Analysis of the viability of a radial Double Decker Turbine for application in Oscillating Water Column devices

A. Vega-Valladares, and B. Pereiras

Abstract—**Oscillating Water Column (OWC) devices are the most widespread among the different systems developed to harness the wave energy available on our coasts. Air turbines, normally used as Power Take Off (PTO) in these devices, are their most controversial part. Unidirectional turbines were first used with rectifying valves to take advantage of both characteristic operating stages of the OWC: exhalation and inhalation. The use of valves was quickly discarded and bidirectional turbines, which rotate in the same direction regardless whence the flow comes, were adopted as the most common solution despite reaching lower efficiencies than unidirectional ones. The Twin Turbine Configuration (TTC), based on the use of two unidirectional turbines, emerged then as a promising system, its main drawback being the duplicity of the equipment. The Double Decker Turbine (DDT) concept has been recently introduced to overcome these limitations since it combines in a single design the two typical solutions: self-rectifying behaviour and the use of unidirectional turbines. In this work, the performance of a radial DDT, composed of an InFlow Radial (IFR) turbine and an OutFlow Radial (OFR) turbine, is assessed. A CFD model is used to design and optimize an IFR turbine to be mounted in combination with an OFR turbine taken from the literature. Finally, a non-steady analysis is carried out assuming sinusoidal flow conditions. Results demonstrate that this radial version of the DDT could compete with its axial version and with other alternatives.**

Keywords—**CFD, DDT, InFlow Radial (IFR) turbine, OutFlow Radial (OFR) turbine, OWC, wave energy.**

I. Introduction

OVER the decades, much research has focused on the development of new energy sources. Despite having an enormous potential [1], the ocean is nowadays an untapped renewable energy source. Due to their simplicity, Oscillating Water Column (OWC) devices are one of the most studied among the different technologies developed to harness wave energy [2]. Apart from that, in these devices the power transformation always takes place out of the sea, and they can be installed onshore. As a result, the lifespan of the facilities is enlarged, and their maintenance costs are significantly reduced in comparison to other wave energy devices.

Typically, OWC systems are composed of three parts [2]: the OWC chamber, the Power Take Off (PTO) and the electric generator. Fig. 1 [3] shows the principle of action of an OWC device. Basically, these devices consist of a chamber opened to the sea at the bottom and to the atmosphere at the opposite end. With the passage of the waves, the water free-surface raises and falls generating, respectively, an air flow from the chamber to the atmosphere (exhalation) and from the atmosphere to the chamber (inhalation).

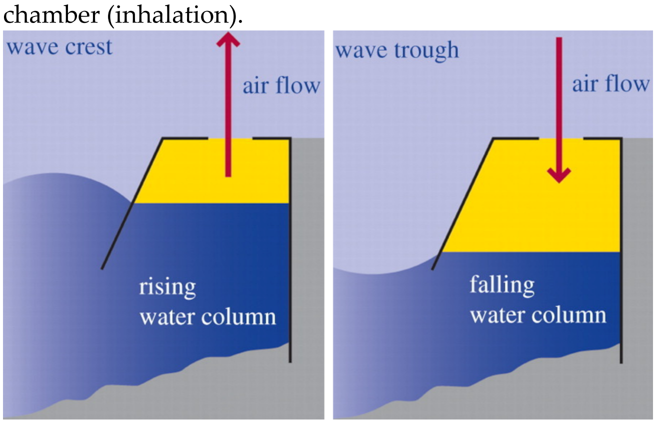

Fig. 1. Principle of action of an OWC device [3].

Despite having been extensively studied, the PTO is still the most controversial part of these devices. The use of an air turbine, always placed above the water free-surface and near the opening to the atmosphere, has been adopted as the most common solution [2].

Unidirectional turbines were first used in combination with a set of rectifying valves that allowed to take advantage of both characteristic operating stages of an OWC device: exhalation and inhalation. However, the use of any type of valves was quickly discarded because it contributed to reduce the lifespan of the whole system and increased the maintenance costs [2].

Nowadays, most of the prototypes installed around the world, mainly for research purposes, incorporate

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bidirectional turbines. These turbines, also known as self-rectifying turbines, can always rotate in the same direction regardless whence the flow comes [2]. Although they have been extensively studied and many researchers have tried to improve their performance, these bidirectional turbines traditionally reach lower efficiencies than unidirectional ones [2]. Patented by Alan Arthur Wells in 1976 [4], Wells turbines are probably the self-rectifying turbines most frequently used. These turbines can reach high peak efficiencies, sometimes very close to those reached by unidirectional ones. Nevertheless, they present a very narrow range of operating conditions and their efficiency drops dramatically working at high flow rates [2]. Patented by Ivan A. Babintsev in 1975 [5], impulse turbines are the most popular alternative to the Wells turbines. Although they can maintain acceptable efficiencies over a broader range of operating conditions than the Wells turbines, the peak efficiency they can reach is quite lower [2].

Twin Turbine Configuration (TTC) systems were first proposed by Jayashankar in 2009 [6]. Based on the use of a pair of unidirectional turbines, each one destinated to extract energy in one of the operating stages of the OWC, these systems also have their own disadvantages [6]. Firstly, the duplicity of equipment makes them economically less competitive. Secondly, while one of the turbines is efficiently working in Direct Mode (DM), extracting energy from the flow, the other one is working in Reverse Mode (RM), acting as an aerodynamic backflow preventer. Nevertheless, part of the flow will always leak through the turbine working in RM and the efficiency of the whole system will be reduced since only a certain percentage of the total flow will be really exploited.

In order to deal with these drawbacks, the Double Decker Turbine (DDT) concept has been recently introduced [7]. Based on the TTC, this concept consists of a single turbomachine composed of two unidirectional turbines working in parallel. Fig. 2 shows the operating scheme of the DDT concept in its radial version.

Based on the radial TTC, the radial DDT presented by the authors in this work is composed of two unidirectional radial turbines named as InFlow Radial (IFR) turbine and OutFlow Radial (OFR) turbine. On the one hand, when the OWC device is working in Outflow Mode (OM), the OFR turbine is efficiently working in DM and the IFR turbine is working in RM. On the other hand, when the OWC device is working in Inflow Mode (IM), the IFR and OFR turbines switch their roles.

Although recent studies demonstrate that the axial version of this new DDT concept could reach efficiencies in the range of bidirectional impulse turbines, some of its limitations should be further improved [8].

Previous studies show that OFR turbines allow an extremely good blockage of the flow working in RM and they can compete with axial turbines in the case of TTC systems [9].

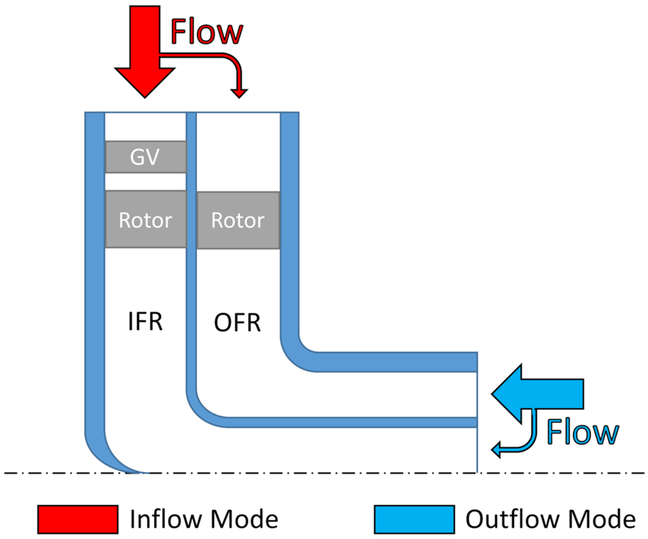

Fig. 2. Flow distribution and operating modes of the radial DDT.

In this work, the authors present the radial version of the DDT concept. This radial DDT is composed of an IFR turbine specifically created for this work and an OFR turbine taken from the literature [10]. The steady performance of the IFR turbine is assessed using CFD techniques. Since the results were unsatisfactory, a new design of the IFR turbine was created to overcome some limitations. Both, the results for the first and the optimized version of the IFR turbine are shown in this work. Finally, the viability of this preliminary radial DDT is analysed in terms of its non-steady efficiency.

II. DATA POSTPROCESSING

The steady performance curves of the IFR and OFR turbines is analysed in terms of the classical coefficients for OWC applications. The torque, input and flow coefficients (C_T, C_A, φ) as well as the total-to-static efficiency (η) are defined as:

$$C_T = \frac{4T_0}{\rho(v_R^2 + u_R^2)\pi b D_m^2} \tag{1}$$

$$C_A = \frac{2\Delta P_{t-s}\, Q}{\rho(v_R^2 + u_R^2)\pi b D_m\, v_R} \tag{2}$$

$$\varphi = \frac{v_R}{u_R} \tag{3}$$

$$\eta = \frac{\omega T_0}{\Delta P_{t-s}\, Q} = \frac{C_T}{C_A\, \varphi} \tag{4}$$

Where ρ is the air density, T_0 is the mechanical torque, ΔP_{t-s} is the total-to-static pressure drop, Q is the flow rate, D_m is the mean turbine diameter, b is the blade span, v_R is the reference radial velocity at the mean turbine radius (r_m) and u_R is the blade speed at r_m. Note that, the efficiency, which is the ratio of shaft power output to pneumatic power input, can be expressed in terms of the previous non-dimensional coefficients.

Once the steady performance curves are obtained, a non-steady analysis is conducted to assess the performance of the radial DDT assuming sinusoidal flow conditions, being the calculations based on the steady performance of each turbine. The formulation can be seen in (5)-(7), applied for different total-to-static pressure drops. Applications of this method can be seen in [7]–[11]. Note that, this formulation had to be rewritten to adapt this new topology to the radial DDT presented in this work, also considering the presence of two turbines.

$$\Delta P_{t-s} = \Delta P_{MAX} \sin\left(\frac{2\pi t}{T}\right) \tag{5}$$

Where ΔP_{MAX} is the amplitude total-to-static pressure drop, T is the wave period and t is the time. A typical wave period of 10 s was taken in this case for the analysis.

$$\bar{\eta}_{NS} = \frac{1/_{T} \int_0^T \omega\, T_{TOT}\, dt}{1/_{T} \int_0^T \Delta P_{t-s}\, Q_{TOT}\, dt} \tag{6}$$

Where $\bar{\eta}_{NS}$ is the non-steady efficiency of the radial DDT, and T_{TOT} and Q_{TOT} are, respectively, the sum of the mechanical torque and the sum of the flow rate of both OFR and IFR turbines.

$$\Gamma = \frac{\Delta P_{MAX}}{\omega u_R^2} \tag{7}$$

Where Γ is a non-steady pressure coefficient that increases as the amplitude total-to-static pressure drop increases.

III. GEOMETRIES STUDIED

The OFR turbine of the radial DDT presented in this work is composed of a single rotor with 24 blades. Neither downstream nor upstream Guide Vanes (GV) have been considered for this turbine. More information about this OFR turbine can be found in [10].

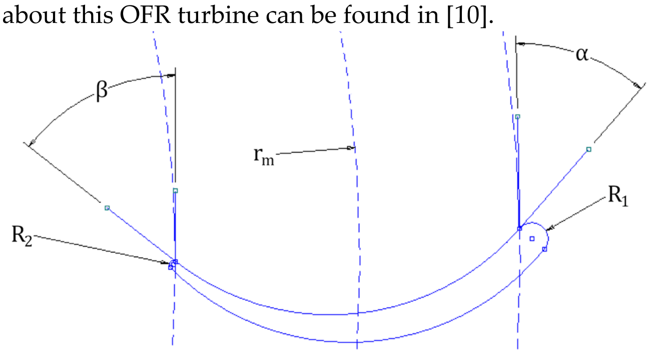

Fig. 3. Main view and geometric parameters for the rotor blades of the IFR turbine.

The IFR turbine studied in this work is composed of a single rotor with 48 rotor blades. Moreover, as in a conventional unidirectional turbine, 72 upstream GV have been added in order to increase its efficiency. Keeping the size of the OFR turbine, this IFR turbine has a mean radius of 87.5 mm. This and other geometric parameters of the IFR turbine can be seen in Fig. 3, being β the relative flow angle, α the absolute flow angle, R_1 the rotor blade leading edge radius and R_2 the rotor blade trailing edge radius.

The geometric parameters values are collected in Table I for both versions of the IFR turbine studied in this work, the first version and the optimized one. Note that, the most important difference between both versions is the value of the relative flow angle.

TABLE I
GEOMETRIC DATA OF THE IFR TURBINE STUDIED

Version	r_m (mm)	β (°)	α (°)	R_1 (mm)	R_2 (mm)
First	87.5	51.9	42.1	1	0.25
Optimized	87.5	40	42.1	1	0.25

Geometric parameters values of the first and the optimized version of the IFR turbine studied.

IV. NUMERICAL MODEL

The OFR turbine was previously studied using a well-validated CFD model [11], so those results were assumed to be reliable as a starting point for this work. Another CFD model has been developed and applied for the IFR turbine of the radial DDT presented in this work.

The mesh was built using ANSYS TurboGrid v16.2, which is a commercial software specifically designed to create high quality hexahedral meshes for turbomachinery applications. Fig. 4 shows the blocking strategy used for both blades and GV to obtain 2D mappable sub-domains. The mapped mesh generated was then extruded along the span direction to create the 3D structured mesh. Special attention was paid to the boundary layer mesh near the walls to obtain y⁺ values in the correct range around 1.

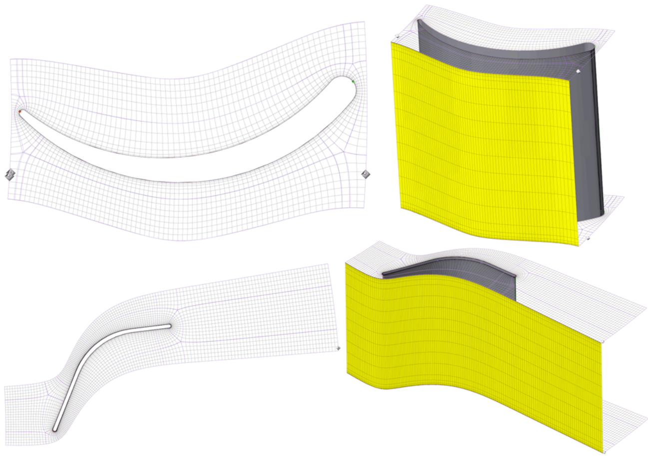

Fig. 4. Detail view of the mesh in the rotor blades and GV of the IFR turbine.

As can be seen in Fig. 5, which shows the boundary conditions, the model has been built over an angular sector of 1/24 of the full annulus, comprising 2 blades and 3 GV. Note that, periodic boundaries, which are a common solution used for CFD models to save computational resources, are typically implemented following the contour of blades and GV.

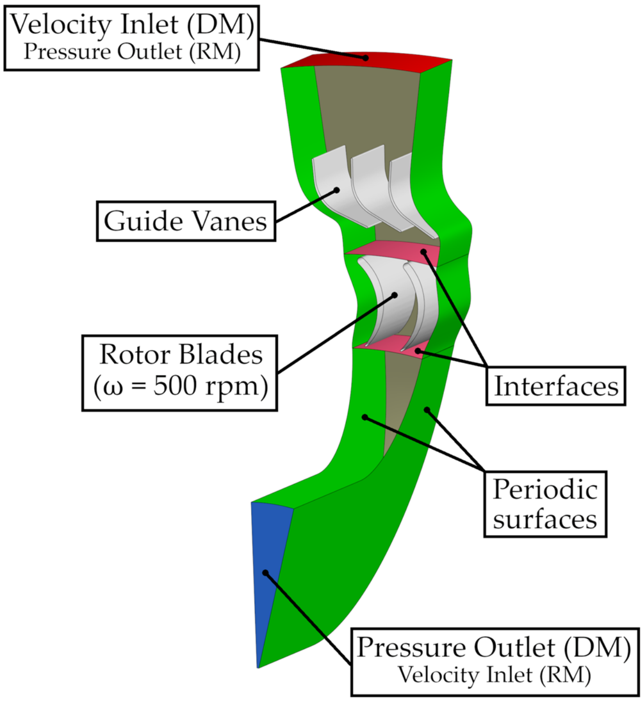

Fig. 5. Periodic domain and boundary conditions.

Two set of interfaces have been introduced in the computational domain. These interfaces separate the domain in three parts: elbow, rotor and GV+diffuser. The sliding mesh technique was used to simulate the relative displacement of the blades and the rotation speed of the rotor was set equal to 500 rpm.

The numerical simulations were made using the commercial software ANSYS Fluent v16.2 in steady flow conditions since the rotation speed of the turbine is much larger than the wave frequency. In order to set the appropriate flow rate, a pair of velocity-inlet/pressure-outlet boundary conditions was introduced in the model. The velocity at the inlet was in the range of 0.6 to 20 m/s depending on the flow coefficient desired, whereas the pressure at the outlet was set as atmospheric. Note that, inlet and outlet switch their location depending on the turbine operating mode, DM or RM.

The rest of surfaces of the model were set as non-slip walls.

With respect to the turbulence modelling, a robust, two-equation k-ε Realizable model was selected for the simulations, using the enhanced wall treatment function consistent with the y+ parameter, which takes values around 1 on all domain walls.

The simulations were carried out with SIMPLE scheme for the pressure-velocity coupling, a second order discretization for pressure and a third order discretization for momentum and the transient formulation, turbulent kinetic energy and turbulent dissipation rate. Residuals were set to 10^{-5} and the time step size was fixed to $5{\cdot}10^{-5}$ s. With these settings, each simulation lasted about 17 hours to complete 30 rotation cycles of the domain in a CPU equipped with 32 Gb RAM and Intel i7-5820 K (6 cores 3.30 GHz) processors.

V. Results and Discussion

As a starting point, the CFD results for both versions of the IFR turbine studied are compared in Fig. 6. In addition, the numerical results for the OFR turbine taken from the literature [11] are also included.

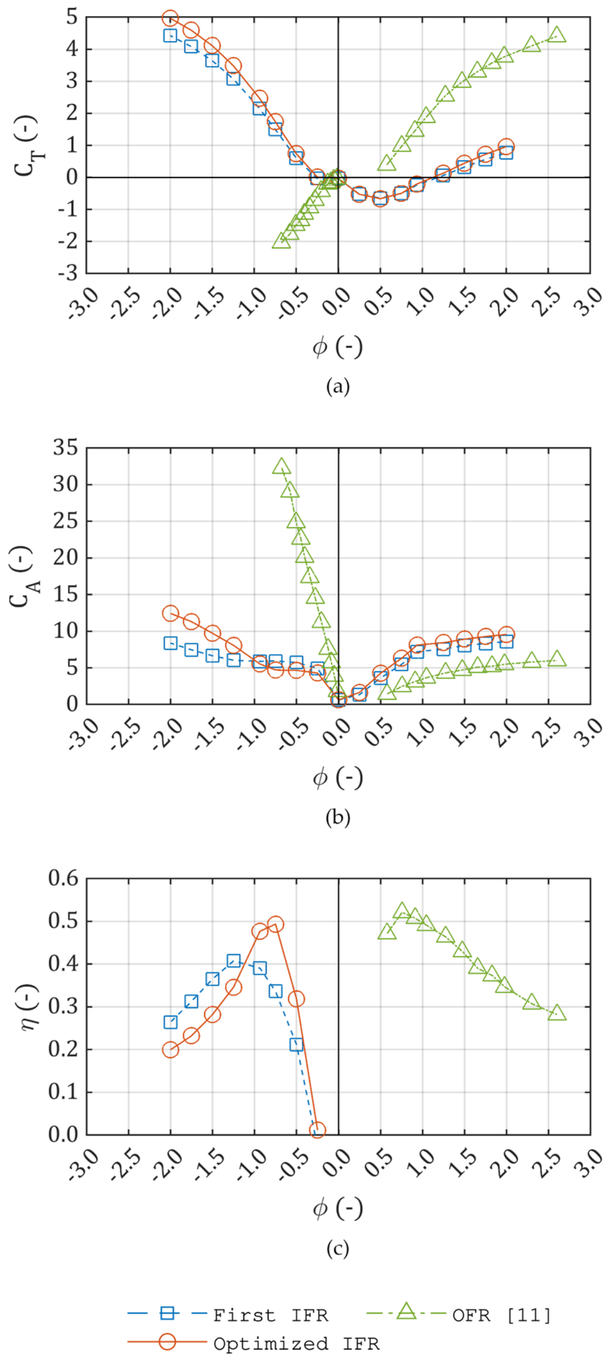

Fig. 6. Comparison of CFD results between both versions of the IFR turbine studied in this work (first and optimized) and the OFR turbine from [11].

The comparison is performed for the torque coefficient (C_T) in Fig. 6 (a), for the input coefficient (C_A) in Fig. 6 (b) and for the total-to-static efficiency (η) when the turbines work in DM in Fig. 6 (c). Note that, the positive flow coefficients ($\phi > 0$) correspond to the Outflow Mode (OM) and the negative ones ($\phi < 0$) to the Inflow Mode (IM) of the radial DDT presented in this work.

Remember that, when the DDT is working in OM, the OFR turbine is working in Direct Mode (DM) and the IFR turbine in Reverse Mode (RM), while when the DDT is working in IM, the IFR turbine is working in DM and the OFR turbine in RM, switching their roles. This can be noticed in Fig.6 (c) since while the OFR turbine is working efficiently in OM, the IFR turbine does it in IM.

Analysing the DM performance of each unidirectional turbine, it can be seen in Fig. 6 (a) that both the IFR turbine, in any of its versions, and the OFR turbine, present a similar C_T evolution, even the IFR turbine reaching higher values. However, it can be seen in Fig. 6 (b) that the OFR turbine reaches lower C_A values than any version of the IFR turbine, especially for high ϕ values. This explains why, in comparison with any version of the IFR turbine, the OFR turbine achieves, at $\phi = 0.75$, a higher peak efficiency of around 54 %.

Focusing on the IFR turbine, it can be seen in Fig. 6 (a) that the optimized version reaches higher C_T values than the first one. In addition, it can be seen in Fig. 6 (b) that for low ϕ values ($|\phi| \leq 1$), the optimized version reaches lower C_A values than the first one. All this makes the optimized version achieve higher efficiencies in this first part of the curve. While the optimized version of the IFR turbine achieves a peak efficiency of around 49 %, the first one only achieves around 41 %. Moreover, the peak efficiency has shifted to lower ϕ values since, while the first version reaches this point at $\phi = -1.25$, the optimized one reaches it at $\phi = -0.75$. However, as can be seen in Fig. 6 (b), the optimized version reaches higher C_A values than the first one for high ϕ values ($|\phi| > 1$). This means that the efficiency of the optimized version drops more abruptly than in the case of the first one when ϕ increases.

Finally, Fig. 7 shows the non-steady efficiency for both versions of the radial DDT presented in this work, the one that incorporates the first version of the IFR turbine and the one that incorporates the optimized one, both combined with the same OFR turbine from [10].

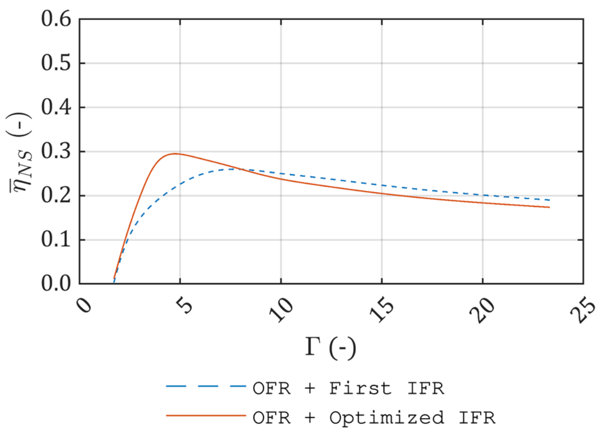

Fig. 7. Non-steady efficiency for both versions of the radial DDT presented in this work.

As could be seen in Fig. 6 (b), the C_A values reached by the OFR turbine when it is working in RM ($\phi < 0$) are much higher than the ones reached by this turbine when it is working in DM ($\phi > 0$). However, this phenomenon is not seen so clearly in the case of any version of the IFR turbine. This means that the OFR turbine blocks the flow better than the IFR turbine when they work in RM.

As a result of this bad behaviour as a backflow preventer by any version of the IFR turbine, the peak efficiency achieved under non-steady flow conditions by the radial DDT is quite lower than the one achieved separately by the two unidirectional radial turbines that compose it when they work in DM under steady flow conditions.

The radial DDT that incorporates the first version of the IFR turbine achieves a peak efficiency under non-steady flow conditions of around 26 %, while, although there is still a deficit compared to its competitors, the radial DDT that incorporates the optimized version of the IFR turbine achieves a higher peak. Moreover, this peak efficiency of around 30 % has shifted to lower Γ values, this meaning that the machine will work better with lower pressure drops, which could be more likely real operating conditions.

VI. Conclusions

In this work, the radial version of the Double Decker Turbine (DDT) concept is presented for its application in Oscillating Water Column (OWC) devices.

A CFD model has been developed to assess the performance, under steady flow conditions, of an InFlow Radial (IFR) turbine (first and optimized version) designed to be mounted in combination with an OutFlow Radial (OFR) turbine taken from the literature. Based on the numerical results obtained in the simulations, a non-steady analysis has been carried out assuming sinusoidal flow conditions, promising results for a first attempt been obtained.

Despite the not entirely satisfactory results, the authors of this work esteem that there is room for improvement since this is the first attempt on the radial DDT and the most important drawback has been found. In order to improve the overall performance of the radial DDT, it is necessary to enhance the behaviour of the IFR turbine when it works in Reverse Mode (RM) as a backflow preventer, as well as further improve its efficiency when it works in Direct Mode (DM).

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