristics, solidity or swept area can be presented in customary C_p-λ or C_t-λ diagrams, as those shown in Figure 6.1. More specifically, the peak power and peak thrust coefficients can be compared in terms of configuration/number of blades/solidity or swept area. It has been observed in studies of vertical and horizontal axis turbines that the performance of a turbine increases while increasing the number of blades ((Morris, 2014), (McAdam., 2011), (Gretton, 2009)). When changing the characteristics of a rotor not only will the performance be modified, the operational regions will also shift to low/high tip speed ratios, or the regions for possible power extraction will be altered.

Turbine configurations can also be compared in terms of torque variation as it has been shown in Consul et al. (2009). This is especially useful for VATs, due to large torque ripples. Blade loading can also be used in a comparative analysis, for example, in terms of power against blade deflections. These measurements are not frequently measured in a TEC testing programme, especially if the device is of small scale e.g. 1/25. If however, the turbine is equipped with strain gauges in the root section this would give useful insight on the blade loading.
It has been observed in HATs that a change of solidity would imply a substantial inverse correlation on the load on each of the blades (Morris, 2014).

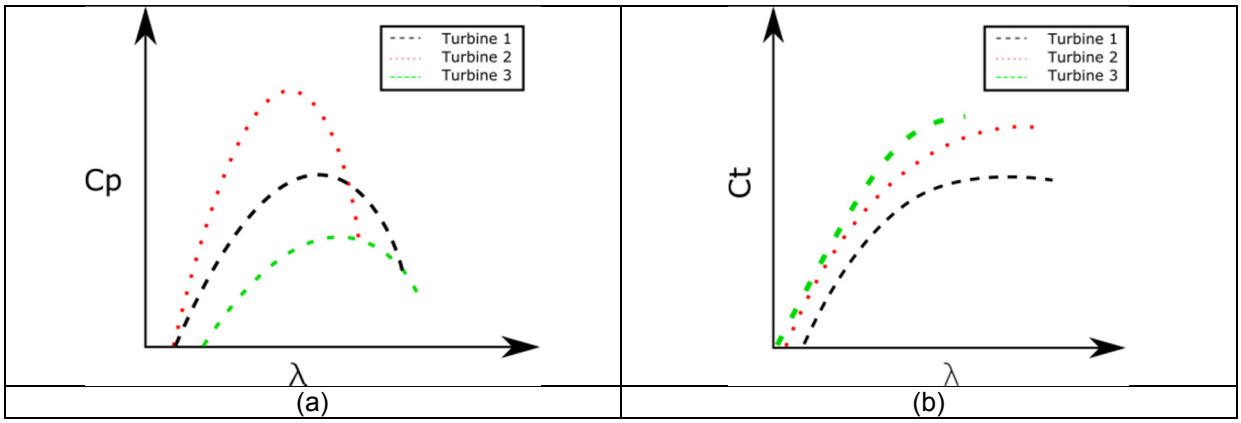

Figure 6.1 Performance characteristics a) C_p-λ and b) C_t-λ

6.2 WAKE STRUCTURE

The wake structure of the turbine can also be used as a comparative parameter for TECs with different rotor configurations. A common practice is to analyse the near and far wake of the turbine by measuring velocity points forming 'velocity grids' in the cross sectional plane at various downstream locations. If simpler current meters are used in the analysis or time is very limited, the velocity grids can be reduced to the measurement of a single point on

the centreline of the turbine at various downstream locations. Then velocity deficits from near to far wake should be shown in the analysis comparing each type of configuration, as shown in Figure 6.2. If instruments able to measure 2 or 3 velocity components and an adequate sampling rate are used, e.g. ADV or LDV; turbulence intensity variations should also be included in the analysis.

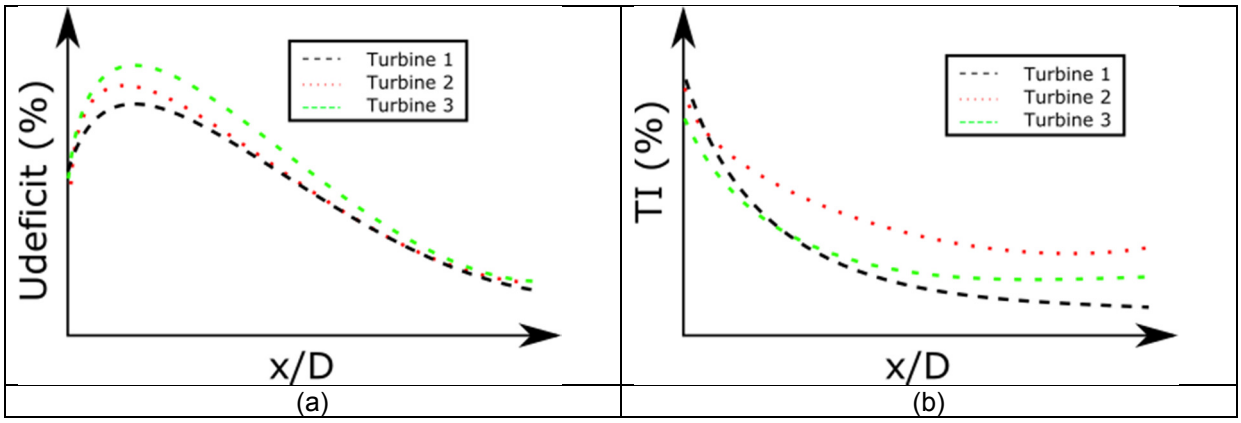

Figure 6.2 Wake characterisation in terms of a) velocity deficit and b) turbulence intensity for different turbine configurations.

If the testing considers the measurement of various velocity points forming grids at various downstream locations (D) velocity maps relating the axial velocity, turbulence and Reynolds stresses should be included in the analysis for comparison purposes, as those shown in (Mycek, et al., 2014). This information would give an insight into wake merging and/or lateral and vertical wake expansion.

6.3 Parametric Normalisation

Data normalisation is a common practice undertaken to obtain relationships between different types of systems and turbine configurations and may also be important in the case of testing the same turbine in different test facilities. Using normalisation enables comparison of the characteristics of each converter type or test facility regardless of the measurement range. For example, the performance of the turbines can be normalised in terms of the maximum C_p and maximum λ, as shown in Figure 6.3. The same can be done to compare characteristics of thrust, torque and blade loading. Moreover, normalisation techniques can also be employed when comparing the wake structure of several devices. For example, this can be done by comparing the mean velocity to the velocity components of the data sets. Currently, there are not many experimental investigations comparing the wake of different TECs; however, the computational analysis undertaken by Morris (2014) showed that the wake of a turbine with a larger number of blades presents a larger velocity deficit in the near wake region than the others with fewer blades; however, the velocity of each turbine wake recovers almost at the same downstream location.

Even though these sections have shown the different ways to compare turbines in terms of the most common parameters measured in the marine energy field, other parameters can be included in this type of analysis. Ordonez-Sanchez (2013), for example, compared the dynamic response and vibrations of different types of tethered turbines. Walker (2014) compared the wake structure of a rotor when using different station keeping/supporting structures for TECs. Therefore, comparing TECs should not be restricted only to the parameters mentioned in this report.

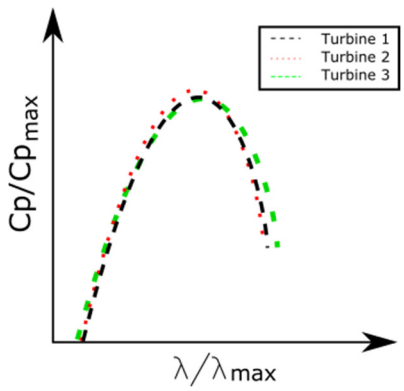

Figure 6.3 Normalised C_p-λ

7 COMPARATIVE RESULTS FROM THE 'ROUND ROBIN' CASE STUDY

A Round Robin testing programme was undertaken during the MARINET project. The Round Robin testing was undertaken using a three bladed horizontal axis turbine, as shown in (Gaurier, et al., 2015) and in MARINET deliverable D2.24 (Iyer et al., 2015). The turbine was tested in four different facilities, two tow tanks and two flume tanks, as seen in Table 4.2. A turbulence intensity of around 3% was used in both flume tanks to resemble the operation of the tow tank. The main results obtained from this set of tests are summarised in the following sections.

Facility	IFREMER	KHL	CNR-INSEAN	CNR-INSEAN
Type of Facility	Flume	Tow	Flume	Tow
Turbulence Intensity (%)	3	N/A	2.5	N/A
Blockage Ratio (%)	4.8	3.3	4.8	1.2
Flow Speed (m/s)	0.6-1.2	0.6-1.2	0.6-1.2	0.6-1.2

Table 4.2 MARINET facilities used in the Round Robin Testing

7.1 POWER COEFFICIENT

Figure 7.1 shows good agreement between the results obtained in the four different tanks. However, one discrepancy is observed at λ values of 2.5 obtained in the IFREMER flume. The averaged C_p curves can also be divided into two different groups with the IFREMER and KHL tanks on one side having the highest values and the CNR-INSEAN facilities on the other side with the lowest values. Higher values of standard deviation of the power coefficient are clearly observed for the IFREMER flume. There is good agreement between the two towing tanks in terms of the standard deviation and as expected, the results from the flume tanks show a higher variation in the power data.

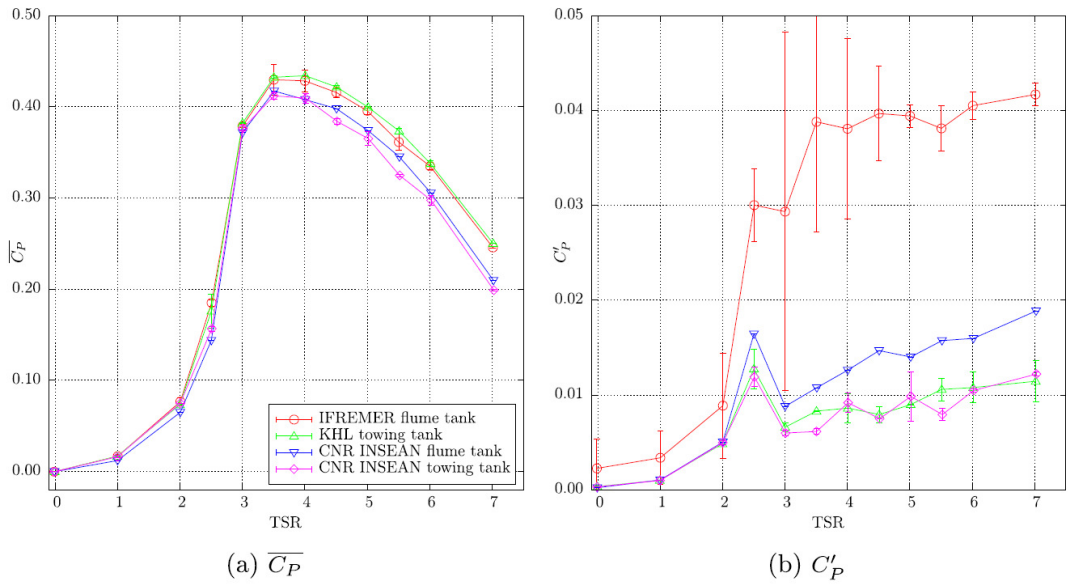

Figure 7.1 (a) Mean and (b) standard-deviation of the power coefficient obtained for every run at every tank for U=1m/s

7.2 Thrust Coefficient

Similarly to the power coefficient curves, there is good agreement in the C_T curves shown in Figure 7.2 between the test facilities for the averaged thrust coefficient. However, the data can be grouped into two distinct sets where slight differences can be observed after λ values of 2.5. The CNR INSEAN flume and KHL tank show similar values of thrust coefficients with higher values than those observed for the IFREMER flume and CNR INSEAN towing tank. On the contrary, there is greater discrepancy between the results from the four facilities when considering the standard deviation of the thrust coefficient (Figure 7.2b). Three groups can be distinguished in Figure 7.2b, with the CNR-INSEAN towing tank having smaller values of standard deviation than the other facilities. The CNR-INSEAN flume tank and KHL towing tank present values just slightly higher. And finally values obtained at the IFREMER flume tank reflect the highest variations of the thrust coefficient. Further details on the power and thrust performance curves for the Round Robin test programme can be found in (Gaurier, et al., 2015).

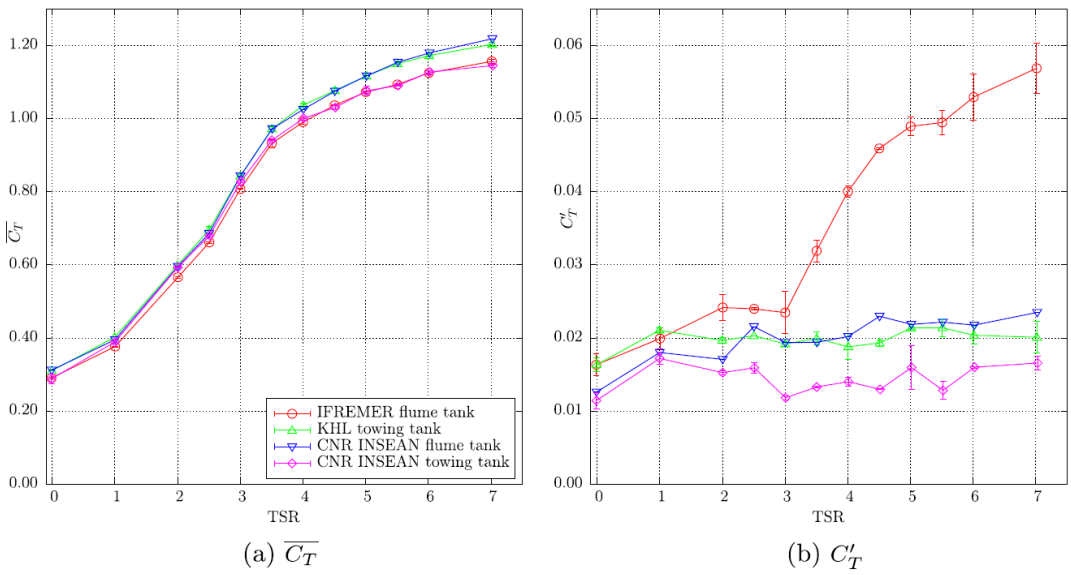

Figure 7.2 (a) Mean and (b) standard-deviation of the thrust coefficient obtained for every run at every tank for U=1m/s

8 CONCLUSIONS

This report outlined the procedures to follow when comparing different types of rotor configurations. The report includes two main types of turbine configurations, horizontal and vertical axis rotor devices. Parameters such as solidity and swept area can be used to compare the performance characteristics of different devices. Additional information considering the wake structure should also be used when comparing TECs.

The information presented in this report included a case study taken from the Round Robin testing programme conducted during the MARINET project. It was observed that power and thrust coefficients obtained for a 3 bladed horizontal axis turbine were similar in each of the tow and flume tank facilities where it was tested. This gives confidence that the facilities used in MARINET are of high quality and can reproduce similar results for similar flow conditions. Further experiments should be undertaken in the future considering other parameters such as turbulence length scales and wave-current interactions on turbines.

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